Understanding Psychrometrics

Third Edition

Donald P. Gatley

ASHRAE™

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Psychrometrics—the physics of moist air:
an applied science dealing with the properties and processes of moist air.

Preface

Psychrometrics is a subscience of physics dealing with the *properties* and *processes* of *moist air.* Moist air is defined as a mixture of two gases: *dry air* and *water vapour* (the gas phase of H_2O). *Dry air* within the troposphere is treated as a non-varying mixture of nitrogen (78.1%), oxygen (20.9%), argon (0.9%), and other trace gases, including carbon dioxide (<0.04%). Some broaden the definition of psychrometrics to cover mixtures of the gas of one substance (any *dry gas* component) and the condensable vapour of a second substance.

The clock in the margin of this paragraph and elsewhere in this book indicates text that should be read by a student or new user who wants to acquire most of the basics of psychrometrics in four to six hours of reading.

Psychrometrics is a basic science that underlies agricultural and aeronautical engineering; air conditioning; drying of crops, grains, and pharmaceuticals; dehydration; dehumidification; humidification; moisture control; meteorology; weather reporting; food science engineering; and piloting of aircraft. It is possible to work in these fields without a good understanding of fundamental-level psychrometrics by the use of shortcut formulae, tables, and charts. While it is true that designers can survive in these fields with minimal and incomplete psychrometric skills without creating many problems for themselves, their employers, and clients, there is no justification for this lack of knowledge when psychrometrics can be quickly learned.

Many readers are frustrated in their pursuit of fundamental psychrometric knowledge because existing texts are incomplete, overly complicated, not well organized into learning elements, or contain obsolete terms and calculations that have little relevance in today's world of fast personal computers, psychrometric software, and computer-generated psychrometric charts tailored to a site-specific barometric pressure or altitude.

Psychrometrics can be simply explained and is solidly based on: (1) the ideal gas equation, (2) Dalton's model of partial pressures, (3) conservation of energy, and (4) conservation of mass. In the author's opinion, there is every reason to pursue a solid foundation in psychrometrics when it takes less than four hours to acquire this background for a lifetime of use. Such knowledge may not be required in typical designs, but it can prevent costly mistakes when altitude or barometric pressure differ from standard sea level values, when subfreezing temperatures exist, or when the specific volume of the moist air differs substantially from that of *Standard Temperature and Pressure (STP)* air (arbitrarily fixed at 15°C and 101,325 Pa).

What is different about this psychrometric text? It is devoted solely to psychrometrics. Psychrometrics is usually covered in one or two chapters of a thermodynamics textbook. This text is written for ease of understanding and not with brevity in mind, but at the same time the information is presented in such a way that the reader can choose not only the chapters to explore but also the level of detail within each chapter. It also includes an extensive listing of the pioneers of psychrometrics.

The underlying reasons for this new psychrometric publication are: (1) to address the changes brought on by the shift from manual calculations and plotting on printed psychrometric charts to software-based psychrometric calculations and graphics, (2) to present in one text a thorough coverage of psychrometric fundamentals, and (3) to assist designers and practitioners in the transition to the Système International system of units and calculations. Therefore, the objectives of this publication are to:

- Provide a resource equivalent to a reference book as well as a basic refresher course for those who use psychrometrics on a recurring basis.

- Provide students and air-conditioning designers, meteorologists, process engineers, and other users with a four-hour *complete* psychrometrics learning module.

- Put to rest the impression that psychrometric calculations based on ideal gas formulations (at normal air-conditioning temperatures and pressures) are imprecise, inaccurate, or based largely on empirical formulae. The text includes comparative data for ideal gas calculations and the ultimate-in-accuracy real moist air formulations by Herrmann-Kretzschmar-Gatley (ASHRAE RP-1485). In contrast to statements in some texts, ideal-gas-based psychrometric calculations are far more accurate than heat gain, duct pressure loss, and other calculations used by air-conditioning engineers and meteorologists.

- Provide practitioners and those new to the field with multiple definitions of basic terms to help in their understanding and communication.
- Provide those fluent in *inch-pound* or *metric* psychrometric calculations with an easy transition to *Systéme International* psychrometrics.
- Provide *all* of the algorithms required for psychrometric calculations; e.g., few texts provide the correlation between barometric pressure and altitude, nor do they provide formulas for wet-bulb and dew-point temperatures below freezing. Algorithms can be input into hand-held programmable calculators and personal computers to aid engineers and meteorologists in their work.
- Motivate manufacturers of cooling coils and desiccant dehumidifiers to utilize proven algorithms in their selection and psychrometric software to eliminate inconsistency.

Endnote

Each reader will determine the degree to which this book meets the author's objectives. Once the basic principles are understood, the psychrometric practitioner will undoubtedly find progressively easier and more efficient ways to apply them in the solution of meteorological, air-conditioning, drying, dehumidification, humidification, and other problems. For the air-conditioning and drying industries, this may lead to the development of compound or hybrid cycles that are more efficient and less costly.

You, the reader, will be the ultimate test of my four-hour learning conviction. Hopefully, that conviction will prove true for the majority. For all, it is hoped that this book will be the resource that (1) takes the mystery out of psychrometric calculations, (2) makes a convincing case for the accuracy of calculations based on the ideal gas equation, (3) gives a far better grasp of dew-point, moisture, and psychrometric processes, and (4) allows practitioners to serve their clients and the public better.

Please e-mail suggested corrections, comments, and criticism to the author at don.gatley@mail.ashrae.org.

Donald P. Gatley, P.E.
January 2002 (1st edition)

Preface to the Third Edition

In the late 1990s, when the first edition of this book was written, psychrometrics was a mature science, and I thought that little if anything would change over the following century. Since then, the universal gas constant has been revised by CODATA, the molar mass of dry air has increased by 0.0001 kg/kmol every four to five years due to the increase of CO_2 in Earth's atmosphere, IAPWS has issued new models for the calculation of H_2O properties, and in 2009 ASHRAE replaced the 1983 Hyland-Wexler real-moist-air numerical model with the research project RP-1485 LibHuAirProp model. These changes have little effect on air-conditioning and meteorological psychrometric calculations, but students and practitioners should have a reference that agrees with the latest international standards.

The third edition includes the above changes, a straightforward and more elegant equation for the adiabatic saturation process in the wet-bulb temperature chapter, an appendix summarizing the 2009 RP-1485 ASHRAE research, as well as minor edits to the text.

Supporting Files

This publication is accompanied by a limited demonstration version of the ASHRAE LibHuAirProp add-in. Also included are the hw.exe program from the second edition and PDF files of 13 ultra-high-pressure and 12 existing ASHRAE psychrometric charts, plus 3 new 0°C to 400°C, 0–1.0 humidity ratio charts for 5.53, 101.325, and 2000 kPa.

The LibHuAirProp add-in allows for duplication of portions of the real moist-air psychrometric tables in *ASHRAE Handbook—Fundamentals* for both standard sea-level atmosphere pressure and pressures from 5 to 10,000 kPa. The hw.exe program is included to enable users to compare the 2009 ASHRAE numerical model real moist-air psychrometric properties with the 1983 ASHRAE-Hyland-Wexler properties.

These files can be downloaded at www.ashrae.org/UP3. If the files or information at the link are not accessible, please contact the publisher.

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Introduction

Give a man a fish, provide food for a day.
Teach a man to fish, provide food for a lifetime.
—Chinese proverb

1 How to Use this Book

vMIn general, each chapter is written as a stand-alone section so that it is possible to pick and choose chapters and, as such, the text serves as a reference for those with prior knowledge of psychrometrics.

The book also serves as a four- to six-hour introduction to the science of psychrometrics for those new to the subject, and this influenced the sequencing of chapters. The clock in the margin of this paragraph and elsewhere in this book indicates the text that should be read in a four- to six-hour learning session.

Since this book was written as both a learning module and a reference book, many of the definitions, physical phenomena, and equations are explained in more detail than found in a basic book. The reader should have a general background knowledge *equivalent* to a high school physics or general science course. For those somewhat familiar with psychrometrics in inch-pound (I-P) units, this text will not only provide an easy transition to psychrometrics in Système International (SI) units but also a quick refresher on the underlying principles of psychrometrics. Learning psychrometrics today is easier because of the SI coherent system of units, which eliminates duplicate units and conversions, and because of the personal computer and software that has all but eliminated not only manual computations using approximate formulae but also graphical solutions using the psychrometric chart.

Psychrometrics brings with it many new terms and concepts. Some texts introduce all of the terms and concepts in a single chapter. For many, including the author, this can produce brain overload and confusion and is an obstacle to the learning process, which is, after all, the goal. In this book, definitions and new concepts are introduced *as needed* and each psychrometric property is treated in a separate chapter. In general, equations that interrelate the properties are introduced with the property.

An exception to the *as needed* introduction of terms and concepts is Chapter 5, “Basics of the Psychrometric Chart,” which precedes the individual chapters on psychrometric properties. The psychrometric chart is a convenient and useful tool for determining psychrometric properties and visualizing the changes to moist air properties in psychrometric processes. This early chapter is intended to give the reader an overview of the chart to help in overall understanding and to add clarity and meaning as each psychrometric property is developed and discussed in subsequent chapters.

The choice of whether or not to read Chapter 8, “Psychrometric Pioneers and Charts from the First 100 Years,” is left to the reader. It may pique the reader’s interest. It may help to explain the evolution of this science and the plethora of sometimes overlapping and confusing terms introduced into psychrometrics from the fields of meteorology, drying of materials, air conditioning, thermodynamics, chemical engineering, and agricultural engineering.

Each of the chapters dealing with individual psychrometric properties defines and discusses the property at several levels of detail. Those new to psychrometrics and those interested in a quick refresher or overview may want to read only the definitions and the first level of detail and then move on to the next chapter. In most cases, multiple definitions are provided. The first definition is suitable for general lay-person discussion. Subsequent definitions are more complete and technically more accurate. A definition serves no purpose unless it provides meaning and understanding for the reader. Multiple definitions allow the reader to choose the definitions that have the most meaning. Readers are encouraged to customize definitions.

A skeleton psychrometric chart highlighting only the isolines of the property under discussion is included in each property chapter.

Chapter 25 on *altitude effects* for locations other than sea level is required reading for those readers who live in or design for deep underground applications or applications for projects at high elevations; but, for most users concerned with psychrometrics within ±500 metres of sea level, this chapter can be scanned for future reference or skipped altogether. This chapter also illustrates the change in altitude (or local barometric pressure) effect on the appearance of the psychrometric chart.

JARGON

The two quotations below are from a paper presented by Frank C. Quinn at the *1985 International Symposium on Moisture and Humidity Control*. The first quotation suggests why psychrometrics has multiple and sometimes confusing terms. The second, as a consideration for students and occasional users of psychrometrics, may provide the motivation for others to use consistent terms and eliminate improper terminology.

There are many ways to measure and express moisture and humidity, and often the term, or value (units) used is a carryover from the early days of a particular industry.

The measurements and consequently appreciation for the influences of water are unnecessarily complicated by improper terminology (and symbols) and usage and the proliferation of measurement techniques which have led to ambiguity in the rationale of many moisture/humidity instruments.

Since psychrometrics evolved from many fields, it should come as no surprise that different or substitute words are often used to describe the same psychrometric property. This is unfortunately true in air-conditioning and meteorology psychrometrics. Much of the jargon is the result of pioneers approaching psychrometrics from the different fields of meteorology, agricultural engineering, air-conditioning, drying, aeronomy, and aeronautical engineering. Other jargon was introduced by theoretical scientists who borrowed and truncated technical terms from the allied sciences of chemistry and physics.

Some substitute words do not qualify as synonyms as defined by *Webster's International Dictionary*, *ASHRAE Terminology of HVAC&R*, meteorology books, or scientific dictionaries. This text uses only the author's preferred word, which is most often the meteorologist's and ASHRAE's preferred word. The goal is not to limit or restrict the reader or to redefine the technology; rather, it is felt that consistent word usage will eliminate possible reader confusion. Table 1-1 lists some preferred words and substitutes.

Table 1-1—Psychrometric Vocabulary

Preferred Word	Words Sometimes Used as Substitutes
air, *the mixture of two components: water vapour and dry air.* No subscript is used for mixture properties.	moist air (Consideration was given to the term *moist air*, which was not chosen because the abbreviation MA might be confused with *mixed air*, the blend of outside and return air.)
dry air, DA, *one of the two components of the mixture* air.	air (Since this could be confused with the previous term, *air*, this text will always use the term *dry air.*) Note that *dry air* in the troposphere has nearly constant constituents of 78% nitrogen, 21% oxygen, and 1% other gases. Also called "bone dry air."
water vapour, WV*, one of the two components of the mixture* air.	vapour, gas phase of water, humidity, moisture, moisture vapour, low pressure, or ultra low pressure steam

Table 1-1—Psychrometric Vocabulary (Continued)

Preferred Word	Words Sometimes Used as Substitutes
barometric pressure, p_{BAR}, units Pa (pascal)	total pressure, atmospheric pressure. (Note: if a process operates at significantly higher or lower pressure than the location average barometric pressure, then that process should be analyzed at actual pressure.)
humidity ratio, *W*, which equals $\mathrm{m_{WV}}/\mathrm{m_{DA}}$, units $\mathrm{kg_{WV}}/\mathrm{kg_{DA}}$	(a combination and contraction of *water vapour* (humidity) and *mixing ratio*), moisture content, mixing ratio, water vapour to dry air mixing ratio, and, incorrectly, specific humidity or absolute humidity
wet-bulb temperature, t_{WB}, units °C or K	thermodynamic wet-bulb temperature, temperature of adiabatic saturation (equals psychrometer wet-bulb temperature within 1% at 3 to 5 m/s air velocity across the sensing element)
water vapour pressure, p_{WV}, units Pa	partial pressure or partial pressure of water vapour
specific enthalpy, *h*, with units of energy/$\mathrm{m_{DA}}$, which in SI units is $\mathrm{kJ/kg_{DA}}$	enthalpy, total heat, and heat content (Heat is a form of energy crossing a boundary due to temperature difference; enthalpy is a composite of three properties: $h = u + p \cdot v$ where u is internal energy, p is pressure, v is specific volume, and the product "$p \cdot v$" is flow work. Enthalpy difference and heat transfer happen to be equal in a constant pressure steady flow process with no shaft work.)
specific volume, v_X, units of $\mathrm{m^3/kg_{DA}}$ (Note: The *x* of v_x can be DA for *dry air*, *WV* for *water vapour*, or none for air.)	the reciprocal of density, ρ; i.e., $v_{DA}=1/\rho_{DA}$ (Note: When dealing with a gas or a vapour, specific volume is generally used in preference to density. To avoid confusion, do not use both. *Important:* If density is used in psychrometrics, its units are mass of dry air per unit volume of the moist air.)
adiabatic process	a process with no transfer of heat (due to temperature difference in either direction) across the system boundary. An adiabatic process may include shaft work or "$p \cdot v$" flow work. For an open system, an adiabatic process may include mass transfer of substance(s) across the system boundary.

Obsolete Words and Words That Should Not Be Used Because of Dual Meanings

In the last 100 years, authors, scientists, researchers, and manufacturers have used various words to describe the many psychrometric properties. Some of these words have dual meanings and their use is

Table 1-2

Word	Reason That Use of the Word Is Discouraged
specific humidity, units of $kg_{WV}/kg_{(DA+WV)}$	Dual meaning. ASHRAE and the World Meteorological Organization define specific humidity as the ratio of the mass of water vapour in moist air to the total mass of the dry-air and water-vapour components. It is also used incorrectly as the ratio of water vapour in moist air to mass of dry air, i.e., *humidity ratio*.
absolute humidity, units of kg_{WV}/m^3	Possible dual meaning. Absolute humidity is the water vapour density expressed as the mass of water vapour per unit volume. Today it is a rarely used term. In some cases it has also been used synonymously with *humidity ratio*.
degree of saturation, $\mu = m_{WV}/m_{WVS}$, units none (Also called *percentage humidity* and *saturation ratio*; a decimal ratio.)	Now rarely used. It was an important property used in interpolation of tabular statepoint properties. *Degree of saturation* is the ratio of the humidity ratio at a defined statepoint to the humidity ratio at saturation at the same dry-bulb temperature. It is equal to relative humidity only at 100% and 0%. It differs from relative humidity by several percent in the 50% RH range. Many people and some texts incorrectly use the definition of degree of saturation as the definition for relative humidity.
Sigma function	Now rarely used. This term was an aid in the days of manual psychrometric calculations. *Sigma function* is the enthalpy at the saturated wet-bulb temperature less the enthalpy of liquid water at the same temperature. Its need has been superseded by psychrometric charts with enthalpy edge scales or enthalpy deviation curves and by software algorithms and personal computers.

discouraged. Some are still accurate terms; however, they are no longer needed in instruction or calculations. These were valuable to the user in the pre-handheld calculator, pre-personal computer era when a printed psychrometric chart and a table of property values were indispensable. This text is confined to terms that are in common everyday psychrometric use today. The terms in Table 1-2 are not used in this text.

Subscripts

Multiple letter subscripts are used in many equations. The intent is not to influence the science of psychrometrics or naming conventions but rather to aid in faster recognition for the first-time reader.

PSYCHROMETRIC ALGORITHMS

Chapter 26, “Psychrometric Program Listing and Comparison Table,” includes a listing of SI algorithms for calculation of statepoint properties along with SI to IP conversion equations. The algorithms can be solved by equation solver software such as EES (Engineering Equation Solver from F-Chart Software) or they can be reconfigured into FORTRAN or some other language. They can also be modified and used in spreadsheet software. Looping or iteration routines are required for the solution of some properties. Tables of selected accurate property values are included so that the user can verify the accuracy of these and other algorithms or psychrometric software.

PHYSICAL LAWS, MODELS, ASSUMPTIONS, AND THE ACCURACY OF PSYCHROMETRIC CALCULATIONS

Equations in this text are based on *ideal (perfect) gas* behavior for (1) the *dry air* component, (2) the *water-vapour* component, and (3) the moist air mixture of the two components. This requires that the behavior of each constituent of the mixture can be modeled by the perfect gas equation:

$$p_X \cdot v_X = R_X \cdot T_X$$

The equations use Dalton’s model (also called Dalton's law or rule) of an ideal gas mixture, which states that the gas and water-vapour molecules share the same volume and that the total pressure of the mixture is equal to the sum of the pressures of the individual components. Each of the constituent gases exerts a pressure (called partial pressure), which is the pressure that the constituent gas would exert if it alone occupied the volume.

Air-conditioning psychrometric processes are almost always modeled as steady flow, constant pressure processes.

Dry air, *water vapour*, and the combined moist air mixture closely follow perfect gas behavior in the temperature range of –40°C to 65°C and at total pressures up to 300 kPa.

Additional accuracy is built into the equations in the appendix by the use of a correction factor (enhancement factor) to account for the slight interaction of *dry air* and *water vapour* molecules. In addition, the specific heat capacity of water vapour in the enthalpy algorithm is based on curve fits of the ASHRAE RP-1485 (2009) moist-air-enthalpy values as opposed to the use of Keenan, Keyes, and Moore low-pressure-steam-table values. RP-1485 values are more accurate in psychrometrics because they account for the slight influence of N_2 and O_2 molecules.

The tables in Chapter 26, “Psychrometric Program Listing and Comparison Table,” attest to the accuracy of ideal gas-based psychrometric calculations in the –40°C to 65°C temperature range.

Looking for an excuse to avoid psychrometrics? Webster's Dictionary does not even list psychrometrics as a word!

Psychrometrics—the science that involves the properties of moist air (a mixture of dry air and water vapour) and the processes in which the temperature and/or the water vapour content of the mixture are changed.

2 Etymology

THE ETYMOLOGY OF "PSYCHRO" AND ITS DERIVATIVE WORDS

Psychrometer

The beginning of the word *psychrometrics* occurred in 1825 when Ernest Ferdinand August of Germany named his wet-bulb thermometer a psychrometer using the Latin words *psychro*, to make cold, and *meter*, to measure. Today a psychrometer includes both dry-bulb and wet-bulb thermometers and is classified as a form of a hygrometer. A hygrometer is used to measure the humidity in the atmosphere. Thus, there is a contradiction of sorts: the name of the instrument suggests the measurement of cold and yet the instrument is used to measure humidity. The answer to this contradiction is: The difference between psychrometer dry-bulb and wet-bulb readings is called the wet-bulb depression, which can be interpreted as additional coldness (*psychro*—to make cold).

Psychrometry

The suffix *metry* converts a word such as *psychrometer* to *the science of using a psychrometer*. Psychrometry as a science is, therefore, the science of measurement of dry-bulb and wet-bulb temperatures using a psychrometer. Gradually the word was also applied to the scientific theory underlying the thermodynamic wet-bulb temperature in an adiabatic saturator.

Webster's Third New International Dictionary, published in 1993, lists *psychrometry* as a noun and defines it as *the science dealing with the physical laws governing the mixture of air and water vapour*. ASHRAE defines psychrometry as *that branch of physics concerned with the measurement or determination of atmospheric conditions, particularly the moisture in the air.*

For some, the term *psychrometry* has broadened to *the science of measuring and determining (calculating) all of the physical properties of an air-water vapour mixture, not only the dry-bulb and wet-bulb temperature but also barometric pressure, dew-point temperature, vapour pressure, relative humidity, humidity ratio, specific enthalpy, and specific volume.*

Wet-bulb thermometers are used not only in measurements involving the gas "*dry air*" and the condensable vapour "*water vapour*" but also in wet-bulb measurements of hundreds of other gas and condensable vapour combinations. Therefore, some might expand the definitions of psychrometry to include all gas and condensable vapour combinations. A search of the web reveals *Handbook of Psychrometric Charts—Humidity Diagrams for Engineers* by David C. Shallcross (Blackie Academic and Professional and Kluwer Academic Publishers, London, 1997; ISBN 0751404764; 317 pages plus front matter and index). The book description is as follows: "In chemical, petroleum, air conditioning and refrigerating engineering, the engineer often encounters systems where gases and condensing vapours co-exist. Key data describing the behavior of such mixtures can be obtained by consulting an appropriate psychrometric chart, but up until now such charts were difficult or impossible to obtain. This new book brings together for the first time over 300 such charts, covering most of the systems likely to be encountered by chemical, petroleum, air conditioning and refrigeration engineers." This description infers that the word *psychrometric* applies to all mixtures of a dry gas and a condensable vapour.

Psychrometric

Webster's classifies the word *psychrometric* as an adjective within its definition of *psychrometer* but lists no further definition. It does define the suffix "ic" as a combining form that converts a basic word to an adjective with the meaning *of or relating to the art, process, or science,* which, in this case, is psychrometry.

ASHRAE defines *psychrometric* as an adjective relating to the measurement or determination of atmospheric conditions, particularly regarding water vapour mixed with air. The most common use of *psychrometric* is as an adjective in "psychrometric chart" or "psychrometric tables."

In 1904, Willis H. Carrier named his moist air properties chart a hygrometric chart but, by 1911, the name was changed to psychrometric chart.

Psychrometrics

Webster's does not list *psychrometrics* as a word; however, it lists the suffix "ics" as *a branch of knowledge (science) associated with that preceding "ics"* (as in *economics, ethics*).

The *McGraw-Hill Encyclopedia of Science and Technology* published in 1992 defines psychrometrics as the study of the physical and thermodynamic properties of the atmosphere.

The 1992 ASHRAE publication *ASHRAE Terminology of HVAC&R* does not recognize psychrometrics as a word; however, in 1996, ASHRAE published the book *Psychrometrics: Theory and Practice*.

For many, ***psychrometrics is the science that involves the properties of moist air*** *(a mixture of dry air and water vapour)* ***and the processes*** *(meteorological, air conditioning, drying, humidification, dehumidification, agricultural soil evaporation)* ***in which the temperature and/or the water vapour content of the mixture are changed.*** This book uses this broad definition.

Aeronomy and Aerology

A more fitting word for the science of psychrometrics dealing with the properties and processes of mixtures of air and water vapour might be the word *aeronomy*. Unfortunately, scientists in the 1940s reserved that word for the science of the upper regions of the atmosphere where dissociation and ionization of molecules are important.

Another related word, *aerology*, is defined as a branch of meteorology that deals with atmospheric conditions away from ground level, especially with the description and discussion of the phenomena of the free air as revealed by kites, balloons, airplanes, and clouds.

RELATED WORDS

Other related words use the Greek *hygro* combining form. *Hygr* or *hygro* relates to the vapour phase of H_2O. The vapour phase of H_2O is often called *humidity*. The related words include *hygrometer*, meaning any of several instruments for measuring the humidity of the atmosphere; *hygrology*, a branch of physics that deals with the phenomena of humidity; *hygrometry*, a branch of physics that deals with the measurement of humidity, especially of the atmosphere; *hygroscopic*, an adjective describing a substance tending to adsorb water vapour; and *hygrometric*, an adjective relating to hygrometry or to humidity.

Hydr or *hydro* is another combining form that infers the liquid phase of H_2O. *Hydrophobic* means water fearing. *Hydrophilic* means water loving.

DIVERSIONS

The two words most commonly misspelled in this field are "psychrometrics" and "desiccant." Fortunately most word processing spell-checkers will catch and correct dessicant (the incorrect spelling); however, spell checkers do not include the word *psychrometrics*, and even if they did they would not highlight *psychometrics* for correction because *psychometrics* is also a word. On second thought, there may be some justification for using the word psychometric based on the fact that *one who spends too much time peering at psychrometric charts, whirling a sling psychrometer, or delving into the intricacies of psychrometrics is probably a good candidate for psychometrics*.

CONCLUSION

The word *psychrometrics* in this book is defined as the science that involves the properties of moist air (a mixture of dry air and water vapour) and the processes in which the temperature and/or the water vapour content of the mixture are changed. Psychrometric processes occur in the fields of meteorology, air conditioning, drying, humidification, dehumidification, food science, and agricultural soil evaporation.

> *Water vapour constitutes only a small fraction (3.5% maximum) of the atmosphere but the importance of water vapour in the air is far greater than this small percentage would indicate. Indeed scientists agree that water vapour is the most important gas in the atmosphere when it comes to understanding atmospheric processes such as thunderstorms.*
> *—Fredrick Lutgens, 1979*

3 Moist Air—The Psychrometric Substance

WHAT IS THE SUBSTANCE THAT WE DEAL WITH IN PSYCHROMETRICS?

Psychrometrics usually involves the *air* within our homes, factories, retail establishments, entertainment centers, hospitals, offices, and places of work, as well as outside *air*. We call this substance *moist air* (which we shorten to *air*) because air, even in the driest deserts or the coldest arctic, has a tiny amount of *water vapour* associated with it. Psychrometrics treats the substance *air* as a mixture of two nonreacting nearly *ideal* gases: ***dry air*** and ***water vapour***. *Ideal gases* (sometimes called *perfect* gases) are those that conform to the ideal gas equation of state, which can be written in any of the following forms with the understanding that the *units* of the chosen equation must balance (cancel):

$$p_X \cdot v_X = R_X \cdot T_X \qquad R_X = R_U / M_X$$

$$p_X \cdot v_{MOL_X} = R_U \cdot T_X \qquad n_X = m_X / M_X$$

$$p_X \cdot V_X = m_X \cdot R_X \cdot T_X$$

$$p_X \cdot V_X = n_X \cdot R_U \cdot T_X$$

where

p = pressure, Pa
v = specific volume, m^3/kg
v_{MOL} = molar specific volume, m^3/mol
V = total volume, m^3
R_X = gas constant for a specific gas, J/(kg·K)
R_U = universal gas constant, J/(kmol·K)
T = absolute temperature, K in kelvin
m = mass, kg
n = number of kilomoles in which $n = m/M$
M = molecular mass, kg/kmol
X = subscript for a specific substance, e.g., N_2, O_2, H_2O

Some psychrometric texts label the psychrometric substance *air* as *atmospheric air* or *moist air*. The term *atmospheric air* is not used herein because it can also mean the total envelope of air surrounding the earth, including the troposphere and the stratosphere. This could be misleading since psychrometrics usually involves air only in the lower regions of the troposphere. The term *moist air* could be used to distinguish *air* from its two components, *dry air* and *water vapour*. Moist air will not be used in this text because *moist* or *moisture* can be defined as either the liquid or the gas (vapour) phase of water, and in psychrometrics we are dealing with the *water vapour* mixed with the *dry air* component except for the rare case of analysis dealing with fog.

The working substance in psychrometrics is simply called *air* and, as used herein, *air* will always refer to a two-component mixture of *dry air* and *water vapour*.

Air within the troposphere always contains a variable quantity of *water vapour*, the gas phase of H_2O. The highest naturally occurring recorded quantity of *water vapour* in air is 0.035 kilogram of *water vapour* per kilogram of *dry air* (34°C dew point), which was recorded at Sharjah, Arabia, on the shores of the Persian Gulf. In the eastern United States, the maximum dew-point temperature is approximately 28°C, which, at sea level altitude, is equivalent to 0.025 kilogram of *water vapour* per kilogram of *dry air*. The lowest recorded quantity of *water vapour* in atmospheric air is at Vostoc, Antarctica, with 0.00000001 kilogram of *water vapour* per kilogram of *dry air*. Obviously the mass of *water vapour* in the air varies over a significant range.

The amount of *water vapour* associated with *air* varies seasonally and from day to day. It is quite small; nevertheless, it is significant because its latent heat of vaporization is about 2450 kJ/kg (at 21.5°C) compared with the specific heat capacity of *dry air* of approximately 1.006 kJ/(kg·°C). The *water vapour* content of *air* has a significant effect on our weather as well as in the science of psychrometrics. The energy transformations that occur as *water vapour* is condensed to its liquid or solid (ice) phases, as when a moist air mass is lifted to 9000 metres by hot afternoon air currents, is a major mechanism in thunderstorms and lightning, not to mention hurricanes. When H_2O changes from one phase to another, it absorbs or releases heat and this energy is termed *latent* (hidden) *heat*. *Water vapour* in the atmosphere transports this latent heat from one region to another and is the major energy source that helps drive many storms. The energy transfer associated with cooling and dehumidifying outside air from average summer conditions of 30°CDB, 25°CWB in the U.S. Southeast to 24°CDB, 45% RH average office building conditions is 80% related to dehumidification (condensing and removing *water vapour*) and only 20% related to reducing the temperature of the air.

The Dry-Air Component

The *dry-air* component of air is actually a mixture of gases containing (by volume) approximately 78% nitrogen, 21% oxygen, 1% argon, and at least eight other gases comprising an inconsequential 0.034%. All of the *dry air* constituent gases are well above their critical temperatures and are not subjected to condensation at temperatures higher than –100°C; therefore, no error is introduced in treating the constituents as a single gas. Most psychrometric calculations assume that the composition of *dry air* does not vary. Table 3-1 identifies the constituents of *dry air* and the calculation of its apparent molecular mass as used by Kretzschmar, Herrmann, and Gatley in their 2009 major revision of the 1983 Hyland and Wexler *Psychrometric Properties of Real Moist Air* (ASHRAE Research Project RP-1485).

Minor differences in the composition of air have little effect on psychrometric values. Hyland and Wexler (1983a, 1983b) used 28.9645 kg/kmol and Nelson and Sauer (2001) used 28.95849 kg/kmol as the apparent relative molar mass of air, compared with 28.966 kg/kmol in Table 3-1.

The Water-Vapour Component

The other component of *air* for purposes of psychrometric analysis is *water vapour.* As noted above, the mass of *water vapour* associated with the *dry air* is not constant. *Water* is constantly evaporating from the earth's oceans and land and entering the atmosphere. It returns to the earth as precipitation. It is one of the primary causes of weather and a major factor in hurricanes and thunderstorms. *Water vapour* is added to the air in our homes and buildings by infiltration, perspiration, respiration, cooking, bathing, dishwashing, and clothes drying, from plants, and by desorption of water from building materials and furnishings. *Water vapour* can also be added to the air by humidification or evaporative cooling processes. Cooling and dehumidification equipment or dehumidification equipment alone is often used to remove *water vapour* from the air for reasons of comfort, indoor air quality, safety, preservation of building components, and quality control in manufacturing and storage of certain materials.

Table 3-2 identifies the constituents of the gas *water vapour* and the calculation of its molecular mass.

Air—A Mixture of Dry Air and Water Vapour

In summary, the substance around us is called *air*. In psychrometrics, it is treated as a two-component mixture consisting of *dry air* and *water vapour.* The *water vapour* is a condensable component.

Table 3-1 Projected 2036 Composition of Dry Air and Calculation of M_{da}

Two Inert Gases and Two Reacting Gases			
Constituent	**Molar Mass, M**	**Mole Fraction, x_i**	**Contribution ($M \cdot x_i$)**
	kg/kmol	**ppmv = $x_i \cdot 10^6$**	**kg/kmol**
N_2 [Eq. 1]	28.0134	780818	21.873367
O_2 [Eq. 2]	31.9988	209382.45	6.699987
A_r [Eq. 1]	39.9480	9332	0.372795
CO_2 [Eq. 2]	44.0095	437.55	0.019256
Subtotals		**999970**	**28.965405**
Eight Trace Gases			
Ne	20.1797	18.2	0.000367
He	4.0026	5.2	0.000021
CH_4	16.0425	1.5	0.000024
Kr	83.7980	1.1	0.000092
H_2	2.0159	0.5	0.000001
N_2O	44.0128	0.3	0.000013
CO	28.0101	0.2	0.000006
Xe	131.2930	0.1	0.000013
Subtotals		**27.1**	**0.000537**
Trace to 30 ppmv [1, 2]	**19.8254**	**2.9**	**0.000057**
Grand Totals		**1000000**	**28.966000**
Notes [ ]			
Eq. 1	$x_iN_2 + x_iAr = 790150$ ppmv		Two inert gases
Eq. 2	$x_iO_2 + x_iCO_2 = 209820$ ppmv		Two reacting gases
1	Mean molar mass of the eight trace gases is 19.82541182 kg/kmol		
2	Adding 2.9 ppmv of trace gases brings the total of all gases to unity		

The CO_2 concentration of 437.5 ppmv is based on a 2008 concentration of 385 ppmv and an increase of 1.9 ppmv per year. If this average rate of increase actually occurs the CO_2 concentration of 437.5 will occur in 2036. The ASHRAE Research Project 1485 rounded M_{da} value of 28.966 kg/kmol will apply from the present until approximately 2058.

M_{da} *for any abundance of CO_2 may be calculated from the following equation:*

$$M_{da} = 28.960745 + CO_2\ [\text{ppmv}] \cdot 12.0107/1000000$$

Table 3-2—Composition of Water Vapour

Substance	Atoms	Atomic Mass[*]	Molecular Mass
Hydrogen (H_2)	2	1.0079817	2.015963
Oxygen (O)	1	15.9993046	15.999305
Total			18.015268

[*] *Atomic mass values are specific for the VSMOW isotopic composition of hydrogen and oxygen.*

Ten important statepoint properties of *air* are:

- Barometric or total pressure. This is the pressure measured with a barometer and is often called atmospheric or barometric pressure. It is the sum of the partial pressure of the *dry air* and the partial pressure of the *water vapour*: This property is called b*arometric pressure* in this text and its symbol is P_{BAR}.
- Dry-bulb temperature or simply temperature: t_{DB}
- Relative humidity: RH
- Wet-bulb temperature: t_{WB}
- Humidity ratio: w
- *Water vapour* (partial) pressure: p_{WV}
- *Dry air* (partial) pressure: p_{DA}. Note that $(p_{BAR} - p_{WV}) = p_{DA}$
- Dew-point temperature t_{DP}
- Specific volume (the reciprocal of density): v
- Specific enthalpy: h

In a two-component mixture of ideal gases, the statepoint properties are fixed if the total (barometric) pressure and two other *independent* statepoint properties such as dry-bulb temperature and wet-bulb temperature, dry-bulb temperature and relative humidity, or dry-bulb temperature and dew-point temperature are known. These property combinations are most often used to determine the statepoint since they can be measured.

Certain psychrometric processes involve constant humidity ratio or constant specific enthalpy. For these processes, the end-of-process statepoint properties are determined using the beginning statepoint calculated humidity ratio or specific enthalpy.

Humidity ratio, dew-point temperature, and *water vapour* pressure are not *mutually independent* properties; therefore, only one of these three may be used in determining psychrometric statepoints. Wet-bulb temperature and enthalpy are theoretically *independent* but they are very closely related and only one of these should be used in determining psychrometric statepoints.

Endnote Quotation

The atmosphere consists of a mixture of gases held by gravitational attraction to the earth. It is compressed under its own weight, and it varies in density with altitude and with temperature and water vapour content. The atmosphere contains water vapour in amounts up to 3.5% by weight.[1] *This amount is highly variable, depending greatly on temperature and previous history of contact with water sources. Water is a highly influential and important component, since it can exist as vapour, liquid, or solid within the range of atmospheric temperatures encountered. Clouds are formed when excess vapour condenses, producing suspended droplets and ice crystals or precipitation in the form of snow, hail, or rain. Changes in the phase of the water are accompanied by changes in energy content of the surrounding air, which adds to the complexity of the role of the atmosphere in energy exchanges*. (Extracted from *Building Science for A Cold Climate*, NRC Canada, 1995.)

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[1] A water vapour content of 3.5% (equivalent to about 34°C dew-point temperature) is rare and only occurs for short time periods adjacent to wet soil and shallow bodies of water in hot climates. The highest average summer dew-point temperature record is 29°C at Assab, Eriteria, on the Red Sea. The highest unofficial recorded dew point was 35°C reported in Dhuahran, Saudi Arabia at 3 p.m., July 8, 2003.

Air, rather than being an elemental substance is a simple mixture of gases,
primarily nitrogen and oxygen with variable smaller amounts of
argon, carbon dioxide, ozone, nitric acid, ammonia and
of great importance for meteorology, water vapour.
—Cavendish, Black, Priestly, Scheele, Lavoisier,
Rutherford, De Saussure, Macbride, & Dalton

Whence comes the dew that stands on the outside of a tank
and that has cold water in it in the summertime?
—Benjamin Franklin, 1797

4 Water Vapour

WATER VAPOUR (THE GAS PHASE OF H_2O): THE VARIABLE COMPONENT OF THE AIR AROUND US

In order to perform psychrometric property and process calculations, it is important to understand the properties of H_2O in its solid, liquid, and gas phases. The previous chapter established that *moist air* is a mixture composed of two gases: *dry air* and *water vapour. Water vapour* and ultra low pressure steam are precisely the same substance. When this substance is used in vacuum or low pressure steam heating systems, it is called *steam*. In psychrometrics, the substance is called *water vapour*.

H_2O, like most substances, exists in each of three phases: solid (ice), liquid (commonly called water), and gas (*water vapour*). H_2O is a very unusual substance in that all three of its phases can be observed in everyday life. For example, in the summer, getting a glass of ice and water involves combining *liquid H_2O* and *solid H_2O;* then, after a few minutes, condensation of *H_2O vapour* often results on the outside of the glass. Condensate (liquid H_2O or dew) results when *water vapour* in the air surrounding the glass is cooled below its dew-point temperature. *Water vapour* is an invisible gas and to say that it can be *observed* is incorrect; nevertheless, observation of condensate is ample proof that *water vapour* is a component of the air around us.

Figure 4-1, the pressure vs. temperature plot of the substance H_2O, illustrates the three phases of H_2O. Two phases of H_2O can coexist with each other in equilibrium at any point on a line and all three phases can exist in equilibrium at the triple point.

The three lines between phases meet at the triple point. The ice-vapour and the liquid-vapour line appear to be one continuous line, but they are really two lines that meet at the triple point. The equilibrium lines continue beyond the end points plotted in the figure.

The **bold** ice-vapour equilibrium line at the lower left side of the figure is the line between the solid phase (above the line) and the vapour phase (below the line). At any point on the curve, *water vapour* and ice coexist in equilibrium and the *water vapour* is designated as *saturated water vapour.*

The nearly vertical line is the ice-liquid equilibrium line, which separates ice on the left from liquid H_2O on the right. At any point on this nearly vertical line, ice and water coexist in equilibrium. The ice-liquid line is of little interest in psychrometrics because *water vapour* is the phase of concern.

The **bold** liquid-vapour equilibrium line starts at the triple point and curves upward to the right and separates liquid H_2O above the line and vapour H_2O below the line. The last plotted point on this curve at 100°C and 101.325 kPa is familiar to most as the point at which water boils at sea level atmospheric pressure. At any point on this curve, water and *water vapour* coexist in equilibrium and the *water vapour* is designated as *saturated water vapour.*

Figure 4-1 is important in understanding the psychrometric chart and the terms (1) dew-point temperature, (2) saturation temperature and pressure, and (3) equilibrium. In psychrometrics, the interest is in *water vapour* and, therefore, our concern is with the two lower lines (which meet at the triple point and appear to be one continuous bold line). These two lines separate *water vapour* from its solid and liquid phases. The area below these two lines is called the *vapour (gas)* region. A substance in this region is often referred to as *superheated.* Superheated refers to a vapour at a given vapour pressure that is at a higher temperature than its *saturation temperature.*

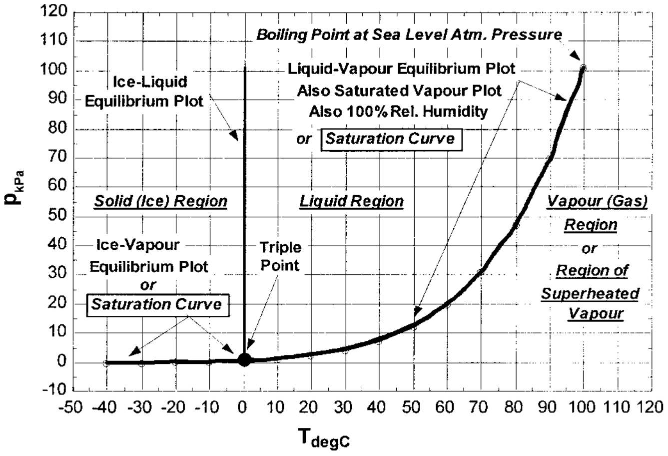


Figure 4-1—Pressure vs. temperature diagram for H_2O.

The Phase Changes of H_2O

Figure 4-2 illustrates and names the processes associated with the phase changes of H_2O.

- A change of phase from solid water to liquid water at the same temperature is called a *melting* process and requires the addition of 333.47 kJ/kg of energy to the ice. The reverse process is called *freezing* and requires the removal of 333.47 kJ/kg of the energy from the liquid water.
- A change of phase from liquid water to *water vapour* at a constant temperature is called an *evaporation* process (sometimes called vaporization) and requires the addition of approximately 2465 kJ/kg (at 15°C) of energy. (The latent heat of evaporation of H_2O varies from 2256.47 kJ/kg at 100°C to 2500.93 kJ/kg at 0°C.) The same quantity of energy must be removed from *water vapour* to condense it to its liquid phase in the reverse process of *condensation*. The quantity of energy required to change a substance from its liquid phase to its gas phase at constant temperature and pressure is called the latent heat of evaporation (or vaporization) and, in the reverse process, it is called the latent heat of condensation. The two values are the same but one is positive and one is negative.
- Figure 4-2 also shows the *sublimation* process and the reverse process called *deposition*. Sublimation involves a change in phase from ice, frost, or snow directly to *water vapour* at constant temperature. Logically, the latent heat of sublimation is the sum of the latent heat of evaporation plus the latent heat of melting. At 0°C, the latent heat of sublimation is 2500.90+ 333.47 = 2834.36.

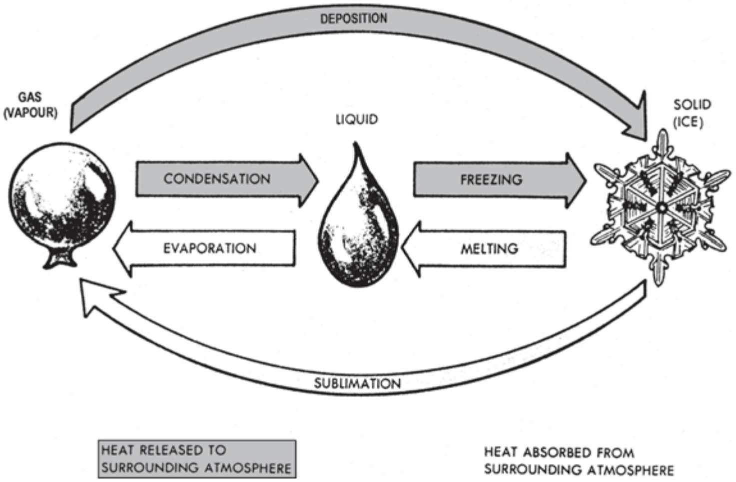


Figure 4-2—Phase change processes for H_2O.

Saturation (with Respect to Water Vapour)

The word *saturation* comes from the Latin word *satar,* which means "full," i.e., the volume can contain no more. The concept of saturation is more meaningful if the word *saturation* is expanded to the phrase "saturation with respect to." In psychrometrics, *water vapour* is of major interest and the expanded phrase is *saturation with respect to water vapour. Water vapour* and *dry air* share the same volume in the atmosphere. Saturation *with respect to water vapour* means that volume contains the maximum possible number of *water vapour* molecules. Another vapour, such as ammonia gas, could be added to the same volume and, if the quantity of ammonia gas molecules was the maximum possible at the existing temperature, then the vessel would also be *saturated with respect to ammonia vapour.*

If a sample of superheated *water vapour* is cooled at constant pressure to the temperature on the *water vapour* equilibrium curve, the resultant vapour is called saturated *water vapour.* If the temperature of the now saturated vapour is lowered by additional cooling, some of the vapour will condense to the liquid form. If some way was found to inject additional *water vapour* molecules into the saturated sample at the existing pressure and temperature, then either those additional *water vapour* molecules or other *water vapour* molecules in the volume would change to the liquid phase because the sample is already saturated and contains the maximum number of *water vapour* molecules.

Dew-Point Temperature

A substance at a pressure and temperature condition lying on the lower curve(s) of Figure 4-1 is said to be saturated. Technically, saturation is defined as a condition of coexistence in stable equilibrium of a vapour and the liquid or solid phase of the same substance. The overall curve is called the *water vapour saturation curve* or the *dew-point temperature curve. Dew-point temperature* for a given substance is the temperature at which saturation of that substance occurs without change in vapour pressure. For *water vapour* it is the surface temperature at which dew forms (strictly speaking, it is the temperature infinitesimally above that at which the first dew forms). For meteorologists, *dew-point temperature* applies only to the saturation temperature above 0°C and *frostpoint* applies to saturation temperatures below freezing; however, in this text, both are called *dew-point temperature*.

The Terms "Dew-Point Temperature" and "Saturation" Apply Only to Water Vapour, Not Air

The vapour saturation curve (t_{WVS} vs. p_{WVS}) is independent of atmospheric pressure or elevation. The *water vapour* saturation pressure at any given temperature does not change with a change in atmospheric pressure or elevation. Saturation *water vapour* pressure depends on the temperature only. The vapour saturation curve is not affected by the presence or absence of other gases including *dry air.* In psychrometrics, the terms *dew-point temperature, saturation temperature, saturated,* and *saturation* refer exclusively to the *water-vapour* component.

The phrase "the air is saturated with water vapour" is often heard. This phrase can be confusing because saturation has nothing to do with the air. A saturated *water vapour* condition exists when the volume contains the maximum possible number of *water vapour* molecules. The fact that the *dry air* and the *water vapour* occupy the same volume has no bearing on the fact that the *water vapour* is in a saturated state. Such usage is widespread in phrases such as "the air is saturated" or "the dew-point temperature of the air." No great harm is done and it would be wishful thinking that comments in this text could even nudge people toward more correct usage. The following quotation will reinforce these comments:

> *Faulty Conceptions. A false notion that the air has a certain capacity for moisture is widely prevalent, and is perpetuated by all such expressions as "The air is partly saturated with moisture," "Weight of aqueous vapour in a cubic foot of saturated air," etc. It should always be clearly observed that the presence of the moisture in any given space is independent of the presence or absence of air in the same space (except that the air may slightly retard the diffusion of the vapour molecules). It is more correct to say, in the above cases, that "the space is partly saturated with moisture," or that "the moisture is in a partly saturated condition or is superheated." By all means use the phrase: "Mass of a cubic foot of saturated aqueous vapour," not "Mass of aqueous vapour in a cubic foot of saturated air."*
>
> —C.F. Marvin, 1900, in "Psychrometric Tables for Obtaining the Vapour Pressure, Relative Humidity, and Temperature of the Dew Point" (U.S. Dept. of Commerce, Weather Bureau).

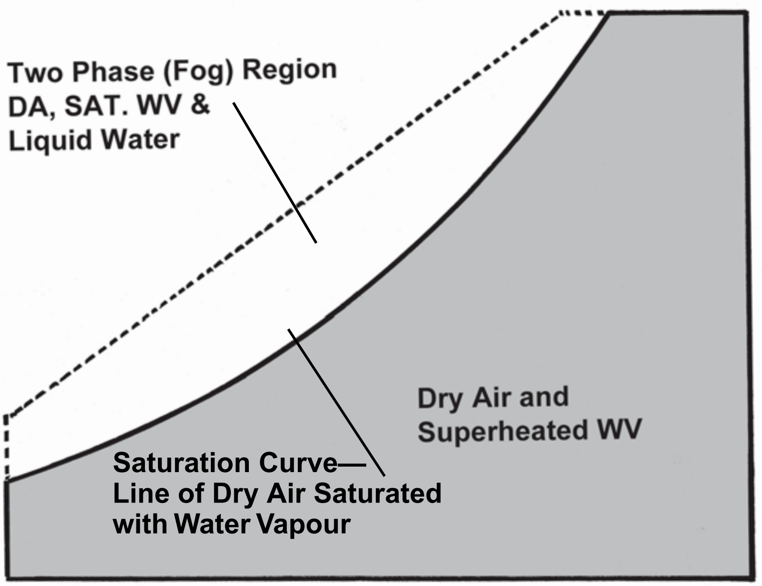


Figure 4-3—H_2O regions of the psychrometric chart.

The curve on the left side of the psychrometric chart (Figure 4-3) is the same as the curve on Figure 4-1. Some psychrometric charts label this the saturation curve. Later, in the section on relative humidity, the basic definition of relative humidity will reveal that the saturation curve for *water vapour* and the 100% relative humidity curve are the same.

RECAP

The important facts that should be retained from the first part of this chapter are: (1) the definitions and concepts of the saturation curve, (2) the concept and definition of dew-point temperature, (3) the shape of the *water vapour* vs. ice and *water vapour* vs. liquid water saturation curves, and (4) the fact that these terms and the curve apply exclusively in air-H_2O psychrometrics to the substance H_2O. In order to reinforce the importance of the shape of the curve, the reader should refer to any psychrometric chart and examine the curved left-hand boundary of the chart. It should come as no surprise that this curved line is called *the saturation curve* and can be quickly used to determine *the dew-point temperature*. Some charts provide a *water vapour* pressure scale on the right side of the chart that applies to this curve.

EQUATIONS FOR CALCULATING THE SATURATION CURVE

The saturation curve is important in psychrometric calculations. It defines the equilibrium or saturation conditions in terms of saturated *water vapour* pressure and saturated *water vapour* temperature. It is also called the *dew-point temperature curve*.

Many psychrometric texts direct the reader to steam tables for H_2O saturation pressures and temperatures. In 2009 *ASHRAE Handbook—Fundamentals* these tables are found within Chapter 1, "Psychrometrics," and Chapter 30, "Thermophysical Properties of Refrigerants." In Chapter 30 the table is titled "Refrigerant 718 (Water/Steam) Properties of Saturated Liquid and Saturated Vapour." (Water is the refrigerant in many absorption water chillers and in steam jet refrigeration. It has also been used as the refrigerant in a few multistage vapour compression centrifugal chillers.)

One of the objectives of this text is to make the reader comfortable with psychrometric software algorithms. This text therefore departs from most psychrometric texts by defining the saturation curve not from tables but from equations.

The saturated pressure temperature curves for ice, water, and *water vapour* shown in Figure 4-1 have been determined by researchers over the last two hundred years, starting with Dalton and Regnault. Numerous curve fit equations are available to define the *water vapour* saturation curve. In the early 1980s when computer processing speed was slow and memory and hard disk space were precious, very simple equations were used in psychrometric routines in order to conserve disk space and speed calculations. Anyone who doubts this only needs to think about the origin of the Y2K problem.

The simple, reasonably accurate Antoine equation [$\mathrm{LN}(p_{wvs}) = A - B/(t_C + C)$] is used in some psychrometric models. The coefficients A, B, and C are very sensitive to temperature range. Meteorologists utilize vapour pressure with respect to liquid at all temperatures, including below 0°C. The Antoine equation over liquid water for the range of –40°C to 50°C is $p_{wvs} = \mathrm{EXP}[23.658571 - 4082.14836/(236.727698 + t_C)]$ with units of Pa for p_{wvs} and °C for t_C (maximum deviation 0.745Pa; minimum deviation –1.102 Pa). This simple equation is suitable for a programmable calculator and advantageously can be rearranged to solve directly for dew-point temperature knowing vapour pressure.

This text displays the more accurate and complex equations of state from IAPWS-97 for *liquid* water and IAPWS-06 for the solid phase of H_2O (e.g., frost, ice, and snow).

Two equations are required: (1) to define the p versus t saturation curve for ice and *water vapour* below 0°C and (2) to define the p vs. t saturation curve for liquid water and *water vapour* from 0°C and above.

IAPWS-IF97 Equation for Saturation Vapour Pressure over Liquid Water as a Function of Temperature

$$p_{ws} = 1000 \cdot (2 \cdot C/(-B + (B^2 - 4 \cdot A \cdot C)^{0.5}))^4 \text{ [kPa]} \qquad (4\text{-}1)$$

where

$$Q = T + \mathrm{N}_9/(T - \mathrm{N}_{10})$$

$$A = Q^2 + \mathrm{N}_1 \cdot Q + \mathrm{N}_2$$

$$B = \mathrm{N}_3 \cdot Q^2 + \mathrm{N}_4 \cdot Q + \mathrm{N}_5$$

$$C = \mathrm{N}_6 \cdot Q^2 + \mathrm{N}_7 \cdot Q + \mathrm{N}_8$$

$T = t_C + 273.15$, where t_C is temperature in degrees Celsius

$\mathrm{N}_1 = 1167.05214527670$

$\mathrm{N}_2 = -724213.167032060$

$\mathrm{N}_3 = -17.0738469400920$

$\mathrm{N}_4 = 12020.8247024700$

$\mathrm{N}_5 = -3232555.03223330$

$\mathrm{N}_6 = 14.9151086135300$

$\mathrm{N}_7 = -4823.26573615910$

$\mathrm{N}_8 = 405113.405420570$

$\mathrm{N}_9 = -0.238555575678490$

$\mathrm{N}_{10} = 650.175348447980$

The range of validity of Equation 1 is $273.15 \text{ K} \le T \le 647.096 \text{ K}$ ($0°\text{C} \le t_C \le 373°\text{C}$).

IAPWS-06 Equation for Sublimation Pressure as a Function of Temperature

$$p_{ws} = 0.611657 \cdot \mathrm{EXP}((1/\theta) \cdot (a_1 \cdot \theta^{b_1} + a_2 \cdot \theta^{b_2} + a_3 \cdot \theta^{b_3})) \text{ [kPa]} \qquad (4\text{-}2)$$

where

$$\theta = T/273.16$$

$T = t_C + 273.15$, where t_C is temperature in degrees Celsius

$a_1 = -0.212144006\text{E}2$

$a_2 = 0.273203819\text{E}2$

$a_3 = -0.610598130\text{E}1$

$b_1 = 0.333333333\text{E-}2$

$b_2 = 0.120666667\text{E}1$

$b_3 = 0.170333333\text{E}1$

The range of validity of Equation 2 is $50 \text{ K} \le T \le 273.16 \text{ K}$ ($-223°\text{C} \le t_C \le 0.01°\text{C}$).

IAPWS-IF97 So-Called *Backward Equation* for Saturation Temperature, t_{C_dp}, from Saturation Pressure [kPa]

$$t_{Cdp} = (0.5 \cdot (\mathrm{N}_{10} + D - ((\mathrm{N}_{10} + D)^2 - 4 \cdot (\mathrm{N}_9 + \mathrm{N}_{10} \cdot D))^{0.5})) - 273.15 \quad [°\mathrm{C}] \qquad (4\text{-}3)$$

where

$$\beta = (p_{ws}/1000)^{0.25} \quad [p_{ws} \text{ units in kPa}]$$

$$D = 2 \cdot G/(-F - (F^2 - 4 \cdot E \cdot G)^{0.5})$$

$$E = \beta^2 + \mathrm{N}_3 \cdot \beta + \mathrm{N}_6$$

$$F = \mathrm{N}_1 \cdot \beta^2 + \mathrm{N}_4 \cdot \beta + \mathrm{N}_7$$

$$G = \mathrm{N}_2 \cdot \beta^2 + \mathrm{N}_5 \cdot \beta + \mathrm{N}_8$$

***Backward equation* to calculate frost point t_{wvs_ice} knowing p_{wvs} in kPa.** As of 2012, IAPWS had not issued a backward equation to calculate p_{wvs} [kPa] from t_{wvs_ice} [°C]. The following curve fit equation was developed by the author in March 2012.

$$t_{wvs\_ice} = i_1 + i_2 \cdot S + i_3 \cdot S^2 + i_4 \cdot S^3 + i_5 \cdot S^4 + i_6 \cdot S^5 + i_7 \cdot S^6 \quad [°\mathrm{C}] \qquad (4\text{-}4)$$

where

$$S = \ln(p_{wvs\_ice} \cdot 1000) \quad [p_{wvs\_ice} \text{ kPa}]$$

$i_1 = -60.5768$

$i_2 = 7.35177$

$i_3 = 0.25247$

$i_4 = 0.00891115$

$i_5 = 3.22863\text{E-}04$

$i_6 = 9.70878\text{E-}06$

$i_7 = 1.48541\text{E-}07$

Table 4-1 lists other important properties and constants required for psychrometric calculations. The data specific to H_2O are shaded. There is no need to memorize these constants. They are used in developing the psychrometric algorithms and form a part of most software calculations; however, once correctly entered into a program listing, the program user no longer needs to enter or refer to them.

C_{P_WV} values in Table 4-1 and throughout this text are based on a curve fit of moist air properties of ASHRAE RP-1485 (2009) The curve fit applies to the temperature range of –15°C (+5°F) to 30°C (86°F). These C_{P_WV} values differ slightly from the published values for pure water because they account for the slight interaction between *water vapour* molecules and between *water-vapour* and *dry air* molecules. Others have used values of

C_{P_WV}-SI from 1.805 to 1.89 J/(kg_wv·°C) and C_{P_WV}-IP from 0.44 to 0.45 Btu/(lb_{WV}·°F). The accuracy of psychrometric calculations at normal temperatures is slightly improved using the curve fit values herein; also the SI and I-P values are exact conversions, which eliminates the mysterious inconsistency of the ASHRAE SI and I-P C_{P_WV} values that existed prior to 2005.

Some psychrometric property software programs utilize polynomial equations for C_{P_WV} and C_{P_DA} values. The improved accuracy in the range of normal air-conditioning process calculations is not significant, but no harm is done. Polynomial equations may be justified for calculations outside of the normal air-conditioning range.

Dry Air, Saturated Water Vapour, and Fog Mixtures

Discussions in this chapter are based on moist air being a mixture of *dry air* and either saturated or superheated *water vapour*. This covers 99%+ of psychrometric applications. It is also possible for the mixture to contain suspended minute particles of liquid water or solid water. Fog and clouds are examples of this. Fog conditions are briefly covered in this book in Chapter 24, “Frequently Asked Questions.”

Mixtures of moist air and suspended minute ice particles apply to high altitude clouds and are not covered in this text. In some processes, ice is formed on the surfaces of heat exchangers during the process. Readers are cautioned that calculation of refrigeration capacity must account for the latent heat of the ice formed in the process.

Table 4-1—Physical Constants or Assumed Constants

Systéme International (SI) Units	Inch-Pound (I-P) Units
$R_U = 8314.472$ J/(kmol·K)	$R_U = 1545.349$ ft lb_F/(lbmol·R)
$R_{WV} = 461.524$ J/(kg_{WV}·K)	$R_{WV} = 85.780$ ft lb_F/(lb_{M_WV}·R)
$R_{DA} = 287.042$ J/(kg_{DA}·K)	$R_{DA} = 53.350$ ft lb_F/(lb_{M_DA}·R)
$M_{WV} = 18.015268$ kg/(kg·mol)	$M_{WV} = 18.015268$ lb/(lb·mol)
$M_{DA} = 28.966$ kg/(kg·mol)	$M_{DA} = 28.966$ lb/(lb·mol)
$M_{WV}/M_{DA} = 0.621945$; $M_{DA}/M_{WV} = 1.607858$	$M_{WV}/M_{DA} = 0.621945$; $M_{DA}/M_{WV} = 1.607858$
$C_{P\_WV} = 1.84$ kJ/(kg_{WV}·°C)	$C_{P\_WV} = 0.44$ Btu/(lb_{WV}·°F)
$C_{P\_DA} = 1.006$ kJ/(kg_{DA}·°C)	$C_{P\_DA} = 0.24$ Btu/(lb_{DA}·°F)
$C_{P\_ICE} = 2.0$ kJ/(kg_{ICE}·°C)	$C_{P\_ICE} = 0.50$ Btu/(lb_{ICE}·°F)
$C_{P\_W} = 4.186$ kJ/(kg_{WATER}·°C)	$C_{P\_W} =1.0$ Btu/(lb_{WATER}·°F)
$h_{G\_WV\_0^\circ C} = 2499.86$ kJ/(kg_{wv}) and 101.325 kPa	$h_{G\_WV\_0^\circ F} = 1060.63$ Btu/(lb_{WV}) and 14.695959 psia; [32°F = –17.78°C]
$h_{IF\_H2O\_0^\circ C} = 333.42$ kJ/kg	$h_{IF\_H2O\_32^\circ F} = -143.34$ Btu/lb
$h_{F\_H2O\_0.00^\circ C} = 0.00$ kJ/kg	$h_{F\_H2O\_32^\circ F} = 0.000$ Btu/lb
$T_{ABS\_ZERO} = -273.15^\circ C$	$T_{ABS\_ZERO} = -459.67\ ^\circ F$
$p_{BAR\_\text{SeaLevel}} = 101.325$ kPa	$p_{BAR\_\text{SeaLevel}} = 14.695959$ psia (29.92126 in. *H*)
p_wv_@ t °C (See equation of state)	p_wv_@ t °F (See equation of state)
AIR $T_C = -140^\circ C$; $P_C = 3.76$ MPa	AIR $T_C = -220^\circ F$; $P_C = 545.3$ psia
N_2 $T_C = -146.96^\circ C$; $P_C = 3.3978$ MPa	N_2 $T_C = -232.5^\circ F$; $P_C = 492.8$ psia
O_2 $T_C = -118.57^\circ C$; $P_C = 5.043$ MPa	O_2 $T_C = -181.4^\circ F$; $P_C = 731.4$ psia
Ar $T_C = -122.29^\circ C$; $P_C = 4.8979$ MPa	Ar $T_C = -188.1^\circ F$; $P_C = 710.4$ psia
H_2O $T_C = 373.946^\circ C$; $P_C =$ 22.064 MPa	H_2O $T_C = 705.1^\circ F$; $P_C = 3200.1$ psia

M *in the table is the molecular mass of the substance.*
T_C *and* P_C *are critical temperature and critical pressure.*
Values in Table 4-1 are consistent with the Herrmann-Kretzchmar-Gatley ASHRAE Research Project 1485 (RP-1485) in 2009.

At first glance a psychrometric chart appears ominous
with its myriad of tiny lines crisscrossing in every direction.
Actually, once understood, this apparent complexity is readily dismissed
and the psych chart can become a relatively simple and invaluable design tool.
—Julian R. Fellows, Emeritus Professor of M.E., University of Illinois

5 Basics of the Psychrometric Chart

The psychrometric chart is a convenient and useful tool for determining statepoint moist air psychrometric properties and visualizing the changes of moist air properties in a sequence of psychrometric processes; e.g., (a) as the outside and return air mixes and (b) proceeds through heating and cooling coils, (c) the supply fan, and (d) supply duct to the conditioned space. This chapter is intended to give the reader an overview of the chart to help in overall understanding and to add clarity and meaning as each psychrometric property is developed and discussed in following chapters.

The main reader objective in this chapter should be to become familiar with the *look* of the psychrometric chart, the *names* of the lines plotted on the chart, and their *general orientation* with respect to one another. It is not important at this point to know the definition of each term or even its pronunciation. If the reader is especially curious about one term, then he or she may go directly to the chapter on that property.

A psychrometric chart is based on a specified barometric pressure or elevation with respect to sea level. The most common chart is based on 101.325 kPa barometric pressure. This is the normal barometric pressure at sea level and 15°C. The *sea level* psychrometric chart can be used without significant error for elevations between 400 metres above and below sea level. ASHRAE publishes SI psychrometric charts for sea level (101.325 kPa), +750 metre altitude (92.66 kPa), +1500 metre altitude (84.54 kPa), and +2250 metre altitude (77.04 kPa). Barometric pressure is related to altitude using the equation shown in Chapter 14, “Barometric Pressure.”

Haines (1961) and Threlkeld (1970) provide instructions for calculating and plotting a psychrometric chart for any specific altitude above or below sea level. Some psychrometric software programs will instantly develop and display a printable psychrometric chart for any altitude.

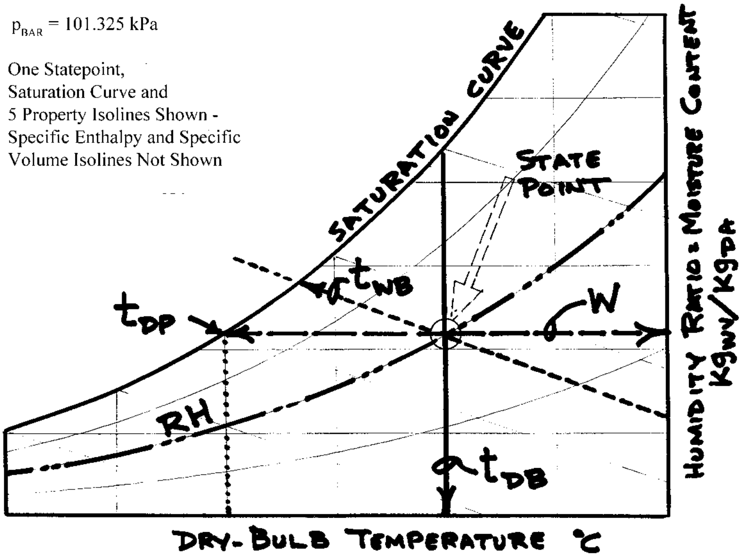


Figure 5-1—Basic psychrometric chart.

Statepoints and processes may be added and calculations made, and the calculation results and the displayed psychrometric chart may then be printed to paper or to a computer file.

Figure 5-1 illustrates a skeleton psychrometric chart including selected property lines and the saturation curve for water. Many users of psychrometric charts are only interested in two or three properties and naturally develop the habit of looking at the psychrometric chart and seeing only those property lines of interest. It is an unconscious action that many of us do everyday as we occasionally look at our automobile speedometer. Some see kilometres per hour and others see miles per hour and yet both values appear on most speedometers. The reader may be somewhat confused when examining a psychrometric chart for the first time, but as Professor Fellows's quotation at the chapter heading implies, those who spend time with the chart become very comfortable using it. The properties isolines plotted on the chart are:

- Dry-bulb temperature isolines in °C, t_{DB}
- Humidity ratio isolines in kilograms (sometimes in grams) *water vapour* per *kilogram dry air*, *W*
- Thermodynamic wet-bulb temperature isolines in °C, t_{WB}
- Relative humidity isolines in percent, RH
- *Water vapour* saturation curve plotted as humidity ratio versus temperature at which the gas phase of water and the liquid phase are in equilibrium, ~t_{DP}

For chapter simplification, these important property isolines are omitted:

- Specific enthalpy (an important energy content property) isolines in kilojoule per *kilogram dry air*, *h*
- Specific volume isolines in cubic metres of moist air per *kilogram dry air*, *v*

Dew-point temperature is an important property; however, dew-point temperature isolines are seldom plotted on a psychrometric chart. This is also true for *water vapour* pressure isolines. Both dew-point temperature and *water vapour* pressure isolines, if plotted, would be horizontal lines and could be confused with the horizontal humidity ratio isolines. Some charts display scales for dew-point temperature and *water vapour* pressure on the far right of the chart. Dew-point temperature isolines can always be easily drawn (or visualized) on any psychrometric chart because dew-point temperature and dry-bulb temperature are equal at the intersection of the *water vapour* saturation curve. All points on a horizontal line through an intersection of the saturation curve and a dry-bulb temperature isoline have the same dew-point temperature.

Knowing barometric pressure (or its equivalent in altitude) and any two of the properties in the previous lists will establish the psychrometric statepoint and thereby all other psychrometric properties. Dry-bulb temperature is easily measured and is normally used along with one other property to fix the statepoint location on the psychrometric chart and then determine all other properties. Relative humidity or psychrometer wet-bulb temperature are often the second property used with dry-bulb temperature and altitude to fix a statepoint.

The psychrometric chart conceived by Dr. Willis Carrier in 1904 used *dry-bulb* temperature and water vapour density as rectangular plotting coordinates. With these coordinates, *dry-bulb* temperature isolines are all exactly vertical and evenly spaced.

In 1908, William Grosvenor was the first to use humidity ratio as a plotting coordinate (in his case, the horizontal scale). This simplified calculations. A majority of later psychrometric charts followed the lead of Grosvenor. He also included auxiliary lines of (a) constant cooling; (b) specific heat, specific volume, and density of humid (saturated) air; (c) specific volume and density of dry air; and (e) latent heat of vaporization of steam.

In 1923, Richard Mollier of Dresden, Germany, proposed a chart that looked quite similar to the Grosvenor chart but used *specific enthalpy* as an oblique coordinate and *humidity-ratio* as the ordinate with both plotted using evenly spaced grid lines. It is possible that Mollier patterned the appearance of his 1923 chart after Grosvenor's 1908 chart. Mollier's significant contribution was the use of a skewed enthalpy coordinate replacing the vertical temperature coordinate. ASHRAE adopted the skewed enthalpy coordinate in 1961. Mollier is more famous for other charts in the field of thermodynamics and most of these are also called Mollier charts. Mollier's 1923 hx diagram (originally named "ix diagram") did not resemble the Carrier chart; however, Keppler, by carefully selecting the oblique angle for specific enthalpy lines and using an evenly spaced *dry-bulb* scale along the zero *humidity-ratio* isoline, made the Mollier diagram appear at first glance identical to the original Carrier psychrometric chart. Careful examination of the current ASHRAE *enthalpy-humidity ratio* psychrometric chart reveals that only one *dry-bulb* isoline is vertical. All other dry-bulb temperature isolines are straight but not parallel to one another. They diverge slightly as they extend into regions of higher *humidity ratios*.

Some practitioners in Europe and Asia use the current Mollier hx diagram. Nevertheless, the ASHRAE and Mollier diagrams produce identical results. To convert from one to the other, first rotate the chart 90° and then look at the rotated image in a mirror or, better yet, plot one on semitransparent material and then flip it over and rotate.

Many psychrometric texts cover construction of the psychrometric chart. A most informative presentation by Professor W.F. Stoecker (1972) starts with a dry-bulb abscissa (horizontal) scale and a *water vapour* pressure ordinate (vertical) scale. Saturation *water vapour* pressure versus temperature is plotted, which makes it perfectly clear that the curved line at the left of the psychrometric chart is the saturation curve for *water vapour*. Stoecker then converts the ordinate scale to humidity ratio using the equation presented in Chapter 17, "Humidity Ratio."

Prior to the advent of personal computer psychrometric software, the psychrometric chart was used (1) for determining statepoint properties, (2) as a graphical display of processes, and (3) as an extremely useful graphic tool for solving psychrometric process problems. The psychrometric chart with oblique *enthalpy-humidity ratio* coordinates provides slightly more accurate graphical solutions of mixing processes than the older-style charts using *dry bulb-humidity ratio* coordinates.

In 1990, approximately 75% of practitioners used the psychrometric chart as a tool in solving psychrometric problems, and the balance used computer software. In the first decade of the twenty-first century, most practitioners will rely on psychrometric software based on algorithms with 99% or greater accuracy. The chart itself will no longer be used as a graphical solution tool. As a secondary function, the software program will generate a psychrometric chart showing the statepoints and the psychrometric process lines connecting the statepoints. It will be displayed to provide a visual picture of the cycle of psychrometric processes and statepoints. Since the chart will no longer be used as a graphical solution tool by the practitioner, the plotting grid will probably revert to rectangular plotting coordinates of *dry bulb* and *humidity ratio*. All statepoints will either be original data entries or the results of accurate calculations—not the result of graphical plotting.

The demise of printed psychrometric charts will probably be similar to the demise of the slide rule (replaced by the handheld calculator) and the “K & E” log-log and other special plotting papers (replaced by plotting capabilities of computer spreadsheet software programs). It is only a matter of time before the once familiar pads of psychrometric charts from major air-conditioning and dehumidification manufacturers will no longer be available. Put a collection away for your grandchildren (some day they may have the value of baseball cards of the ‘40s and ‘50s).

PSYCHROMETRIC CHART COORDINATES: MUCH ADO ABOUT VERY LITTLE

The plotting coordinates for most graphs of any kind are based on a horizontal scale (the abscissa) and a vertical scale (the ordinate) and are called *rectangular* coordinates. At first glance, the ASHRAE psychrometric chart appears to be drawn or plotted using rectangular coordinates with dry-bulb temperature as the horizontal scale and humidity ratio as the vertical scale. Closer examination reveals that the dry-bulb temperature isolines are not precisely parallel to one another and that the distance between dry-bulb isolines in the upper portion of the chart (at high humidity ratio values) is greater than this distance at the bottom of the chart (where the humidity ratio is zero). In other words, the dry-bulb temperature lines fan out or diverge as the humidity ratio increases.

The common psychrometric chart in use today is actually drawn using (1) *specific enthalpy* values as an oblique coordinate (the *specific enthalpy* scale has uniform scaling and is at a slant) and (2) *humidity ratio* values as a vertical coordinate. (The ASHRAE psychrometric chart enthalpy scale is at an angle of 29° to the horizontal.)

The psychrometric chart using oblique enthalpy-humidity ratio coordinates has several minor advantages over a rectangular coordinate humidity ratio-dry-bulb temperature chart:

- Enthalpy isolines are straight and parallel and evenly (uniformly) spaced. Identical edge scales are used on each side of the psychrometric chart rather than plotting enthalpy isolines. In contrast, the enthalpy scale on a rectangular coordinate *dry bulb* and *humidity ratio* psychrometric chart is not uniform. The use of edge scales eliminates the confusion of showing both thermodynamic wet-bulb temperature and enthalpy lines and permits direct reading of an enthalpy value in the middle of the chart by the use of a straight edge or the edge of a piece of paper correctly aligned on the left and right enthalpy scales.

- Thermodynamic wet-bulb temperature isolines are straight; however, they are not precisely parallel to one another. In contrast, thermodynamic wet-bulb temperature lines on a rectangular coordinate humidity ratio-dry-bulb temperature chart are slightly curved.
- Mixture processes can be correctly and accurately represented as straight lines and the resultant statepoint from mixing masses of moist air from two different statepoints can be quickly and accurately shown.
- A perceived disadvantage (by some) is that dry-bulb temperature isolines, although straight, are not parallel.

From an educational and also an overall appearance standpoint there is an insignificant difference between the oblique-coordinate specific enthalpy-humidity ratio psychrometric chart and the rectangular coordinate chart. If the reader is more comfortable thinking about a chart with rectangular coordinates and *true* vertical dry-bulb temperature isolines, then no harm is done. *If the reader encounters a psychrometric-chart-coordinate-final question in his/her appearance on a billion-ruble TV quiz show, then the winning response should be that most psychrometric charts are drawn with oblique enthalpy-humidity ratio coordinates ... but who knows how educated the answer judge may be?* (The reader was forewarned that I promised *much ado about very little*!)

QUICK SUMMARY

- Different psychrometric chart plotting coordinates are used but differences in chart appearance or utility are minor.
- Early in the first decade of the twenty-first century, virtually all psychrometric calculations will use accurate software algorithms and the difference between charts will be a moot point.
- The psychrometric chart is simply the graphic representation of the property interrelationships of mixtures of air and *water vapour.*

One has to pay attention to the way in which specific enthalpy and specific volume are expressed for moist air psychrometrics. Specific enthalpy, specific volume, and humidity ratio are defined as properties of the moist air mixture per unit mass of the "dry air" component *rather than per unit mass of the moist air.*

6 Underlying Theory and Algorithms

There is no correct location for this chapter for the reasons listed below. The reader is therefore left to his/her own decision. This chapter can be read or overviewed now, read after the chapters covering individual psychrometric properties as a consolidating overview, or (perish the thought) skipped altogether.

- For those having some familiarity with psychrometrics, a location before the individual property chapters may be helpful in that it gives an overall perspective.
- For those new to the subject and those who have never delved into statepoint property calculations, a location after the individual property chapters may be less threatening because the reader can comprehend each definition and equation one at a time without being overwhelmed by the number of new terms.

THE FUNDAMENTAL LAWS, MODELS, OR PRINCIPLES UNDERLYING PSYCHROMETRICS

Contrary to the belief of some, psychrometrics is not based on witchcraft, smoke and mirrors, or fog (pun intended). Empirical formulae based on curve fit of meteorological records and experimental data were used in the past, but since Willis H. Carrier presented his treatise on rational psychrometrics in 1911, the science of psychrometrics has been solidly based on fundamental laws of physics. In this book, *scientific laws, principles, rules,* and *models* all have the same meaning. They are statements or equations that have never proven to be false. Psychrometrics makes use of the following laws, principles, rules, or models:

- The *ideal gas* equation of state. This subject is so important that it is covered in a separate chapter (Chapter 7, "Ideal Gas"). The ideal gas equation of state applies to the substances *dry air*, *water* *vapour*, and their *moist air mixture*. Note that in this text, the terms *ideal gas* and *perfect gas* are synonymous.

- Conservation of energy. Energy cannot be created or destroyed. It can only be changed or transformed from one form of energy to another.
- Conservation of the mass of dry air and the mass of H_2O.
- Dalton’s law of partial pressures. In a mixture of different ideal gases, the pressure of the mixture is equal to the sum of the (partial) pressures of each of the component gases. The individual gases share a common volume and temperature. Each gas behaves as if it alone occupied the volume.
- The Gibbs-Dalton law for the energy, enthalpy, and entropy of a mixture, which states that for mixtures that comply with Dalton's law the energy, the enthalpy, and the entropy are equal to the sum of the energies, the enthalpies, and the entropies of their individual components.

Psychrometric calculations require the properties of H_2O in each of its three phases. These properties are based on extensive laboratory measurement and the calculations of theoretical and statistical thermodynamics at the molecular level. From these come two *water vapour* equilibrium or saturation curves showing saturation temperature vs. *water vapour* pressure (one for *water vapour* and liquid water in equilibrium and one for *water vapour* and ice in equilibrium).

For a given barometric pressure, by use of the ideal gas law, the water vapour pressure (p_{WVS}) of the p_{WVS} vs. t curve can be converted to humidity ratio (W) of the mixture of *dry air* and *water vapour*.

The relationship between humidity ratio and water vapour saturation temperature is exponential—not linear. For the range of temperatures at the earth’s surface, the maximum possible mass of *water vapour* per unit mass of *dry air* approximately doubles for each 10°C rise in temperature.

PHYSICAL CONSTANTS, ASSUMPTIONS, AND CUSTOMS OF CONVENIENCE

With any science or calculation methodology there are physical constants, conventions (sometimes called conveniences), and assumptions, and the science of psychrometrics is no exception. The following apply in psychrometrics based on the RP-1485 (2009) ASHRAE research:

- The universal gas constant on the basis of the carbon 12 scale is 8314.472 J/(kmol·K).
- The N_2 and O_2 composition of *dry air* up to 11000 metres elevation above sea level is constant.

- *Dry air* behaves as a perfect gas in which $p_G \cdot v_G = R_G \cdot T_G$. *Dry air* has a molecular mass of 28.966 kg/kmol. The individual gas constant for *dry air* is 287.042 J/(kg·K), which is the result of dividing the universal gas constant by the molecular mass of *dry air.*
- *Water vapour* behaves as a perfect gas with a molecular mass of 18.015268 kg/(kmol). The individual gas constant for *water vapour* is 461.524 J/(kg·K).
- The ratio $R_{DA}/R_{WV} = M_{WV}/M_{DA} = 0.621945$.
- The total pressure of the *dry-air* and *water-vapour* components is equal to the mean annual barometric pressure at the location of interest, i.e., $p_{TOTAL} = p_{BAR} = p_{DA} + p_{WV}$, where p_{DA} and p_{WV} are called the partial pressure of *dry air* and the partial pressure of *water vapour.*
- The total pressure in most psychrometric processes is constant. Therefore, if the partial pressure of *water vapour* decreases during a process, it follows that the partial pressure of the *dry air* must increase. The exception to a constant total pressure assumption is a process involving the compression of a gas.
- By custom and convenience, calculations are based on the unit mass of the *dry air* component. This is logical because the mass of *dry air* is constant in most psychrometric processes, whereas the mass of *water vapour* associated with the mass of *dry air* varies throughout the year and throughout many processes as a consequence of humidification or dehumidification.
- By tradition and for convenience in calculations, many air-conditioning process heat and mass transfer equations contain a volumetric flow term—not a mass flow term. Because the ultimate calculation is based on the mass flow of the *dry air* component, it follows that the equation must include a specific volume term to convert volumetric flow to the mass flow of *dry air.*
- In many shortcut equations, the specific volume term is hidden in an equation constant. This creates no problem as long as the user of the shortcut abbreviated equation verifies that the hidden specific volume in the equation is reasonably close to the actual specific volume.

THE ALGORITHMS

Simple algorithms are developed within some chapters. They are shown here for the purpose of overview and to demonstrate how few algorithms are actually required. Readers new to psychrometrics should grasp that there are only *six principal equations* without trying to understand the development of any of the equations.

The six principal psychrometric equations identified below in numbered parentheses, (#), are used to compute statepoint properties of moist air (the mixture of *dry air* and *water vapour*). Three property values are required to fix the statepoint of moist air. These are usually p_{BAR} and t_{DB} and a third property relating to *water vapour*. The third property is quite often t_{WB}, t_{DP}, or RH, all of which are measurable properties. Optionally, the third property could be a previously established value of W, h, or v (properties that are difficult to measure). These equations may be used in different sequences depending on which properties are known.

For identification of symbols and subscripts, refer to the inside back cover.

Write the p, v, T Relationship for Each of the Components

$$p_{DA} \cdot V_{DA} = m_{DA} \cdot R_{DA} \cdot T_{DA} \text{ for } \textit{dry air} \qquad \textbf{(ideal gas law)}$$

$$p_{WV} \cdot V_{WV} = m_{WV} \cdot R_{DA} \cdot T_{WV} \text{ for } \textit{water vapor} \qquad \textbf{(ideal gas law)}$$

Note in a mixture of *dry air* and *water vapour*

$$V_{DA} = V_{WV} \qquad \text{and} \qquad T_{DA} = T_{WV} \qquad \textbf{(Dalton's law)}$$

Calculation of Relative Humidity from Its Definition

$$\mathbf{RH} = \boldsymbol{p_{WV}/p_{WVSAT}} \text{ both at the same } p_{BAR} \text{ and } t_{DB} \qquad \textbf{(6-1)}$$

(Basic definition of relative humidity)

Calculation of Humidity Ratio First from Its Definition and Then as a Function of p_{WV}

$$\boldsymbol{W = m_{WV}/m_{DA}} \qquad \textbf{(Basic definition of humidity ratio)}$$

$$W = 0.621945\, p_{WV}/p_{DA} \qquad \text{from ideal gas laws above)}$$

And since

$$p_{BAR} = p_{TOT} = p_{WV} + p_{DA} \qquad \text{(Dalton's law)}$$

$$\boldsymbol{W = 0.621945\, p_{WV}/(p_{BAR} - p_{WV})} \qquad \textbf{(6-2)}$$

See Chapter 14, "Barometric Pressure," for the calculation of p_{BAR} from an elevation above or below sea level.

Calculation of Enthalpy

$$h = h_{DA} + h_{WV} \qquad \text{(Gibbs-Dalton)}$$

$$\mathbf{h = C_{P\,DA} \cdot (t - t_{REF}) + W \cdot [(h_{G\,WV@\,t\,REF} + C_{P\,WV} \cdot (t - t_{REF})]} \qquad \textbf{(6-3)}$$

and by custom and for convenience, t_{REF} is normally chosen as 0°C for *DA* and *WV.* Note that C_{PDA} and C_{PWV} are assumed to have constant values in the range of –40°C to +65°C. Readers familiar with psychrometrics in I-P units should note that the I-P t_{REF} temperature for water is liquid water at 32°F and the I-P t_{REF} temperature for dry air is 0°F.

Note that t_{REF} is 0°C in SI, which allows the t_{REF} term to be omitted.

Calculation of Specific Volume of the Dry Air Component

$$\mathbf{p_{DA} \cdot v_{DA} = R_{DA} \cdot T \text{ and rearranging } v_{DA} = R_{DA} \cdot T/p_{DA}} \qquad \textbf{(Ideal gas law)}$$

and substituting $p_{DA} = p_{BAR} - p_{WV}$

$$\mathbf{v_{DA} = R_{DA} \cdot T/(p_{BAR} - p_{WV})} \qquad \textbf{(6-4)}$$

Calculation of Thermodynamic Wet-Bulb Temperature in an Adiabatic (Zero Heat Transfer) Saturation Process

$$h_1 + (W_{SAT2} - W_1) \cdot h_{F2} = h_{SAT2}$$

where

W_{SAT2}, h_{F2}, and h_{SAT2} are functions of t_{WB2} and

$$t_{WB2} = t_2 = t_{DP2} = t_{DB2} = t_{\,H2O\,ADDED}$$

By substituting and rearranging:

$$h_1 = C_{PDA} \cdot t_{DB1} + W_1\,(h_{GREF} + C_{PWV} \cdot t_{DB1})$$

$$h_{SAT2} = C_{PDA} \cdot t_{DB2} + W_{SAT2}\,(h_{GREF} + C_{PWV} \cdot t_{DB2})$$

$$h_{F2} = C_{P\,H2O} \cdot t_{DB2} \text{ if } H_2O \text{ is added as a liquid}$$

$$\mathbf{W_1 = (W_{WVSAT@\,TWB}\,(h_{GREF} + C_{PWV} \cdot t_{WB} - h_{F2}) - C_{PDA}\,(t_{DB1} - t_{WB}))/(h_{GREF} - h_{F2} + C_{P\,h20} \cdot t_{DB1})} \qquad \textbf{(6-5)}$$

Note that different equations are required below 0°C.

Calculation of Dew-Point Temperature or, Alternatively, Water Vapour Pressure

This calculation uses the IAPWS relationship of p_{WV} *vs.* T_{WV} for water vapour over solid ice or liquid water as appropriate. If T_{WV} is known, the equation is solved for p_{WV}. If p_{WV} is known, the equation is used to solve for T_{WV}.

The IAPWS equations are shown on page 26 of Chapter 4, "Water Vapour," and are not repeated here.

Some of the equations require specific heat capacity values for: c_{P_DA}, c_{P_WV}, c_{P_H2O}, and $c_{P\ ICE}$. Note that the specific values may be constant over the range of temperatures being analyzed or they may vary as a function of temperature. For the substances *dry air*, *water vapour, liquid water,* and *ice* in the range of –40°C to +65°C, the variance in specific heat capacities is so small that unless an extreme degree of accuracy is required, an average constant value for each of the specific heat capacities is normally used.

Some may question whether or not a substance having a variable specific heat capacity in the gas phase qualifies as an ideal gas. The only qualification for ideal gas behavior is that at any statepoint the substance must follow the relationship $p_G \cdot v_G = R_G \cdot T$. There are many substances with varying specific heat capacities that, nevertheless, are still classified as ideal gases.

> *We like the ideal gas law because it is the very simplest and*
> *leads to all sorts of nice consequences and because it well approximates*
> *the behavior of all gases from hydrogen to sulfur hexafluoride*
> *at not too high a pressure, say up to 500,000 Pa.*
> *Let us see what interesting and useful treasures appear*
> *when we apply the first law to ideal gases.*
> *—Octave Levenspiel*

7 Ideal Gas

THE IDEAL GAS EQUATION OF STATE

For those who wish to quickly peruse psychrometrics, this chapter may be skipped; however, the reader may find it of interest because many psychrometric equations are easily derived from the *ideal gas equation of state.* This chapter will discuss the *ideal gas equation of state* and then show the derivation of equations for *humidity ratio, relative humidity,* and *specific volume.*

Preliminary Comments

Most psychrometric processes involve the addition or removal of heat and/or H_2O as liquid, solid, or vapour from a steady flow of air. *Moist air* is a mixture of two gases discussed in previous chapters: *dry air* (N_2, O_2, and Ar, plus traces of other gases) and *water vapour* (the gas phase of water). In most psychrometric processes, the mass flow of the *dry air* component is constant throughout the process while the mass flow of the *water vapour* is changing. Therefore, the *total* mass flow of *moist air* in a process is variable, making it somewhat difficult to track.

In order to simplify calculations, psychrometric pioneers in the period from 1850 to 1910 borrowed a technique from chemical engineering and based many of the psychrometric properties on the unit mass of the *dry air* component of the *moist air* mixture. The psychrometric properties of *specific volume* ($\mathrm{m^3/kg_{DA}}$), *humidity ratio* ($\mathrm{kg_{WV}/kg_{DA}}$), and *specific enthalpy* ($\mathrm{J/\ kg_{DA}}$) are expressed per unit mass of the *dry air* component of the mixture. This also applies to the specific heat capacity of air (moist air), C_P ($\mathrm{J/[kg_{DA}\cdot K]}$). Expressing these properties in this manner simplifies process calculations; e.g., in order to compute the total heat transfer in a process it is only necessary to determine the mass flow of the *dry air* component (which is usually constant throughout the process) and multiply this times the difference between the entering and leaving specific enthalpies:

$$q_{TOT} = \dot{m}_{DA} \cdot (h_{ENT} - h_{LVG}) .$$

Adding units shows that the units balance:

$$q_{TOT}\ \{\mathrm{J/s}\} = \dot{m}_{DA}\ \{\mathrm{kg_{DA}/s}\} \cdot (h_{ENT} - h_{LVG})\ \{\mathrm{J/kg_{DA}}\}.$$

It is important to recognize the subtle difference between the mass flow of the *dry air* component and the total mass flow when deriving or using psychrometric equations.

Ideal Gas

Some scientists make distinctions between an *ideal gas* and a *perfect gas*. In this text they are treated as identical and the reader may use the terms interchangeably. An ideal gas is one that satisfies Equation 7-1 over a wide range of conditions. Expressed in words: at any given state, the product of gas pressure times its specific volume is equal to the product of the gas constant (for the gas under investigation) and the absolute temperature of the gas. This equation is a combination of Boyle's and Charles' gas laws. No real gas precisely follows this equation, but many follow it with better than 99% accuracy over a wide range of conditions.

$$p_X \cdot v_X = R_X \cdot T_X \qquad (7\text{-}1)$$

where

p = pressure
v = specific volume
R = the gas constant, i.e., a proportionality constant
T = the absolute temperature
X = a subscript to denote the specific gas

R_X is a proportionality constant for a particular gas and is called the *gas constant.* This equation is designated an *equation of state* in that it relates the state properties p, v, and T. *Any gas for which this equation is valid is called an ideal gas (or a perfect gas*). Note that when using the ideal gas equation of state, the pressure and temperature must be expressed as absolute pressure and absolute temperature.

The gas constant, R_X, is related to the *universal gas constant,* R_U (which has the same value for all gases), by the relationship:

$$R_X = R_U/M_X$$

where

R_X = gas constant for the specific gas
R_U = universal gas constant that equals 8314.472 J/(kmol·K)
M_X = relative molecular mass (mass of one mole of the substance)

For *dry air, M* is 28.966 kg/kmol, so that for *dry air* R_{DA} is 287.042 J/(kg·K). For *water vapour, M* is 18.015268 kg/kmol. *Water vapour* R_{WV} is 461.524 J/(kg·K).

Table 7-1—The Universal Gas Constant—Different Units

8.314472 J/mol·K	0.0624 m^3·mm Hg/(mol·K)
8.314472 kJ/(kmol·K)	1.986 cal{IT}/(mol·K)
8.314472 Pa·m^3/(mol·K)	1545.349 ft lbf/(lbmol·R)
8.314472 N·m/(mol·K)	10.732 psia ft^3/(lbmol·R)
8314.472 J/(kmol·K)	0.7302 ft^3 atm/(lbmol·R)
0.082057 litre·atm/(mol·K)	1.986 Btu{IT}/(lb mol·R)
82.057 litre·atm/(kmol·K)	
0.082057 m^3·atm/(kmol·K)	*{IT} = International Table*

Caution—the units of the ideal gas equation of state are no different than any other equation. The units *must* balance! The universal gas constant is often converted to other units, as shown in Table 7-1. Some texts present ten or more forms of the ideal gas equation of state. It is probably better to remember the single SI constant 8314.472 J/(kmol·K) and use conversions rather than memorize or retain a long list. The 1545.349 ft lbf/(lbmol·R) is probably familiar to anyone who studied science in inch-pound units.

A *mole* is a counting unit for molecules or other defined atomic or subatomic particles (examples of counting units include the terms *dozen* and *gross*). Using gas molecules, a mole (meaning a gram mole) of substance contains $6.023 \cdot 10^{23}$ molecules (Avogadro's number) of the substance. A kilomole contains $6.023 \cdot 10^{26}$ molecules of the substance. The *mole* is also that quantity of a substance (i.e., that number of atoms or molecules) having a mass that, measured in grams, is numerically equal to the atomic or molecular weight of the substance. In SI it is convenient to use a kilomole (kmol) instead of a mole (gmol). One kmol of carbon has a mass of 12 kg (exactly); 1 kmol of oxygen is 32 kg (very nearly). Stated otherwise, $M = 12$ kg/kmol for carbon, and $M \cong 32$ kg/kmol for O_2.

Using the relations $m = n \cdot \mathrm{M}$; $R_G = R_U/\mathrm{M}$; and $m = V/v$, other forms of the ideal gas equation are:

$$p \cdot V = m \cdot R_G \cdot T \qquad p \cdot V = n \cdot R_U \cdot T$$

where

n = number of moles
m = mass of substance
V = volume occupied by the mass
v = specific volume of the substance.

The ratio of molar (molecular) masses, $M_{WV}/M_{DA} = 18.015268/28.966 = 0.621945$, appears frequently in psychrometrics equations, as does its reciprocal, 1.607858.

Applicability of Equation 7-1 to the Gases Dry Air and Water Vapour

In general, when the gas (sometimes called a vapour) phase of a substance has a relatively high specific volume, experimentation has shown that the pressure, specific volume, and temperature are related by simple Equation 7-1, the *ideal gas equation of state*. A relatively high specific volume results when the gas is at a low pressure relative to its critical pressure or has a temperature high relative to its critical temperature. For dry air and water vapour near one atmospheric pressure and room temperature, this is the case. See Table 4-1 for critical temperature and pressure.

In general, *dry air* qualifies as an ideal gas because its temperature is high compared to the critical temperatures of its constituent gases. The ideal gas equation is surprisingly accurate for *dry air* over a wide range of temperatures and pressures. Less than one percent error is encountered for pressures as high as 3000 kPa at room temperature or for temperatures as low as –130°C at atmospheric pressure.

Water vapour in the atmosphere, even in a saturated state, qualifies as an ideal gas because the partial pressure of the *water vapour* is quite low at temperatures below 50°C. For *water vapour*, when the *water vapour* is mixed with *dry air* at temperatures between –40°C and +50°C at or near atmospheric pressure, the *ideal gas equation of state* has an accuracy of better than 99%.

The following equations for humidity ratio (W) and specific volume (v_{DA}) are easily derived using the ideal gas equation of state. These equations are used extensively in psychrometric property calculations.

Derive W—The Humidity Ratio Expressed as kg_{WV}/kg_{DA}

The relationship between W, p_{WV}, and RH is developed using the explicit definition of W and the two ideal gas equations of state for *dry air* and *water vapour*.

$$W = m_{WV} / m_{DA} \qquad \text{(the explicit definitions of humidity ratio)}$$

$$p \cdot V = m \cdot R_G \cdot T \qquad \text{and rearranging} \qquad m = p \cdot V/(R_G \cdot T)$$

$$m_{WV} = (p_{WV} \cdot V_{WV})/(R_{WV} \cdot T_{WV}) \text{ and}$$

$$(m_{DA} = (p_{DA} \cdot V_{DA})/(R_{DA} \cdot T_{DA}))$$

Using Dalton's law, the *water vapour* and *dry air* occupy the same volume ($V_{DA} = V_{WV}$) and are at the same temperature ($T_{DA} = T_{WV}$). Substituting for m_W and m_{DA} in the W equation and eliminating the volume and temperature terms that cancel leads to

$$W = ((R_{DA} \cdot p_{WV})/(R_{WV} \cdot p_{DA})) \text{ and } W = (287.042 \cdot p_{WV})/(461.524 \cdot p_{DA})$$

Therefore,

$$W = 0.621945 \cdot p_{WV}/p_{DA}$$

Also,

$$P_{TOTAL} = P_{BAR} = p_{WV} + p_{DA} \qquad \text{and} \qquad P_{DA} = P_{BAR} - p_{WV}$$

Therefore,

$$\boldsymbol{W = 0.621945 \cdot p_{WV}/(p_{BAR} - p_{WV}).}$$

Also,

$$RH = (100 \cdot p_{WS})/p_{WVS} \qquad \text{and} \qquad p_{WV} = (RH/100) \cdot p_{WVS}$$

and

$$W = 0.621945 \cdot p_{WVS} \cdot (\mathrm{RH}/100)/(p_{BAR} - p_{WVS} \cdot (\mathrm{RH}/100))$$

where

p_{WV} = partial pressure of the *water-vapour* component of the moist air mixture

p_{DA} = partial pressure of the *dry air* component of the moist air mixture

p_{BAR} = total pressure, i.e., atmospheric or barometric pressure

p_{WVS} = partial pressure of saturated *water vapour* at dry-bulb temperature

RH = relative humidity expressed in percent

Specific Volume and the Ideal Gas Equation of State

The equation for specific volume is a simple example of applying the ideal gas equation of state as shown below:

$$p_{DA} \cdot v_{DA} = R_{DA} \cdot T \qquad \text{and} \qquad p_{DA} = p_{BAR} - p_{WV}$$

$$v_{DA} = \frac{R_{DA} \cdot T}{P_{BAR} - p_{WV}} = \frac{287.042\ \mathrm{J}/(kg \cdot K) \cdot (t_{CELSIUS} + 273.15)}{P_{BAR} - p_{WV}}$$

Also, by substituting the relationship between W, p_{BAR}, and p_{WV}, the following relationship can be shown:

$$v_{DA} = \frac{287.042\ \mathrm{J}/(\mathrm{kg} \cdot \mathrm{K}) \cdot (t_{CELSIUS} + 273.15) \cdot (1 + 1.607858 \cdot W)}{p_{BAR}}$$

Specific Volume and Density of Dry Air at Sea Level Standard Pressure at Temperatures of 0°C and 20°C

$$v_{DA\_0°C} = 8314.472\{(\mathrm{Pa}\cdot\mathrm{m}^3)/(\mathrm{kmol}\cdot\mathrm{K})\}\cdot 273.15\{K\}/\rightarrow$$

$$(28.966\{\mathrm{kg/kmol}\}\cdot 101325\{\mathrm{Pa}\}) = 0.7738036\ \mathrm{m}^3/\mathrm{kg}_{\mathrm{DA}}$$

and

$$\rho_{DA\_0°\mathrm{C}} = 1.2923176\ \mathrm{kg}_{DA}/\mathrm{m}^3$$

$$v_{DA\_20°C} = 8314.472\{(\mathrm{Pa}\cdot\mathrm{m}^3)/(\mathrm{kmol}\cdot\mathrm{K})\}\cdot 293.15\{\mathrm{K}\}/\rightarrow$$

$$(28.966\{\mathrm{kg/kmol}\}\cdot 101325\{\mathrm{Pa}\}) = 0.8304614\ \mathrm{m}^3/\mathrm{kg}_{\mathrm{DA}}$$

and

$$\rho_{DA\_20°\mathrm{C}} = 1.2041499\ \mathrm{kg}_{\mathrm{DA}}/\mathrm{m}^3$$

The R_U in these equations is 8314.472 Pa·m^3/kmol·K. It is 1000 times greater than the value in the third row of Table 7-1 because it is expressed per kmol rather than per mol. R_U is expressed in these units for this equation to simplify the presentation. The derivation of the table's third row first column R_U from the basic value is shown below for those who have interest.

$$\frac{8314.472\ \mathrm{J}}{\mathrm{kmol}\cdot\mathrm{K}}\cdot\frac{\mathrm{Nm}}{\mathrm{kJ}}\cdot\frac{\mathrm{Pa\ m}^2}{\mathrm{N}} = \frac{8314.472\ \mathrm{Pa\ m}^3}{\mathrm{kmol}\cdot\mathrm{K}}$$

Other Consequences of the Ideal Gas Law

When gases or gas mixtures follow the ideal gas law, the gas internal energy property, *u*, and the gas enthalpy, *h*, are both functions of temperature only. Also, the specific heat capacity is a function of temperature only. In many psychrometric problems the additional assumption is made that the specific heat capacity is constant over the temperature range of the process.

… giants on whose shoulders we stand.
—Gerald Holton

8 Psychrometric Pioneers and Charts from the First 100 Years

PART I: PSYCHROMETRIC PIONEERS

At the beginning of the 21st century, the four principles or laws on which the science of psychrometrics is based were firmly established, as were the equations that interrelate the properties. This was not always the case. Early scientists and practitioners were confronted with the same two invisible gases: *dry air* and *water vapour*. The fact that these gases are invisible substances was itself a challenge, but early scientists were also confronted with three scientific beliefs of the eighteenth century—all incorrect—about air, water, and heat: (1) many scientists believed that water dissolved in air in much the same way that some solids dissolve in a liquid, (2) others held that *water vapour* was a chemical formed from liquid water and heat (fire), and (3) the majority of scientists held that heat was a substance itself (the now disproved *caloric theory*). It was not until the 1750s that scientists became aware that elements had a gaseous form.

Early pioneers had to invent the barometer and thermometer to perform crude experiments. They had no means of producing temperatures below about –20°C. The early pioneers had to discover the fundamentals and values of latent heat and the specific heat capacity values for air, ice, water, and *water vapour*. They did not have accurate steam (*water vapour*) equilibrium property tables for temperatures above freezing and had no property tables for *water vapour* in equilibrium with ice, the solid phase of water. They were confronted with punishment or death for pronouncing theories not endorsed by the church (e.g., Galileo was tried by the Inquisition in Rome in 1633 and placed under house arrest for his pronouncement that the earth revolved around the sun). Imagine Torricelli's predicament in discussing the barometer, which relied on the principle of a vacuum, when the church firmly held that a vacuum was not possible (*horror vacui*). The early pioneers used crude adding machines, slide rules, and log tables, and had to manually plot each data point. They did not have the hindsight of existing psychrometric charts to guide them in choosing the format for property plotting coordinates.

Prior to 1900, the early pioneers were located in Europe. After 1900, the pioneers were primarily from North America. The eastern United States' summertime climate with dew-point temperatures from 15°C to 27°C was an environment in need of cooling and dehumidification to improve processes, productivity in the workplace, comfort in assembly buildings such as theaters and meeting places, and, finally, comfort in the home. The need for air conditioning coupled with a population concentration and a rising standard of living undoubtedly contributed to the shift from European psychrometric pioneers to North American pioneers.

The lives of these pioneers included other accomplishments, disappointment, tragedy, love, jealousy, and politics. Many pioneers were accomplished in other fields such as medicine, government, and the arts.

This chapter presents a partial list of pioneers who made contributions (in some cases, erroneous or misleading) to the science of psychrometrics. This list was extracted from encyclopedias, texts on physics and meteorology, and perusal of the publications of ASRE and ASHVE, the predecessor societies of ASHRAE. The date of discovery, birth, and death dates are not consistent in the source material and therefore should be taken as *approximate*. Likewise, disputes remain about who first invented an instrument or developed a theory. Undoubtedly there are many contributors missing from this listing. The missing will be found in the fields of meteorology, agriculture, aeronautics, printing, textile manufacturing, and other technologies that rely on psychrometrics.

The list is by no means complete nor are the contributions of those listed equal in value, or in some cases, even worthwhile. Most certainly the list is missing many psychrometric pioneers from Europe and other areas of the world. Readers having additions and corrections are encouraged to send them to the author so that revisions to this book may include the most representative list of psychrometric pioneers.

Observation and Philosophical Period

Anaximenes (6th century B.C.) of Miletus, Greek philosopher. In 570 B.C. he suggested *aer* (meaning mist or vapour as well as air) as the primary substance. Thales (624 B.C.–548 B.C.), Anaximander (611 B.C.–547 B.C.), and Anaximenes were considered to be the first Greek philosophers. They, along with later philosophers, helped in the transition from mystical and mythological explanations of nature to the scientific explanations of natural phenomena. Anaximenes believed that *aer* was the basic building block of all matter. This proved to be incorrect. He did have a good understanding of *water vapour* condensation when he stated, "invisible air of the atmosphere ... by condensation it becomes visible, first as a mist or cloud, then as water, and finally as solid matter such as earth or stones ... further rarefied it turns to fire."

Empedocles (490–430 B.C.) of Acragas on the south coast of Sicily. Some references list Empedocles as the discoverer of air as a separate substance when he observed that when a bucket or similar vessel is put upside down in water, the water does not enter into the bucket. He wrote: "When a girl playing with a water-clock of shining brass, puts the orifice of the pipe upon her comely hand, and dips the water-clock into the yielding mass of silvery water, the stream does not then flow into the vessel, but the bulk of air inside, pressing upon the close-packed perforations, keeps it out till she uncovers the compressed stream; but the air escapes and an equal volume of water runs in" (an explanation of respiration). He died by leaping into the crater of Etna to prove that he was a god: "Great Empedocles, that ardent sole, leapt into Etna, and was roasted whole." *This practice is now banned by the Environmental Protection Agency!*

Aristotle (384–322 B.C.), Greek philosopher. Aristotle and Plato, a predecessor, are regarded as the two greatest intellectual figures produced by the Greeks. Aristotle produced 47 works and is credited with writing the earliest treatises on meteorology in his famous *Meteorologica* (334 B.C.). He taught that matter consists of four basic elements (*earth, water, air,* and *fire)* that interact with each other through four *essences* (*moist, dry, warm,* and *cool*). A fifth element, *ether,* was added to explain happenings in the universe beyond the air. Heat was regarded as one of the active qualities of a body. Hotness and dryness and their opposites, coldness and moisture, were the qualities that defined an element. Aristotle held that air and water were mutually interconvertible. He considered invisible air to be formed from water and fire. **Empedocles** is also credited with many of these same teachings. While Aristotle's conception of the basic elements was far short of the mark, his reasoning that the earth was spherical and not flat proved his genius. Aristotle held that a vacuum was not actually or logically possible, that light could not penetrate a vacuum, and that nothing could be seen through a vacuum. He believed that air had weight.

Philo (possibly 2nd Century B.C.) of Byzantium, Greek philosopher. He is credited with making the earliest-recorded thermal instruments. These instruments used the heat of the sun to expand air in a vessel. This class of instruments later received the name *thermoscope,* meaning a hotness indicator. These instruments lacked accuracy because they did not account for variations in atmospheric pressure.

Hero (circa 62 A.D.) of Alexandria. The Latin translation of his *Pneumatics* in 1575 described experiments with air including its expansion on heating.

Gaius Plinius Secundus (23–79 A.D.), Roman naturalist. Believed that steady winds fell from the stars or, alternatively, are created by the stars and Earth as they travel in opposite directions.

The Dark Ages

From **Aristotle**'s death in 322 B.C. until the 1400s, there was little progress in the sciences underlying meteorology and psychrometrics.

The Experimental Science Period (Invention of Instrumentation, Identification of Elements, and Elementary Physical Laws)

This period had to wait for the invention of instruments to measure the physical parameters and properties of a substance. During this period, the components of moist air were identified as nitrogen, oxygen, minor trace gases, and a variable quantity of *water vapour.*

Alberti, Leone Battista (1414–1472), Italian painter, poet, philosopher, musician, and architect. In about 1450, he described a hygroscopic hydrometer and also invented a simple anemometer. He wrote: "We have found that a sponge is moistened by the humidity of the air, and so we make a balance with which we measure the heaviness and the dryness of the air and the winds."

Cusanus, Nicholas (1401–1464), also known as Nicolas Cryfts or Nicolas of Cues or Cusa. He was born at Cues on the Moselle and became a Cardinal and Bishop of Brixen. In 1450 in the *Codex Cusanas, Folio 132*, he wrote "Idiota de Staticus Experiments," in which he described a balance hygrometer as follows: "If someone should band a good deal of dry wool, tied together, on one end of a large pair of scales, and should balance it with stones at the other end, in a place where the air is temperate, it would be found that the weight of the wool would increase when the air became more humid, and decrease when the air tended to dryness. Whence, by such difference, he might weigh the air, and make probable conjectures about a change in the weather."

da Vinci, Leonardo (1452–1519), Italian painter, sculptor, architect, and engineer. In about 1500, he is credited with inventing and providing drawings of the first hygrometer, which he called a hygroscope. This hygrometer utilized a ball of wool that was weighed after exposure to the environment. If the humidity was high, the ball would weigh more because of the water adsorbed from the atmosphere.

Santorre, Santorio (1561–1636), professor of medicine at Padua. Santorre is probably the inventor of the thermoscope. Giuseppe Biancani published a figure of Santorre's thermoscope, which included movable threads around the tube to serve as a pseudo scale in making comparative readings. Biancani is responsible for its name: *thermoscopium.* In the same writing and separately in 1614, Santorre described three different types of hygrometers: (1) the weighing of dried salt of tartar before and after overnight exposure to the air, (2) the greater or lesser warping of very thin boards such as those of pear wood, and (3) the contraction of lyre strings or hempen cords. His description of the string-hygrometer is as follows: "a cord is stretched out, or a lyre string if you prefer, provided that it is a thick one. The string is fastened to a wall at two locations. In the middle a lead ball is fixed, and a scale may be drawn nearby. When the air gets moister the cord contracts, but when the north winds dries it up, it loosens. Sometimes air from the south wets and shortens the cord so much that the ball rises to the letter A, highest point on the scale, but while the north winds blow they dry it until the ball lowers to B, the lowest point on the scale. In this way even if no wind is blowing, the degree of moisture or dryness that there may be in the air can be observed every day."

Galilei, **Galileo** (1564–1642), Italian scientist, mathematician, astronomer, and physicist. In 1593 he constructed an air thermoscope. This was perhaps the first attempt in thermometry during the Renaissance period; however, it proved inaccurate because accuracy was affected by changes in atmospheric pressure. It did, however, establish the principle of change in gas volume with change in temperature. In 1615 Galileo stated that air weighs nothing at all and that it exerts no pressure. While wrong in these beliefs, he was nevertheless quite sure that a vacuum was possible. Later, in 1638, he determined that air has weight.

Rey, Jean (d. 1654), French country doctor. Rey devised an open-top, liquid-filled bulb and stem thermometer in about 1630, but nothing ever came of his instrument.

Descartes, René (1596–1650), the greatest of French philosophers. He held that all substances are composed of tiny particles. He envisioned that "water was composed of long, smooth shaped atoms easily separable and that evaporation of water was like the raising of dust from the ground in a windstorm. While the dust in the air is visible the water vapour is not; nevertheless, the tiny particles of water vapour keep their long smooth shaped form and are different from the air particles." The scientists of his age began to accept *water vapour* as a distinct substance. Descartes also contributed to the formation of the mechanical theory of heat. He stated, "Heat is no quality of bodies but a violent motion of the corpuscles," and he established the connection between light and heat by showing that the rays of one could be transformed into rays of the other. He differed with **Galileo**, stating that a vacuum was impossible because all space must be filled with *something*.

Ferdinand II (1610–1670), Fifth Grand Duke of Tuscany (now a part of Italy) and founder of the Florentine Academy of Science. He sponsored artisans in the construction of sealed-stem, alcohol-filled Florentine thermometers in 1641. These were truly works of art and were based on the expansion of alcohol within the glass. The basic design of the liquid, sealed-in-a-glass thermometer (the predecessor of the hermetically sealed, liquid-containing thermometer so familiar to us today) was perfected under Ferdinand's patronage prior to 1654. He also commissioned Francesco Folli of Poppi and others to construct hygrometers along the designs suggested by **Santorre**. In 1657, he is credited with the invention of the condensation hygrometer, a device that attempted to measure the *water vapour* content in air by the amount of condensation collected over a stated period of time. It is said: "He noticed the condensed moisture on the outside of his drinking glass. It was observed that when something iced was put in a glass, the surrounding air seemed to change itself into water." Cardinal Giovanni Carlo de'Medici wanted to be sure that the water collected on the outside of the condensation hygrometer came from the air and not from the ice water mixture on the inside of the glass. He used vinegar and snow and then alcohol and snow on the inside, but the water on the outside tasted the same.

Torricelli, Evangélista (1608–1647), Italian physicist and mathematician. He invented the mercury barometer in 1643. Torricelli moved to Florence at the invitation of **Galileo** in October 1641; **Galileo** died just four months later and **Ferdinand II** made Torricelli his Philosopher and Mathematician. Torricelli said: "We live submerged at the bottom of an ocean of elementary air, which is known by incontestable experiments to have weight, and so much weight, that the heaviest part near the surface of the earth weighs about one four-hundredth as much as water." Torricelli's general concept was, of course, true but he missed badly in his estimate of the weight of air—the actual value at sea level is closer to one eight-hundredth. **Descartes** deserves mention in connection with the barometer because of his 1647 suggestion to attach a scale to the barometer "so that we may also know if changes of weather and of location make any difference to it."

Pascal, Blaise (1623–1662), French scientist and mathematician. He developed what is referred to as Pascal's principle: *Pressure exerted anywhere on a confined liquid is transmitted unchanged to every portion of the interior and to all the walls of the containing vessel; and is always exerted at right angles to the walls*. In 1647, Pascal used **Torricelli**'s barometer concept and investigated what was in the space above the mercury in a closed-tube barometer. He wrote: "After having demonstrated that none of the substances that can be perceived by the senses, or of which we have knowledge, fill this apparently empty space, it will be my opinion, until someone has demonstrated to me the existence of some matter that fills it, that it is truly empty, and destitute of all matter." In 1648, with the help of his brother-in-law, Florin Périer, using two mercury barometers, he demonstrated that air pressure decreases with increasing altitude. Both barometers read 28 in. Hg (94.82 kPa) in a garden in Clermont-Ferrand (330 m above sea level). Périer then carried the second barometer to the top of Puy de Dôme (1465 m) where the barometer read 24.7 in. Hg (83.53 kPa). After descending to the garden, both barometers read 28 in. Hg. Prior to 1654, Pascal conceived the idea of the aneroid barometer, which relies on the mechanical measurement of force exerted by the atmosphere as opposed to the hydrostatic equilibrium principle of the mercury barometer. Pascal's *Traité de l'équilibre des liquers et de la Pesanteur de la Masse de l'air* is one of the classics of physical science. In 1642, Pascal built the first cogged-wheel adding and subtracting machine, called a Pascaline, as a tool for his father, the local tax collector.

Boyle, Robert (1627–1691), English natural philosopher and scientist. In 1657, he read of Otto von Guericke's air pump and, together with Robert Hooke in 1659, he developed the *Machina Boyleona (pneumatical engine)* and began a series of experiments on the properties of air. In 1659, he developed *Boyle's law*: If the temperature *of **dry** gas* is constant then its *volume varies inversely with the pressure exerted on it.* This law is one of the components of the ideal gas equation, one of the four principles on which psychrometric calculations are based. In 1660, in *New Experiments Physico-Mechanical Touting the Spring of Air and its Effects,* he enunciated this law. He also investigated the expansive qualities of freezing water. In 1661, in *Skeptical Chemist,* he distinguished between mixtures and compounds and conceptualized *elements*. He is credited by some with defining the kinetic theory of matter. In 1662, Boyle observed that air pressure changed with elevation.

Mariotte, Edme (1620–1684), French physicist. In 1676, he independently discovered Boyle's law. In Europe, the pressure-volume relationship of a gas is often called *Mariotte's law* rather than *Boyle's law* because of the contribution of Mariotte in emphasizing the importance of constant temperature. Mariotte also performed some research on the freezing of water. Mariotte revealed his findings in his *Second Essais de Physique* titled *De la Nature de le' Air.*

Eschinardi, Francesco (1623–1699), a Jesuit. In 1681, he was perhaps the first to write about the underlying principle of the psychrometer: "He found a most singular thing with the thermometer. When this was immersed in a glass of water … in a closed room, it showed a greater cold than the air of the room; but when it was taken out, it then showed a greater cold still." He mistakenly attributed this to the possibility that the thin film of water covering the bulb of the thermometer might be denser than ordinary water and, therefore, might have more power to cool the thermometer than the water in the glass.

Halley, Edmund (1656–1742), English astronomer. In 1693, he suggested that mercury be used for thermometers because of its density and the fact that it does not "wet" the tube (some texts suggest that mercury was first used in thermometers circa 1670). He also suggested that temperatures be measured down from the boiling point. (In 1655, **Boyle** suggested that temperatures be measured up from the freezing point.) Halley proposed that trade winds arise from the westward movement of the subsolar area of rising warm air at the equator together with the inflow of cool air from the temperate regions to replace it. Halley is famous for observing a comet in 1682 and, using **Sir Isaac Newton**'s laws, predicting the comet's return in 1758. Although he did not live to see the comet's return, it continues to return as predicted and is known as *Halley's comet*. He encouraged **Newton** in his writings and published Newton's *Principia*.

Renaldini, Carlo (1615–1698). In 1694, he suggested that thermometers be calibrated at the fixed points of water at freezing and boiling.

Amontons, Guillaume (1663–1705), French physicist. In 1699, he was the first to study the expansion of gases with temperature change and showed that as gas contained in a closed container was heated, its pressure increased by a fixed amount for a given temperature rise regardless of the mass of gas involved. From these observations he is said to have proposed that there was an absolute zero temperature. Apparently this was ignored and, more than a century later, **William Thompson (Lord Kelvin)** received the credit for establishing absolute zero. At Amontons's time, air was the only gas readily available. He noted in 1699 "that when the bulb of his air thermometer was taken out of water the mercury index descended and afterward rose again, little by little, as the bulb dried." He was known for his work in thermometry. In 1687, he invented a hygrometer based on the volumetric change of a leather bag tied to one end of a glass apparatus filled with mercury. In 1695, he published *Remarques et Expériences Physiques sur la Construction d'une Nouvelle Clepsydre, sur les Barométres, Thermométres, et Hygrométres*. Amontons stated that readings of a mercury barometer should be corrected for temperature.

Rømer, Ole (1644–1710), Danish astronomer. He wrote on the need for uniformity in the bore of the glass tubes used for thermometers and included instructions for the construction of a sealed thermometer. He was the first to make accurate reproducible thermometers using two fixed points for the scaling of the thermometer (the ice point and the boiling point of water). He developed a thermometer scale with a span of 60° with the ice point at 7.5 and the boiling point at 60. He apparently chose 60 because of its use in astronomy. The 7.5 value allowed for temperatures lower than freezing without negative numbers.

Fahrenheit, Gabriel Daniel (1686–1736), German meteorological instrument maker who lived in England and Holland. History indicates that he was using mercury in thermometers by 1714 when he invented the temperature scale that bears his name. Prior to this, in 1709, he made thermometers filled with alcohol. He said that he changed to mercury upon reading **Amontons**'s statement that mercury barometer readings should be corrected for temperature. Fahrenheit apparently used three fixed points in the scaling of his thermometers: the temperature of an ice-salt (sal ammoniac) mixture as his zero point (0°F, –17.8°C), a mixture of ice and water (32°F, 0°C), and his perceived value of the temperature of the human body (blood temperature) (96°F, 35.6°C). Fahrenheit's thermometers were made for medical practitioners and meteorologists. He did not use the boiling point of water in scaling his thermometers. Fahrenheit used four scale divisions for every one of **Rømer**'s. After his death, others extended Fahrenheit's scaling uniformly to water's boiling point, which turned out to be 212°F (100°C) at normal atmospheric pressure. In 1724, he invented an improved hygrometer.

Newton, Sir Isaac (1642–1727). In 1717, Newton clearly stated that "moist air was lighter than *dry air* at the same temperature and pressure." His pronouncement proved to be correct; however, it was ignored for years by others because it defied their logic that adding any substance to air would naturally make it heavier. Newton's logic was that a volume of air at a given pressure could only contain a given quantity of molecules. Adding the lighter *water vapour* molecules displaced an equal quantity of air molecules and, thus, moist air is lighter than *dry air* at the same temperature and barometric pressure. Previously, in 1687, he had developed his three laws of motion.

Boerhaave, Herman (1668–1738), Dutch physician who exchanged letters with **Fahrenheit**. He described heat as a subtle fluid, later called *calor* or *caloric*, that could pass from one object to another.

Hadley, George (1685–1768), English lawyer and climatologist. Hadley improved on **Halley**'s trade wind theory by substituting deflection of the inflowing temperate air by the earth's rotation for the westward motion of the subsolar point. This concept was named *Hadley cell circulation*.

Réaumur, René (1683–1857), French physicist and naturalist. In 1730 he developed an alcohol fluid thermometer with each degree mark above 0° (the freezing temperature of water) equal to one-thousandth of the alcohol volume below the 0° mark. For this specific thermometer the boiling point of water was 80°. The 0° to 80° Réaumur thermometric scale was used for many years.

Bernoulli, Daniel (1700–1782), Swiss academic doctor and scientist. As early as 1738, in his book *Hydrodynamica,* he suggested *the rational kinetic theory of gases* by stating that the indefinite expansion of gas was due to its flying molecules.

Celsius, Anders (1701–1744), Swedish astronomer and inventor. In 1741, he developed the centigrade (now called Celsius in his honor) temperature scale with 100 units between the *ice point of 100°C* and the *steam point of 0°C*. The scale was inverted to its present sequence by the Swedish botanist Carl Linnaeus in 1745. At about the same time, Jean Pierre Christin (1683–1755) of Lyons independently developed a thermometer with 0° for the ice point of water and 100° for the boiling point.

Le Roy, Charles (1726–1779), professor of medicine at Montpellier. In 1751, he described the *degré de saturation*: "He sealed up a bottle of moist air and placed the bottle in a place where the temperature was falling. At some temperature, always the same as long as the bottle remained sealed, dew would appear in the bottle, and would disappear later when the temperature rose again." He stated: "There is at all times a certain degree of cold at which the air is ready to release part of the water that it holds in solution. I call this temperature *degré de saturation* of the air." Le Roy's degree of saturation is now called *dew-point temperature*. Le Roy believed that water dissolved in air.

Black, Joseph (1728–1799), Scottish chemist. Black is responsible for making a clear distinction between heat and temperature. He experimented with the heat consumed in melting ice and developed the doctrine of latent heat in 1757. In this doctrine, he stated that when ice melts it takes up a quantity of heat without undergoing a change in temperature. He further stated that the heat must combine with the particles of ice and thus become latent in its substance. *"Latent" refers to something that is present in essence but not in such a fashion as to be apparent or visible.* He taught the doctrine of latent heat in lectures from 1761 onward. Black noticed that equal masses of liquid and solid substances require different amounts of heat to raise them to the same temperature and, in 1760, founded the doctrine of specific heat capacity. Black also showed that heat was something distinct from temperature and stated that both could be measured. In an essay he discussed "The supposed effect of boiling on water in disposing it to freeze more readily." (Does this sound like the familiar physics question regarding the use of hot water in the freezer tray to speed the making of ice cubes?) In 1754 he heated magnesium carbonate and anticipated **Antoine Laurent de Lavoisier** and modern chemistry by indicating the existence of a gas distinct from common air, which he detected by the use of a balance. In 1764, Black measured the latent heat of steam but did not publish his research; consequently, **Jean Andre Deluc** and others were able to claim credit as the first researchers to determine the latent heat of steam. James Watt, of steam engine fame, was a student of Black.

Scheele, C.W. In 1772, he recognized the existence of nitrogen and stated that air was a mixture of two gases called *foul air* (nitrogen) and *fire air* (oxygen).

Deluc, Jean André (1727–1817), Swiss geologist, meteorologist, and physicist who relocated to England in 1773. In 1761, he discovered that water was more dense at about 4°C than at higher or lower temperatures and that heat was required in the melting of ice. He originated the theory that the quantity of *water vapour* that any space can contain is independent of the presence of any other gas in the space. **John Dalton** later advanced this theory, which is known as *Dalton's law of partial pressures*. Deluc invented a new hygrometer and advanced important rules for measuring height by means of a barometer: *"the logarithm of the pressure decreases proportionally with increase in height and with the inverse of the temperature of the air."* In 1772, he authored the paper *Recherches sur les modifications de l'atmosphere.* Deluc made important contributions in spite of the fact that he actively promoted *that calor or heat was a substance*.

Priestly, Joseph (1733–1804), English minister and scientist and a Fellow in the Royal Society. Priestly was trained as a theologian but is best known for his scientific work in electricity, his six-volume publication *On Different Kinds of Air*, his *History of Optics*, and his *History of the Corruption's of Christianity.* In the 1760s, he met Benjamin Franklin, who encouraged him to complete his first scientific work, *The History and Present State of Electricity*, in 1767. In that same year, while a pastor at Mill Hill chapel, near Leeds, he investigated *fixed air* (carbon dioxide), the gas that lay above the fermenting liquid in the brewery vats next to his home. In 1774, he produced oxygen and later met with **Antoine Laurent de Lavoisier** in Paris. He discovered nitrogen (co-discovered independently by Daniel Rutherford in 1770), ammonia, nitrogen dioxide, nitrous oxide, hydrogen chloride, and sulfur dioxide. In 1779, he wrote about the inter diffusion of mixed gases. He took up the cause of the American colonists and later the French Revolution, which resulted in the burning of his church, house, and laboratory; three years later, in 1794, he moved to America.

Cleghorn, William. In 1779, he proposed the (now discredited) *caloric theory* with five statements: (1) the caloric is a fluid whose particles repel each other—to account for thermal expansion, (2) the caloric is conserved—to account for the science of calorimetry, (3) caloric particles are attracted differently to different particles of matter—to account for different specific heat capacities of the various substances, (4) the caloric may be sensible or latent—to account for phase changes, (5) the caloric has weight—to account for the change of weight of oxidation.

Lavoisier, Antoine Laurent de (1743–1794), French chemist. In 1774 he confirmed the law of conservation of matter (mass): *Matter can be neither created nor destroyed.* In 1777, at the Academy of Royal Scientific in Paris, he wrote in support of the theory that heat (or calor) was a substance. In 1783, he determined the true nature of atmospheric air. He is credited along with **Priestly** and **Scheele** with the discovery of oxygen, which he called dephlogisiticated air (1774). He proposed names for oxygen (from *acid-maker,* Greek) and nitrogen (*azote*, because it does not support life). Lavoisier overturned the 100-year-old *phlogiston* theory of combustion. He is called the father of modern chemistry. He stated that the oxygen element contained caloric. He refuted the belief that water by repeated distillation is converted into earth. Unfortunately, Lavoisier was a public figure and early in the French Revolution was arrested and the next day guillotined. His friend Joseph Lagrange commented: "It required only a moment to sever that head and perhaps a century will not be sufficient to produce another like it." Lavoisier's widow married **Sir Benjamin Thompson (Count von Rumford)**; however, that marriage lasted only a few years. One of Lavoisier's assistants was a founder of the DuPont Company.

Saussure, Horace Bénédict de (1740–1799), Swiss physicist and professor at the University of Geneva. In 1783, he invented a hygrometer that used human hair. The hair length increased as the humidity increased. Unfortunately, his hair was supplied by hospitals and was not healthy hair. After several months, his hair hygrometer changed calibration. He later employed healthy hair, minimized and made uniform the amount of degreasing, and was careful to limit the stretching or tension of the hair and obtained better results. In the same year, he established through experimentation that *water vapour* exercises a pressure of its own with the maximum value solely dependent on temperature. While others described the amount of *water vapour* in the air in terms of *water vapour* pressure, he specified *water vapour* content as the weight of *water vapour* in unit volume of air, a term now called *water vapour density* or *absolute humidity.* His tables used units of grains per cubic foot of air. His *water vapour* pressure experiments were made in atmospheres of air, hydrogen, and carbon dioxide. He is credited with developing a wet-bulb thermometer in 1787 for the purpose of measuring the evaporative power of the air. In 1783 he published a book of four "Essais sur l'Hygrometrie," which covered basic and new hygrometer theories, the theory of evaporation, and applications in meteorology.

Hutton, James (1726–1797), Scottish philosopher and geologist. In about 1784 (possibly as late as 1792), he may have been the first person to measure humidity ("estimate the dryness of the air") by wetting a thermometer bulb and then comparing the reading with a previous dry-bulb reading.

Charles, Jacques Alexander César (1746–1823), French physicist. In 1787, Charles, through experiments, demonstrated that *the volume of a dry gas is directly proportional to its absolute temperature, provided that the pressure remains constant.* Charles never published his experiments or conclusions. In 1802 this same principle was independently arrived at by **Joseph Louis Gay-Lussac**, who published his experiments and conclusions. In Europe, this law is called either *Charles' law* or *Gay-*

Lussac's law. This law is one of the components of the ideal gas equation—one of the four principles on which psychrometric calculations are based. In 1783, Charles used hydrogen for the inflation of balloons.

Conte, Nicholas Jacques (1755–1805). In 1795, he investigated the aneroid (without fluid) method of measuring pressure.

Thompson, Benjamin (Count von Rumford) (1753–1814), British-American scientist and later a Tory exile from the U.S. He questioned the *caloric theory* based on observations while boring brass cylinders for the construction of cannon barrels in Bavaria in 1798. Prior to his observations, it was thought that the heat produced in boring a cannon was furnished by the latent heat release of the metal chips or, put another way, that heat was a fluid that had to be squeezed out of the metal. Writing on the demise of the caloric theory, he stated in his *Enquiry Concerning the Source of Heat which is Excited by Friction*: "It is unnecessary to add that anything which any insulated body or system can continue to furnish without limitation cannot possibly be a material substance ... the heat produced could not possibly have been furnished at the expense of the latent heat of the metallic chips ... it was simply produced by the friction."

Leslie, John (1766–1832), English compatriot of **Hutton**. In 1794–1795 (possibly 1797), while experimenting on the evaporation of ice, he developed a differential thermometer that he used as a hygrometer. This hygrometer was the first of that type to be accepted as a reliable instrument. It could be called an early non-aspirated wet-bulb thermometer. In his *A Short Account of Experiments and Instruments, Depending on the Relations of Air*, he wrote "that in equilibrium, the heat lost by unit mass of air in falling from the temperature of the dry-bulb to that of the wet-bulb must equal the latent heat of evaporation of the moisture taken up."

Arnim, Ludwig Achim von (1781–1831). In 1800, he used the terms *relative Menge Wasser* and *degree of saturation* synonymously.

Böckmann, Carl Wilhelm, Jr. (1773–1821), German. In 1802, he used two thermometers—one with a dry bulb and one with a wet bulb—to make measurements of humidity.

The Beginning of a Unifying Theory Regarding the Behavior of Air and Water Vapour

Dalton, John (1766–1844), English scientist and chemist. Dalton is known as the father of atomic theory. In 1800, he established the physical basis of modern meteorology by explaining the variations of *water vapour* in the atmosphere and the relation between the expansion of air and atmospheric condensation. Dalton postulated in the years 1803 to 1808 that each chemical element is composed of a different kind of atom; his ideas are the basis of modern atomic theory. In 1801, he developed the *law of* *partial pressures*—one of the four principles on which modern psychrometrics is based. In 1802, he identified *water vapour* as a gas. Dalton kept a meteorological diary from 1787 to 1844 (with over 200,000 observations) and made his own instruments. In 1803, he published his law of partial pressures in *Absorption Of Gases By Water And Other Liquids*. Independent of **Joseph Louis Gay-Lussac**, he discovered that all gases have nearly the same coefficient of thermal expansion. He studied the physical properties of the atmosphere and other gases. He developed the first table of atomic weights. Dalton's accomplishments were not without error. He asserted *that the whole atmosphere, from top to bottom, must be almost saturated in every place.* A year later he stated that the density of the aqueous atmosphere at any height is totally independent of the density of the compound mass of air and is to be ascertained by knowing the density of vapour at the earth's surface and its specific gravity. This part of Dalton's law was fatally flawed and, in the present era, Dalton's law or model is limited to *the total pressure in a space is the sum of the partial pressures of the gaseous substances contained in the space*. Dalton was the first to define relative humidity.

Gay-Lussac, Joseph Louis (1778–1850), French chemist and a pioneer in the study of the gaseous state. In 1802, in his early research into the properties of gases, vapour pressures, hygrometry, and capillarity, he determined the coefficient of expansion for air and also showed that oxygen, nitrogen, and hydrogen all had nearly the same coefficient of cubical expansion. Gay-Lussac published his work. In Europe, *Charles' law* is called *Gay-Lussac's law.* In 1804 with Jean Biot, he made the first balloon ascent to 4000 metres and later in 1804 to 7000 metres. On these ascents, he observed magnetism, temperature, and humidity of the air and collected air samples at different heights. He observed that the nitrogen and oxygen composition of air was the same at all heights in his 7000 metre ascent. He is often called the father of meteorology. In 1809, he wrote that when gases combine with one another, they do so in the simplest proportions by volume and that the volume of any gaseous product formed bears a simple ratio to that of its constituents. In 1815 or 1822, at a meeting of the Academy of Sciences in Paris, he stated his theory of the wet bulb, pointing out that the cooling is a function of pressure, temperature, and humidity, and proposed that very extensive tables relating these properties would be required (*Sur le Froid Produit par l'evaporation des Liquides*, Ann. Chem. Physique, Vol. 21, p. 82).

Young, Thomas (1773–1829), English physicist and physician. In 1805–1807 he was the first to use the word *energy* in its modern sense in his *Bakerian Lecture* to the Royal Society. The word *energy* comes from the Greek word *energeia*, meaning *in work*, *work within*, or *containing work*. Young defined energy as that property of a body that enables it to do work. (Gaspart Gustave de Coriolis in 1829 was the first to use the term *work* in the scientific sense.)Young was also known for determining the size of molecules of a number of elements based on a study of the surface tension and tensile strength of liquids. Young also had fame in deciphering the hieroglyphs of the 196 B.C. Egyptian Rosetta Stone. He was the first to prove that the elongated ovals contained a royal name written phonetically. This laid the foundation for the interpretation of the hieroglyphic stone writings of ancient Egypt.

Avogadro, Amedeo (1776–1856), Italian physicist and professor of physics at the University of Turin. In 1811, he recognized that at the same temperature and pressure equal volumes of different gases contain the same number of particles (molecules) and that the mass densities of different gases are proportional to their molecular weights. He also postulated that the number of molecules in one gram-molecular weight (mole) of a substance was the same as that in a mole of any other substance, one of the basic concepts of modern chemistry. Later researchers defined a mole as the amount of substance containing the Avogadro number of particles of that substance. The Avogadro number is taken as 6.0232 E23 molecules in a gram-molecular weight of substance. He also conceived the atomic-molecular theory. He deduced that molecules have very high speed in the order of 1000 feet per second (305 m/s) and collide often. He recognized the distinction between atoms and molecules. Stanislas Cannizzaro in 1858 was instrumental in gaining acceptance for Avogadro's theories when he built a logical system of chemistry based on it.

Wells, William Charles. In 1838, he wrote an essay on dew and the several appearances of it. In 1847, he proposed the first theory of dew emphasizing the importance of radiative cooling.

Dulong, Pierre Louis (1785–1838), French physicist, and **Petit, Alexis Thérés** (1791–1820), French chemist and physicist. In 1819, they developed the general rule that for most elements the specific heat capacity multiplied by the relative mass of its atoms yields a number that is approximately the same for all elements. This is known as the law of Dulong and Petit. In 1819, Dulong developed a list of the atomic weights of twelve elements. In 1829, he published the specific heat capacities of gases.

Daniell, J.F. In 1820, he presented the paper "On a New Hygrometer Which Measures the Force and Weight of Aqueous Vapour in the Atmosphere, and the Corresponding Degree of Evaporation" in *The Quarterly Journal of Science Literature and the Arts (London).*

Döbereiner, Johann Wolfgang (1780–1849), German chemist. In 1822, he suggested a device for measuring dew-point temperature. With his device, the surface on which dew condensed was cooled by bubbling air through liquid ether, which resulted in the evaporation of a portion of the ether. His work significantly improved on the concept of the dew-point instruments suggested by J. von Soldner (1777–1833) in 1809, the chemist Jöns Jacob Berzelius (1779–1848) in 1809, earlier, W.H. Wollaston (1766–1828), and **Daniell** in 1820.

Ivory, James (1765–1842). In 1822, he analyzed **Hutton**'s wet-bulb experiments and is credited with publishing the first theory of the wet bulb in *The Hygrometer by Evaporation* (Phil. Mag, Vol. 60, p. 81). His theory assumed a moderate flow of air across the wet bulb and assumed that there would be negligible conduction of heat down the stem of the wet-bulb thermometer. He assumed the attainment of a steady state in which the air coming into contact with the wet bulb was saturated. For this condition he stated that the latent heat of vaporization of the water added to the air was equal to the (sensible) heat given up by the air in falling from the temperature shown by the dry-bulb thermometer to the temperature shown by the wet-bulb thermometer. He developed the following empirical formula for *water vapour* pressure:

$$e = e' - b \cdot (t - t')/1200$$

where

e = actual *water vapour* pressure in inches of mercury

e' = saturation *water vapour* pressure at the wet-bulb temperature in inches of mercury

t = dry-bulb temperature in Celsius

t' = wet-bulb temperature in Celsius

b = barometric pressure in inches of mercury

The basic form of this formula remains intact today. The b factor has been replaced by an aP term in which P is the barometric pressure and a is a modifying factor. The U.S. Weather Bureau also applies a small correction that takes into account the change in the latent heat of the evaporation of water with temperature.

The mass-based equation $(C_{P\_DA} + C_{P\_WVS}) \cdot (t_{DB} - t_{WB\_SAT}) = h_{FG\_WV\_@WB} \cdot (w_{WB} - w_{\_ACT})$ can easily be developed from Ivory's volume-based equations and is still used in psychrometric calculations today.

Poisson, Siméon Denis (1781–1840), French mathematician. He studied under Pierre-Simon Laplace and Joseph Lagrange at the École Polytechnique and in 1806 was appointed full professor at that institution in succession to Jean Baptiste Joseph Fourier. In 1823, he deduced the correct expression relating changes in pressure and temperature for an ideal gas undergoing adiabatic processes ($T \cdot v^{(\gamma - 1)}$ = constant; $T \cdot p^{(\gamma - 1)/\gamma}$ = constant; $p \cdot v^{\gamma}$ = constant). These equations are used today in the field of meteorology and in air conditioning in determining fan temperature rise.

August, Ernest Ferdinand (1795–1870), German. In 1825, August independently arrived at the same results as **Ivory**. He had in his possession the theories proposed and the instruments used by **Dalton**, **Hutton**,

Leslie, and **Daniell** but did not have **Ivory**'s formula. He pointed out: "The mathematical results that I have set out above agree so well with those found by Mr. Gay-Lussac for the cold produced by evaporation in *dry air*, which he tested by experiment, and with Mr. Ivory's formula, about which I learned through the kindness of the Editor, that I think it worth the trouble to set them out in full, for as far as I know nothing of the sort has hitherto appeared in German." August and **Ivory** are credited with the earliest empirical formulae relating dry-bulb and wet-bulb temperature readings to vapour pressure and relative humidity. L.S. Marks, in his comments on the 1911 Carrier treatise, indicates that in 1825 August completely understood the theory of adiabatic saturation and deduced the best equation possible given the existing physical knowledge at the time. August named the wet-bulb and dry-bulb instrument the *psychrometer,* taken from the Greek words *psychro* (cold) and *meter* (to measure) and, in the wayward path of word usage and development, this inappropriate term (to measure cold) has further developed into *psychrometrics,* the study of the thermodynamic properties and processes involving moist air. August may have regarded the psychrometer as an instrument for measuring the cooling effect produced by the evaporation of moisture from the wet-bulb thermometer. Later, **William Ferrel** improved on the work of August. Even later, **Willis H. Carrier** made a giant leap with his *Rational Psychrometric Formulae*, in which he provided a thermodynamically sound basis in lieu of the previous empirical formulae.

Apjohn, James (1796–1886). In 1834, he suggested the theory of the adiabatic absorption (mixing) of water into air in a saturation chamber, the theory that explains evaporative cooling. He was not able to correlate his theory with observations due in part to inaccurate values for the specific heat capacities of air and the latent heat of vaporization of water in use at the time ("On the Theory of the Moist-Bulb Hygrometer," Royal Irish Academy Trans., 1834). Also in 1834, Apjohn proposed an equation for calculating the actual *water vapour* pressure using the dry-bulb temperature reading, the wet-bulb temperature reading, the saturated *water vapour* pressure at wet-bulb temperature, and the barometric pressure reading. This equation was used for more than fifty years. It is still sufficiently accurate when the temperature readings cannot be read with a precision closer than ±0.2°C. **William Ferrel** in 1886 and **Willis H. Carrier** in 1911 proposed equations with slightly more accurate correction terms. With the advent of psychrometric charts and tables in the early 1900s and the availability of psychrometric software in the 1990s, these equations are no longer widely used.

Espy, James Pollard (1785–1860), American meteorologist. Espy was a pioneer weather forecaster who put together the first annual weather reports. He contributed to the theory of thunderstorm formation in his

1841 "Philosophy of Storms" dissertation, which contained the following: "An upward draft of warm air will increase in force and height by virtue of the energy of latent heat released in cloud formation and rainfall." In 1847, he correctly surmised the importance of latent heat release in cloud development and paved the way for the theory of moist adiabatic processes, which was later brought to fruition by **William Thompson (Lord Kelvin)** in 1862 and Reye in 1864. The prevailing theory for cloud and rain in Espy's time was that of **Hutton**, who in 1784 proposed, after **Dalton**, that it was the mixing of air masses of different temperatures and humidities rather than the adiabatic cooling of a single air mass that lead to saturation.

Mayer, Julius Robert von (1814–1878), German physicist and physician. In 1842, he advanced the theory that heat is a form of energy and that a calorie will produce a definite quantity of work. This theory was based on his attempt to determine the mechanical equivalent of heat from the heat produced when warm air is compressed. He deduced his theory using the old principle *causa aequat effectum*. He was the first to state the law of conservation of energy.

Joule, James Prescott (1818–1889), English experimental physicist. Joule received some instruction from **Dalton**, but, for the most part, he was self-taught in science. In 1843, he established the quantitative relationship between heat and mechanical energy (the interconvertibility of the various forms of energy and the first law of thermodynamics). Joule's work routed the *caloric* scientific theory of the period in which heat was thought to be a mysterious caloric substance without mass. He studied the relationships between electrical, mechanical, and chemical effects. In 1853, with **William Thompson (Lord Kelvin)**, he investigated the work done in compressing gases and on the thermal changes they undergo when forced under pressure through small apertures.

Regnault, Henri Victor (1810–1878), French chemist and physicist. He invented one form of dew-point apparatus (or perfected **Döbereiner**'s hygrometer). This hygrometer and an improvement of it by Alluard were used at large observatories for decades. In 1845, Regnault published his paper on hygrometry. In 1847, he began a long series of accurate measurements of the properties of various gases for the French government. His work on the properties of *water vapour* built upon the earlier work of **Dalton** and others. During the 1850s, Regnault researched and published vapour pressure tables of water and steam for temperatures to 230°C, *Mémoires de L'Académie des Sciences de L'Institute de France, Tome XXI, 1847*. He determined that the specific heat capacity of air was a constant equal to 0.2375 Btu/(lb·°F) [0.994 J/(kg·°C)]. **W.F.G. Swann**, in 1908, pointed out a defect in Regnault's method for the determination of the specific heat capacity of air. Regnault's law states that the C_P of a gas is the same, whatever the pressure. Regnault's property tables, physical constants, and functions represented a leap of several plateaus in the accuracy of the properties of water, *water vapour*, air, and other substances. Regnault tested **Ivory**'s formula in 1835 or 1840 and also demonstrated that wind speed influenced the formula. He developed psychrometric formulas for strong wind, moderate wind, and calm wind, which were used by meteorologists of this era. Regnault also developed an empirical formula giving the enthalpy of steam in units of calories per gram: $h = 606.5 + 0.305 \cdot t$ with t in degrees Celsius.

Helmholtz, Herman von (1821–1894), German physicist and biologist. He is generally credited by the scientific community as the discoverer of the *law of the conservation of energy (force)*. In 1847, he stated clearly and emphatically what **Joule** and **Mayer** had implied from their research. In 1851, he invented the opthalmoscope. **Heinrich Hertz** was his pupil.

Thompson, William (Lord Kelvin) (1824–1907), English baron and physicist. Initially he worked in the lab of **Regnault**. In 1848 (or 1851), he developed the concept of absolute zero temperature (–273°C). The Kelvin name for the scale of absolute temperature in SI units honors this contribution. Thompson also established the theory of atmospheric tides. He also deserves major credit for bringing the work of **Joule** to the attention of the scientific community. In 1849, he coined the word *thermodynamics* in an account of Sadi Carnot's *theory of heat*. In 1851, he presented a paper on the dynamical theory of heat reconciling the work of Carnot, **Rumford**, Davy, **Mayer**, and **Joule** and placed this theory and the fundamental theory of the conservation of energy in a position to command universal acceptance. In the same paper, he briefly summarized the second law of thermodynamics in his principle on the dissipation of energy.

The following quotation from Lord Kelvin when he was president of the Royal Society in 1895 should be ample proof that neither fancy titles nor past accomplishments preclude erroneous statements: "Heavier than air flying machines are impossible."

Joule, **Helmholtz**, **Mayer**, and **Lord Kelvin** are often listed as the founders of the law of conservation of energy. **Newton** had previously divined the principle of conservation of energy with respect to mechanics.

Glaisher, James (1809–1903), engineer, astronomer, and meteorologist with the Royal Observatory in England. In 1847, he developed the first reliable tables (*Glaisher's Hygrometrical Tables*) relating *water vapour* pressure, atmospheric pressure, and *stationary* wet-bulb and dry-bulb temperatures (*Philosophical Transactions of the Royal Society*, 1851). Glaisher based his tables on Greenwich Magnetical and Meteorological Observations from 1843 to 1845. He calculated, arranged, and printed, at his own cost, an elaborate series of tables showing, at a glance, the relation of the temperature of evaporation to that of the dew point at every degree of air temperature from 10°F to 90°F (–12°C to 32°C); he also listed the *water vapour* pressure, the *water vapour* density expressed in both pounds and grains per cubic foot, the additional quantity of H_2O required for complete saturation, and the relative humidity.

Bravais, Auguste (1811–1863), French astronomer. In 1853, he developed the *psychrométre fronde* (sling psychrometer), which consisted of a dry-bulb and a wet-bulb thermometer attached to a string or chain so that they could be whirled in the air. He was also instrumental in cloud measurement.

Gibbs, Josiah Willard (1839–1903), mathematical physicist. Circa 1865, he described the phenomenon of thermodynamic equilibrium. Also, he formulated the Gibbs-Dalton principle, in which the specific heat capacity, enthalpy, and entropy of a mixture of ideal gases is equal to the sum of the values of the individual components of the mixture. He is credited with a writing style in which *there is not a word more than necessary* and *everything is reduced to its essence.*

Stewart, Balfour (1828–1887) LL.D., F.R.S., Scottish physicist, superintendent of the Kew Observatory and professor of natural philosophy, Owens College, Manchester. In the second edition (1866) of his text *An Elementary Treatise on Heat* he introduced the inverse of the term *mixing ratio* and thus, like Celsius, deserves most of the credit for the term "mixing ratio." In his words: "...for meteorological purposes it may be desirable to investigate what may be termed the hygrometric quality of air. This may be defined to be the ratio between the weight of one cubic foot of air and that of the aqueous vapour present in it. Since dry air and aqueous vapour are affected by temperature and pressure according to the same laws, it is evident that the hygrometric quality will not vary unless there be either evaporation, precipitation of mixture (mixing of air parcels) going on in the air." He received the Rumford Medal of the Royal Society in 1868 for his work in radiant heat.

Hertz, Heinrich (1857–1894), German physicist and professor at Bonn. The SI frequency unit—one cycle per second—is named after him. He published an article in *Meteor, Zeitschrift* (Nov.–Dec. 1884, pp. 421–431) on a graphic method to determine the adiabatic changes of state of moist air. He is also credited with the creation of the *emagram*, a graphic used in meteorology. The *emagram* and his adiabatic change of state diagram may be the same.

Ferrel, William (1817–1891), American meteorologist and professor and assistant on psychrometric tables for use by the U.S. Army Signal Service. He developed the empirical Ferrel formula in 1886 for computing *water vapour* pressure from dry-bulb temperature, *sling* psychrometer wet-bulb temperature, and atmospheric pressure. The Ferrel formula built on the earlier work of Stefan, Maxwell, and Sworykin, who developed a theoretical calculation in still air including the effects of radiation, conduction, and diffusion. Ferrel's tables were regarded as more reliable than the **Glaisher** tables and were used by the U.S. Weather Bureau (cited in the Annual Report of the Chief Signal Officer of the Army to the Secretary of War for the year 1886). In 1859–1860, he presented the first analytic treatment of cyclonic storms and atmospheric circulation using principles developed by Pierre-Simon Laplace for tides and those of **Poisson** for projectiles. He further developed the theories of **Espy**, which became known as the Ferrel and Espy theories. He anticipated **Warren Kendall Lewis** when he stated, *"the comparative success of the earlier formulae such as those of August and of Ivory was due to a completely fortuitous numerical equality between the coefficient of thermal conductivity of air and the coefficient of diffusivity of water vapour. This may be the greatest piece of luck (or the most serious misfortune, according to the point of view) that has ever befallen a scientific theory"* (emphasis added).

Juhlin, Julius, Swedish meteorologist. In the late 1800s, he conducted experiments to determine the vapour pressure over both ice and water below 32°F. He determined the law of atmospheric circulation.

Jouglet, A. In 1873, he wrote on the various methods of cooling air to achieve comfort cooling and mentioned cooling by ice, refrigeration machines, compression-cooling-reexpansion, and cooling by passing air though underground ducts.

Hazen, William B. (1830–1887), U.S. soldier. In 1880, he was appointed chief signal officer and served in that post until his death. Beginning in 1870, the Signal Corps acted as the U.S. Weather Service, transmitting weather data over military telegraph lines. (Note the similarity with the establishment of the Internet by the Department of Defense.) He employed **Cleveland Abbe** as a meteorologist. Hazen introduced Abbe's standard time meridians and improved upon weather reporting methods and forecasting procedures that provided advance warning of storms and cold waves.

Abbe, Cleveland (1838–1916), American meteorologist. In 1869, he was the first to prepare weather maps for the region around Cincinnati. He was also the first meteorologist to send out weather bulletins. He later became a scientific consultant to the U.S. Weather Bureau, a branch of the Signal Corp. He eventually headed the bureau when it separated from the army. In 1891, he authored "The Mechanics of the Earth's Atmosphere." In 1901, he suggested that forecasting should be approached as a physical problem by seeking the solution of a full set of simultaneous equations governing the atmosphere.

Assmann, Richard (1845–1918), German meteorologist. In 1892, he developed the aspirated psychrometer, which consisted of a centrifugal fan that drew air past the bulbs of the wet- and dry-bulb thermometers and also between two polished tubes surrounding the thermometer bulbs. This latter provision eliminated most of the radiation effect on the bulbs.

Rietschel, Herman, German professor at the Berlin Royal Institute of Technology. In 1894, he published his *Guide to Calculating and Design of Ventilating and Heating Installations,* which included a chapter on the cooling of closed rooms. Some historians claim that this chapter was the first to present the rudiments of cooling in scientific form.

ASHVE (American Society of Heating and Ventilating Engineers), founded in 1894.

Marvin, Charles F. (1858–1943), professor. In 1900 and 1910, he authored *Psychrometric Tables for Obtaining the Vapour Pressure, Relative Humidity, and Temperature of the Dew Point* (U.S. Department of Commerce Weather Bureau). He is cited in the Annual Report of the Chief Signal Officer (U.S. Army), 1891. Marvin also developed a sunshine recorder in 1919.

Teisserenc de Bort, Leon Philippe, French scientist. In 1902, he confirmed previous measurements that temperature decreases with altitude but showed that the temperature leveled off at an altitude between 8 and 12 km. In 1908, he labeled the bottom layer of the atmosphere the *troposphere,* which means the region where air turns over and refers to the changes as air masses move under or over and displace other air masses.

ASRE (American Society of Refrigeration Engineers), founded in 1904.

Swann, W.F.G., English scientist. In 1908, he pointed out a defect in **Regnault**'s method for the determination of specific heat capacity of air.

Transition from Empirical to Rational Theory Period

Carrier, Willis H. (1876–1950). In 1904, just two years after his graduation from Cornell, while an employee of Buffalo Forge Company, he developed a blueprint version of a psychrometric chart very similar to today's charts. A 1906 Buffalo Forge catalog included the Carrier hygrometric (now called psychrometric) chart. In 1911, he presented "Rational Psychrometric Formulae," paper number 1340, at the annual meeting of the American Society of Mechanical Engineers. Among other things, this 35-page treatise substituted a thermodynamically based formula for **Ferrel**'s formula. This formula linked the transformation of sensible heat into latent heat in the adiabatic saturation of air. Carrier was able to correlate his formula with actual measurements. His formula was more accurate than that of **Apjohn** because Carrier used recently determined, more accurate values for the latent heat of vaporization of water (C.H. Peabody, ASME 1909) and the specific heat capacity of air (**Swann**, Phil. Trans. Royal Soc., 1909). Carrier was the editor of the first Buffalo Forge *Fan Engineering Handbook* published in 1914.

Cramer, Stuart W., American engineer and manufacturer. He is credited with coining the term *air conditioning,* which first appeared in published form in his 1906 publication *Useful Information for Cotton Manufacturing.* **Carrier** at one time said that Cramer co-discovered (but did not publish) many of Carrier's psychrometric relations.

Grosvenor, William M. In 1908, in a paper on dryer design, he presented a humidity (psychrometric) chart to the American Institute of Chemical Engineers was the first to use humidity-ratio as a plotting coordinate. This simplified calculations. A majority of later psychrometric charts followed the lead of Grosvenor. He also included auxiliary lines of (a) constant cooling, (b) specific heat, specific volume and density of humid (saturated) air, (c) specific volume and density of dry air, and (e) latent heat of vaporization of steam.

Thornthwaite, Charles W. (1889–1963), American climatologist. He classified climatic types by the moisture they contain and developed the concept of global water balance.

Lyle, J. Irving, one of **Carrier**'s partners. Wrote the article "How to Use the Psychrometric Chart" for the January 1918 issue of *Heating and Ventilating* magazine.

Köppen, Wladimar. In 1918, he introduced his climate classification scheme.

Mollier, **Richard** (1863–1935), German mechanical engineer and scientist of Dresden. In 1923, he introduced a psychrometric chart using enthalpy and humidity ratio as the coordinates (*Ein neues Diagram fur Dampfluftgemische*, ZVDI, Vol. 67 September 8, 1923). **James L. Threlkeld** states that *"the use of enthalpy and humidity ratio as basic coordinates presents many advantages. Thermodynamic wet-bulb temperature lines are identically straight. A majority of common psychrometric processes appear as straight lines on h-W coordinates. In general, the Mollier type of chart allows the most fundamentally consistent treatment of air-conditioning problems with a minimum of approximations"* (emphasis added). The Mollier psychrometric chart was not only applicable to mixtures of *water vapour* and air but also included a fog region that is applicable to some psychrometric mixing processes.

Hill, E. Vernon. He published a psychrometric chart in the October 1920 issue of *Heating and Ventilating* magazine. Essentially this chart was a plot of dew-point temperature versus dry-bulb temperature. The chart was triangular in form and used auxiliary scales alongside the primary coordinates to display secondary variables.

Lewis, Warren Kendall. (1882–~1974), professor and head of the Department of Chemical Engineering, Massachusetts Institute of Technology. With regard to psychrometrics, his name is preserved in the dimensionless *Lewis number* (or *ratio*), which expresses the ratio of convective heat transfer to the mass diffusivity of a vaporizable substance and a noncondensable gas and is widely used in processes of combined heat and mass transfer and in wet-bulb psychrometry. In 1922, he surmised for substances and conditions having a Lewis number of 1.0 (which is approximately true for air and *water vapour*) that the psychrometric wet-bulb temperature and the temperature of adiabatic saturation would be identical. In 1933, he showed that for other combinations of a noncondensable gas and a liquid-vapour substance the psychrometric wet-bulb temperature and the temperature of adiabatic saturation are different. It is fortunate that for water-*water vapour* and air the Lewis number using a sling psychrometer is approximately 1.0 and, consequently, the psychrometer wet-bulb reading and the temperature of adiabatic saturation (thermodynamic wet-bulb temperature) are essentially the same. The psychrometer wet-bulb temperature and the temperature of adiabatic saturation are unequal for mixtures such as air and benzene or air and tolunol. Serge J. Zaroodny explains this by stating that "*the evaporation (in an adiabatic saturator) is essentially a molecular diffusion process and that the molecules of water (which of all the liquids has about the lowest molecular weight) are more agile and apt to diffuse into the air than are the heavier molecules of organic liquids"* (emphasis added).

Bulkeley, Claud A. (1875–1939), American engineer of Wilmington, Delaware, associated with Niagara Blower. He presented the Bulkeley Psychrometric Charts to ASHVE in 1926. This chart replaced the **Carrier** psychrometric chart in ASHVE publications from 1926 to 1938. Bukleley's chart used a logarithmic scale of *water vapour* pressure and a nonuniform dry-bulb temperature scale of decreasing modulus in order to present the water-*water vapour* saturation parameter as a straight line. Auxiliary scales were used for enthalpy, specific volume, and humidity ratio. These charts were not suitable for graphically presenting air-conditioning problems and therefore fell into disuse and were replaced by charts similar to the original **Carrier** chart or the chart developed by **F.O. Urban**.

Faust, Frank H. (1905–1995), engineer. During his long career with the air-conditioning division of General Electric, he is credited with designing a direct-reading psychrometric chart, circa 1930.

Mackey, Charles Osborn (1902–1965), American engineer and professor of mechanical engineering at Cornell University. In 1931, he provided a review of psychrometric charts in *Heating and Ventilating* (June and July 1931). He also co-authored with **Carrier** an ASME paper in 1939 titled "A Review of Existing Psychrometric Data in Relation to Practical Problems."

Arnold, J. Howard. In 1933, he wrote extensively on the theory of the psychrometer in *Physics* (Volume IV, July 1933, pp. 225–262 and September 1933, pp. 334–340).

Bergeron, Tor, Swedish meteorologist. In 1933 in *Physics of Clouds and Precipitation*, he described the Bergeron process, which explains rain formation in the middle latitudes. In these latitudes cold clouds consist of *water vapour*, ice crystals, and supercooled water droplets at temperatures as low as –20°C.

ASHVE Guide. In 1922, the first annual edition of the *Guide* was published. It contained a chapter titled "Properties of Air" reprinted from the *Buffalo Fan Engineering Handbook* and a general data section. The "Properties of Air" chapter was not included in the 1923–1929 *Guides*. The 1930 edition included a chapter on "Air Conditioning Fundamentals," which included basic psychrometric equations and the **Carrier** equation relating humidity ratio and psychrometer wet-bulb temperature. In 1933, these equations were found in the "Thermodynamics of Air Conditioning" chapter. In about this same period, ASHVE and ASRE publications included a **Mollier** diagram for moist air that was printed on the same page as the **Bulkeley** psychrometric chart.

Keppler, Ferdinand, engineer, Puerto Mexico (now Caotzacoalcos), Veracruz, Mexico. He introduced North American engineers to the 1923 Mollier psychrometric chart in a March and April 1934 *Refrigerating Engineering* article. Keppler cleverly transposed the Mollier chart coordinates to show a chart of roughly the same proportions as the Carrier chart, which was more familiar to the readers.

Urban, F.O., an employee of General Electric Company, an early manufacturer of air conditioners. In 1934, he produced a psychrometric chart with wet-bulb temperature lines in the body of the chart and showed an enthalpy at saturated wet-bulb scale using extensions of the wet-bulb lines from the saturation curve. Determining statepoint enthalpy involved reading the saturated wet-bulb enthalpy and correcting it by deducting the difference in enthalpy of liquid water at saturation and at the statepoint.

Keenan, Joseph H., and **Keyes, Fred G.,** professors at the Massachusetts Institute of Technology. In 1936, they published the *Thermodynamic Properties of Steam.*

The Mature Science Period

Goodman, William (1903–1993), consulting engineer, Chicago, Illinois. He was associated with the Trane Company in the 1930s and 1940s and was the principal author of the *Trane Air Conditioning Manual* in 1937. His other books were *Cooling and Air Conditioning for Comfort* (1932) and *Air Conditioning Analysis with Psychrometric Charts and Tables* (The Macmillan Company, 1943). He also wrote a series of articles dealing with psychrometrics for *Heating, Piping and Air Conditioning* magazine from 1936 to 1940, which provided useful tables of psychrometric data for the practicing engineer.

Torok, Elmer, mechanical engineer and superintendent of power, North American Rayon Company (NARC), Elizabethton, Tennessee. Torok authored what may be the first psychrometric handbook: *Psychrometric Notes and Tables—A Handbook on Psychrometric Principles, Tables and Calculations for Textile Manufacturers, Engineers and Students* (NARC, New York, 1935). A revised edition was published in 1941. Both editions contained extensive tables that later appeared (without reference to Torok) in *Handbook of Air Conditioning, Heating and Ventilating* by Industrial Press. Torok also authored *Air Conditioning in the Home* (Industrial Press, 1937). Torok was a member of ASHVE but no records from that era exist. NARC is no longer in existence and the successor company has no information about Torok.

Goff, John A., and **Gratch, Serge,** professors and researchers at the University of Pennsylvania Thermodynamics Research Laboratory. In 1943, 1945, and 1949, under a research grant from ASHRAE, they published papers on accurate psychrometric formulae and thermodynamic properties of *dry air*, *water vapour*, water, and ice.

Wile, Dan D. (1902–1984), American engineer. In 1944, he authored an article on psychrometrics in the frost zone in the magazine *Refrigerating Engineering*. In 1946, Wile and **Everett P. Palmatier** produced a psychrometric chart similar to the **Urban** chart that showed enthalpy at saturation. They also cleverly added enthalpy deviation curves in the body of the chart that enabled the user to quickly determine the true enthalpy at any point. The Palmatier-Wile psychrometric chart was used in the ASRE data book from 1947 until ASRE merged with ASHRAE. Wile is the author of many articles on psychrometrics, including a *must read* article from the August 1959 *ASHRAE Journal* titled "Psychrometric Charts, Past and Present."

Palmatier, Everett P. (b. 1911). In 1946, he co-authored an article in *Refrigerating Engineering* with **Wile** entitled "A New Psychrometric Chart." This chart included curves of enthalpy deviation from saturated wet-bulb enthalpy values. He was also a member of ASHRAE TC 1.2 and the principal researcher responsible for the 1963 series of ASHRAE psychrometric charts, as well as the author of *ASHRAE Transactions* paper 1802, which described the features and construction of the new normal temperature ASHRAE psychrometric chart.

Ashley, Carlyle M. (1899–1993), engineer with Carrier Corporation. In 1949, he authored a paper on psychrometric factors in the air-conditioning estimate. He was a member of ASHRAE TC 1.2, which was responsible for the 1963 series of ASHRAE psychrometric charts.

Carpenter, James H., professional engineer. In 1948, he authored the Carrier Corporation publication *Fundamentals of Psychrometrics*. This publication, including updates, is still available from Carrier's literature sales department in both I-P and SI versions.

Fellows, Julian, professor emeritus at the University of Illinois. He authored a series of practical articles on the properties of moist air and psychrometrics in the magazine *American Artisan*.

Barenbrug, A.W.T., mining engineer. In 1947 he was awarded the gold medal of the Chemical, Metallurgical and Mining Society of South Africa for his paper "Psychrometry and Psychrometric Charts." This paper was of considerable interest to engineers concerned with the ventilation and cooling of deep-level mines in the Transvaal, the Orange Free State, and elsewhere throughout the world. Expanded versions of this paper, complete with examples, were published in book form in 1955 (1st edition—IP) and 1974 (3rd edition—SI). A 2nd edition provided oversized psychrometric charts but no text. The 3rd edition included 31 charts, including 21 psychrometric charts covering barometric pressures from 80 kPa to 130 kPa in 2.5 kPa increments.

Everetts, John Jr. In 1959, he led ASHRAE TC 1.2 in a three-year research and development effort to produce the 1963 series of ASHRAE psychrometric charts.

Nevins, Ralph G. (1925–1974), American scientist and educator. In 1961, he authored a paper on psychrometrics and modern comfort. He is well known for his early research in the field of indoor air quality. An ASHRAE annual award honors his name.

Haines, Roger W. (b. 1916), consulting engineer. In 1961, he authored the article "How to Construct High Altitude Psychrometric Charts," published in *Heating, Piping and Air Conditioning* magazine. He was a member of ASHRAE TC 1.2, which was responsible for the 1963 series of ASHRAE psychrometric charts.

Threlkeld, James L., professor of mechanical engineering, University of Minnesota. He was a member of ASHRAE TC 1.2, which was responsible for the 1963 series of ASHRAE psychrometric charts. In 1962, he authored the book *Thermal Environmental Engineering,* which included an extensive section on the thermodynamics of moist air and a section on the theory and accuracy of the wet-bulb thermometer.

Chaddock, Jack B. (b. 1924), chairman of the Department of Mechanical Engineering, Duke University. He was a member of ASHRAE TC 1.2, which was responsible for the 1963 series of ASHRAE psychrometric charts. In 1965, he presented the paper "Moist air properties from tabulated virial coefficients."

Harrison, Louis P., researcher at the U.S. Weather Bureau and later at NBS. In 1965 he comprehensively covered ideal and real moist air psychrometric formulae, including those of Goff and Gratch in pages 3–256 of Volume 3 of *Humidity and Moisture*.

Ramsey, Melvin A., consulting engineer with Worthington Air Conditioning Company. In 1966, he authored the book *Tested Solutions to Design Problems in Air Conditioning and Refrigeration*, which included psychrometric principles and their application.

Kusada, Tamami, researcher at National Bureau of Standards (now NIST). He was a member of ASHRAE TC 1.2, which was responsible for the 1963 series of psychrometric charts. In 1970, he authored *Algorithms for Psychrometric Calculations Including Skeleton Tables for the Thermodynamic Properties of Moist Air* (NBS BSS 21).

Stoecker, Wilbert F., professor of mechanical engineering at the University of Illinois. In about 1970, he published the text *Refrigeration and Air Conditioning* (2nd edition 1982), which includes an excellent chapter on psychrometrics and air-water processes.

Sun, Tseng-Yao (b. 1931), consulting engineer with Ayres and Hayakawa. In an October 1971 article in *Heating, Piping and Air Conditioning,* he published a FORTRAN psychrometric subroutine using ASHRAE algorithms. These algorithms were used in the APEC (Automated Procedures for Engineering Consultants) software programs involving psychrometrics. These programs are reprinted in Sun's book *Air Handling System Design* (McGraw Hill, 1994).

Bullock, Charles E. (b. 1937), engineer with Carrier Corporation. In 1977, he authored a major section, "Thermodynamic Properties of Dry and Moist Air and Water," of the *ASHRAE Brochure on Psychrometry.* He was the ASHRAE liaison for the 1978–1983 **Richard W. Hyland** and **Arnold Wexler** research and contributed to the FORTRAN code for the new psychrometric charts.

Rudoy, William (1923–1984), professor of mechanical engineering, University of Pittsburgh. In 1979, he authored the *ASHRAE Cooling and Heating Load Calculation Manual*, which included a chapter and an appendix on psychrometric processes.

Hyland, Richard W., and **Wexler, Arnold,** physicists and researchers at the National Institute of Technology (formerly the National Bureau of Standards). In 1983, they completed two ASHRAE research projects, "Formulations for the Thermodynamic Properties of Moist Gases" and "Formulations for the Thermodynamic Properties of Dry Air."

Stewart, R.B. (b. 1924), **Jacobsen, R.T.** (b. 1941), and **Becker, J.H.,** researchers at the Thermodynamic Center at the University of Idaho. They took the work of **Hyland** and **Wexler** and produced a FORTRAN psychrometric program that was used to construct the current ASHRAE psychrometric charts.

Sonntag, Dietrich, German meteorologist and author. In 1990 he updated psychrometric formulae and constants to the ITS-90 temperature scale. (1990). He also published an excellent 16-page treatise: Advancements in the field of hygrometry (Meteorol. Zeitschrift, N.F. 3, 51–66, April 1994).

Olivieri, Joseph; Singh, Trilochan; and **Lovodocky, Steve.** They completed ASHRAE research project RP-872, which was abandoned by two Ohio State University faculty members. The product of their research is the 1996 ASHRAE publication *Psychrometrics: Theory and Practice*.

Nelson, H.F., and **Sauer, H.J.** In 2001, they completed ASHRAE RP-1060 for psychrometric properties above 0°C, which built upon the research of **Goff** and **Gratch** and **Hyland** and **Wexler**. Their research updated virial coefficients and extended the modeling of moist air as a mixture of real gases to 320°C, humidity ratios to 1.0, and barometric pressures to 5000 kPa.

Herrmann, Sebastian; **Kretzschmar, Hans-Joachim**; and **Gatley, Donald.** During 2003–2009, they completed two major psychrometric research projects: (a) “Advanced Adiabatic Compressed Air Energy Storage” (AA-CAES) “Determination of Thermodynamic and Transport Properties of Humid Air for Power-Cycle Calculations” for Pysikalisch-Technische Budesanstalt and (b) ASHRAE RP-1485, “Thermodynamic Properties of Real Moist Air, Dry Air, Steam, Water, and Ice.” RP-1485 is a comprehensive revision of the previous 1983 ASHRAE Hyland-Wexler model, including new physical constants from IUPAC, new models for H_2O from IAPWS, and a new relative molar mass for dry air. This research broadened the range of the real moist air ASHRAE model from –143.15°C to 350°C, 0.01 kPa to 10 Mpa, and humidity ratio from 0 to 10 kg_w/kg_a and produced new moist air and saturated H_2O tables for the 2009 *ASHRAE Handbook—Fundamentals*. Subsequently, they developed LibHuAirProp, software libraries of easy-to-use add-in psychrometric and transport property functions and user manuals for use in Excel®, MATLAB®, and EES®.

The following individuals have contributed to the science of psychrometrics by developing psychrometric software (in some cases with extensive documentation). All programs should be thoroughly tested and compared with ASHRAE real moist air psychrometric data by the user over the range of conditions for which the program will be used. Many software programs are accurate between 0°C and 50°C but do not include the correct algorithms for temperatures below 0°C, above 50°C, or for pressures outside of the normal 70–101 kPa range.

- **Edward F. Sowell**, PsyChart (now distributed by Elite Software)
- **Richard Grout**, Action Psychrometrics (discontinued)
- **Frank Pizzimenti**, Akton Psychrometric Chart
- **Paul J. Milligan** and **Andy Schoen**, PMtherm

- **Rex Raiza**, AhuPsyc (coil curve and space iteration)
- **Jim Judge** (Linric Company), PsychPro, PsyCalc, and PsyFunc
- **Robert Hanna** (Hands Down Software), Psychrometric Analysis CD (ASHRAE)
- **Bruce Wernick**, TechniSolve Software, Randburg, South Africa, PsyChart and many other air-conditioning, refrigeration, coil and equipment design software programs.
- **Sebastian Herrmann et. al.**, LibHuAirProp, software libraries psychrometric and transport property add-in functions for Excel®, MATLAB®, Mathcad®, and EES®, LabView® and DYMOLA®.

Suggested Additional Reading

"Psychrometric Charts in Review" by D.D. Wile (*ASHRAE Transactions* paper 1770, 1961), *The World of Measurement* by H.A. Klein, *Invention of the Meteorological Instruments* by W.E. Knowles Middleton (Johns Hopkins Press, Baltimore, 1969), "Psychrometric Chart Celebrates 100th Anniversary," *ASHRAE Journal* (2004) and "Grosvenor Humidity Chart (100th Anniversary)", *ASHRAE Journal* (2008). The books on measurement and instruments are suggested for further reading.

Psychrometric charts are probably the most widely used graphical devices of the refrigerating and air conditioning industry. Their principal purpose is to show the relation between six major properties of moist air: dry-bulb temperature, wet-bulb temperature, humidity ratio, relative humidity, enthalpy, and specific volume; and to provide graphical solutions to processes where air changes from one condition to another.
—D.D. Wile, 1959

Instead of using the word psychrometric use some simple word that all of us could understand, because to most of us who are not accustomed to using the psychrometric chart continuously it seems like a very complicated proposition. As a matter of fact the chart is really quite simple, after you once get the primary idea involved.
—H.P. Gant, 1926

PART II: A SELECTION OF PSYCHROMETRIC CHARTS FROM THE FIRST 100 YEARS

Humankind has an insatiable passion for producing squiggly diagrams, graphs, and other aids to help the originator and possibly others understand and visualize a process that is, for the most part, beyond human comprehension. The thermodynamicist is particularly adept at producing graphs for steam and refrigeration processes. The vast quantity of graphs can bewilder the uninitiated unless one comprehends that the choice and scale of each plotting coordinate is important in demonstrating one or more important characteristics of the process.

The two quotations and the previous paragraph should provide some food for thought about the psychrometric chart and perhaps its history. People of the Americas and Western Europe generally use the ASHRAE-style “psych” chart that resembles Willis H. Carrier's original 1904 *Hygrometric Chart*. Some people in Eastern Europe and China use a *Mollier* psych chart. The *Mollier* chart may look different; however, if one prints an ASHRAE-style psych chart on translucent paper, views it from the reverse side, and rotates it clockwise 90°, the result is a *Mollier* chart.

The history of the psych chart probably involved jealousy, politics, luck, and fierce competition similar to the development of the overall science. ASHRAE first adopted the Carrier-type chart, then the Bulkeley-style chart from 1926 to 1938, then returned to its original chart until 1961 when it developed a slightly more accurate chart with enthalpy-humidity ratio plotting coordinates based on Mollier's concept but configured similarly to the original chart.

Were there charts prior to 1904? Yes. W. Edson’s 1865 patent number 48,620 for a hydometer (later called a hygrodeik) included a printed chart as part of the instrument. The instrument included dry-bulb and wet-bulb thermometers, and its chart displayed dew-point temperature, humidity ratio, water vapour density, and relative humidity. The chart was probably plotted from tables similar to Glaisher's psychrometric observations. These early charts do not qualify as true psychrometric charts, which include specific enthalpy scales and the rationally correct thermodynamic wet-bulb temperature.

What future does the new millennium hold for the "psych" chart? The age of the computer and graphic psychrometric software spells the demise of the paper or laminated psych chart in much the same way that the wide variety of K&E log and other plotting papers were displaced by the graphic plotting capabilities of spreadsheet software. We will still see the psych chart on our computer screens and in training materials because it provides a convenient method of visualizing an air-conditioning or meteorological process. The screen plot will sometimes be printed so that a hard copy is available in the project file.

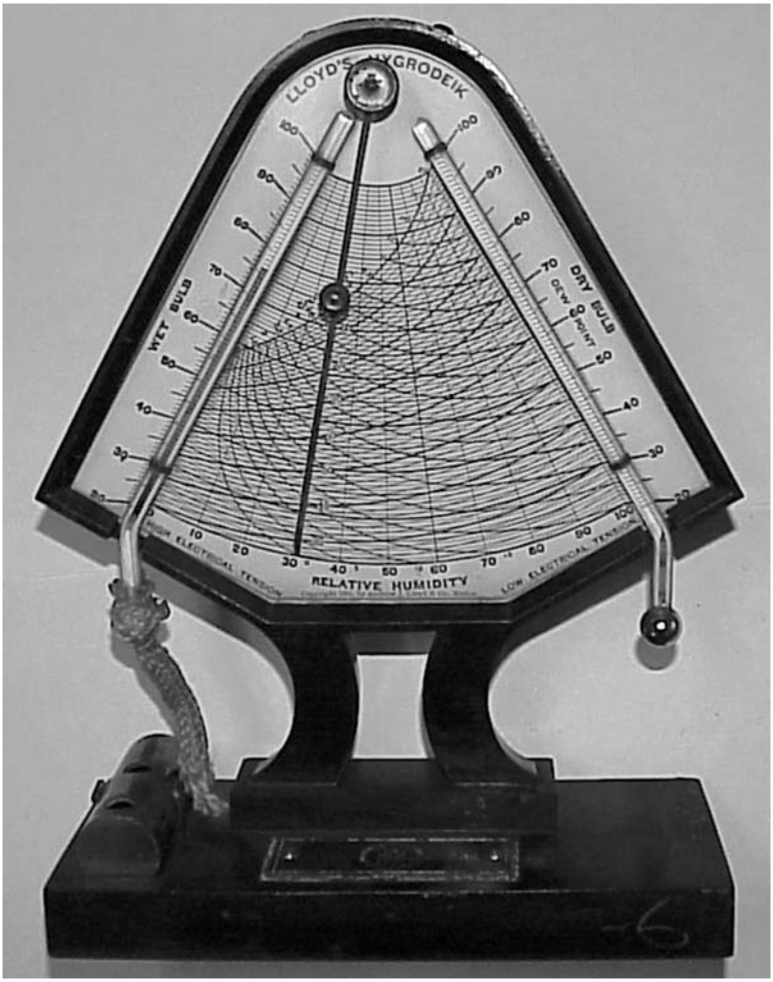


Photo courtesy of David A. Burba, Ph.D., Department of Physics and Astronomy, Vanderbilt University, Nashville, Tennessee.

Figure 8-1—Hygrodeik Instrument (circa 1865).

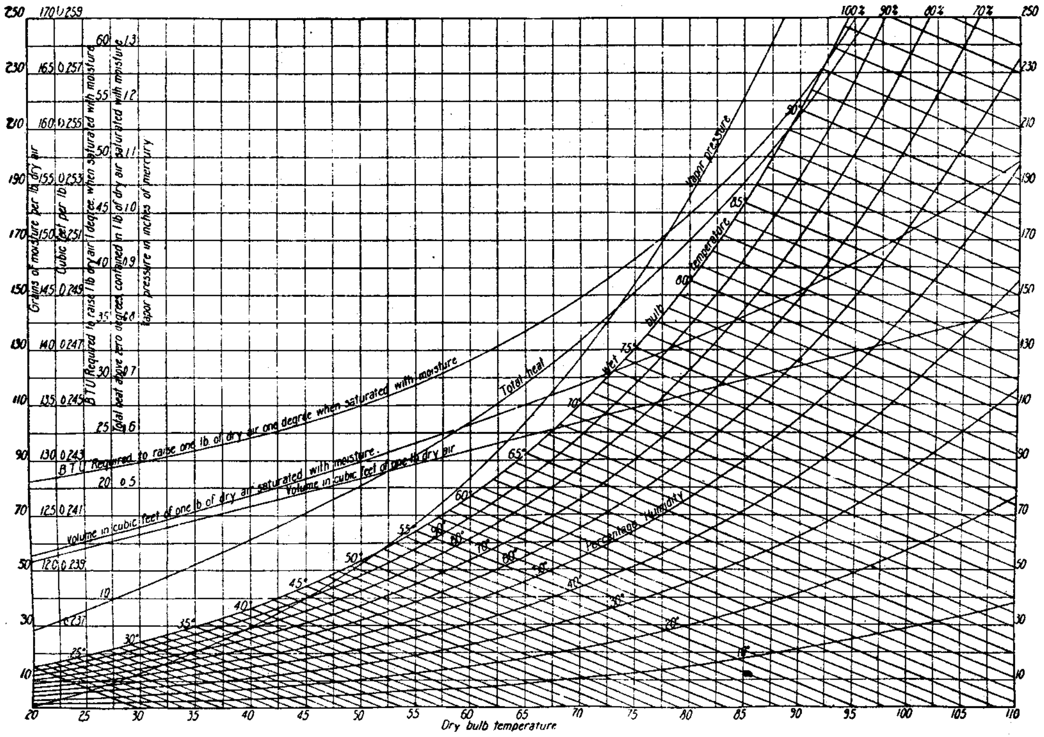
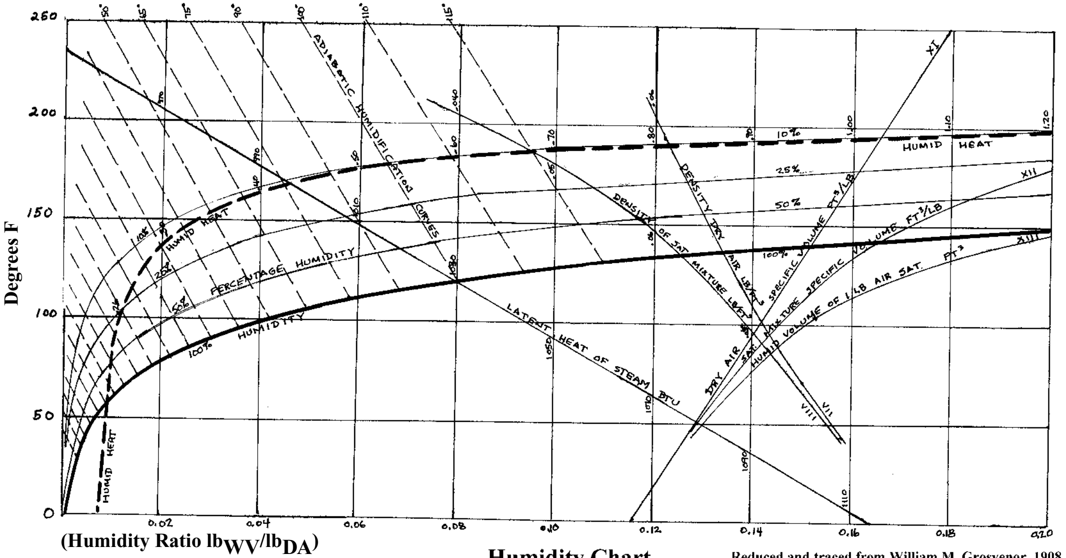


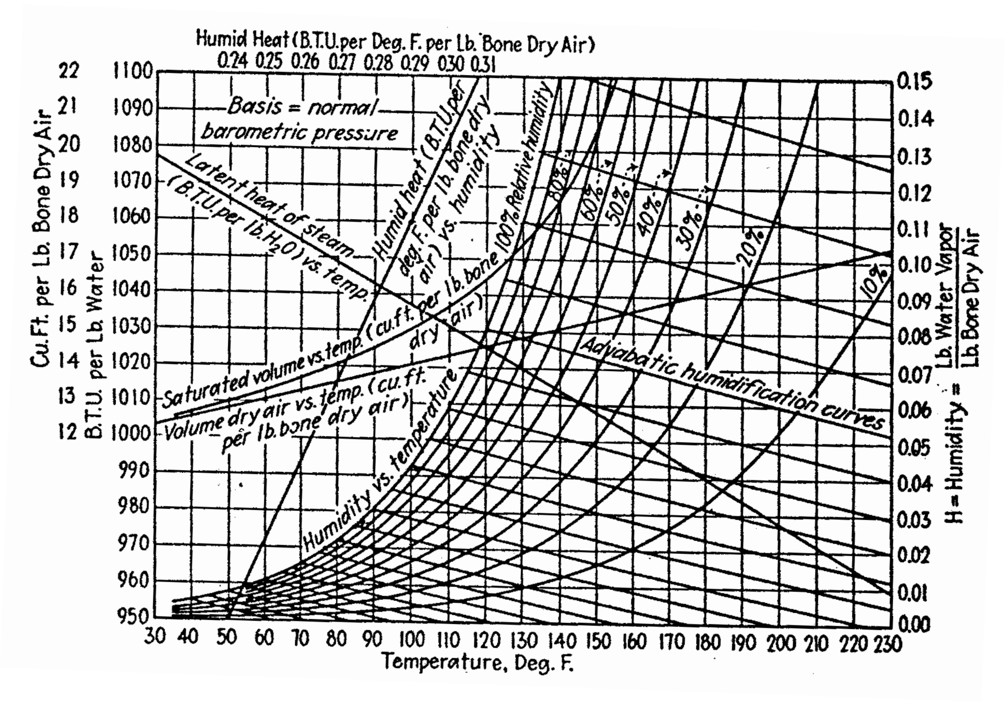
Figure 8-2—Willis H. Carrier historical chart, 1904 and 1911.



Humidity Chart

Reduced and traced from William M. Grosvenor, 1908

Figure 8-3—Grosvenor historical chart, 1908.



—Humidity Chart, ordinary range.

Figure 8-4—Modified Grosvenor chart—original chart 1908, modification date unknown.

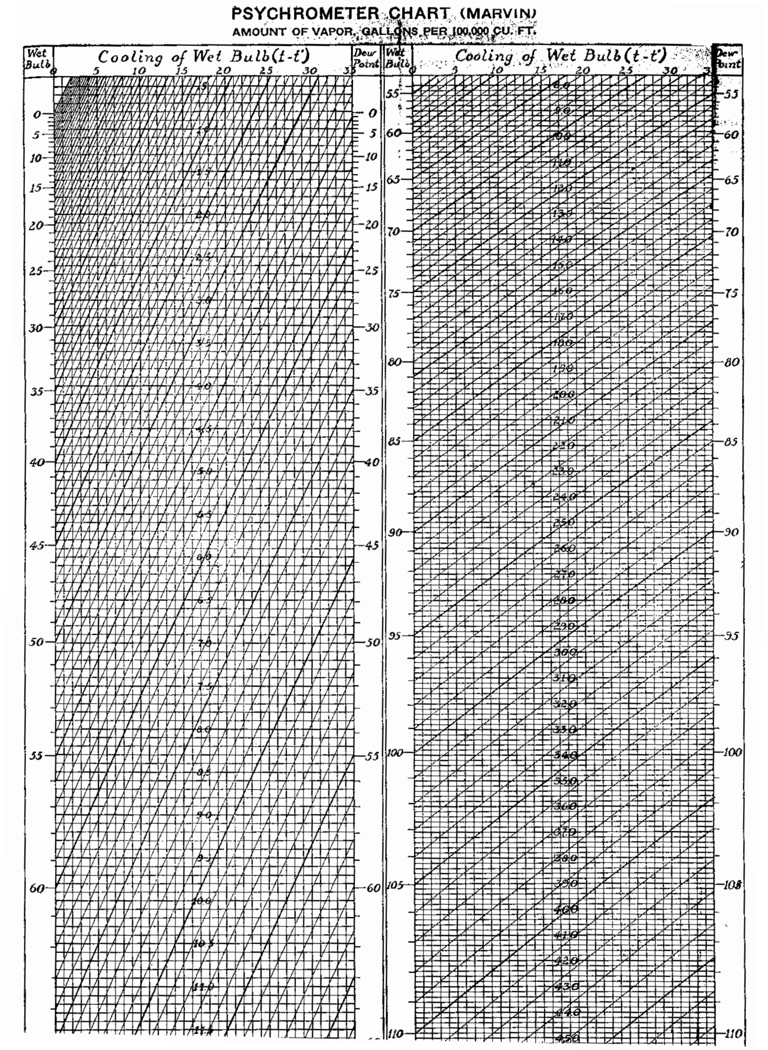
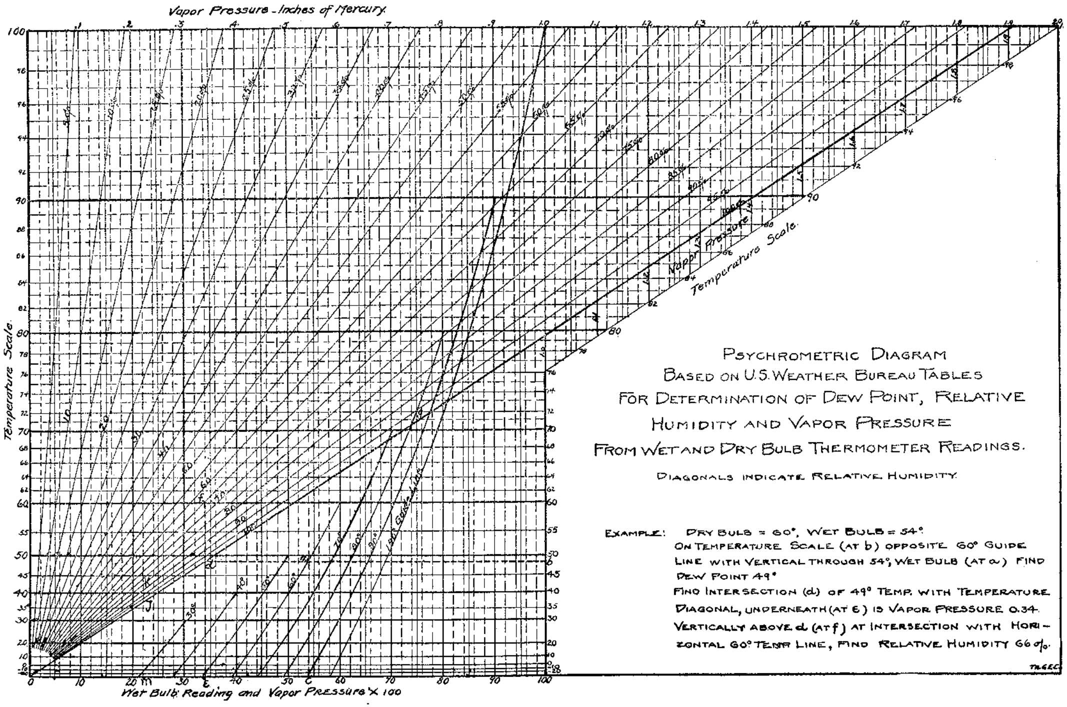


Figure 8-5—Marvin's historical "psychrometer" chart, circa 1920.



PSYCHROMETRIC DIAGRAM
BASED ON U.S. WEATHER BUREAU TABLES
FOR DETERMINATION OF DEW POINT, RELATIVE
HUMIDITY AND VAPOR PRESSURE
FROM WET AND DRY BULB THERMOMETER READINGS.

DIAGONALS INDICATE RELATIVE HUMIDITY.

EXAMPLE: DRY BULB = 60°, WET BULB = 54°
ON TEMPERATURE SCALE (AT b) OPPOSITE 60° GUIDE LINE WITH VERTICAL THROUGH 54°, WET BULB (AT a) FIND DEW POINT 49°
FIND INTERSECTION (d) OF 49° TEMP. WITH TEMPERATURE DIAGONAL, UNDERNEATH (AT e) IS VAPOR PRESSURE 0.34.
VERTICALLY ABOVE d (AT f) AT INTERSECTION WITH HORIZONTAL 60° TEMP LINE, FIND RELATIVE HUMIDITY 66%.

Figure 8-6—R.E. Horton historical chart, 1921.

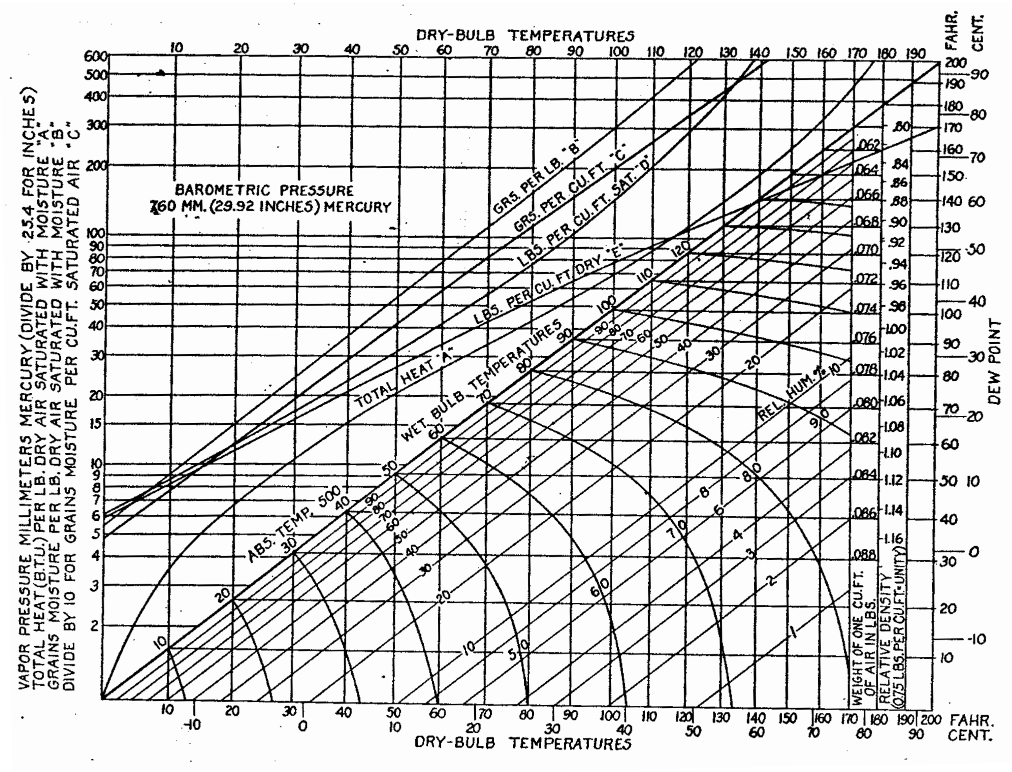


Figure 8-7—Claud A. Bulkeley historical chart, 1926.

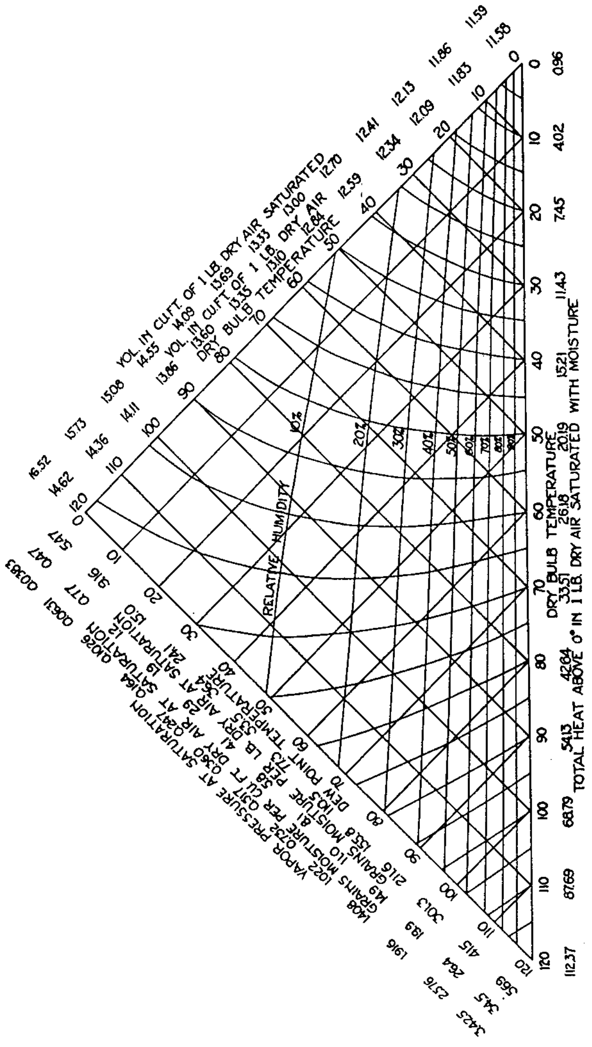


Figure 8-8—E.V. Hill historical chart, 1931.

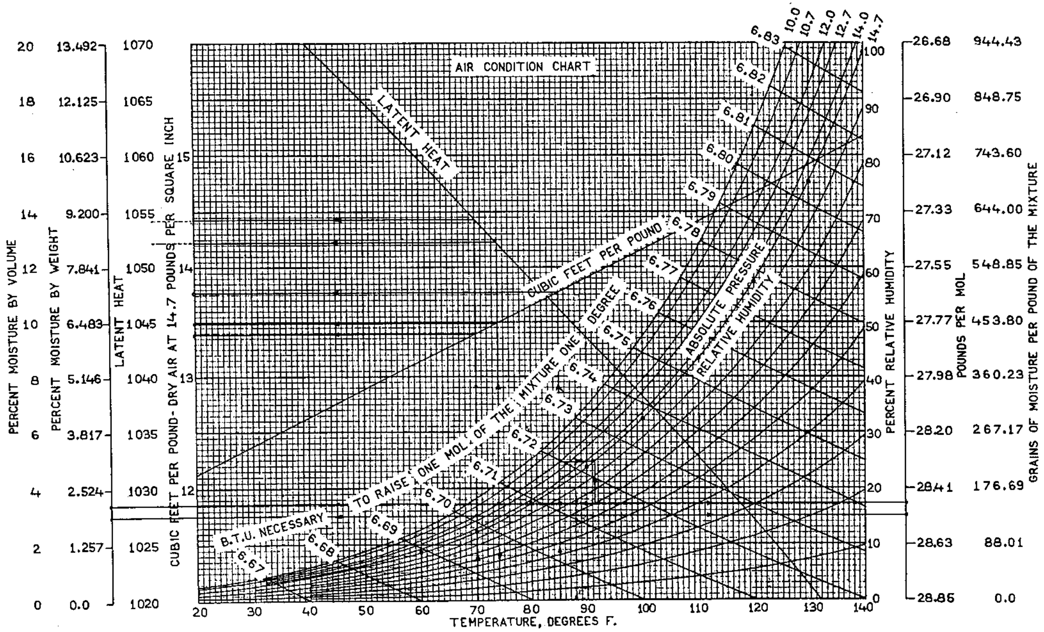


Figure 8-9—C.M. Ware historical chart, 1931.

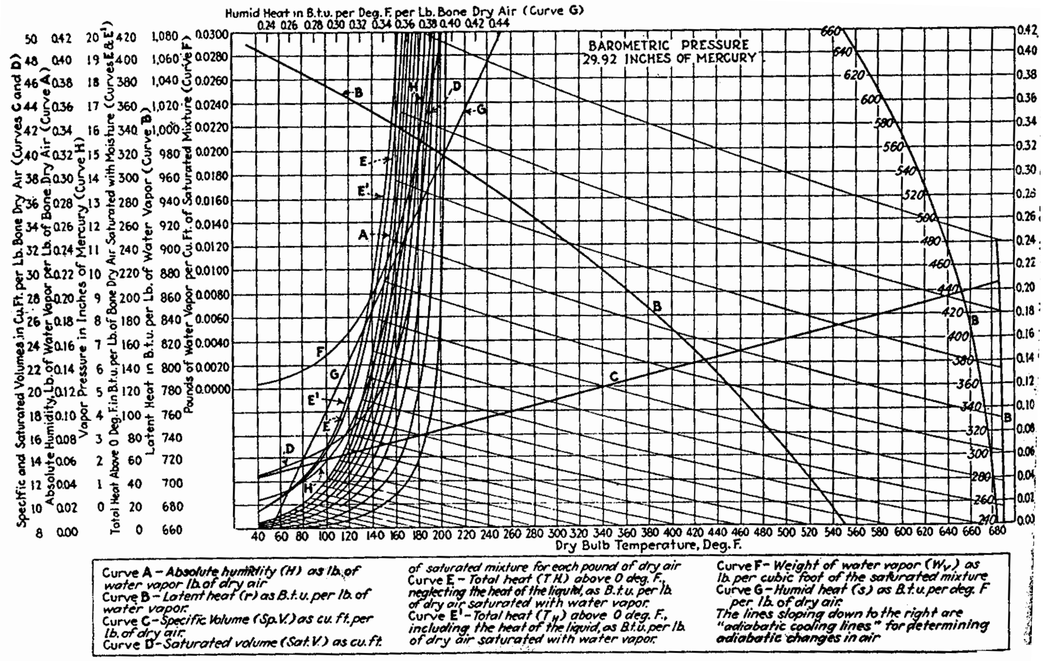


Figure 8-10—Lavine and Sutherland historical chart, 1931.

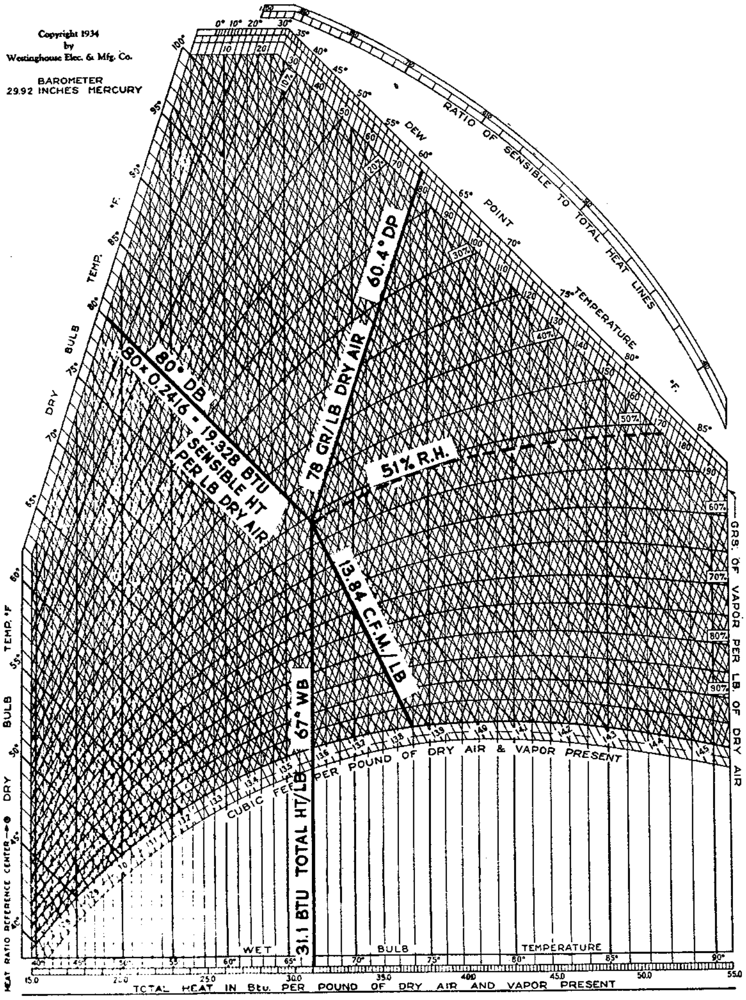


Figure 8-11—A.M. Norris psychrograph, 1934.

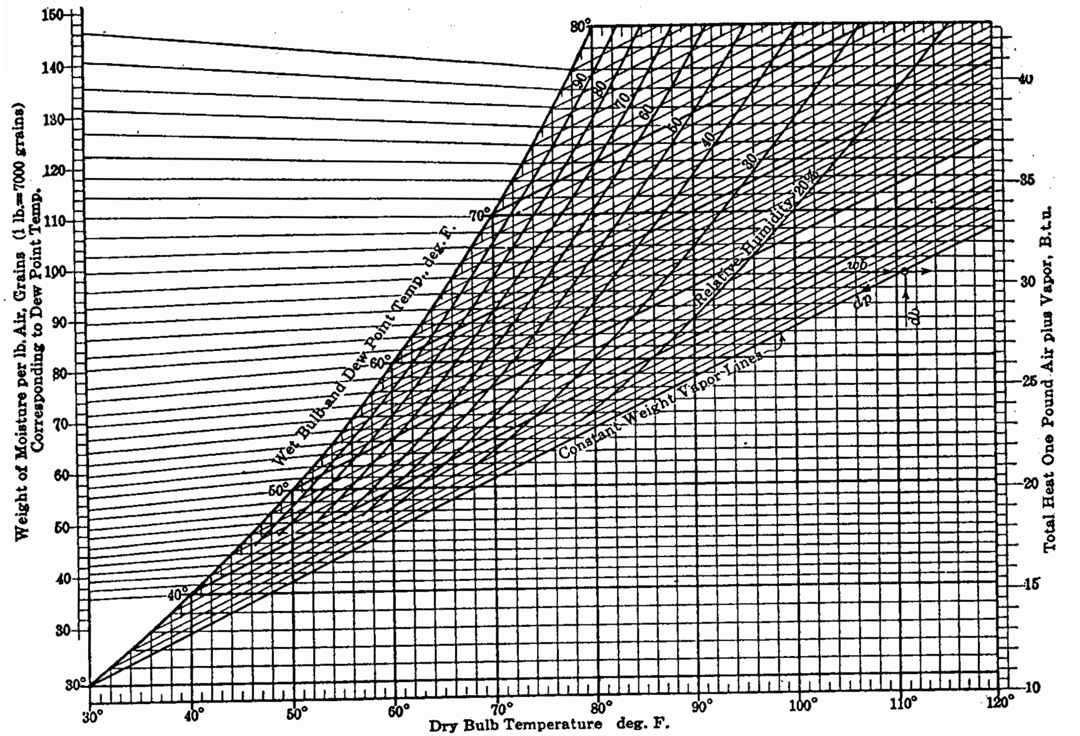


Fig. 1. Psychrometric Chart

Figure 8-12—Historical chart from Kent's handbook, 1936.

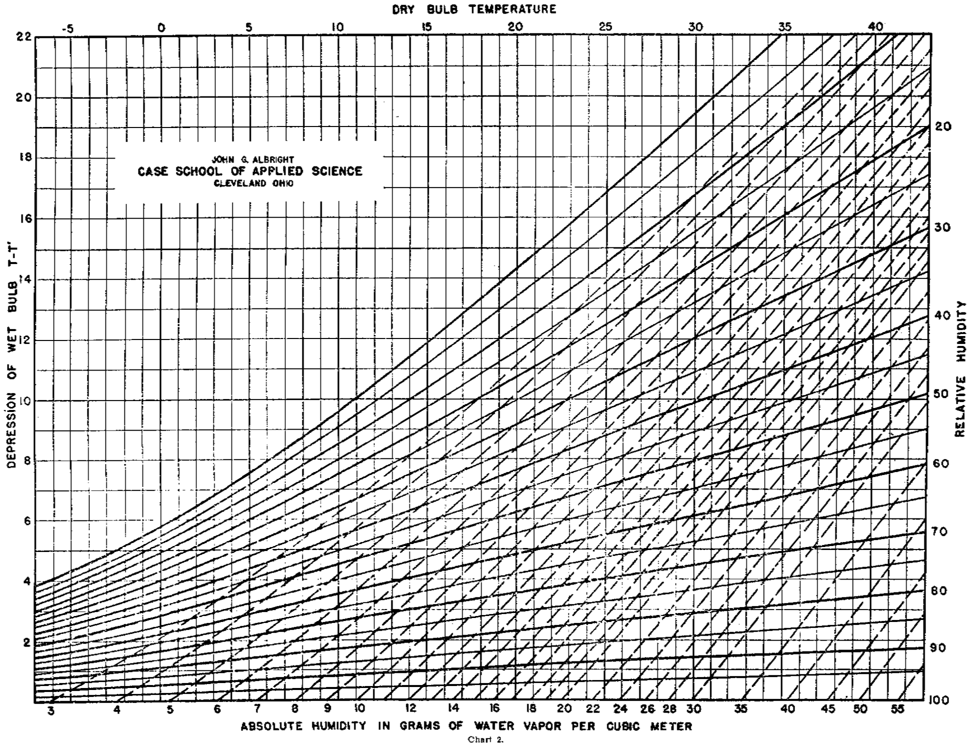


Figure 8-13—John G. Albright historical chart, date unknown.

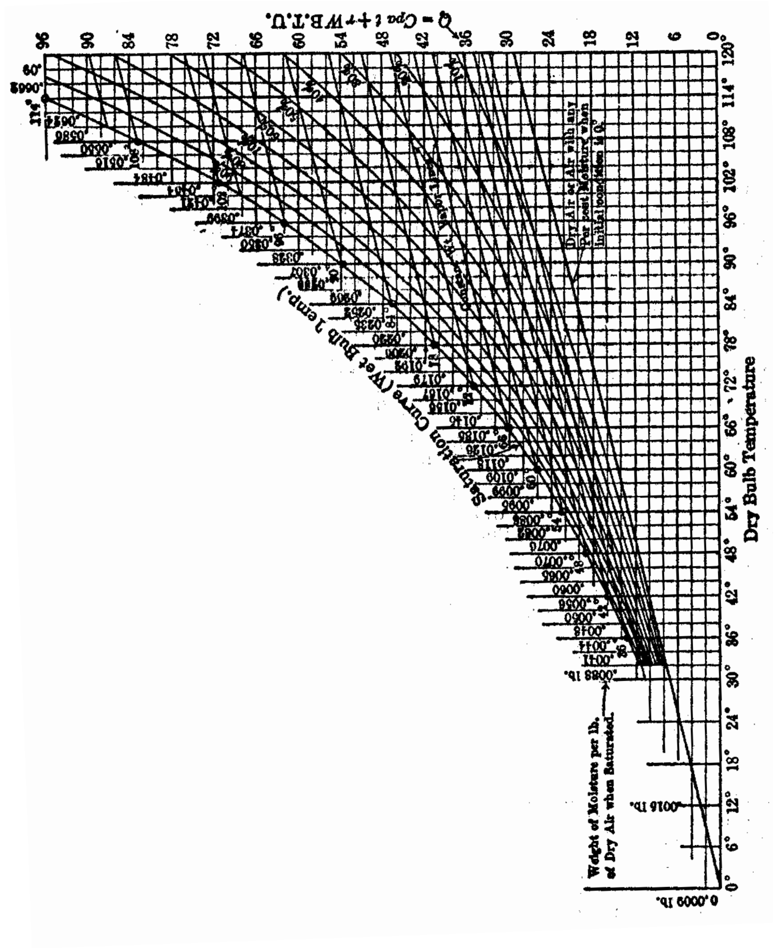


Figure 8-14—L.A. Harding historical chart, date unknown.

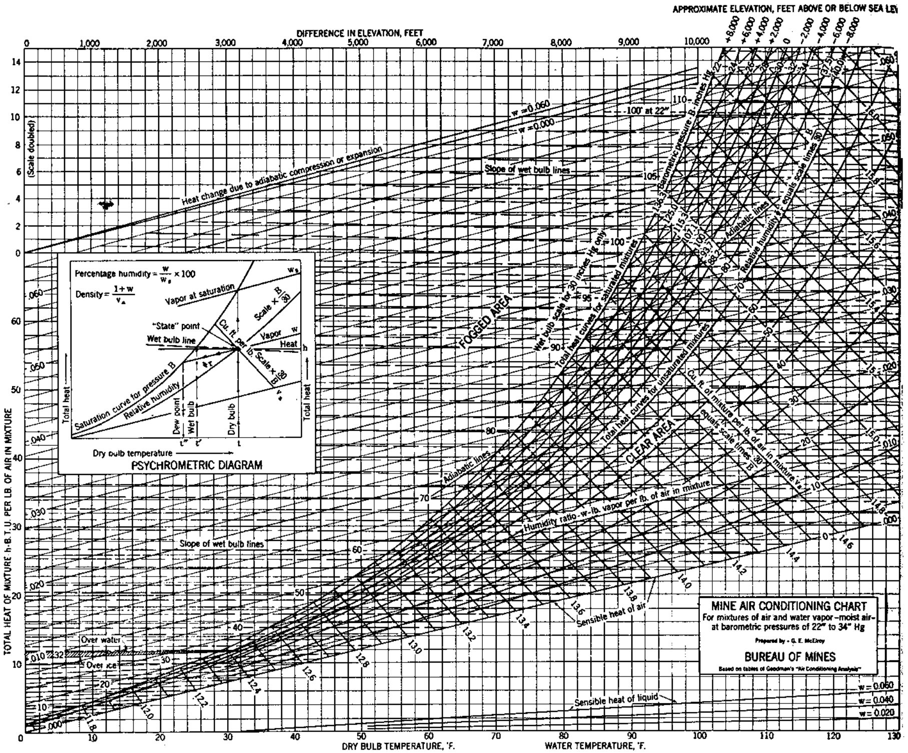


Figure 8-15—Multi-pressure chart, G.E. McElroy, Bureau of Mines, 1947.

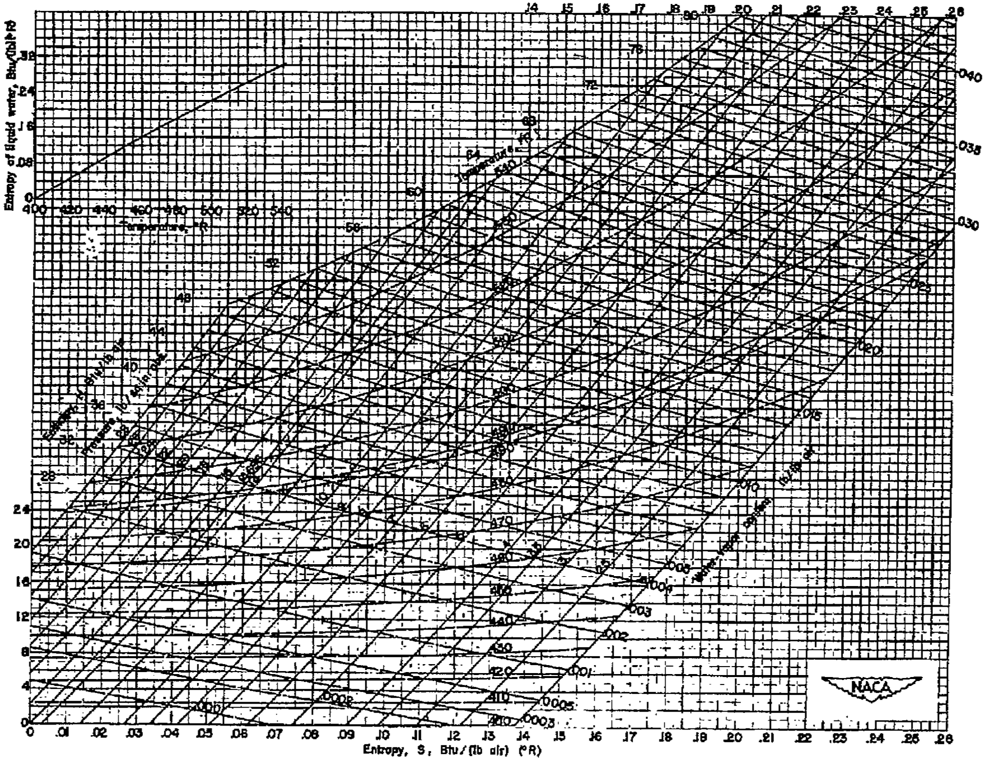


Figure 8-16—NACA historical psychrometric chart (NACA TN 1715), 1948.

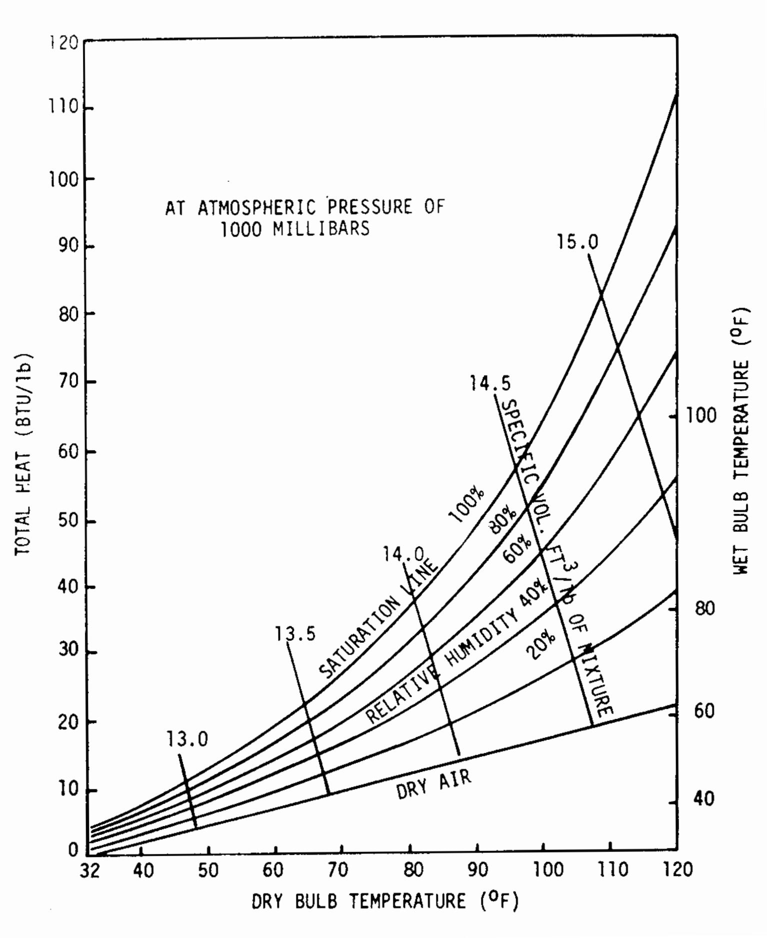


Figure 8-17—B. Woods and P. Betts historical charts, 1950.

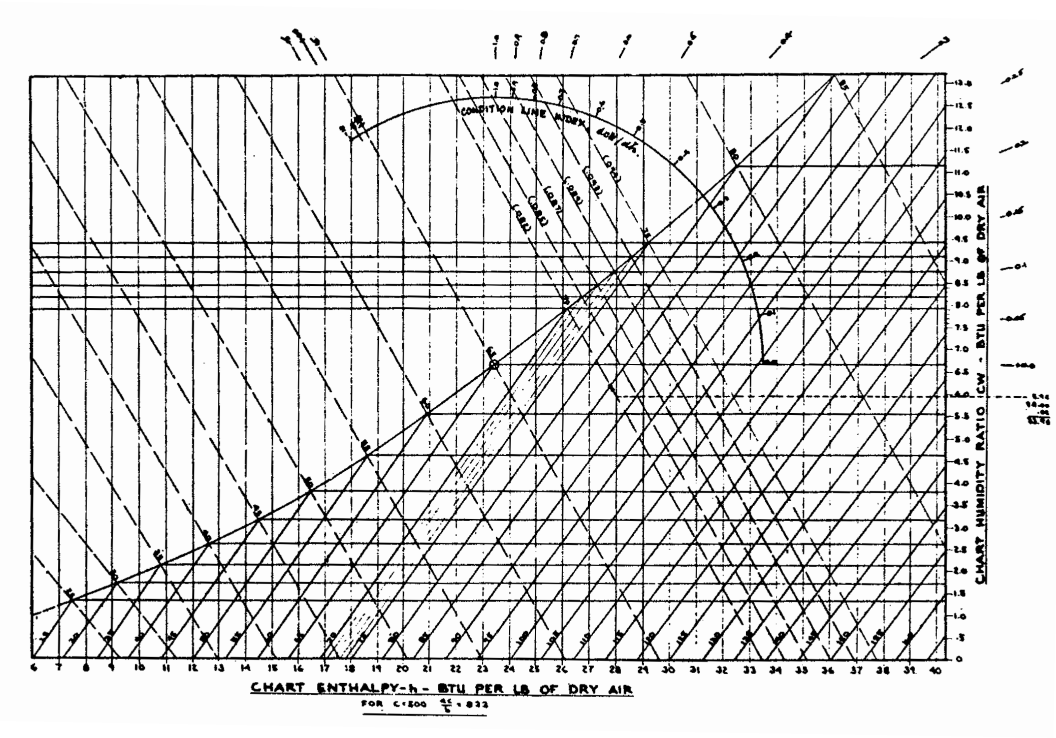


Figure 8-18—J.A. Goff historical chart, 1960.

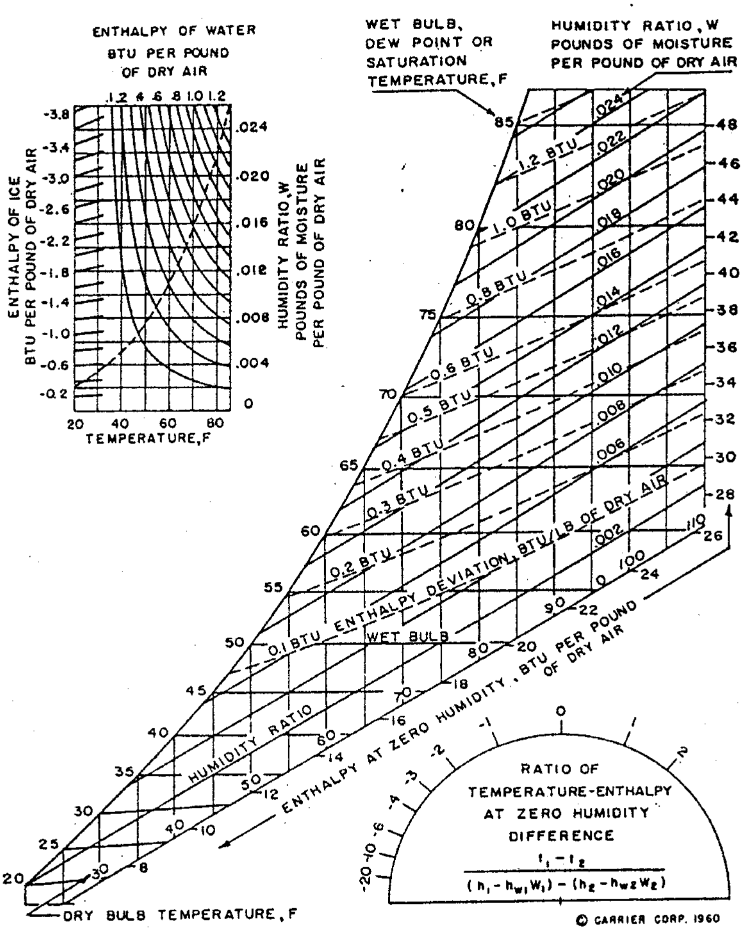


Figure 8-19—C.M. Ashley historical chart, 1960.

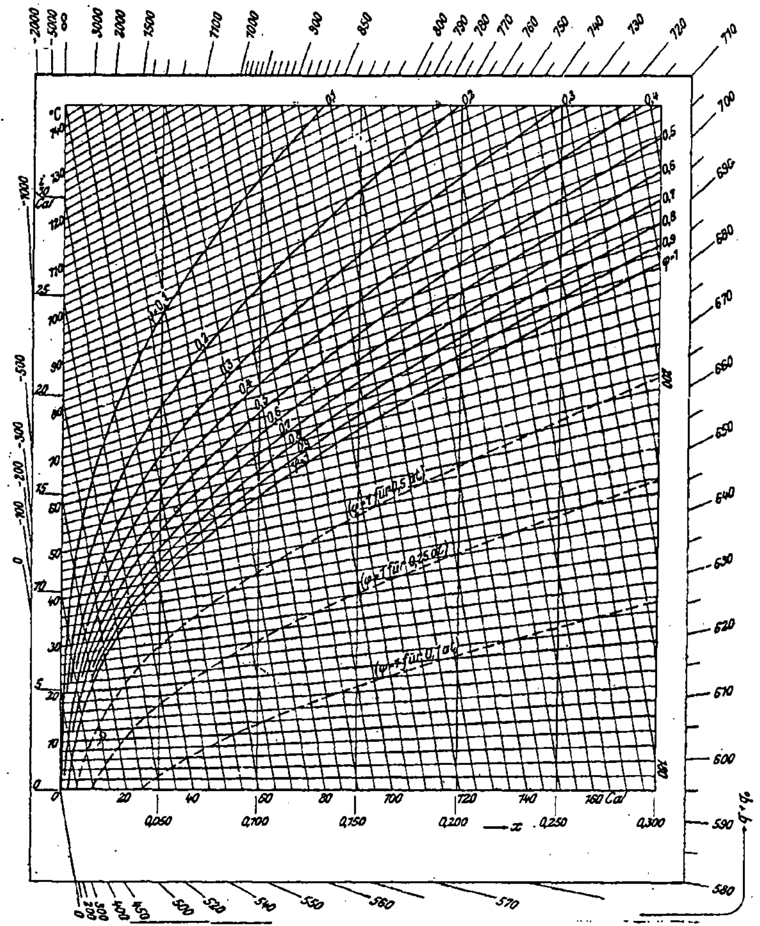


Abb. 1. Diagramm für Dampfluftgemische.

Figure 8-20—Mollier ix or hx (psychrometric) diagram.

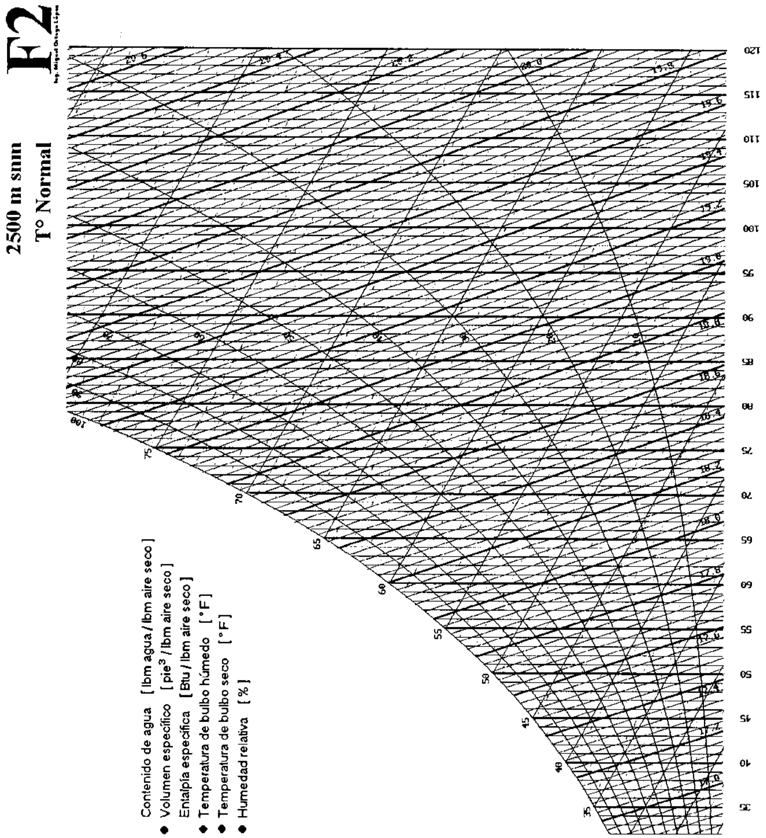


Figure 8-21—M. Ortega (Ecuador) Spanish chart, 1997.

Psychrometric Properties

> *I often say that when you can measure what you are speaking about, and express it in numbers, you know something about it, but when you cannot express it in numbers, your knowledge is of a meager and unsatisfactory kind;*
> *it may be the beginning of knowledge, but you have scarcely in your thoughts, advanced to the stage of science, whatever the matter may be.*
> *—William Thompson (Lord Kelvin), 1883*

9 Psychrometric Properties and Conventions

A property in physics is any observable, measurable, or calculable characteristic of a substance that depends only on the state of the substance (a property always has the same value under a given set of physical conditions).

The psychrometric state of a mixture of dry air and water vapour is fixed when the values of three independent properties are known, e.g., (1) barometric pressure or altitude, (2) dry-bulb temperature, and either (3a) wet-bulb temperature, (3b) relative humidity, or (3c) dew-point temperature.

A psychrometric chart (Figure 9-1) displays up to nine properties of moist air. There are many other psychrometric properties that are not displayed on the chart.

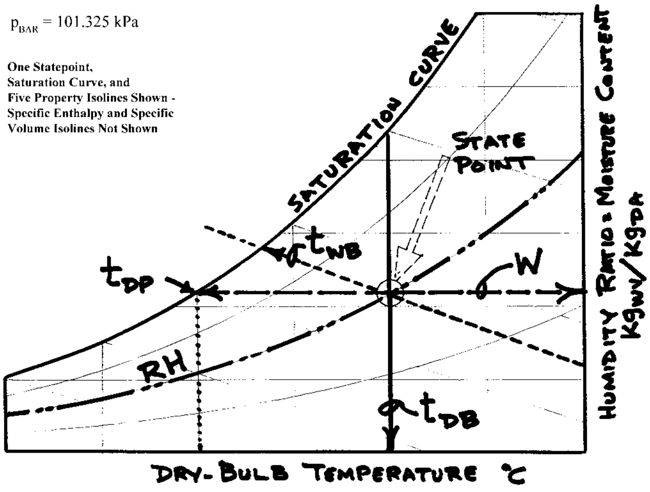


Figure 9-1—Basic psychrometric chart.

Table 9-1

Measurable Psychrometric Properties			Calculable Psychrometric Properties		
Dry-bulb temperature	t_{DB}	°C	Specific enthalpy	h	$\mathrm{kJ/kg_{DA}}$
Wet-bulb temperature	t_{WB}	°C	Specific volume	v	$\mathrm{m^3/kg_{DA}}$
Dew-point temperature	t_{DP}	°C	Humidity ratio	W	$\mathrm{kg_{WV}/kg_{DA}}$
Relative humidity	RH	%	Water vapour pressure	p_{WV}	Pa
Barometric pressure	P_{BAR}	Pa			

MEASURED VS. COMPUTED PROPERTIES

Some properties can be observed or measured using simple instrumentation, some properties can be computed from other properties or measured in the laboratory with expensive instrumentation, and certain properties can only be computed from other properties. Table 9-1 categorizes the nine common psychrometric properties displayed on a psychrometric chart into (1) those that are easily measurable with field instrumentation and (2) all others listed under calculable properties. The measurable properties on the left are most often used to fix the statepoint of the moist air and thus determine the calculable properties. The calculable properties are used in process calculations to determine energy and *water vapour* changes.

OTHER PSYCHROMETRIC PROPERTIES

The number of possible psychrometric properties is greater than 50 (one psychrometric chart software program lists 63). A property can be a combination of properties such as enthalpy ($h = u + p \cdot v$), the reciprocal of a property such as the relationship between density and specific volume ($\rho_{DA} = 1/v_{DA}$), the logarithm of a property, or a basic property raised to a power. Properties often used in psychrometric process calculations include: C_{P_DA} (specific heat capacity of *dry air*), C_{P_WV} (specific heat capacity of *water vapour*), C_{P_W} (specific heat capacity of liquid water), C_{P_ICE} (specific heat capacity of solid water), h_{FG_H2O} (latent heat of vaporization from the liquid phase to the gas phase of water at a specific temperature), and h_{SG_H2O} (latent heat of sublimation from the solid phase to the gas phase of water at a specific temperature). Many of these values are listed in the properties and constants table in Chapter 4, "Water Vapour."

PSYCHROMETRIC CONVENTION FOR THE PROPERTIES SPECIFIC ENTHALPY AND SPECIFIC VOLUME

Early psychrometric calculations were patterned after chemical calculations involving mixtures and solutions because in most psychrometric processes, the mass flow rate of the *dry air* component is unchanged throughout a single or sequential series of processes while at the same time the mass flow rate of the *water-vapour* component changes throughout many processes (e.g., water vapour is either added as in humidification or removed as in dehumidification or precipitated from the atmosphere as rain). As a matter of custom and convenience, the equations used in psychrometric process calculations are based on the mass flow rate of the *dry air* component.

This, in turn, requires that the units of *specific enthalpy* are energy/ mass_{DA} and the units of *specific volume* are volume/mass_{DA}. Also note that *humidity ratio* units are mass_{WV}/mass_{DA}.

INDIVIDUAL PSYCHROMETRIC PROPERTY CHAPTERS

The following chapters present the most common psychrometric properties in the order listed in Table 9-1.

> *The development of a standardized temperature scale*
> *was a long and bumbling process.*
> *—Octave Levenspiel*

10 Dry-Bulb Temperature

DRY-BULB TEMPERATURE
Symbol: t_{DB} with Units of °C or K

Definition

Dry-bulb temperature is the temperature of air measured by an ordinary thermometer or a sensor-transducer that gives identical values. In the SI system of units, the temperature is expressed in either degrees Celsius (°C) or kelvin (K). Dry-bulb temperature is the true temperature of moist air at rest, i.e., the temperature of air measured by a thermometer not subjected to evaporation, condensation, or radiation and when air velocity impacting the temperature sensor is not so high that the energy conversion from impact on the sensor increases the reading. The term *dry-bulb* is customarily added to *temperature* to distinguish it from *wet-bulb* and *dew-point* temperature. If *temperature* only is used without a prefix it usually means *dry-bulb* temperature.

The dry-bulb temperature reading is specific to the air in the vicinity of the temperature sensing device. The term *air* as used in this text is defined to mean *ambient* or *moist air,* which simply means a mixture of *dry air* and *water vapour*. Air in the troposphere (the atmosphere from the surface of the earth up to an altitude of approximately 11 km) always contains *water vapour* from a low value of about 0.005% to a high of about 4% compared to the dry-air mass.

Since air (in this text) means a mixture of *dry air* and *water vapour*, it follows that *dry-bulb temperature* is the *temperature* of not only the *dry air* component but also the temperature of the *water-vapour* component.

Measurement and Use

Dry-bulb temperature sensing element types include thermocouple, thermister, RTD, bi-metal, and mercury bulb. Dry-bulb temperature is the easiest psychrometric property to accurately measure and is usually more accurate than the measure of other properties such as relative humidity or psychrometer wet-bulb temperature.

Dry-bulb temperature is useful:

- As one of the three properties required to fix the psychrometric statepoint of moist air.
- In calculating the ending dry-bulb temperature of a sensible energy process knowing energy transfer and the mass flow of air, specific heat, and statepoint of entering moist air.
- In calculating the sensible energy transfer knowing the beginning and ending statepoints, the mass flow of air, and the specific heat capacity of the moist air.
- In calculating heat flow through building elements knowing the *dry-bulb* temperature difference between the inside air and the outside air and the thermal conductance of the intervening wall including the inside and outside air layers.

Sensible heat transfer is, by definition, the change in dry-bulb temperature without a change in the mass of *water vapour* associated with the air during the process. Sensible heat (transfer) in a psychrometric process is:

$$q_{SEN} = m_{DA} \cdot (h_{ENT} - h_{LVG}) = m_{DA} \cdot C_{P\_MOIST\_AIR} \cdot (t_{DB\_ENT} - t_{DB\_LVG}).$$

Dry-Bulb Temperature and the Psychrometric Chart

A uniform scale for *dry-bulb* temperature is shown at the bottom (abscissa) of the psychrometric chart along the horizontal zero *humidity ratio* isoline. Dry-bulb temperature lines on many psychrometric charts are *vertical* (for example, the original 1902 Carrier chart) or *vertical for all practical purposes* (the current ASHRAE psychrometric charts). Dry-bulb temperature lines are shown in Figure 10-1. (See Chapter 5, "Basics of the Psychrometric Chart," for a discussion of the slight differences between the Carrier and ASHRAE charts and the Mollier hx diagram.)

Using Equation 10-1 for calculation of specific enthalpy, observe that the sensible specific enthalpy difference between two adjacent dry-bulb temperature isolines is greater at increased values of W, such as at $W = 0.03$ rather than at $W = 0.00$. This accounts for the slight divergence of *dry-bulb* temperature lines with increasing humidity ratio on an ASHRAE or Mollier psychrometric chart using enthalpy-*humidity ratio* coordinates.

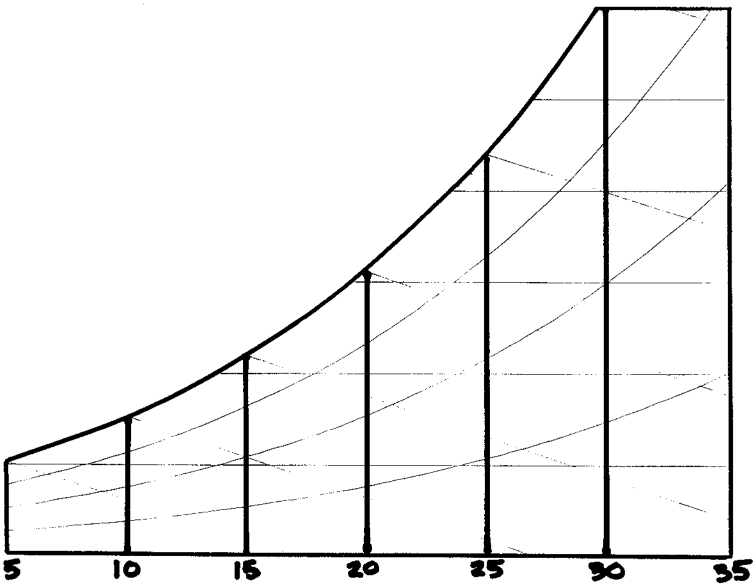


Figure 10-1—Dry-bulb temperature (t_{DB} in units of °C).

Development of the Plotting Equation for Dry-Bulb Temperature Chart Isolines

Dry-bulb temperature is a measure of the temperature of both *dry air* and the *water vapour* mixed with the air. At any statepoint on a *dry-bulb* temperature isoline, the enthalpy is as shown in Equation 10-1:

$$h = t_{DB}\,(C_{P-DA} + C_{P-WV} \cdot W) \qquad (10\text{-}1)$$

where

h = specific enthalpy [kJ/kg$_{DA}$]
t_{DB} = dry-bulb temperature [°C]
C_{P-DA} = specific heat of dry air [1.006 kJ/kg$_{DA}$]
C_{P-WV} = specific heat of water vapour [1.84 kJ/kg$_{WV}$]
W = humidity ratio [kg$_{WV}$/kg$_{DA}$]

Cooling by evaporation must have been observed first
when a naked man swam a river and emerged on the other side.
—W.E. Knowles Middleton, 1969

The analysis of the wet-bulb thermometer is a complex phenomenon that involves simultaneous convective heat and mass transfer considerations.
—K.L. Christianson and K.N. Newhouse, 1985

It is fortuitous for the mixture of dry air-water vapour that psychrometer and thermodynamic wet-bulb temperatures are for practical purposes equal provided that the psychrometer is correctly used.

11 Wet-Bulb Temperature

WET-BULB TEMPERATURE
Symbol: t_{WB} with Units of °C or K

Layperson Definitions

- The lowest dry-bulb temperature that can be attained by evaporatively cooling air at 100% saturation efficiency.
- The temperature indicated by an ordinary thermometer having its sensing section (bulb) covered with a porous wetted sleeve.
- The temperature of evaporation.

Definitions

In psychrometrics, there are two wet-bulb temperatures:

- *thermodynamic wet-bulb temperature* (also called the *temperature of adiabatic saturation*). For any given moist air state, thermodynamic wet-bulb temperature has a fixed mathematical value based on thermodynamic properties.
- *psychrometer wet-bulb temperature*, which is not a thermodynamic property because it is dependent on heat and mass transfer rates which are variable.

For *dry air-water vapour* mixtures, the term *wet-bulb temperature* in this text means either *thermodynamic wet-bulb temperature* or *psychrometer wet-bulb temperature* with the understanding that the psychrometer measurement technique must conform with industry recommendations.

Thermodynamic Wet-Bulb Temperature

Thermodynamic wet-bulb or adiabatic saturation temperature can be defined in a number of ways, including:

- the saturation temperature to which moist air can be adiabatically (without gain or loss of heat) cooled by evaporation of H_2O (liquid or ice) at that same (saturation) temperature into the moist air
- the lowest temperature attainable when liquid water or ice (at the final temperature) is brought into intimate contact with moist air
- the temperature at which liquid water or ice may be evaporated or sublimated into air to bring the *water vapour* to a state of saturation at exactly the same wet-bulb temperature and pressure
- a mathematical expression of the *conservation of energy* and *conservation of mass* for an adiabatic constant pressure process in which a mixture of a dry gas and the vapour of a volatile substance (H_2O in the case of moist air) is brought to a state of saturation as it is passed over the liquid or solid phase of the volatile substance

Thermodynamic wet-bulb temperature is calculated using mass and energy balance equations presented in the adiabatic saturator discussion later in this chapter. In theory, it could be measured using an adiabatic saturator and for this reason it is sometimes called *temperature of adiabatic saturation*. Realistically, the adiabatic saturator is a theoretical instrument because it must be perfectly insulated, be very long, and have intimate contact between the air and water.

The wet-bulb temperature isolines on Mollier and ASHRAE (after 1963) psychrometric diagrams are plotted using the thermodynamic temperature property of the adiabatic saturator.

Psychrometer Wet-Bulb Temperature

Psychrometer wet-bulb temperature for the mixture of dry air and water vapour is the temperature measured by an ordinary thermometer that has its sensing element covered with a clean, soft cotton sleeve wetted with distilled water and exposed to moist air moving at 4 ± 1 m/s. The thermometer bulb should be shielded from direct solar radiation but exposed to normal background radiation. The purpose of the wetted sleeve is to provide a means of keeping the thermometer bulb wet. One end of the sleeve, called the wick portion, usually continues to a small reservoir of distilled water. A portion of the wick must be exposed to the moving air so that the temperature of the water reaching the sensor is close to the final wet-bulb temperature prior to taking the final readings.

When the wet-bulb depression, $(t_{DB} - t_{WB})$, is greater than 10°C, it becomes increasingly difficult to maintain the temperature of the free wick (between the sensor and the liquid reservoir) at the final wet-bulb temperature because the reservoir is usually exposed to the moving air dry-bulb temperature, which warms the liquid in the reservoir. If the exposed and wetted free wick is not able to reduce this higher temperature water to the wet-bulb temperature, then the final reading will be slightly high.

Psychrometer wet-bulb temperature is not a true thermodynamic property because it is dependent on heat and mass transfer rates, which are functions of the following variables: air velocity, external radiation and view factor, water temperature as it arrives at the bulb-wick interface, diffusivity of the sock, convection coefficient, and heat conduction up the thermometer stem.

The psychrometer wet-bulb thermometer's combination heat and mass transfer process is not the same as the adiabatic saturation process; however, for the air and H_2O mixture, the deviation between the two wet-bulb temperatures is relatively small when the psychrometer is correctly used. This is not true for other volatile compounds in contact with other mixtures of dry gases and the vapour phase of the volatile compound.

Use

Wet-bulb temperature is used for rating cooling coils, cooling towers, evaporative condensers, evaporative coolers, air washers, and evaporative-type humidifiers. The chemical dehumidification process, using a solid adsorbent desiccant, closely follows the entering wet-bulb isoline as the air becomes progressively warmer and drier (the opposite of the evaporative cooling process).

Measurement

Psychrometer wet-bulb temperature is measured using either a sling psychrometer (Figure 11-1) or an aspirating psychrometer (Figure 11-2). These instruments use two identical thermometers—one covered with a porous wetted sleeve.

The user must ensure that the sleeve remains wet and that air moves over it at approximately 4 m/s. Once the water on the wick attached to the reservoir water cools, the temperature is read at 15- to 30-second intervals. The lowest observed temperature is the psychrometer wet-bulb temperature. If the wet-bulb temperature is below freezing, the wick is removed and the sensing bulb is coated with a layer of ice prior to exposing it to moving air. Users of these instruments should understand their accuracy, limitations, and use. The theory and accurate use of sling or aspirating psychrometers is extensively covered in ASHRAE Standard 41.6 and NOAA and NIST literature.

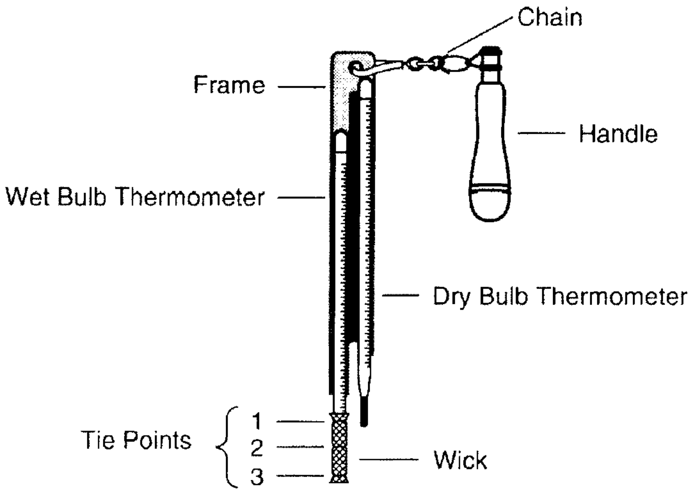


Figure 11-1—Sling psychrometer (wet-bulb and dry-bulb thermometers).

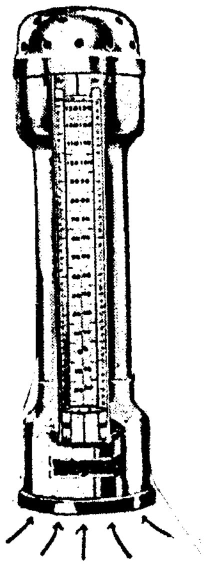


Figure 11-2—Aspirating psychrometer.

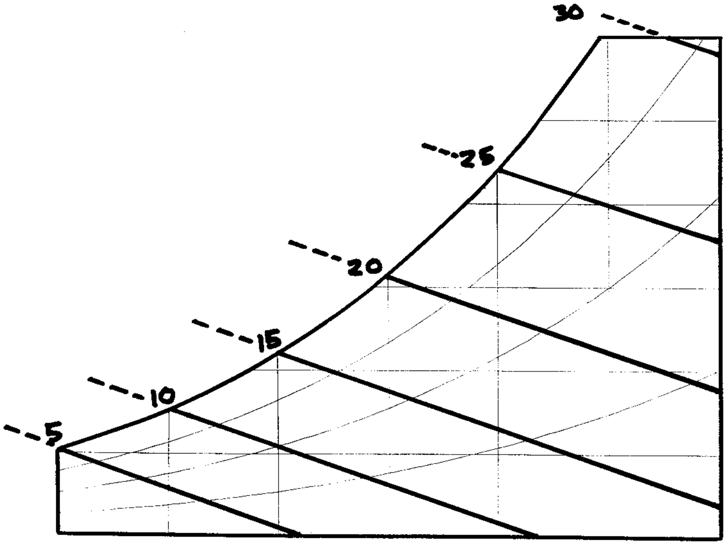


Figure 11-3—Wet-bulb temperature isolines (t_{WB} in °C).

Wet-Bulb Temperature and the Psychrometric Chart

Thermodynamic wet-bulb temperature isolines on an ASHRAE (Figure 11-3) or a Mollier psychrometric chart are straight and nearly parallel with specific enthalpy isolines. The nearly identical slopes of the wet-bulb temperature and specific enthalpy isolines indicate that these two properties are closely related. *Wet-bulb temperature* provides a good indication of the relative magnitude of the enthalpy of moist air.

In order to avoid a multiplicity of lines on the psychrometric chart, most charts do not show enthalpy lines. Most charts include enthalpy edge scales so that enthalpy can be graphically determined.

Adiabatic Saturator Discussion

In a perfectly insulated adiabatic saturator (Figure 11-4) the moist air passing through the saturator will take on additional *water vapour* (the dew-point temperature and humidity ratio will increase) and the dry-bulb temperature of the air will decrease until the air is fully saturated with *water vapour*. The latent energy necessary to evaporate the water (or sublimate the ice) comes from the sensible energy of the moist air passing through the saturator. The amount of water (or ice) in the saturator is large

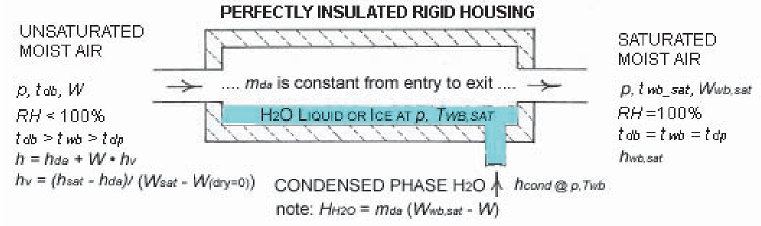


Figure 11-4—Adiabatic saturator.

in comparison to the amount of water evaporated into the air. The air leaving the saturator is fully saturated (100% relative humidity) and the leaving dry-bulb, wet-bulb, and dew-point temperatures are equal. Exhaustive laboratory analysis has shown that the wet-bulb temperature and the temperature of the water or ice are constant throughout the saturator. If the saturator is operated continuously, it will be necessary to replenish the water (or ice). If the body of water or ice in the saturator is not at the final temperature, then the saturator should be operated until this occurs.

$$h + (W_{wb,\,sat} - W) \cdot h_{cond.\,@p,\,twb} = h_{wb,\,sat} \tag{11-1}$$

where

h = enthalpy of entering moist air at pressure p, temperature t_{db}, and humidity ratio W

$W_{wb,sat}$ = mixing ratio of exiting saturated moist air at pressure p and temperature $t_{wb,sat}$

W mixing ratio of entering unsaturated moist air at pressure p and temperature t_{db}

$h_{cond.@p,\,t_{twb}}$ = enthalpy of 1 kg of condensed water at pressure p and temperature t_{wb}

$h_{wb,sat}$ = enthalpy of saturated moist air at pressure p, temperature $T_{wb,sat}$, and humidity ratio W_{sat}

and

$$h = h_{da} + W \cdot h_{v,\,H_2O} \tag{11-2}$$

$$h_{v,\,H_2O} = (h_{db,\,sat} - h_{da}) / (W@p,\,t_{db,\,sat} - W_{da\,=\,0}) \tag{11-3}$$

Note: $W_{da\,=\,0} = 0$ *and can be eliminated.*

The final equation is therefore:

$$h_{da} + W \cdot h_{v,\,H_2O} + (W_{wb,\,sat} - W) \cdot h_{cond.@p,\,twb} = h_{wb,\,sat} \qquad (11\text{-}4)$$

which, after rearranging to solve for *W*, yields

$$W = \frac{(h_{wb,\,sat} - W_{wb,\,sat} \cdot h_{cond@p,\,Twb} - h_{da})}{(h_{v,\,H_2O} - h_{cond@p,\,Twb})} \qquad (11\text{-}5)$$

Note: The development of the adiabatic saturator equations in many texts, including previous editions of this text, differs from the above. Many versions include specific heat capacity and latent heat of vaporization terms, which may impact accuracy. The revision above follows this quote from Albert Einstein: "Explanations should be made as simple as possible, but not simpler." The resulting equations are as simple as possible and have absolute accuracy. Accuracy is ensured because Equation 11-2 accounts for the slight interaction of water vapour and air molecules and actual atmospheric pressure.

The preceding equations are applicable for calculations based on moist air as an ideal gas and also as a real gas providing of course that enthalpy and humidity ratio values are obtained from real gas models.

The equation for $h_{v,H2O}$ utilizes proven psychrometric equations for *h* and *w* and ensures rational consistency because the value of $h_{v,H2O}$ is based on actual atmospheric pressure and the interactions between water vapour and dry air molecules.

When *p*, *t*, and t_{wb} are known then all terms except *W* are known and *W* can be directly calculated without iteration. In other cases *p*, *t,* and a humidity property (*W*, *RH*, or t_{dp}) are known and t_{wb} is the unknown, in which case iteration of Equation 11-5 is necessary.

Equation 11-5 is a rationally consistent equation that results in the most accurate calculation of the adiabatic saturation temperature. Many publications from 1911 through the 1990s presented other equations using temperatures, average specific heat, and latent heat terms that were better suited to the manual calculations of that era. These equations produced acceptable results; however, for ultimate accuracy and internal consistency Equation 11-5 should be used.

Other Facts Relating to Wet-Bulb Temperature

Thermodynamicists do not regard psychrometer wet-bulb temperature as a thermodynamic property because it does not have a mathematically fixed value for a given state of moist air.

The psychrometer with its two thermometers gives simultaneous readings of dry-bulb temperature and wet-bulb temperature. Some psychrometers include a scale used to compute relative humidity. From the

1890s through the 1980s, wet-bulb temperature as measured by a sling or aspirated psychrometer was widely used in determining relative humidity and other properties of moist air, particularly in industrial environments for printing, textile processes, and agriculture laboratories. With the advent of electronic measuring instruments, psychrometers are being replaced by instruments having separate dry-bulb and relative humidity sensors. These instruments have computer chips with built-in algorithms for computing wet-bulb temperature and, in some cases, dew-point temperature.

Wet-bulb temperature has also been called “screen” or “stationary” wet-bulb temperature, terms used to describe a wet-bulb thermometer in a small louvered and screened instrument housing used by weather bureaus in the early 1900s. These terms could also be used to describe the inexpensive wet-bulb thermometers sold for use in the home and sometimes as one of a package of weather components, including a barometer, a dry-bulb thermometer, and a wet-bulb thermometer. Screen or stationary wet-bulb thermometer readings are subject to significant errors because of the unpredictable airflow across the wet bulb.

A constant wet-bulb temperature process from an initial lower to a final higher humidity ratio differs slightly from a constant enthalpy process. The reason for this is the slight gain in enthalpy associated with the mass of added H_2O.

For a 0.00°C wet-bulb process, the wet-bulb line and the enthalpy line of the psychrometric chart coincide because liquid water at 0.00°C has an enthalpy value of zero; thus, the mass of added water at 0.00°C does not increase the enthalpy of the mixture.

Wet-bulb processes can also occur in the presence of H_2O in the solid phase (ice), in which case the process described herein will experience a somewhat significant loss in enthalpy because the mass of added water vapour is from ice, which has a negative enthalpy value.

Wet-bulb thermometer use is not limited to measuring the wet-bulb temperature of moist air. It is also used in the chemical industry to measure the wet-bulb temperature of other mixtures of a dry gas and a volatile substance, including hydrogen-water, hydrogen-carbon tetrachloride, oxygen-carbon tetrachloride, dry air-alcohol, dry air-toluene, dry air-benzene, etc. Thermodynamic wet-bulb temperature and psychrometer wet-bulb temperature are not equal for these and most other volatile compounds in contact with other dry gases and the vapour phase of the volatile compound.

... the lowest temperature to which air with a given mass of water vapour may be cooled without precipitation of a portion of the water vapour.
—Willis H. Carrier, 1911, ASME: "Rational Psychrometric Theory"

Dew-point is the most vivid way of indicating the moisture content of the air, if not a direct measure of this content.... The curve of the moisture content vs. dewpoint (saturation curve) is very nearly an exponential curve, doubling approximately every 10°C, so that at higher temperatures a degree increment in dewpoint means a higher increment in the moisture content than at lower dewpoints.
—Serge J. Zaroodny, 1939 (°F changed to °C)

12 Dew-Point Temperature

DEW-POINT TEMPERATURE
Symbol: t_{DP} with Units of °C or K

Layperson Definitions

- The surface temperature at which dew (liquid condensation) or frost* (solid deposition) first forms.
- The temperature at which the volume is saturated with and can contain no more *water vapour* and below which dew, fog, or clouds can form.
- The surface temperature at which *water vapour* just starts to condense.

Scientific Definition

For *any* condensable vapour, *dew-point temperature* is the temperature at which the vapour phase of a substance will be saturated without a change in pressure, i.e., the saturation temperature corresponding to the actual vapour pressure. Dew-point temperature of a vapour and saturation temperature of a vapour are synonymous. *This definition applies to any vapour!* If applied to *water vapour,* the dew-point temperature is the temperature to which the *water vapour* must be cooled in order to become saturated with respect to *water vapour.* In a mixture of dry gases and a condensable vapour, *dew-point temperature* applies only to the vapour component.

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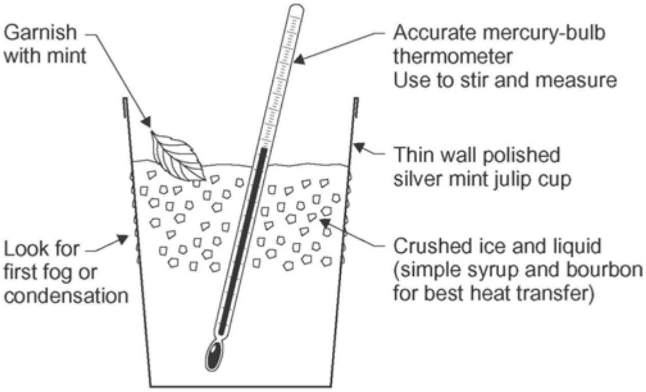
* *Technically, the temperature at which frost forms is called "frost-point temperature." In this text they will both be called "dew-point temperature."*

Measurement and Use

National Weather Service stations have recorded and published dew-point temperature every three hours for over 40 years. The three-volume final report for ASHRAE research project RP-754 dated April 1, 1995, by Donald G. Colliver, Ph.D, P.E., contains extensive dew-point temperature data for most U.S. and Canadian cities. Colliver used weather bureau tapes to prepare these data, which also appear in a much condensed form in *ASHRAE Handbook—Fundamentals*.

A dew-point hygrometer utilizes a temperature-controlled, highly polished observable surface. In the instrument's simplest form, crushed ice is slowly added to a liquid in a thin-walled silver container such as a mint julep cup (Figure 12-1). An accurate mercury bulb thermometer is used to constantly stir the liquid in the cup. When the first sign of condensation (dew) is observed on the outside of the cup, the temperature of the liquid in the cup is read as the dew-point temperature. This method requires that the temperature of the outside surface of the silver cup and the temperature of the liquid in the cup be essentially the same. In actual practice, the temperature of the liquid in the cup will be slightly lower than the outside surface temperature of the cup.



Instructions
1. Slowly add ice, stir and observe temperature and evidence of condensation.
2. Dew-point is thermometer reading when fog or condensation first appears.
3. Dispose of contents of cup in compliance with all regulations.

Figure 12-1—Dew-point temperature apparatus.

Digital recording dew-point temperature instruments use optical devices to detect the first formation of dew on a highly polished surface. Once dew is detected, the surface temperature is repeatedly raised and lowered to vaporize and then recondense the water in order to provide accuracy and a record of dew-point temperature throughout the recording period. When the readings stabilize, the dew-point temperature is the average of the lower surface temperature observed at first sign of condensation and the higher surface temperature observed as condensation is evaporated.

Many electronic instruments measure dry-bulb temperature and relative humidity and compute dew-point temperature from these measurements. Dew-point temperature obtained from a true dew-point temperature instrument is more accurate than dew-point temperature computed from dry-bulb temperature and relative humidity because of slight errors in measuring dry-bulb temperature and more significant errors in accurately measuring relative humidity. A dew-point temperature instrument is often used to calibrate the less expensive, more portable electronic temperature-humidity instrument.

Dew-point temperature is an extremely useful property for relating (1) the potential for and direction of movement (diffusion) of *water vapour*, (2) the potential for condensation, and (3) the relative magnitude of the latent cooling or latent heating load associated with either infiltration or deliberate introduction of unconditioned outside air into a conditioned space as well as the impact of that latent load on space relative humidity.

Water vapour moves or diffuses from regions of higher concentration to regions of lower concentration (concentration can be expressed as water vapour density, pressure, or ppmv). Dew-point temperature and water vapour pressure are directly related using IAPWS, Hyland and Wexler (1983), Antoine, and other equations. It follows that water vapour moves or diffuses from zones of high dew-point temperature to zones of low dew-point temperature.

Latent energy transfer ($q_{LAT} = m_{H2O} \cdot h_{FG} = m_{DA} \cdot \Delta W \cdot h_{FG}$) is a function of the change in dew-point temperature and the latent heat of vaporization (itself a function of dew-point temperature). Humidity ratio or *water vapour* pressure may be substituted for dew-point temperature in the previous sentence because, at fixed elevation or barometric pressure, the three are mutually dependent properties.

Dew-Point Temperature and the Psychrometric Chart

Dew-point temperature isolines on a traditional psychrometric chart (single altitude with humidity ratio as the ordinate) (Figure 12-2) are horizontal lines extending from the saturation (100% RH) curve on the left

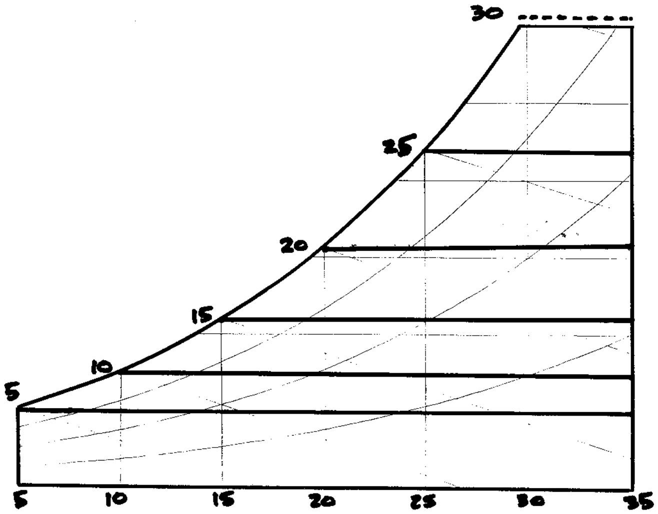


Figure 12-2—Dew-point temperature isolines (t_{DP} in units of °C).

to the right-hand boundary of the chart. Horizontal dew-point temperature lines and *water vapour* pressure lines are usually not plotted on the psychrometric chart because they are parallel to the humidity ratio lines and would add clutter, making the chart difficult to read. On some psychrometric charts, edge scales are used to display dew-point temperature and/ or *water vapour* pressure. If the edge scales are not shown on a psychrometric chart, the dew-point temperature may be determined by following a horizontal line originating at the statepoint to the saturation curve and reading the wet-bulb temperature or dry-bulb temperature at the saturated statepoint.

Discussion

The saturation curve, which is sometimes called the dew-point temperature curve, and the 100% relative humidity isoline are the same curve. The saturation curve is a plot of *water vapour* saturation temperature versus *water vapour* pressure. At saturation (100% RH), dry-bulb temperature, wet-bulb temperature, and dew-point temperature are equal. Within the body of the psychrometric chart, the *water vapour* is superheated and dry-bulb temperature is higher than wet-bulb temperature, which is higher than dew-point temperature.

Humidity ratio, dew-point temperature, and *water vapour* pressure are directly linked by equations. These properties are not independent of one another; therefore, only one of the three may be used in determining the statepoint of a moist air sample. Humidity ratio approximately doubles for each 10°C rise in dew-point temperature.

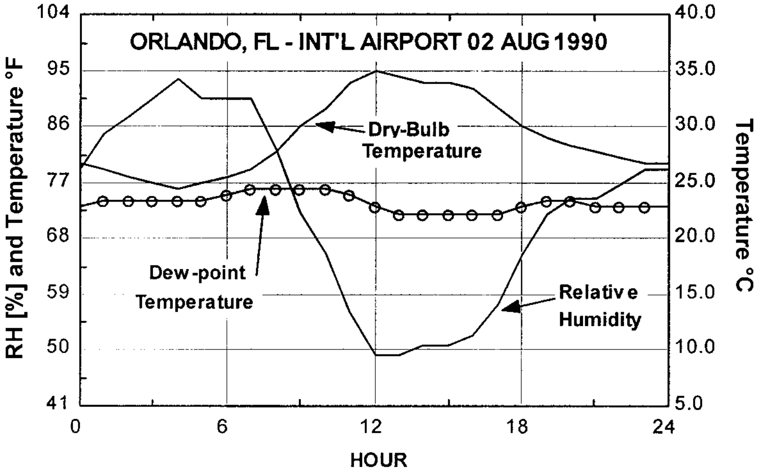


Figure 12-3—Twenty-four hour dew-point temperature and relative humidity.

In a temperate and humid climate typical of the summer climate over the eastern United States, the following approximations may be useful:

- In the absence of rain or a frontal system, the dew-point temperature is generally constant within ±1°C throughout the day and can be roughly approximated as 1°C less than the 5 a.m. dry-bulb temperature as evidenced by the 24-hour typical day plot for Orlando, Florida (See Figure 12-3).
- At any one relative humidity value, the difference between dry-bulb temperature and dew-point temperature, sometimes called dew-point temperature depression, is approximately constant over a wide range of dry-bulb temperatures as noted in Table 12-1. At 75% RH, the dew-point temperature depression is approximately 5°C. At 55% RH, the dew-point temperature depression is approximately 10°C. Within this range, a 1% decrease in RH is accompanied by a 0.25°C increase in dew-point temperature depression. Given dry-bulb temperature and relative humidity, it is possible to approximate the dew-point temperature. For example, if outside conditions are 35°C, 65% RH, the mental arithmetic follows this equation:

$$t_{DP} = 35 - [5 + (75-65) \cdot 0.25] = 27.5°\mathrm{C}$$

(versus 27.43°C using the table)

Dew-point temperature is frequently reported by TV weather reporters. In their commentary intended for the average audience, they state that

Table 12-1—(Dry-Bulb—Dew-Point) Temperature Depression °C

Relative Humidity	Dry-Bulb Temperature °C					
	Avg.	40	35	30	25	20
80%	3.83	4.12	3.97	3.83	3.69	3.55
75%	4.92	5.29	5.10	4.92	4.74	4.56
70%	6.07	6.53	6.30	6.07	5.85	5.63
65%	7.30	7.84	7.57	7.30	7.03	6.77
60%	8.62	9.25	8.93	8.61	8.30	7.99
55%	10.03	10.77	10.39	10.02	9.66	9.30
50%	11.56	12.41	11.97	11.55	11.13	10.73
45%	13.23	14.20	13.70	13.22	12.74	12.28

Values in table are for sea level conditions with P_{BAR} *= 101,325 Pa.*

dew-point temperature is a good indication of the amount of moisture (*water vapour*) in the air. As shown by the saturation curve, the relationship between the mass of *water vapour* in the air and the dew-point temperature is exponential (not linear). TV weather reporters sometimes relate the low overnight dry-bulb temperature to the dew-point temperature. Once the dry-bulb temperature lowers to the dew-point temperature (100% RH and *water vapour* saturation), it is more difficult to further lower the temperature because of the additional cooling required to condense a portion of the *water vapour.*

Endnote Quotation

As we vary the temperature of a given sample of air at constant barometric pressure the quantities that remain constant (besides, of course, the ratio of the masses of water vapour and air), are the dew-point temperature and the partial pressure of the water vapour. The direct measure of the dew-point temperature would therefore seem to be the most logical way of determining the moisture content of the air. (Serge J. Zaroodny, Municipal Architect's Office, Washington, D.C., 1939.)

Note that the portion of this quotation stating that the mass of water vapour, the dew-point temperature, and the water vapour pressure remain constant as we vary the dry-bulb temperature of the air obviously requires that the dry-bulb temperature not be reduced below the entering air dew-point temperature because that would change the mass of water vapour, the dew-point temperature, and the water vapour pressure.

There is a marked contrast between the behavior of the substance H_2O *in a space and* H_2O *in materials: the maximum water vapour content in a given volume of a gas rises steeply with temperature approximately doubling with every 10°C rise while the* H_2O *content of materials depends mainly on relative humidity and is largely unaffected by temperature.*
—restated from Tim Padfield, Ph.D.

13 Relative Humidity

RELATIVE HUMIDITY
Symbol: RH, Dimensionless

Definition—Ideal Gas Model

In the temperature range of –40°C to +65°C where experimental evidence shows that *water vapour* can be modeled as an ideal gas with acceptable results, relative humidity is *the ratio of the actual water vapour pressure to the saturated water vapour pressure at the same dry-bulb temperature*. It is customarily expressed in percentage form.

$$\text{RH}\ (\%\ format) = 100 \cdot p_{WVS}(t_{dp}) / p_{WVS}(t_{db})\Big|_p$$

p_{WVS} (*pressure of water vapour at saturation*) means that the sample, parcel, or volume when saturated with respect to *water vapour* contains the maximum possible amount of *water vapour* at the specified temperature.

By using the ideal gas law, $p \cdot v = R \cdot T$, and the fact that density (ρ) is the reciprocal of specific volume ($\rho = 1/v$), relative humidity at low pressures can also be defined in terms of either the specific volume or the density of *water vapour*. In 1911, Willis H. Carrier defined relative humidity as the ratio of the actual and saturated *water vapour* densities and called it *percent of isothermal saturation*.

Measurement and Use

Relative humidity, like dry-bulb temperature, can be measured with inexpensive instruments, which may explain why relative humidity is a widely used and reported property. Unfortunately, the accuracy of the best instruments is ±2% and, to consistently maintain this accuracy, the

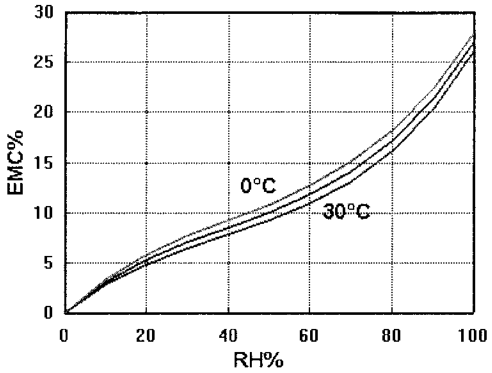


Figure 13-1—Equilibrium moisture content of wood vs. relative humidity (by permission of Tim Padfield, PhD; www.natmus.min.dk/cons/tp).

measuring device must be frequently recalibrated. The accuracy of inexpensive relative humidity instruments is ±5% to 20%.

Relative humidity is often used along with dry-bulb temperature and barometric pressure to define a psychrometric statepoint.

Relative humidity is an important and very useful property for determining the equilibrium moisture content (EMC) of hygroscopic materials. The equilibrium moisture content of materials is almost solely dependent on relative humidity and is largely unaffected by temperature, as shown in Figure 13-1

Relative Humidity and the Psychrometric Chart

The upward-curving left boundary of the psychrometric chart (Figure 13-2) is the *water vapour saturation curve,* which is the locus of points representing saturated *water vapour*, i.e., the p_{WVS} vs. t plot for H_2O. The saturation curve and 100% RH curve are the same. The 0% relative humidity line is the horizontal 0.000 humidity ratio line at the bottom of most psychrometric charts. Other relative humidity lines are usually displayed at 10% intervals.

Other Definitions—Real Gas Behavior (i.e., Nonideal)

Scientists use the most fundamental definition: φ is the ratio of the mole fraction of the water vapour in the mixture, x_{WV}, to the mole fraction in a saturated mixture, x_{WVS}, at the same dry-bulb temperature and total pressure.

The following equation shows equivalencies:

$$\varphi = x_{wv}/x_{wvs} \equiv p_{wv\_actual}/p_{wvs\_actual} \equiv f(p,\, t_{dp}) \times e(t_{dp})/(f(p, t_{db}) \times e(t_{db})$$
$$\text{with the constraint } |p$$

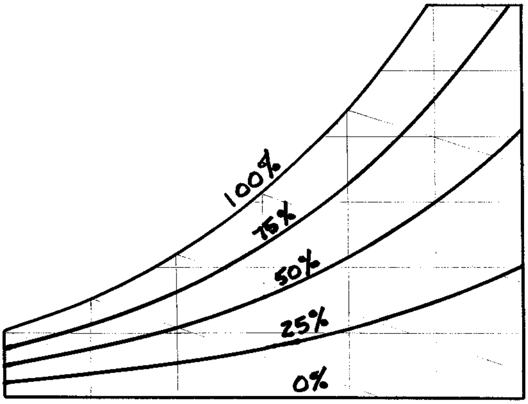


Figure 13-2—Relative humidity isolines.

where

φ = decimal relative humidity
x_{wv} = mole fractions of water vapour in moist air
x_{wvs} = mole fraction of saturated water vapour in moist air
p_{wv_actual} = actual partial pressure of water vapor in moist air
p_{wvs_actual} = actual partial pressure of saturated water vapor in moist air
f = is the water vapour pressure enhancement factor (see pp. 299)
p = total (barometric) pressure
t_{dp} = dew-point temperature
t_{db} = dry-bulb temperature
e = saturation pressure of pure H_2O

For a mixture of ideal gases $x_{WV} = p_{WV}/p_{BAR}$ and $x_{WVS} = p_{WVS}/p_{BAR}$ and for an ideal gas it follows that $\varphi = p_{WV}/p_{WVS}$.

The water vapour pressure enhancement factor for ideal H_2O vapour is unity and $\varphi = e/es$.

Incorrect Definitions

The most common definition of relative humidity is *not exactly* correct. It is used in *Webster's Collegiate Dictionary*, which defines relative humidity as the ratio of the amount (the mass) of *water vapour* present in the air to the greatest possible amount at the same temperature. If the words *and same volume* are added at the end, then this definition would be complete. The problem with the incorrect definition is that some practitioners convert their understanding of it to: RH = m_{WV}/m_{WVS}, which is not the formula for relative humidity. The ratio m_{WV}/m_{WVS} is the formula

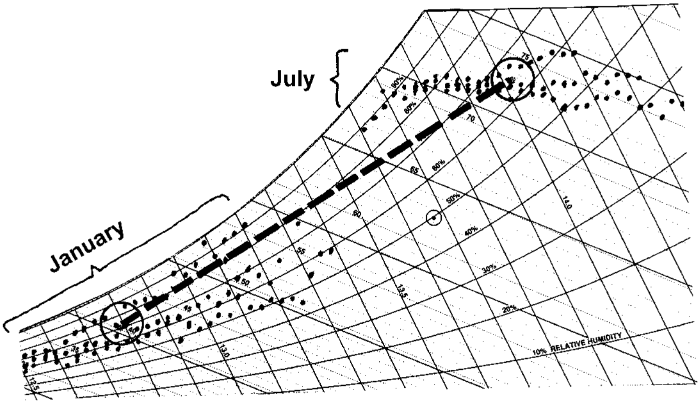


Figure 13-3—Outside air relative humidity is higher in winter.

for the (practically obsolete) psychrometric term *degree of saturation* (also called *percentage humidity*).

The *not exactly* correct definition is frequently used by TV weather reporters and others to give the layperson a rough grasp of the term *relative humidity.* The task of helping the layperson grasp the significance of the term *relative humidity* is difficult because definitions involving *water vapour* pressures, densities, or specific volumes have little meaning for the layperson.

A correct definition of relative humidity for the layperson is: the ratio, expressed in percent, between the actual mass of water vapour present in a given volume, space or parcel and the maximum mass of water vapour in that same volume, space or parcel at the same dry-bulb temperature.

From an intuitive standpoint, relative humidity can be a misleading indicator of the mass of *water vapour* in a given volume, space, or parcel of air because (1) temperature alone defines the *water vapour* saturation curve and (2) the maximum mass of *water vapour* in a given volume approximately doubles for each 10°C rise in temperature. Therefore, a volume saturated with *water vapour* at a warm temperature contains significantly more *water vapour* than an equal volume saturated with *water vapour* at a cold temperature. This leads to confusion for the layperson because winter RH is often higher than summer RH, yet it is a common perception that high humidity is a summer problem and low humidity a winter problem. For example, in the eastern United States, the noon relative humidity in winter is often in the 80% to 100% range, and in summer it is often in the 40% to 60% range; yet the mass of *water vapour* associated with the dry air is much higher in the summer, as shown in Figure 13-3. At one time there was some truth in the association between winter and low

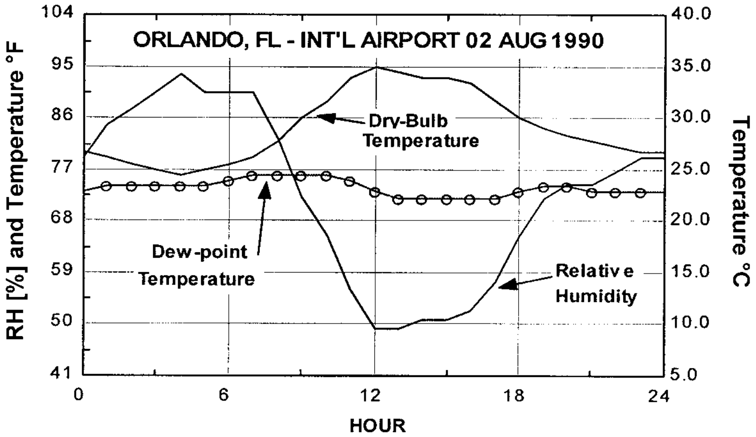


Figure 13-4—Twenty-four-hour relative humidity vs. dew-point temperature.

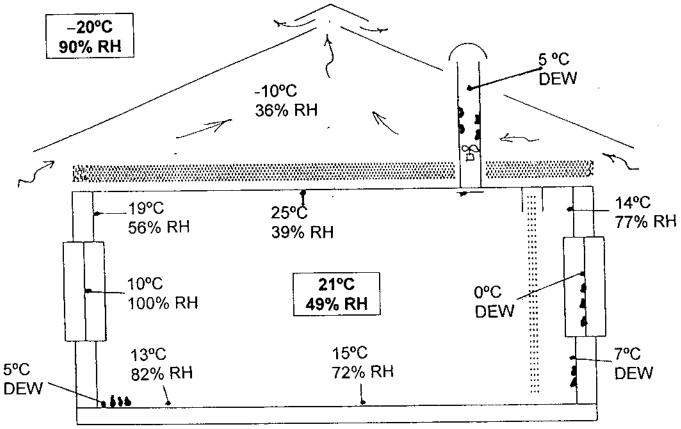
relative humidity, but it applied to the relative humidity inside leaky homes that had high infiltration rates of dry outside air.

Dew-point temperature is perhaps a more intuitive description of *water-vapour* content. Figure 13-4 shows that relative humidity in the eastern United States on a summer day varies from near 100% at 5 a.m. to 50% at 2 p.m., while dew-point temperature tends to be nearly constant throughout the 24-hour period unless rain occurs or a weather front moves through.

The dew-point temperature in a room is uniform throughout the room. On the other hand, the relative humidity varies throughout the room depending on the temperature of the air or the surface location at which the relative humidity is measured. Many texts caution that mold proliferates at relative humidities above 70%. Figure 13-5 shows that relative humidity at the center of the room may be 50%, but the relative humidity in the carpet pad on a cold floor in the same room may be 90% or higher. This demonstrates that stating a relative humidity value without stating or implying the dry-bulb temperature is meaningless. If an interior space has a constant dry-bulb temperature, then charting of relative humidity over time is a means of determining if the HVAC system and its controls are correctly managing the removal or addition of *water vapour* (but only when t_{DB} is constant!).

WHAT IS
THE RELATIVE HUMIDITY IN THIS ROOM?

Dew-point Temperature is 10°C At All Locations In The Room



ROOM$_{CTR}$: 21°CDB, 10°CDP, 49% RH, p_{wv} 1228 Pa (0.3626 "H$_{G}$), W 0.00766 kg$_{WV}$ / kg$_{DA}$
OUTSIDE: -20°CDB, -21°CDP, 90% RH, p_{wv} 93 Pa (0.0275 "H$_{G}$), W 0.00058 kg$_{WV}$ / kg$_{DA}$

Figure 13-5—Inside relative humidities in cold weather.

TRIVIA

Hundreds of early hygrometers are described in the literature of early meteorological instruments beginning with the inventions of Nicholas Cusanus in 1450 and Leonardo da Vinci in 1500. Contrast this with the fact that there was no term related to today's relative humidity until about 1803. The credit for the term RH belongs to the collective efforts of Saussure (1783), Jean-Andre Deluc (1791), Ludwig Wilhelm Gilbert (1803), John Dalton (1802), Joseph Louis Gay Lussac (~1804), and Jean-Baptiste Biot (~1804).

Noi viviamo sommersi nel fondo di un pelago d'aria elementare,
la quale per esperienze indubitate si sa che pesa.
(We live submerged at the bottom of an ocean of elementary air,
which is known by incontestable experiments to have weight.)
—Evangélista Torricelli, 1649

14 Barometric Pressure

BAROMETRIC PRESSURE
Symbol: p_{BAR} with Units of Pa

Definition

Barometric pressure is the force per unit area exerted by the weight of a column of air above a specific location. Weight is the force exerted by gravity on unit mass (in this case, the air mass). In psychrometrics the term *barometric pressure* is synonymous with *atmospheric* (or *total*) *pressure.* The descriptor *barometric pressure* is used in this text because of its wide use in meteorology and to avoid confusion with the term *total pressure* used in the fan and air moving industries.

Measurement and Use

Barometric pressure is measured with a barometer (often a mercury barometer).

Barometric pressure is one of three psychrometric properties required to fix or determine a psychrometric statepoint. Dry-bulb temperature is usually the second property. A humidity property such as dew-point temperature, relative humidity, or wet-bulb temperature is normally the third property used to determine a psychrometric statepoint.

If barometric pressure is not known for a location but altitude above or below sea level is known, then barometric pressure for the *standard atmosphere* may be calculated using the following International Civil Aviation Organization (ICAO) equation originally developed in 1925:

$$p_{BAR} = 101.325\,(1 - 2.25577 \cdot 10^{-5} \cdot Z)^{5.256}$$

where

p_{BAR} = yearly mean local barometric pressure at location with units of kPa

Z = altitude (elevation) above (+) or below (–) mean sea level in m

Appendix I, "ICAO Equation Relating Barometric Pressure and Altitude," describes the *standard atmosphere* and includes the derivation of this equation, the constants used, and the underlying assumptions. The concept of a *standard atmosphere* originated in the field of aeronautics in the calibration of altimeters and in calculating the performance of aircraft at different altitudes.

The equation is limited to the troposphere and to an average moist air lapse rate of 6.5°C per kilometre (temperature decreases 6.5°C for each kilometre increase in altitude). ICAO defines the upper limit of the troposphere as –56.5°C and *standard atmosphere* temperature at sea level as 15°C. At about 40° north latitude, the lapse rate and sea level temperature fix the tropopause at 11 km. The ICAO equation should not be used for unusual psychrometric process applications. For these applications, use the best available estimate of process barometric pressure, the actual temperature, and a humidity property.

Psychrometric calculations require barometric pressure at the location altitude. If a U.S. Weather Bureau barometric pressure reading is used, it *may* have to be corrected to the location altitude depending on whether or not the reported value has or has not been adjusted to sea level altitude. The U.S. Weather Bureau makes three modifications to a mercury barometer reading: (1) An adjustment to modify the instrument reading to sea level conditions at 45° latitude (this modification can be significant depending on altitude. It accounts for the altitude of the barometer and for the temperature of the assumed column of air down to sea level and is required so that the U.S. Weather Bureau can graph the horizontal pressure gradients (isobars of pressure) across an area of the world in order to indicate the strength and direction of winds and to help forecast weather); (2) A thermal expansion correction, specific to each barometer, to adjust the reading to the standard temperature reference of 0°C (this is a small adjustment that should be made to all barometer readings); (3) An insignificant latitude adjustment (which may safely be ignored in psychrometric calculations) to account for the difference in gravitational acceleration between the location of the measuring barometer and the gravitational acceleration at 45° latitude.

Barometric Pressure and the Psychrometric Chart

The barometric pressure and/or altitude should be stated on any psychrometric chart. ASHRAE publishes psychrometric charts for sea level pressure (101.325 kPa), 750 m altitude (92.66 kPa), 1500 m altitude (84.54 kPa), and 2250 m altitude (77.04 kPa). The better psychrometric software programs perform calculations and display a psychrometric chart based on the barometric pressure input by the user. Optionally, the user can input altitude above or below sea level and the program will compute barometric pressure using the ICAO *standard atmosphere* equation.

Barometric pressure is sometimes not presented as a basic psychrometric property, probably because the majority of psychrometric calculations throughout the world are for applications at or near sea level with a standard barometric pressure of 101.325 kPa. If barometric pressure is not stated, standard sea level barometric pressure of 101.325 kPa is usually implied. The record highest and lowest barometric pressures *at sea level* are 108.38 kPa and 87.00 kPa. These record variations from standard sea level barometric pressure are small compared with the change in barometric pressure due to altitude, which ranges from 142.95 kPa at –3000 m to 22.63 kPa at +11000 m. Changes in barometric pressure of less than 10 kPa have an insignificant effect on psychrometric calculations.

Psychrometric processes are traditionally assumed to be at constant pressure throughout the process. This is typical of most air-conditioning calculations. If pressure varies significantly in the process, the calculations can be modified to reflect this. The assumption of constant barometric pressure for psychrometric calculations is based on the fact that a psychrometric process has an insignificant effect on the total mass of air in the surroundings and, therefore, the barometric pressure is not changed because of a local psychrometric process such as the conditioning of building air for the comfort of the occupants.

In accordance with Dalton's law, the barometric (total) pressure is the sum of the pressures of the component gases *dry air* and *water vapour*:

$$p_{BAR} = p_{DA} + p_{WV}$$

With the barometric pressure fixed, it follows that addition of *water vapour* to the air mass with a commensurate increase in the *water vapour* pressure results in a decrease in the pressure of the *dry air* gases. Therefore, in a psychrometric process involving the addition of *water vapour* to the air, the pressure of the *water vapour* increases and with a constant barometric pressure it follows that the pressure of the dry air gas components decreases by the amount of the increase in the pressure of the water vapour.

Table 14-1 lists the altitude of the plotted curves and the *standard atmosphere* calculated barometric pressure at that altitude. Figure 14-1 shows the effect of changes in barometric pressure on the location of the saturation curve on a psychrometric chart. Higher barometric pressures move the saturation curve to the right and lower pressures move the curve to the left.

Table 14-1—Barometric Pressure at Altitude for the Standard Atmosphere

Z in km	–3	–1.5	Sea Level	+1.5	+3	+7	+11
P_{BAR} in kPa	142.95	120.69	101.325	84.56	70.11	41.06	22.63

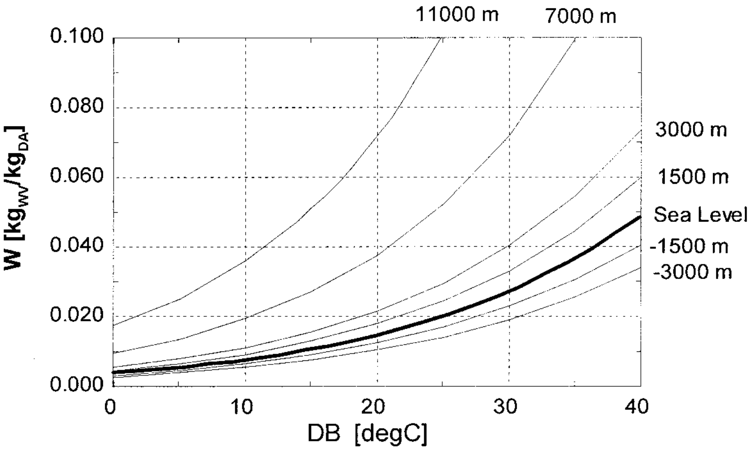


Figure 14-1—Psychrometric chart saturation curve vs. altitude.

People ask for a "reality" definition of enthalpy on the assumption that this mathematically defined quantity describes some reality. This leads to all sorts of mental gymnastics creating learning obstacles and rampant linguistic confusion. Enthalpy is nothing other than the convenient sum of two components of the energy balance equation: internal energy, U, and flow-work energy, p · V.

15 Specific Enthalpy

SPECIFIC ENTHALPY
Symbol: *h* with Units of kJ/kg$_{DA}$
(Note: kg of Dry Air Component)

Definition

Specific enthalpy (i.e., enthalpy per unit mass) is the mathematical sum of two energy components associated with a substance. These energies are the specific internal molecular energy (u) and specific flow work energy ($p \cdot v$). Specific enthalpy is defined by humankind as

$$h = u + p \cdot v$$

The product ($p \cdot v$) is called *flow work energy* or *the energy of intrusion* (or *extrusion*). The properties u and $p \cdot v$ occurred together so often in energy balance equations for steady flow processes that our predecessors combined them for convenience into this new property.

Using the Gibbs-Dalton rule, the specific enthalpy of moist air is the sum of the specific enthalpies of the *dry air* component and the *water-vapour* component of air. The prefix *specific* is often dropped and it is usually obvious from the equations or the units whether or not the prefix is implied. Specific enthalpy is calculated using the following equation (see the end of this chapter for development of this equation):

$$\begin{array}{ccccc} \text{enthalpy} & & \text{enthalpy dry air} & & \text{enthalpy water vapor} \\ h & & t_{DB} \cdot 1.006 & & W \cdot (2499.86 + \quad t_{DB} \cdot 1.84) \\ \left\{\dfrac{\text{kJ}}{\text{kg}_{\text{DA}}}\right\} & = & \left\{\dfrac{^\circ\text{C} \cdot \text{kJ}}{\text{kg}_{\text{DA}} \cdot {}^\circ\text{C}}\right\} & + & \left\{\dfrac{\text{kg}_{\text{WV}}}{\text{kg}_{\text{DA}}}\right\}\left[\left\{\dfrac{\text{kJ}}{\text{kg}_{\text{WV}}}\right\} + \left\{\dfrac{^\circ\text{C} \cdot \text{kJ}}{\text{kg}_{\text{DA}} \cdot {}^\circ\text{C}}\right\}\right] \end{array}$$

Measurement and Use

Enthalpy cannot be directly measured; however, enthalpy differences between the initial and the final statepoints of a process can be measured. Accurate values of enthalpy have been determined in research laboratories by measuring the changes of properties during a carefully controlled process and, from these properties, enthalpies relative to an arbitrary reference statepoint (0°C for moist-air psychrometrics) are calculated. The value of enthalpy at the arbitrary reference point is usually designated as having zero enthalpy. The resultant enthalpy values are published in the form of tables, graphs, and curve fit equations.

Specific enthalpy is a most important property in psychrometric calculations. With no shaft work (and no changes in potential or kinetic energies), the required heat transfer in a steady flow constant pressure process is the product of the mass flow rate of air and the difference in the enthalpy of the entering and leaving air, i.e., $q = m_{DA} \cdot \Delta h$. Thus, the product of the mass flow of the air and the enthalpy difference gives the heat added in a process or the heat removed (negative heat added) in a process (e.g., air passing through a cooling and dehumidifying coil).

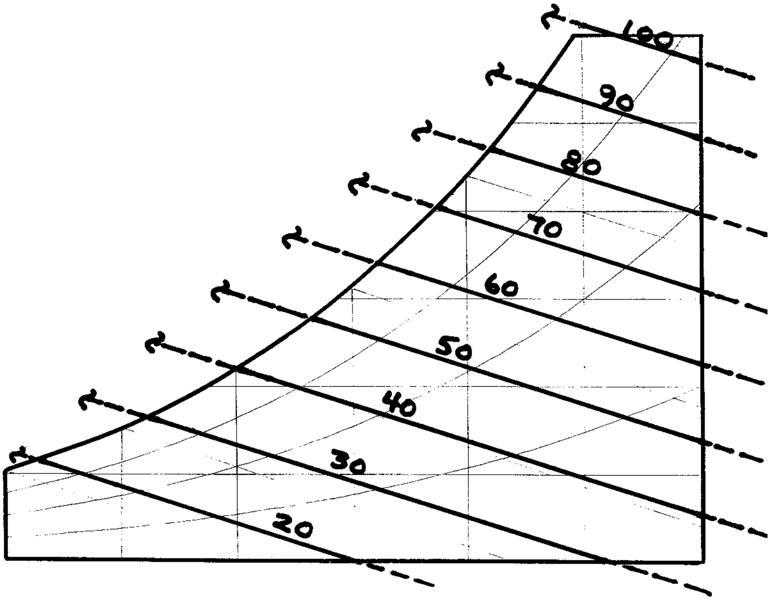


Figure 15-1—Specific enthalpy isolines (h in units of kJ/kg_{DA}).

Enthalpy and the Psychrometric Chart

Enthalpy isolines (Figure 15-1) on ASHRAE (since 1963) and Mollier psychrometric charts are straight, evenly spaced, and parallel to each other. These charts use an oblique enthalpy grid or scale as one plotting coordinate with horizontal humidity ratio as the other plotting coordinate (the ordinate). The oblique enthalpy coordinate was developed by Richard Mollier of Germany in 1923. It was introduced to North America in a reconfigured form by Ferdinand Keppler in 1934. The Mollier enthalpy coordinate improved the accuracy of graphically solving process calculations and resulted in a chart appearance closely resembling the popular 1904 Willis H. Carrier chart, which used vertical dry-bulb temperature lines as the second plotting coordinate (the abscissa).

Few psychrometric charts display enthalpy isolines through the body of the chart because enthalpy isolines are nearly parallel to wet-bulb temperature isolines. Some charts present edge scales of enthalpy so that a straight edge may be used to determine the enthalpy of a statepoint by aligning equal values of enthalpy at each edge with the statepoint within the body of the chart. Other charts present values of enthalpy at the saturation curve and show enthalpy deviation curves within the body of the chart.

The value of enthalpy is strongly, but not totally, dependent on wet-bulb temperature. This can be observed by comparing the slope of enthalpy and wet-bulb temperature lines on a psychrometric chart.

Other Facts and Help in Understanding and Applying Enthalpy

Specific enthalpy indicates that enthalpy is expressed as energy per unit mass. Specific enthalpy in psychrometrics is different than the conventional definition of specific enthalpy in most other fields of science. In psychrometrics, specific enthalpy is the enthalpy of the moist air substance divided by the *mass of the dry air component* (kJ/kg_{DA}). In most other fields, specific enthalpy is the enthalpy divided by the *total mass of all of the components* (kJ/kg). On rare occasions, to make a clear distinction, the term *psychrometric specific enthalpy* can be used.

Enthalpy, in one sense, could be called *relative enthalpy* or *relative specific enthalpy* since the zero enthalpy value has an *arbitrary reference datum* from which the enthalpy is determined. The word *relative* here means the energy relative to a reference datum. The use of a reference datum for enthalpy is similar to the use of sea level, or some other arbitrary elevation, to determine the potential energy of a body. The psychrometric reference datum or zero enthalpy value for dry air is 0°C. Note that the inch-pound system enthalpy datum for dry air is 0°F (–17.78°C). This makes it difficult, but not impossible, to convert statepoint enthalpy values between the two unit systems. Chapter 24, “Frequently Asked Questions,” covers this conversion, which has little if any practical application. Process calculations should be completed using either SI or I-P units and then, if dual units are required, the final results can be converted to the alternate system of units.

For moist air psychrometrics, the reference datum or zero enthalpy for H_2O, in both SI and I-P unit systems, is liquid water at the triple ice point of water (0.00°C or 32.00°F) and sea-level atmospheric pressure. This reference datum differs slightly from the IAPWS triple point datum of 0.01°C and saturation pressure. Enthalpy values are shown in tables of the thermodynamic properties of water, *water vapour*, and ice, e.g., the enthalpy of saturated *water vapour* at 0°C and 101.325 kPa is 2499.86 kJ/kg.

The result of transferring heat into a parcel of air at constant barometric pressure is an increase in enthalpy equal to the sum of the increase in internal energy and the increase in flow work energy associated with the volumetric expansion of the parcel.

Chris Glosser, in a thermodynamics lecture posted on the web, states: “Enthalpy is basically the energy required to create a system out of nothing in a constant pressure environment.”

Derivation of Enthalpy Equations for Moist Air and Psychrometrics

The enthalpy of moist air, using the Gibbs-Dalton law for a mixture of two non-reacting ideal gases (dry air and *water vapour*), is the sum of H_{DA}, the enthalpy of the dry air component, and H_{WV}, the enthalpy of the *water-vapour* component, expressed by Equation 15-1. Fortunately, the specific heat capacities of both dry air and *water vapour* are nearly constant over the range of temperatures encountered in most psychrometric applications and Equation 15-1 can be expressed in the form of Equation 15-2a.

By using 0°C as the t_{REF} value, Equation 15-2a converts to the more familiar Equation 15-2b. By modeling *water vapour* as an ideal gas, the enthalpy of *water vapour* is a function of temperature only. Thus, the enthalpy of superheated *water vapour* at a given temperature is identical to the enthalpy of saturated *water vapour* at the same temperature. The equivalency of these two enthalpies can be confirmed by examining the low-pressure, low-temperature range of a TS property plot for steam or by examining steam tables at low pressure and temperature.

$$H = H_{DA} + H_{WV} = H_{DA} + m_{WV} \cdot h_{WV} \qquad (15\text{-}1)$$

substituting $m_{WV} = W \cdot m_{DA}$ and dividing by m_{DA}

$$h = h_{DA} + W \cdot h_{WV}$$

$$h = C_{P-DA} \cdot (t_{DB} - t_{REF}) + W \cdot [h_{G\,WV\,0°C} + C_{P-WV} \cdot (t_{DB} - t_{REF})] \qquad (15\text{-}2a)$$

$$h = C_{P-DA} \cdot t_{DB} + W \cdot (h_{G\,WV\,0°C} + C_{P-WV} \cdot t_{DB}) \qquad (15\text{-}2b)$$

where

C_{P-DA} = specific heat capacity of dry air, 1.006 kJ/(kg_{DA}·°C)
C_{P-WV} = specific heat capacity of *water vapour*, 1.84 kJ/(kg_{WV}·°C)
DA = dry air subscript
H = total enthalpy of a given mass of moist air with units of kJ
h = specific enthalpy of the mixture of dry air and *water vapour*, units kJ/kg_{DA}
H_{DA} = enthalpy of the mass of dry air with units of kJ
h_{DA} = specific enthalpy of the dry air with units of kJ/kg_{DA}
$h_{G\,WV\,0°C}$ = enthalpy of *water vapour* at 0°C and 101.325 kPa, value of 2499.86 kJ/kg_{WV}
H_{WV} = enthalpy of the mass of *water vapour* with units of kJ
h_{WV} = specific enthalpy of *water vapour* with units of kJ/kg_{DA}
m_{DA} = mass of dry air with units of kg_{DA}
m_{WV} = mass of *water vapour* with units of kg_{WV}
t_{DB} = dry-bulb temperature in °C
t_{REF} = 0°C for both dry air and *water vapour*; 0.00°C for liquid water
W = humidity ratio with units of $\text{kg}_{\text{WV}}/\text{kg}_{\text{DA}}$
WV = water vapour subscript

Enthalpy calculated using Equation 15-2b is based on average specific heat capacity values for both the dry air and the *water vapour*. The specific heat capacity of both components varies slightly with temperature. For a minute improvement in accuracy, some psychrometric property calculations utilize equations incorporating the specific heat capacity of dry air and/or *water vapour* as a function of temperature.

Substituting Table 4-1 values into Equation 15-2b gives:

$$h = 1.006 \cdot t_{db} + W \cdot 2499.86 + W \cdot 1.84 \cdot t_{db}$$

The $W \cdot 1.84 \cdot t_{db}$ part of the last equation contributes less than one per cent to the value of h. Therefore small changes in the value of C_{p_wv} will have insignificant effect.

Readers may encounter “ideal gas model” enthalpy equation using different values for the term (2499.96 + 1.84 · t_{db}). In many cases the different values are the result of curve fitting “Real Moist Air” property values for a specific temperature range, pressure, and humidity parameter.

Equal volumes of all gases (or mixtures of gases) under similar conditions of pressure and temperature contain the same number of molecules.

16 Specific Volume

SPECIFIC VOLUME
Symbol: v with Units of m^3/kg_{DA}
(Note: kg of Dry Air Component)

Definition

Specific volume in psychrometrics is the volume per unit mass of the *dry air* component. In SI units, the specific volume is expressed as cubic metres per kilogram of *dry air*, (m^3/kg_{DA}). Using (1) Dalton's model in which the volume is shared by the *dry air* gas and the *water vapour* and (2) the psychrometric convention of expressing specific volume properties per unit mass of the *dry air* component, it follows that $v_{DA} = v_{WV} = v_{AIR}$. In psychrometrics, the three properties are equal and, thus, the subscripts are rarely used.

Measurement and Use

Specific volume can be measured by experiment in a research laboratory. It is not easily measured in the field and, consequently, it is calculated from other properties. Specific volume (m^3/kg_{DA}) is used in process calculations in converting between moist air volumetric flow (m^3/s) and the mass flow (kg_{DA}/s) of the *dry air* component.

Specific Volume and the Psychrometric Chart

The inclination of specific volume isolines on a psychrometric chart (Figure 16-1) is steeper than that of enthalpy or wet-bulb temperature isolines. The volume lines are usually drawn as straight lines even though they actually have a visibly undetectable slight curvature. Specific volume isolines are not parallel to one another although in a cursory examination they appear to be parallel.

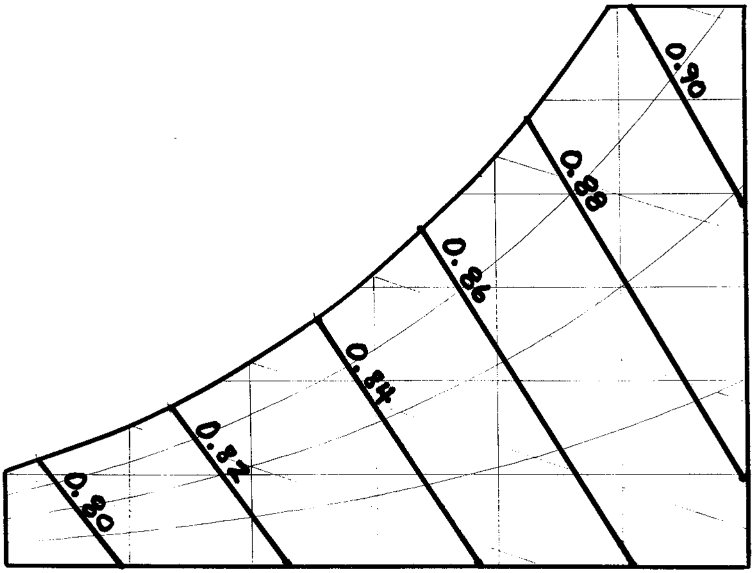


Figure 16-1—Specific volume isolines (v in units of m^3/kg_{DA}).

Other Facts Relating to Specific Volume

Specific volume in psychrometrics is different than the conventional definition of specific volume in other fields of science. In psychrometrics, specific volume is the volume of the moist air substance divided by the *mass of the dry air component* (m^3/kg_{DA}). In other fields, specific volume is the volume divided by the *total mass of all of the components* contained within the volume (m^3/kg_{AIR}). On rare occasions, to make a distinction, this term can be called *psychrometric specific volume*.

Using Dalton's model for mixtures of perfect gases, the *dry air* component, the *water-vapour* component, and the moist air mixture of the two gases occupy (share) the same volume—i.e., the *dry air* volume, the *water vapour* volume, and the mixture (moist air) volume are the same. In equation form: $V_{DA} = V_{WV} = V_{MIX}$. Following the psychrometric convention of referencing all specific properties to the unit mass of *dry air*, it follows that

$$V_{DA}/m_{DA} = V_{WV}/m_{DA} = V_{MIX}/m_{DA} = v_{DA} = v_{WV} = v_{MIX}.$$

The *dry air*, *water vapour*, and the mixture also have the same dry-bulb temperature.

In order to avoid possible confusion and the errors resulting therefrom, it is best to avoid use of the property *density* (ρ) in psychrometrics—

first because it is somewhat redundant, and second because density is normally thought of as the *total mass* per unit volume and is the reciprocal of conventionally defined specific volume (see previous paragraph). Although it is possible in psychrometrics to calculate the density of moist air ($\rho = [1+W]/v$), the total mass of moist air is of little interest because the important equations in psychrometrics are based on unit mass of *dry air*. A minor reason for avoiding the term *density* in psychrometrics is that its symbol, ρ, can easily be confused with the pressure symbol p.

The air property density is of interest to golfers and baseball home run hitters. A ball travels a greater distance in less dense, warmer air than in colder air. It is not intuitive to laypersons that at a given temperature air density decreases with increasing humidity and, thus, higher humidity helps golfers and home run hitters.

Specific Volume and the Ideal Gas Equation of State

The equation for specific volume is a simple example of applying the ideal gas equation of state as shown below:

$$p_{DA} \cdot v = R_{DA} \cdot T \qquad \text{and} \qquad p_{DA} = p_{BAR} - p_{WV}$$

$$v = \frac{R_{DA} \cdot T}{p_{BAR} - p_{WV}} = \frac{287.042\ \text{J}/(kg \cdot K) \cdot (t_{CELSIUS} + 273.15)}{(p_{BAR} - p_{WV})}$$

Also, by substituting for p_{WV} using the relationship between W, p_{BAR}, and p_{WV}, the following can be derived:

$$v = \frac{287.042 \cdot (t_C + 273.15) \cdot (1 + 1.608 \cdot W)}{p_{BAR}} \quad [\text{m}^3/\text{kg}_{\text{DA}}]$$

> *The benefits and problems of moisture/humidity are many;*
> *they can be complex, yet they often can be unknowingly simple,*
> *when properly described and interpreted .*
> *—F.C. Quinn*

17 Humidity Ratio

HUMIDITY RATIO
Symbol: W with Units of kg_{WV}/kg_{DA}
(Some Use g_{WV}/kg_{DA})

Definition

Humidity ratio, a defined property, is the ratio of the mass of *water vapour* to the mass of *dry air* in a parcel, sample, or volume of moist air. *Humidity ratio* is represented by the following equation:

$$W = m_{WV}/m_{DA} \text{ with units of } \mathrm{kg_{WV}/kg_{DA}}$$

(see end of chapter for other units)

where

W = humidity ratio
m_{WV} = mass of *water vapour* in the space or sample of moist air
m_{DA} = mass of *dry air* in the space or sample of moist air

Measurement and Use

Humidity ratio is difficult, but not impossible, to measure. Instead, low-cost portable instruments are used to measure t_{DB}, t_{DP}, t_{WB}, or RH and, from any two of these values and p_{BAR} or *altitude*, humidity ratio and all other psychrometric properties are calculated. The National Institute of Standards and Technology (NIST) utilizes a gravimetric hygrometer that measures the mass of *water vapour* and the mass of *dry air* and from these measurements calculates *humidity ratio*. The gravimetric hygrometer is used for very accurate calibration of other types of hygrometers. It is large, nonportable, and expensive. Under low relative humidity conditions, a single calibration may require more than a day!

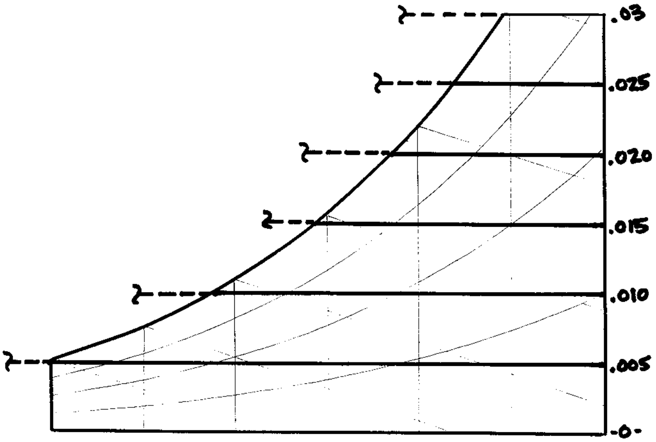


Figure 17-1—Humidity ratio isolines
(W in units of kg_{WV}/kg_{DA} [lb_{WV}/lb_{DA}]).

Humidity ratio provides a simple, effective, and most convenient means of accounting for the mass of *water vapour* in a psychrometric process because it allows the tracking of *water vapour*, a variable quantity in many psychrometric processes, by relating it to the nonvarying mass of *dry air* in a psychrometric process.

Humidity ratio, *water vapour* pressure, and dew-point temperature are mutually dependent properties (all represented by horizontal lines on a psychrometric chart) and only one of the three (along with barometric pressure and one other property such as dry-bulb temperature) may be used to determine the statepoint.

Humidity ratio is a function of *water vapour* pressure and barometric pressure as shown by the equations presented later in this chapter.

Water vapour pressure is a function of dew-point temperature; therefore, considering the previous sentence, humidity ratio is a function of dew-point temperature and barometric pressure. (Latent heat transfer in a process is a function of humidity ratio and the latent heat of vaporization, both of which are functions of dew-point temperature.)

Humidity Ratio and the Psychrometric Chart

Humidity ratio is one of the two psychrometric chart coordinates (the ordinate). *Humidity ratio* isolines on the psychrometric chart (see Figure 17-1) are horizontal, evenly spaced, and extend from the right side of the chart left to the *water vapour* saturation curve (the saturation curve is also the 100% RH isoline). Since humidity ratio is a plotting coordinate, the humidity ratio isolines can be extended to the left of the saturation curve into the fog region of the psychrometric chart. Chapter 24, “Frequently Asked Questions,” has additional information on the fog region. The scale for *humidity ratio* is the uniform vertical scale usually shown at the right side of the psychrometric chart. The other plotting coordinate for the ASHRAE chart (since 1963) is specific enthalpy using an oblique coordinate.

Other Facts Relating to Humidity Ratio

Humidity ratio has also been called *moisture content*, *mixing ratio, humidity x* (Europe), and *water vapour to dry air mixing ratio*. Although *water vapour to dry air mixing ratio* is an elegant definition because it is self-defining, *humidity ratio* will be used throughout this text because it is widely used and accepted in psychrometrics and meteorology.

Before the science of psychrometrics developed, *mixing ratio* was used by chemists and chemical engineers in tracking substances in chemical processes. It conveniently allowed the practitioner or scientist to track a variable quantity of one substance (such as a condensable vapour) as the ratio of a nonvarying quantity of a second substance (such as a noncondensable “dry” gas—meaning that the gas is noncondensable in the range of normal temperatures and pressures). The term *humidity ratio* probably developed as practitioners in the field of psychrometrics shortened and abbreviated *water vapour to dry air mixing ratio*.

Humidity ratio ($W = m_{WV}/m_{DA}$) is sometimes incorrectly called *specific humidity* or *absolute humidity*. *Webster’s International Dictionary*, meteorological texts, and *ASHRAE Terminology of HVAC&R* definitions for these two terms are different from the definition for *humidity ratio*. To avoid confusion, *specific humidity* or *absolute humidity* should not be used to describe the mass of *water vapour* associated with a unit mass of *dry air*.

Development of Humidity Ratio Equations

The relationship between W, p_{WV}, and RH is developed using the basic definition of W and the ideal gas equations of state for *dry air* and *water vapour*.

$$W = m_{WV}/m_{DA}$$

$$pV = mRT \qquad \text{and rearranging} \qquad m = \frac{pV}{RT}$$

$$m_{WV} = \frac{p_{WV}V_{WV}}{R_{WV}T_{WV}} \qquad m_{DA} = \frac{p_{DA}V_{DA}}{R_{DA}T_{DA}}$$

$$R_{DA} = 8314.472/28.966 = 287.042\ \text{J/(kg·K)}$$

$$R_{WV} = 8314.472/18.015268 = 461.524\ \text{J/(kg·K)}$$

Using Dalton's law, the *water vapour* and *dry air* occupy the same volume and are at the same temperature. Substituting for m_W and m_{DA} and eliminating the volume and temperature terms that cancel leads to:

$$W = \frac{R_{DA}}{R_{WV}} \cdot \frac{p_{WV}}{p_{DA}} \qquad W = \frac{287.042}{461.524} \cdot \frac{p_{WV}}{p_{DA}}$$

$$W = 0.621945 \cdot \frac{p_{WV}}{p_{DA}}$$

$$p_{TOTAL} = p_{BAR} = p_{WV} + p_{DA}$$

Therefore

$$p_{DA} = p_{BAR} - p_{WV}$$

and

$$W = \frac{0.621945 \cdot p_{WV}}{p_{BAR} - p_{WV}}$$

and using the RH equation: $100 \cdot p_{WV} = \mathrm{RH} \cdot p_{WVS}$

$$W = \frac{0.621945 \cdot \mathrm{RH} \cdot p_{WVSAT}/100}{p_{BAR} - \mathrm{RH} \cdot p_{WVSAT}/100}$$

where

p_{WV} = partial pressure of the *water-vapour* component of moist air mixture

p_{DA} = partial pressure of the *dry air* component of the moist air mixture

p_{BAR} = total pressure, i.e., atmospheric or barometric pressure

p_{WVSAT} = partial pressure of saturated *water vapour* at dry-bulb temperature

RH = relative humidity expressed in percent

Note that *humidity ratio* is a function of barometric pressure (or altitude) as is evidenced by the preceding equations. This contrasts with other humidity parameters including *dew-point temperature, water vapour partial pressure,* and *relative humidity,* which are independent of barometric pressure.

In SI units, *humidity ratio* is expressed in kilograms of *water vapour* per kilogram of *dry air* or in grams of *water vapour* per kilogram of *dry air*. In I-P units, *humidity ratio* is expressed in pounds of *water vapour* per pound of *dry air* or, alternatively, in grains of *water vapour* per pound of *dry air* (7000 grains = one pound). The *humidity ratio* in pounds of *water vapour* per pound of *dry air* and in kilograms of *water vapour* per kilogram of *dry air* is identical. The use of grams (SI) or grains (I-P) to measure the mass of *water vapour* results in *humidity ratios*, which are respectively 1000 times larger (SI) or 7000 times larger (I-P). Humans abhor negative and fractional numbers and have a proclivity for substituting, shortening, abbreviating, and deleting and occasionally creating a new property or unit using a contraction of their own name. In repetitive situations this is sometimes beneficial, but, in most cases, it creates an unnecessary obstacle to the student or occasional user. In today's world of software calculations, there is little justification for departing from SI standard units and dimensions.

At the earth's surface, the *humidity ratio* (in kilogram$_{\mathrm{WV}}$ per kilogram$_{\mathrm{DA}}$ or pound$_{\mathrm{WV}}$ per pound$_{\mathrm{DA}}$) varies from 0.0005 to 0.035 in the course of a year. In SI units of gram$_{\mathrm{WV}}$ per kilogram$_{\mathrm{DA}}$, these values become 0.5 to 35, which are easier to work with but in the strictest sense no longer represent a ratio but 1000 times a ratio. It is obvious that practitioners using *humidity ratio* in calculations must ensure balanced or consistent units.

Water vapour exercises a pressure of its own
with the maximum value solely dependent on temperature.
—Horace Bénédict de Saussure, 1783

18 Water Vapour Pressure

WATER VAPOUR PRESSURE
Symbol: p_{WV} with Units of Pa

Water vapour pressure is the pressure exerted by *water vapour* molecules whether in a container by themselves or sharing a container (or the atmosphere) with nitrogen, oxygen, or other gaseous molecules. It is the pressure indicated by a pressure-measuring instrument if *water vapour* alone occupies a closed container.

The simplest way to envision this measurement is to imagine a closed container void of any substance (at an absolute vacuum). The container is then partly filled with liquid water and/or ice without admitting air or other gases. A pressure-sensing probe is inserted above the level of the ice or water so that the pressure of the *water-vapour* (H_2O gas) molecules colliding with the wall of the container is obtained. A temperature-sensing probe measures the temperature of the H_2O in the container. Heat is then transferred to the H_2O in the container and simultaneous readings are made for p_{WV} and t. Henri Victor Regnault and others used this type of apparatus to measure the pressure of *water vapour* and showed that as the temperature of the H_2O in the container increased, the *water vapour* pressure above the water or ice also increased.

It is not possible to measure the pressure contribution of one constituent gas in a mixture of gases and it follows that it is not possible to measure p_{WV}, p_{N2}, or p_{O2} in the air around us. The only pressure that our instruments will measure is the total (combined) pressure, which we call barometric pressure, p_{BAR}, of all the gases occupying the container or volume. Using Dalton's law, barometric (or total) pressure of air equals the sum of the pressure of the *water vapour* molecules and the pressure of the *dry air* (N_2, O_2, Ar, etc.) molecules as shown in this equation:

$$p_{BAR} = p_{WV} + p_{N2} + p_{O2} + p_{Ar} = p_{WV} + p_{DA}$$

Water vapour pressure can also be defined as the saturation pressure corresponding to the dew-point temperature of the *water vapour*. It is often referred to as *the partial pressure of the water vapour* or simply *partial pressure*. In this text, the term *water vapour pressure* is used exclusively.

Measurement and Use

In the 1800s, *water vapour* pressures versus temperatures-at-saturation were tabulated by Regnault from extensive measurements in his laboratories. For most of the 1900s, tables from the later research of Joseph H. Keenan and Fred G. Keys have been used. Tabulated values are suitable for manual calculations, but for software calculations, equations are preferred. There are numerous curve fit equations of the tabulated data. The most accurate equations for moist air psychrometrics are those from ASHRAE RP-1485 (2009) listed in Chapter 4, “Water Vapour.”

In psychrometrics, the value of *water vapour* pressure is calculated from other properties such as humidity ratio, dew-point temperature, or combinations of properties such as dry-bulb and wet-bulb temperatures or dry-bulb temperature and relative humidity.

Water vapour pressure differences constitute the driving force in the diffusion of *water vapour* molecules between ambient air and hygroscopic materials or between a surface of water and ambient air. *Water vapour* moves from a zone or region of higher *water vapour* pressure to a zone of lower *water vapour* pressure as shown by the following equation:

$$M_{\mathrm{H2O}} = \mu \cdot A \cdot \Theta \cdot (p_1 - p_2)/l$$

where

M_{H2O} = total mass of water vapour transmitted in units of nanograms

μ = average permeability in nanograms per second per square metre for one metre thickness and one pascal pressure difference, $\mathrm{ng \cdot m/(s \cdot m^2 \cdot Pa)}$

A = cross-sectional area of flow path in square metres, m^2

Θ = time during which flow occurs in seconds, s

$(p_1 - p_2)$ = the vapour pressure difference in pascals, Pa

l = length (thickness) of flow path in metres, m

In many calculations involving a material of known thickness, a permeance value representing μ/l is used in lieu of separate values for μ and l.

Water Vapour Pressure and the Psychrometric Chart

Water vapour pressure isolines on a traditional psychrometric chart (Figure 18-1) are horizontal lines extending from the saturation (100% RH) curve on the left to the right-hand boundary of the chart. *Water*

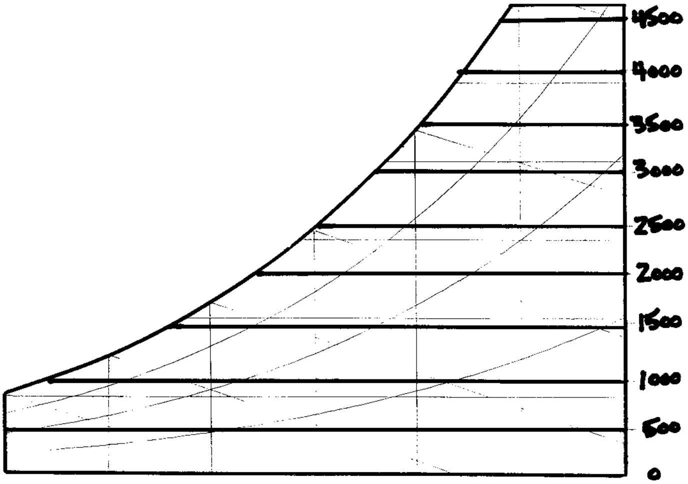


Figure 18-1—Water vapour pressure isolines (P_{WV} in units of Pa [in. Hg]).

vapour pressure isolines and dew-point temperature isolines are rarely plotted on the chart because they are both parallel to the humidity ratio lines and would add clutter, making the chart difficult to read. On some psychrometric charts, edge scales are used to display *water vapour* pressure and/or dew-point temperature. If the edge scales are not shown on a psychrometric chart, the *water vapour* pressure may be found (1) in a table, (2) from a calculation using dew-point temperature, or (3) from a calculation using dry-bulb and relative humidity.

Discussion

Water vapour pressure is not commonly used by the general public or TV weather reporters, but it is important in the calculation of *water vapour* diffusion. Dew-point temperature is a widely used humidity measure and is useful in understanding the concentration of *water vapour.*

Using the relationship between W, p_{BAR}, and p_{WV} derived in Chapter 17, "Humidity Ratio," and taking into consideration that p_{WV} for temperatures under 50°C is inconsequential when compared to p_{BAR}, it can be demonstrated that p_{WV} is essentially proportional to W for ordinary temperatures. Many equations used to calculate evaporation from swimming pools and hot tubs utilize W values in lieu of p_{WV} values with little or no loss in accuracy because water temperatures are under 50°C. The following equations show the W vs. p_{WV} relationship and provide the rationale for the accuracy at low temperatures.

$$W/0.621945 = p_{WV}/(p_{BAR} - p_{WV})$$

$$\mathrm{W}/0.621945 \cong p_{WV}/p_{BAR} \text{ when } p_{WV} \text{ is tiny in relation to } p_{BAR}.$$

Basic Psychrometric Processes

In teaching the thermodynamics of heating and ventilating, a thoughtful teacher will not allow students to use charts (or simplified equations) until they understand the fundamental thermodynamic principles upon which these charts are based. The practicing engineer, also, unless he wishes to become simply a handbook engineer, will not use charts until he is sure of the underlying principles. It is our opinion that the careless use of psychrometric charts has resulted in covering up many fundamental psychrometric principles.
—C.O. Mackey

19 Psychrometric Processes

A psychrometric process occurs when air at an *initial state* undergoes a transformation and ends up at a *final state*. The transformation involves the transfer *to or from the air* of (a) heat or work and/or (b) mass (H_2O). A process has an initial statepoint and a final statepoint. The process occurs between the two.

Table 19-1 lists *some* of the possible psychrometric processes. Theoretically, if process vectors originated from the initial statepoint in one-degree increments, 360 different process vectors could originate and fan out from the initial statepoint.

Table 19-1

Psychrometric Process	Comment
The Four Basic Processes	
Sensible Heating Only (Heat transfer into the air), Figure 19-1, process A	No change in water-vapour content
Sensible Cooling Only (Heat transfer from the air), Figure 19-1, process B	No change in water-vapour content
Humidification Only (Latent energy addition) (Latent heating), Figure 19-1, process C	No change in dry-bulb temperature
Dehumidification Only (Latent energy removal) (Latent cooling), Figure 19-1, process D	No change in dry-bulb temperature
Single Processes Involving Both Heat and Water Vapour Transformation	
Cooling And Dehumidification Coil (or an air washer with chilled spray water)	Curved process path line
Evaporative Cooling (Adiabatic) (Sensible Cooling +Humidification), Figure 19-2, process E	Constant wet-bulb temperature
Water Spray Processes, Figures 19-6 and 19-7	Many variations
Active Desiccant (Chemical or Sorbent) Dehumidification (Adiabatic) Figure 19-2, process F	Constant wet-bulb temperature (ideal)

Table 19-1 (Continued)

Single Processes Involving Both Heat and Water Vapour Transformation	
Mixing of Two Airstreams	Adiabatic (no heat transfer)
Room Effect (Changes to supply air due to sensible and latent gains in room)	Vector direction varies by season
Fan Heat including fan, motor, and drive (No change in water vapour)	Similar to sensible heating
Enthalpy Wheel	Vector ~ mixing process
Examples of Two or More Processes in Sequence	
(Ordinary) Face and Bypass* (F&BP of Mixed Air)	2 processes in sequence
Return Air Face and Bypass*	2 processes in sequence
Recuperative Runaround Precooling-Reheat with Cool and Dehumidification	3 processes in sequence
Desiccant Dehumidification Followed by Sensible Cooling	2 or 3 processes in sequence
Sensible Precooling Followed by Evaporative Cooling	2 processes in sequence
Sensible Heating Followed by Humidification	2 processes in sequence
Typical Air-Conditioning Cycle Example	A cycle of 4 processes in sequence

* *F&BP involves a face damper capable of modulating the airflow quantity through a device such as a cooling coil, a heating coil, or a rotary wheel device and a bypass damper that modulates the remainder of the airflow around the device. The dampers are usually interconnected so that as one closes the other opens. The first process involves the coil or device; the second process is the mixing of the air that passes through the coil (or device) with the portion that bypasses the coil (or device).*

Figure 19-1 illustrates the four basic processes that involve either a transfer of heat *only* or a transfer of mass (water vapour) *only*. Transfer of heat *into* air is a *sensible heating* process. Transfer of heat *from* air is a *sensible cooling* process. Transfer of water vapour *into* air is a *latent heating* (humidification) process. Transfer of water vapour *from* air is a *latent cooling* (dehumidification) process.

Figure 19-2 illustrates other psychrometric processes, paths, or vectors. The processes in the 0° to 90° and 180° to 270° quadrants have sensible heat and latent heat components with similar signs. The processes in the 90° to 180° and the 270° to 360° quadrants have sensible heat and latent heat components with opposite signs. The processes in the 90° to 180° and the 270° to 360° quadrants, which follow a wet-bulb line, are adiabatic (there is no heat transfer into the process from an external source). In the 90° to 180° process (active desiccant dehumidification, process F), the gain in sensible energy is offset by a loss in latent energy. In the 270° to 360° process (evaporative cooling, process E), the gain in latent energy is offset by a loss in sensible energy.

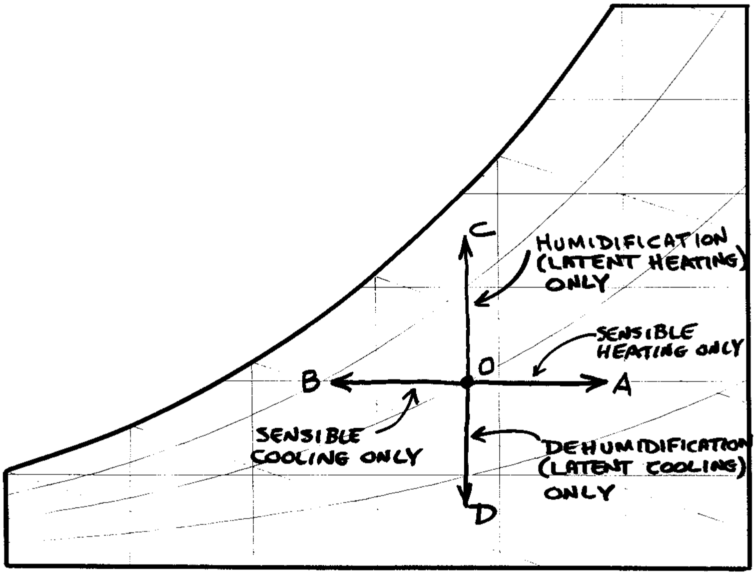


Figure 19-1—Four basic processes.

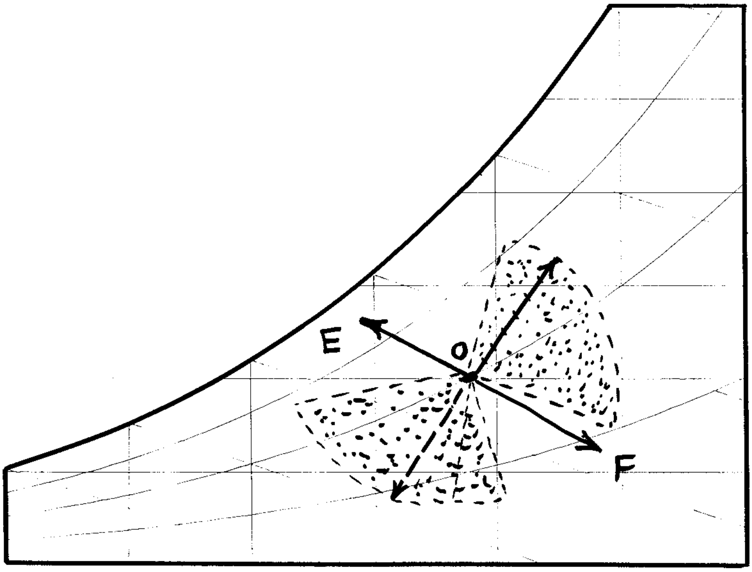


Figure 19-2—Other process vectors.

Not all of the Figure 19-2 process paths are possible. The practically obtainable *single* process vectors are represented by solid lines; the more difficult or theoretical single processes are shown as dashed vectors. The shaded areas of Figure 19-2 indicate that a sequence of processes is usually required (as opposed to a single process) in order to transform the air from the initial to the final statepoint.

Sequential processes involve a series or sequence of two or more processes in which the final state of air in the first process is the beginning state of the second process, etc. The possible combination of sequential processes is almost limitless. Seven of the more common sequential processes are shown in the last grouping in Table 19-1.

A cycle involves a sequence of processes in which the final state of the last process is also the initial state of the first process. Figure 19-3 illustrates a common psychrometric cycle with the following processes: mixing of two airstreams, cooling and dehumidification, fan heat, and room effect.

The Four Basic Processes

The psychrometric chart in Figure 19-1 illustrates the four basic processes, each beginning at statepoint "0."

The sensible heating *only* and sensible cooling *only* processes require the transfer of heat into or out of the air with no change in *water-vapour* content.

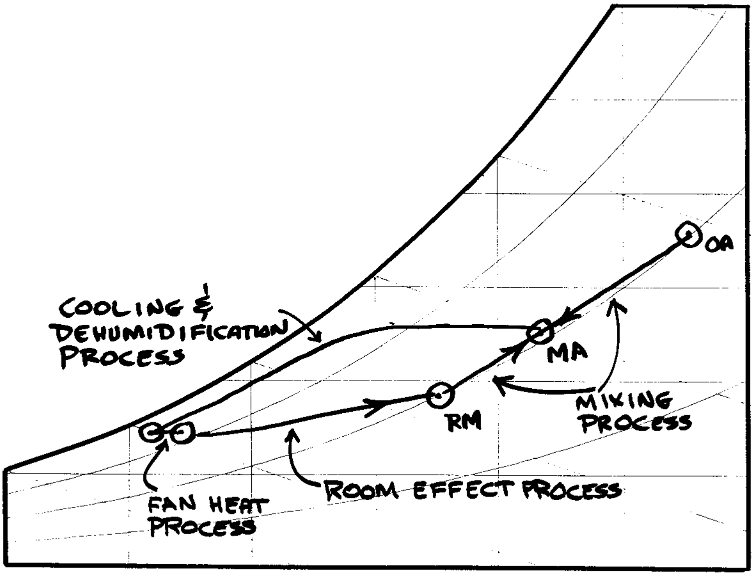


Figure 19-3—A four-process air-conditioning cycle.

These processes occur on a horizontal line of constant humidity ratio (and, therefore, constant dew-point temperature and constant *water vapour* pressure).

The humidification *only* and dehumidification *only* processes involve a change in the *water-vapour* content of the air but do not involve any sensible heat transfer. They proceed along a line of constant dry-bulb temperature.

Sensible Heating *Only* Process (Heat Addition Without Change in Water-Vapour Content). The transfer of heat into air using a steam coil, a hot water coil, a heat pipe, an air-to-air heat exchanger, a sensible-only rotary heat wheel, an electric heating coil, a furnace, or some other heat transfer device is an example of this process. On a psychrometric chart, the sensible heating process proceeds horizontally to the right along a line of constant humidity ratio. The final statepoint has higher dry-bulb temperature, higher wet-bulb temperature, and lower relative humidity. There is no change in dew-point temperature, *water vapour* pressure, or humidity ratio in this process, indicating no change in *water-vapour* content. Vector A in Figure 19-1 shows this process.

Sensible Cooling *Only* Process (Cooling Without Change in Water-Vapour Content). The transfer of heat from air using a chilled water or refrigerant cooling coil, an indirect evaporative cooler, a heat pipe, an air-to-air heat exchanger, or a sensible-only rotary heat wheel is an example of this process. On a psychrometric chart, the sensible cooling process proceeds horizontally to the left along a line of constant humidity ratio toward the saturation line. The final statepoint has a lower dry-bulb temperature, a lower wet-bulb temperature, and higher relative humidity. There is no change in dew-point temperature, *water vapour* pressure, or humidity ratio in this process, indicating no change in *water-vapour* content. Vector B in Figure 19-1 shows this process.

The lower dry-bulb temperature limit of the sensible cooling process occurs at approximately 85% relative humidity. Attempts at additional sensible cooling usually result in the condensation of a portion of the *water vapour* from the air onto the surfaces of the cooling coil or heat exchanger; when this occurs the process is called a *cooling and dehumidification* process.

Humidification *Only* (Latent Heating) Process (Water Vapour Addition Without Temperature Change). The addition of *water vapour* (ultra low pressure steam) to air using a steam grid humidifier or an electric-replaceable-cylinder-water-vapour-generator are examples of this process. In actual practice the steam may add a small quantity of sensible heat. (A question in Chapter 24, "Frequently Asked Questions," provides more detail on this subject.) On a psychrometric chart, the humidification *only* process proceeds up a line of constant dry-bulb temperature. The final statepoint has a higher humidity ratio, higher dew-point temperature, higher relative humidity, and higher wet-bulb temperature. The dry-bulb temperature of the initial and final state are equal, indicating that the process involves no sensible heating or cooling. Vector C in Figure 19-1 shows this process.

Dehumidification *Only* (Latent Cooling) Process (Water Vapour Removal Without Temperature Change). The removal of *water vapour* from air using an industrial sprayed coil liquid desiccant dehumidifier is an example of this process. In one type of industrial liquid desiccant dehumidifier, air passes through 10 to 20 rows of bare-pipe-cooling coils over which concentrated lithium chloride (the liquid desiccant) is sprayed. Some may claim that this is a chemical dehumidification process followed by a sensible cooling process; however, it should be regarded as a single process since cooling and dehumidification occur simultaneously and the cooling enhances the effectiveness of the desiccant. Also, there is no definable intermediate statepoint to delineate a sequence of two processes. On a psychrometric chart, the dehumidification *only* process proceeds down a line of constant dry-bulb temperature. The final statepoint has a lower humidity ratio, lower dew-point temperature, lower relative humidity, and lower wet-bulb temperature. The dry-bulb temperature of the initial and final statepoints are equal, indicating that the process involves no sensible heating or cooling. Vector D in Figure 19-1 shows this process.

Other Basic Processes

Cooling and Dehumidification Process (Figure 19-4). The transfer of sensible energy and *water vapour* from air using a chilled water, glycol, or refrigerant cooling coil or an air washer supplied with chilled water are examples of this process. On a psychrometric chart, this process starts as a *sensible cooling* process. When the air being sensibly cooled reaches approximately 85% relative humidity, the remainder of the process follows a vector curving toward the entering coolant temperature (assuming a counterflow coil arrangement) at the saturation curve. ARI Standard 410 provides algorithms for calculating the theoretical process path and final statepoint of the cooled and dehumidified air leaving the cooling coil. Coils produced by each manufacturer deviate by approximately ±1°C dry-bulb temperature at any point along the theoretical ARI-calculated process line. The *leaving relative humidity* at the actual coil leaving dry-bulb temperature is a function of the entering air humidity ratio, the number of coil rows, fin pattern and spacing, the velocity of the air, the velocity of the coolant, and the log mean temperature difference of the entering and

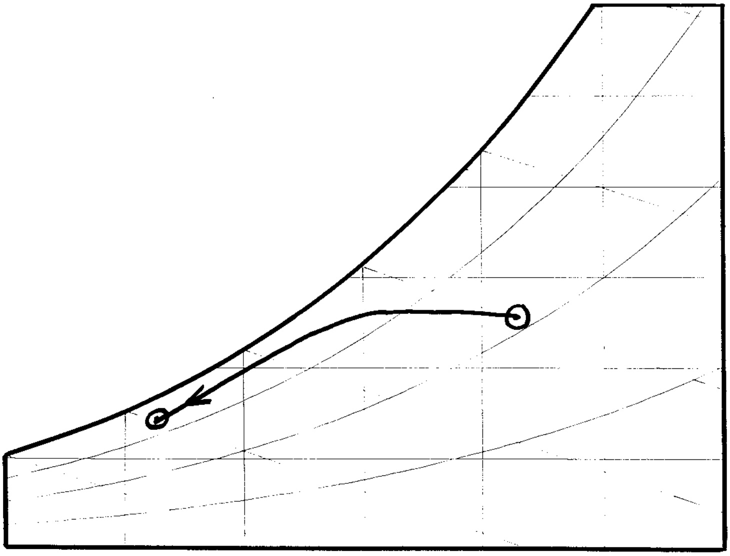


Figure 19-4—Cooling and dehumidification process.

leaving coolant and the entering and leaving air. As a rough approximation with 2.5 m/s (500 ft/min) coil air face velocity, with 27°C EDB, 19°C EWB, 7°C EWT, and 13°C LWT, a four-row cooling coil will achieve 86% to 90% leaving RH, a six-row cooling coil 92% to 95% RH, and an eight-row coil 94% to 96% RH. In psychrometric analysis, it is important that the statepoint of the air leaving the cooling coil matches the statepoint of the installed cooling coil—i.e., the coil leaving dry-bulb temperature and the leaving relative humidity used in the analysis must match the actual coil leaving values.

Cooling and dehumidification can also be accomplished using an air washer with externally cooled water. The 1902 air conditioning of the Sackett-Wilhelms Lithographing and Printing Company by Willis H. Carrier used this system. This is a common application in textile manufacturing where the lint in the air tends to clog standard cooling coils. A later section of this chapter describes another technology for cooling air to the same statepoint as the cooling and dehumidification process, i.e., desiccant dehumidification followed by sensible cooling.

The two following paragraphs discuss occasional errors in the specification or scheduling of cooling and dehumidifying coils. The first relates to specifying or scheduling a leaving dry-bulb temperature to the left of the actual cooling and dehumidification process curve shown in Figure 19-5. The second relates to specifying or scheduling a leaving dry-bulb temperature to the right of the actual curve.

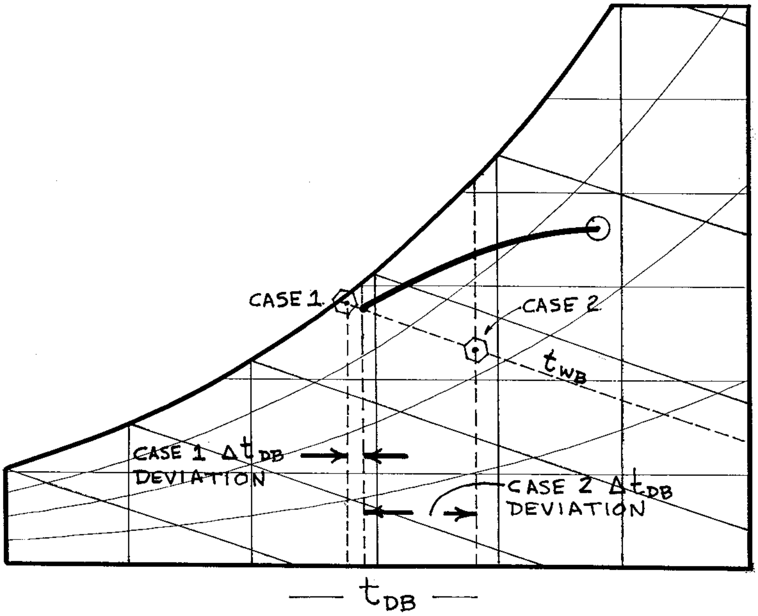


Figure 19-5—Deviations between scheduled and actual coil performance.

Case 1. Typically, in accordance with ARI Standard 410, the coil manufacturer (or the coil manufacturer's selected software) selects for and guarantees capacity at the leaving wet-bulb temperature only (not at the leaving dry-bulb temperature). If the designer specifies a leaving dry-bulb temperature closer to the leaving wet-bulb temperature than obtainable from the manufacturer, the net result will be a coil that provides the required total capacity at the specified leaving wet-bulb temperature; but, unfortunately, the coil leaving dry-bulb temperature will be slightly higher than desired and the coil's sensible cooling capacity will be less than desired. If the actual dry-bulb temperature deviation is +0.5°C, the calculated supply air quantity will be approximately 5% low.

Case 2. In this case, the designer specifies a leaving cooling coil condition far to the right of the cooling and dehumidification coil process line in the lower crosshatched area of Figure 19-2. An all-too-frequent example of this is a humid or mixed climate outside air makeup air cooling unit providing conditioned outside air to the corridors of a hotel or school. The designer desires to supply the outside air a few degrees cooler than the room temperature with a humidity ratio slightly below the room humidity ratio. This process is not physically attainable in a cooling and dehumidification coil process. No amount of words or legal language will overcome mother nature and make the process possible. The result is a coil having a significant surplus of sensible cooling capacity and a significant deficiency in *water vapour* removal (latent cooling) capacity. This leads to high humidity, clammy spaces, and the possibility of mold and bacteriological growth.

Evaporative Cooling Process (Adiabatic). Ideal evaporative cooling is an adiabatic process (no heat is added or removed), which follows the entering air wet-bulb line up and to the left toward the saturation curve. Air is brought into intimate contact with *recirculating* water at the wet-bulb temperature. If there is insignificant heat gain through the apparatus casing and water piping and the heat gain from pump energy is minor, then the process will be close to a true adiabatic process. The dry-bulb temperature decreases as the humidity ratio increases. The energy transferred from the air to the water in the sensible cooling of the air is equal to the latent heating energy required to vaporize the water. No energy is transferred to or from the surrounding environment except for a small amount of energy transferred to the air by the enthalpy of the mass of makeup water required to sustain the process. Vector E in Figure 19-2 shows this process.

Evaporative coolers are rated by percentage saturation effectiveness, which expresses the actual dry-bulb temperature decrease relative to the maximum possible dry-bulb temperature decrease in a perfect saturator. The somewhat theoretical adiabatic saturator discussed in the section on wet-bulb temperature is an ideal evaporative cooler operating at 100% saturation effectiveness. Industrial air washers use multiple opposing spray banks to achieve intimate mixing of the air and water and achieve saturation effectiveness of 95% to 98%. Rigid media (Munter's fill) evaporative coolers with 12 in. fill can achieve saturation effectiveness of 88% to 91%. Some residential evaporative coolers use aspen wood excelsior media resembling coarse steel wool. Other residential evaporative coolers use a plastic mesh referred to as a "hogs hair filter media." The saturation effectiveness of residential evaporative media may be in the 50% to 60% range.

Water Spray Processes. These processes utilize water sprays or wetted media to achieve intimate contact between air and water. Equipment includes cooling towers, dense-water-spray air washers, wetted fluted cellulose or fiber glass paper-like media, and sprayed coil equipment. Cooling towers are available with both counterflow of air and water and crossflow. Air washers are generally parallel flow. The process condition lines are dependent on the flow arrangement, the mass flow and state of entering air, the mass flow and temperature of the entering water, and the heat and mass transfer coefficients. A thorough discussion of the basic heat and mass transfer and the plotting of process condition lines for all options could be the subject of a separate book.

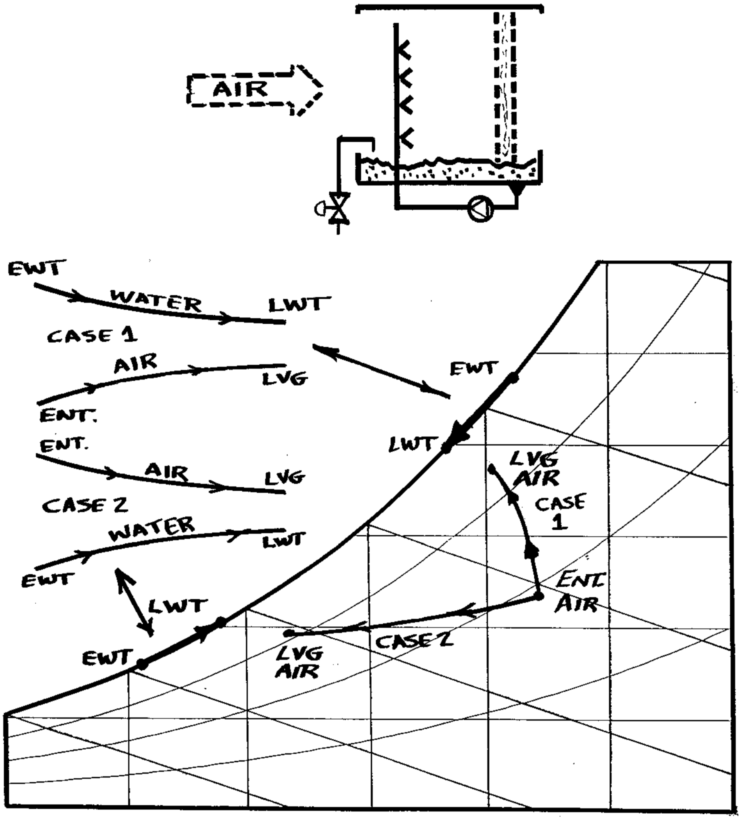


Figure 19-6—Parallel flow water spray process.

Figure 19-6 shows the relationship between water and air as they pass through a parallel flow apparatus for a cold water and a warm water process. Figure 19-7 shows similar information for a counterflow apparatus.

The process lines in Figures 19-6 and 19-7 are curved because the water changes temperature as it passes through the apparatus. Two air washer processes have straight condition lines: (1) the adiabatic evaporative cooling process in the preceding section and (2) a process such as a sprayed coil unit in which heat is either added or removed during the process to maintain the spray water at constant temperature.

Desiccant Dehumidification Process (Adiabatic) (Chemical Dehumidification). This process vector is the opposite of the adiabatic evaporative cooling process. *Water vapour* is adsorbed from the airstream by the desiccant and the *water vapour* is condensed to its liquid form within the pores of the desiccant. The latent heat of condensation released

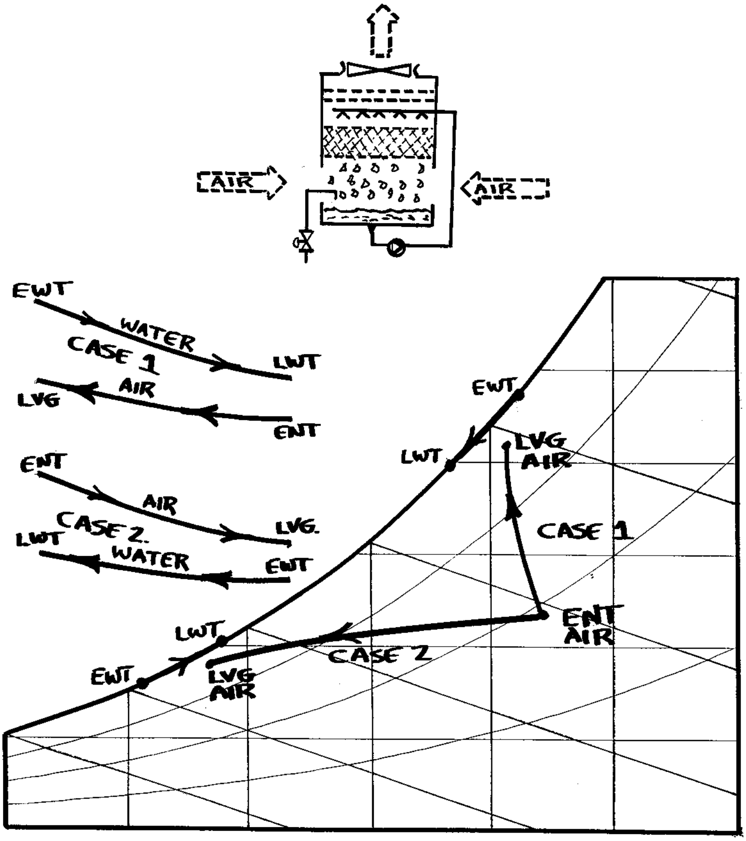


Figure 19-7—Counterflow water spray process.

as the *water vapour* condenses within the desiccant causes the desiccant and the air in contact with the desiccant to warm. The ideal adiabatic process follows the entering wet-bulb line down and to the right on the psychrometric chart. The latent heat released in condensing the *water vapour* becomes a sensible heat gain to the air. The dry-bulb temperature of the leaving air is warmer and the humidity ratio is lower. In actual practice, the dry-bulb temperature gain in the desiccant process is 20% to 30% greater than that predicted by a pure conversion of latent heat to sensible heat due to (1) the residual heat carried over or retained by the desiccant and its supporting structure from the hot regeneration process and (2) a lessor phenomenon called heat of wetting. Vector F of Figure 19-2 illustrates this process.

When the desiccant approaches a condition of saturation, it must be removed (or rotated) from the process airstream and regenerated. Regeneration is a separate sequence of processes in which heated air (usually heated outside air) flows across the water-saturated desiccant. The first process in regeneration is sensible heating of the air, which is followed by the second process in which the heated air takes up the adsorbed water in an evaporative cooling process. The regenerative airstream is heated to a temperature exceeding 120°C for silica gel desiccants and to as low as about 65°C for some molecular sieve desiccants. The coefficient of performance (COP) of the regeneration process varies from 0.3 for inefficient regenerators to 0.5 for industrial regenerators to approximately 0.7 for regenerators that incorporate heat pipes or rotary heat wheels to transfer a portion of the *carry-over-heat* from the leaving process airstream to the entering recuperative airstream.

Mixing of Two Airstreams Process (Figure 19-8). This is an adiabatic process involving the mixing of air at state 1 with a second stream of air at state 2. The resultant mixture is at state 3. The most common example is the mixing of a *mass* flow of outdoor air with a *mass* flow of return air in the mixing box of an air-handling unit. Other examples covered later include the *(ordinary) face and bypass damper process* and the *return air face and bypass damper process*. Other examples of the mixing process include dual-duct mixing boxes and cold deck-warm deck or cold deck-neutral deck-warm deck multizone air-handling units.

On a psychrometric chart using enthalpy-humidity ratio plotting coordinates, the final state of the mixture process lies on a straight line connecting the two original states. The enthalpy of the resultant mixture

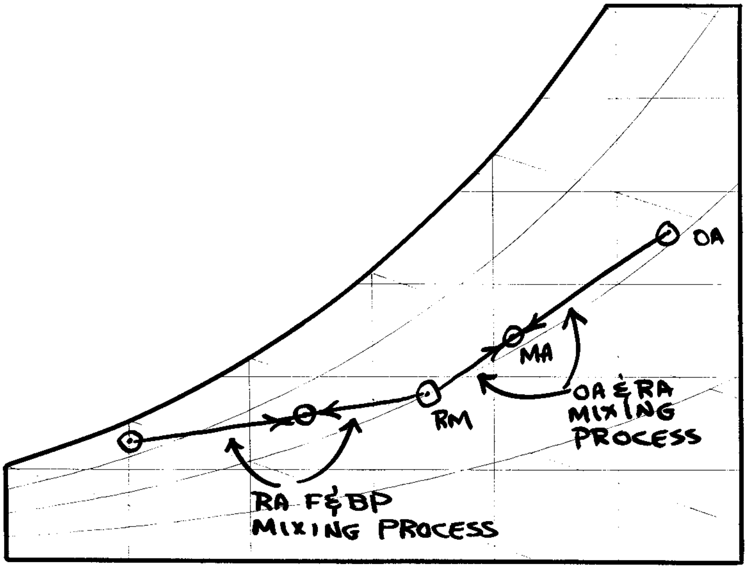


Figure 19-8—Mixing of two airstreams process.

is determined by summing the product of each mass flow times its enthalpy and dividing by the total mass flow. The humidity ratio of the mixture is determined by summing the product of each mass flow times its humidity ratio and dividing by the total mass flow. An approximation of the final dry-bulb temperature (suitable for most air-conditioning work) can be obtained by summing the product of each volumetric flow times its dry-bulb temperature and dividing by the total volumetric flow. The small error introduced in this approximation is caused by the differences in specific volumes of the two airstreams and, to a minor extent, by differences in the air specific heat capacities. For maximum accuracy, the final dry-bulb temperature should be determined using mass flows and the calculated enthalpy and humidity ratio as input into psychrometric algorithms.

Room Effect Process (Figure 19-9). This process vector shows the progression of states as the parcel of supply air passes through and becomes intimately blended with the air in the room(s) or space(s). The vector in Figure 19-9 shows the parcel being heated and humidified by the room air (in a cooling application). The room effect process vector varies throughout the day and seasonally as the space sensible and latent loads vary. The room effect process vector can proceed from the supply air statepoint in 360 (in reality an infinite number of) different directions depending on the time of day, the season, the climate, interior loads, exterior loads, and the air leakage characteristics of the exterior walls, windows, roof, and floor. The possible changes in both space sensible and latent load have a major impact on the ability of the HVAC system to maintain space

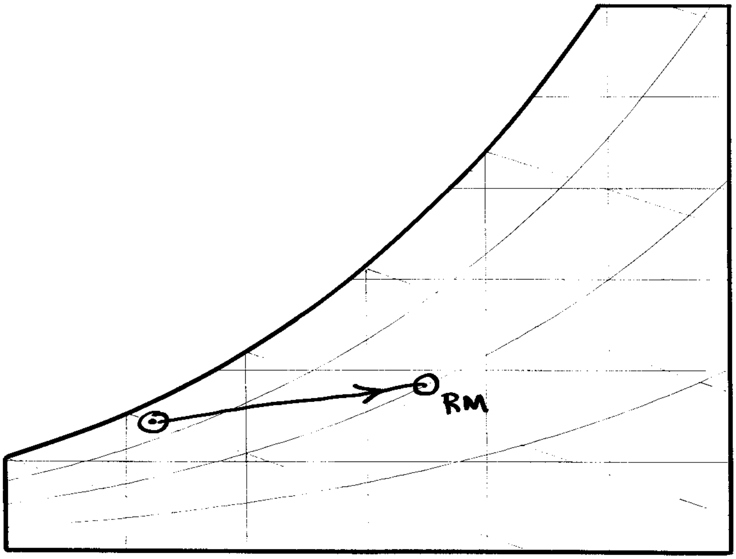


Figure 19-9—Room effect process.

comfort conditions as well as the energy requirements of the HVAC system at partial loads. These issues are addressed in Chapter 20, "Process Calculations and Definitions of Sensible and Latent Enthalpy Change," under the topic ***part-load operating conditions.***

In the case of summer air conditioning, sensible heat from solar effects, envelope conduction, lights, business machines, infiltration, and people is transferred to the supply air. *Water vapour* gain (latent heat) from people, infiltration, and other *water vapour* sources in the conditioned space is also transferred to the supply air. The summer air-conditioning peak design room effect process vector moves from the supply air state upward and to the right on the psychrometric chart as the supply air takes up sensible heat and *water vapour* (latent heat) from the room. In effect, this is a combination sensible heating and humidification process. The difference is that the room effect process originates at the statepoint of the supply air and ends at the statepoint of the room air, whereas the heating and humidifying process originates at the room air and ends at the statepoint of the supply air. The final state, which should be the desired room temperature and humidity ratio, has a higher dry-bulb temperature and a higher humidity ratio. The peak summer load process vector is shown in Figure 19-9.

In cold weather, the room effect process includes sensible cooling from the envelope due to infiltration and conduction (offset to some extent by heat gains from solar effects, lights, business machines, and people) and a reduction in *water vapour* due to infiltration of cold *dry air* leaking into the conditioned space (offset by minor *water vapour* gains from people). The cold weather room effect vector proceeds from the state of the supply air entering the room toward a lower dry-bulb temperature and a lower humidity ratio. In effect, this is a combination cooling and dehumidification process.

Other Processes

Fan Heat Process (Figure 19-10). The fan heat process is a special category of a *sensible heating* process. It represents the energy supplied to the fan shaft and the v-belt drive and the energy associated with motor losses. In some cases, the motor and/or the v-belt drive are located out of the airstream; in such cases, the energy associated with these losses is assigned to the room or airstream in which these components are actually located. Motors supplied by adjustable-frequency, variable speed drives experience greater heating effect than the same motor supplied by a pure sine wave power supply.

Enthalpy Exchange Process (Figure 19-11). The enthalpy exchange process involves the exchange of sensible heat and *water vapour* between two airstreams using a rotary wheel heat-and-*water-vapour* transfer media that rotates between the two airstreams. Membrane enthalpy heat

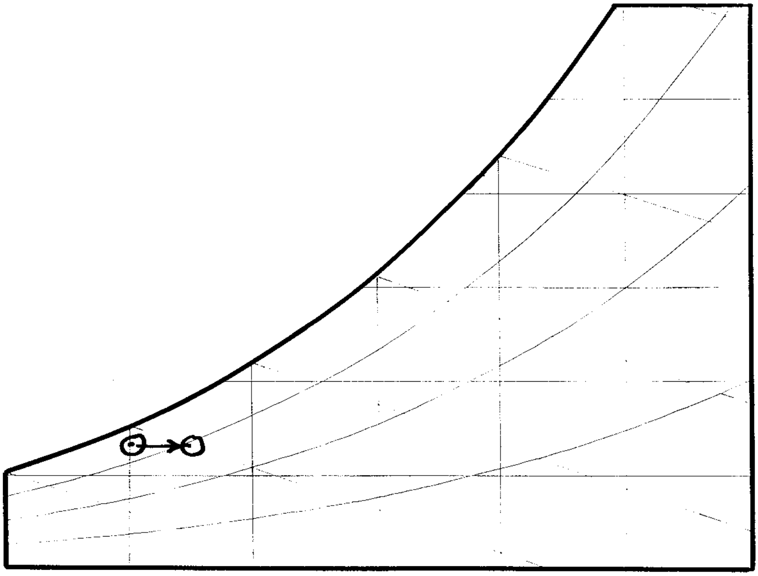


Figure 19-10—Fan heat process.

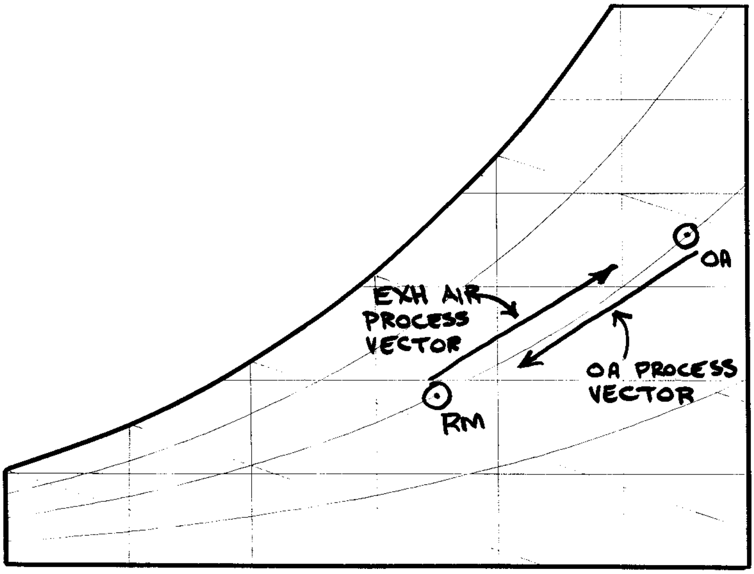


Figure 19-11—Enthalpy exchange process.

exchangers are also used. Psychrometrically, the enthalpy exchange process is two separate processes that are closely interrelated because the enthalpy transfer occurs from one airstream to the media in the first process and from the media to the other airstream in the second process. The combination of the two processes is adiabatic.

Enthalpy wheel construction is similar in some respects to an active (heat powered) desiccant dehumidifier wheel. Both wheels are coated with a desiccant. The active desiccant wheel is usually much thicker in direction of airflow than the enthalpy exchange (passive desiccant) wheel and contains about five times the mass of desiccant for the same process air quantity. The active desiccant wheel rotates at approximately 10 revolutions per hour and the enthalpy (passive desiccant) wheel rotates at approximately 600 revolutions per hour. About 75% of the active desiccant wheel face area receives process air and 25% receives heated regeneration air. The enthalpy wheel proportions are usually 47% and 47%, and neither airstream receives heat from an external source. About 6% of the wheel segment area is partitioned for purge air to ensure that none of the exhaust air reaches the fresh airstream.

One-half of the enthalpy wheel takes on sensible heat from the warmer airstream and rotates into and transfers that sensible heat to the colder airstream. Simultaneously, one-half of the wheel takes on *water vapour* (latent heat) from the airstream having the higher *water vapour* pressure and transfers that *water vapour* to the lower *water vapour* pressure airstream.

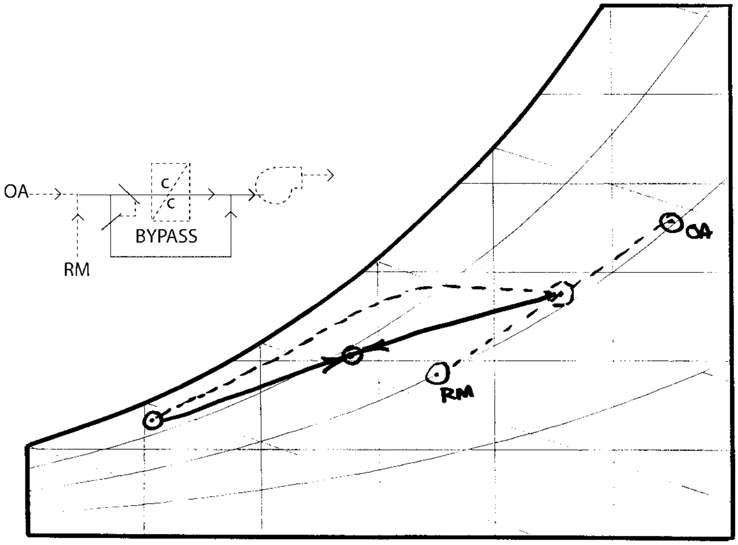


Figure 19-12—Ordinary (mixed air) F&BP process.

The air paths are arranged in a counterflow direction. If the manufacturer has designed the wheel so that its sensible and latent (*water vapour*) transfer effectiveness values are equal, which is usually the case, then the process vector of the makeup airstream proceeds in straight line from the makeup air state toward the exhaust air state. The length of the vector compared to the total distance between the makeup air and exhaust air states is the enthalpy wheel's effectiveness factor. The process vector for the exhaust airstream can be similarly described except that it originates at the exhaust air state.

Two or More Processes in Sequence

(Ordinary) (Mixed Air) Face and Bypass (Damper) Process (Figure 19-12). This process involves two interrelated modulating dampers that vary the portion of air that goes through a device and the portion that bypasses the device. The first process involves the device (a cooling coil, heating coil, air washer, active desiccant dehumidifier, etc.). The second process involves the mixing of the portion of the air passing through the device and the portion of the air that bypasses the device. The air entering the first device is often a mixture of outside air and return air from the space. Ordinary face and bypass is built into central station air-handling units at the factory.

Return Air Face and Bypass (Damper) Mixing Process (Figure 19-13). The difference between this process and the (ordinary) face and bypass (damper) process in the previous paragraph is that this process

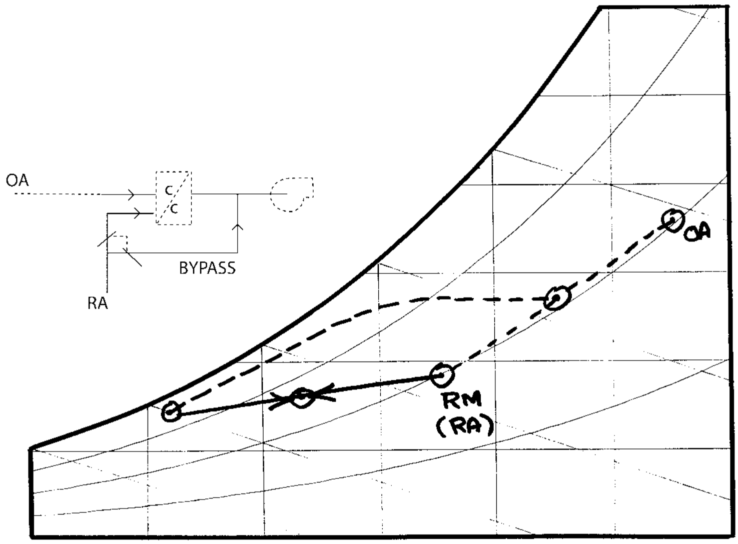


Figure 19-13—Return air F&BP process.

bypasses only return air from the space and, therefore, the outside air should always go through the cooling coil. In the *ordinary* face and bypass process, the bypass air is a mixture of outside air and return air. Return air face and bypass of a cooling coil results in lower space relative humidity than ordinary bypass. Return air face and bypass often involves ductwork added in the field from the return air duct to a plenum in the air-handling unit downstream of the cooling coil.

New Energy Reheat (Sensible Heating) Following the Cooling and Dehumidification Process (Figure 19-14). This system was popular in the era before the 1973 energy crisis. Hospital operating suites, laboratory test rooms, and other spaces requiring close control of temperature and relative humidity still use this process. The air first undergoes the cooling and dehumidification process, followed next by the fan heat process (assuming that the fan is located downstream of the cooling and dehumidification apparatus), and finally the cooled and dehumidified air undergoes a *new energy* sensible heating process. The cooling and dehumidification process is controlled so that it removes sufficient *water vapour* to maintain the space relative humidity. In many cases, the coil leaving air temperature necessary to satisfy the room relative humidity controller results in cooling the room below the desired setpoint. This sometimes occurs at full-load conditions and frequently occurs at part-load conditions. Thus, the cold air is sensibly heated to avoid overcooling of the space. New energy reheat is often accomplished with a hot water or steam heating coil or an electric heating coil. The source of the reheat energy is *new energy* in the form of fossil fuel to a boiler or electricity.

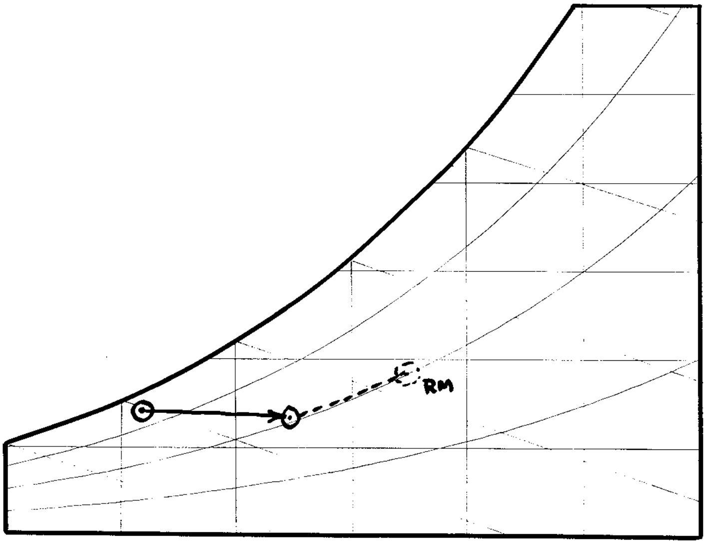


Figure 19-14—“New energy” reheat process.

Recuperative Runaround with Cooling and Dehumidification (Figures 19-15 and 19-16). This is a sequence of processes that can displace a portion of the new energy reheat process. Other processes that might be used to replace the new energy reheat process involve the use of solar or recovered heat.

The recuperative runaround with cooling and dehumidification process is often used with 100% outside air-conditioning units. It involves the use of coupled recuperative runaround heat exchanger devices to sensibly cool the air upstream of the main cooling and dehumidification coil and then sensibly heat the air downstream of the cooling and dehumidification coil. The recuperative runaround heat exchanger can be interconnected water or glycol coils, heat pipes, air-to-air heat exchangers, or rotary wheel heat exchangers. Figure 19-15 shows the arrangement of coils. A heat transfer fluid circulating between the upstream and downstream heat exchange devices transfers heat from the precooling device to the postheating device. The air passing over the postheating coil cools the heat transfer fluid, which then circulates through the precooling coil. The heat transferred to the heat transfer fluid in the precooling coil is equal to the heat transferred from the fluid in the downstream coil. Figure 19-16 shows this sequence of processes.

The so called *free* reheat available in this sequence of processes declines from a maximum at summer design conditions to zero when the outside air temperature equals the temperature leaving the cooling and dehumidification coil.

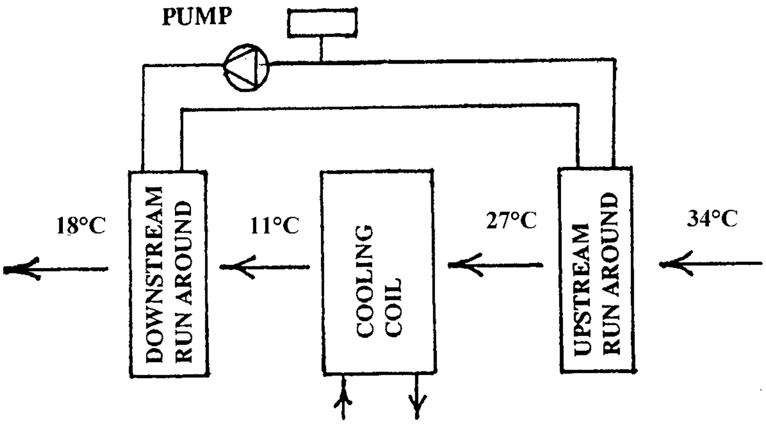


Figure 19-15—Recuperative run-around equipment schematic.

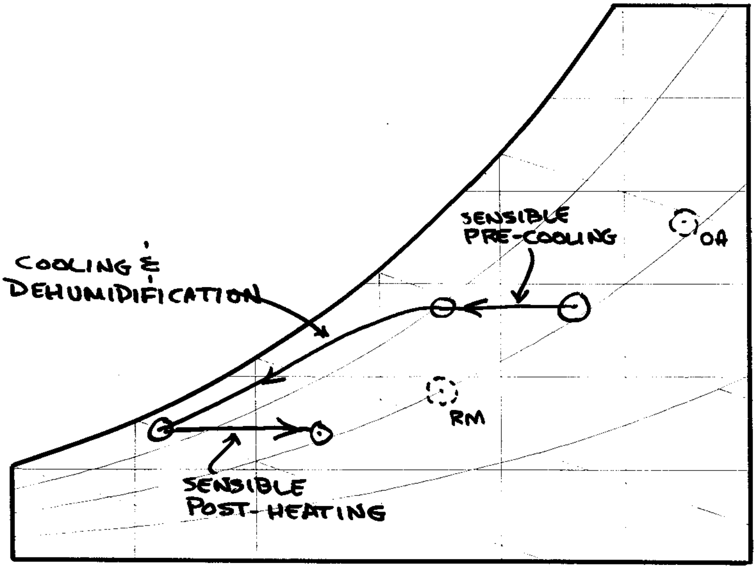


Figure 19-16—Recuperative run-around with cooling and dehumidification.

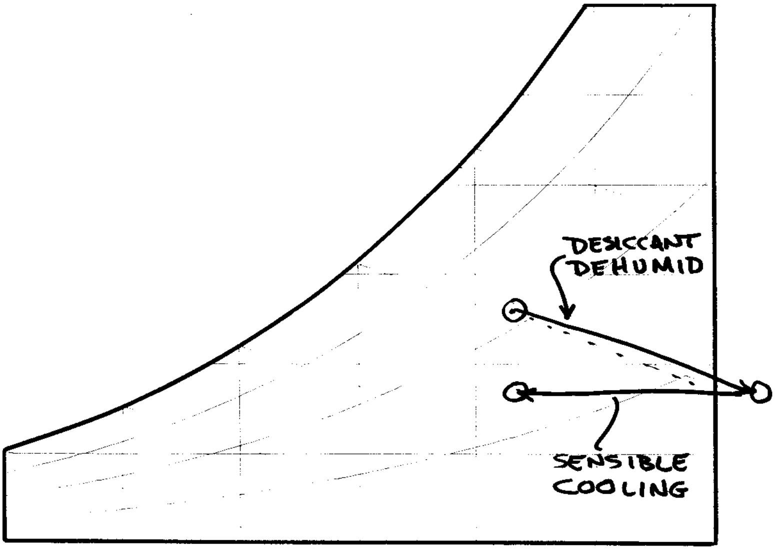


Figure 19-17—Desiccant dehumidification followed by sensible cooling.

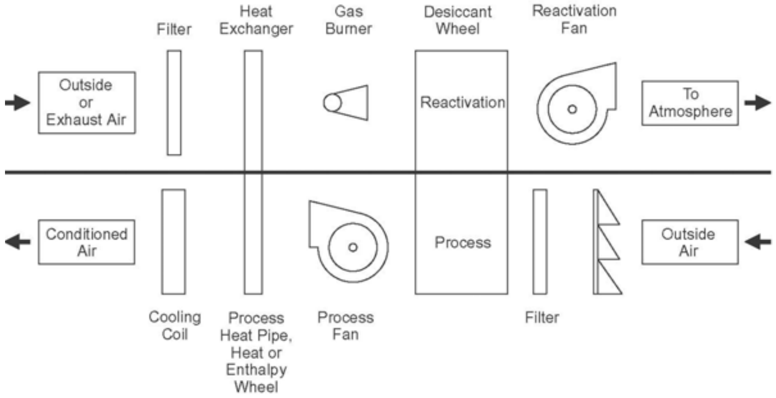


Figure 19-18—Desiccant equipment schematic.

Desiccant Dehumidification Followed by Sensible Cooling (Figure 19-17). Another alternative to the sequences discussed in the two prior sections is active heat-powered *desiccant dehumidification* followed by *sensible cooling*. This sequence and the sequences in the two prior sections all begin by removing *water vapour* down to the final humidity ratio. (In the cooling and dehumidification process, the air is simultaneously cooled and dehumidified and then sensible heat is added to achieve the final state.) In the active desiccant dehumidification process, the air is simultaneously dehumidified and heated (by conversion of latent heat to sensible heat the heat transferred from the mass of the desiccant rotor or liquid returning from reactivation) and then, in the second process, the warm air undergoes a sensible cooling process to achieve the final state.

A large portion of the sensible cooling following active desiccant dehumidification should be accomplished using a coolant that requires minimal "new" (as opposed to recovered or waste) energy. Cooling tower water in a finned coil is a common source of low-energy sensible cooling. Another source of low-energy sensible cooling uses an air-to-air heat exchanger (AAHX) in which evaporatively cooled outside airflows through one path of the AAHX and cools the hot *dry air* from the desiccant dehumidifier in the other path. Figure 19-18 is a schematic of this equipment. Some recent designs accomplish the evaporative cooling by spraying or dripping water onto the heat transfer surfaces of the regeneration section of the heat exchanger.

Sensible Cooling Followed by Evaporative Cooling (Figure 19-19). Sensible cooling of air on the entering air side of an evaporative cooling process lowers the evaporative cooler inlet wet-bulb temperature with the result that the dry-bulb temperature leaving the evaporative cooler is lower than that obtainable using only evaporative cooling. This sequence of pro-

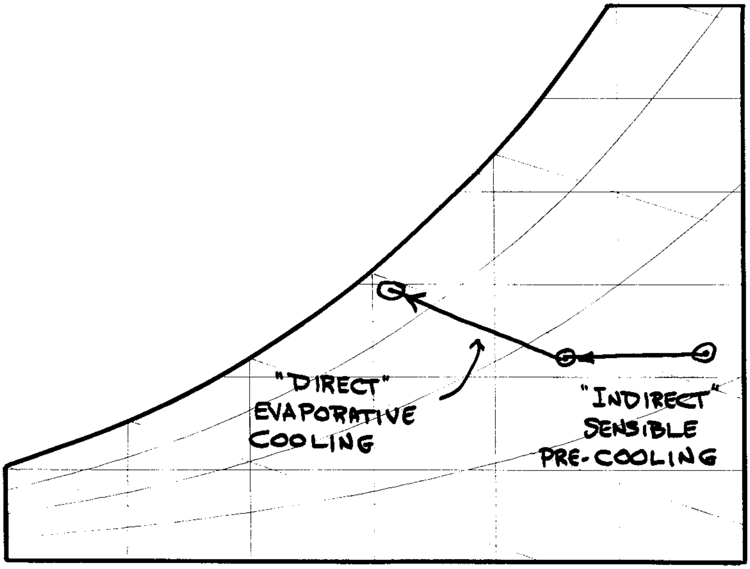


Figure 19-19—Sensible cooling followed by evaporative cooling.

cesses is often referred to as *indirect-direct evaporative cooling.* At 100% saturation effectiveness, the lowest leaving air dry-bulb temperature achievable with an evaporative cooler is the entering wet-bulb temperature; however, with a precooler, the lowest theoretical temperature obtainable is the dew-point temperature of the entering air.

The sensible cooling should be accomplished using a coolant that requires minimal "new" energy, as described in the paragraph preceding this section.

Sensible Heating Followed by Humidification (Figure 19-20). This is a common cold climate application in which cold outside air or a mixture of return air and cold outside air is heated in the first process and then humidified in the second process. Theoretically, this could be accomplished in the single process of heating and humidification using an air washer with externally heated water. This single process is rarely used because the water temperature necessary to accomplish heating of the air by direct contact could result in adding significantly more *water vapour* to the air than desired. Heating and humidification are easily accomplished by a sequence of the sensible heating *only* and humidification *only* processes.

Typical Air-Conditioning Cycle. A cycle is a sequence of processes in which the final state of the last process in the sequence is the same as the initial state of the first process. In this example, there are four sequential processes: (1) mixing of OA and RA airstreams, (2) cooling and dehu-

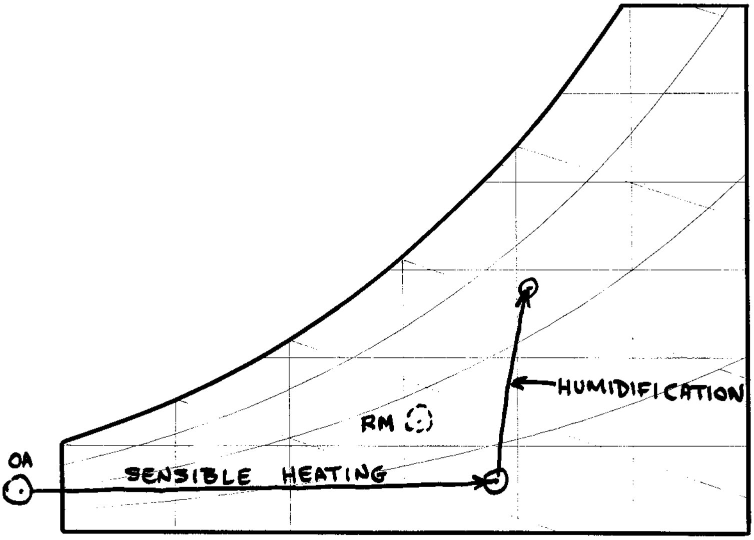


Figure 19-20—Sensible heating followed by humidification.

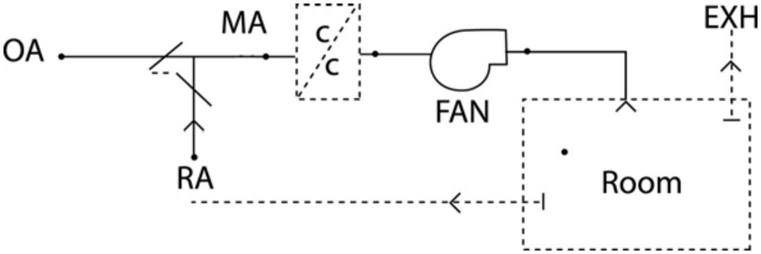


Figure 19-21—Four-process air-conditioning cycle—equipment schematic.

midification, (3) fan heat, and (4) room effect. Figure 19-21 shows this sequence of processes. Figure 19-3 shows these processes on a psychrometric chart.

Other Combinations of Processes in Sequence. The seven combination processes presented in this section are ample evidence that an almost limitless number of combinations is possible. Some are cost-effective at the present time because of current energy cost. Some await the development of technology. Some may simply be waiting for a manufacturer to recognize and cash in on the opportunity.

The following procedure is typical for analyzing and solving problems involving steady-flow processes in heating, ventilating, dehumidification, and air conditioning: The boundaries of the system are first established, then all the forms of energy and mass crossing these boundaries are noted, and finally all the known properties of the fluid streams (including mass flow) and the amount of heat and work crossing the boundaries are calculated.
—anonymous

20 Process Calculations and Definitions of Sensible and Latent Enthalpy Change

The basic principles applicable to psychrometric process calculations are (1) conservation of energy, (2) conservation of the mass of the dry air component, and (3) conservation of the mass of the H_2O component.

Process calculations can be divided into two groups:

1. Mixing processes (Figure 20-1), which usually involve multiple input streams and a single output stream. The properties and mass flow rates for all but one of the streams are known, and equations presented later in this chapter are utilized to solve for the unknown mass flow and properties. Mixing processes are often assumed to be adiabatic, meaning that no heat is transferred into or out of the system.

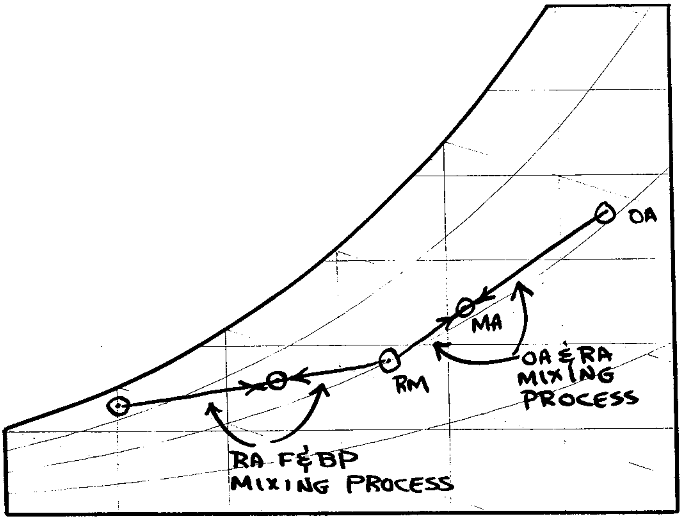


Figure 20-1—Mixing processes.

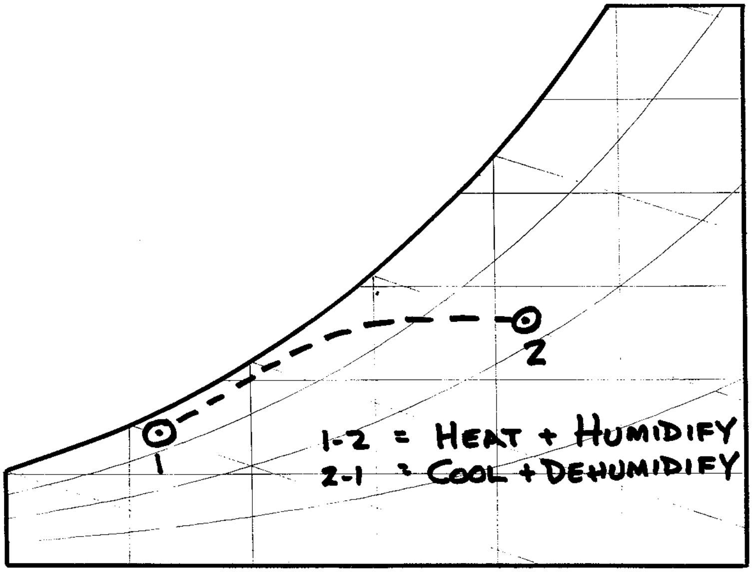


Figure 20-2—Heating (cooling) and/or humidifying (dehumidifying) processes.

Table 20-1—Heating (Cooling) and/or Humidifying (Dehumidifying) Processes

Dry-Bulb Temperature Change	H_2O Component		
	Add H_2O	No Change in H_2O	Remove H_2O
$+\Delta t_{DB}$	$+ \Delta h_{SENS}$ & $+ \Delta h_{LAT}$ Heat* and humidify	$+ \Delta h_{SENS}$ (only) Sensible heat* only	$+ \Delta h_{SENS}$ & $- \Delta h_{LAT}$ Heat* and dehumidify
$t_{DB\ ENT} = t_{DB\ LVG}$	$+ \Delta h_{LAT}$ (only) Humidify only	No Change in t or W (Not a process)	$- \Delta h_{LAT}$ (only) Dehumidify only
$- \Delta t_{DB}$	$- \Delta h_{SENS}$ & $+ \Delta h_{LAT}$ Cool and humidify	$- \Delta h_{SENS}$ (only) Sensible cool only	$- \Delta h_{SENS}$ & $- \Delta h_{LAT}$ Cool and dehumidify

** In physics and psychrometrics, "heat" is strictly defined as the transfer of energy across a boundary due to a difference in temperature. The term "heat" is not used to represent energy stored in a substance.*

2. Heating (cooling) and/or humidifying (dehumidifying) processes (Figure 20-2), which involve (1) a beginning statepoint, (2a) the addition (removal) of heat and/or (2b) the addition (removal) of H_2O, and (3) a resultant final statepoint. Eight combinations are possible, as shown in Table 20-1.

The equations used to solve for the heat and H_2O transfers during these processes utilize the sign convention that heating (adding heat) is positive and humidification (adding H_2O) is positive. Cooling is therefore negative heating; dehumidification is negative humidification. The use of this sign convention allows one set of equations to be used for all eight processes. This is convenient for the practitioner and the student. The term H_2O is used because H_2O can be added as a solid (ice), a liquid (water), or a gas (steam).

Process Sensible and Latent Enthalpy (Energy) Changes

> *Although small in concentration, water vapour constitutes a very significant portion (constituent) of the atmosphere, varying from very little in the desert and polar regions to as much as 4 percent by volume in jungle tracts. Water is a major factor in the making of weather. It can exist in a gaseous, liquid, or solid state. In its gaseous state, water is an invisible vapour; in its liquid state, rain. If temperatures are below freezing, water changes directly into solid ice crystals—sleet, hail, or snow. This solid form may be converted to a liquid state by melting or to a gaseous state by evaporation (sublimation).*
>
> *When water changes from one phase to another, the rather mysterious exchange of latent, or "hidden," heat is involved. Unlike "sensible" heat—heat that we feel, or sense, and measure as temperature—latent heat is stored in the molecular structure of liquid and solid water. When water vapour is cooled and condenses into a cloud or rain droplet, latent heat is released to the air, where it becomes sensible heat, raising the air's temperature.*—David M. Ludlum

In psychrometric processes, it is often convenient from accounting and calculation standpoints to separate the process *total enthalpy* change into *sensible enthalpy* change and *latent enthalpy* change. Following the practice in the previous psychrometric property chapters, several explanations are provided.

Sensible enthalpy change implies that the outcome or the effect of a sensible (only) enthalpy change process can be *sensed*; that is, a person can *sense* a difference in dry-bulb temperature between the entering and leaving process airstream—accurately with a dry-bulb thermometer or less accurately with one's hand.

Before continuing with *sensible enthalpy change,* the next two paragraphs discuss *latent enthalpy change* to help the reader grasp the difference between *sensible enthalpy change* and *latent enthalpy change*.

The word *latent* in *latent enthalpy change* was originally used by Joseph Black in 1757 (see Chapter 8, "Psychrometric Pioneers and Charts from the First 100 Years"). *Latent* means *hidden* or *not capable of being readily sensed*. It could just as well have been called *insensible enthalpy change*. With a latent-only enthalpy change process, a person cannot readily *sense* a difference between the entering and leaving process airstream with a dry-bulb thermometer or with one's hand because the dry-bulb temperatures of the entering and leaving airstreams are the same. Note that instruments other than a dry-bulb thermometer (e.g., a dew-point instrument or hygrometer) could be used to sense a change in a latent enthalpy process and, therefore, it can be argued that the term *latent* (hidden) or *insensible* is not exactly correct. Arguments aside, the term *latent enthalpy change* or *latent heat* is here to stay. If this preliminary paragraph helps in understanding, then use it; otherwise, ignore it.

Latent enthalpy change (often called latent heat) is energy in the form of heat transferred to a substance that is not evidenced or not apparent by a change in dry-bulb temperature. In the case of H_2O, it refers to the latent heat of vaporization, which is the heat required to change liquid H_2O to its gas (vapour) phase at constant temperature. This hidden heat is obviously present in the vapour because when the vapour is condensed back to its liquid phase, the energy reappears.

Sensible enthalpy change can be defined as a change in dry-bulb temperature of the *dry air* and the associated *water vapour* with no change in the humidity ratio (i.e., no change in the mass of *water vapour* associated with the *dry air*). Sensible enthalpy change is the energy transfer required to increase or decrease the dry-bulb temperature of an air mass without changing its humidity ratio. Sensible heating involves an increase in dry-bulb temperature; sensible cooling, a decrease.

Latent enthalpy change is the change in enthalpy associated with an increase or decrease in the humidity ratio of an air mass without changing its dry-bulb temperature. In psychrometrics, a latent enthalpy change means that a mass of H_2O vapour is transferred to the airstream or from the airstream during a process. An increase or decrease in humidity ratio is another way of stating that the mass of *water vapour* associated with the *dry air* increases (humidification) or decreases (dehumidification) during the process.

A common reader query at this point is: "O.K., I understand that latent enthalpy change involves either adding or removing *water vapour* from the process airstream, but what does that have to do with a change in enthalpy?"

One way to answer this is to use the process of increasing the water-vapour content of the air in a closed and sealed room by heating an open pan of water. This requires the transfer of approximately 2465 kJ/kg to the liquid water in the pan in order to vaporize one kilogram of water vapour. One kilogram of water vapour molecules escaping from the pan includes this 2465 kJ/kg of enthalpy, which is called the *latent enthalpy* change.

Table 20-2—Solid Phase to Vapour Phase

Temperature	–10°C	–5°C	0°C
h_{SG} kJ/kg	2837	2836	2834
h_{SG} Btu/lb	1220	1219	1219

Table 20-3—Liquid Phase to Vapour Phase

Temperature	0°C	5°C	10°C	15°C	20°C	25°C	30°C
h_{FG} kJ/kg	2501	2489	2477	2465	2454	2442	2430
h_{FG} Btu/lb	1075	1070	1065	1060	1055	1050	1045

Another way to illustrate a *latent enthalpy* change is to use the process of removing water vapour from an airstream. This requires that (1) the water vapour be condensed to its liquid or solid phase and then (2) the liquid or solid water be removed from the apparatus. Condensation of water vapour to liquid water at constant dry-bulb temperature requires the removal of approximately 2465 kJ per kg_H_2O (a value that varies slightly with the temperature at which the H_2O is evaporated or condensed). Thus, it can be said that air containing 1 kg of water vapour has a latent enthalpy content of approximately 2465 kJ.

Previous paragraphs have used the approximate value of 2465 kJ/kg (~1060 Btu/lb) as the latent heat of vaporization at constant temperature. Tables 20-2 and 20-3 give the exact value at the temperature of evaporation.

In the bygone era of simplified (or shortcut) manual psychrometric calculations in inch-pound units, the air-conditioning industry approximated latent heat change per unit mass of H_2O as the enthalpy of 75°F (26.1°C) water vapour with a value of 1093.92 Btu/lb (2544.46 kJ/kg) minus the enthalpy of condensed water leaving the system at 50°F (10°C) having a value of 18.06 Btu/lb (42.01 kJ/kg). The resultant value of 1075.86 Btu/lb (2502.45 kJ/kg) was, by coincidence, essentially equal to h_{FG} at 32°F (0°C). The ~1076 Btu/lb value multiplied by the density of standard air (0.075 $\text{lb}_{DA}/\text{ft}^3$) and the conversion from minutes to hours (60 min/h) produced the somewhat familiar constant of ~4840 $\text{Btu}\cdot\text{lb}_{DA}/(\text{h}\cdot\text{cfm}\cdot\text{lb}_{WV})$ used in the equation $q_{LAT} = 4840 \cdot scfm \cdot \Delta W$ with ΔW units of $\text{lb}_{WV}/\text{lb}_{DA}$. If ΔW is expressed in units of $\text{grains}_{WV}/\text{lb}_{DA}$ then the constant becomes 4840/7000 or ~0.69. The comparable SI equation is $q_{LAT} = 3013 \cdot Q_{L/s} \cdot \Delta W$. The I-P constants 4840 and 0.69 and the SI of 3013 in the equations were useful approximations in the era of manual psychrometric calculations. They may still have some value in quick checks or approximations; however, the reader must understand that these equations are accurate only for (1) air at a density of 0.075 $\text{lb}_{DA}/\text{ft}^3$ (1.204 $\text{kg}_{DA}/\text{m}^3$), (2) evaporation or condensation at 75°F (26.1°C), and (3) liquid H_2O entering or leaving the system at 50°F (10°C).

The comparable SI constants are not fully derived in the previous paragraph because this would only add credence to the continuation of equations based on approximations in this era of more accurate software-based calculations

Humidification and Dehumidification

An increase in the mass of *water vapour* associated with a parcel of air is called *humidification* and requires that an energy transfer (or internal exchange of energy) occur to cause the added H_2O (ice or water phase) to evaporate (liquid to gas) or sublimate (solid to gas) to the vapour phase at constant temperature. A humidification process is often called *latent heating*. In the process of *humidification* with low pressure steam, the latent energy transfer to the H_2O occurs in a preceding process.

Dehumidification, the reverse process of humidification, is often called *latent cooling*. The latent enthalpy change is the energy transfer from the air mass required to condense the H_2O mass from the vapour phase to its final (liquid or solid) phase without change in temperature.

Latent enthalpy change (the enthalpy of evaporation or condensation) is the energy transfer required to increase or decrease the mass of *water vapour* associated with an air mass without changing its dry-bulb temperature. Latent heating involves an increase in humidity ratio; latent cooling, a decrease.

A Side Note on the Technically Incorrect Use of the Term "Heat"

If one appends the word *heat* to the terms *sensible* or *latent* in the presence of a thermodynamics or physics purist, chastisement may follow. Here is the background. Over the last fifty years, the enthalpy changes discussed in the previous paragraphs have come to be called *sensible heat* and *latent heat,* which is, in a few cases, technically incorrect according to strict thermodynamic definition of the word *heat*. From the early 1800s, it took the scientific community a century to discard the incorrect *caloric* scientific theory of the period in which heat was thought to be a mysterious caloric substance or fluid without mass. According to the caloric theory, heat (the mysterious calor) could be stored in a substance.

The scientific community now rigidly defines *heat* as a form of energy crossing a system boundary from the *surroundings* to the *system* due to a temperature difference. *Heat* is like *work* in that it can only cross a system boundary—heat is not a form of energy stored in a substance.

Now, in most processes, sensible and latent enthalpy changes involve heat transfer to or from an air mass across a boundary due to a temperature difference, so there is some justification for the word *heat*. However, there are also the adiabatic processes of evaporative cooling and, its reverse, adiabatic chemical dehumidification. These two processes occur with no heat transfer across a system boundary (using the apparatus casing as the system boundary). The adiabatic evaporative cooling process involves a decrease in sensible enthalpy, which is exactly offset by an increase in latent enthalpy. In other words, the exchange occurs within the air mass and without heat transfer across a boundary. These processes are adiabatic, meaning there is no transfer of heat across a boundary. For these two processes, the word *heat* is an oxymoron because in an adiabatic process, by definition, there can be no heat transfer. A clever way around this dilemma is to designate the boundary as the film of saturated air surrounding every water droplet. Because of long usage in psychrometrics, the terms *sensible heat* and *latent heat* will no doubt continue for adiabatic and all other processes.

The use of the term *heat* in connection with stored energy will probably be forever debated by practitioners and thermodynamics purists. Some tolerate the terms *sensible heat* and *latent heat* but, at this point, they occasionally draw the line. The sum of sensible heat (sensible enthalpy change) per unit of dry air mass and latent heat (latent enthalpy change) per unit of dry air mass would seem by most to equal the total heat (total enthalpy change) per unit of dry air mass, but it is here that the scientific community discourages use of the term *total heat* and recommends instead that it be called the *total change in specific enthalpy* or the *total enthalpy change*.

Sensible Heat Ratio and Enthalpy Difference to Humidity Ratio Difference

These two ratios have been primarily used in the era of manual psychrometric calculations as an aid in plotting process lines on a psychrometric chart. (Please note that the term *heat* is incorrect but its use is well ingrained, as noted above.) In one sense, these paragraphs may be historical because the need for these ratios vanishes with the use of graphic psychrometric software. Many psychrometric charts include instructions on the front or back of the chart for plotting these ratios.

Either of these ratios defines the slope of a process line on a psychrometric chart. Many charts include a nomograph that aids in plotting the process line. By knowing (1) either the beginning or ending statepoint of a psychrometric process and (2) either the *SHR* or the $\Delta h/\Delta W$ ratio, which defines the characteristics of the process, the process line can be plotted. The unknown statepoint can be at any location on the plotted process line with the understanding that the required mass flow of *dry air* will increase as the distance between the beginning and ending statepoints is decreased.

The **sensible heat ratio in a process** is the ratio of the sensible enthalpy difference to the total enthalpy difference in the process. The term can be applied to the *room effect* process, in which case it is often called the *room sensible heat ratio*. It can also be applied to the *cooling and dehumidifying coil process* or any other process. Some cooling equipment manufacturers list sensible heat ratio along with total cooling capacity for their equipment at various entering air conditions and air quantities. Many psychrometric charts include a sensible heat ratio scale and an index point somewhere in the body of the chart.

$$SHR = \Delta h_{SEN} / \Delta h_{TOT}$$

The **enthalpy difference to humidity ratio difference in a process** is self-defining. It is a ratio used for the same purpose as the sensible heat ratio. Practitioners who work in the humidification field generally choose to use this ratio in lieu of sensible heat ratio. The ASHRAE psychrometric charts include a protractor scale diagram with enthalpy humidity ratio plotted on the outside of the protractor and sensible heat ratio plotted on the inside.

$$q' = \Delta h / \Delta W$$

Equations

One set of equations is utilized for solving mixing processes. A second set of equations is used for solving heating (cooling) and humidifying (dehumidifying) processes. Only one form of each equation is shown in which heat addition is positive and *water* addition is positive. These same equations apply to a cooling process, which is simply negative heating, and also to a dehumidification process in which the mass of water vapour change is negative.

Psychrometric processes are assumed to occur at constant barometric (i.e., total) pressure.

Mixing Process Equations

The following equations are stated for two entering streams and one leaving stream. Given sufficient input data, these equations may be solved to determine the missing properties.

$$m_{DA1} \cdot h_1 + m_{DA2} \cdot h_2 = m_{DA3} \cdot h_3 \qquad \text{(conservation of energy)}$$

$$m_{DA1} + m_{DA2} = m_{DA3} \qquad \text{(conservation of the mass of dry air)}$$

$$m_{DA1} \cdot W_1 + m_{DA2} \cdot W_2 = m_{DA3} \cdot W_3 \qquad \text{(conservation of the } H_2O \text{ mass)}$$

Heating (Cooling) and/or Humidification (Dehumidification) Process Calculations

Most psychrometric systems involve a series of individual processes. Equations are solved for each individual process in series, usually starting with the first process.

There are two sets of equations for each individual process. The first set solves for the heat transfer to or from the heat transfer fluid circulating through the heating or cooling coil (Figure 20-3). The second set of equations solves for the heat transfer into or out of the air (Figure 20-4). The first set is important because it represents the refrigeration or chilled water capacity required. The second set is useful for checking the "air side" capacity of an individual unit.

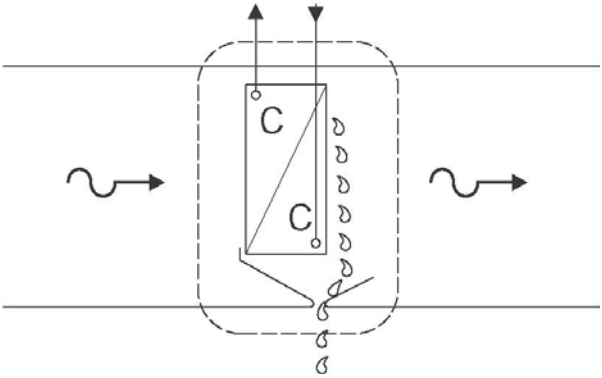


Figure 20-3—"System" is the apparatus casing.

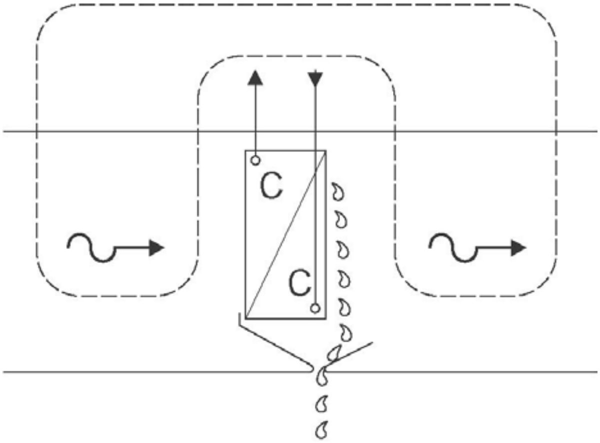


Figure 20-4—"System" is the entering and leaving air.

The difference between the heat transfer by the cooling coil (first set) and the heat transfer into or out of the air (second set) is the enthalpy difference associated with the added or removed H_2O. The enthalpy of H_2O is measured from a reference datum of liquid water at 0°C. Assume a cooling and dehumidifying process with the liquid condensate, leaving the drain pan at 10°C. The enthalpy of the leaving H_2O usually represents an insignificant amount (1.5% to 3%) of the total enthalpy difference between the entering and leaving air. In the era of manual calculations, the enthalpy of liquid water leaving a cooling coil was often ignored. If the H_2O enters or leaves the system in the form of ice, hot water, or steam, this term is more significant and should not be ignored.

Equations in Which the System (or Control Volume) is the Complete Apparatus

$$m_{DA1} \cdot h_1 + m_{H2O} \cdot h_{H2O} + q_{COIL} = m_{DA2} \cdot h_2 \quad \text{(conservation of energy)}$$

$$m_{DA1} = m_{DA2}$$

and by substituting and rearranging:

$$q_{COIL} = m_{DA} \cdot (h_2 - h_1) - m_{H2O} \cdot h_{H2O}$$

$$m_{DA1} \cdot W_1 + m_{H2O} = m_{DA2} \cdot W_2 \quad \text{(conservation of H2O mass)}$$

and by substituting and rearranging:

$$m_{H2O} = m_{DA}\ (W_2 - W_1) \quad \text{H2O added (or removed)}$$

Equations in Which the System (or Control Volume) is the Air Only

$$m_{DA1} \cdot h_1 + q_{AIR} = m_{DA2} \cdot h_2 \quad \text{(conservation of energy)}$$

$$m_{DA1} = m_{DA2}$$

and by substituting and rearranging:

$$q_{AIR} = m_{DA} \cdot (h_2 - h_1)$$

$$m_{DA1} \cdot W_1 + m_{H2O} = m_{DA2} \cdot W_2 \quad \text{(conservation of H2O mass)}$$

and by substituting and rearranging:

$$m_{H2O} = m_{DA}\ (W_2 - W_1)$$

Iterative or Looping Calculations for Psychrometric Cycles

Iterative or looping process calculations involve a *cycle* of sequential psychrometric processes in which, by definition, the ending statepoint properties of the last process in the cycle are the same as the beginning statepoint properties of the first process of the cycle. Iteration is required in a cycle of processes when some factor(s) in the series of sequential processes causes the cycle ending statepoint to not be the same as the cycle beginning statepoint. Thus, to satisfy the definition of a cycle, additional cycle calculations (iterations) are required with a new beginning statepoint. The cycle must be reanalyzed until the beginning and ending statepoints are essentially the same.

An example of a cycle that requires iteration at full load is an ordinary air-conditioning system in which the designer arbitrarily fixes the coil leaving dry-bulb temperature rather than fixing it at the intersection of the coil process line and the room sensible heat ratio line. In this hypothetical example, the air-conditioning load (heat gain) calculations are made for room conditions of 24°CDB, 50% RH. For purposes of discussion, simplify the problem and assume that (1) calculations result in a room sensible heat ratio of 0.75 (room sensible heat gain to total room heat gain), (2) outside air for the space is supplied at neutral conditions by another system, (3) the fan heat is ignored for the purpose of simplifying the example, and (4) the cooling coil supplied for this project has heat transfer characteristics that result in the air leaving the cooling coil having a relative humidity of exactly 90%.

In this example, the *psychrometrically ideal* cooling coil process line will begin at 24°CDB, 50% RH and end at the point where the curving coil process line intersects the *room effect* line at point 2 of Figure 20-5. Air supplied at point 2 will exactly satisfy the room sensible and latent heat gains. So far everything looks simple and direct.

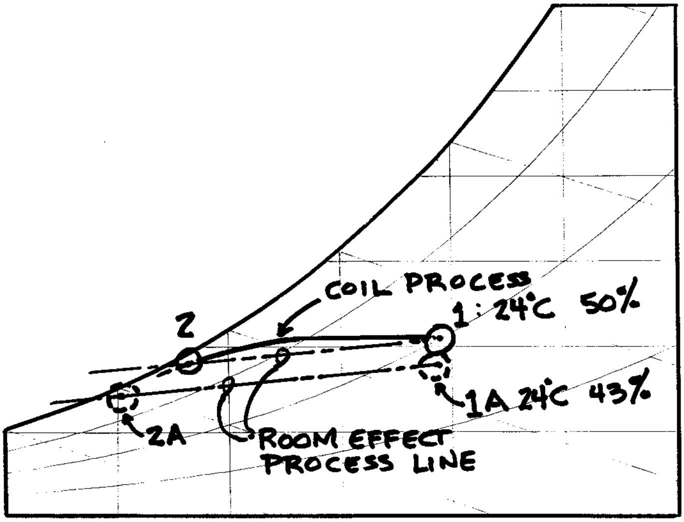


Figure 20-5—Iterative looping calculations.

There are often significant first and operating cost benefits in supplying colder air to the space than indicated by point 2 because colder air reduces the size of the air-handling equipment and ducts, reduces fan power, and results in lower room relative humidity, which may be beneficial from comfort and productivity standpoints. Of course a lower supply air temperature, point 2A, requires more cooling coil surface and/or lower chilled water or refrigerant temperature. The point of this example is not to debate the merits of this change but to illustrate that iterative or looping calculations are required.

The *room effect* line starts at the supply air statepoint, which, in our example, is the coil leaving air relative humidity of 90% and the new (usually lower) supply air dry-bulb temperature. The slope of the *room effect* line remains at 0.75 because the room sensible and latent heat gains are unchanged. The *room effect* process ends at 24°CDB, point 1A, which has a lower RH than the room design condition of 50%. This then becomes the new coil entering condition and the resultant coil process curve will be slightly different from the original curve and, in this case, assume that the relative humidity leaving the coil at the new supply air dry-bulb is 88%. The new supply air condition is point 2B (2B is not shown but is approximately midway between points 2 and 2A). After a number of iterations, there will be little change in points 1A and 2B and these should be the final conditions scheduled for the performance of the cooling coil.

The iteration becomes more complex in applications involving fan heat and/or a fixed amount of outside air. It should be obvious that the space will end up at less than 50% RH at 24°C conditions because the supply air humidity ratio is lower than required by the *room effect* process line.

The solution to cycle processes requires a series of looping or iterative psychrometric calculations, which progressively change the room RH (or some other parameter) in 1% or smaller increments until the assumed value equals the result obtained by the cycle of process calculations.

As an aside, at this point some readers may question the use of colder than "psychrometrically exact" supply air because it (1) implies an increase in total cooling load and (2) requires an increase in coil heat transfer surface, a decrease in water or refrigerant temperature, and/or an increase in water or refrigerant flow. Point two is correct; however, as to point one, in the majority of cases there is no total cooling load increase in supplying colder air because the increase in outside air dehumidification load due to the resultant lower room humidity ratio (or dew-point temperature) is (by coincidence) approximately the same as the decrease in fan heat load because of the reduction in fan air quantity.

Experienced designers intuitively can usually guess or "plug" the final room RH within 1% or 2% and often make their load calculations assuming a lower room relative humidity. Some integrated heat gain psychrometric software calculation programs perform the iterative calculations automatically if a coil leaving air temperature or room supply temperature is specified.

Part-load operating conditions in many air-conditioning systems commonly require iterative solutions because the room sensible heat ratio generally decreases at part-load conditions (in many applications the room latent load stays relatively constant; therefore, as the sensible heat gain to the room declines, the room sensible heat ratio declines). In many of these systems, the room temperature is sensed and the system is controlled to produce the design temperature; but the room relative humidity is not directly controlled, and, thus, the actual room relative humidity may be higher or lower than the design value. The system may achieve the design room relative humidity at full load; however, at part-load conditions, the room sensible heat ratio is often significantly less than at full load and, as a consequence, the iterative psychrometric analysis of the series of sequential processes shows a higher room relative humidity than used as the beginning statepoint.

Figure 20-6 is an example of a cycle that satisfies full-load conditions but requires part-load iteration to solve for the resulting humidity level.

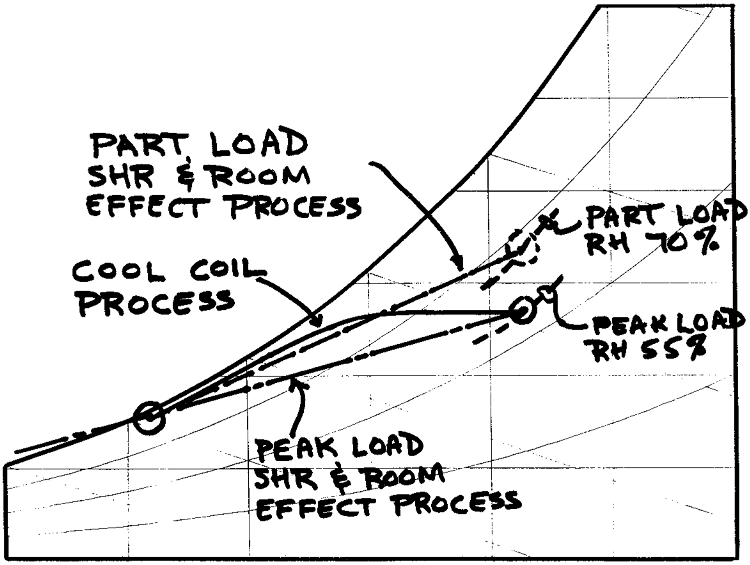


Figure 20-6—Part-load psychrometric analysis.

Psychrometrics is used in the conversion of cooling and heating loads into coil entering and leaving conditions and air quantities which are then used in duct design.
—William Rudoy

21 Why Do Air-Conditioning Engineers Need Psychrometrics?

PSYCHROMETRICS ⇔ LOAD CALCULATIONS

For the four- to six-hour basic psychrometrics, read this chapter in its entirety.

This chapter uses a simplified summer air-conditioning example to demonstrate how an air-conditioning designer uses psychrometrics and load calculations to determine equipment performance requirements. The number of significant digits in the example is more than customary but is provided herein for readers wishing to check accuracy.

The basic function of an air conditioner (in the cooling mode) is to remove sensible heat and water vapour from the room at the same rate at which sensible heat and water vapour are added. The air conditioner does this by circulating room air (in many systems mixed with outside air) across a cooling coil, which transfers sensible heat and water vapour from the circulating air.

Ventilation standards and codes mandate the delivery of outside air to each room to dilute and remove volatile organic compounds, odors, and carbon dioxide and to replenish oxygen. In order to keep the example problem simple, we stipulate that the outside air is supplied to the room at room temperature and room humidity ratio by an independent system.

Load Calculation

To select or specify an air conditioner it is first necessary to calculate the rate at which sensible heat and water vapour are added to the room. This is traditionally done for the one hour during which peak gains occur. Our example involves a junior ballroom in the interior of a hotel. This particular room has no external walls and the temperature of surrounding spaces is maintained by other systems at the same temperature as the junior ballroom. Thus, there is no heat transfer across the room boundaries (sometimes called the *room enclosure*). The load estimate is shown in Table 21-1. Also for this example we stipulate zero heat gain or loss in the supply and return ducts, and fan heat gain is included in the lighting, appliance, and motors component of the room load.

Table 21-1—Air-Conditioning Load Estimate

Load Component	kW_{GAIN}
Wall and glass (transmission and solar)	0
Roof and floor	0
Lighting, appliances, and motors	4.75
People – sensible: 70 @ 0.075 kW	5.25
Subtotal room sensible heat gain	10.0
People – latent (water vapour): 70 @ 0.055 kW	3.85
Total room load	13.85
Outside air load	0
Grand total system load	13.85

The room is to be maintained at 24°CDB and 50% RH. Three psychrometric properties—(1) sea level location with a barometric pressure of 101,325 Pa, (2) dry-bulb temperature, and (3) relative humidity—fix the room statepoint. For reference, other psychrometric statepoint properties are: $t_{DP} = 12.946$°C, $t_{WB} = 17.057$°C, $h = 47.911$ kJ/kg_{DA}, $v = 0.8545$ m^3/kg_{DA}, and $W = 0.009341$ kg_{WV}/kg_{DA}.

The air-handling unit will circulate m_{DA} kilograms of air per second across the cooling coil, then to the junior ballroom where the air will be warmed and humidified by the load (lights, appliances, motors, and people), after which the air will return back to the air conditioner to complete the cycle.

Air Quantity Calculation

The following equation is used to calculate the air quantity m_{DA}:

$$m_{DA} = q_{SEN} /(C_P \cdot \Delta t_{DB})$$

where

m_{DA} = mass of air circulated across the cooling coil and through the room in kg_{DA}/s

q_{SEN} = room sensible heat gain with units of kW (= kJ/s)

Cp = the specific heat capacity of air with units of kJ/(kg_{DA}·°C), where $Cp = 1.006 + W_{AVG} \cdot 1.84$

Δt_{DB} = temperature difference in degrees Celsius between the air leaving and entering the room; in this example Δt_{DB} is also equal to the temperature difference between coil entering and leaving air

W_{AVG} = average humidity ratio of room air and supply air

In order to calculate *Cp* for this example we need W_{AVG}. Many psychrometric process software programs will iterate to find this value. For our example, we will assume $W_{AVG} = 0.008\ \mathrm{kg_{WV}/kg_{DA}}$. Therefore, *Cp* = 1.02056 kJ/($\mathrm{kg_{DA}}$·°C). [The *Cp* value changes slightly from 1.02238 kJ/($\mathrm{kg_{DA}}$·°C) at 0.009 $\mathrm{kg_{WV}/kg_{DA}}$ to 1.01874 kJ/($\mathrm{kg_{DA}}$·°C) at 0.007 $\mathrm{kg_{WV}/kg_{DA}}$)]. While the slightly greater accuracy of an iterative determination of *Cp* might be desired, it must be recognized that the maximum error in *Cp* for this problem is less than 0.2%. Given that the variance in load estimates can be as much as ±10%, an error of 0.2% for *Cp* is insignificant.

The Δt_{DB} value depends on the equipment heat exchange characteristics and the psychrometric limitations of the process. In order to simplify the problem, it is stipulated or assumed that the Δt will be a whole number value in the range of 5°C to 15°C. At 10°C Δt_{DB}, the calculation is

$$m_{DA} = (10\ \mathrm{kJ\ s^{-1}})/(1.02056\ \mathrm{kJ/kg_{DA}{}^{\circ}C} \cdot 10^{\circ}\mathrm{C}) = 0.9799\ \mathrm{kg_{DA}/s}.$$

Air Quantity in Volumetric Flow vs. Mass Flow

Air-conditioning designers and manufacturers traditionally list the flow rate in cubic metres per second rather than kilograms per second. In order to convert from the mass flow rate in kilograms per second to the volumetric flow rate in cubic metres per second, the former must be multiplied by the specific volume of the flowing fluid.

$$Q = m_{DA} \cdot v \qquad (\mathrm{m^3/s}) = (\mathrm{kg_{DA}} \cdot \mathrm{s^{-1}} \cdot [\mathrm{m^3/kg_{DA}}])$$

In this example, a specific volume (ASHRAE standard air) of 1 $\mathrm{m^3}$/1.204 $\mathrm{kg_{DA}}$ is used and the required volumetric flow rate is 0.8138 $\mathrm{m^3/s}$. Table 21-2 shows the mass and volumetric flow rates with changes in Δt_{DB} from 15°C to 5°C.

Each of the table flow rate and Δt_{DB} combinations provides the sensible cooling necessary to offset the 10 kW sensible heat gain in the room. Obviously, in looking at the equation and the table, a Δt of 5°C requires twice the airflow of a 10°C Δt. The 10°C Δt requires smaller air-handling and duct components, but more coil surface and probably lower temperature coil-heat-transfer fluid is required to achieve the 10°C Δt_{DB}.

Add Latent Heat Gain to the 10 kW Sensible Heat Gain

The latent heat gain generated in the space by 70 people, each releasing 55 W latent heat, totals 3.850 kW. The 55 W of latent heat can also be expressed as the release of 0.000 0225 $\mathrm{kg_{H2O}}$ per second per person or 0.001575 $\mathrm{kg_{H2O}}$ per second total for the 70 people. This is the m_{H2O} of water vapour added to the space as a result of respiration and perspiration.

Table 21-2—Mass and Volumetric Flow Rates for a 10 kW Sensible Heat Gain

Δt_{DB}, °C	t_{Coil} Lvg, °C	m_{DA}, kg/s	Q_{STD}, m³/s	Q_{ACT} $CoilLvg$, m³/s
15	9	0.6532	0.5426	0.5280
14	10	0.6999	0.5813	0.5678
13	11	0.7537	0.6260	0.6138
12	12	0.8165	0.6782	0.6675
11	13	0.8908	0.7398	0.7309
10	14	0.9799	0.8138	0.8070
9	15	1.0887	0.9043	0.9000
8	16	1.2248	1.0173	1.0163
7	17	1.3998	1.1626	1.1658
6	18	1.6331	1.3564	1.3652
5	19	1.9597	1.6277	1.6442

The problem is now more complex because the combination of flow rate and Δt_{DB} required for sensible heat removal must also condition that air with a humidity ratio sufficiently lower than the room humidity ratio so that the air will offset both the 10.0 kW sensible heat gain and the 3.850 kW latent heat gain (m_{H2O} = 0.001575 $\mathrm{kg_{H2O}}$) in the room.

The ΔW required at each of the mass flows (m_{DA}) in Table 21-2 is determined using the following equation:

$$\Delta W = m_{\mathrm{H2O}}/m_{DA} \qquad [(\mathrm{kg_{H2O}} \cdot \mathrm{s}^{-1})/(\mathrm{kg_{DA}} \cdot \mathrm{s}^{-1})] = (\mathrm{kg_{H2O}}/\mathrm{kg_{DA}})$$

The supply air humidity ratio (leaving the cooling coil) for a dry air mass flow rate of 0.9799 $\mathrm{kg_{DA}}$/s is calculated below. Recall that this is the mass flow rate for a Δt_{DB} of 10°C.

$$W_{COIL\ LVG} = W_{COIL\ ENT} - \Delta W\ (\mathrm{kg_{WV}}/\mathrm{kg_{DA}})$$

$$W_{COIL\ LVG} = 0.009341 - 0.001575\ \mathrm{kg_{WV}}/0.9799\ \mathrm{kg_{DA}}$$
$$= 0.007733\ \mathrm{kg_{WV}}/\mathrm{kg_{DA}}$$

Table 21-2 can now be expanded to include the latent load effects.

The 11 coil leaving air statepoints and the room statepoint are plotted on the skeleton psychrometric chart (see Figure 21-1). Note that the 11 statepoints and the room statepoint are on the same line. This line is called *the room condition line*.

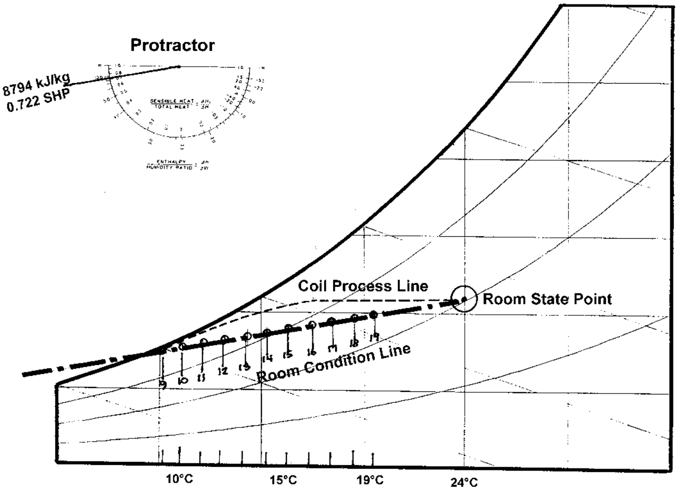


Figure 21-1—Coil leaving statepoint properties.

Graphic Solution with the Psychrometric Chart Protractor

The calculation of 11 statepoints to establish the room condition line is somewhat cumbersome. For graphic solutions, easier methods are available to establish this line. In the past when most problems like this were graphically solved using the psychrometric chart, there were two solution techniques employed (either technique sets the slope of the room condition line, which is then drawn through the room statepoint):

- The total enthalpy difference/total moisture difference ratio technique can also be expressed as the specific enthalpy difference divided by the humidity ratio difference. For this example:

$$q' = \Delta H/\Delta W = 13.85\ \text{kW}/0.001575\ \text{kg}_{H2O} \cdot \text{s}^{-1} = 8794\ (\text{kJ/kg}_{H2O})$$

- The sensible heat ratio technique. For this example:

$$\text{SHR} = q_{SEN}/q_{TOT}$$

$$SHR = 10.0\ \text{kW}/13.85\ \text{kW} = 0.722\ \text{(a ratio of kW/kW)}.$$

Both techniques produce satisfactory results. The first is theoretically more accurate because it is a ratio that uses the two psychrometric chart property plotting coordinates. These solution methods often used a psychrometric chart plotting protractor with both a $\Delta h/\Delta W$ and a sensible heat ratio (*SHR*) scale. The *SHR* scale on the protractor or the *SHR* scale on an ASHRAE-type or Mollier psychrometric chart is precisely accurate at only the one evaporation (or condensation) temperature used in constructing the *SHR* scale for the psychrometric chart in use.

Table 21-3—Flow Rates and Coil Leaving Statepoint Properties

Δt_{DB}, °C	t_{Coil} Lvg, °C	m_{DA}, kg/s	Q_{STD}, m^3/s	Q_{ACT} CoilLvg, m^3/s	ΔW, kg_{H2O}/kg_{DA}	W_{Coil} Lvg, kg_{H2O}/kg_{DA}	t_{DP} CoilLvg, °C
15	9	0.6532	0.5426	0.5280	0.002412	0.006929	8.522
14	10	0.6999	0.5813	0.5678	0.002251	0.007090	8.857
13	11	0.7537	0.6260	0.6138	0.002090	0.007251	9.185
12	12	0.8165	0.6782	0.6675	0.001929	0.007412	9.507
11	13	0.8908	0.7398	0.7309	0.001768	0.007573	9.822
10	14	0.9799	0.8138	0.8070	0.001608	0.007733	10.132
9	15	1.0887	0.9043	0.9000	0.001447	0.007894	10.436
8	16	1.2248	1.0173	1.0163	0.001286	0.008055	10.734
7	17	1.3998	1.1626	1.1658	0.001125	0.008216	11.028
6	18	1.6331	1.3564	1.3652	0.000965	0.008376	11.316
5	19	1.9597	1.6277	1.6442	0.000804	0.008537	11.599

Both of the graphic methods establish a *room condition line* with an end point at the room statepoint condition. Any point on this line will provide a supply air condition capable of simultaneously offsetting both the room sensible heat gain and the room latent heat (moisture) gain. As shown in Table 21-3, the quantity of supply air increases the closer the supply air condition is to the room statepoint.

Most air-conditioning designers use the *SHR* technique because their load calculations usually display q_{SEN} and q_{TOT}. Most humidification designers use the $\Delta H/\Delta W$ technique because their load calculations usually display m_W and q_{TOT}. The possible inaccuracy of the *SHR* method is insignificant.

Caution—The Coil Leaving Statepoint Must Be Physically Attainable!

While it is true that any supply air condition on the room condition line will satisfy the mathematical equations presented herein, another requisite is that the cooling coil must be physically capable of delivering that exact condition. Refer to Chapter 19, "Psychrometric Processes," and the section on the cooling and dehumidification process for more on this subject.

This simplified chapter covers the equations and concepts. Obviously, in all cases, the cooling coil must be sized to transfer the sensible heat and condense the water vapour from the air flowing across the cooling coil. Also, the air moving equipment must be able to deliver the required air quantity when operating with the installed components and duct system.

Shortcut Calculation Methodology

Many publications include “shortcut” equations for calculating *SI standard* airflow. These are often based on a specific volume of 1 m^3/ 1.204 kg_{DA}, a moist air specific heat of 1.0216 kJ/(kg_{DA} °C), and a latent heat value of 2500 kJ/kg_{H2O}. The moist air specific heat value is representative of air having a humidity ratio of 0.00848 kg_{H2O}/kg_{DA}. The latent heat value represents the refrigeration effect associated with 24°C water vapour less the enthalpy of 10°C condensed water. For processes at 101,325 Pa, these values are reasonably close to actual values for most air-conditioning problems and, therefore, calculations using these values will be reasonably close to the most accurately determined values using the previous equations in this chapter. The “shortcut” equations express the cooling coil heat transfer in kW (kJ/s) and are developed below.

$$q_{TOTAL} = Q \cdot \Delta h / v_{STD} = \mathbf{1.204}\ \mathrm{kg_{DA}/m^3} \cdot Q \cdot \Delta h \qquad (\mathrm{kW = kJ/s})$$

$$q_{SEN} = Q \cdot \mathbf{1.204}\ \mathrm{kg_{DA}/m^3} \cdot 1.0216\ \mathrm{kJ/kg_{DA}\ ^\circ C} \cdot \Delta t_{DB}$$

$$q_{SEN} = 1.23 \cdot Q \cdot \Delta t_{DB} \qquad (\mathrm{kW = kJ/s})$$

$$q_{LAT} = Q \cdot \mathbf{1.204}\ \mathrm{kg_{DA}/m^3} \cdot \Delta W\,(\mathrm{kg_{WV}/kg_{DA}}) \cdot 2500\ \mathrm{kJ/kg_{H2O}} \quad (\mathrm{kW = kJ/s})$$

$$q_{LAT} = \mathbf{3010} \cdot Q \cdot \Delta W \qquad (\mathrm{kW = kJ/s})$$

If ΔW is expressed in units of g_{WV}/kg_{DA} this equation changes to:

$$q_{LAT} = Q \cdot \mathbf{3.010} \cdot \Delta W \qquad (\mathrm{kW = kJ/s})$$

Some texts round the bolded equation constants. Users of shortcut equations must understand the underlying assumptions and must ensure that the assumptions fit their specific application.

Endnote Quotation

A quotation from *Principles of Heating, Ventilating and Air Conditioning* (Sauer and Howell 1989) restates the principles of this chapter using slightly different language:

Cooling Coil Load from Space Cooling Load. *In order to properly design and size either all-air or air-and-water central air conditioning systems, it is not enough to calculate the cooling load in the space to be conditioned. Such items as fan energy, location of fans, duct heat gain and loss, duct leakage, heat extraction lighting systems, and type of return air system all affect component sizing, as well. The only adequate way to design a system and properly size the components is to analyze the system performance as a series of psychrometric processes. In most cases, it requires defining all known statepoints on a psychrometric chart and then based on the zone or space load, amount of outside air introduced into the system, and size of load due to the other items described above, solve for the entering and leaving conditions for the pertinent components such as the cooling and/or heating coils.*

Processes in Meteorology

When we compress a gas slowly, the temperature of the gas increases.
So, under slow compression, a gas will increase in temperature,
and under slow expansion, a gas will decrease in temperature.
—Richard P. Feynman

22 Adiabatic Expansion and Adiabatic Compression

Adiabatic Expansion—Accompanied by Temperature Decrease

When a gas expands due to lower pressure *surroundings,* with no heat transfer to or from the gas (adiabatically), the gas experiences a temperature decrease. The *flow work energy* transferred from the expanding volume of gas (the *gas* is the *system* in this case) to the *surroundings* is at the expense of the *internal energy* of the gas; therefore, in the adiabatic expansion of a gas caused by a lowered pressure, the volume of the gas (the *system*) expands and its temperature is lowered as it pushes back the *surroundings* to make room for its expansion. One example of the effect of adiabatic expansion occurs when air is let out of a tire. The air flowing through the valve expands and *does work in pushing aside some of the surrounding atmosphere* formerly occupying that volume. The temperature of the expanding gas decreases.

Some meteorology texts label this process *adiabatic expansional cooling* (and the opposite process, *adiabatic compressional heating*). To a thermodynamicist, this is unfortunate word choice and an oxymoron because the process is adiabatic, which means no external heat transfer, yet to the thermodynamicist the word *cooling* (or its opposite, *heating*) means that heat is transferred from the air mass. In everyday context, the term *cooling* means *lowering of temperature*.

A primary means of cumulus cloud formation in the atmosphere is by the process of adiabatic expansion. As an air parcel ascends in the atmosphere, it expands due to lower surrounding pressure and experiences a temperature decrease. The temperature change will be approximately –10°C per 1000 m (*the dry air lapse rate*) until the temperature of the parcel reaches the dew-point (saturation) temperature of the *water vapour* in the parcel. As the parcel ascends higher, condensation of a portion of the *water vapour* occurs. Rising air saturated with *water vapour* experiences a temperature change known as the *moist adiabatic lapse rate,* which averages –6.5°C per 1000 m in the earth's troposphere.

It may be difficult to think of a large mass of air expanding or contracting and not mixing with adjacent air masses. Imagine a monstrous balloon with a skin that can stretch and shrink with no associated tension or force. If that balloon is lifted several thousand metres, the air within the balloon will expand and *flow work* will be accomplished as some of the *surrounding* atmosphere is pushed aside. The *internal energy* and the temperature of the air in the balloon will decrease.

Adiabatic Compression—Accompanied by Temperature Increase

This process is the reverse of adiabatic expansion. Adiabatic compression results in an increase in temperature when a volume of air contracts because of increased pressure of the *surroundings*. The *flow work* of compressing the air is converted into *internal energy* of the air mass and, thus, a temperature rise. One example of adiabatic compression is the bicycle tire pump, which experiences warming of the pump's cylinder wall mainly due to adiabatic compression of the air. The temperature rise accompanying the adiabatic compression process is also evident in air compressors, refrigerant compressors, and fans.

The Chinook ("snow eater") winds of the Rocky Mountains, the Santa Ana winds of California, and the foehn winds of Germany and Austria are meteorological examples of temperature rise accompanying adiabatic compression. In these cases, a low pressure instability pulls air down the leeward slopes of mountain ranges. A Chinook wind descending on Pincer Creek, Alberta, Canada, resulted in a temperature increase of 21°C in only four minutes! In the Great South Dakota Chinook*,* many plate glass windows cracked when blistering Chinook winds descended on the town of Spearfish in the early morning hours and raised the temperature from –20°C to +7°C in two minutes. This warm and dry wind has been known to melt 500 mm of snow in a few hours (hence the expression "snow eater"). Boulder, Colorado, experiences about $1 million of property damage each year because of these Chinook winds, which gust up to 160 km per hour.

As an air parcel descends in the atmosphere, it is compressed and experiences a temperature rise. In meteorology, the rate of temperature rise is known as the *dry adiabatic lapse rate* and has an approximate value of 10°C per 1000 m. Chapter 26 of the *1999 ASHRAE Handbook—HVAC Applications*, "Mine Air Conditioning and Ventilation," states that air will increase in temperature 1°C for every 102.48 m as it descends a mine shaft, which is equivalent to 9.757°C per 1000 m (the *dry air adiabatic lapse rate*).

The town of Kopperl, Texas, may hold the record high temperature accompanying adiabatic compression. On the night of June 14, 1960, the temperature rose almost instantly to near 60°C due to a sudden downblast of air produced by a distant thunderstorm. The colorful local reports of this adiabatic compression phenomena indicate that "ears of corn were roasted on their stalks!"

Endnote Quotation

As air rises, pressure on it decreases and in response it expands. The act of expansion to encompass (fill) its new and larger dimensions requires an expenditure of energy; since temperature is a measure of internal energy, this use of energy makes its temperature drop. A mass of rising air—whether large and rising gently or small and rising abruptly—cools (at an average moist air adiabatic lapse rate of 6.5°C per kilometre.) Conversely, sinking air is compressed and warms (at the dry adiabatic lapse rate of 10°C per kilometre). This activity is of great meteorological significance: it determines the altitude at which a cloud will form or evaporate. (Vincent J. Schaeffer, Director Emeritus, Atmospheric Sciences Research Center, State University of New York, 1981.)

Additional Information

There is a natural internal energy gain from compression
as the pressure increases and a corresponding internal energy loss
in adiabatic expansion as the pressure decreases.
—G.E. McElroy, Bureau of Mines (updated)

The heat from return-air or supply-air fans goes into the refrigeration load.
This energy is not part of the room sensible heat with the exception of
supply fans downstream of the conditioning apparatus.
—p. 11.6, 1999 ASHRAE Handbook—HVAC Applications

23 Fan Temperature Rise

WHERE DOES FAN WORK ΔT OCCUR?

Background

Dry-bulb temperature and humidity properties are required at the following system locations in order to properly design and specify air-conditioning equipment: (1) at the supply air outlet as the air enters the room to be conditioned (this together with room temperature and room sensible heat gain determine the mass flow rate of the air required to cool the room), (2) at the cooling coil inlet, and (3) at the cooling coil outlet.

If the cooling coil is located downstream of the supply air fan (often called a blow-through arrangement), the designer must know the dry-bulb temperature at the fan inlet and the dry-bulb temperature of the fan discharge air as it reaches the cooling coil inlet. These determinations should be straightforward using thermodynamic principles.

Unfortunately, confusion exists and many designers, instructors, and even some manufacturers in their load and energy analysis software depart from thermodynamic principles when it comes to determining where fan energy (work) input is revealed as a temperature rise in the system.

For purposes of discussion, most readers will agree that the temperature rise associated with motor, drive, and bearing losses occurs at the location of the loss. Also, *real* fans depart from reversible adiabatic (isentropic) compression. The temperature rise associated with this departure occurs at the fan. We make two other assumptions: velocity in and out are identical and there are no elevation changes. (Later we test the velocity assumption.) This simplifies the discussion because the only question remaining is: Where does the temperature rise associated with the work of reversible adiabatic compression occur?

The correct thermodynamic answer is presented below—but first, an incorrect answer and its accompanying incorrect explanation: Many claim that the fan work energy input for reversible adiabatic compression is utilized to increase the pressure of the airstream and that the temperature rise associated with this energy is revealed throughout the duct system as friction reduces the pressure and the frictional energy converts to an increase in internal energy and, therefore, a temperature rise. This sounds plausible but is not thermodynamically correct for a gas.

Correct Thermodynamic Analysis

Using the first law of thermodynamics (conservation of energy) for each unit mass of flowing fluid:

$$u_{ENT} - u_{LVG} + \Delta\,(gz/g_C) \cancel{+ \Delta\,(vel^2/(2g_C))} +$$
$$(p \cdot v)_{ENT} - (p \cdot v)_{LVG} = \cancel{q} - w_{SHAFT}$$

Using the stated assumptions and the fact that the q term is zero in an adiabatic process eliminate the terms shown with strikethroughs. Also, $u + p \cdot v = h$ and the equation simplifies to

$$W_{SHAFT} = h_{LVG} - h_{ENT} = c_{P\_AIR} \cdot \Delta t.$$

Thermodynamically, all energy input at the fan shaft is revealed as a temperature increase from the fan inlet to the fan discharge duct. Siméon Denis Poisson's (see Chapter 8, "Psychrometric Pioneers and Charts from the First 100 Years") 17th-century expression relating changes in pressure and temperature for an ideal gas undergoing adiabatic processes ($T \cdot p^{(\gamma - 1)/\gamma}$ = constant) gives the same result and simplifies to

$$T_2/T_1 = (p_2/p_1)^{(k-1)/k},$$

where

T = absolute temperature, Kelvin
p = absolute pressure in Pa
k = C_P/C_V; for air $k = 1.4$

Figure 23-1 shows the temperature entropy plot for the adiabatic reversible (constant entropy) process. For discussion on fans and the temperature entropy plot, see Gerald J. Williams's article, "Fan Heat: Its Source and Significance" (*Heating/Piping/Air Conditioning*, January 1989).

Effect of Velocity Changes

Table 23-1 shows the temperature increase that results when air velocity is reduced from the table velocity to zero. Velocity differences have a small and insignificant effect on the previous analysis.

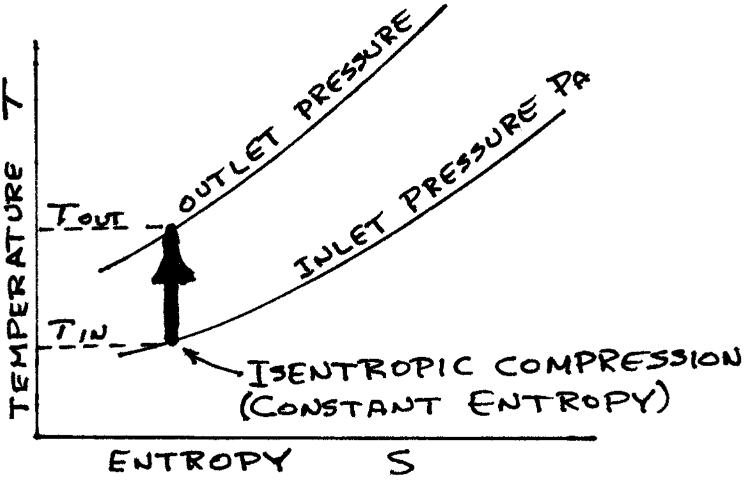


Figure 23-1—Temperature-entropy diagram for air.

Table 23-1—Temperature Rise Associated with the Adiabatic Compression of Air

vel, m/s	vel_{IP}, feet/min	ΔT, °C	ΔT_F, °F	*KE*, J/kg_{air}
20	3937	0.199	0.358	200
18	3543	0.161	0.290	162
16	3150	0.127	0.229	128
14	2756	0.098	0.176	98
12	2362	0.072	0.129	72
10	1969	0.050	0.090	50
8	1575	0.032	0.057	32
6	1181	0.018	0.032	18
4	787	0.008	0.014	8
2	394	0.002	0.004	2

Another Analysis

The following analysis comes from W. Peter Jones, an English consulting engineer and educator (Jones 1973). I personally like the analysis but, in the words of the meteorologist Alistair B. Fraser, this may be a case of *Bad Science* on which Fraser comments, "*the fact that you sometimes will get the correct answer from applying the reasoning is not a vindication of the logic.... Just because you get the correct answer by trying to* *reduce the fraction 16/64 by canceling the two sixes does not vindicate the technique or assure one that it will work under other circumstances. In short, it is so easy to get the right answer for the wrong reason, that one should always be skeptical of any assertion that a correct result implies that the process by which it was obtained was also correct.*" I leave it to future authors of this book to determine the scientific validity of Jones's explanation (stated below in my words).

Perhaps the easiest way to answer the question (*Where is fan energy work input revealed as a temperature rise in the system?*) is to examine where temperature rise does *not* occur. A very long (say 1000 m) constant elevation, constant area, super insulated duct, under steady flow conditions, provides a good example, illustrated in Figure 23-2.

The process is adiabatic because of the super insulation. The mass flow is constant. The duct area is constant. The density of air can be assumed constant and then this assumption can later be tested for sensitivity. With constant mass flow and density, the air velocity is constant; thus, the kinetic energy is constant. If the thermodynamic system boundary is the inner wall of the duct where the air molecules are not moving, then there is no work transferred across the system boundary and, as noted above, there is no heat transferred across the system boundary. Under these steady flow conditions without heat or work transfer, the enthalpy of the air leaving the system and that of the air entering the system are equal. Air at normal pressures behaves as an ideal gas. For an ideal gas, the temperature is dependent on the enthalpy and since the enthalpies are equal, inlet and outlet temperatures of the 1000 m duct are equal.

So what is going on in the duct? Answer: With a little stretch it could be imagined as two concurrent processes.

In the first process there is a layer of stagnant air at the inner duct wall and the velocity increases from zero at the duct wall to a maximum at the center of the duct. Viscous shearing is taking place between the so-called layers of moving air in the duct. Viscous shearing (often referred to as *fric-*

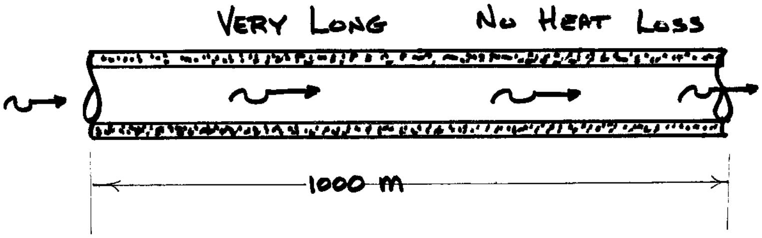


Figure 23-2—Super insulated, constant area, very long duct.

tion) results in a temperature increase and also causes a gradual decrease in pressure and, thus, a decrease in flow work ($p \cdot v$) energy. Applying the conservation of energy law, the decrease in the flow work energy of the fluid is offset by an increase in internal energy. This increase in internal energy should be revealed by an increase in temperature except for the second simultaneous process in this duct. This is discussed in the following paragraph.

One of the effects in the first process is a gradual decrease in pressure. The process involving decrease of pressure for an ideal gas is called *expansional cooling* by meteorologists. As the pressure decreases, the gas expands and its temperature decreases. Meteorologists are quite familiar with this process as it is one of the major ways in which cumulus clouds and thunderstorms develop.

The decrease in temperature accompanying the expansional cooling process equals the increase in temperature due to the viscous shearing process. The combination of these two concurrent processes for a gas results in no change in the temperature of air passing through a long perfectly insulated duct. Analysis of the two concurrent processes produces the same result as that of the previous constant enthalpy analysis.

If there is no air temperature rise in a perfectly insulated duct, then where does the temperature rise from fan energy input occur? Answer: Certainly the inefficiencies of the belt drive, variable speed drive, motor, and fan (including the departure from isentropic compression) occur at the location of the component affected by the inefficiency. Aside from these inefficiencies, the fan imparts energy into the air in the form of increased pressure and, for many fans, increased velocity due to a smaller fan outlet than inlet. The process of increasing the pressure is that of compressional heating (the opposite of the expansional cooling process previously described) and the air experiences a temperature increase. Thus, all of the energy associated with (1) component inefficiencies and (2) increasing the air pressure is revealed as a temperature rise at the location of the fan.

The only energy not accounted for in this analysis is the increase in velocity (kinetic energy) at the fan. In a typical duct system, velocities are reduced as the air passes from the fan to the outlets and the kinetic or velocity energy is first converted to pressure energy, which, in turn, converts to internal energy as it overcomes viscous shearing forces in the duct and, finally, as it pushes aside room air in order to enter the room.

While it is possible to track the static and velocity pressures in the entire supply duct downstream of the supply fan, the following analysis shows that a simpler procedure is adequate. There are only two possible locations for the supply fan with respect to the cooling coil.

1. *The draw-through arrangement* (Figure 23-3), wherein the fan is located downstream of the cooling coil and draws air through the cooling coil. In this arrangement, the majority of the velocity energy is converted to static energy in the duct and the remaining velocity energy is converted to static energy in the room. Designers assign all of the associated temperature rise to the air in the duct since there is nothing to be gained by separately analyzing the gain in the duct or the gain in the room. It is clear that all of the energy input of a draw-through fan and coil arrangement is revealed as a temperature increase downstream of the cooling coil and, for all practical purposes, upstream of the supply air grille.
2. *The blow-through arrangement* (Figure 23-4), wherein the fan is located upstream of the cooling coil and blows or discharges air into the cooling coil. In this arrangement, the relatively high velocity air at the fan outlet is converted to static pressure

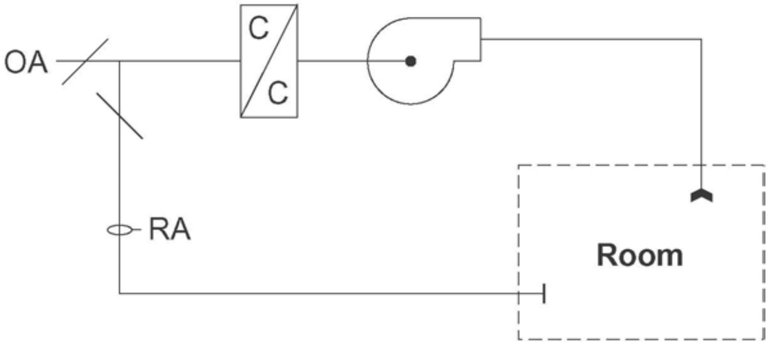


Figure 23-3—The fan “draws” air through the coil.

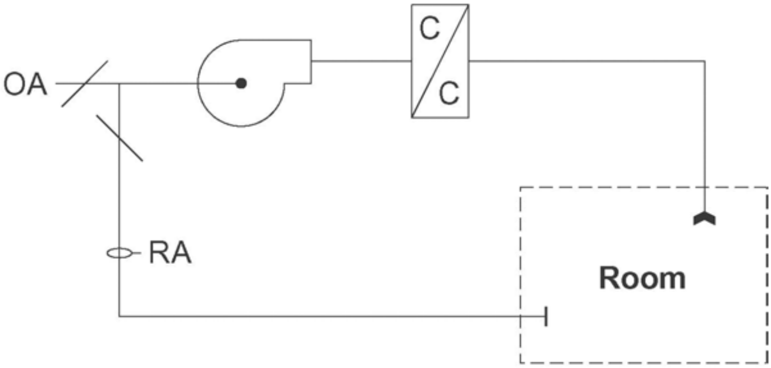


Figure 23-4—The fan “blows” air through the coil.

upstream of the cooling coil, where the air velocity is at a much lower value of ~2.5 m/s. The conversion of velocity energy to static pressure energy is an additional compressional heating process and the temperature of the air entering the cooling coil experiences another temperature rise. The pressure conversion from velocity to static energy is not 100% efficient, and the inefficiency would be revealed as a conversion of part of the energy to internal energy and temperature rise; however, the total temperature rise associated with the conversion of velocity energy to static pressure energy upstream of the cooling coil is the same, i.e., the temperature rise is either all from compressional heating assuming 100% efficiency or part from compressional heating and part from the inefficiency and conversion to internal energy.

The only velocity energy not revealed as a temperature rise upstream of the cooling coil is that energy associated with the velocity energy of the air passing through the cooling coil at a velocity of approximately 2.5 m/s or less. What is the temperature rise associated with this remaining energy component? It is an inconsequential 0.003°C. Therefore, in a blow-through fan and coil arrangement, all of the fan energy input is revealed as a temperature rise upstream of the cooling coil. There is no temperature rise in the duct or in the room associated with a blow-through fan energy input.

There are other examples of temperature rise associated with compressing air, such as the simple tire pump and the industrial air compressor. Another example involves the ground support fan and coil equipment used at many airports to cool airplanes at the terminal. The fan in these units operates at 5000 – 7000 Pa (20 – 30 in. w.g.) in order to force the air through the relatively small ducts in the aircraft cabin. The fan temperature rise is approximately 10°C—an unacceptable rise for a fan located in a draw-through arrangement. This convincingly shows why these units are all of the blow-through type.

For those who wish to delve deeper into the subject of temperature effects in a constant area, perfectly insulated long duct, Fox and McDonald (*Introduction to Fluid Mechanics*, 3rd edition, John Wiley and Sons, 1985) provide an analysis using six simultaneous equations and the Fanno curve for both supersonic and subsonic flows. Their analysis includes the effect of a slight density change accompanying the small change in specific volume of the air in the constant area duct and shows that the density change has insignificant effect for applications in which the change in pressure is less than 5000 Pa.

The believer is happy; the doubter is wise.
—Hungarian proverb

24 Frequently Asked Questions

What is the Relationship between the Pressure and the Specific Volume (the Reciprocal of Density) of a Gas?

In the temperature range of –50°C to +65°C and at the barometric pressures found on earth, both *dry air* and *water vapour* behave essentially as *ideal gases,* which means that the following equations interrelating gas properties are applicable:

$$p_G \cdot v_G = R_G \cdot T_G$$

and since $v_G = V_G/m_G$ it follows that

$$p_G \cdot V_G = m_G \cdot R_G \cdot T_G.$$

Substituting $m_G/V_G = \rho_G$ and rearranging results in:

$$p_G = \rho_G \cdot R_G \cdot T_G.$$

If T_G is constant, then $R_G \cdot T_G$ is a constant in which case p_G is proportional to ρ_G. *Thus, for ideal gases, vapour pressure and vapour density can be said to be two different ways of expressing vapour concentration.*

A historical example is the fact that psychrometric pioneers sometimes defined relative humidity as the ratio of actual water vapour density to saturated water vapour density at the same temperature. Assuming ideal gas behavior, relative humidity is defined by either of the following equations:

$$\text{RH} = 100 \cdot p_{WV}/p_{WVS@}\, t_{DB}$$

$$\text{RH} = 100 \cdot \rho_{WV}/\rho_{WVS@}\, t_{DB}$$

Is It Possible to Have "Supercooled" (Undercooled) Liquid H_2O at Temperatures Less Than 0°C?

Supercooled liquid can exist when a substance remains in its liquid phase at temperatures below its normal freezing temperature. It has particular significance in the field of meteorology. It is known that liquid H_2O can exist in nature at temperatures below 0°C at normal atmospheric pressure. Some meteorologists and researchers report liquid H_2O at temperatures of –20°C or lower. Practitioners using a psychrometer at temperatures below freezing must be aware of this phenomenon (see Chapter 11, "Wet-Bulb Temperature") because different equations apply when the H_2O remains in the liquid phase below 0°C.

All tabulations and algorithms in this text for temperatures below 0°C are based on water vapour pressure over ice.

Psychrometric property and process calculations involving condensation of water vapour in the –20°C to 0°C range first require a knowledge of whether the condensed water vapour is in the liquid or solid (ice or frost) phase. This is particularly significant in calculating the heat removal in these processes. It is also important to conservators in the preservation of waterlogged materials and in the restoration of books and other artifacts damaged by flooding.

Table 24-1—Water Vapour Pressure at Saturation Over Plane Surfaces of Pure Water or Pure Ice

Temperature, °C	Vapour Pressure Over Ice, Pa	Vapour Pressure Over Water, Pa
–20	103.2	125.5
–15	165.3	191.3
–10	259.9	286.4
–9	283.9	309.9
–8	310.0	335.0
–7	338.2	362.0
–6	368.7	390.9
–5	401.7	421.8
–4	437.5	454.8
–3	476.0	490.1
–2	517.7	527.9
–1	562.7	568.2
0	611.2	611.2

This table uses the IAPWS-08 vapour pressure equation along the sublimation line and extrapolates the IAPWS-IF-97 vapour pressure equation along the vaporization line. It is similar to a table in "Building Science for a Cold Climate" (Hutcheon and Handegord), 1983

What Is Fog?

Hopefully it represents but a small part of this book.

Fog is a mixture of (1) *dry air*, (2) *saturated water vapour* (100% RH), and (3) *extremely small particles of suspended liquid phase* H_2O. This can be stated another way by combining (1) and (2) above: Fog is a mixture of 100% relative humidity *moist air* and *extremely small particles of suspended liquid phase* H_2O.

Perhaps the simplest and still accurate description is that fog is *a cloud at ground level.*

Fog is an important meteorological phenomena, the cause of many traffic accidents, and a frequent problem at some airports. Fog in air-conditioning and refrigeration psychrometrics does not command the attention that it gets in meteorology, but it does occur. Examples include cold storage entry vestibules, fog formation at airplane air-conditioning outlets when the airplane is on the ground and loading passengers on a humid day, and in some air-mixing applications when saturated air at, say, 30°C mixes with saturated air at 10°C. Fog may also occur for much the same reason on the downstream side of a blow-through cooling coil. Each of these examples can be predicted by plotting the mixing process on a psychrometric chart and checking to see if the calculated result of the mixing process lies in the *fog region* of the psychrometric chart.

Is the Psychrometric Chart Useful When Fog is Present?

This is a good question. The answer is *YES* and it may lead to an even better understanding of the psychrometric chart. Many charts and, specifically, the ASHRAE psychrometric charts include a shaded region to the left of the saturation curve that is labeled “Fog Region.” The left-hand boundary of the fog region is arbitrary and the fog region could extend even further to the left.

For purposes of example, use an ASHRAE-type psychrometric chart, which is constructed using h and W plotting coordinates.

Since h and W are the chart plotting coordinates, it should be obvious that the enthalpy (h) and the humidity ratio (W) lines cover the entire page, and h and W can be correctly determined for a statepoint in the fog region by extending the lines and reading the appropriate scale value. Note that a humidity ratio value in the fog region represents the sum of (1) the liquid or solid H_2O suspended in the air and (2) the saturated water vapour value as read at the wet-bulb temperature and the saturation curve.

The wet-bulb temperature, t_{WB}, can also be read by extending wet-bulb lines from the body of the chart.

The relative humidity, RH, is 100% because, by definition, RH is the ratio expressed as a percentage of the actual water vapour pressure to the

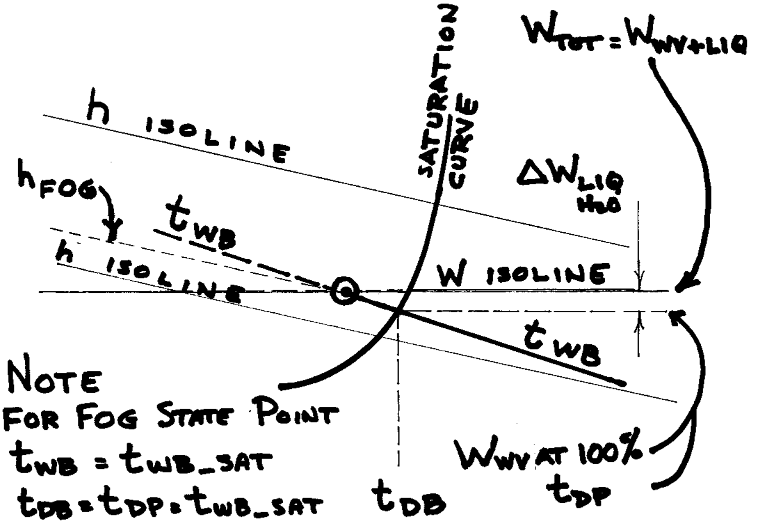


Figure 24-1—Magnified fog region of the psychrometric chart.

saturated water vapour pressure at the same temperature. The addition of finely dispersed, minute H_2O particles in the liquid or solid phase does not change the actual water vapour pressure from its saturated value.

The specific volume, *v*, is for all practical purposes equal to *v* at t_{WB} and *saturation* (i.e., 100% RH) because the liquid phase molecules take up only a small fraction of the space occupied by an equivalent number of gas phase molecules. The same logic can be applied to *water vapour pressure*, p_{WV}, which is essentially equal to p_{WVS} at t_{WB} and *saturation*.

The important distinction to be made about the psychrometric chart and the fog region is that neither dry-bulb temperature (t_{DB}) nor dew-point temperature (t_{DP}) lines extend into the fog region. That is, these two temperature lines stop at the saturation curve. As previously noted, the wet-bulb temperature lines can be correctly extended into the fog region and in this region, as at saturation, $t_{WB} = t_{WB\_SAT} = t_{DB\_SAT} = t_{DP\_SAT}$.

The annotated partial psychrometric chart, Figure 24-1, shows a fog region statepoint and illustrates the correct humidity ratio values for W_{TOTAL}, $W_{MOIST\,AIR}$, and the *mass of liquid or solid H_2O per unit mass of dry air*. The partial chart has been purposefully distorted for the purpose of clearly showing the relative slopes of wet-bulb temperature and specific enthalpy isolines in the fog region.

Does Steam Humidification Increase Dry-Bulb Temperature?

William Goodman, in his 1943 book *Air Conditioning Analysis,* provides an eloquent answer to this question:

> *... if saturated steam at 218°F (2 psig) is added, little change will take place in the temperature of the air. The moisture that is being added to the air has already been evaporated; the latent heat required for this evaporation has been acquired elsewhere. Hence, this case represents nothing more than the mixing of two gases, air and steam.*
>
> *When steam at 218°F is sprayed into the air (at 70°F) and the two are thoroughly mixed, the temperature of the steam will fall and the temperature of the air will rise until both are at the same temperature. During the mixing process the steam surrenders the heat that raises the temperature of the air. Because the weight of steam that can be added to the air without saturating it is very small, the steam can surrender only a little heat. Consequently, the increase in the temperature of the air is very small as long as the air remains unsaturated.*

An excellent means of testing Goodman's claim is to create your own example, calculate the mass of H_2O required to raise a given air volume from 10% to 50% RH, and then, using the energy and mass balance equations of Chapter 20, “Process Calculations and Definitions of Sensible and Latent Enthalpy Change,” calculate the final enthalpy of the air after mixing. Then, using calculations or a psychrometric chart, determine the final temperature of the air.

What Is Apparatus Dew-Point Temperature (ADP)?

ADP is a theoretical temperature used in the era of manually calculating and selecting cooling and dehumidification coils. ADP is sometimes defined as the average temperature of the cooling coil surface and water film at the leaving air side of the cooling coil. If ADP is plotted on a psychrometric chart, it is located on the saturation curve.

Knowing (1) entering air properties, (2) actual coil (entering air) bypass factor, and (3) ADP, it is possible to approximate the coil leaving air properties. Coil bypass factor is unique to each coil depending on coil face velocity, tube arrangement, fin configuration and spacing, number of rows, and wetness over the fin surface. The *estimated* coil selection bypass factor must agree with the *actual* final selection coil bypass factor. If it does not, then a second calculation is required. Since 1990, most cooling coils have been selected using manufacturer’s software, which eliminates the use of ADP and coil bypass factor.

Explain the (Small) Difference in Cooling Coil Sensible Enthalpy Removal Between a Software Load Calculation Program and the Cooling Coil Manufacturer's Tabulated Sensible Enthalpy Removal When Both Have Identical Entering and Leaving Statepoints.

First, this difference should be less than two or three percent, which, for most applications, is insignificant given the approximations in the overall load estimate.

Second, the total enthalpy removed from the air should agree within less than one-half percent. If total enthalpy removed is not in close agreement, the difference may be (1) that the coil manufacturer listed the total enthalpy transferred to the refrigerant, chilled water, or glycol circulating through the tubes —this value is less than the total enthalpy removed from the air by the amount of energy (enthalpy) associated with the (now) condensed water vapour leaving the drain pan ($H_{CONDENSATE} = m_{\mathrm{H2O}\_COND} \cdot h_{\mathrm{H2O}}$) or (2) that the two software calculation programs are not using the same mass flow rate of dry air ($\mathrm{kg_{DA}/s}$). If both programs have identical volumetric flow, then the mass flow rate of dry air should be identical ($m_{DA} = Q/v_{DA}$). Check that both programs are either using *standard* air ($0.83\ \mathrm{m^3/kg_{DA}}$) or *actual* air (specific volume at actual temperature and barometric pressure).

Also take into account that at least one coil manufacturer defines *actual volumetric flow rate* (in I-P units *acfm*) using actual altitude but at the fixed state of 15°C and 100% RH. Although the coil entering air properties are known, this coil manufacturer determines the mass flow of dry air through the coil using the specific volume at 15°C and 100% RH. In effect, this treatment defines *acfm* as *scfm* at actual altitude (sometimes called *altitude cfm*) while most of us think of *acfm* as actual *cfm* determined using actual temperature, a humidity parameter, and barometric pressure. The consequence of these differences is generally small, but it can be confusing and it is not always easy to determine the coil manufacturer's definition.

Now to address the sensible enthalpy removal difference assuming that the total enthalpy of both calculations is in close agreement and that both programs use the same mass flow rate of dry air. The sensible enthalpy difference between a load calculation program and a coil report may be explained by the fact that there is no universal or standard method for calculating the sensible enthalpy transfer in a process.

ARI Standard 410 for rating forced circulation air cooling and air heating coils suggests that coil sensible enthalpy be calculated using a moist air specific heat capacity of $1.018\ \mathrm{kJ/kg_{DA}}$ ($0.243\ \mathrm{Btu/lb_{DA}}$). Using the equation $C_P = C_{P\_DA} + W \cdot C_{P\_WV}$ it is possible to "back out" the W value used by ARI since C_{P_DA} and C_{P_WV} are for all practical purposes constants in the range of normal air conditioning. The W (humidity ratio) value used by ARI is ≈ 0.007 $\mathrm{kg_{WV}/kg_{DA}}$ to arrive at 1.018 $\mathrm{kJ/kg_{DA}}$ (0.243 $\mathrm{Btu/lb_{DA}}$). The value 0.007 $\mathrm{kg_{WV}/kg_{DA}}$ may not represent the average humidity ratio of the air in the actual cooling and dehumidifying process.

In calculating the sensible enthalpy transfer, the load calculation or psychrometric software program may be using a humidity ratio at (1) coil leaving air condition, (2) an average of entering air humidity ratio and leaving air humidity ratio, or (3) humidity ratio at 0°C dew-point temperature. The effect of these differences can be judged by the equation for calculating sensible enthalpy transfer:

$$q_{SEN} = Q \cdot [C_{P\_DA} \cdot \Delta t_{DB} + C_{P\_WV} \cdot (W_{ENT} \cdot t_{DB\_ENT} - W_{LVG} \cdot t_{DB\_LVG})]/v_{DA}$$

In the future, ARI could consider revising this standard. The standard was originally developed in the late 1940s when the instrumentation and laboratory setup for coil testing were not as accurate or available as they are today. Coil testing in the field is difficult, as is reading temperatures to an accuracy of ±0.05°C. It is not much easier in the laboratory and requires the averaging of several tests. Also, it may come as a surprise to many that ARI does not certify the leaving dry-bulb temperature or the sensible enthalpy removed by the coil. An ARI coil rating only certifies the leaving wet-bulb temperature and the total enthalpy removed by the coil.

How Can I-P Specific Enthalpy Values Be Converted to SI?

Conversion of individual statepoint specific enthalpy values between Inch-Pound (I-P) and Systéme International (SI) units is complicated because I-P and SI use different zero enthalpy temperature reference statepoints for the *dry-air* component. For I-P, the reference temperature for the dry air component of moist air is 0°F (–17.78°C) and for SI, the reference is 0°C (32°F). The reference for the H_2O component is the ice point of liquid H_2O at 32°F (0°C) for I-P and SI. *The conversion complication is similar to the complication in converting temperature between °F and °C because the 0° (reference) temperatures are not equal.*

Because of the possible confusion in converting individual I-P specific enthalpy values to SI, practitioners are advised to convert I-P barometric pressure and two other independent I-P statepoint properties to SI and then use these SI values to calculate all other SI psychrometric properties. The output SI property values can then be converted to I-P values. This is the procedure least likely to result in errors.

The following equation can be used to convert an individual I-P specific enthalpy values with units of Btu (lb°F) to an SI value with units of kJ/(kg°K).

$$h\_SI = [h\_IP - 7.687] \cdot 2.326.$$

The bracketed term includes the I-P-specific enthalpy value minus the specific heat of dry air (0.2403 Btu/(lb°F) times the difference in reference temperatures (32°F). In words, the minus 7.687 Btu (lb_{da}°F) adjustment resets the I-P zero specific enthalpy dry-air (not the water vapour) reference temperature to 32°F (0°C). In lieu of the rounded 7.687 value high accuracy psychrometric algorithms should subtract the specific enthalpy of dry air in I-P units at the I-P equivalent of 101.325 kPa at 0°F from that at 32°F. For example, the unique value for the ASHRAE RP-1485 (2009) algorithm is 7.68565957 Btu/lb.

The final term in the equation converts the units from Btu/(lb°F) to kJ/(kg°C).

I recommend that this equation only be used when debugging software programs or comparing an SI version of a program with a previous I-P version. Anyone who chooses to convert individual specific enthalpy values will have to manipulate many numbers, and eventually errors will occur—hence the earlier recommendation to convert the entering statepoint P_{BAR}, t_{DB}, and a humidity property to SI and then use SI psychrometric software for the remainder of the calculations. As a matter of interest, the 1983 Hyland-Wexler and the 2009 Herrmann-Kretzschmar-Gatley water and ASHRAE *Real Moist Air* psychrometric algorithms and research both used SI units. This is one more tiny reason to calculate in SI.

Why Are There So Many Different Values for C_{P_WV}?

First, a little background. The specific heat capacity of water vapour, C_{P_WV}, is used in virtually every psychrometric statepoint and process calculation. In I-P units, values of 0.435 Btu/(lb·°F), 0.44 Btu/(lb·°F), and 0.45 Btu/(lb·°F) are often used. In SI units, values of 1.805 kJ/(kg·°C), 1.84 kJ/(kg·°C), 1.86 kJ/(kg·°C), 1.88 kJ/(kg·°C), and 2.005 kJ/(kg·°C) appear in psychrometric literature.

Second, no matter which value is used, the differences in final property or process values or calculations are relatively minor.

Third, the values should be consistent in both the SI or I-P system of units. Psychrometrics jumped several plateaus with Willis H. Carrier's 1911 ASME paper "Rational Psychrometric Theory." If we follow Carrier's *rational* suggestion, it follows that the SI C_{P_WV} value should be 4.1868 times the I-P value because this is the exact conversion unit listed in ASTM E380 *Standard Practice for Use of the SI International System* *of Units* for converting I-P specific heat capacity values to SI, assuming that the I-P specific value was based on the 1956 Fifth International Conference on the Properties of Steam (International Table, abbreviated IT). The ASTM document also lists an exact conversion constant of 4.184 based on the "thermochemical" definition of the calorie (*thermochemical* is abbreviated TH). So, in choosing the conversion factor, one must first know the basis under which the I-P specific heat capacity value was measured or stated.

Here I borrow and slightly modify a quotation from my friend and esteemed building scientist Joe Lstiburek, Ph.D., P.E.: *Finding the original basis (IT or* TH*) requires the wisdom of Solomon, the intelligence of Newton, and the wit of Wilde— the reason we have trouble with this is that none of these people is alive to actually ask. Wars have been fought over less. Sometimes no choice is possible. We will assume that some type of rational choice will be made.*

Now where does one go to get C_{P_WV} values? Answer: lots of places, but confine the search to the *2009 ASHRAE Handbook—Fundamentals,* where there are at least three locations: (1) Chapter 1, "Psychrometrics," (2) Thermophysical Properties of Refrigerant 718 (Water), p. 30.37, and (3) Chapter 33, "Physical Properties of Materials" (the water vapour information). Unfortunately, the values are slightly different. A minor difference can be explained by the fact that water and water vapour properties change slightly in the presence of air at atmospheric pressure versus properties of pure H_2O. See below for the values used by the author.

What C_{P_WV} Value Do You Recommend and Why?

For building heating and air-conditioning psychrometrics, I use the following values: In SI units, C_{P_WV} = 1.84 kJ/(kg °C); in I-P units, 0.44 Btu/(lb·°F). These values are rationally consistent using either the 4.1868 (IT) or the 4.184 (TH) conversion factor.

The ultimate source for these values is Table 2 from Chapter 1, "Psychrometrics," from both the I-P and SI *2009 ASHRAE Handbook—Fundamentals* or the ASHREA LibHuAirProp_SI Excel add-in. The SI values come indirectly from the Herrmann-Kretzschmar-Gatley ASHRAE RP-1485 2009 Research Report. Using equation solver software (EES) or a spreadsheet program, it is possible using the h_{AS} and W_S values to back out the change in enthalpy of the H_2O component per unit mass of the H_2O component for every degree change in temperature. When these data are plotted and a curve fit is applied over the ordinary air-conditioning range of temperatures, my recommended values are the result.

Hopefully, Willis H. Carrier will rest more peacefully in his grave when our I-P and SI equations are rationally consistent!

What are the Units of $C_{p\_MoistAir}$?

$C_{p\_MoistAir}$ represents the enthalpy to increase 1 kilogram of moist air one degree Celsius. However, in psychrometrics it is expressed per kilogram of the dry air component with units of kJ/(kg_{da}·°C).

Note that $C_{p\_MoistAir}$ is sometimes listed along with other transport properties of moist air, in which case it may have units of kJ/(kg·°C). To convert to psychrometric units, this value must be multiplied by (1+W).

Will Future Editions of this Book Include Problems?

Possibly, if there is a third edition and if there is a demand for problems and a solutions manual. Many college-level thermodynamics textbooks that include a chapter on *gas-vapour mixtures* include problems. The *mother lode* of psychrometric problems is probably in *Thermodynamics, An Engineering Approach*, 3rd edition (Yunus A. Cengel and Michael A. Boles, McGraw-Hill, 1998); there are 129 problems listed at the end of Chapter 13.

What Are Other Uses of Psychrometric Charts?

I am certain that students and architects have a number of choice suggestions!

Several authors use the psychrometric chart to display the ASHRAE comfort zones from Standard 55.

Some authors and organizations show summer and winter climate zone areas on a psychrometric chart. A Hong Kong University Web page shows Hong Kong weather data on a psychrometric chart. A recent CD-ROM update of the *Engineered Weather Data Manual* (AFM-88) plots annual weather data for many cities in the U.S. on a temperature-humidity ratio grid, which is, for all practical purposes, a psychrometric chart.

Bill Rose at the University of Illinois makes the absorption and desorption of water vapour to and from hygroscopic materials more understandable by plotting the material equilibrium moisture content on a psychrometric chart.

The psychrometric chart is also useful for delineating the control modes or modes of operation for systems that include energy recovery and other options. Some of these concepts are illustrated on the following pages.

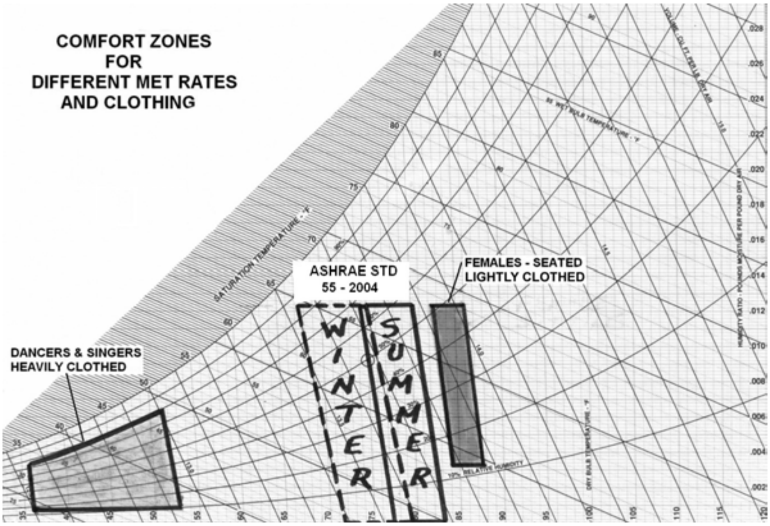


Figure 24-2—ASHRAE Standard 55 comfort range on psychrometric chart.

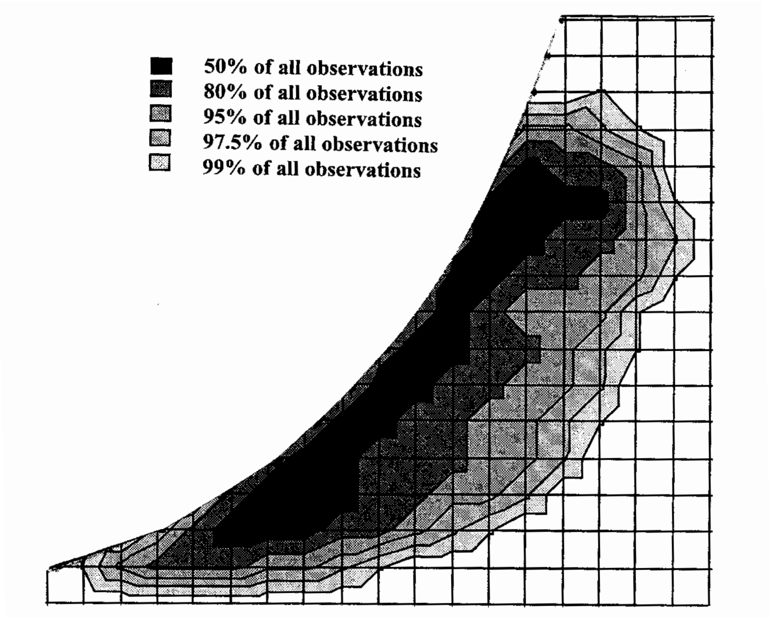


Figure 24-3—Frequency of occurrence plotted on psychrometric chart.

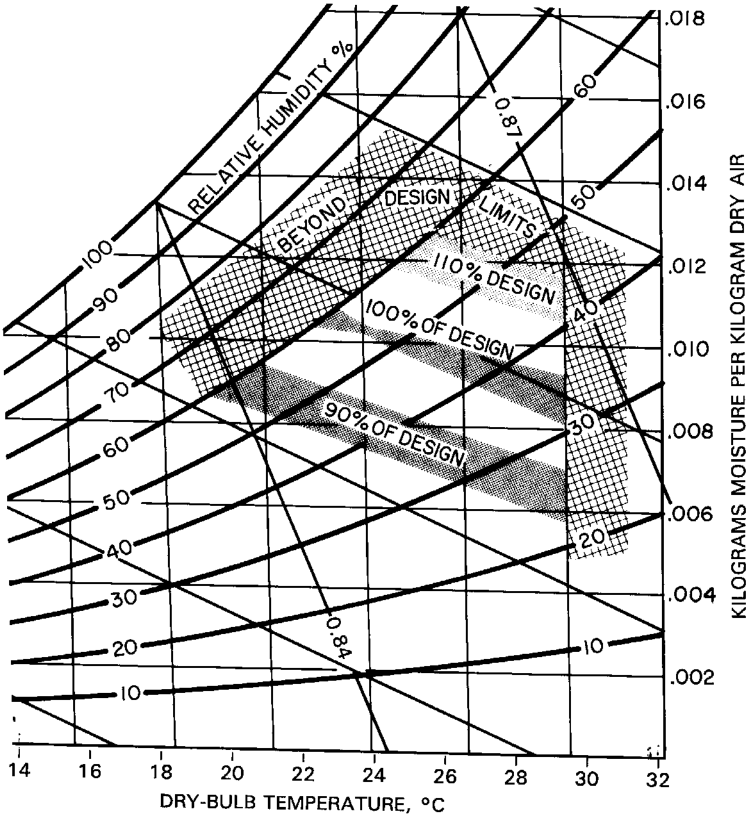


Figure 24-4—Open display refrigerator load factors on psychrometric chart

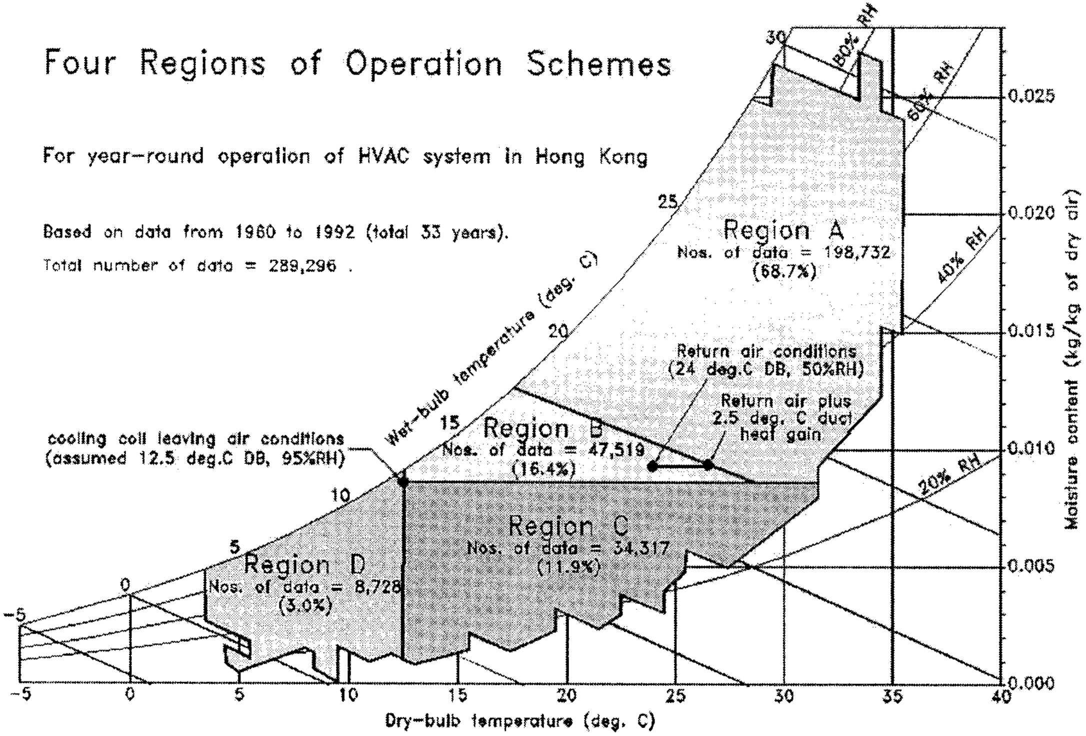


Figure 24-5—Control modes superimposed on a psychrometric chart.
(Source: Hong Kong University School of Architecture.)

We have no trouble with Bernoulli's equation so long as we use
absolute pressures. We do have trouble when we use gage pressures.
The conclusion might be that we should always use absolute pressures.
In many ways this would be an excellent custom, but in fact it is not the custom.
—C. Harold Berry, 1954

25 Altitude Effects on Psychrometrics and HVAC

ALTITUDE (OR P_{BAR}) EFFECTS ON PSYCHROMETRICS AND HVAC

The main thrust of this chapter is psychrometric and HVAC design and equipment selection considerations when the location barometric pressure or the process pressure is other than 101,325 Pa (i.e., when location altitude is other than sea level). The major psychrometric properties of concern are specific volume, v, and specific enthalpy, h. Air moving systems (fans and ducts) are the main focus. Determination of psychrometric properties, cooling equipment selection, and load calculation will receive minor coverage. For more extensive coverage, refer to publications of the major HVAC equipment manufacturers.

A subtitle of this chapter could be *the effect that a change in specific volume from 0.83* m^3/kg_{da} *(≈13.33* ft^3/lb_{da}*) has on HVAC design and equipment selection*. In order to standardize equipment ratings, manufacturers and industry associations often use a specific volume of 0.83 m^3/kg_{da} as a rating standard for use in performance tables. (Note that many texts use the density property in lieu of specific volume but, following the practice outlined earlier, this text uses only one psychrometric property for a given parameter. Specific volume and density are interrelated by the equation $\rho = 1/v$ [$\rho = 1/0.83 \cong 1.205\ kg_{da}/m^3$]). (As of the date of publication of this book the industry has not settled on a single SI standard air density value. Some equipment rating standards use a density of 1.2 kg_{da}/m^3 [$\cong 0.833333\ m^3/kg_{da}$] and other standards use a density of 1.205 kg_{da}/m^3 [$\cong 0.83000\ m^3/kg_{da}$].) (Standard air is usually based on 20°C dry air at 101,325 Pa; the exact temperature for 0.83 m^3/kg_{da} is 19.9476°C.)

Psychrometric Properties That Can Affect Specific Volume

The specific volume of the working fluid (air) is dependent on the following properties, which are interrelated by the ideal gas equation:

- absolute pressure, p_{BAR}
- absolute temperature, T
- water vapour pressure (indicator of moisture content), p_{WV}
- molecular mass of the working fluid, M

$$p_G \cdot v_G = R_G \cdot T_G, \quad \text{rearranging} \quad v_G = R_G \cdot T_G/p_G$$

$$p_G = p_{BAR} - p_{WV} \quad \text{and} \quad R_G = R_U/M$$

which leads to

$$\boldsymbol{v_{G\_DA} = (R_U/M) \cdot T_G/(p_{BAR} - p_{WV})} \quad \text{and} \quad T_G = t\_°C + 273.15$$

The ideal gas equation of state shows the property interrelationships, which can also be described in words:

- Specific volume is directly related to absolute temperature, T.
- Specific volume is inversely related to the difference between absolute pressure p_{BAR} and p_{WV}. As noted in Chapter 14, "Barometric Pressure," p_{BAR} is related to altitude above or below sea level.
- At sea level and dew-point temperatures less than 30°C, the impact of water vapour pressure is relatively minor because p_{WV} is tiny in relation to p_{BAR}. At higher elevations where p_{BAR} is significantly lower and in drying applications with higher dew-point temperatures for which p_{WV} has a greater value, water vapour pressure has a major impact on the specific volume of the working fluid. (See Chapters 13, 17, and 18 for the interrelationship between the values RH, W, and p_{WV}.)
- The impact of the molecular mass, M, of the working fluid is important when the molecular mass of the working fluid is significantly different from 28.966 kg/mol.

The specific volume per unit mass of the *dry air* component is used in psychrometric heat transfer equations because specific enthalpy and humidity ratio are expressed per unit mass of dry air. Calculations involving fan power should use specific volume per total mass of air.

$$v_{G\_DA+WV} = v_G/(1+W)$$

Processes Requiring a Constant Mass Flow of Air

The underlying equations for most HVAC applications require a constant mass flow of the dry air component. Recall from the chapter on process calculations (Chapter 20) that:

$$q_{AIR} = (h_1 - h_2) \cdot m_{DA} \qquad \text{and} \qquad Q(\text{Volumetric\_}Flow) = v \cdot m_{DA}$$

$$q_{AIR} = (h_1 - h_2) \cdot Q/v$$

These equations show that an increase in specific volume, which occurs at higher altitudes, higher temperatures, or higher water vapour pressures, requires an increase in volumetric flow rate in order to achieve the same mass flow rate (m_{DA}) and, therefore, the same energy transfer.

Processes Requiring a Constant Volumetric Flow Rate

Some processes such as that of exhaust hoods are more dependent on capture velocity and, therefore, volumetric flow rate rather than mass flow rate. If these processes are at higher altitudes (or the specific volume of the working fluid is greater), then the required mass flow rate of air at the higher altitude will be less than at sea level.

Determination of Psychrometric Properties

Some pamphlets covering altitude effects provide correction factor tables to determine properties at conditions other than 101,325 Pa (sea level), 20°C, and 0.0 humidity ratio (dry air). Since psychrometric properties at any altitude in the troposphere with temperatures between –40°C and 65°C and any humidity parameter can be easily obtained using software, the use of correction factor tables is antiquated.

It is assumed that the designer or technician can obtain sufficiently accurate psychrometric properties using a software program or the software built into an instrument or other device.

Selecting the Air Moving Device (the Fan)

How does a change in specific volume affect a fan originally operating with a specific volume of 0.83 m^3/kg?

The assumptions underlying the answer are that (1) the type, model, size, fan blade pitch (if applicable), and fan rotational speed are fixed and (2) the air conveyance system, comprising the air passage through equipment and ducts, is fixed. Additional minor stipulations are (3) that the airflow in the air conveyance system is turbulent and (4) that there are no fixed or linear pressure drop components in the system. Using *fan laws* and these assumptions, a change in specific volume has the following effects:

- The volumetric flow capacity remains constant (is not changed) with a change in specific volume: $Q_{ACT} = Q_{TBL}$.

- Fan pressure is inversely proportional to specific volume: $SP_{ACT} = SP_{TBL} \cdot (v_{TBL}/v_{ACT})$; note that this relationship also applies to total pressure, *TP*, or velocity pressure, *VP*.
- Fan power requirements are inversely proportional to specific volume: $PWR_{ACT} = PWR_{TBL} \cdot (v_{TBL}/v_{ACT})$.
- Fan efficiency is approximately the same.

The above relationships are shown in Table 25-1.

The phrase is often heard that fans are "essentially constant volume devices." This is true for a given fan at a constant fan rotational speed and a fixed system with turbulent airflow and no fixed or linear pressure drop components. Change the density of the working fluid and the volumetric flow capacity remains exactly the same. Figure 25-1 shows fan and system curves with different densities of working fluid.

Table 25-1

Table Value @ 0.83 m³/kg	Multiply By	Actual Value @ v_{ACT}
Q_{TBL}	1.0	Q_{ACT}
SP_{TBL}	v_{TBL}/v_{ACT}	SP_{ACT}
PWR_{TBL}	v_{TBL}/v_{ACT}	PWR_{ACT}

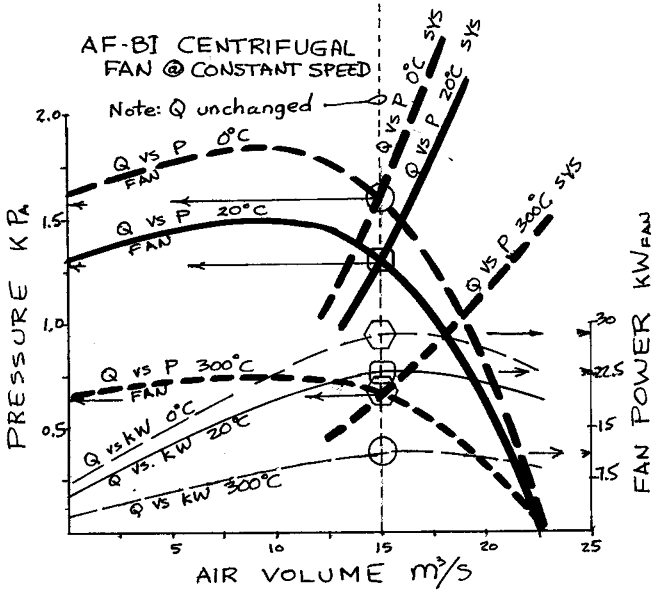


Figure 25-1—Fan performance at three different specific volume conditions.

General Rules for Selecting Fans at Other Than Standard v

Given actual air quantity, Q_{ACT}, and actual fan static pressure, SP_{ACT}, required at actual specific volume of the working fluid, v_{ACT}:

1. Convert SP_{ACT} to SP_{TBL} using the relationship cited above.
2. Using the fan manufacturers' rating table for standard (table) conditions, select the fan size at Q_{ACT} and SP_{TBL}. Note the fan speed and fan power as listed in the rating table.
3. Fan speed and Q are identical for v_{ACT} and v_{TBL}.
4. Convert PWR_{TBL} to PWR_{ACT} using the relationship cited above. Carefully evaluate PWR required for starting and warm-up conditions. If the fan is started with a specific volume of the working fluid of 0.83 m^3/kg, then the PWR at table or standard rating conditions will be required to ensure adequate power at startup. As the specific volume increases, the PWR will decrease.

Fan selection software provided by the fan or air-handling unit manufacturer is usually programmed to accept actual values for air quantity, pressure, and specific volume.

General Rules for Designing Ducts at Other Than Standard v

Air conveyance system performance with changes in specific volume of the working fluid follows the same relationships as the fan. The air quantity remains constant even though the specific volume changes and

$$SP_{ACT} = SP_{TBL} \cdot (v_{TBL}/v_{ACT}).$$

The air conveyance system can be designed using standard sea level tables or, alternatively, some software design programs may process the calculations using sophisticated algorithms at the actual specific volume.

Heat Gain and Heat Loss Calculations

At higher altitudes, the air is less dense, there are fewer dust particles to scatter the sun's rays, and the sun's intensity increases approximately 0.23% for each 100 m increase in altitude above sea level.

The dew-point temperature is less at higher altitudes; thus, there are fewer water vapour molecules in the air to scatter and reflect the sun's rays. For each 1°C decrease in dew-point temperature below 20°C, the solar intensity increases approximately 1.26%.

The thin, stagnant films of air that constitute a portion of the total resistance to the transmission of heat across building elements have fewer molecules at higher altitudes and are therefore better insulators. This

Table 25-2—Airflow at Altitude Multipliers

Altitude	–750 m	0 m	750 m	1500 m	2250 m	3000 m
Δh	20.257	21.440	22.763	24.246	25.913	27.792
v_ENT	0.7845	0.8583	0.9407	1.0329	1.1361	1.2522
$\Delta h/v\_ENT$	25.821	24.979	24.198	23.475	22.808	22.195
Multiplier	1.034	1.000	0.969	0.940	0.913	0.889

Table applicable only to 25°C EDB, 18°C EWB, 10°C LWB, 100% LRH.

decreases transmission of heat through walls by 0.2% to 0.5% for every 100 m increase in altitude above sea level.

If the volumetric infiltration flow rate is the same at sea level and at altitude, then the mass flow rate of this infiltration will be less at elevation and the associated heating load will be lower.

Many computerized load calculation programs factor in these changes.

Coil Selection Considerations: v_{DA} and Δh

Most coil selections are made using manufacturers' selection software, which, if properly used, accounts for altitude and psychrometric effects. Cooling coils are usually selected and specified to cool a given airflow rate, Q, from entering air dry-bulb and wet-bulb temperatures to leaving wet-bulb temperature. The airflow at altitude must be increased to maintain the sea level mass flow of the dry air component as discussed earlier in this chapter. Table 25-2 illustrates that the specific enthalpy difference between entering and leaving air for fixed *EWB* and *LWB* increases with altitude. The multiplier row in the table shows the combined effect of the change in v_{DA} and Δh.

Cooling Tower Selection at Altitude

Changes in altitude have only a minor effect on cooling tower (CT) capacity given fixed (1) fan rpm, speed, or blade pitch—i.e., constant air quantity; (2) entering wet-bulb and dry-bulb temperatures; (3) water flow rate; and (4) entering water temperature.

Ramsey (1966) and others analyze CT performance at altitude vs. sea level using a simplified model that accounts for (1) the decrease in the mass flow rate of air due to the increase in specific volume at altitude and (2) the increase in the specific enthalpy difference between leaving and entering air due to changes in specific enthalpy at altitude. This simple analysis predicts a *decrease* in capacity at altitudes above sea level, as shown in Figure 25-2. At 3000 m elevation Ramsey predicts a decrease in capacity of approximately 5%. Ramsey's analysis assumes no change in the heat and mass transfer effectiveness.

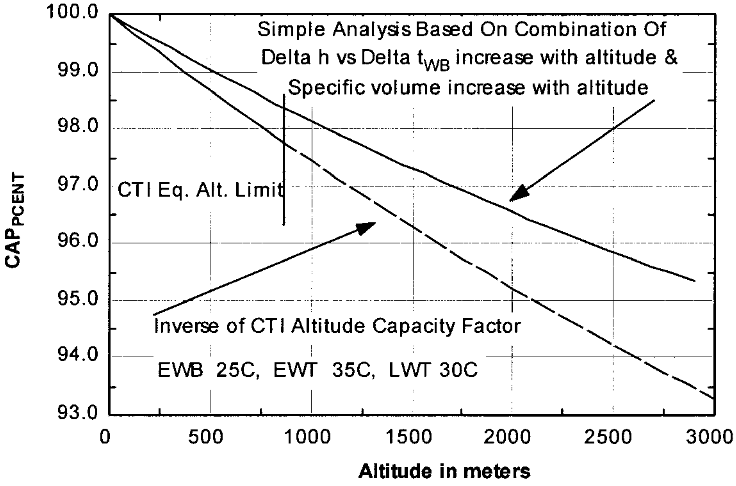


Figure 25-2—Cooling tower capacity vs. altitude.

The Cooling Technology Institute (CTI) (formerly the Cooling Tower Institute) Standard STD-201 dated November 1996 predicts the opposite of Ramsey's analysis. Equation (1) of STD-201 predicts a capacity *increase* of approximately + 2.5% per 1000 m *increase* in elevation with the restriction that the equation only applies for barometric pressures from 105 kPa (300 m elevation *below* sea level) to 91.4 kPa (860 m elevation above sea level). The *inverse* of this equation is also plotted in the figure. There are no supporting references for Equation (1) or the limited range of barometric pressures in either CTI technical papers or CTI STD-201.

Most CT selection software uses modifications of the Merkel theory (circa 1925) for predicting CT capacity. This requires stepwise integration of complex heat and mass transfer equations as the air and water proceed through the CT. The combined-heat-and-mass-transfer coefficient used in this integration is based on tests of CT fill at the manufacturer's plant or at an independent test facility. The test locations are generally close to sea level elevation. There is no known test data for high elevations.

An ultraconservative no-added-cost method of selecting a packaged CT at higher elevations is to increase fan speed by 1.5% (i.e., increase the airflow 1.5%) for each 1000 m increase in elevation above sea level. The cataloged motor will be more than adequate because of the increase in specific volume at elevation. Contact the CT manufacturer to determine if a smaller-than-cataloged motor will provide the required airflow.

James E. Braun, Ph.D. of Purdue's Mechanical Engineering Faculty, has provided me with a draft of his paper on this topic. Braun's analysis is based on the Merkel theory, a Lewis number of unity and other simplifying assumptions. He defines cooling tower volumetric capacity, q_v, with this equation:

$$q_v = \varepsilon \cdot \rho_a \cdot (h_{s,w} - h_{a,i})$$

where

q_v = heat rejection per unit volume of flow

ε = 1-e^{-Ntu} (Refer to Braun's 1989 ASHRAE paper for ε from catalog data)

ρ_a = air density

$h_{s,w}$ = saturated air enthalpy at the air-water interface (evaluated at T_w)

h_a = the local air enthalpy away from the surface

Ntu = $h_c \cdot A / (m_a \cdot C_{pm})$

T_W = average temperature of inlet and outlet water

m_a = air mass flow rate

The *effectiveness*, ε, increases with altitude in all cases, but it increases more for cooling towers having *Ntu* = 1 than for towers with *Ntu* = 4. Braun's ρ and Δ*h* effects on capacity are in agreement with Ramsey's analysis except that Braun's analysis covers a broad range of entering conditions. Figure 25-3, from Braun's draft paper, shows that capacity at altitude for a specific cooling tower may either increase or decrease depending on the (1) entering air conditions, (2) entering water conditions, or (3) the *Ntu* value of the specific cooling tower at sea level.

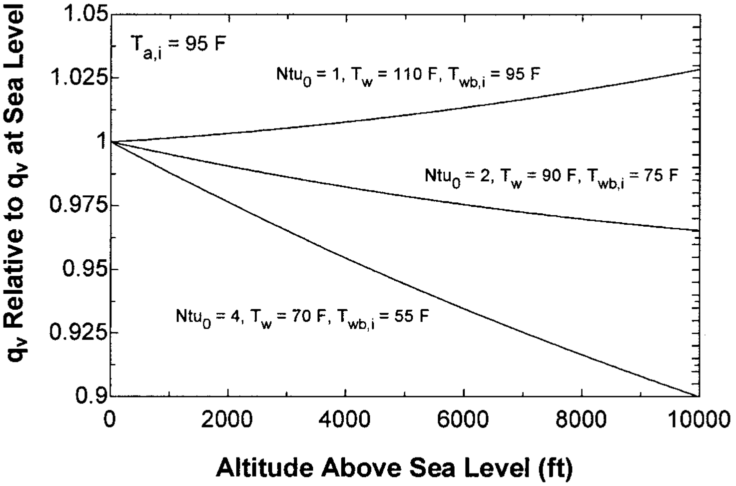


Figure 25-3—Effect of altitude on volumetric heat rejection capacity (Braun).

Chapter Closure

- Adjust air-handling and heat transfer equipment used for conditioning air for specific volume and enthalpy differences at non-sea level elevations. Applications must be thoroughly checked.
- Calculations using absolute pressure and temperature, humidity ratio, and mass flow rates (vs. volumetric) will minimize errors.
- Psychrometric properties vary with altitude, temperature, and water vapour pressure (or humidity ratio). Good psychrometric software easily computes the changes.
- If the application requires a given mass flow of air, then the required volumetric flow capacity will be greater at higher elevations.
- If the volumetric flow rate is held constant, then duct friction is reduced at higher altitudes compared with duct friction at sea level.
- Heat gains and losses must be corrected to obtain accurate loads at altitudes other than sea level.
- The air quantity and stack requirements for a combustion device must be adjusted for the change in specific volume at elevated altitudes in order to retain the sea level heat transfer rating.
- Some DX HVAC equipment utilizes direct drive condenser fans and, in certain cases, direct drive conditioned air fans. The mass flow rate for these fans decreases as the specific volume of the air increases. The lower air mass flow rate causes a change in the condensing temperature, which impacts the refrigeration cycle. The total capacity adjustment is thus complex. Contact the manufacturer for ratings.
- Closed-circuit equipment such as pumps, refrigeration compressors, and associated piping will not be affected by changes in elevation. Absolute pressures must be used in diagnosis and selection.

26 Psychrometric Program Listing and Comparison Table

This section includes the author's 2012 listing of an SI psychrometric property calculation program in EES (Engineering Equation Solver). EES is on the network of many college campuses. In some ways it resembles a spreadsheet. EES was written by Sanford Klein (University of Wisconsin).

This equation solver software has at least two advantages over spreadsheet software: (1) it will automatically iterate to find the solution, which eliminates the need to use looping routines, and (2) once the program is given barometric pressure (or ± altitude ref. sea level) and any two other psychrometric properties, it solves for all other properties.

A feature of the equation solver is that the equations can be in any form and in any order. For someone not trained in FORTRAN or another programming language, entering the equations is simplified, which may be an advantage. On the other hand, the fact that equations can be entered in any order, including helter-skelter, may make it more difficult for others to follow the program listing.

The program on the following pages was originally written in I-P units and a year later an SI program was patterned after the I-P format. Today the program is written in SI only and includes an I-P conversion routine.

These EES files were created to compare commercial psychrometric software and to provide documentation for accurate and simple psychrometric routines using the Ideal Gas assumption. Options for writing psychrometric routines include: (a) IAPWS equations in lieu of the Antoine water vapour pressure equations, (b) curve fit equations in lieu of single value specific heats, (c) use of Fortran modules, interpolation from tables, or use of simplified equations in lieu of a single value enhancement factor.

SI PROGRAM LISTING

"SI PSYCHROMETRIC ALGORITHMS (Ideal Gas); DPG VERSION APR 2012

USER IS RESPONSIBLE FOR TESTING, VERIFYING ACCURACY, & FINAL RESULTS; PROGRAM DOES NOT INCLUDE ERROR CHECKING OF INPUT VALUES. THE PROGRAM WILL SOMETIMES RUN WITH AN ERRONEOUS INPUT SUCH AS DB LESS THAN WB OR WB LESS THAN DEWPOINT. USER IS RESPONSIBLE FOR INPUTTING REALISTIC VALUES."

"*****SUBPROGRAMS***********************************"

SUBPROGRAM p_ws_liq97(T, p_WS) "Calculates vapour pressure in Pa over liquid water at temperature using IAPWS-IF97 equation"

$N_1 = 1167.05214527670$
$N_2 = -724213.167032060$
$N_3 = -17.0738469400920$
$N_4 = 12020.8247024700$
$N_5 = -3232555.03223330$
$N_6 = 14.9151086135300$
$N_7 = -4823.26573615910$
$N_8 = 405113.405420570$
$N_9 = -0.238555575678490$
$N_{10} = 650.175348447980$

$$Q = T + N_9/(T - N_{10})$$

$$A = Q^2 + N_1 \cdot Q + N_2$$

$$B = N_3 \cdot Q^2 + N_4 \cdot Q + N_5$$

$$C = N_6 \cdot Q^2 + N_7 \cdot Q + N_8$$

$$p_{ws} = 1000000 \cdot (2 \cdot C/(-B + (B^2 - 4 \cdot A \cdot C)^{0.5}))^4$$

END

SUBPROGRAM p_ws_ice08 (T, p_ws_ice) "Calculates vapour pressure in Pa over ice (solid) water at temperature using IAPWS-06 equation. Note that p_ws output values above 0°C do not represent an achievable state"

$a_1 = -0.212144006E2$
$a_2 = 0.273203819E2$
$a_3 = -0.610598130E1$
$b_1 = 0.333333333E-2$
$b_2 = 0.120666667E1$
$b_3 = 0.170333333E1$

$THA = T/273.16$

$$p_{ws\_ice} = 611.657 \cdot EXP((1/THA) \cdot (a_1 \cdot THA^{b_1} + a_2 \cdot THA^{b_2} + a_3 \cdot (THA)^{b_3}))$$

END

```
" ******CONSTANTS FOR THIS PROGRAM***************** "

 MM_H2O=18.015268              MM_DA=28.966
 R_U=8314.472                  EPSILON=MM_H2O/MM_DA
 FS=1.00475
 Cp_DA=1.006                   Cp_WV=1.84
 Cp_liq_H2O=4.186              Cp_ice_H2O= 2.0
 DltCp_L_V=    Cp_liq_H2O    -
 Cp_WV
 DltCp_I_LV=Cp_ice_H2O-
 DltCp_L_V
 h_g_ref=2499.86               h_g_ref_ice=2833.28

"*******PBAR & ALTITUDE ********************************"

"NACA Eq. applicable in the troposphere"
PBAR=101325*(1-0.0000225577*ALT)^5.2559    "Pa with ALT in m"

"******* KELVIN & CELSIUS *******************************"

KDB=CDB+273.15"Converts deg C dry-bulb to deg Kelvin"
KWB=CWB+273.15"Converts deg C wet-bulb to deg Kelvin"
KDP=CDP+273.15"Converts deg C dew-point to deg Kelvin"

"******* p_ws & DB temperature *****************************"

"Calculates PWS at DB and saturation at temperatures above and below
0 °C"

PWS=IF(KDB,273.15,PWSL,PWSX,PWSX)
CALL p_ws_ice08(KDB, PWSL)
CALL p_ws_liq97(KDB, PWSX)

"******* phi & RELATIVE HUMIDITY **********************"

phi=PW/PWS   "decimal ratio"
RH=100*phi  " % "

"******* WETBULB TEMPERATURE & p_ws @  WB **********"

PWWB=IF(KWB,273.15, PWWBL,PWWBX,PWWBX)  "PWWB is at
saturation"
CALL p_ws_ice08(KWB, PWWBL)
```

```
CALL p_ws_liq97(KWB, PWWBX)

"******* HUMIDITY RATIO & p_ws @  WB ******************"

WSWB=(MM_H2O/MM_DA)*FS*PWWB/(PBAR-FS*PWWB)

"******* HUMIDITY RATIO  & DB & WB ********************"

W=IF(KWB,273.15,WL,WX,WX)
WL=((h_g_ref_ice+DltCp_I_LV*CWB)*WSWB-Cp_DA*(CDB-
CWB)) /
            (h_g_ref_ice+Cp_WV*CDB-Cp_ice_H2O*CWB)
WX=((h_g_ref-DltCp_L_V*CWB)*WSWB-Cp_DA*(CDB-CWB))/
            (h_g_ref+Cp_WV*CDB-Cp_liq_H2O*CWB)

"******* HUMIDITY RATIOS & pw and p_ws   ****************"

WS=(MM_H2O/MM_DA)*FS*PWS/(PBAR-FS*PWS)
W=(MM_H2O/MM_DA)*PW*FS/(PBAR-PW*FS)

"******* ENTHALPY*************************************"

h=Cp_DA*CDB+W*(h_g_ref+Cp_WV*CDB)

"******* p_ws & DEWPOINT TEMPERATURE ***************"

"Calculates PW at dewpoint at temperatures above and below 0°C"

PW=IF(PW,611.2,PWKDPL,PWKDPX,PWKDPX)
CALL p_ws_ice08(KDP, PWKDPL)
CALL p_ws_liq97(KDP, PWKDPX)

"******* SPECIFIC VOLUME******************************"

"Calculates specific volume with units of cubic meters per kilogram of
dry air"
v=R_U*KDB*(1+W*MM_DA/MM_H2O)/(MM_DA*PBAR)
v_da=R_U*KDB/(MM_DA*PBAR)

"******* SI to I-P CONVERSIONS ************************"

FDB=(CDB+40)*9/5-40; FWB=(CWB+40)*9/5-40"°F"
FDP=(CDP+40)*9/5-40
```

```
ALT_IP=ALT/0.3048"FT"
PBAR_IP=PBAR*14.695948775513/101325"psia"
PBAR_InchHg=PBAR*29.921259843/101325"inchHg"
PW_IP=PW*14.695948775513/101325"psia"
PW_InchHg=PW*29.921259843/101325"inchHg"
h_IP=(h/2.326)+7.68565957 "Btu/lb_da"
v_IP=v*0.45359237/(0.3048^3)"ft^3/lb_da"
```

Checking Accuracy of This and Other Psychrometric Software

It is prudent to check the accuracy of any software that you use. This is true for psychrometric software and most particularly when the software is being used at altitudes other than sea level, temperatures below freezing, and temperatures above 40°C.

The seed for this book was planted in 1990 when a reviewer of a symposium paper rejected the paper because of minor property differences between the reviewer's software vendor and my vendor. At the time I was using a software accessories package from a major equipment manufacturer. I tediously checked several properties and, indeed, there were minor differences. I then tried a similar program from another major manufacturer and discovered larger errors. Ironically, the most accurate psychrometric software I found in 1990 was a $5.00 demo disk.

In order to check the accuracy of any psychrometric software, I suggest a comparison of property values with the tables in Appendix V of this book, which were calculated in EES using the ASHRAE LibHuAirProp add-in functions based on ASHRAE Research Project 1485 (2009) "real" dry-air gas and "real" water vapour equations.

Appendix V in this third edition is a summary of the 2007–2009 ASHRAE RP-1485 real moist air psychrometric research. The tables include the most accurate psychrometric property and enhancement factor values available.

Accuracy Comparisons

Table 26-2 compares *ideal gas psychrometric values* (Table 26-1) computed using the SI Program Listing in this chapter with the highly accurate ASHRAE RP-1485 real moist air values for 8 statepoints at barometric pressures of 101325, 84556, and 70108 Pa. Figure 26-1 provides a comparison from –50°C to +50°C at 101,325 Pa barometric pressure increase is that the author's April 2012 EES program utilizes (a) a single FS (enhancement factor) and the earlier program in the second edition interpolated FS for each pressure and temperature and (b) the 2012 program utilizes fixed values for the four specific heat values in the program. The earlier file used curve fit equations for the four specific heats.

Table 26-1—Psychrometric Properties For Software Validation (SI)

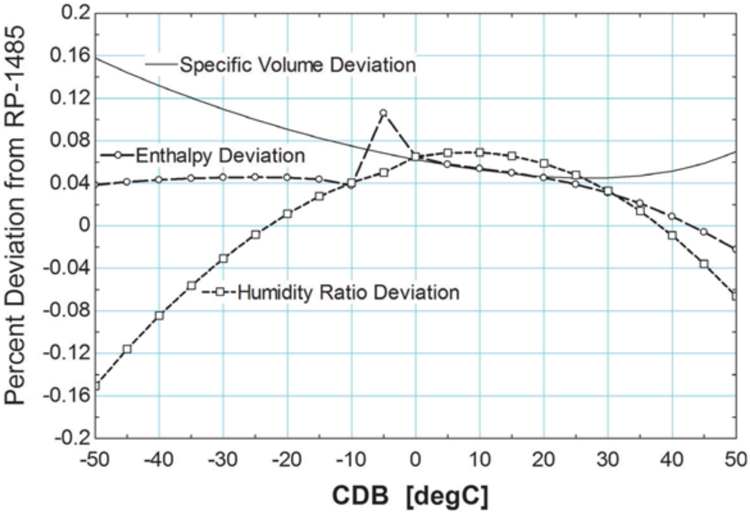
Altitude: 0 m (Sea Level)				**101.325**	**kPa**	**h = 0 m**	
t_{db}	**t_{wb}**	**t_{dp}**	**RH**	**p_{WV}**	**W**	**h**	**v**
°C	**°C**	**°C**	**%**	**Pa**	**kg_{WV}/kg_{DA}**	**kJ/kg_{DA}**	**m^3/kg_{DA}**
5	5	5.00	100.0	876.1	0.00542	18.639	0.7944
5	2	–2.16	58.6	513.1	0.00317	12.971	0.7915
5	–1	–11.91	25.1	219.9	0.00135	8.424	0.7893
5	–3	–37.17	2.0	17.7	0.00011	5.301	0.7877
25	25	25.00	100.0	3183.3	0.02017	76.503	0.8716
25	20	17.60	63.5	2021.4	0.01266	57.384	0.8615
25	15	7.73	33.2	1057.5	0.00656	41.853	0.8532
25	10	–10.40	7.9	251.9	0.00155	29.096	0.8464
Altitude: 1500 m				**84.556**	**kPa**	**h = 1500 m**	
t_{db}	**t_{wb}**	**t_{dp}**	**RH**	**p_{WV}**	**W**	**h**	**v**
°C	**°C**	**°C**	**%**	**Pa**	**kg_{WV}/kg_{DA}**	**kJ/kg_{DA}**	**m^3/kg_{DA}**
5	5	5.00	100.0	875.6	0.00651	21.402	0.9537
5	2	–1.42	62.3	545.3	0.00404	15.204	0.9499
5	–1	–9.32	31.6	277.0	0.00204	10.204	0.9469
5	–3	–20.99	10.8	94.2	0.00069	6.813	0.9449
25	25	25.00	100.0	3181.8	0.02432	87.097	1.0513
25	20	18.03	65.2	2075.5	0.01565	65.038	1.0373
25	15	9.19	36.7	1166.8	0.00870	47.350	1.0260
25	10	–4.65	13.1	415.6	0.00307	33.011	1.0168
Altitude: 3000 m				**70.108**	**kPa**	**h = 3000 m**	
t_{db}	**t_{wb}**	**t_{dp}**	**RH**	**p_{WV}**	**W**	**h**	**v**
°C	**°C**	**°C**	**%**	**Pa**	**kg_{WV}/kg_{DA}**	**kJ/kg_{DA}**	**m^3/kg_{DA}**
5	5	5.00	100.0	875.2	0.00786	24.840	1.1528
5	2	–0.82	65.5	573.1	0.00513	17.974	1.1478
5	–1	–7.45	37.3	326.2	0.00291	12.408	1.1437
5	–3	–15.40	18.3	159.8	0.00142	8.677	1.1410
25	25	25.00	100.0	3180.4	0.02956	100.466	1.2783
25	20	18.39	66.7	2122.2	0.01941	74.656	1.2584
25	15	10.35	39.6	1260.9	0.01139	54.230	1.2427
25	10	–1.17	17.5	556.5	0.00498	37.896	1.2302

Source: ASHRAE LibHuAirProp_SI Add-In

Table 26-2—Percentage (%) of Deviation SI Psych 2012 vs. RP-1485—Real Moist Air

Altitude: 0 m (Sea Level)				101.325	kPa	*h* = 0 m	
t_{db}	t_{wb}	t_{dp}	RH	p_{WV}	W	h	v
°C	°C	°C	%	Pa	kg_{WV}/kg_{DA}	kJ/kg_{DA}	m^3/kg_{DA}
5	5	0.00	0.00	–0.40	0.18	0.06	0.05
5	2	–0.46	0.00	–0.39	0.00	0.06	0.06
5	–1	0.00	0.00	–0.36	0.00	0.05	0.05
5	–3	–0.24	0.00	0.56	0.00	0.08	0.05
25	25	0.00	0.00	–0.43	0.05	0.04	0.05
25	20	0.00	0.00	–0.39	0.08	0.04	0.03
25	15	0.13	0.30	–0.31	0.15	0.05	0.04
25	10	–0.48	0.00	–0.04	0.65	0.06	0.04
Altitude: 1500 m				84.556	kPa	*h* = 1500 m	
t_{db}	t_{wb}	t_{dp}	RH	p_{WV}	W	h	v
°C	°C	°C	%	Pa	kg_{WV}/kg_{DA}	kJ/kg_{DA}	m^3/kg_{DA}
5	5	0.00	0.00	–0.34	0.15	–0.10	0.04
5	2	0.00	0.00	–0.31	0.00	–0.19	0.05
5	–1	–0.11	0.32	–0.29	0.49	–0.34	0.04
5	–3	–0.05	0.00	–0.21	1.45	–0.57	0.04
25	25	0.00	0.00	–0.38	0.08	0.03	0.04
25	20	0.00	0.15	–0.34	0.13	0.02	0.03
25	15	0.11	0.00	–0.27	0.23	0.00	0.03
25	10	–0.65	0.00	–0.12	0.33	–0.03	0.03
Altitude: 3000 m				70.108	kPa	h = 3000 m	
t_{db}	t_{wb}	t_{dp}	RH	p_{WV}	W	h	v
°C	°C	°C	%	Pa	kg_{WV}/kg_{DA}	kJ/kg_{DA}	m^3/kg_{DA}
5	5	0.00	0.00	–0.30	0.25	–0.20	0.03
5	2	–1.22	0.00	–0.28	0.19	–0.32	0.03
5	–1	–0.13	0.00	–0.21	0.00	–0.52	0.04
5	–3	–0.13	0.00	–0.13	0.70	–0.81	0.04
25	25	0.00	0.00	–0.34	0.14	0.03	0.04
25	20	0.00	0.15	–0.30	0.21	0.01	0.04
25	15	0.10	0.25	–0.24	0.26	–0.02	0.03
25	10	–1.71	0.00	–0.13	0.20	–0.08	0.02

p_{WV} values are pure H_2O multiplied by f.



1. *v, h* and *W* values are for saturated air at 101.325 kPa barometric pressure.
2. Values generated using "SI PSYCHROMETRIC ALGORITHMS (Ideal Gas) DPG VERSION APR 2012" are compared with values generated using ASHRAE LibHuAirProp_SI add-in functions which use the ASHRAE RP-1485 (2009) Real Moist Air algorithm.
3. The APR 2012 Ideal Gas algorithm utilized fixed values for the enhancement factor, the specific heat of water vapour, the specific heat of dry air, the specific heat of liquid H_2O and the specific heat of Ice.
4. The deviations is this figure can be reduced by utilizing curve fit equations for the specific heats and by interpolating FS from an extensive table of FS values.

Figure 26-1—Difference between ideal gas and real moist air (RP-1485) algorithms.

Conversion Equalities: I-P to SI Units

CONVERSION METHODOLOGY

Conversion tables herein show *equal* values that can be easily applied using "factor-label" conversion methodology. *Factor-label* methodology is based on the fact that multiplying a number and its units by 1.0 (with proper units) does not change its value. Using this method a fraction with units is created with a numerator and a denominator using a table equality. The fraction has a value of 1.0. The value to be converted (including its units) is multiplied by the fraction and its units. (See the example on the endsheet.)

The significant digits displayed in the table are more than usually required. This allows the user to develop other equalities. Users should adjust the number of significant digits to the accuracy of the problem.

The Btu in the table is based on the International Table (IT) definition. Unless otherwise stated, 1 Btu = 251.996 (IT) calorie. Prior to SI there were at least five definitions of the calorie in terms of joules: (1) the Fifth International Congress on the Properties of Steam in 1956 defined the calorie (IT) as 4.186 800 J (Exact) and therefore 1 Btu = 1055.055852 62 J, (2) the thermochemical (TH) calorie, which defined the calorie as 4.184000 J (Exact) and therefore 1 Btu (TH) = 1054.350 J, (3) the mean calorie, (4) the 15°C calorie, and (5) the 20°C calorie. The calorie qualifies as one of the most confusing technical terms. The food and diet industries also have the Calorie (which should always have a capital "C"), which is called a "food calorie," "large calorie," or "kilo calorie." It is equal to 1000 calories. The original calorie is sometimes called a "small calorie." SI eliminates the confusion and the slight differences in definitions of energy.

Unfortunately, until researchers update the property tables for all substances in current use, engineers and scientists must use older tables of properties in their calculations. Correct use of older property tables demands that the user know the definition of the calorie as used by the original author or researcher. In many cases this information is not available. In general, engineers and physicists utilized the International Table definition, while chemists and some scientists more commonly used the thermochemical definition.

A property that impacts the engineering world is the specific heat capacity of substance, C_P and C_V. Fortunately, most ASHRAE psychrometric data have already been updated to the IT definition for specific heat capacities of air and the three phases of H_2O. For other substances, if the definition of the calorie is not available, it is probably safer to assume that the data are based on one calorie equals 4.184000 J.

SI Prefix Nomenclature— Orders of Magnitude in Steps of 1000 (i.e., 1.0E3)

Sym.	Prefix	Pronunc.	Value	Example	Source of Prefix	
Y	yotta	YOTE-uh	1.0E+24	yottajoule	from *otto*, 8 in Italian	(8 steps)
Z	zetta	ZETT-uh	1.0E+21	zettamole	from *sette*, 7 in Italian	(7 steps)
E	exa	EX-uh	1.0E+18	exaweber	from *hex*, 6 in Greek	(6 steps)
P	peta	PET-uh	1.0E+15	petahertz	from *pente*, 5 in Greek	(5 steps)
T	tera	TAIR-uh	1.0E+12	teracandela	from *tettares*, 4 in Greek	(4 steps)
G	giga	JIG-uh	1.0E+09	gigawatt	gigas, *giant* in Greek	
M	mega	MEG-uh	1.0E+06	megakelvin	megas, *great* in Greek	
k	kilo	KILL-oh	1.0E+03	kilovolt	chilioi, *thousand* in Greek	
m	milli	MILL-ih	1.0E-03	milliampere	mille, *thousandth* in Latin	
µ	micro	MIKE-roe	1.0E-06	microohm	mikros, *small* in Greek	
n	nano	NAN-oh	1.0E-09	nanosecond	nanos, *dwarf* in Greek	
p	pico	PEEK-oh	1.0E-12	picofarad	pico, *little* or *tiny bit* in Spanish	
f	femto	FEM-toe	1.0E-15	femtonewton	femten, 15 in Danish or Norwegian	
a	atto	AT-toe	1.0E-18	attogram	atten, 18 in Danish or Norwegian	
z	zepto	ZEP-toe	1.0E-21	zeptohenri	from *sept*, 7 in Greek	(–7 steps)
y	yocto	YOCK-toe	1.0E-24	yoctolitre	from *okto*, 8 in Greek	(–8 steps)

Pronunciation, example, and source of prefix courtesy of Phillip Gibb's "Physics and Relativity FAQ" Web page.

In general, the prefixes in the following SI prefix nomenclature table should not be used. In restricted circumstances involving area or volume, these prefixes may be marginally acceptable. The centimetre may be used for body measurements and for clothing sizes.

Sym.	Prefix	Pronunc.	Value	Example	Source of Prefix
h	hecto	HECK-toe	1.0E+02	hectoradian	hekaton, *hundred* in Greek
da	deka	DECK-uh	1.0E+01	dekapascal	deka, 10 in Greek
d	deci	DESS-ih	1.0E-01	decisievert	decimus, *tenth* in Latin
c	centi	CENT-ih	1.0E-02	centimetre	centum, *hundredth* in Latin

CONVERSION FROM I-P TO SI UNITS— Table of Equalities

atmosphere, std press	=	101,325 Pa = 760 mm Hg = 29.92126 in. Hg @ 0°C
atmosphere, std press	=	14.69595 psi = 33.9 ft H_2O @ 39.2°F
atmosphere, std press	=	407 in. H_2O = 28,216 ft air @ 0.075 lb/ft^3
bar	=	100,000 Pa
Btu (IT)	=	1055.05585262 J (Exact)
Btu (tc)	=	1054.350 J
Btu/(lb·°F) (IT) *heat capacity*	=	4186.800 J/(kg·K) (Exact)
Btu/(lb·°F) (tc) *heat capacity*	=	4184.000 J/(kg·K) (Exact)
Btu/h (IT)	=	0.293071 1 W
Btu/lb (IT)	=	2326.000 J/kg (Exact)
Btu·ft/(hour·ft^2·°F) *thermal conductivity IT*	=	1.731 W/(m·°C)
Btu·in./(hour·ft^2 ·°F) *thermal conductivity IT*	=	0.1442 W/(m·°C)
calorie (IT)	=	4.186800 J (Exact)
calorie (tc)	=	4.184000 J (Exact)
°C + 273.15	=	K {degree Kelvin}
(°F – 32) / 1.8	=	°C
Δ enthalpy (specific) Btu/lb (IT)	=	2326.000 J/kg (Exact)
(h_{DA} in Btu/lb$_{DA}$– 7.687)*2326 (IT)	=	h_{DA} in J/kg_DryAir {Note units of joule; for kilojoule (kJ) use 2.326}
ft	=	0.3048 m (Exact)
ft·lb$_F$	=	1.355818 J {Note: 778.169 ft·lb$_F$ =Btu}
ft liquid H_2O (39.2°F)	=	2988.98 Pa
ft/min (fpm)	=	0.00508 m/s (Exact)
ft/s (fps)	=	0.3048 m/s (Exact)
ft^2	=	0.09290304 m^2 (Exact)
ft^3	=	28.31685 L (litre)
ft^3	=	0.02831685 m^3
ft^3/lb	=	0.062428 m^3/kg
ft^3/min (cfm)	=	0.4719474 L/s = 0.000 471 947 4 m^3/s
g (std. grav. free fall)	=	9.806650 m / s^2 (Exact)
g_C (grav. constant)	=	1.0 kg m / (N s^2)
gallon (US liquid)	=	3.785412 L = 0.003785412 m^3

CONVERSION FROM I-P TO SI UNITS— Table of Equalities (Continued)

GPM (gal. per min.)	=	0.06309020 L/s
joule = J	=	N·m
horsepower (550 ft lb/s)	=	745.6999 W
horsepower (electric)	=	746.000 W (Exact)
inch	=	25.4 mm (Exact)
inch^2	=	6.451600E-04 m^2 (Exact)
inch **Hg** @ 32°F	=	3386.38 Pa
inch **Hg** @ 60°F	=	3376.85 Pa
inch $\mathbf{H_2O}$ @ 39.2°F	=	249.082 Pa
inch $\mathbf{H_2O}$ @ 60°F	=	248.84 Pa
kilogram-force	=	9.806650 N (Exact)
kWh	=	3 600000 J (Exact)
lb (mass avoirdupois)	=	0.45359237 kg (Exact)
lb (mass avoirdupois)	=	7000 grain
lb$_F$ (pound-force)	=	4.4482216152605 N (Exact)
lb/ft^3	=	16.01846 kg/m^3
m bar (milli bar)	=	100 Pa (Exact)
mm Hg (0°C)	=	133.322 Pa
newton = N	=	kg·m/s^2
ounce (mass avoirdupois)	=	28.34952 g
pascal = Pa	=	N/m^2 = kg/(m·s^2)
perm @23°C (gr/(h·ft^2·in. Hg))	=	57.45 ng/(Pa·s·m^2) *WV permeance*
perm inch @23°C (gr·in./(h·ft^2·in. Hg))	=	1.46 ng/(Pa·s·m) *WV permeability*
psi (lb$_F$/in^2)	=	6894.757 Pa
ton (refrigeration)	=	3517 W
watt = W	=	J/s
Wh	=	3600.000 J

{tc} = thermochemical; {IT} = International Table; gr = grain

EXAMPLE OF FACTOR-LABEL METHOD

Convert 108 km/hour to miles/hour

$$\frac{108\ \text{km}}{\text{hour}}\ \frac{1000\ \text{m}}{\text{km}}\ \frac{\text{ft}}{0.3048\ \text{m}}\ \frac{\text{mile}}{5280\ \text{ft}} = \frac{67.108\ \text{mile}}{\text{hour}}$$

SYMBOLS

A = area, m^2

C_P = specific heat capacity of air, i.e., moist air

$C_{P\text{-}DA}$ = specific heat capacity of *dry air*, 1.006 kJ/(kg$_{\text{DA}}$·°C)

$C_{P\text{-}H2O}$ = specific heat capacity of *liquid H_2O,* 4.186 kJ/(kg$_{\text{H2O}}$·°C)

$C_{P\text{-}ICE}$ = specific heat capacity of *ice*, 2.0 kJ/(kg$_{\text{ICE}}$ ·°C)

$C_{P\text{-}WV}$ = specific heat capacity of *water vapour*, 1.84 kJ/(kg$_{\text{WV}}$·°C)

f_S = factor to adjust p_{WV} over pure water to p_{WV} in association with air

H = enthalpy, kJ

h = psychrometric specific enthalpy, kJ/kg$_{\text{DA}}$

H_{DA} = enthalpy of *dry-air* component of air, kJ

h_{DA} = specific enthalpy of *dry air*, kJ/kg$_{\text{DA}}$

$h_{G\,WV\,REF}$ = specific enthalpy of *water vapour* at 0°C, 2499.86 kJ/kg$_{\text{WV}}$

H_{WV} = enthalpy of *water vapour* component of air, kJ

h_{WV} = specific enthalpy of *water vapour*, kJ/kg$_{\text{DA}}$

l = length, m

M = molecular (molar) mass (formerly molecular weight), kg/mol

M_{H2O} = total mass of *water*, kg

m_{DA} = mass of *dry air*, kg$_{\text{DA}}$

$\dot{m}$ = mass flow rate, kg/s

m_{WV} = mass of H$_2$O, kg$_{\text{WV}}$

p_{BAR} = total pressure; barometric pressure, Pa

p_{DA} = partial pressure of *dry air*, Pa

p_{WV} = partial pressure of *water vapour*, Pa

p_{WVSAT} = partial pressure of saturated *water vapour*, Pa

Q = air quantity, m^3/s

q = heat transfer in process from *surroundings* to *system*, W = J/s

q^1 = ratio of $\Delta h/\Delta W$

R_U = universal gas constant, 8.314472 J/(mol · K)

R_G = gas constant for a specific gas = R_U/M, J/(kg · K)

RH = relative humidity in percent

SHR = sensible heat ratio, dimensionless

T = absolute temperature, Kelvin (K) where K = °C + 273.15

t_{DB} = dry-bulb temperature, °C

t_{DP} = dew-point temperature, °C

t_{WB} = wet-bulb temperature, °C

t_{REF} = 0°C for both *DA* and *WV*; 0.00°C for liquid water

μ = *water vapour* permeability, ng /(s·m·Pa)

V = volume, m^3

v = psychrometric specific volume, m^3/kg$_{\text{DA}}$

W = humidity ratio, kg$_{\text{WV}}$/kg$_{\text{DA}}$

Z = altitude above (+) or below (-) sea level, m

SUBSCRIPTS

DA = *dry air*

WV = *water vapour*

SAT = saturated

F = liquid phase (in some equations either liquid or solid phase)

H_2O = liquid phase (in some equations either liquid or solid phase)

$_I$ or $_S$ = ice or solid phase

G = gas phase

1 = statepoint 1 or entering statepoint

2 = statepoint 2 or leaving statepoint

3 = statepoint 3 (resultant of mixing 1 and 2)

COIL = "*system*" encompasses coil and airstream

AIR = "system" encompasses airstream only

SENS = sensible (energy, enthalpy, or heat)

LAT = latent (energy, enthalpy, or heat)

TOT = total (energy, enthalpy, or heat) = same as q with no subscript

GREEK

α = average temperature lapse rate in the troposphere, 0.0065°C/m

Δ = delta or difference, e.g., $\Delta h = h_1 - h_2$ or $\Delta W = W_1 - W_2$

ϕ = alternate for *RH*; units may be a decimal in lieu of a percent

Θ = time, s

ρ = density, kg/m^3 for air moving or kg$_{\text{DA}}$/m^3 for psychrometrics

μ = average permeability, ng ·m/(s·m^2·Pa)

Glossary

True communication, particularly in the transfer of knowledge, demands that the instructor or writer and the pupil or reader use words that have identical meanings for both parties. When the teacher and the student comprehend words differently, misinterpretation and confusion are often the result. This can create significant obstacles to learning.

Instructors and authors must be alert to these obstacles and strive to separately list difficult words. Students should ask for clarification or definition in a classroom situation, otherwise the balance of the classroom session may be ineffective because of miscomprehension.

In general, the definitions that follow are taken from ASHRAE and meteorology texts. In some cases, the author has modified the definition in the hope that the modified definition adds clarity and increases comprehension. There is no intent to discredit the original sources or even suggest that the author's definitions are superior to those of the source documents.

acfm: a variation of *scfm* used with the I-P system of units. Unfortunately, it has at least two meanings: (1) actual cubic feet of air per minute (referenced to a specific location in the airflow path), which equals the mass flow of dry air times the specific volume at a specific point in the flow path, i.e., p_{BAR}, t_{DB}, and RH defined; (2) Altitude Adjusted scfm (**scfm** is based on $\rho = 0.075\ \mathrm{lb_{DA}/ft^3}$ ($1/\rho = v = 13.3333\ \mathrm{ft^3/lb_{DA}}$), which occurs at 59.95°F, 100% RH at seal level barometric pressure). Altitude Adjusted scfm is based on adjusting the density, ρ, for the change in only the barometric pressure at the new altitude.

absolute humidity (vapour concentration; water vapour density number): in a mixture of water vapour and dry air, the mass of water vapour in a specific volume of the mixture.

absolute pressure: pressure above a perfect vacuum; the sum of gage pressure and atmospheric pressure.

absolute temperature: temperature as measured above absolute zero.

absolute zero temperature: zero point on an absolute temperature scale. See *Kelvin temperature*.

absorption: process whereby a porous material extracts one or more substances from an atmosphere, a mixture of gases, or a mixture of liquids.

adiabatic expansion: expansion of a fluid during which no heat is exchanged with the surroundings. **Isenthalpic expansion** is expansion of a fluid during which no heat or mechanical energy is exchanged with the surroundings; the enthalpy remains constant.

adiabatic process: thermodynamic process during which no heat is extracted from or added to the system.

adsorbent: material that has the ability to cause molecules of gases, liquids, or solids to adhere to its surfaces without changing the adsorbent physically or chemically. Certain commercially available solid materials, such as silica gel, activated carbon, and activated alumina, have this property.

adsorption: (1) process in which fluid molecules are concentrated on a surface by chemical or physical forces or both; (2) surface adherence of a material in extracting one or more substances present in an atmosphere or mixture of gases and liquids, unaccompanied by physical or chemical change.

air: the atmosphere; the mixture of invisible, odorless, tasteless gases (nitrogen, oxygen, and others) that surrounds the earth. Air is also refrigerant R-729.

air mass: in meteorology, refers to an enormous volume or parcel of air covering thousands of square kilometres that originated in one climate zone and is moving to or has moved to (and is stationary over) another climate zone. It is characterized by being relatively uniform horizontally in temperature and water-vapour concentration. A (weather) front occurs where two air masses collide.

air parcel: see *air mass*.

air, standard (I-P): dry air at ~69.2°F and 14.696 psia. Under these conditions, dry air has a mass density of exactly 0.075 lb/ft^3. This is not the same as SI standard air.

air, standard (SI): dry air at 20°C and 101.325 kPa absolute. Under these conditions, dry air has a mass density of 1.204 kg/m^3. This is not the same as I-P standard air.

air-conditioning process: in enclosed spaces, combined treatment of the air to control temperature, relative humidity, velocity of motion, and radiant heat energy level, including consideration of the need for removal of airborne particles and contaminant gases. Some partial air conditioners that may not accomplish all of these controls are selected for their capability to control specific phases of air treatment.

air infiltration (building): uncontrolled inward leakage of air with its associated water vapour through cracks and interstices in any building element and around windows and doors of a building, caused by the pressure effects of wind or the effect of differences in the indoor and outdoor air density.

air, ambient: surrounding air (usually outdoor air or the air in an enclosure under study).

air, saturated: air that has the maximum associated water vapour possible at a specified temperature and pressure.

ambient temperature: temperature of the medium surrounding an object.

apparatus dew point: usually the average cooling coil surface temperature of a cooling and dehumidifying air-conditioning apparatus. With the advent of software-based cooling coil selection programs, this variously defined term is no longer of use.

atmosphere: a thin envelope of gases (also containing suspended solid and liquid particles and clouds) that encircles the earth. An analogy is the thin skin of an apple.

atmospheric pressure: (1) pressure of the outdoor atmosphere; (2) standard atmospheric reference pressure (assumed sea level) is defined by the ICAO (International Civil Aeronautics Organization) as 101.325 kPa. In I-P units, the value is *approximately* 14.696 psi, or 29.921 inches of 0°C (32°F) mercury.

Avogadro's law: under the same conditions of pressure and temperature, equal volumes of all gases contain equal numbers of molecules.

blow-through (coil or heat exchanger): refers to the arrangement of air-side components in which the coil or heat exchanger is located downstream (on the leaving air side) of the air moving device (fan or blower). Hence the air moving device *blows-through* the coil.

boiling: phenomenon involving violent and raging agitation or bubbling up of a liquid under the influence of heat. In physics, the boiling temperature or boiling point is defined as the temperature at which a liquid's vapour pressure equals the local barometric pressure. Evaporation is a phenomenon that occurs when the vapour pressure of the liquid exceeds the partial pressure of the water vapour in the surrounding atmosphere.

boundary conditions: physical conditions (values of physical quantities, conditions of energy interchange, etc.) imposed on a system at the boundaries separating the system from its surroundings.

Boyle's law: the product of the volume of a gas times its pressure is a constant at fixed temperature. Also known as Mariotte's law.

British thermal unit (Btu) (an I-P unit): heat energy in a Btu. The Fifth International Conference on the Properties of Steam (1956) defined the International Table calorie as 4.1868 J. Therefore, the International Table Btu is exactly 1055.055 852 62 J. The mechanical equivalent energy of a Btu is 778.169 262 ft lb. The heat energy of a Btu is approximately that required to raise the temperature of a pound of water from 59°F to 60°F.

Btuh: properly, Btu/h.

Celsius temperature: temperature scale used with the SI system in which the freezing point of water is 0°C, the triple point is 0.01°C, and the boiling point is 99.9743°C. (Formerly referred to as the *centigrade scale*.)

change of phase: change from one of the three phases—solid, liquid, or gas—to another.

Charles' law: at constant pressure, the volume of a fixed mass or quantity of gas varies directly with the absolute temperature; a close approximation. Also known as Gay-Lussac's law.

climate: weather of some locality averaged over some time period plus extremes in weather behavior observed during the same period of time during the entire period of record. Time periods are generally 30 years if records are available.

cloud: a visible aggregate of minute water droplets and/or ice crystals in the atmosphere above the earth's surface. Clouds differ from fog only in that the latter is, by definition, in contact with the earth's surface. Clouds form in the free atmosphere as a result of condensation or deposition of water vapour in ascending air.

comfort chart: chart showing operative temperatures with dry-bulb temperatures, relative humidities, and air motion by which the effects of the various conditions on human comfort may be compared.

concentration (ratio) (by mass): ratio of the mass of one constituent to the mass of the total mixture (PPM_M) (parts per million by mass).

concentration (ratio) (by volume): ratio of the mols or molecules of one constituent to the mols or molecules of the total mixture (PPM_V) (parts per million by volume).

condensate: liquid formed by condensation of a vapour. In steam heating, water condensed from steam; in air conditioning, water extracted from air, as by condensation on the cooling coil.

condensation: change of phase of a vapour into its liquid or solid phase by extracting heat from the vapour.

condensation point: temperature at which a vapour liquefies if the latent heat is removed at a stated pressure.

condensation, solid phase: direct formation of the solid phase by cooling a vapour below its triple point; called **deposition** in the field of meteorology.

condensed phase: the liquid or solid phase of a substance.

condensing temperature: temperature of a fluid at which condensation occurs.

condition line, room: on a psychrometric chart, the infinite number of statepoints (dry-bulb temperatures and humidity ratios) that will satisfy the requirements of an air supply for a given room temperature.

condition line, process: on a psychrometric chart, the infinite number of statepoints (dry-bulb temperatures and humidity ratios) that trace the path of a psychrometric process from initial to final statepoint.

conditioned space: space within a building provided with heated or cooled air or both (or surfaces) and, where required, with humidification or dehumidification means to maintain conditions for an acceptable thermal environment.

conservation of mass law: the mass of a specific substance in the system undergoing a process equals the original mass of the substance in the system plus the mass entering the system minus the mass leaving the system.

conservation of energy law: energy can neither be created nor destroyed.

constant of proportionality: A number expressing a relationship and remaining the same in all circumstances for the same substance and the same units of a given equation. The constant's units and its numerical value are such that the units of the equation in which the constant is used balance and the relationship of the properties is correct, e.g., constant of proportionality appearing in the equation of state of an ideal gas, equal to the absolute pressure of the gas times its molar volume divided by its absolute temperature. Also known as the gas-law constant. (It might have more meaning to the layperson if called a finagle factor, but this implies dishonesty and there is no dishonesty associated with this constant.) These constants are required because our systems of units were derived before scientists understood nature's relationships. Octave Levenspiel, for example, points out that the gas law constant would not be required (it would equal 1.0) if science adopted a new absolute temperature scale.

cooling capacity (total cooling capacity): design maximum rate at which equipment removes heat from the air passing through it under specified conditions of operation.

cooling capacity, latent: design maximum rate at which equipment removes latent heat (reduces the moisture content of air passing through it) under specified conditions of operation.

cooling capacity, sensible: design maximum rate at which the equipment removes sensible heat.

cooling effect, latent (dehumidifying effect): the product of the mass of water vapour condensed during cooling times the latent heat of condensation at the temperature at which the condensation occurs.

cooling effect, sensible: (1) amount of sensible heat removed from air during cooling; (2) difference between total cooling effect and dehumidifying effect.

cooling effect, total: (1) difference between the total enthalpy of the mixture of dry air and water vapour entering and leaving the cooler; (2) amount of sensible and latent heat removed from the conditioned space.

cooling load: amount of cooling per unit time required by the conditioned space.

cycle: (1) process or series of processes wherein the initial and final states of the system are identical. At the conclusion of a cycle, all the properties have the same values as at the beginning; (2) a complete course of operation of working fluid back to a starting point, measured in thermodynamic terms (functions).

Dalton's law: each constituent of a mixture of gases behaves as if it alone occupied the space. The sum of the individual pressures of the constituents equals the total pressure of the mixture.

dehumidification: removal of water vapour from air.

dehumidifying effect: mass of water condensed during cooling or the equivalent refrigerating capacity expressed in terms of the latent heat of the water condensed per unit of time.

density: mass per unit of volume. In air moving systems, density is expressed as mass per unit volume, kg/m^3 (lb/ft^3). In psychrometrics it is undefined because it is expressed both as (1) mass per unit volume of the moist air mixture and (2) as mass per unit volume of the dry-air component only. In psychrometrics it is recommended that the reciprocal of **specific volume** be used in lieu of density to avoid confusion and the resultant errors.

deposition: in meteorology, the process by which H_2O changes phase directly from a vapour into a solid without first becoming a liquid. Antonym of **sublimation**.

desiccant: absorbent or adsorbent, liquid or solid, that removes water or water vapour from a material.

design air temperature: air temperature that an HVAC system or apparatus is designed to maintain (inside design) or to operate against (outside design).

design airflow: required airflow when the system is operating under assumed maximum conditions of design, including diversity.

design conditions: specified environmental conditions, such as temperature and humidity, required to be produced and maintained by a system.

design load (peak load): peak instantaneous load that a system is expected to meet.

dew: (1) deposit of water droplets on cold surfaces formed by the condensation of water vapour; (2) hydrometeor produced on objects at or near the ground.

dew-point temperature: temperature at which water vapour has reached the saturation point (100% relative humidity). Compare **frost point**. *Not*e*:* It is improper to refer to the dew point as the temperature at which condensation starts to occur because condensation at the dew point requires removal of latent heat from the vapour to induce condensation, and this can occur only if the vapour is cooled below the dew point.

diffusion: displacement of the molecules of a fluid within another fluid.

draw-through (coil or heat exchanger): refers to the arrangement of air-side components in which the coil or heat exchanger is located upstream (on the entering air side) of the air moving device (fan or blower). Hence the air moving device *draws-through* the coil.

dry-bulb temperature: temperature of air indicated by an ordinary thermometer.

energy: (1) capability for doing work; (2) capacity for producing an effect. Energy has several forms, which may be either *stored* or *transient*, and can be transformed from one into another. Forms include heat, mechanical work (shaft work), mechanical work (flow work), internal (molecular), potential, kinetic, electrical, radiant, and chemical.

energy, internal (molecular): sum of all kinetic and potential energies contained in a substance or system due to the states of motion and separation of its molecules, atoms, and electrons. It includes sensible heat (vibration energy) and latent heat represented by the increase in energy during evaporation.

energy, kinetic (molecular): energy caused by the velocity of the molecules.

energy, mechanical: capacity for doing work, usually expressed in work units (foot-pounds or newton-metres), sometimes in heat units (Btu or joule). Energy may be inherent in the speed of a body (**kinetic energy**) or in its position relative to another body (**potential energy**).

energy, potential (molecular): energy possessed by a system caused by the attractive forces existing between molecules.

energy, thermal: a dual-meaning term, which can be a synonym for **heat** (energy crossing a system boundary due to a difference in temperature) or **internal energy** (energy possessed by a system caused by the motion of the molecules and/or intermolecular forces).

enhancement factor: in moist air psychrometrics, the factor fs in the equation $W_S = 0.622 \cdot fs \cdot p_{WVS} / (p_{BAR} - fs \cdot p_{WVS})$ that accounts for the non-ideal behavior of saturated water vapour in air due to (a) the effect of dissolved gases on the properties of the condensed (liquid or solid H_2O) phase, (b) the effect of pressure on the properties of the condensed phase, and (c) the effect of intermolecular force (gas imperfections) on the properties of the moist air itself. See Appendix V, “Real Gas Models.”

enthalpy: thermodynamic quantity equal to the sum of the internal energy of a system plus the product of the pressure-volume work done on the system; $h = u + pv$, where h = enthalpy or total heat content, u = internal energy of the system, p = pressure, and v = volume. Enthalpy cannot be directly measured; however, enthalpy differences between the initial and the final statepoints of a process can be measured. Accurate values of enthalpy have been determined in research laboratories by measuring the changes of properties during a carefully controlled process, and from these properties enthalpies relative to an *arbitrary reference statepoint* (0°C for moist-air psychrometrics) are calculated. The value of enthalpy at the arbitrary reference point is usually designated as having zero enthalpy. The resultant enthalpy values are published in the form of tables, graphs, and curve fit equations.

enthalpy humidity ratio: the increase of enthalpy per unit mass of air, with its associated moisture, to the rise of humidity ratio under conditions of constant pressure.

enthalpy, specific: enthalpy per unit mass of a substance. In psychrometrics, the enthalpy of air (moist air) is per unit mass of the *dry-air* component of the mixture of *dry air* and *water vapour.*

equation of state: thermodynamic expression relating the volume, pressure, and temperature of a given substance.

equilibrium: steady-state condition, during which the fluctuations of variables remain within prescribed operating tolerances.

evaporating temperature: temperature at which a portion of a fluid evaporates (vaporizes).

evaporation: change of phase from liquid to vapour. Antonym of **condensation**.

exfiltration: opposite of **air infiltration**.

flow work: energy carried into or transmitted across the system boundary because a pumping process occurs somewhere outside the system, causing fluid to enter the system. Flow work also occurs as fluid leaves the system.

flux: amount of some quantity flowing across a given area (often a unit area perpendicular to the flow) per unit time. *Not*e: The quantity may be, for example, mass or volume of a fluid, energy, or number of particles.

fog: fine airborne droplets usually formed by condensation of vapour.

freezing: process of changing a liquid substance or the liquid content of a food or other commodity to a solid phase by removing heat.

freezing point: for a particular pressure, the temperature at which a given substance will solidify or freeze upon removal of heat.

frost point: temperature at which visible frost forms on a surface being chilled.

fusion: change of phase from liquid to solid. Antonym of **melting**.

gas: a phase (form of aggregation of molecules) of matter in which substances exist in the form of nonaggregated molecules and which, within acceptable limits of accuracy, satisfy the ideal gas laws; usually a highly superheated vapour.

grain(s) of moisture: a convenient I-P unit for expressing the humidity ratio of the mass of water vapour associated with the dry-air component of air; 1 grain = 1/7000 pound avoirdupois (historically, the average weight of a grain of wheat).

heat: (1) energy that is transferred in the direction of lower temperature; (2) form of energy that is exchanged between a system and its surroundings or between parts of the system induced by temperature difference existing between them.

heat flow: passage of heat from one point to another or one space to another by one or more of the three modes: conduction, convection, and radiation. Note that energy can also be transferred across a boundary by the mechanism of mass transfer across a boundary. In psychrometrics, this is sometimes called *latent heat flow*; however, since **heat** is rigidly defined in thermodynamics as energy crossing a boundary due to a temperature difference, the energy transfer associated with a mass transfer should be designated *latent energy flow*.

heat flux (density of heat flow rate): heat flow rate from a surface per unit area.

heat gain (heat uptake): quantity of heat absorbed by an enclosed space or system.

heat gain, solar (sun effect): solar energy transferred into a building, through both windows and structural materials.

heat of fusion (fusion heat): (1) latent heat involved in changing between the solid and the liquid phases; (2) heat energy required to cause a change of phase from solid to liquid at constant temperature. For ice to water: 333.39 kJ/kg (143.33 Btu/lb).

heat of the liquid: enthalpy of a mass of liquid above an arbitrary zero.

heating load: heating rate required to replace heat loss from the space being controlled.

humidify: to add water vapour to the atmosphere.

humidifying effect: product of the mass of water evaporated times the latent heat at the evaporating temperature.

humidity, percentage: the ratio in percent of the mass of moisture at a given temperature (humidity ratio) to the maximum possible (humidity ratio saturated) at the same temperature.

humidity ratio (mixing ratio) (humidity *x*): ratio of the mass of water vapour to the mass of dry air in a parcel or volume of air. See **mixing ratio**.

humidity ratio, saturation: ratio of the mass of water vapour to the mass of dry air in a parcel or volume of air that is saturated with water vapour.

hygrometer: from the Greek *hygros* (wet) and *metron* (a measure); a device (instrument) for determining the humidity of the atmosphere. There are at least five types of hygrometers: (1) those based on the principle of evaporation, which include the sling psychrometer and the aspirating psychrometer, both based on the fact that the less humid the air the greater the evaporation and the associated cooling effect; (2) dew-point temperature apparatus, which carefully regulates the temperature of a surface to make water vapour from the air alternatively condense on a surface and then re-evaporate (the **dew-point temperature** is the temperature when condensation is first detected or observed); (3) apparatus that uses hygroscopic materials that change in dimension with a change in humidity (human hair, wood, paper, or gold beaters skin made from ox gut); (4) chemical absorption and gravimetric determination; (5) apparatus that measures the difference in electrical capacitance or resistance of a material with respect to the amount of water vapour in the air.

hygrometry: a branch of physics that deals with the measurement of humidity, especially humidity in the atmosphere.

hygroscopic: capable of absorbing and retaining/losing moisture.

ideal gas (perfect gas): gas whose internal energy and enthalpy depend solely on temperature and which is defined by $pv = RT$.

ideal gas law (perfect gas law): the equation of state of a named ideal gas is a good approximation to real gases at sufficiently high temperatures and low pressures; that is, $p \cdot v = \mathrm{R} \cdot T$, where p = pressure, v = specific volume, T = absolute temperature, and R = gas constant for the named gas. The gas constant for the named gas (R) is equal to the universal gas constant ($\mathrm{R_U}$) divided by the molecular mass (M) of the named gas.

I-P units (inch-pound units): units using inches, pounds, and other designations, as opposed to **SI units**. Examples are foot, Btu, horsepower, and gallon.

isenthalp: thermodynamic change at constant enthalpy.

isenthalpic expansion: the expansion of a fluid during which no heat or mechanical energy is exchanged with the surroundings—the enthalpy remains constant.

isentrope: line of equal or constant entropy.

isentropic process (reversible adiabatic): thermodynamic change at constant entropy.

isobar: (1) an isoline of constant pressure on a graph; (2) line at one equivalent sea level barometric pressure on a weather map.

isobaric: a thermodynamic change at constant pressure.

isochor: an isoline of constant volume on a graph that shows the variation of one quantity with another; e.g., the variation of pressure with temperature.

isolated system: in thermodynamics, a system that can interchange neither matter nor energy with its surroundings.

isopsychric: state of constant relative humidity.

isothermal: process at constant temperature.

kelvin temperature: SI absolute temperature scale on which the triple point of water is 273.16 K (a 1 K temperature difference is also a 1°C temperature difference). See **triple point**.

lapse rate: the decrease in temperature of the atmosphere at increasing altitudes. ICAO and NASA define a "standard" lapse rate within the troposphere in defining the **standard atmosphere**.

lapse rate, dry adiabatic: the rate at which a rising parcel of air not saturated with water vapour cools as it ascends. The dry adiabatic lapse rate in the troposphere is about 10°C per 1000 m increase in altitude.

lapse rate, moist adiabatic: the rate at which a rising parcel of air saturated with water vapour cools as it ascends. The moist adiabatic lapse rate in the troposphere is about 6.5°C per 1000 m increase in altitude. This rate is less than the dry adiabatic lapse rate because some of the expansional cooling is compensated for by the release of latent heat that accompanies the condensation or deposition of the water vapour.

latent heat: change of enthalpy during a change of phase. In psychrometrics, only the H_2O component of the moist air mixture changes phase and the change in enthalpy is the product of the mass of H_2O evaporated (or condensed) times the latent heat of evaporation (or condensation) at the temperature at which the phase change occurs.

latent heat load (moisture): cooling load to remove latent heat.

latent heat of condensation: quantity of heat released on change of unit mass of vapour to liquid with no change in temperature.

latent heat of evaporation: heat energy required to cause a change of phase from a saturated liquid to a saturated vapour, measured in J/kg (Btu/lb mass). Also known as **vaporization**.

latent heat of sublimation: heat energy required to cause a change of phase from a saturated solid to a saturated vapour, measured in J/kg (Btu/lb).

liquefaction: change of phase from gas to liquid; generally used instead of the term **condensation** in cases of substances such as nitrogen and oxygen that are gases at the temperatures and pressures at the earth's surface.

liquid: phase of matter intermediate between crystalline substances and gases in which the volume of a substance, but not the shape, remains relatively constant.

load: amount of heat per unit time imposed on a refrigeration system by the required rate of heat removal.

mass: quantity of matter in a body; measured in terms of resistance to acceleration by a force, i.e., its inertia. In SI, the standard unit of mass is the kilogram (kg); in I-P, it is the slug.

mass flow rate: mass of a substance flowing per unit time.

mass transfer (mass transport): transfer of one component of a mixture relative to the motion of the mixture. It is the result of a concentration gradient. Compare **heat**.

mechanical (shaft) work: energy delivered or absorbed by a mechanism, such as a turbine, air compressor, or internal combustion engine.

mechanical equivalent of heat: in SI, one joule of heat equals one joule of mechanical work.

melting: change of phase from solid to liquid caused by absorption of heat.

melting point: for a given pressure, the temperature at which the solid and liquid phases of the substance are in equilibrium.

meteorology: scientific study of the atmosphere and atmospheric processes.

mixing ratio: in meteorology and psychrometrics, a contraction of "mass of water vapour to mass of dry air mixing ratio." A further contraction substitutes the word *humidity* for *water vapour* and omits the words *mass* and *dry air* to arrive at **humidity ratio**.

moisture: (1) water vapour; (2) water in a medium such as soil or insulation, but not bulk water or flowing water. The term *moisture* is essentially undefined. It is often used to represent a small quantity of H_2O in either the liquid or vapour form.

molar (molecular) mass: the mass of one mole of a substance. A mole of gas contains approximately $6.022 \cdot 10^{23}$ molecules (Avogadro's number). One mole is also defined as the number of atoms in a 12 g sample of carbon 12.

mol fraction water vapour: in humid air, the ratio of the number of water vapour mols (or molecules) to the total number of mols (or molecules) in the mixture.

mole (mol): amount of substance of a system that contains as many elementary entities as there are atoms in 0.012 kg of carbon 12 (approximately, $6.022 \cdot 10^{23}$). *Not*e: When the mole is used, the elementary entities must be specified and may be atoms, molecules, ions, electrons, other particles, or specified groups of such particles.

molecular diffusion: diffusion of the molecules of a substance in the absence of convective movement of the substance.

open system: in thermodynamics, a system that involves the transfer of mass across a boundary to and/or from the system.

partial pressure: portion of total gas pressure of a mixture attributable to one component (Dalton's law of gases).

particle: (1) any relatively small subdivision of matter, ranging in diameter from a few nanometres (as with gas molecules) to a few millimetres (as with large raindrops); (2) any very small part of matter, such as a molecule, atom, or electron. Also known as *fundamental particle*.

particle concentration: number of individual particles per unit volume of air.

Pascal's law: confined fluid transmits externally applied pressure uniformly in all directions, without change in magnitude.

phase: in thermodynamics, one of the three phases of matter (solid, liquid, or gas).

phase change: change of a material or system from one phase to another such as liquid to gas, liquid to solid, or solid to gas.

phase, gas: one of the three phases of matter characterized by the greatest freedom of molecules and the lack of any inherent fixed shape or volume.

phase, liquid: one of the three phases of matter characterized by limited freedom of molecules and by substantial incompressibility. See also **liquid**.

phase, solid: one of the three phases of matter characterized by stability of dimensions, relative incompressibility, and molecular motion held to limited oscillation.

physical laws: properties of physical phenomena or relationships between the various quantities or qualities that describe the phenomena and apply to all members of a broad class.

physical properties: properties discussed in physics, exclusive of those described under mechanical properties; e.g., density, electrical conductivity, coefficient of thermal expansion.

pressure: (1) thermodynamically, the normal force exerted by a homogeneous liquid or gas, per unit of area, on the wall of the container; (2) force exerted per unit area. *Not*e: The terms *head* and *pressure* often are used interchangeably; however, *head* is the height of a fluid column supported by fluid flow, while *pressure* is the normal force per unit area. For liquids, it is convenient to measure the head in terms of the flowing fluid. With a gas or air, however, it is customary to measure pressure on a column of liquid. In SI psychrometrics, the pascal or kilopascal is the unit of pressure. Due to a century of use, it is represented by meteorologists in the *head* form expressed as mm $\text{Hg}_{0°\text{C}}$ (101,325 Pa = 760 mm Hg = 29.92125 in. Hg = 14.69595 psia).

pressure, air: the cumulative force exerted on any surface by the molecules composing air; usually expressed as the weight of a column of air per unit surface area.

process: change in thermodynamic state that can be defined as any change in the properties of a system. *Note*: A process is described by specifying the initial and final equilibrium states, the path (if identifiable), and the interactions that take place across system boundaries during the process.

property: of a system, any observable characteristic of the system. The **state** of a system is defined by listing its properties.

psychometrics: the scientific measurement of mental capacities and processes and of personality.

psychrometer: instrument for measuring wet- and dry-bulb temperatures. Many psychrometers include an integral slide rule used to display relative humidity given wet- and dry-bulb temperatures.

psychrometer, aspirated: psychrometer having mechanical means for rapidly circulating air to be tested over dry and wet bulbs.

psychrometric: *adj.*, as in “psychrometric chart.”

psychrometric chart: graphical representation of the properties of moist air, usually displaying isolines of dry-bulb, wet-bulb, and dew-point temperatures, humidity ratio, relative humidity, specific enthalpy, and specific volume.

psychrometrics: subscience of physics dealing with the properties and processes of moist air. Moist air is defined as a mixture of two gases: dry air and water vapour (the gas phase of H_2O). Dry air within the troposphere is treated as a nonvarying mixture of nitrogen, oxygen, argon, and other trace gases including carbon dioxide (for exact composition see Table 3-1). Some broaden the definition of psychrometrics to cover mixtures of the gas of one substance (any dry gas component) and the condensable vapour of a second substance.

psychrometry: branch of physics concerned with the measurement or determination of atmospheric conditions, particularly the moisture in air.

refrigeration: process of extracting heat from a substance or space by any means, usually at a low temperature.

relative humidity: (1) ratio of the partial pressure of water vapour in moist air to the saturation pressure at the same dry-bulb temperature and barometric pressure of the ambient air; (2) ratio of the mole fraction of water vapour to the mole fraction of water vapour saturated at the same temperature and barometric pressure.

room dew-point temperature: that temperature at which the water vapour reaches saturation.

saturated: contains the maximum amount possible.

saturated liquid: liquid existing at the saturation temperature and pressure.

saturation: condition for coexistence in stable equilibrium of a vapour and liquid or a vapour and solid phase of the same substance (e.g., steam over the water from which it is being generated). See **vapour pressure**.

saturation pressure: for a pure substance *at a given temperature*, the pressure at which vapour and liquid, or vapour and solids, can exist in equilibrium.

scfm (standard cubic feet of air per minute): volumetric flow rate of air having a density of 0.075 lb_{DA}/ft^3 or its reciprocal v = 13.3333 ft^3/lb_{DA}. This density occurs for psychrometric statepoints from 60.15°F and 100% RH to 69.41°F and 0% RH at 14.69595 psia barometric pressure. The somewhat comparable SI values are 1.204 kg_{DA}/m^3 or its reciprocal v = 0.83056 m^3/kg_{DA} at 15.17°C and 100% RH to 20.15°C at 0% RH. A large majority of air-conditioning systems are designed for this range of conditions at or near sea level altitude. The scfm concept permits the use of standardized rating tables for equipment. It was also a useful approximation or simplification for design of systems at or near sea level in the pre-computer era of air conditioning. Innocent or blind reliance using equations based on scfm leads to a shortfall in capacity for projects at altitudes above sea level.

sensible heat: heat that causes a change in temperature that can be sensed. The substance in psychrometrics is air (moist air). Sensible heat includes not only the change in the temperature of the dry-air component but also the change in temperature of the water-vapour component.

sensible heat load: cooling load to remove the sensible heat.

sensible heat ratio: ratio of the sensible heat to the sensible plus latent heat (i.e., the total) to be removed from a conditioned space.

SI units: Le Système International d' Unités; the international agreement on the metric system of units.

specific (as a prefix to a property): in most branches of science, the prefix to a substance property (e.g., volume; thus, *specific volume*) results in the division of that property by the mass of that substance associated with the volume. In the case of a volume with units of m^3, the *specific volume* (or the volume per unit mass) has units of m^3/kg. In psychrometrics, the properties of *specific enthalpy, specific volume*, *humidity ratio*, and *specific heat of moist air* are all expressed per unit mass of the dry-air component of the moist air mixture.

specific heat: ratio of the quantity of heat required to raise the temperature of a given mass of any substance one degree to the quantity required to raise the temperature of an equal mass of a standard substance one degree (usually water at 59°F [15°C]).

specific heat capacity: amount of heat necessary to raise the temperature of a given mass one degree. Specific heat capacity is usually stated at an average value over a specified temperature range.

specific humidity: ratio of the mass of water vapour to the total mass of a moist air sample. Often incorrectly used as a synonym for **humidity ratio** (the mass of water vapour to the mass of the dry-air component of a moist air sample).

specific volume: usually defined as the volume of a unit mass of a material and expressed in cubic metres per kilogram (cubic feet per pound). Usually the reciprocal of density. In psychrometrics, specific volume is defined as the volume of a moist air sample per unit mass of the dry-air component.

specific volume, saturated: specific volume of an air parcel saturated with water vapour.

standard airflow (See also scfm): an I-P term expressing the volumetric rate of flow of air at arbitrary standard conditions. There is no single equivalent term in SI units. Some use as the reciprocal of specific volume 1.204 kg_{DA}/m^3 (dry air at 20°C and 101.325 kPa) while others use 1.2 kg_{DA}/m^3. This was a useful concept in the era of manual calculations and is still useful when manufacturers provide tables of equipment performance. In the current era of software calculations and software programs for equipment selection, the concept of standard air should probably be abandoned because it serves no useful purpose and can lead to errors when the reciprocal of specific volume differs significantly from the "standard" values above.

standard atmosphere: in 1925 aeronautical and meteorological researchers and practitioners developed the concept of the standard atmosphere in order to standardize calculations used in aircraft altimeter calibration and in the design and performance analysis of aircraft and missiles. The standard atmosphere (within the troposphere) is defined to have a sea level temperature of 15°C, a sea level barometric pressure of 101,325 Pa, and a lapse rate of 6.5°C per 1000 m increase in altitude above sea level, and the upper boundary of the troposphere (i.e., the tropopause) is the altitude at which the temperature has decreased to –56.5°.

state: condition or stage in the physical being of something. In moist-air psychrometrics, the state of the moist air is fixed knowing barometric pressure and two other independent properties such as dry-bulb temperature and one of (a) wet-bulb temperature, (b) relative humidity, (c) dew-point temperature, or (d) humidity ratio. Also referred to as *statepoint*.

steady state: state of a system in which movement of matter or energy phenomena are taking place when the various physical phenomena are independent of time.

subcooling: cooling a substance below its saturated condition or its freezing temperature.

sublimation: change of phase directly from solid to gas without passing through a liquid phase. Antonym of **deposition**.

superheat: extra heat in a vapour when at a temperature higher than the saturation temperature corresponding to its pressure.

superheated vapour: vapour at a higher enthalpy than the enthalpy at saturation.

surroundings: in thermodynamics, everything beyond the boundary that separates the system from the surroundings.

system, thermodynamic: in thermodynamics, a region in space or a quantity of matter bounded by a closed surface in which thermal and/or work actions occur. The surroundings include everything external to the system, and the system is separated from the surroundings by the system boundaries. These boundaries can be either movable or fixed, either real or imaginary.

temperature: thermal state of two adjacent substances that determines their ability to exchange heat. Substances in contact that do not exchange heat are at the same temperature. Temperatures are indicated on defined scales, such as Kelvin and Rankine for absolute temperatures and Celsius and Fahrenheit for ordinary temperatures.

thermodynamic equilibrium: equilibrium in a system when the physical variables have uniform values that do not change in time. Furthermore, if the system is not an isolated one, these variables should have the same values for both the system and its surroundings.

thermodynamic properties: basic qualities used in defining the condition of a substance, such as temperature, pressure, volume, enthalpy, entropy, etc.

thermodynamic wet-bulb temperature: the saturation temperature to which moist air can be adiabatically (without gain or loss of heat) cooled by evaporation of H_2O (liquid or ice) at that same (saturation) temperature into the moist air. For any given moist air state, thermodynamic wet-bulb temperature has a fixed mathematical value based on thermodynamic properties. Also known as *the temperature of adiabatic saturation*.

thermodynamics: the science in which the storage, transformation, and the transfer of energy are studied. Energy forms for HVAC analysis include shaft work, flow work, heat, internal energy, enthalpy, kinetic energy, and potential energy. The terms *system*, *boundary*, and *surroundings* are rigidly defined and used. *System* is a region of space or a quantity of matter set aside for study. The system is separate for the purpose of study or analysis. *Boundary* distinguishes the limits of the system. *Surroundings* are everything outside of the boundary.

thermodynamics, first law of (conservation of energy): heat and work are mutually convertible; because energy can neither be created nor destroyed, the total energy associated with an energy conversion remains constant.

total heat: a misleading term commonly used in lieu of *enthalpy difference*.

triple point: particular temperature and pressure at which the three phases of a substance can coexist in equilibrium. Water is an example of a substance that has a well-known triple point (273.16 K, 0.01°C).

vaporization: change of phase from liquid to vapour. Also known as **evaporation**.

vapour: (1) substance in the gaseous phase that can also exist as a liquid or solid at normal atmospheric conditions; (2) gas, particularly one near equilibrium with its liquid phase and one that does not follow the gas laws. The term is usually used instead of **gas** to refer to a refrigerant or, in general, to any gas below the critical temperature.

vapour concentration (absolute humidity; water vapour density number): in a mixture of water vapour and dry air, the mass of water vapour in a specific volume of the mixture.

vapour pressure: pressure exerted by a vapour. If a vapour is kept in confinement over its liquid or solid so that the vapour can accumulate above the substance with the temperature constant, the vapour pressure reaches a maximum called the **saturated vapour pressure**. *Note*: For a specific liquid or solid, at constant volume of vapour above it, the vapour pressure depends only on the temperature.

vapour pressure, saturated: pressure at which vapour and liquid, or vapour and solid, can exist in equilibrium at a given temperature; the vapour pressure of the water vapour associated with an air parcel that is saturated with respect to water vapour.

vapour, saturated: vapour at the saturation temperature corresponding to the existing pressure and without any liquid phase.

vapour, superheated: vapour at a temperature greater than the saturation temperature. The pressure and temperature of superheated vapour are independent properties, since the temperature can increase while the pressure remains constant. See **superheat**.

vapour, wet: saturated vapour containing liquid droplets in suspension. In psychrometrics, **fog**.

velocity pressure: in a moving fluid, the pressure due to the velocity and density of the fluid; expressed by the velocity squared times the fluid density, divided by two.

ventilation: process of supplying or removing air by natural or mechanical means to or from any space. Such air may or may not have been conditioned.

virial: a simple function of the positions of and the forces acting upon the molecules of a system, summed over that system.

virial coefficient: in psychrometrics, each of the coefficients in a polynomial series used to approximate the quantity $pv/\mathrm{R}T$ in the virial equation of state. They account for the forces of attraction and repulsion between molecules. The virial coefficients are dependent on temperature only. See Appendix V, “Real Gas Models.”

virial equation of state: an equation that is used to determine the state of a real gas or similar collection of particles. In psychrometric modeling of moist air and its constituents as real gases, the virial equation of state has the form $pv/\mathrm{R}T = 1 + [B/v + C/v^2 + D/v^3]$ or an alternative form $pv/\mathrm{R}T = 1 + [B'p + C'p^2 + D'p^3]$. See Appendix V, “Real Gas Models.”

water: transparent, odorless, tasteless liquid; a compound of hydrogen and oxygen containing 11.188% hydrogen and 88.812% oxygen by mass; freezing point at 0°C (32°F); boiling point near 100°C (212°F). Water is also a refrigerant with the designation R-718.

water vapour: water in the vapour or gas phase.

water vapour pressure: (1) the portion of atmospheric pressure exerted by the associated water vapour at a specified temperature; (2) pressure due to water vapour.

weather: the state of the atmosphere at some place and time in terms of such variables as temperature, humidity ratio, cloudiness, precipitation, solar radiation, etc.

wet-bulb temperature: indicated by a wet-bulb sensing element when the sensor is covered with a water-saturated wick over which air is caused to flow, at approximately 4.5 m/s (900 ft/min), to reach an equilibrium temperature of water evaporating into air, when the heat of vaporization is supplied by the sensible heat of the air. Wet-bulb temperature herein is not a thermodynamic property (see **thermodynamic**

wet-bulb temperature). It is fortuitous for the mixture of dry air-water vapour that psychrometer and thermodynamic wet-bulb temperatures are for practical purposes equal (provided that the psychrometer is correctly used).

work: mechanism that transfers energy across the boundary of systems with differing pressures (or force of any kind), always in the direction of lower pressure; if the total effect produced in the system can be reduced to the raising of a weight, then nothing but work has crossed the boundary.

Bibliography

This bibliography is not intended to be exhaustive. It may help those who wish to explore the subject in more detail. A list such as this would have made the author's research into the subject both quicker and easier. Dan D. Wile included an excellent list of references in his papers, which, unfortunately, were not discovered until the final stages of this book. As a first step in further exploration, read the "Ideal Gas" and "Ideal Gas Mixtures" chapters of one or two college level thermodynamics texts, such as Stoecker (1982) or Threlkeld (1970), and follow these with Goodman (1944) and Carpenter (1983). (The Goodman book is sometimes hard to find but is worth the effort if you can ignore the author's adding ten to all enthalpy values in order to avoid negative numbers.) For practical applications, read Ramsey (1966), Sun (1994), and Haines (1988).

The Internet is another source of psychrometric information; however, most of the material has not been peer reviewed. Expect to find some incorrect terminology. Psychrometrics is used by meteorologists, museum conservators, food science, agricultural and chemical engineers, and, somewhat astonishingly, more Web pages are found in these areas than in air conditioning. In 1996, the author searched the web and found three psychrometric listings. In 2000, a second search produced 1156 hits! Obviously some of the 1156 are duplicates. About five percent of the hits were course lectures dealing with psychrometrics, moist air, or gas-vapour mixtures.

Major air-conditioning, humidification, and dehumidification manufacturers have psychrometric handouts used with educational slide presentations. These generally start with a discussion of the psychrometric chart and avoid underlying principles. As with Internet materials, expect to find some incorrect terminology.

Note: Entries in bold are of particular interest.

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Appendices

Appendix I—ICAO Equation Relating Barometric Pressure and Altitude

THE ICAO EQUATION RELATING BAROMETRIC PRESSURE AND ALTITUDE ABOVE OR BELOW SEA LEVEL

Aeronautical and meteorological researchers and practitioners developed the concept of the *standard atmosphere* in order to standardize calculations used in aircraft altimeter calibration and in the design and performance analysis of aircraft and missiles. In 1925, they developed the equation (shown at the end of this appendix) for the *standard atmosphere* relating p_{BAR} and Z, altitude (above or below sea level), within the troposphere. The derivation, constants, and underlying assumptions for this equation are covered at the end of this appendix.

This equation applies only within the troposphere—the lowest level of the atmosphere. The troposphere is the atmospheric layer of the earth's weather in which clouds form and from which rain, sleet, hail, and snow fall. In the troposphere, the temperature decreases at an average lapse rate of 6.5°C per kilometre of elevation increase above sea level until the temperature reaches –56.5°C. The upper boundary of the troposphere is called the tropopause and is defined as the altitude at which the temperature has decreased to –56.5°C.

The altitude of the tropopause (upper boundary of the troposphere) varies somewhat throughout the year, but on average it is about 11 km in middle latitudes (45° N or S), 16 km in the tropics (22° N to 22° S), and 8 km over the north and south poles. The lower portion of the troposphere is of interest in the field of psychrometrics because all land-based activities occur from 3000 metres below sea level in deep mines to 3000 metres above sea level at high observatories.

The relationship between mean atmospheric pressure and altitude has its history mainly in the field of aeronautics, but long before the airplane it had its beginnings in France in 1648 (see Blaise Pascal in Chapter 8, "Psychrometric Pioneers and Charts from the First 100 Years") and later in the 1860s in connection with ballooning and mountaineering. In J.W. Powell's account of the 1869–1872 mapping of the Colorado river through the Grand Canyon, he discussed the use of mercury barometers for measuring the heights of the surrounding cliffs.

The present definition of the *standard atmosphere* comes from the International Civil Aviation Organization (ICAO) and NASA. ICAO and NASA define the pressure and density of *standard atmosphere* at sea level and at various altitudes. The reference location for the *standard atmosphere* is mid-latitude (45°) at sea level with a mean average temperature of 15°C (288.15 K) and a mean average pressure of 101.325 kPa. The standard assumes a constant composition within the troposphere because the air is constantly circulating and well mixed. The ICAO standard includes the following values. Note that the current RU, MDA, and RDA values shown below may be slightly different than those in the ICAO standard, which are assumed to be constant for the purposes of defining the standard atmosphere.

p_o = $101.325\ \mathrm{kPa}$
T_{KELVIN} = $t_{°C} + 273.15$
T_o = $15°\mathrm{C} + 273.15 = 288.15\ \mathrm{K}$
α = $0.0065°\mathrm{C/m} = 6.5°\mathrm{C/km}$
g = $9.80665\ \mathrm{m^2/s^2}$
R_U = $8314.472\ \mathrm{kJ/(kmol \cdot K)}$
M_{DA} = $28.966\ \mathrm{kg_{DA}/(kmol)}$
R_{DA} = $287.042\ \mathrm{J/(kg_{DA} \cdot K)}$

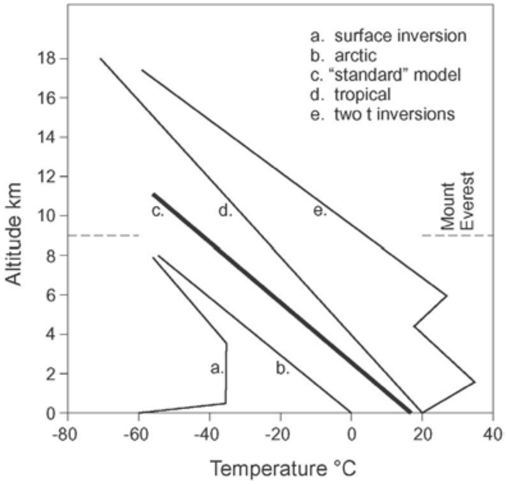

Figure A1-1—Different temperature lapse rates in the troposphere.

Figure A1-1 compares the *standard atmosphere* profile of p_{BAR} vs. Z with profiles over the equator, the poles, and two examples of actual measured profiles. The actual profiles show temperature inversions, which is not uncommon.

A standard atmosphere table (Table A1-1) within the troposphere uses the equation developed at the end of this appendix and includes altitudes from –3000 m to +11,000 m. Some of the values in this table differ insignificantly from earlier tables due to improved accuracy of the constants. The *density at 15°C* column has been added because many psychrometric process and fan performance calculations apply to inside conditions where the air temperature is closer to 15°C than the temperature in the second column. Note that the table is for *dry air*. The mass of water vapour associated with *dry air* has a minor effect on actual density.

Fan application and HVAC designers should note that ICAO and NASA standardize on a dry air density value at sea level and 15°C of 1.2250 kg/m^3. The HVAC industry generally bases "standard" calculations on a temperature of ~20°C at sea level, which, for dry air, results in a density of 1.204 kg/m^3. For 100% RH air, this is the density at ~15°C.

Table A1-1

Altitude, m	Temperature, °C	Pressure, kPa	Density, kg/m^3	Density at 15°C, kg/m^3
–3000	34.50	142.951	1.6187	1.7282
–2000	28.00	127.774	1.4781	1.5448
–1000	21.50	113.929	1.3470	1.3774
–500	18.25	107.478	1.2849	1.2994
0	15.00	101.325	1.2250	1.2250
500	11.75	95.461	1.1673	1.1541
1000	8.50	89.874	1.1116	1.0866
1500	5.25	84.556	1.0581	1.0223
2000	2.00	79.495	1.0065	0.9611
2500	–1.25	74.682	0.9569	0.9029
3000	–4.50	70.108	0.9091	0.8476
4000	–11.00	61.640	0.8191	0.7452
5000	–17.50	54.019	0.7361	0.6531
6000	–24.00	47.180	0.6597	0.5704
7000	–30.50	41.060	0.5895	0.4964
8000	–37.00	35.599	0.5252	0.4304
9000	–43.50	30.742	0.4663	0.3717
10000	–50.00	26.435	0.4127	0.3196
11000	–56.50	22.631	0.3639	0.2736

Density values are at the pressure in column 3; density in 4th column is at temperature in 2nd column; density in 5th column is at 15°C (288.15 K).

Yearly mean barometric pressure has traditionally been used in psychrometric calculations even though sea level temperature and pressure are seldom 15°C and 101.325 kPa and the temperature lapse rate seldom equals 6.5°C/km. If calculations require a precise value of barometric pressure, then pressure as measured by an accurate mercury barometer should be used.

Derivation of the ICAO Equation

The ICAO calculation of standard barometric pressure at altitude and temperature is based on the hydrostatic equation in which the difference in pressure between any two altitudes is equal to the weight of a column of air of unit cross section between the two altitudes. Other equations used in the derivation include the ideal gas equation of state and an equation describing the temperature lapse rate within the troposphere. The lapse rate applies throughout the troposphere. It is sometimes called the *standard* or *average lapse rate* to distinguish it from the *moist air lapse rate* and the *dry air lapse rate.* The standard lapse rate is a yearly average based on long-term observations.

$$dp = -g \cdot \rho \cdot dZ \qquad \text{(the hydrostatic equation)}$$

where

dp = the change in atmospheric pressure
ρ = average density of the layer in mass per unit volume
g = gravitational force on unit mass
Z = altitude above or below sea level

Air is assumed to behave as a ideal gas. One form of the ideal gas equation of state is

$$1/v = \rho = p_{ATM}/(R_{DA} \cdot T),$$

where

v = specific volume of air
ρ = density of air
p_{ATM} = atmospheric pressure at altitude
R_{DA} = gas constant for dry air
T = absolute temperature

Substituting the ideal gas equation in the hydrostatic equation gives

$$dp/p = -dZ \cdot g/(R_{DA} \cdot T),$$

where

Z = altitude above (+) or below (–) sea level
g = force of gravitational acceleration

The moist air temperature lapse rate of air in the troposphere equation is

$$T = T_O - \alpha \cdot Z,$$

where

T = temperature at the new altitude

T_O = original or datum temperature

α = rate of temperature decrease with increase in altitude in the troposphere

Substituting the temperature lapse rate equation into the previous equation gives

$$dp/p = -dZ \cdot g/[R_{DA}\,(T_O - \alpha \cdot Z)].$$

This equation can be changed by multiplying the right-hand side by α/α and by adding the constant T_O inside the differential, which does not change the differential. The resulting equation is

$$dp/p_O = [g/(\alpha \cdot R_{DA})] \cdot [d\,(T_O - \alpha \cdot Z)/(T_O - \alpha \cdot Z)].$$

Integrating between the limits of 0 and Z results in

$$\ln\,(p/p_O) = [g/(\alpha \cdot R_{DA})] \cdot \ln\,[(T_O - \alpha \cdot Z)/(T_O)].$$

This, in turn, results in the final forms of the equation:

$$p = p_O \cdot [(T_O - \alpha \cdot Z)/(T_O)]^{\,g/(\alpha \cdot RDA)}$$

$$p = p_O \cdot [(1 - \alpha \cdot Z/(T_O)]^{\,g/(\alpha \cdot RDA)}$$

Substituting the appropriate values for p_O, T_O, α, g, *and* R_{DA} gives

$$p = 101.325 \cdot [(1 - 0.0065 \cdot Z/(288.15)]^{\,9.80665/(0.0065 \cdot 287.042})$$

$$p = 101.325 \cdot [(1 - 2.25577 \cdot 10^{-5} \cdot Z)]^{\,5.256}$$

Note: The atmospheric pressure, p, is expressed as a function of altitude above sea level, Z, and an *average* lapse rate of 6.5°C per kilometre increase in altitude.

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