Thermodynamics Basics:
Gas Dynamics, Psychrometric Analysis, Refrigeration Cycle and HVAC Systems

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Preface

As the adage goes, "a picture is worth a thousand words;" this book maximizes the utilization of diagram, graphs and flow charts to facilitate quick and effective comprehension of the concepts of thermodynamics by the reader.

This book is designed to serve as a tool for building basic engineering skills in the filed of thermodynamics.

If your objective as a reader is limited to the acquisition of basic knowledge in thermodynamics, then the material in this book should suffice. If, however, the reader wishes to progress their knowledge and skills in thermodynamics to intermediate or advance level, this book could serve as a useful stepping stone.

In this book, the study of thermodynamics concepts, principles and analysis techniques is made relatively easy for the reader by inclusion of most of the reference data, in form of excerpts, within the discussion of each case study, exercise and self assessment problem solutions. This is in an effort to facilitate quick study and comprehension of the material without repetitive search for reference data in other parts of the book.

Certain thermodynamic concepts and terms are explained more than once as these concepts appear in different Segments of this text; often with a slightly different perspective. This approach is a deliberate attempt to make the study of some of the more abstract thermodynamics topics more fluid; allowing the reader continuity, and precluding the need for pausing and referring to Segments where those specific topics were first introduced.

Due to the level of explanation and detail included for most thermodynamics concepts, principles, computational techniques and analyses methods, this book is a tool for those energy engineers, engineers and non-engineers, who are not current on the subject of thermodynamics.

The solutions for end of the Segment self assessment problems are explained in just as much detail as the case studies and sample problem in the pertaining Segments. This approach has been adopted so that this book can serve as a thermodynamics skill building resource for not just energy engineers but engineers of all disciplines. Since all Segments and topics begin with the introduction of important fundamental concepts and principles, this book can serve as a "brush-up" or review tool for even mechanical engineers whose current area of engineering specialty does not afford them the opportunity to keep their thermodynamics knowledge current.

In an effort to clarify some of the thermodynamic concepts effectively for energy engineers whose engineering education focus does not include thermodynamics, analogies are drawn from non-mechanical engineering realms, on certain complex topics, to facilitate comprehension of the relatively abstract thermodynamic concepts and principles.

Each Segment in this book concludes with a list of questions or problems, for self-assessment, skill building and knowledge affirmation purposes. The reader is encouraged to attempt these problems and questions. The answers and solutions, for the questions and problems, are included under Appendix A of this text.

For reference and computational purposes, steam tables and Mollier (Enthalpy-Entropy) diagrams are included in Appendix B.

Most engineers understand the role units play in definition and verification of the engineering concepts, principles, equations and analytical techniques. Therefore, most thermodynamic concepts, principles and computational procedures covered in this book are punctuated with proper units. In addition, for the reader's convenience, units for commonly used thermodynamic entities, and some conversion factors are listed under Appendix C.

Most thermodynamic concepts, principles, tables, graphs, and computational procedures covered in this book are premised on US/Imperial Units as well as SI/Metric Units. Certain numerical examples, case studies or self-assessment problems in this book are premised on ***either*** the SI unit realm ***or*** the US unit system. When the problems or numerical analysis are based on only one of the two unit systems, the given data and the final results can be transformed into the desired unit system through the use of unit conversion factors in Appendix C.

Some of the Greek symbols, used in the realm of thermodynamics, are listed in Appendix D, for reference.

What readers can gain from this book:

- Better understanding of thermodynamics terms, concepts, principles, laws, analysis methods, solution strategies and computational techniques.
- Greater confidence in interactions with thermodynamics design engineers and thermodynamics experts.
- Skills and preparation necessary for succeeding in thermodynamics portion of various certification and licensure exams, i.e. CEM, FE, PE, and many other trade certification tests.
- A better understanding of the thermodynamics component of heat related energy projects.
- A compact and simplified thermodynamics desk reference.

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Segment 1

Gas Dynamics

Topics

- Gas Dynamics
- Steady Flow Equation
- Isentropic Flow
- Critical Point
- Shock Waves

Introduction

This segment is devoted to introduction of ***Gas Dynamics*** and topics within the realm of gas dynamics that are more common from a practical application point of view. Gas dynamics constitutes the study of gases moving at a high velocity. By most standards, a gas is defined as a high velocity gas when it is moving at a velocity in excess of 100 m/s or 300 ft/s. Fluid dynamics tools such as the Bernoulli's equation as well as the momentum and energy conservation laws, traditionally applied in mechanical dynamics study, do not account for the role internal energy plays in gas dynamics. Therefore, they cannot be applied in a comprehensive study of high velocity gases. In this segment, we will examine the behavior of high speed gases on the basis of key thermodynamic entities, such as enthalpy, **h,** and internal energy, **u**. The gas dynamics discussion is premised largely on the fact that a high velocity of a gas is achieved at the expense of internal energy, where the drop in internal energy, **u,** as supported by equation **Eq. 1.1**, results in the drop in the enthalpy, **h**.

$$\mathbf{h = u + p.v} \qquad \textbf{Eq. 1.1}$$

Steady Flow Energy Equation

Consider the high velocity flow scenario depicted in Figure 1.1 below. We will use this illustration to explain important characteristics and components of a high velocity gas flow system.

As shown in Figure 1.1, a high pressure reservoir is located on the extreme right. The properties of gas in this reservoir are referred to as the ***stagnation properties, chamber properties, or total properties.*** The gas possesses kinetic, potential, and thermal energy in all segments of the high velocity gas system. The thermal energy possessed by the gas is in the form of internal energy and enthalpy. The pressure and temperature of the gas in the reservoir are denoted by P_o and T_o, respectively. The gas in the reservoir is high pressure gas. This gas travels through the mid segment, referred to as the ***duct***. In the duct, the gas continues to be considered as high pressure and low velocity. The duct leads to the segment called the ***throat*** where the pressure drops and velocity escalates; thus transforming the gas into high velocity gas.

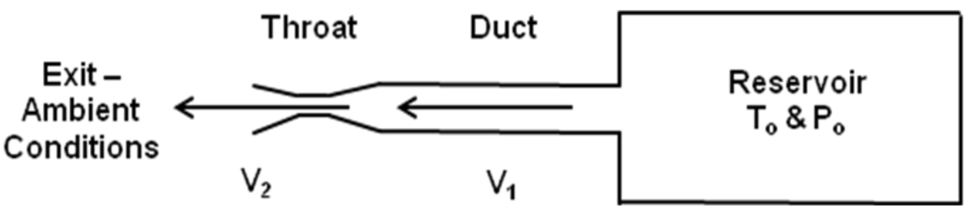

Figure 1.1 - High Velocity Flow

The flow of gas in the high velocity gas system is considered to be adiabatic because the high speed of gas (due to its short residence time in the throat) does not allow significant amount of heat exchange. In addition, in a simplified scenario, the length of the duct is considered to be short enough, such that no significant frictional head loss occurs. Also, as seen in Figure 1.1, the high velocity gas exits out to the ambient atmosphere; signifying its "**open-flow**" characteristic.

Since the high velocity gas system described above is an adiabatic open flow system, the **SFEE, Steady Flow Energy Equations** stated below would apply:

In the SI (Metric) Unit System:

$$\mathbf{g.z_1 + \tfrac{1}{2}.v_1^2 + h_1 = g.z_2 + \tfrac{1}{2}.\, v_2^2 + h_2} \qquad \textbf{Eq. 1.2}$$

In the US (Imperial) Unit System:

$$(g/g_c).z_1 + ½.(v_1^2/g_c) + J.h_1 = z_2.(g/g_c) + ½.(v_2^2/g_c) + J.h_2 \qquad \textbf{Eq. 1.3}$$

Where,

h_1 = Enthalpy of gas entering the throat, in kJ/kg (SI) or BTU/lbm (US).
h_2 = Enthalpy of gas exiting the throat, in kJ/kg (SI) or BTU/lbm (US).
v_1 = Velocity of the gas entering the throat, in m/s (SI) or ft/s (US).
v_2 = Velocity of the gas exiting the throat, in m/s (SI) or ft/s (US).
z_1 = Elevation of the gas entering the throat, in m (SI) or ft (US).
z_2 = Elevation of the gas exiting the throat, in m (SI) or ft (US).
g = 9.81 m/s^2 in the SI realm and 32.2 ft/s^2 in the US unit realm.
g_c = Gravitational constant, 32.2 lbm-ft/lbf-sec^2
J = 778 ft-lbf/BTU

Often, in practical gas dynamics scenarios, the exit velocity of the gas is the desired objective of analysis. Therefore, in practical scenarios, Equations 1.2 and 1.3 can be simplified to compute the exit velocity **v_2**. Since the reservoir area of the cross section is inordinately larger than the orifice or throat area of the cross section, the velocity, **v_1**, of gas in the reservoir is considered to be negligible; or, **$v_1 \cong 0$.** Since the density of gas is small, the potential energy component in Equations 1.2 and 1.3 can be disregarded. With these practical assumptions, Equations 1.2 and 1.3 can be distilled down to the following, simpler, practical forms:

$$v_2 = \sqrt{2(h_0 - h_2)} \qquad \text{\{SI Unit System\}} \qquad \textbf{Eq. 1.4}$$

$$v_2 = \sqrt{2g_c J(h_0 - h_2)} \qquad \text{\{US Unit System\}} \qquad \textbf{Eq. 1.5}$$

Case Study 1.1

A nozzle is fed from a superheated steam reservoir, shown in Figure 1.2. The duct or hose connecting the nozzle to the reservoir is short and the frictional head loss in the hose is negligible. Based on these practical assumptions, the velocity of the superheated steam in the hose can be neglected. The steam exits the nozzle at 150°C (300°F) and 0.15 MPa (21.76 psia). Determine the exit velocity of the steam at the nozzle.

Solution:

Given:

$\mathbf{T_o}$ = 300°C or 572°F

$\mathbf{P_o}$ = 2.0 MPa or 290 psia

$\mathbf{v_1 = v_0 = 0}$

$\mathbf{T_2}$ = 150°C or 300°F

$\mathbf{P_2}$ = 0.15 MPa or 21.76 psia

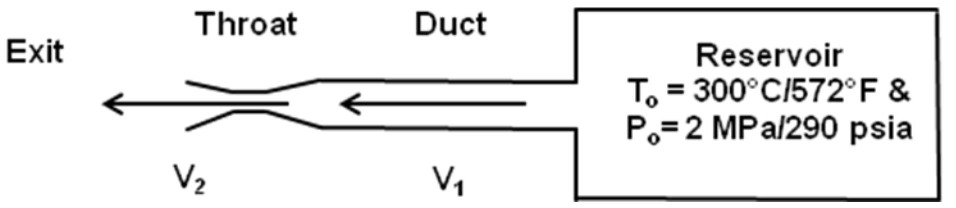

Figure 1.2 - High Velocity Flow, Case Study 1.1

SI Unit System:

Apply Eq. 1.4 to calculate the exit velocity of the superheated steam in the SI units:

$$v_2 = \sqrt{2(h_0 - h_2)} \qquad \{\text{SI Unit System}\} \qquad \textbf{Eq. 1.4}$$

From the steam tables in Appendix B, in the SI units:

$\mathbf{h_o = 3024\ kJ/kg}$

$\mathbf{h_2 = 2773\ kJ/kg}$

Then, by applying **Eq. 1.4**:

$$v_2 = \sqrt{(2).(3024 - 2773 kJ / kg).(1000 J / kJ)}$$

Note: The multiplier 1000J/kJ, in the equation above, is used to convert kJ to Joules as Eq. 1.4 is premised on Joules and not kilo Joules

$$v_2 = 709 m / s$$

US Unit System:
Apply **Eq. 1.5** to calculate the exit velocity of the superheated steam in the US units:

$$v_2 = \sqrt{2g_c J(h_0 - h_2)} \qquad \text{\{US Unit System\}} \qquad \textbf{Eq. 1.5}$$

From the steam tables in Appendix B, in the US units, and through double interpolation:

h_o = 1299 BTU/lbm

h_2 = 1191BTU/lbm

Note: These enthalpies can also be read from the Mollier diagram without interpolation; albeit, the results might differ slightly.

Then, by applying **Eq. 1.5**:

$$v_2 = \sqrt{2.(32.2\frac{lbm - ft}{lbf - s^2}).(778\frac{ft - lbf}{BTU}).(1299 - 1191\frac{BTU}{lbm})}$$

$$v_2 = 2324\, ft/s$$

Isentropic Flow

In gas dynamics, flow of gas is said to be ***isentropic*** when the process is adiabatic, frictionless, reversible, and when the ***change in entropy is negligible***. In many practical, high velocity gas flow scenarios, there is a small entropy change due to the nozzle and discharge loss coefficients.

Critical Point

A gas in flow is said to be at the critical point when its speed equals the speed of sound, i.e. Mach 1, or M=1, or 1130 ft/s (344 m/s) at 70°F/20°C, 1 Atm, or 1090 ft/s (331 m/s) at STP. At the critical point, parameters such as velocity, density, temperature, pressure, etc., are called sonic properties and are annotated by an asterisk, *; for instance; v*, ρ*, T*, and P*, respectively. The ratios of sonic properties to reservoir properties are referred to as ***critical constants*** or ***critical ratios***. For instance, the critical pressure ratio $\mathbf{R_{cp}}$ is represented mathematically as:

$$R_{cp} = \left[\frac{P^*}{P_o}\right] = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \qquad \textbf{Eq. 1.6}$$

Where,

P* = Sonic pressure
$\mathbf{P_o}$ = Reservoir pressure
k = Ratio of specific heats; e.g., **k = 1.4** for air
$\mathbf{R_{cp}}$ **=** Critical pressure ratio

Shock Waves

Shock waves are thin layers of gas and several molecules in thickness that have substantially different thermodynamic properties. Shock waves develop when a gas moving at supersonic speed slows to subsonic speed. Shock waves propagate or travel ***normal to the direction of flow of gas***. Shock waves represent an ***adiabatic*** process and the total temperature of the system stays constant. However, the total pressure does decrease and the process is ***not isentropic.***

Self Assessment Problems and Questions – Segment 1

1. A nozzle is fed from a superheated steam reservoir. The superheated steam in the reservoir is at 500°C (932°F) and 2.0 MPa (290 psia). The duct or hose connecting the nozzle to the reservoir is short and the frictional head loss in the hose is negligible. Based on these practical assumptions, the velocity of the superheated steam in the hose can be neglected. The steam exits the nozzle at 1.0 bar (14.5 psia) and 95% quality. Determine the exit velocity of the steam at the nozzle in SI units.

2. Solve Problem 1 in US Units, use Mollier diagram for all enthalpy identification and compare the resulting steam speed with the results from the computation conducted in SI units in Problem 1.

3. The SFEE Equation 1.2 can be applied to compute the exit speed of gas in high speed gas applications under which of the following conditions?

A. When data is available in US units
B. When data is available in SI units
C. When the reservoir is large enough such that $\mathbf{v_0 = 0}$, applies
D. Both B and C
E. Both A and B

4. Which of the following statements is true about shock waves?

A. Shock waves require superheated steam.
B. Shock waves travel parallel to the direction of the flow of gas.
C. Shock waves travel perpendicular to the direction of the flow of gas.
D. Both A and B

Segment 2

Psychrometry and Psychrometric Analysis

Topics:

- Psychrometry and psychrometric chart
- HVAC analysis.

Introduction

The scope of this segment is to introduce engineers to psychrometry and to provide an understanding of psychrometric concepts, principles, tools and techniques available for analyzing existing and projected psychrometric conditions in an air conditioned environment. Psychrometry, like many other aspects of thermodynamics, deals with the basic elements of thermodynamics such as air, moisture, and heat. Our discussion in this Segment will focus heavily on the use of the psychrometric chart as an important tool for evaluating current psychrometric conditions, defining transitional thermodynamic processes and projecting the post transition psychrometric conditions.

The Psychrometric Chart:

A psychrometric chart is a graph of the physical properties of moist air at a constant pressure (often equated to an elevation relative to sea level). This chart graphically expresses how various physical and thermodynamic properties of moist air relate to each other, and is thus a graphical equation of state. See Figures 10.1, 10.2 and 10.3. Psychrometric charts are available in multiple versions. Some versions are basic and allow analysis involving only the basic parameters, such as the dry bulb, wet bulb, enthalpy, relative humidity, humidity ratio and the dew point. The detailed version psychrometric charts include additional parameters, like the specific volume, sensible heat ratio and higher resolution relative humidity scale for RH level below 10%. Psychrometric charts are available in the US or imperial units as well as the SI or metric units. Psychrometric charts are published by various sources including the major refrigerant and refrigeration systems manufacturers like Dupont, York, Carrier and Trane. Moreover, several tools are available online for psychrometric analysis.

The versatility of the psychrometric chart lies in the fact that by knowing three independent properties of moist air (one of which is the pressure), other unknown properties can be determined.

The thermophysical properties and parameters found on most psychrometric charts are as follows:

Dry-bulb Temperature (DB): Dry bulb temperature of an air sample is the temperature measured by an ordinary thermometer when the thermometer's bulb is dry; hence the term “Dry-bulb.” Dry bulb temperature can also be measured using electronic or electrical instruments such as Resistance Temperature Devices (RTD) and thermocouples. When RTD’s or thermocouples are employed for dry bulb measurement, the temperature sensing tips or junctions of these devices are simply exposed to ambient air. The units for dry bulb temperature are °F (US/Imperial domain) or °C (SI/Metric domain).

Wet-bulb Temperature (WB): Wet bulb temperature is the temperature read by a thermometer whose sensing bulb is covered with a wet sock evaporating into a rapid stream of the sample air. When the air is saturated with water, the wet bulb temperature is the same as the dry bulb temperature and the psychrometric point lies directly on the saturation line. Similar to the dry bulb temperature, the units for dry bulb temperature are °F (US/Imperial domain) or °C (SI/Metric domain).

Dew-Point Temperature (DP): Dew point is the temperature at which water vapor begins to condense into liquid. The dew point temperature serves as an adjunct to and supports other psychrometric properties of moist air, such as the wet bulb and the relative humidity. Similar to the dry bulb and wet bulb temperatures, the units for dew point are °F (US/Imperial domain) or °C (SI/Metric domain).

Relative Humidity (RH): Relative humidity of a sample of moist air (air that holds some measurable quantity of water vapor) is the ratio of the mole fraction of water vapor to the mole fraction of saturated moist air at the same temperature and pressure. Relative humidity is dimensionless and is usually expressed as a percentage.

Humidity Ratio: Humidity ratio is the proportion of the mass of water vapor per unit mass of dry air under a given set of dry bulb, wet bulb, dew point and relative humidity conditions. Humidity ratio is denoted by the symbol "**ω.**" Humidity ratio is dimensionless. However, it is typically expressed in grams of water per gram of dry air (in SI units) or grains of moisture per pound of dry air (in US units).

Specific Enthalpy: Specific enthalpy of a substance is defined as the heat content of the substance per unit mass. In psychrometry, enthalpy represents the heat content of moist air. Enthalpy is measured in kilo Joules per kilogram of dry air (in SI units) or BTUs per pound (in US Units) of dry air. In the SI or metric unit realm, specific enthalpy is sometimes stated in Joules/gram. Of course, enthalpy amounting to 1 kJ/kg of dry air is equivalent to an enthalpy of 1 J/gm of dry air. Specific enthalpy, as alluded to earlier in this text, is denoted by the symbol "**h.**"

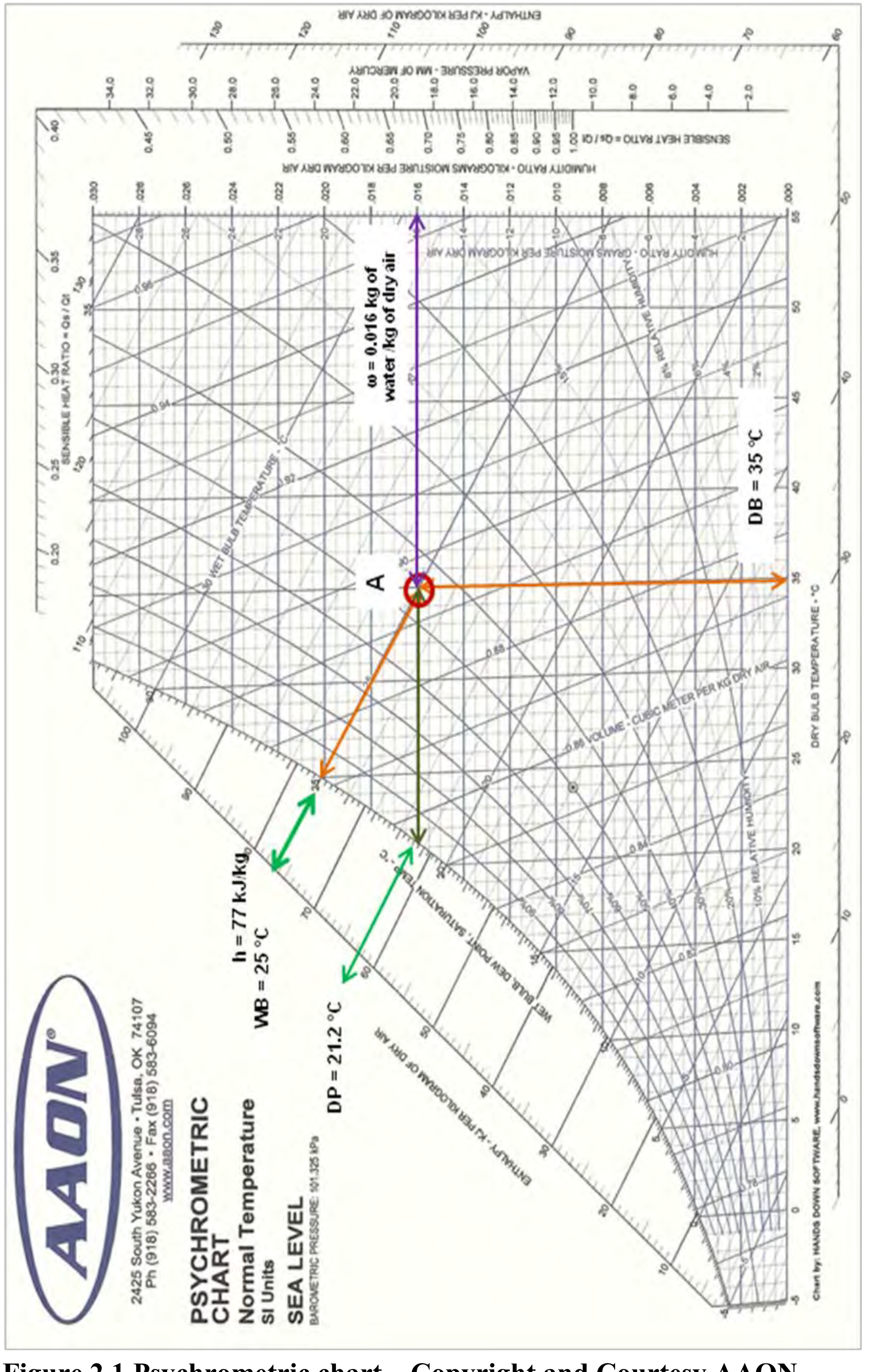

Figure 2.1 Psychrometric chart – Copyright and Courtesy AAON

Method for Reading the Psychrometric Chart:
The psychrometric chart reading guide shown in Figure 2.2 illustrates the general method for reading various psychrometric parameters on a typical psychrometric chart. Navigation of some of the basic psychrometric parameters, utilizing the guide in Figure 2.2 and a simple psychrometric chart shown in Figure 2.1, is outlined below:

- **Dry Bulb:** On the psychrometric chart, the dry bulb temperature scale appears horizontally, along the x-axis, See Figures 10.1 and 10.2. As apparent in these two diagrams, the dry bulb temperature increments from left to right. The scale for dry bulb temperature is graduated in °F (US/Imperial domain) or °C (SI/Metric domain).

- **Wet Bulb**: The wet bulb lines are inclined with respect to the horizontal. In other words, the wet bulb lines emanate diagonally from the psychrometric point and intersect with the saturation curve on the left as they run parallel to the enthalpy lines. The wet bulb temperature shares its scale on the saturation line with the dew point. See Figures 10.1 and 10.2. Like the dry bulb temperature, the scale for the wet bulb temperature is graduated in °F (US/Imperial domain) or °C (SI/Metric domain).

- **Dew Point:** To read the dew point, follow the horizontal line from the psychrometric point to the 100% RH line. The100% RH line is also known as the ***saturation curve***. Note that the psychrometric point is a point on the psychrometric chart where wet bulb (slanted) and dry bulb (vertical) lines meet, or where dry bulb line (vertical) and the line/curve representing a given %RH intersect. The dew point is located where the horizontal dew point line intersects with the 100% relative humidity line on the left. The dew point temperature shares its scale on the saturation line with the wet bulb temperature. Like the dry bulb and wet bulb temperatures, the scale for the dew point temperature is graduated in °F (US/Imperial domain) or °C (SI/Metric domain).

- **Relative Humidity Line**: Relative humidity is depicted in the form of positively sloped curves, or lines, spanning from the bottom left corner of the psychrometric chart to the top right portion of the chart. These relative humidity lines are half parabolic asymptotic lines that are drawn to the right of the saturation curve. The relative humidity lines are typically graduated in 10% increments on most conventional psychrometric charts; ranging from 10% to 100%. The relative humidity scale is graduated in finer 2% increments below the 10% RH level. See Figure 10.1.

- **Humidity Ratio - ω**: Humidity Ratio is read off the graduated vertical line, on the right side of the psychrometric chart, representing the humidity ratio scale. See Figures 10.1 and 10.2. The horizontal humidity ratio lines span from the saturation line side of the psychrometric chart to the extreme right side, intersecting on the right with the vertical humidity ratio scale. The humidity ratio scale ranges from 0.000 to 0.030; defined in kg of moisture per kg of dry air on the metric (SI) psychrometric charts or in pounds (lbm) of moisture per pound (lbm) or dry air on the US unit psychrometric charts.

- **Specific Enthalpy - h**: As shown in Figures 10.1 and 10.2, the specific enthalpy lines run parallel to the wet bulb lines on the psychrometric charts. In other words, the enthalpy lines emerge diagonally from the psychrometric point and intersect with the saturation curve on the left. In the commonly used segment of the psychrometric chart, the specific enthalpy scale ranges from 10 to 55 BTU/lbm of dry air in the US unit realm, and 10 to 110 kJ/kg in the metric (SI) unit realm.

- **Specific volume - υ**: Specific volume lines appear on the psychrometric chart as equally spaced parallel lines representing specific volume ranging from 0.5 to 0.96 m^3/kg of dry air¸ in increments of 0.01m^3/kg of dry air, in the SI (metric) unit system. These lines span diagonally from the bottom left corner of the psychrometric chart to the top right corner. See Figure 2.1. On the US or imperial system psychrometric charts, the specific volume lines range from 13.0 to 15.0 ft^3/lbm of dry, in 1.0 ft^3/lbm of dry air increments. Specific volumes for psychrometric points that do not lie on the designated specific volume lines must be derived through interpolation, as illustrated in Case Study 2.2.

Example 2.1

A basic illustration of the method employed for reading and analyzing psychrometric charts can be seen in Figure 2.1, where, psychrometric point A is identified on the basis of the following two parameters:

a) The given dry bulb temperature of 35°C.
b) The given wet bulb temperature of 25°C.

Once point A is located on the psychrometric chart, the following additional psychrometric properties and attributes are read off the chart:

I. The dew point is read horizontally off to the left on the wet bulb and dew point scale as 21.2°C.
II. The enthalpy is read diagonally to the left, parallel to the wet bulb lines, off the enthalpy scale as 77 kJ/kg.
III. The humidity ratio, ω, is read off the humidity ratio scale on the right as 0.016 kg of moisture per kg of dry air.

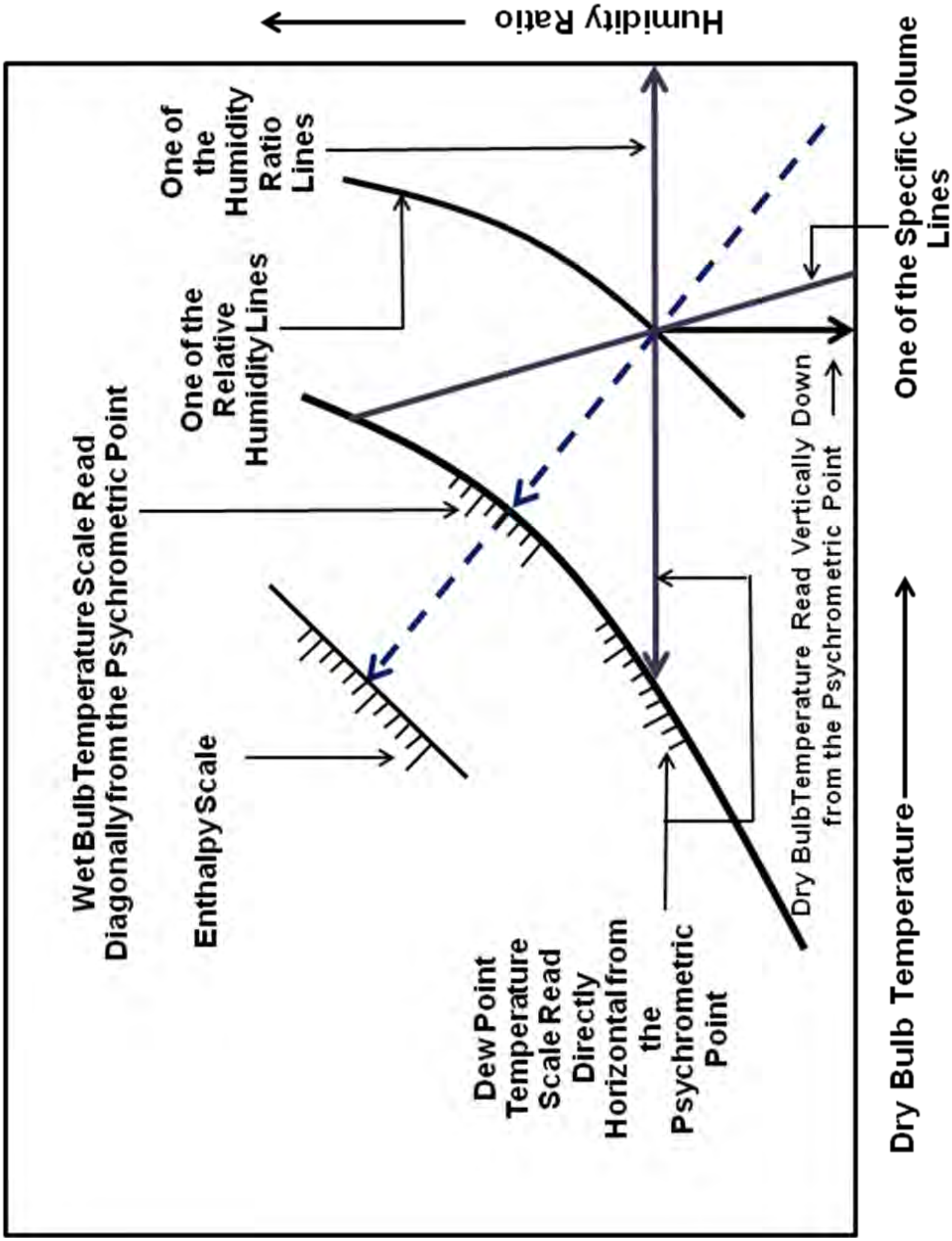

Figure 2.2 Psychrometric Chart Reading Guide

Psychrometric Transition Process

The psychrometric Transition Process is the process involving changes in dry bulb, wet bulb, dew point, relative humidity, humidity ratio, and enthalpy, whereby, a psychrometric point representing a set of psychrometric conditions moves from one point on the psychrometric chart to another.

These psychrometric processes are illustrated in Figure 2.3. The paths representing eight of these processes and the psychrometric significance of each are as follows:

- **Path O-A:** Path O-A, in its absolute vertical configuration as depicted in Figure 2.3, represents a psychrometric transition or process involving an increase in relative humidity or humidity ratio only. Since this path is vertical, the dry bulb stays constant.

- **Path O-B:** Since path O-B is at an upward diagonal attitude, relocation of psychrometric points along this path would involve an increase in dry bulb, enthalpy and humidity.

- **Path O-C:** Path O-B, in its absolute horizontal direction to the right, as depicted in Figure 2.3, represents a psychrometric transition or process involving an increase in sensible heat only. Since this path is horizontal, the dry bulb changes while the humidity ratio stays constant.

- **Path O-D:** Path O-D, with its diagonally downward direction, represents dehumidification with some decline in the dry bulb temperature. This psychrometric process path can be implemented through chemical dehumidification systems, which are ideally suited for ice skating rinks where dehumidification is desired without an increase in the dry bulb temperature.

- **Path O-E:** Path O-E, in its direct downward vertical configuration, as depicted in Figure 2.3, represents a reduction in relative humidity or humidity ratio with no change in the dry bulb temperature.

- **Path O-F:** As obvious from the diagram in Figure 2.3, path O-F with its diagonally downward attitude to the left represents a simultaneous cooling and dehumidification process. This path is ideal for situations where lower dry bulb temperatures and lower dew points are desired.

- Path O-G: While relocation of a psychrometric point from "O" directly to the left, as depicted in Figure 2.3, would result in some increase in the relative humidity, the predominant impact is in the form reduction of sensible heat and the dry bulb temperature.

- Path O-H: This path is a classic representation of the evaporative cooling process where, typically, the dry bulb temperature is reduced through forced air evaporation of water. However, while the latent evaporative process extracts heat from the system, thus lowering the dry bulb temperature, the evaporated moisture increases the RH level and the humidity ratio.

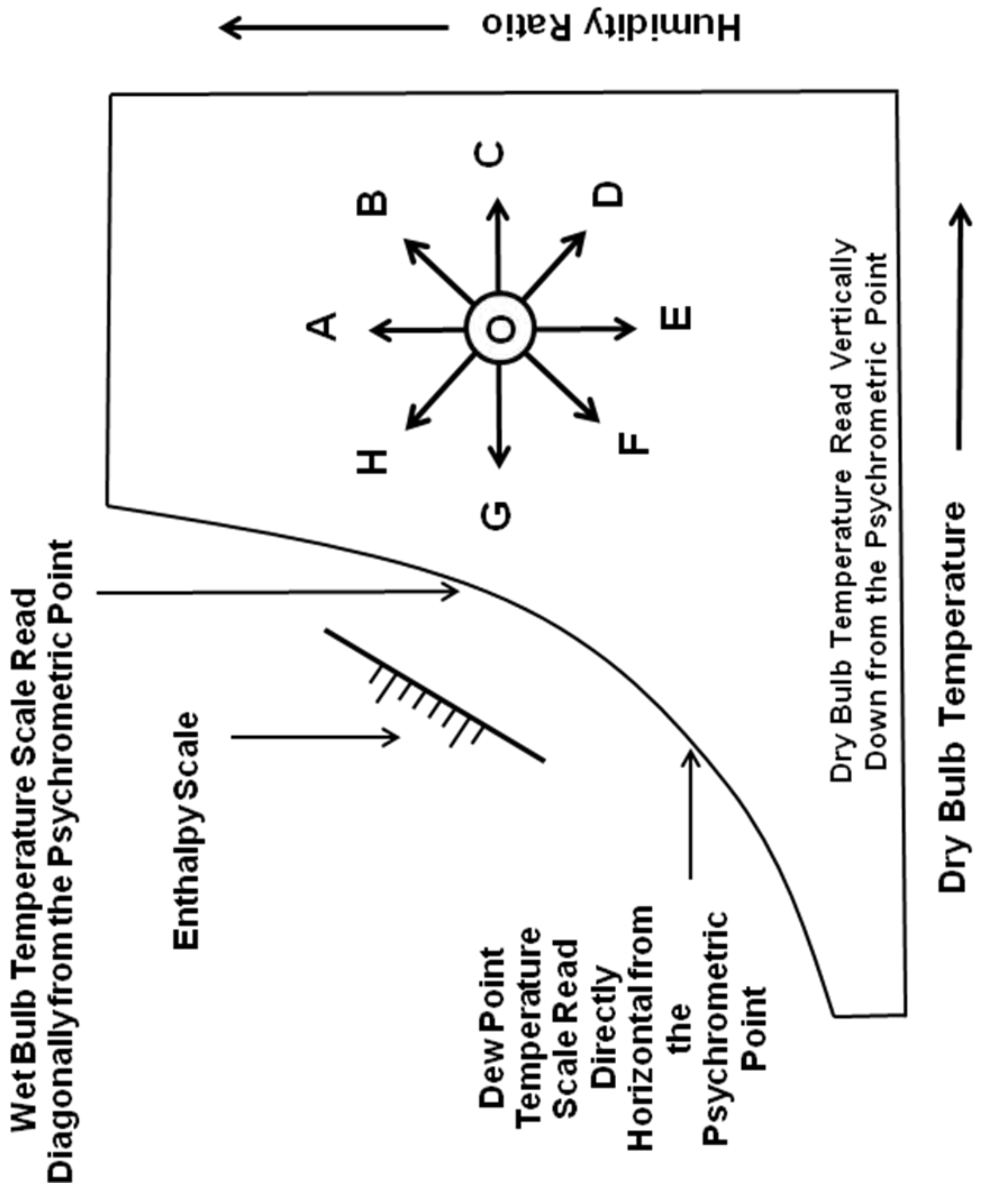

Figure 2.3 Psychrometric Processes

When performing psychrometric analysis pertaining to a scenario that entails the transition from an initial set of psychrometric conditions to a final set of psychrometric conditions, a suitable approach is to begin with the identification or location of the initial and final psychrometric points on the psychrometric chart.

The location or identification of the initial and final psychrometric points on the psychrometric chart requires the knowledge, or field measurement, of at least two of the following important parameters associated with each point:

I. The dry bulb temperature
II. The wet bulb temperature
III. The % relative humidity
IV. The dew point
V. The enthalpy
VI. The humidity ratio

Among the six parameters listed above, the more conventional, measurable and more "likely to be known" parameters are the dry bulb temperature, the wet bulb temperature, the % relative humidity, and the dew point. The enthalpy and the humidity ratio are listed merely as theoretical possibilities.

Once the initial and final psychrometric points have been located on the psychrometric chart, other unknown parameters associated with these two points can be read off the psychrometric chart. Interpolation between known or graphed lines and points may be necessary in certain cases to locate the psychrometric points in question.

Also, once the initial and final psychrometric points have been located on the psychrometric chart, advanced analyses such as the determination of SHR, Sensible Heat Ratio, mass of water removed, amount heat involved, specific volume of the air, etc. can be determined through graphical or geometric analyses performed on the psychrometric chart.

Case Study 2.1: Psychrometrics – SI Unit System

In an environment that is estimated to contain, approximately 450 kg of air, the dry-bulb is measured to be 35 °C and the wet-bulb is at 25 °C. Later, the air is cooled to 13 °C and, in the process of lowering the dry-bulb temperature, the relative humidity drops to 75%. As an Energy Engineer, you are to perform the following psychrometric analysis on this HVAC system:

a) Find the initial humidity ratio, ω_i.
b) Find the final humidity ratio, ω_f.
c) Find the total amount of heat removed.
d) Find the amount of sensible heat removed.
e) Find the amount of latent heat removed.
f) Find the final wet-bulb temperature.
g) Find the initial dew point.
h) Find the final dew point.
i) Find the amount of moisture condensed/removed.
j) Can the amount of electrical power consumed by the A/C System be determined on the basis of the data provided in this case study?

Solution:

A general approach to solving this psychrometric case study problem and other similar ones is premised on the psychrometric transition process paths illustrated in Figure 2.2 and the psychrometric chart interpretation guide shown in Figure 2.2.

As explained in the discussion leading to this case study, we need to begin the analyses associated with this case with the identification or location of the initial and final psychrometric points on the psychrometric chart.

Location of the initial psychrometric point can be established using the following two parameters associated with this point:

- Dry-bulb temperature of 35 °C
- Wet-bulb temperature of 25 °C

This point is shown on the psychrometric chart in Figure 2.4 as point A.

The location of the final psychrometric point can be established using the following two pieces of data associated with this point:

- Dry bulb temperature of 13 °C
- Relative humidity of 75%.

Relative humidity line, representing an RH of 75% is placed through interpolation between the given 70% and 80% RH lines on the psychrometric chart. The final point, thus identified, is shown as point B on the psychrometric chart in Figure 2.4.

a) Find the initial humidity ratio, ω_i.

To determine the initial humidity ratio, draw a horizontal line from the initial point to the vertical humidity ratio scale on the psychrometric chart as shown in Figure 2.4.

The point of intersection of this horizontal line and the humidity ratio scale represents the humidity ratio for the initial psychrometric point, ω_i.

As read from the psychrometric chart in Figure 2.4:

$$\boldsymbol{\omega_i = 0.016} \textbf{ kg of moisture per kg of dry air}$$

b) Find the final humidity ratio, ω_f.

Similar to part (a), the humidity ratio for the final psychrometric point can be determined by drawing a horizontal line from the final point to the vertical humidity ratio scale on the psychrometric chart in Figure 2.4.

The point of intersection of this horizontal line and the humidity ratio scale represents the humidity ratio, $\boldsymbol{\omega_f}$, for the final psychrometric point.

As read from the psychrometric chart in Figure 2.4:

$\boldsymbol{\omega_f = 0.007}$ kg of moisture per kg of dry air

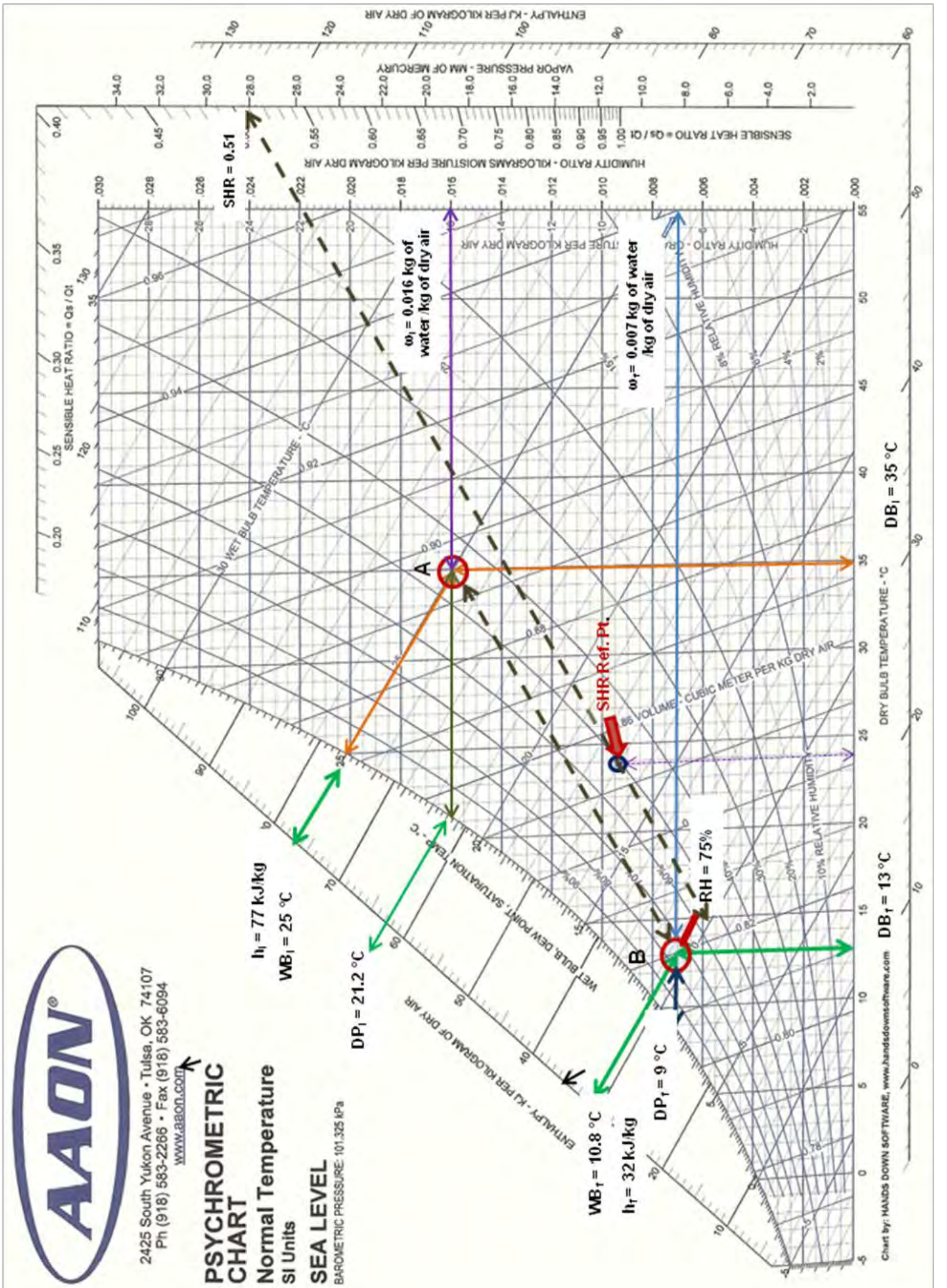

Figure 2.4 Psychrometric Chart – Case Study 2.1, SI Unit System

c) Find the total amount of heat removed.

The first step is to identify the enthalpies on the psychrometric chart at the initial and final points. See Figure 2.4.

At the initial point, the dry-bulb temperature is 35 °C, the wet-bulb is 25 °C, and as shown on the psychrometric chart, h_i = 77 kJ/kg of dry air.

At the final point, dry-bulb is 13 °C, with RH at 75%. The enthalpy at this point, h_f = 32 kJ/kg of dry air. Therfore:

$$\Delta h = h_f - h_i$$
$$= 32 \text{ kJ/kg} - 77 \text{ kJ/kg}$$
$$= -45 \text{ kJ/kg of dry air.}$$

and,

$$\Delta Q = (\Delta h) \,.\, m_{air} \qquad \textbf{Eq. 2.1}$$
$$= (-45 \text{ kJ/kg}) \,.\, m_{air}$$

where, the mass of dry air, m_{air}, needs to be derived through the given combined mass of moisture and air, 450 kg, and the humidity ratio, ω.

Humidity ratio is defined as:

$$\omega = \text{mass of moisture (kg) / mass of dry air (kg)}$$

and as determined from the psychrometric chart earlier in part (a):

ω = 0.016 kg of moisture per kg of dry air, at the initial point

$$\omega = m_{moisture} / m_{dry\ air}$$

Or,

$$\omega = (m_{moist\ air} - m_{dry\ air}) / m_{dry\ air}$$

Through algebraic rearrangement of this equation, we get:

$$(1 + \omega) = m_{moist\ air} / m_{air}$$

Or,

$$m_{air} = \text{mass of dry air} = m_{moist\ air} / (1 + \omega)$$

where the total combined of mass of the moisture and the dry air is given as 450 kg. Therefore:

$$m_{air} = 450 \text{ kg}/ (1 + 0.016)$$
$$= 443 \text{ kg}$$

Then, by applying Eq. 2.1:

$$\mathbf{\Delta Q = Total\ Heat\ Removed}$$
$$= \mathbf{(\Delta h)\ .\ m_{air}}$$
$$= (- 45 \text{ kJ/kg}) . (443 \text{ kg})$$

Or,

$$\Delta Q = \text{Total Heat Removed} = \mathbf{- 19{,}935\ kJ}$$

The negative sign in the answer above signifies that the heat is extracted from, or that it exits, the system as the air conditioning process transitions from the initial psychrometric point to the final psychrometric point.

d) Find the amount of sensible heat removed.

The first step in determining the amount of sensible heat removed is to identify the Sensible Heat Ratio (SHR) from the psychrometric chart. This process involves drawing a straight line between the initial and final points. This line is shown as a dashed line between the initial and final points. Then draw a line parallel to this dashed line such that it intersects with the SHR Reference Point and the vertical scale representing the SHR. The point of intersection reads approximately 0.51. **See Figure 2.4.**

Note: For additional discussion on the significance of SHR, refer to Case Study 2.2, part (g) and self assessment problem 10.2 (g).

The Sensible Heat Ratio is defined mathematicall as:

$$\mathbf{SHR = Sensible\ Heat\ /\ Total\ Heat}$$

In this case,

$$\mathbf{SHR = Q_s / Q_t = 0.51}$$

Or,

Q_s = Sensible Heat = (0.51) . (Q_t)

and, since $Q_t = \Delta Q$ = Total Heat Removed = - 19,935 kJ, as calculated earlier:

Q_s = Sensible Heat = (0.51). (-19,935 kJ)
= -10,167 kJ

e) Find the amount of latent heat removed.

The total heat removed consists of sensible and latent heat components.

Or,

$$Q_t = Q_s + Q_l$$

Q_l = Latent Heat = Q_t - Q_s
= - 19,935 kJ – (- 10,166 kJ)

Q_l = Latent Heat = - 9,768 kJ

f) Find the final wet-bulb temperature.

As explained in the discussion associated with the psychrometric chart interpretation guide in Figure 2.2, wet bulb is read diagonally from the psychrometric point along the wet bulb temperature scale. The wet bulb lines run parallel to the enthalpy lines on the psychrometric chart.

The diagonal line emerging from the final point intersects the wet bulb and dew point scale at approximately 10.8 °C as shown on the psychrometric chart in Figure 2.4. Therefore, the wet bulb temperature at the final point is **10.8 °C.**

g) Find the initial dew point.
To determine the initial dew point, read the dew point temperature for the initial point on the psychrometric chart in Figure 2. 4 using the psychrometric chart interpretation guide in Figure 2.3.

Follow the horizontal "dew point" line drawn from the initial point to the left, toward the saturation curve. The point of intersection of the saturation line and the dew point line represents the dew point. This point lies at 21.2°C.

Therefore, the dew point at the initial point, as read off from Figure 2.4, is **21.2°C.**

h) Find the final dew point.
To determine the final dew point, follow the horizontal "dew point" line drawn from the final point to the left, toward the saturation curve. The point of intersection of the saturation line and the dew point line represents the dew point. This point lies at 9 °C.

Therefore, the dew point at the initial point, as read off from Figure 2.4, is **9 °C.**

i) Find the amount of moisture condensed/removed.
In order to calculate the amount of moisture condensed or removed, we need to find the difference between the humidity ratios for the initial and final points.

Humidity Ratio, ω = mass of moisture (kg) / mass of dry air (kg)

From the psychrometric chart, in Figure 2.4:

$\boldsymbol{\omega_i}$ = **0.016** kg of moisture per kg of dry air, at the initial point
$\boldsymbol{\omega_f}$ = **0.007** kg of moisture per kg of dry air, at the final point

Δω = Change in the Humidity Ratio
= **0.016 - 0.007**
= **0.009** kg of moisture per kg of dry air

The amount of moisture condensed or removed
= (Δω) . (Total mass of Dry Air)
= (Δω) . ($m_{dry\ air}$)

Where, $m_{dry\ air}$ = 443 kg of dry air, as calculated in part (a)

The amount of moisture condensed or removed

= (0.009 kg of moist./kg of dry air) . (443 kg of dry air)

= 3.987 kg

j) Can the amount of electrical power consumed by the A/C System be determined on the basis of the data provided in this case study?

Calculation of the electrical power consumed by the A/C System would require data pertaining to the brake horsepower demanded by the A/C compressor.

The brake horse power can be calculated from the efficiency of the pump, differential pressure, head added and the volumetric flow rate of the refrigerant system. However, since none of these parameters are known, determination of the electrical power consumed is **not feasible** due to insufficient data.

Case Study 2.2: Psychrometrics – US Unit System

As an Energy Engineer, you have been assigned to perform psychrometric analysis on an air conditioned environment. The results of measurements performed are as follows:

Estimated mass of *dry air*: **900 lbm**
Initial Dry Bulb Temperature: **81 °F**
Initial Wet Bulb Temperature: **70.4 °F**
The air is cooled to a final temperature of: **75 °F**
Final Dew Point: **48°F**

a) What is the RH, Relative Humidity, at the initial point?
b) What is the RH, Relative Humidity, at the final point?
c) What is the initial Dew Point?
d) What is the final point Wet Bulb?
e) Find the initial Humidity Ratio, ω_i.
f) Determine the SHR for the change in conditions from the initial to the final point.
g) What is the significance of the low SHR in this scenario as compared to the scenario analyzed in Case Study 2.1?
h) What is the estimated specific volume at the initial point?

i) Estimate the total volume of the air in the system.

Solution:
As explained in Case Study 2.1, we need to begin our analyses of this case study with the identification or location of the initial and final psychrometric points on the psychrometric chart.

Location of the initial psychrometric point can be established using the following two parameters associated with this point:

- Dry-bulb temperature of **81 °F**
- Wet-bulb temperature of **70.4 °F**

This point is shown on the psychrometric chart in Figure 2.5 as point **A**.

Location of the final psychrometric point can be established using the following two pieces of data:

- Dry bulb temperature of **75 °F**
- Final Dew Point: **48°F**

The final point, thus identified, is shown as point **B** on the psychrometric chart in Figure 2.5.

a) Relative Humidity at the initial point:
Locate the initial point, as described above, on the psychrometric chart shown in Figure 2.5. As evident from the psychrometric chart in Figure 2.5, the 60% RH line passes directly through the initial point. Therefore, the RH at the initial point is **60%.**

b) Relative Humidity at the final point:

Similar to the approach used in part (a), locate the final point, as described above, on the psychrometric chart shown in Figure 2.5. Use the psychrometric chart interpretation guide from Figure 2.3 for clarification and review as needed. As shown on the psychrometric chart in Figure 2.5, the final point lies directly on the 40% RH line. Therefore, the RH at the final point is **40%.**

c) Initial Dew Point:
To determine the initial dew point, read the dew point temperature for the initial point on the psychrometric chart in Figure 2.5 using the psychrometric chart interpretation guide in Figure 2.3.

Follow the horizontal "dew point" line drawn from the initial point to the left, toward the saturation curve. The point of intersection of the saturation line and the dew point line represents the dew point. This point lies at 66°F.

Therefore, the dew point at the initial point, as read from Figure 2.5, is **66°F.**

d) Final Point Wet Bulb Temperature:
The wet bulb is read diagonally from the psychrometric point along the wet bulb temperature scale. Note that the wet bulb lines are parallel to the enthalpy lines on the psychrometric chart.

The diagonal line emerging from the final point intersects the wet bulb and dew point scale at approximately 59°F. Therefore, the wet bulb temperature at the final point is 59°F.

e) Initial Humidity Ratio, ω_i:
As shown in Figure 2.5, draw a straight horizontal line from the initial point to the right until it intersects with the vertical scale labeled **Humidity Ratio, ω**. This point of intersection with the vertical humidity ratio line lies at ω = 0.0138. Therefore, the humidity ratio at the initial point is **0.0138 lbm of moisture per unit lbm of dry air.**

f) **SHR for the change in conditions from the initial to the final point**.
Draw a straight line between the initial and final points as shown in Figure 2.5. This line is shown as a dashed line spanning between the initial and final points. Then draw a line parallel to this dashed line such that it intersects with the SHR Reference Point and the SHR scale, at the top. The point of intersection reads approximately 0.18. Therefore, the SHR for the change in conditions from the initial to the final point is **0.18.**

g) **What is the significance of the low SHR in this scenario as compared to the scenario analyzed in Case Study 2.1?**

The **SHR of 0.18,** for the scenario portrayed in this problem, is significantly lower than the **SHR of 0.51** for the scenario in Case Study 2.1 because the dry bulb change in this case study is significantly smaller than the dry bulb change in Case Study 2.1. The dry bulb drop in Case Study 2.1 is **22 °C** (or, 95°F – 54°F = 40°F) while the **dry bulb reduction** in this case study is only 6°F, or **3.3°C** (or, 27.2°C -23.9°C). In "°F," the dry bulb reduction in Case Study 2.1 is **40°F,** versus a rather small reduction of only **6°F** in this case study. While the dry bulb change in this case study is relatively minute, the dew point change is substantial; an **18°F drop**, from 66°F to 48°F. In other words, in this case, while the dew point changes significantly, the dry bulb changes negligibly.

When the dry bulb change is small or negligible, the amount of sensible heat involved in the transition process is much smaller than the latent heat. This explains the reason behind SHR being only 0.18 in this case. Note that the SHR of 0.18 implies that only 18% of the total heat involved in this process transition is sensible heat; the remaining 82% of the heat extracted in this process transition is latent heat. This larger portion of extracted latent heat explains the significant 18°F drop in the dew point

h) **What is the estimated specific volume at the initial point?**
Specific volume at the initial point can be estimated through interpolation between the given specific volume lines, in proximity of the initial psychrometric point, on a standard psychrometric chart. These two lines, as shown on the psychrometric chart in Figure 2.5, represent specific volumes of 13 cu-ft/lbm of dry air and 14 cu-ft/lbm of dry air. On the psychrometric chart in Figure 2.5, a diagonal, specific volume line is drawn such that it passes through the initial point and is parallel to the given specific volume lines. The specific volume represented by this initial point specific volume line is interpolated to be approximately **13.9 cu-ft/lbm of dry air.**

i) **Estimate the total volume of the air in the system.**
The total volume of the air in the system can be estimated on the basis of the specific volume determined in part (h) and the mass of dry air given in the problem statement.

Given:

Estimated mass of *dry air*: **900 lbm**

Specific volume: **13.9 cu-ft/lbm of dry air.**

Since Specific Volume = Volume/Mass

$\therefore$ **Volume of the air in the system = (ν) . (Mass)**
= (13.9 cu-ft/lbm of dry air).(900 lbm)
= **12,510 cu-ft**

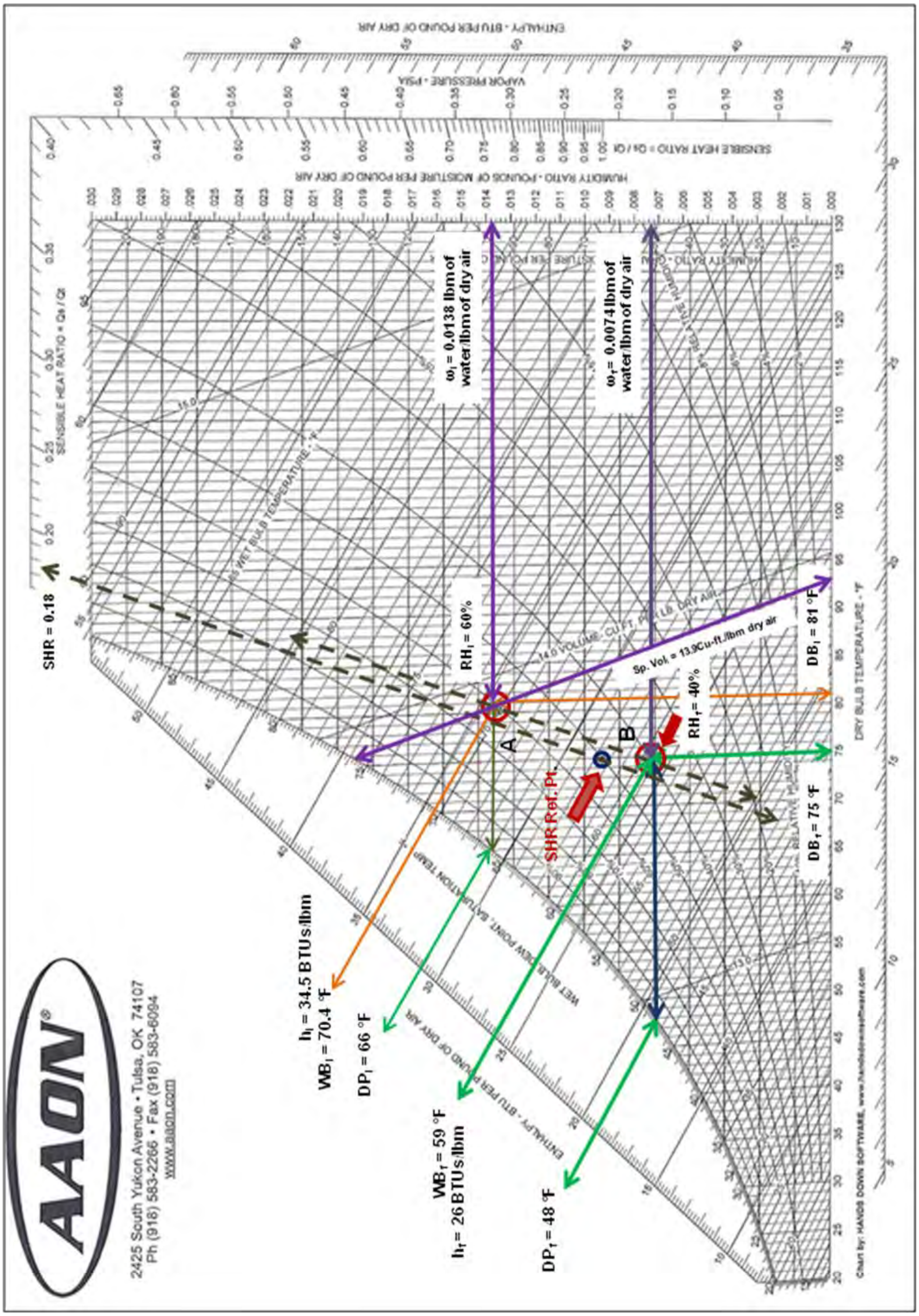

Figure 2.5 Psychrometric Chart – Case Study 2.2

Self Assessment Problems and Questions – Segment 2

1. In an environment that is estimated to contain, approximately, 400 kg of air, the dry-bulb is measured to be 40°C and the wet-bulb is at 27.3°C. Later, the air is cooled to 20°C and, in the process of lowering the dry-bulb temperature, the relative humidity drops to 47%. As an Energy Engineer, you are to perform the following psychrometric analysis on this system using the psychrometric chart in Figure 2.6:

a) Find the initial humidity ratio, ω_i.
b) Find the final humidity ratio, ω_f.
c) Find the total amount of heat removed.
d) Find the amount of sensible heat removed.
e) Find the amount of latent heat removed.
f) Find the final wet-bulb temperature.
g) Find the initial dew point.
h) Find the final dew point.
i) Find the amount of moisture condensed/removed.

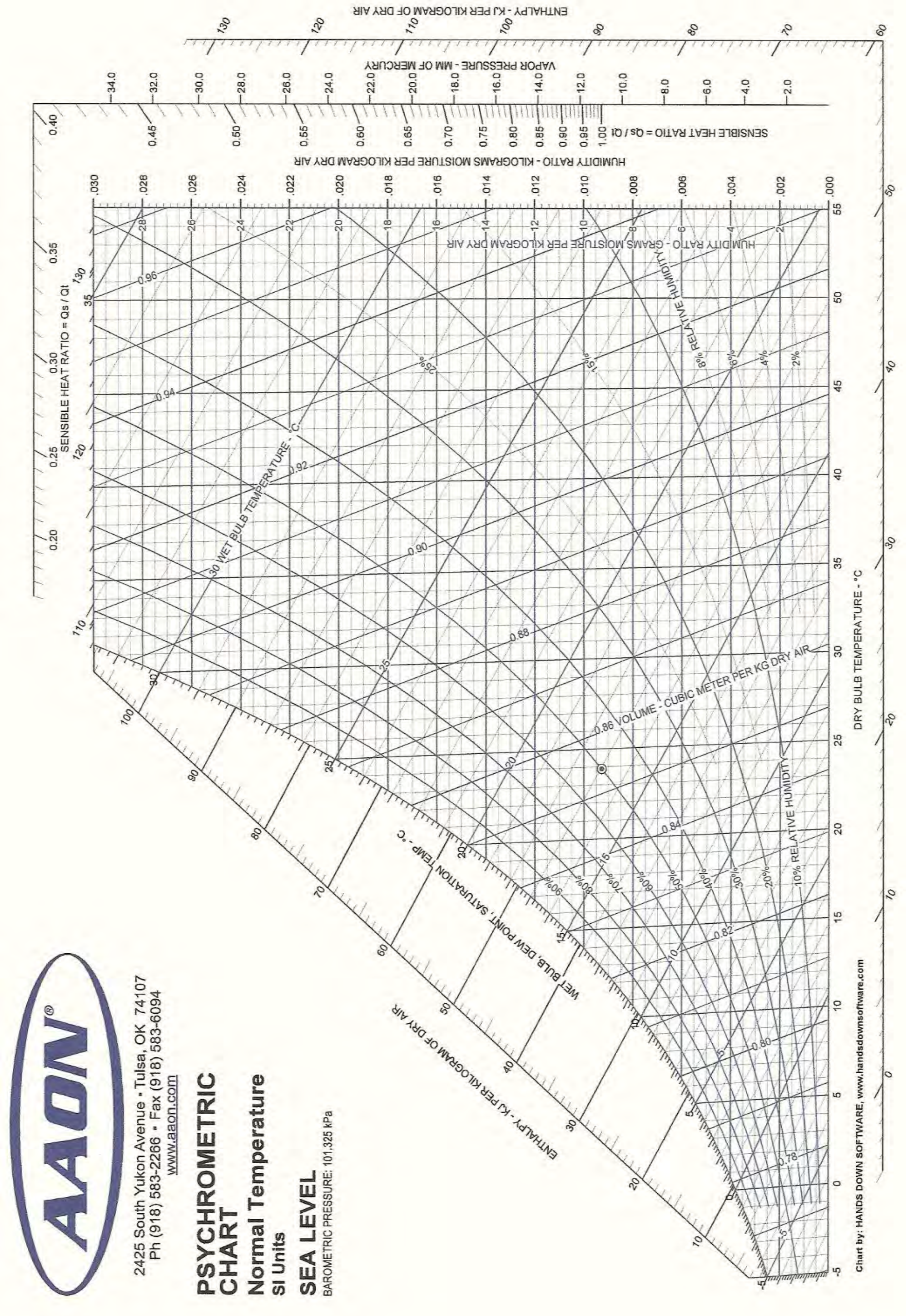

Figure 2.6 Psychrometric Chart – Problem 1

2. The psychrometric chart for the initial and final conditions in a commercial warehouse is shown in Figure 2.7. Dry Bulb and Wet Bulb data associated with the initial and final points is labeled on the chart. Assess the disposition and performance of the HVAC system in this building as follows:

a) What is the initial Dew Point?
b) What is the final Dew Point?
c) Based on the results of dew point determination in parts (a) and (b), define the type of thermodynamic process this system undergoes in the transition from initial point to the final point.
d) What is the RH, Relative Humidity, at the initial point?
e) What is the RH, Relative Humidity, at the final point?
f) Determine the SHR for the change in conditions from the initial to the final point.
g) Comment on why the SHR for this scenario is significantly higher than the scenario analyzed in Case Study 2.2?

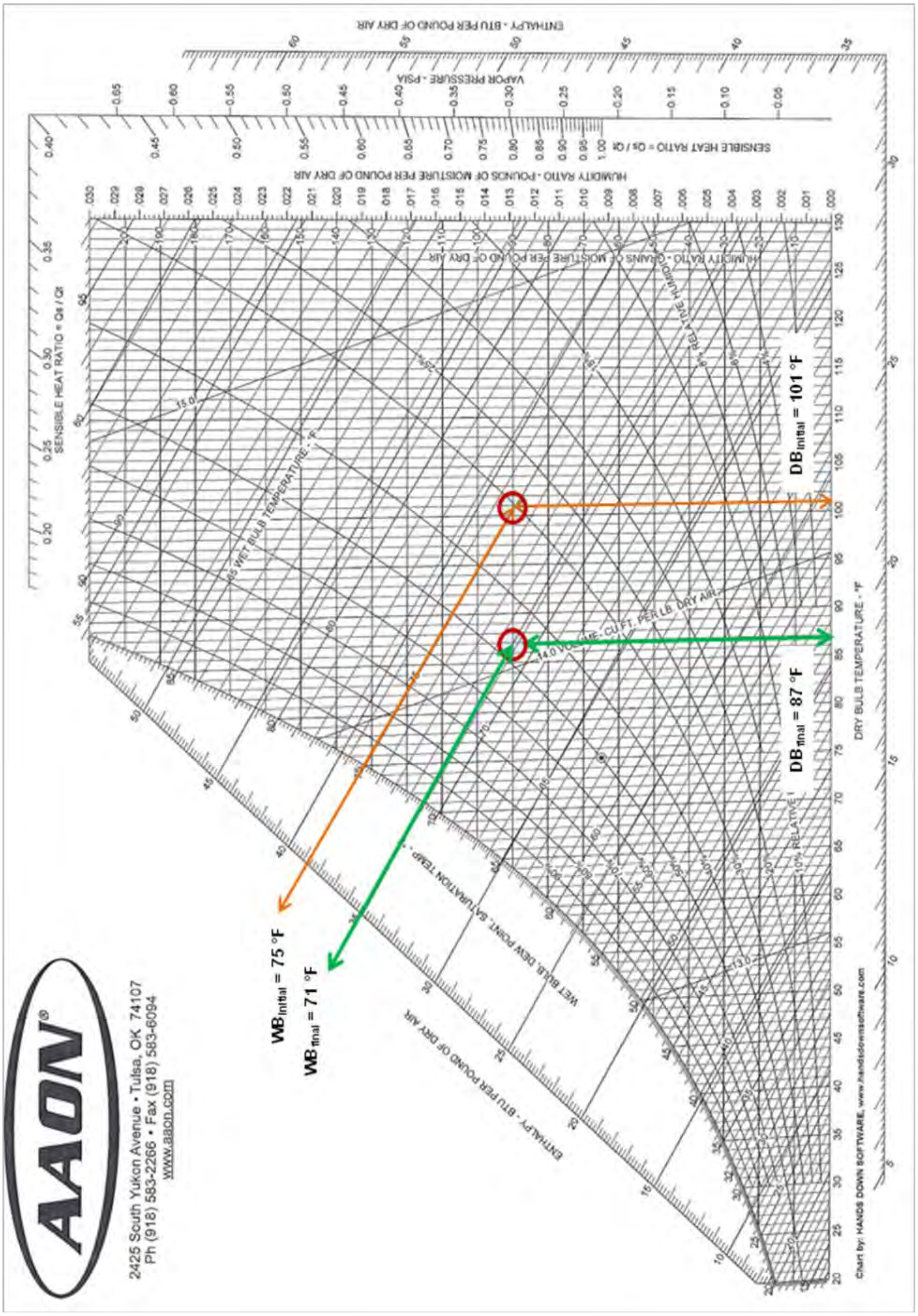

Figure 2.7 Psychrometric Chart – Problem 2

Segment 3

Refrigeration Cycles and HVAC Systems

Topics:

- Basic Refrigeration Cycle
- HVAC and Automated HVAC Systems

Introduction

Study and understanding of the Basic Refrigeration Cycle, HVAC Systems and Automated HVAC Systems is an essential an integral part of thermodynamics. This segment provides the reader an opportunity to learn or review important fundamental concepts, principles, analysis and computational techniques associated with refrigeration and HVAC systems. The study and exploration of refrigeration cycles and HVAC system analysis are illustrated through practical examples, case study and end of segment self assessment problems; formulated with the Energy Engineer's role in mind.

This segment includes definitions and explanations of HVAC terms, concepts and mechanical components not introduced before in this text. Definitions and explanations of several other important HVAC terms and concepts, such as, dry bulb, wet bulb, dew point, enthalpy, specific enthalpy, humidity ratio, Specific Heat Ratio, entropy, saturated liquid, saturated vapor and superheated vapor are covered under Segment 2 and the preceding material.

Types of Air Conditioning Systems

There are several types of air conditioning systems. One could categorize air conditioning systems based on their application and size. The fundamental refrigeration system principles that govern the functionality of a refrigerator versus a typical air conditioning system are the same. Therefore, most of our discussion and engineering analysis in this segment would apply to both of these devices.

Within the air conditioning realm, differences between different types of air conditioning systems are premised on their application and size. In large air conditioning systems, such as those pertaining to industrial and commercial applications, major components of the refrigeration systems are sizeable, somewhat independent, and are located separately. Some of the large industrial and air conditioning systems consist of large single unit chillers. A typical chiller for air conditioning applications is rated between 15 to 1500 tons. This would translate into 180,000 to 18,000,000 BTU/h or 53 to 5,300 kW in cooling capacity. Chilled water temperatures in such systems can range from 35°F to 45°F or 1.5°C to 7°C, depending upon specific application requirements. Figure 3.1 draws a comparison between a large industrial or commercial chiller and a typical refrigerator compressor. The large chiller in the picture is rated over 700 hp, while the small compressor is rated approximately 1 to 3 hp.

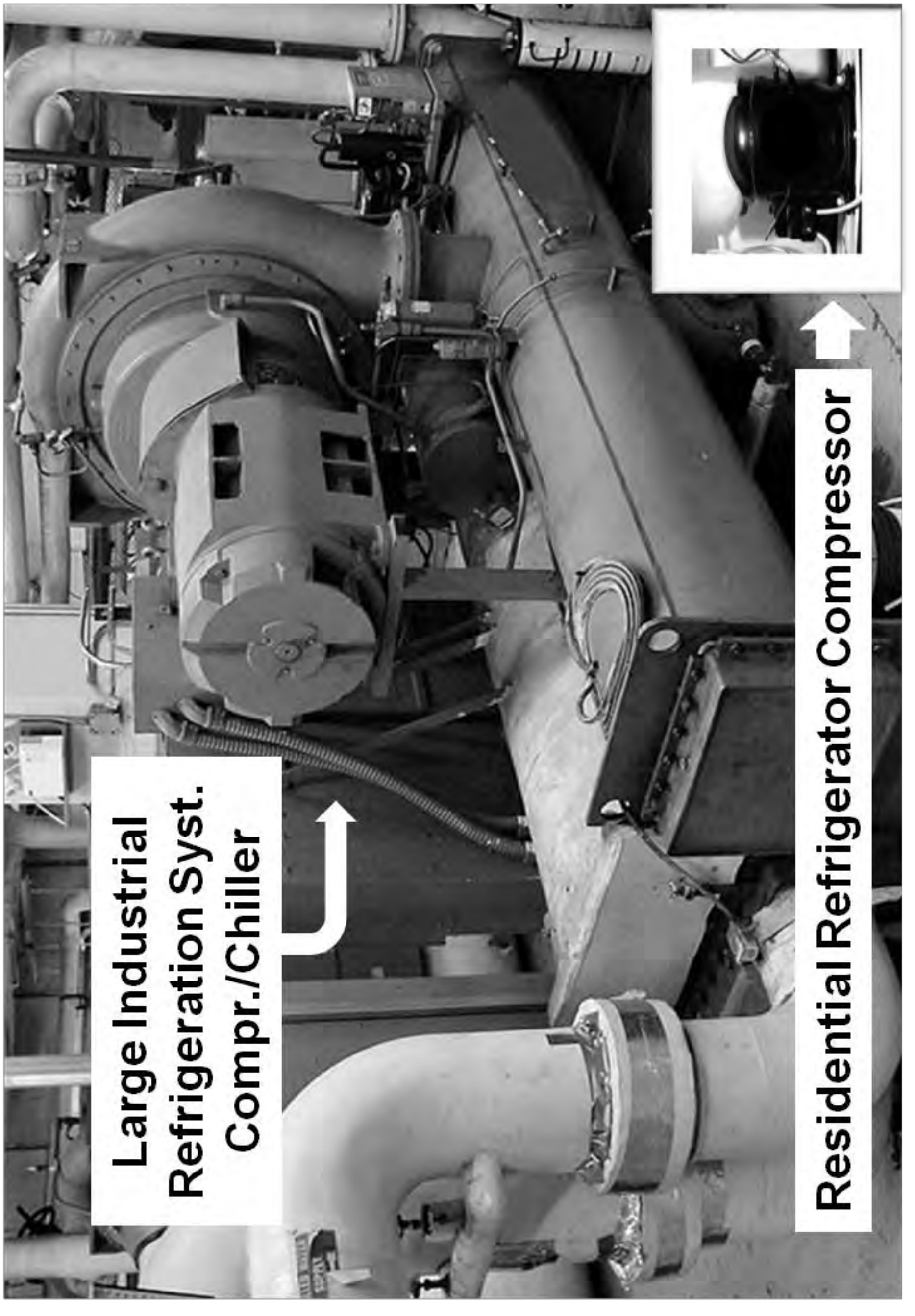

Figure 3.1 Large Refrigeration System Chiller vs. a Refrigerator Compressor

Large chiller based air conditioning systems and process chilled water systems utilize water as a "secondary" working fluid. But, the primary working fluid is still a typical refrigerant, i.e., HFC-134a, ammonia, R-500, etc. Large chiller based air conditioning systems can be categorized as Open Air Conditioning Systems or Closed Air Conditioning Systems. An open air conditioning system utilizes a Freon based refrigerant, in a large chiller, to cool the water to 35°F to 45°F or 1.5°C to 7°C range. This chilled water is then conveyed to Open Air Washers, equipped with chilled water spray nozzles. See Figure 3.2. The return or outside air is passed through a chilled water spray. The high moisture content and higher temperate return or outside air is thus cooled and dehumidified as it passes through the air washer. The supply air exiting the air washer is at lower dry bulb and lower dew point with lower relative humidity. The supply air is then driven by a supply air fan to work spaces, or occupied spaces in general, as conditioned air. A Closed Air Conditioning System, on the contrary, in most cases does not use chilled water as a secondary working fluid to cool and condition the ambient air. Closed air conditioning systems are similar or equivalent to residential air conditioning systems where the Freon or refrigerant is used as the working fluid.

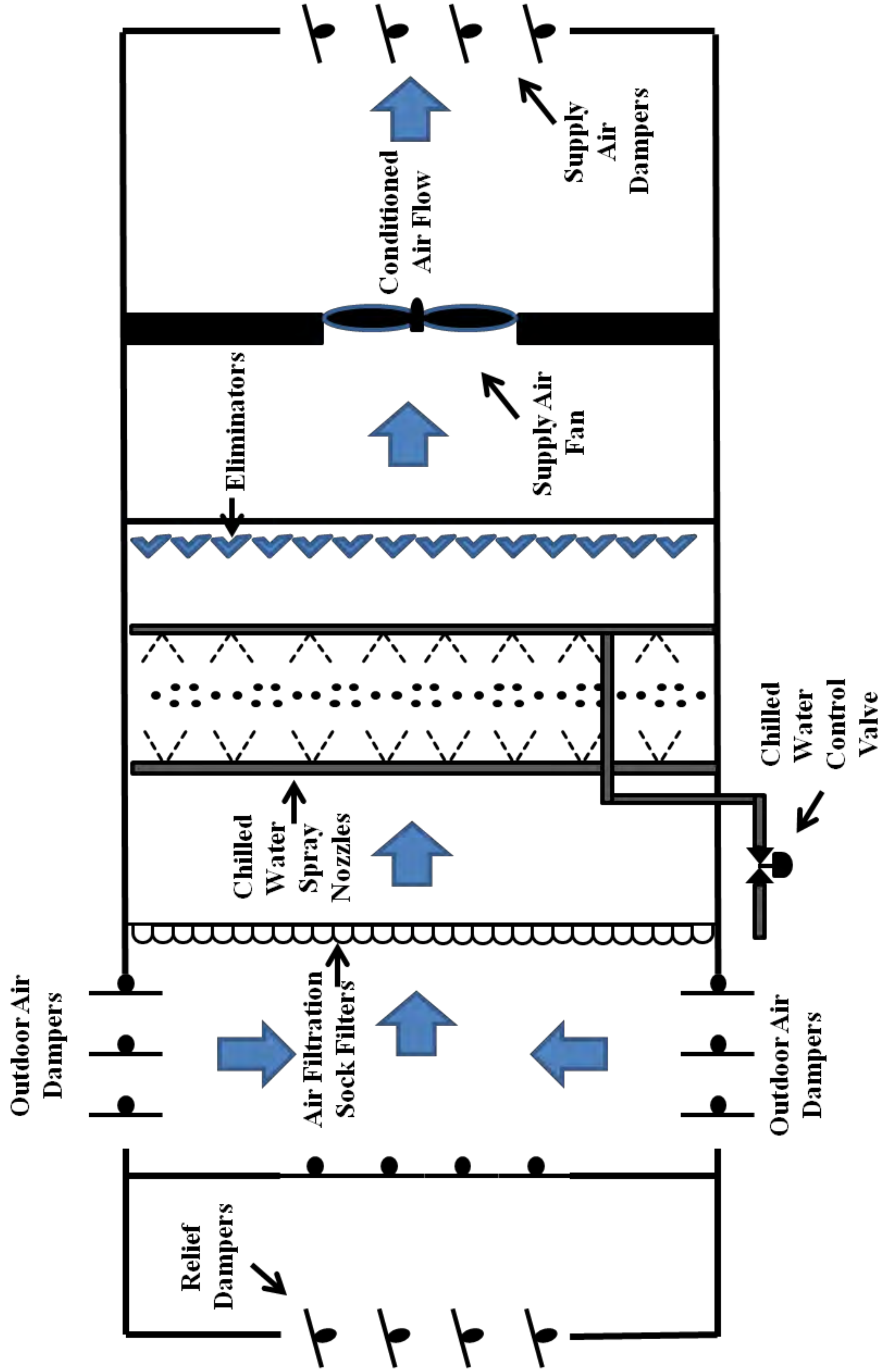

Figure 3.2 Open Air Washer System Architecture

Refrigeration System Compressors

There are five main types of air conditioner compressors:

1. Rotary compressor
2. Reciprocating
3. Centrifugal compressor
4. Screw compressors
5. Scroll compressors

While the function and ultimate output of these different types of compressors is the same, which is high pressure refrigerant vapor, the mechanical components and principles employed to accomplish the compression differ. These differing mechanical approaches and principles are apparent from the names of the compressors listed above.

The most common compressor is the reciprocating compressor. Reciprocating compressors can be open type or hermetically sealed type. A typical refrigerator compressor is hermetically sealed as shown in Figure 3.1.

Common refrigeration compressors range in size from less than 9 kW (approx. 9 hp) to 1 MW (approx. 1,000 hp), with condensing temperatures ranging from 15°C to 60°C, or higher. Mainstream refrigeration compressors power-source specifications, in terms of include voltage, frequency and phases are: 12 VDC and 24 VDC, 115/60/1 (single phase AC), 230/50/1, 208.230/60/1, 208.230/60/3 (three phase AC), 380/50/3, 460/60/3 and 575/60/3.

Refrigeration System Condenser

Condensers, in essence, are heat transfer devices. They permit extraction of heat form the hot, high pressure, refrigerant vapor; thus, allowing the vapor to condense into high pressure liquid phase. While the function of all condensers is the same, which is to condense high temperature, high pressure and high enthalpy refrigerant vapor, like the compressors and chillers, they differ based on their size and specific applications. For example, many large open air washer type air conditioning systems include large, water based, cooling towers for cooling the high pressure, high enthalpy, refrigerant vapor. On the other hand, some large closed air conditioning systems employ dry, forced air, type cooling towers such as the one shown in Figure 3.3.

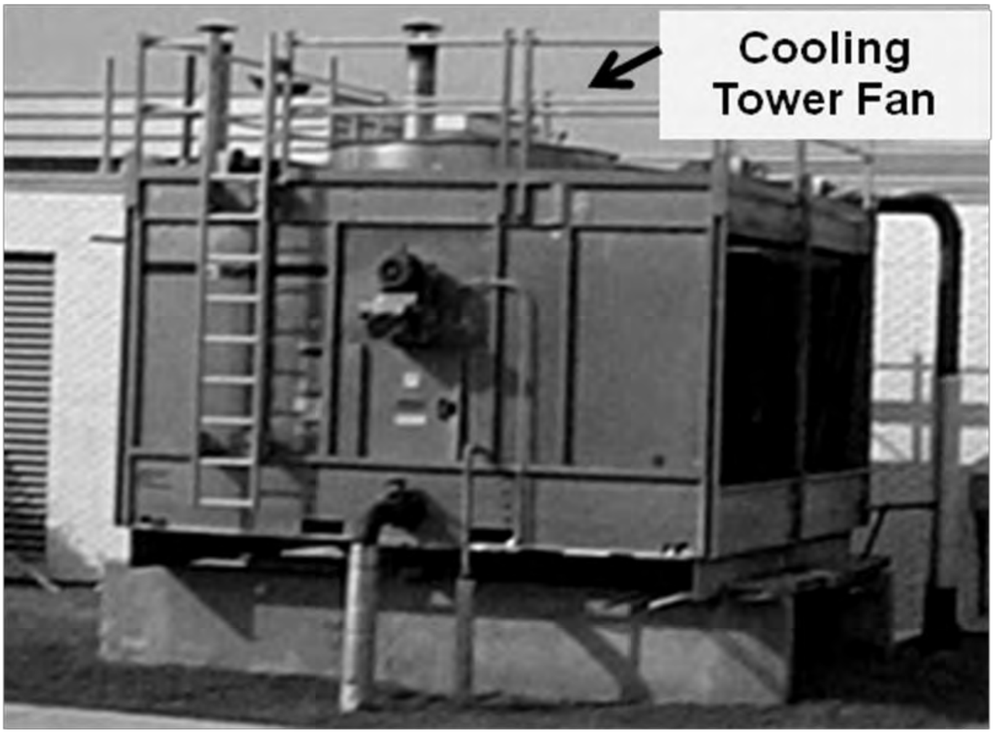

Figure 3.3 Forced Air Type Condenser Cooling Tower for Refrigeration System

Refrigerants

A refrigerant is a substance or medium used in a refrigeration system heat cycle. Refrigerants allow heat exchange and work to be performed in refrigeration systems as they undergo repetitive and cyclical phase changes from liquid to vapor and vapor to liquid, as illustrated in Figure 3.5.

Traditionally, fluorocarbons (FC) and chlorofluorocarbons (CFC) have been used as refrigerants. However, they are being phased out because of their Ozone Depletion Potential (ODP) and, in some cases, Global Warming Potential (GWP. They are being replaced by hydroflourocarbons, i.e. HFC-134a. Other, non-CFC and non-HFC refrigerants used in various applications are non-halogenated hydrocarbons such as methane, and non-hydrocarbon substances such as ammonia or sulfur dioxide.

Tables 3.1 and 3.2 list some of the commonly applied refrigerants and their corresponding important properties. These tables include chemical formulas, boiling points, critical temperatures, chemical properties, ozone depletion potential, global warming potential and likely application for the listed refrigerants.

Refrigerant	Formula	Boiling temperature (°C)	Critical temperature (°C)	Properties	Applications
Amonia	**NH_3**	**-33**	**133**	Penetrating odor, soluble in water, harmless in concentration up to 1/30%, non flammable, explosive	Large industrial plants
R11	**CCl_3F**	**8.9**	**198**	R11 is a single chlorofluorocarbon or CFC compound. ODP* = 1 and GWP** = 4000	Commercial plants with centrifugal compressors.
R12	**CCl_2F_2**	**-29.8**	**112**	Little odor, colorless gas or liquid, non flammable, non corrosive of ordinary metals, stable	Small plants with reciprocating compressors. Automotive, Medium Temperature Refrigeration
R22	**$CHClF_2$**	**-40.8**	**96**	R22 is a single hydrochlorofluorocarbon or HCFC compound. Low chlorine content and ozone depletion potential and only a modest global warming potential. R22 can still be used in small heat pump systems, but new systems can not be manufactured for use in the EU after 2003. From 2010 only recycled or saved stocks of R22 can be used. It will no longer be manufactured. ODP = 0.05 and GWP = 1700 .	Packaged air-conditioning units where size of equipment and economy are important. Air Conditioning, Low and Medium

Table 3.1 Commonly used refrigerants and some of their important properties.

Refrigerant	Formula	Boiling temperature (°C)	Critical temperature (°C)	Properties	Applications
R-134a	**CH_2FCF_3**			R134a is a single hydrofluorocarbon or HFC compound. No chlorine content, no ozone depletion potential. Modest global warming potential. ODP = 0 and GWP = 1300	Automotive replacement for R-12, Stationary A/C,
R417A				R417A is the zero ODP replacement for R22. Suitable for new equipment and as a drop-in replacement for existing systems.	Offers aprox. 20% more refrigeration capacity than R12 for same compressor.
R500	**CCl_2F_2 (73,8%); CH3 CH F2 (26.2%)**	**-33**		Similar to R12	Offers aprox. 20% more refrigeration capacity than R12 for same compressor.
R502	**$CClF_2$ (48,8%), CCl; F2-CF3 (51.2%)**	**-45.6**	**90.1**	Non flammable, non toxic, non corrosive, stable	Capacity comparable to R22.

Table 3.2 Commonly used refrigerants and some of their important properties.

* **ODP** or Ozone Depletion Potential of a refrigerant, or any other substance, is defined as the capacity of a single molecule of that refrigerant to destroy the Ozone Layer. All refrigerants use R11 as a datum reference, with R11 reference ODP of 1.0. The lower the value of the ODP, the less detrimental the refrigerant is to the ozone layer and the environment.

** **GWP** stands for Global Warming Potential. GWP is a measurement based over a 100-year period. It quantifies the effect a refrigerant will have on Global Warming relative to the GWP of Carbon Dioxide (CO_2). Carbon dioxide is assigned a GWP of 1. The GWP of all other substances or chemicals is assessed relative the Carbon Dioxide GWP of 1. The lower the value of GWP, the better the refrigerant is for the environment.

Note: Currently there are no restrictions on the use of R134A, R407C, R410A and R417A in original equipment or for maintenance and repair.

Expansion Valve

Expansion valve is an apparatus or component used in refrigeration systems to throttle the high pressure refrigerant in liquid phase from high pressure liquid state to low pressure liquid state.

Common refrigeration system expansion valves are also referred to as Thermal Expansion Valves or TXV's. The operating principle of the thermal expansion valve is illustrated in Figure 3.1. A thermal expansion valve functions as a metering device for the high pressure liquid refrigerant; it allows small proportionate amounts of the high pressure liquid refrigerant into the discharge side. This permits the refrigerant to transform into low pressure liquid; ready to be converted to vapor phase as it absorbs heat from the warm ambient air passing through the heat exchanger coils. As the refrigerant evaporates to higher temperature, the temperature of the gas in temperature sensing bulb rises. The higher temperature gas develops higher pressure thus pushing the expansion valve open. The valve, in its metering function, stays open only until the temperature in the evaporator section drops. When the temperature in the evaporator section drops, the temperature and the pressure of the gas in the temperature sensing bulb drop, therby resulting in the valve closure. This cycle repeats itself in a closed loop control fashion continuously in a typical refrigeration system.

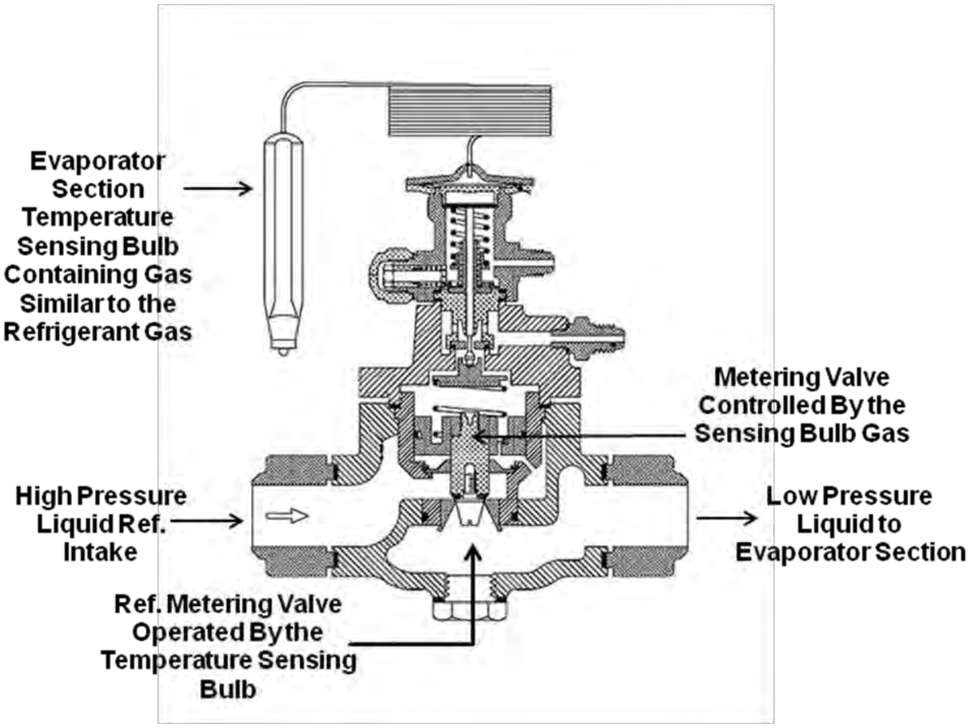

Figure 3.4 Refrigeration System Thermal Expansion Valve

Cooling Capacity of Refrigeration Systems

Cooling capacity of a refrigeration system is essentially the capacity of the refrigerant to exchange heat with the environment or ambient air. While in the absolute sense, cooling capacity represents the capacity of a refrigeration system to ***cool*** the environment or surroundings, whereas in the case of heat pumps, cooling capacity of the system could broadly include the capacity of the system to heat the environment; explained on the basis of role reversal of the evaporator and condenser.

Refrigeration System Capacity Quantification in A/C Tons

The cooling capacity of refrigeration systems is often defined in units called "tons of refrigeration". One ton of refrigeration represents the rate of refrigeration required to freeze a ton of 32°F (0°C) water in a 24-hr period. Stated alternatively, a ton of refrigeration is the rate of heat removal necessary to freeze a 2000 lbm of saturated water, at 32°F (0°C), within a period of 24 hours. For water, one ton of refrigeration amounts to 12,000 BTU/hr (12,660 kJ/hr). This is premised on the heat of fusion of water being 143.4 BTU/lbm as illustrated below:

The amount of heat that must be extracted from 32°F (0°C) water to freeze it to 32°F (0°C) ice is equal to the amount of heat that must be added to 32°F (0°C) ice to melt it to 32°F (0°C) water.

Therefore,

Rate of refrigeration for one ton of 32°F (0°C) water
= (143.4 BTU/lbm).(2000 lbm)/24 hr
= 11,950, or approximately, 12,000 BTU/hr

Since 1 BTU = 1055 Joules in metric or SI units' realm:

Rate of refrigeration for one ton of 0°C water
= (12,000 BTU/hr).(1055Joules/BTU)/(3,600 sec/hr)
= 3,517 watts or 3.517 kW

In the metric unit realm, the unit equivalent to 1 ton (US) of refrigeration is a tonne (Metric/European). One ton of refrigeration is based on freezing 1000 kg of 0°C water to 0°C ice in a 24 hour period. Calculation similar to the one illustrated in US units above equates one ton of refrigeration to 3.86 kW. Note that most residential air conditioning units range in refrigeration capacity from about 1 to 5 tons.

Basic Refrigeration Cycle

As we describe the refrigeration cycle, let's begin tracking the process at the point in the cycle where the refrigerant enters the compressor in the form of saturated or superheated low pressure vapor. Refer to Figure 3.5. The refrigerant, at this point, is high in enthalpy; albeit, the enthalpy is not the highest at the point of entry into the compressor.

The compressor compresses the saturated vapor and raises the pressure of the vapor to the maximum level. In doing so, the compressor packs the vapor or gaseous molecules of the working fluid closer together. The closer the molecules are together, the higher its energy and its temperature. The higher energy of the refrigerant vapor is manifested in its higher enthalpy.

The working fluid leaves the compressor as a hot, high pressure gas with the highest enthalpy and pressure. As the high pressure, high enthalpy refrigerant in vapor or superheated vapor forms and enters the condenser, the cooling and heat exchanging process begins. In the condenser, the condenser's cooling system extracts the heat from the superheated refrigerant vapor. This loss of heat transforms the refrigerant into a saturated liquid phase.

The next stage of the refrigeration cycle involves the expansion or throttling of the high pressure liquid refrigerant. The expansion and the evaporation stages can be explained as two separate stages or they can be viewed as one contiguous stage. Expansion valve, as described in detail earlier and as illustrated in Figure 3.4, meters the high pressure liquid refrigerant out into the evaporator segment. This allows the refrigerant to expand into low pressure liquid form. The low pressure liquid refrigerant then readily evaporates into low pressure saturated vapor form. In doing so, the low pressure liquid refrigerant engages in heat exchange with the ambient air. The ambient air is forced past the refrigerant coils by the system return or supply fan. This heat exchange in the evaporator section results in a substantial increase in the enthalpy of the refrigerant. The end product is a low pressure refrigerant vapor, ready to enter the compressor and repeat the refrigeration cycle.

The refrigeration cycle as described above is explained from the mechanical or fluid dynamics perspective. The thermodynamic properties and thermodynamic changes associated with each segment or stage of the refrigeration cycle are shown on the Pressure-Enthalpy graph in Figure 3.6. As indicated on the Pressure-Enthalpy graph, the compression process is an isentropic, where the entropy stays constant, or, $\mathbf{\Delta s = 0}$. The condensation stage, as illustrated in the Pressure-Enthalpy graph, is a non-adiabatic isobaric process. The next stage involving the expansion of the high pressure liquid refrigerant is an adiabatic and isenthalpic process, with $\mathbf{\Delta Q = 0}$ and $\mathbf{\Delta h = 0}$.

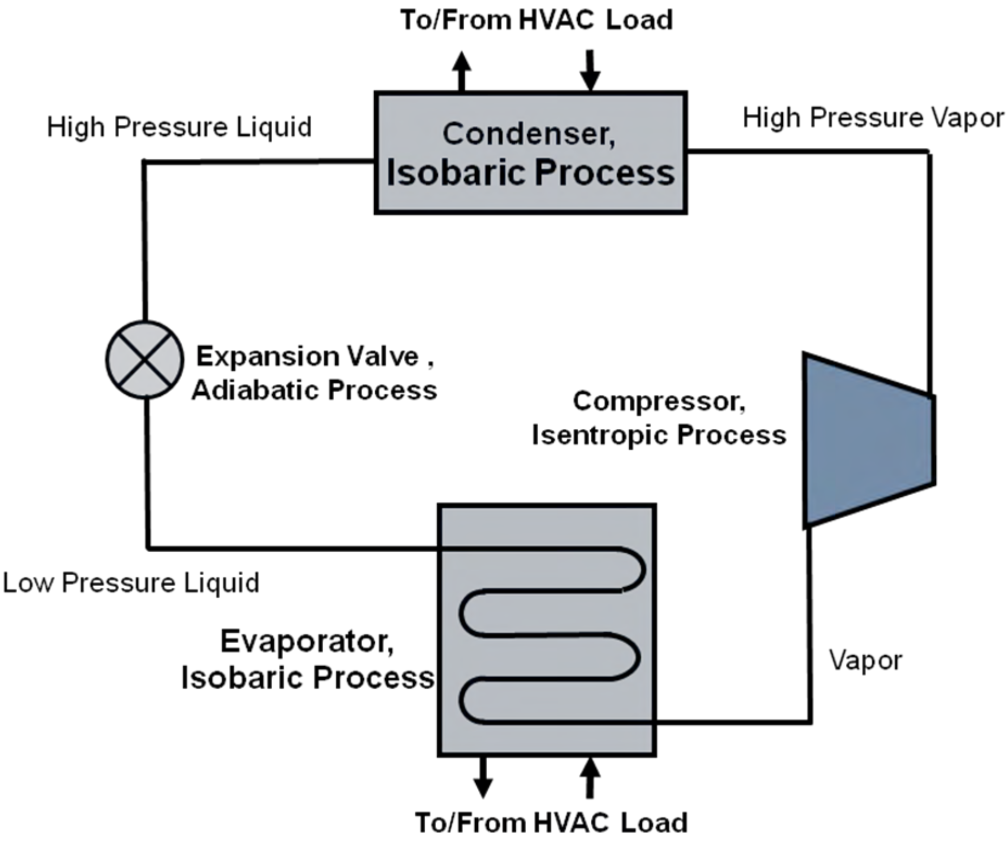

Figure 3.5 Refrigeration Cycle Process Flow Diagram

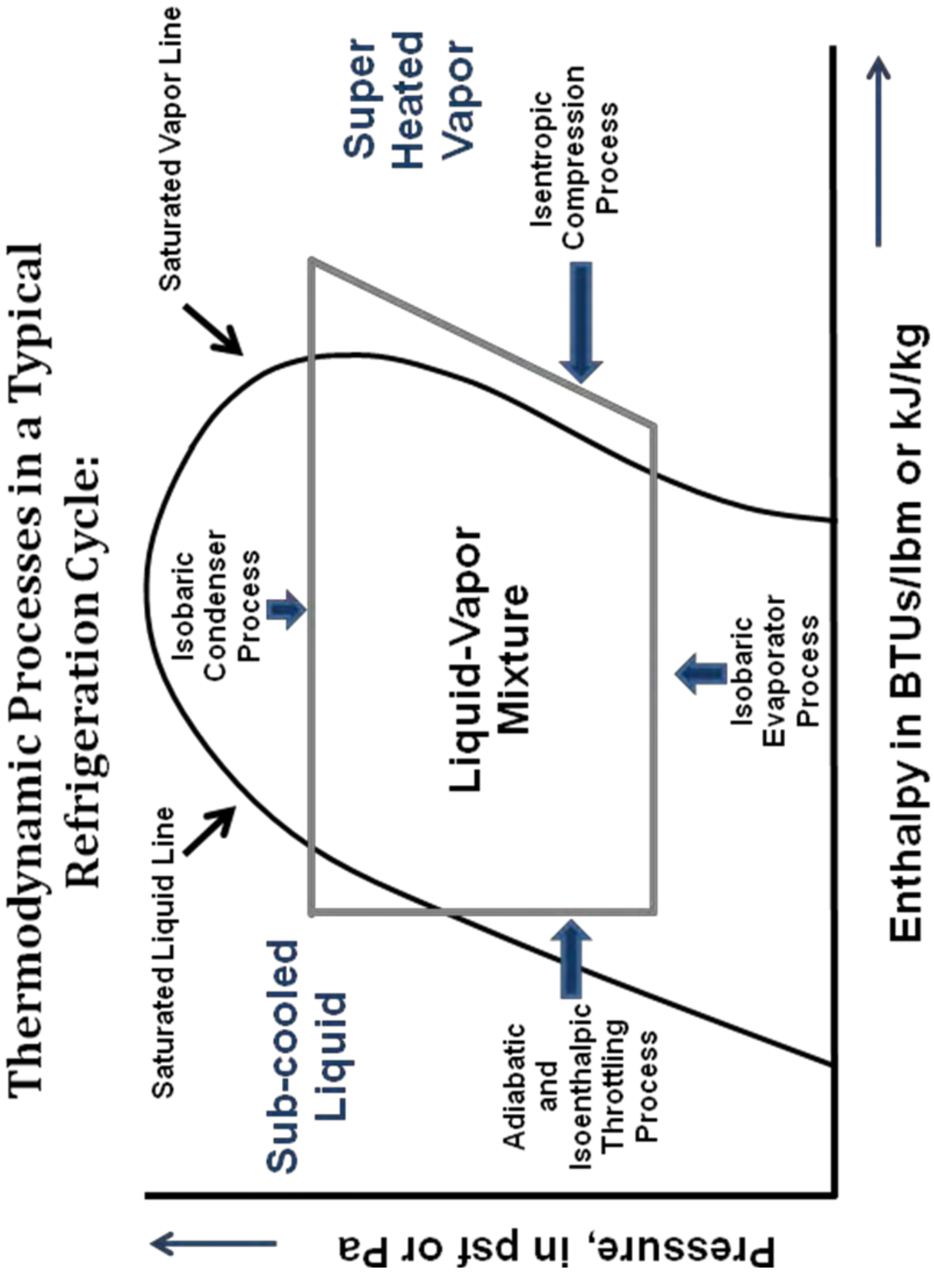

Figure 3.6 Refrigeration Cycle Pressure -Enthalpy Graph

Refrigerant Compression

As the name implies, the compression segment of the refrigeration cycle involves transformation of the low pressure refrigerant vapor into high pressure refrigerant vapor. The low pressure refrigerant vapor can be transferred from the evaporator to the compressor in the following forms:

(a) Mixture of liquid and vapor; also referred to as wet vapor. See Figure 3.7.

(b) Saturated vapor

(c) Slightly superheated vapor

Wet Vapor Compression Process

Figure 3.7 depicts the temperature versus the entropy diagram of a refrigeration cycle that is based on a ***wet compression cycle***. A wet compression cycle involves the compression of a refrigerant before it has evaporated completely into saturated vapor or slightly superheated vapor form. This state of the refrigerant is represented by point 3 in Figure 3.7.The refrigerant, at point 3, exists in vapor and liquid mixture form and cannot be compressed or pumped as efficiently as it can be when it is in a saturated vapor or slightly superheated form. In addition, wet compression results in compressor wear and performance problems.

Refrigerant Vapor Quality Ratio

The refrigerant vapor quality ratio is denoted by **"ω,"** and is defined as the ratio of the mass of pure vapor to the total mass of vapor and liquid mixture. The vapor quality ratio can be defined mathematically as follows:

$$\boldsymbol{\omega = (m_{vapor})/(m_{vapor} + m_{liquid})} \qquad \textbf{Eq. 3.1}$$

Where,

$\mathbf{m_{vapor}}$ = Mass of refrigerant in vapor form
$\mathbf{m_{liquid}}$ = Mass of refrigerant in combined vapor and liquid form

The values of quality ratio **ω** at points 1, 2, 3 and 4, as noted on the graph in Figure 3.7, project the state and composition of the refrigerant as follows:

Point 1: This point lies directly on the saturated liquid line. There is no vaporized refrigerant along this line; $\mathbf{m_{vapor}} = 0$ and, therefore, $\boldsymbol{\omega} = 0$. In other words, the refrigerant is in pure liquid phase all along this line.

Point 2: This point lies in the area of the graph where the refrigerant exists in a vapor-liquid mixture form. The fact that point 2 is closer to the saturated liquid line than it is to the saturated vapor line implies that, at point 2, the percentage of liquid refrigerant is greater than the percentage of vapor; in other words, $\mathbf{m_{liquid}} > \mathbf{m_{vapor}}$. Note that all points on the path from point 2 to point 3 represent wet vapor state.

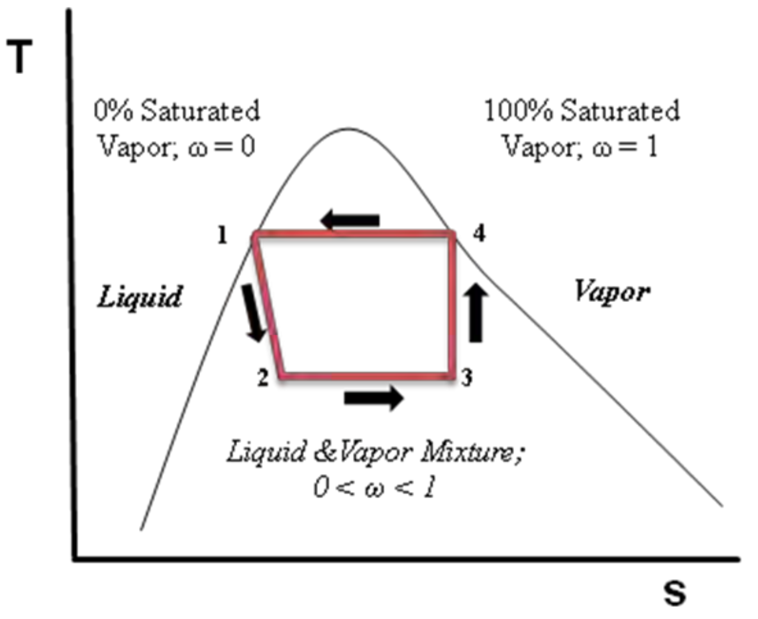

Figure 3.7 Wet Vapor Compression Cycle in Refrigeration Systems

Point 3: This point also lies in the area of the graph where the refrigerant exists in a vapor-liquid mixture form. However, point 3 is closer to the saturated vapor line than it is to the saturated liquid line. This means that, at point 3, the percentage of the vaporized refrigerant is greater than the percentage of liquid; in other words, $\mathbf{m_{vapor}} > \mathbf{m_{liquid}}$. The refrigerant compression process begins at point 3 and extends up to point 4.

Point 4: This point lies directly on the saturated vapor line. There is no liquid refrigerant along this line; $\mathbf{m_{liquid}} = 0$ and, therefore, $\boldsymbol{\omega} = 1$, or 100%. The refrigerant is in pure vapor phase all along this line. However, this vapor state is a saturated vapor state. Any loss of heat at this point would slide the refrigerant back into the condensed, or partially condensed, phase. On the other hand, if this point were to shift to the right of the saturated vapor line, it would be in a superheated vapor phase. Superheated vapor is also referred to as "***dry vapor.***"

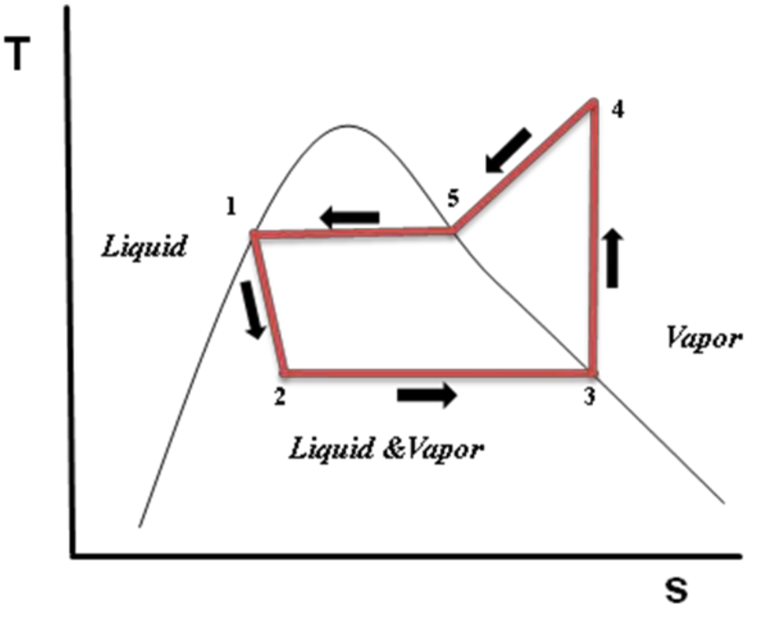

Figure 3.8 Dry Vapor Compression Cycle in Refrigeration Systems

Dry Vapor Compression Process

The compression efficiency in a refrigeration cycle can be enhanced by extending the refrigerant evaporation process all the way to the saturated vapor line or beyond. This is precisely the strategy employed with the dry vapor compression process as shown in Figure 3.8. In dry vapor compression refrigeration systems, the refrigerant leaves the evaporator section in either a saturated vapor or a superheated form as shown by point 3 in Figure 3.8. Furthermore, as obvious from Figure 3.8, the temperature of the refrigerant rises during the compression phase in dry vapor compression systems. As the temperature of the refrigerant rises, the refrigerant vapor becomes superheated. This is affirmed by the location of point 4 in Figure 3.8.

The following equations find a common application in refrigeration cycles:

Computation of a refrigerant compressor power utilizing the compression path 3-4:

$$P = \dot{W} = \dot{m}(h_4 - h_3) \qquad \textbf{Eq. 3.2}$$

Where, ***W*** represents "work flow rate" in BTU/sec or J/s.

Computation of refrigerant mass flow rate **m,** utilizing the evaporation path 2-3:

$$\dot{m} = \frac{\dot{Q}_{in}}{(h_3 - h_2)} \qquad \textbf{Eq. 3.3}$$

Where, $\boldsymbol{Q_{in}}$ represents "heat flow rate" in BTU/sec or J/s.

Rearrangement of Eq. 3.3 yields the equation for heat flow rate calculation:

$$\dot{Q}_{in} = \dot{m}(h_3 - h_2) \qquad \textbf{Eq. 3.4}$$

These equations are premised on the refrigeration cycle depicted in Figure 3.8. The nomenclature used in these equations is specific to the refrigeration cycle shown in Figure 3.8. If letters are used to denote various points in the refrigeration cycle, the numerical subscripts are replaced with letters as illustrated in Case Study 3.1 below.

Coefficient of Performance, or COP, in Refrigeration Systems

Coefficient of performance of a refrigeration system is defined as the ratio of useful energy transfer to the work input. The system is considered to be the refrigerant in the computation of COP. The equations for the COP are as follows:

$$COP_{refrigerator} = \frac{Q_{in}}{(Q_{out} - Q_{in})} \qquad \textbf{Eq. 3.5}$$

Also,

$$COP_{refrigerator} = \frac{Q_{in}}{W_{in}} \qquad \textbf{Eq. 3.6}$$

and,

$$COP_{refrigerator} = COP_{heatpump} - 1 \qquad \textbf{Eq. 3.7}$$

EER = 3.41 x COP **Eq. 3.8**

SEER, Seasonal Energy Efficiency Ratio

SEER or Seasonal Energy Efficiency Ratio is a rating based on the cooling output in BTU during a cooling season divided by the total electrical energy drawn from the utility, during the same period, in Watt-Hours. Therefore, the engineering units for SEER rating are BTU/W-h. The higher the SEER rating of an air conditioning system the more efficient it is.

Example 3.1
As an Energy Engineer, you have been asked by your client to determine the total annual cost of electrical energy consumed and the input power demanded by an air conditioning system with the following specifications:

SEER Rating: 10 BTU/W-h
Air Conditioning System Rating: 10,000 BTU/hr
Total, Annual, Seasonal Operating Period: 130 days, 8 hours per day.
Average, Combined, Electrical Energy Cost Rate: $0.18/kWh

Solution:
Annual Cost of Energy = ($0.18/kWh).(Total Energy Drawn From The Utility, Annually)

Total Power Demanded From The Utility

= (Air Conditioning System Rating, in BTU/hr)/(SEER Rating, in BTU /W-hr

Eq. 3.9

Note: Both BTU values in **Eq. 3.9** are outputs, while the W-hr value represents the input energy drawn from the line (utility) side of the power distribution system.

∴ **Total Power Drawn From The Utility**

= (10,000 BTU/hr)/(10 BTU /W-hr)

= 1,000 Watts, or 1 kW

Then,

Total Energy Drawn From The Utility, Annually

= (1 kW).(Total Annual Operating Hours)

= (1 kW).(130 Days).(8 Hours/Day)

= 1,040 kWh

∴ **Total Annual Cost of Electrical Energy Consumed**

= (1,040 kWh)/($0.18/kWh)

= $187.20

Case Study 3.1 - Refrigeration Cycle:

An air-conditioning system uses HFC-134a refrigerant. The refrigeration system is cycled between 2.0 MPa and 0.40 MPa. A pressure-enthalpy diagram for HFC-134a is presented on the next page.

a) Draw the refrigeration cycle on the given diagram. See Figure 3.9.
b) Determine the change in entropy during the throttling process.
c) Determine the percentages of liquid and vapor at the end of the throttling segment of the refrigeration cycle.
d) How much enthalpy is absorbed by the system (refrigerant) in the evaporation (latent) phase?
e) How much enthalpy is extracted from the system (refrigerant) in the condensation (condenser) phase of the cycle?
f) In which leg of the refrigeration cycle would expansion be used?

g) If the refrigeration capacity of this system was sized based on the enthalpy extracted from the refrigerant, as calculated in part (e), what would the specification be in tonnes (Metric).

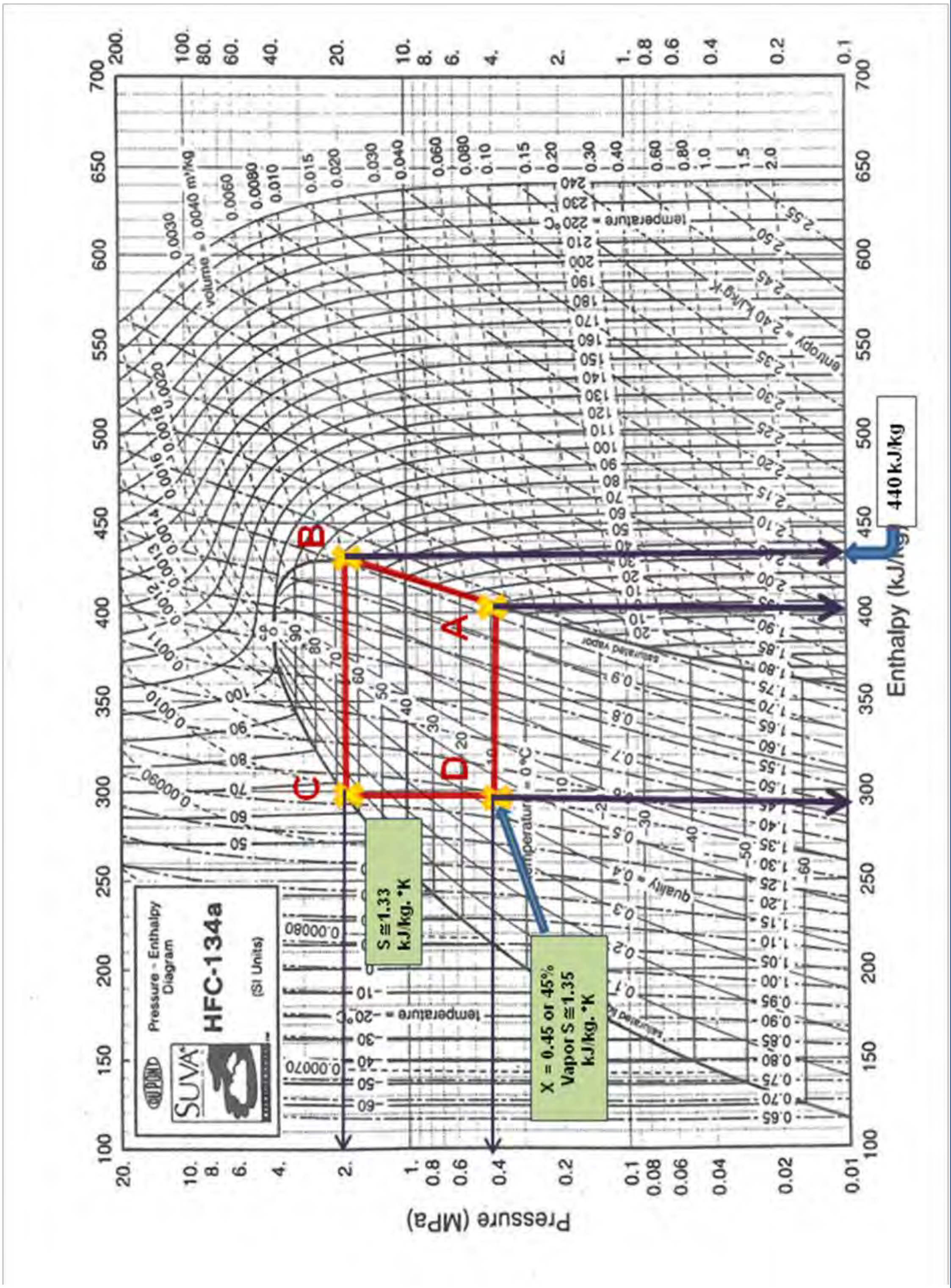

Figure 3.9 Pressure-Enthalpy Diagram, Case Study 3.1

(a) Draw the refrigeration cycle for each of the following paths:

Solution:
The process involved in the drawing of the refrigeration cycle is as follows:

C – D:
See Figure 3.9. Locate the 2 MPa and 0.4 MPa points along the pressure (vertical) axis of the chart; name these points "C" and "D," respectively. This is the throttling portion of the refrigeration cycle. Note: HFC-134a (at point C) is in high pressure, saturated liquid phase.

Throttling process is adiabatic and Δ h = 0

Draw a straight, vertical line down from C to D. At point D, R-134a is in liquid-vapor mixture phase.

D - A:
See Figure 3.9. The next step involves complete transformation of the refrigerant from liquid to gaseous phase through absorption of heat, or Δ h. This is an, non-adiabatic, isobaric process; so draw a straight, horizontal line from D to A. This step is referred to as the ***evaporator*** segment of the refrigeration cycle. This is where the system (refrigerant) performs cooling of the environment as its phase undergoes through latent transformation from liquid to gaseous phase.

A - B:
See Figure 3.9. The next step involves the transformation of HFC-134a from LOW pressure (0.4 MPa) gaseous phase to HIGH (2 MPa) pressure gaseous phase. This is the ***compressor*** segment of the refrigeration cycle. This phase is an isentropic process, **Δ s** = 0. Therefore, draw a straight line from point A to B, asymptotic to S ≅ 1.73 kJ/kg °K.

B - C:
See Figure 3.9**.** The next step involves the transformation of HFC-134a from high pressure (**2.0 MPa**) gaseous phase to high pressure, saturated liquid phase. This segment constitutes the ***condenser*** segment of the refrigeration cycle. This is an **isobaric** process, **Δ P = 0.** Therefore, draw a straight line from point B to C, along **P = 2 MPa** line.

(b) Determine the change in entropy during the throttling process:

Solution: See Figure 3.9.

$$\Delta s = s_D - s_C \cong 1.35 - 1.33 \cong 0.02 \text{ kJ/kg. °K.}$$

(c) Determine the percentages of liquid and vapor at the end of the throttling segment of the refrigeration cycle.

Solution:
This involves reading the value of "x," the quality at point "**D**" from the Pressure-Enthalpy diagram. **See Figure 3.9.**

$$x = (m_{vapor}) / (m_{vapor} + m_{liquid}) = 0.45 \text{ or } 45\%$$

In other words,

$$m_{vapor}\ (\%) = 45,$$

and since:

$$(\%m_{vapor} + \%m_{liquid}) = 100\%,$$

$$m_{liquid}\ (\%) = 100 - 45 = 55\%$$

(d) How much enthalpy is absorbed by the system (refrigerant) in the **evaporation** (latent) phase?

Solution:
This involves step **D - A.** See the Pressure – Enthalpy Diagram and **Figure 3.9**:

$$\Delta h_{D-A\ Phase} = h_A - h_D = 400 - 300 = 100 \text{ kJ/kg}$$

(e) How much enthalpy is extracted from the system (refrigerant) in the **condensation** (condenser) phase of the cycle?

Solution:
This involves step **B** - **C**. **See Figure 3.9.** :

$$\Delta h_{B-C\ Phase} = h_B - h_C = 440 - 300 = 140 \text{ kJ/kg}$$

(f) In which leg of the refrigeration cycle would expansion occur?

Answer: The throttling leg, Step **C - D. See Figure 3.9.**

(g) Determine the refrigeration capacity in tons based on the enthalpy extracted from the refrigerant, as calculated in part (e):

Solution:
Heat extracted from the refrigerant in part (e) = 140 kJ/kg.

Rate of refrigeration for this system

= (140 kJ/kg).(1000 kg)/24 hr
= 140,000 kJ/24 hr
= 140,000 kJ/(24 hr)/(3600sec/hr)
= 1.62 k.J/sec
= 1.62 kW

Since 1 tonne of refrigeration amounts to 3.86 kW, the refrigeration capacity of this system, in tonnes, would be:

= (1.62 kW)/(3.86 kW)
= 0.4 tonne.

Direct Digital Control of HVAC Systems

Like many manufacturing operations and chemical process, nowadays HVAC systems are taking advantage of automation and automated control systems. Automated closed loop control systems permit operator-error-free and reliable operation of HVAC systems. Prior to the 1980's, proper and effective operation of many large HVAC systems in industrial and commercial domains required a sizeable crew of utility engineers and technicians. Their sole responsibility, in many cases, was to monitor, track, audit and optimize the performance of chillers, compressors, air washers, cooling towers, fans and pumps. This approach required continuous manual monitoring, data recording and frequent manual adjustments of HVAC controls.

The advent of industrially hardened computers and Programmable Logic Controllers (PLC) transformed all that and ushered the era of automation in the HVAC realm. Central controllers, whether they consist of PC's or PLC's, are programmable; meaning the control system code/program may be customized for the specific use. Major program features, within the overall application program, include synchronous controlled events, time schedules, set-points, control logic, timers, trend logs, regression analysis based forecasts, alarms, graphs, and graphical depictions of the HVAC systems with ***live*** or ***real-time*** data points.

There are myriad alternative automated HVAC brands in the market to choose from. Some brands are relatively small concerns and others are well established and well supported subsidiaries of large firms. End users in the market for automated HVAC systems are advised to consider the following criteria in the formulation of their decision for a specific brand:

1) Is the automated HVAC system provider well established in the market with reasonable availability of technical support during start-up phase, commissioning phase and post installation operation?

2) Ensure that the technology (hardware, software and firmware) offered by the vendor is well beyond its "***infancy***" period, and therefore, vetted.

3) If possible, avoid technology that is proprietary and offers little compatibility with the mainstream PC's and PLC's.

4) Select brands and technology that are compatible with established, recognized, standard communication protocol, such as that established and sponsored by recognized entities like the Institute of Electrical and Electronic Engineers (IEEE).

5) Choose field input and output hardware that is versatile and compatible with mainstream sensors and output transducers.

6) Preference should be given to PC's and PLC's that operate on, or are compatible with, mainstream operating system platforms, i.e. Microsoft Windows 7, Vista, and equivalent late generation systems. This is to ensure compatibility with useful application software packages, such as the Wonderware ® HMI, Human Machine Interface package and other equivalent software packages.

HVAC control systems are sometimes embedded into comprehensive Energy Management Systems (EMS), or Building Management Systems (BMS). This approach offers the following advantages:

1) Cost reduction through economies of scale.

2) More attractive return on investment and economic justification.

3) Obvious confluence of automated HVAC system projects with other energy productivity improvement projects. This could potentially provide additional advantage of financing such projects through ESCO/EPC programs. **(Reference: Finance and Accounting for Energy Engineers, By S. Bobby Rauf, Fairmont Press.)**

4) Central monitoring and control of all utilities in an industrial or commercial facility.

5) Computerized streamlining and scheduling of the HVAC system predictive maintenance programs along with the predictive maintenance of all other plant equipment.

General approach to the architecture of most automated HVAC systems is illustrated in Figure 3.10. As shown in this system architecture, the core brain of the system is the CPU, Central Processing Unit. This CPU can be a PLC, Programmable Logic Controller, a DDC, Direct Digital Controller, or simply an industrially hardened PC. The language or code utilized to program the PLC or the PC would be proprietary and specific to the type of PLC or PC installed. Specifications such as the size of CPU memory, RAM, Random Access Memory and the ROM, Read Only Memory, would determine the length of code or program that can be written and the number of data points that can be monitored, tracked, trended and controlled. The heart of the CPU is referred to as a ***"microprocessor***." Controllers or CPU's that drive large HVAC systems, or combination of large HVAC and EMS Systems, are sometimes equipped with dual microprocessors in order to limit the program scan times. Program scan times are sometimes also referred to as cycle times. Reasonable scan time for an average automated HVAC system is approximately 30 milliseconds. Shorter scan times are desirable. Longer cycle times can, potentially, cause the control system to become ineffective and disfunctional.

As shown in Figure 3.10, the CPU is sometimes connected to a Monitoring Terminal, which includes a PC, display/monitor, keyboard and a mouse. This PC System serves as an interface between the CPU and the programmers, maintenance technicians and certain qualified production personnel. Often, this is where the control program and application software reside.

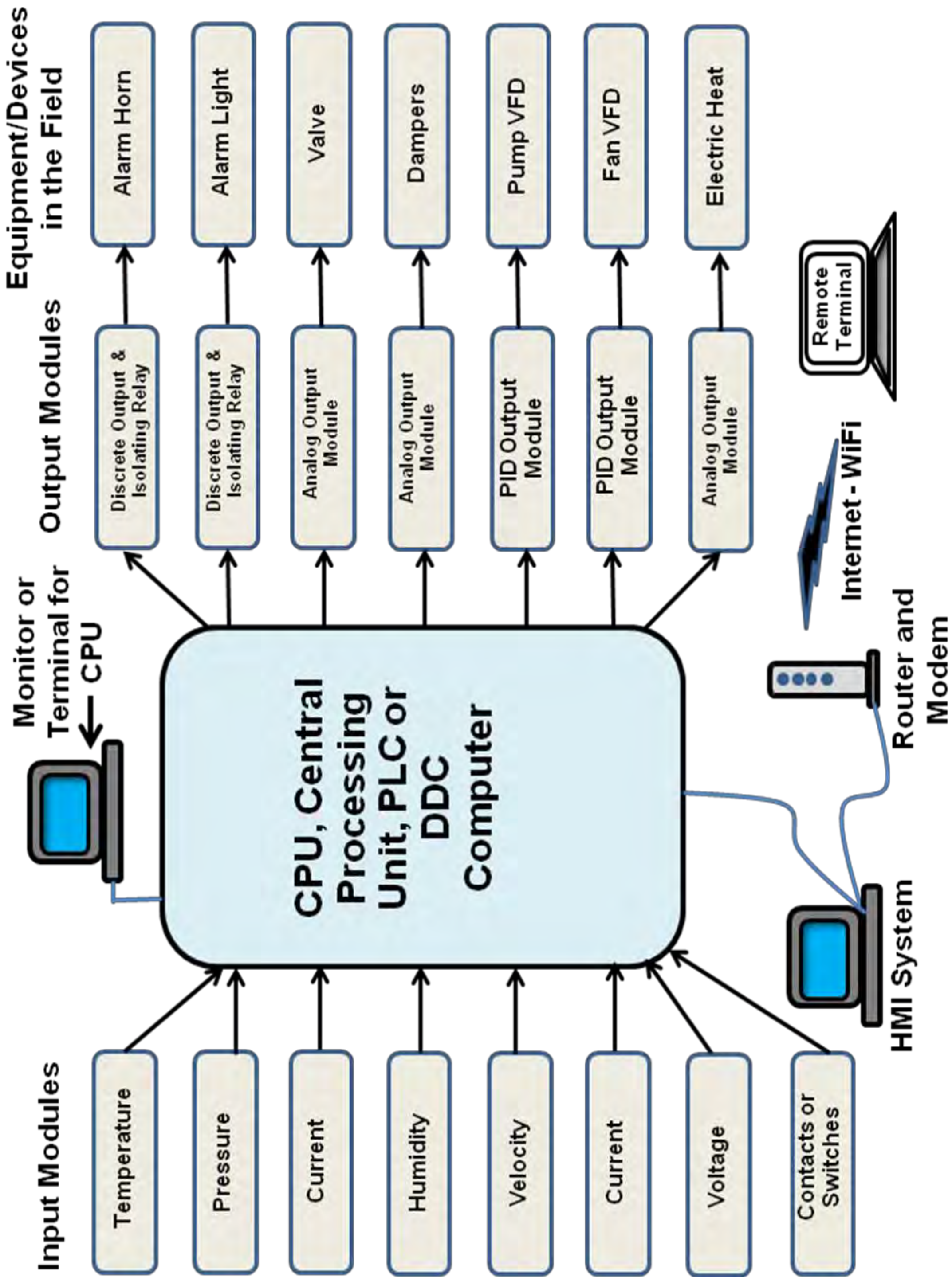

Figure 3.10 Automated HVAC Control System Architecture

The PC and the monitor shown at the bottom of Figure 3.10 serve as the process annunciation system; complete with HVAC graphics, real time data, alarms, trend charts, event logs, and production performance data diagnostics. This system, or sub-system, is referred to as the Human Machine Interface or HMI System. This PC based system is equipped with a suitable operating system such as the Microsoft Windows or Vista, and an HMI software package such as, Wonderware or equivalent. The HMI system and/or CPU Terminal are often connected in network format to other Information Technology (IT) and accounting computers for monitoring of productivity as well as for production cost tracking purposes. As shown in the automated HVAC system architecture diagram, the HMI system and the CPU terminal are sometimes linked to remote or off-site locations, i.e. corporate offices, through Ethernet, wireless routers and modems.

As shown in Figure 3.10, automated HVAC systems also consist of other peripheral equipment, such as input and output modules. Input modules receive different types of signals from peripheral devices/sensors and process them for presentation to the CPU. Output modules take various outputs or commands from the CPU, and package them such that they can be used to turn on or turn off control equipment in the field.

Inputs and outputs, in essence, are signals or commands. The main categories of inputs and outputs are as follows:

1) Digital or discrete inputs

2) Digital or discrete outputs

3) Analog inputs

5) Analog outputs

Digital or Discrete Inputs: The digital inputs, sometimes referred to as discrete inputs, are simply closed or open contact signals that represent the closed or open status of switches in the field. These switches or contacts can be limit switches, safety interlock switches, or auxiliary contacts of motors, sensors, etc. The normally closed safety interlock contact shown in Figure 3.11 presents 110 volts AC to the discrete input block. Closed contact type inputs can present other 'non-zero" voltages, such as 5 volts DC, 10 volts DC and 24 volts AC/DC, or 110 volts AC to the discrete input modules. The end result is a logic level HIGH or "1" to the CPU for computation and decision making purposes. On the other hand, an open contact type input, such as the one shown to represent the normally open pressure switch in Figure 3.11, would be interpreted as a LOW logic level or "0" by the discrete input module and the CPU. Another perspective on the role of input modules would be to view them as intermediary devices that transform and isolate discrete signals being received from peripheral devices.

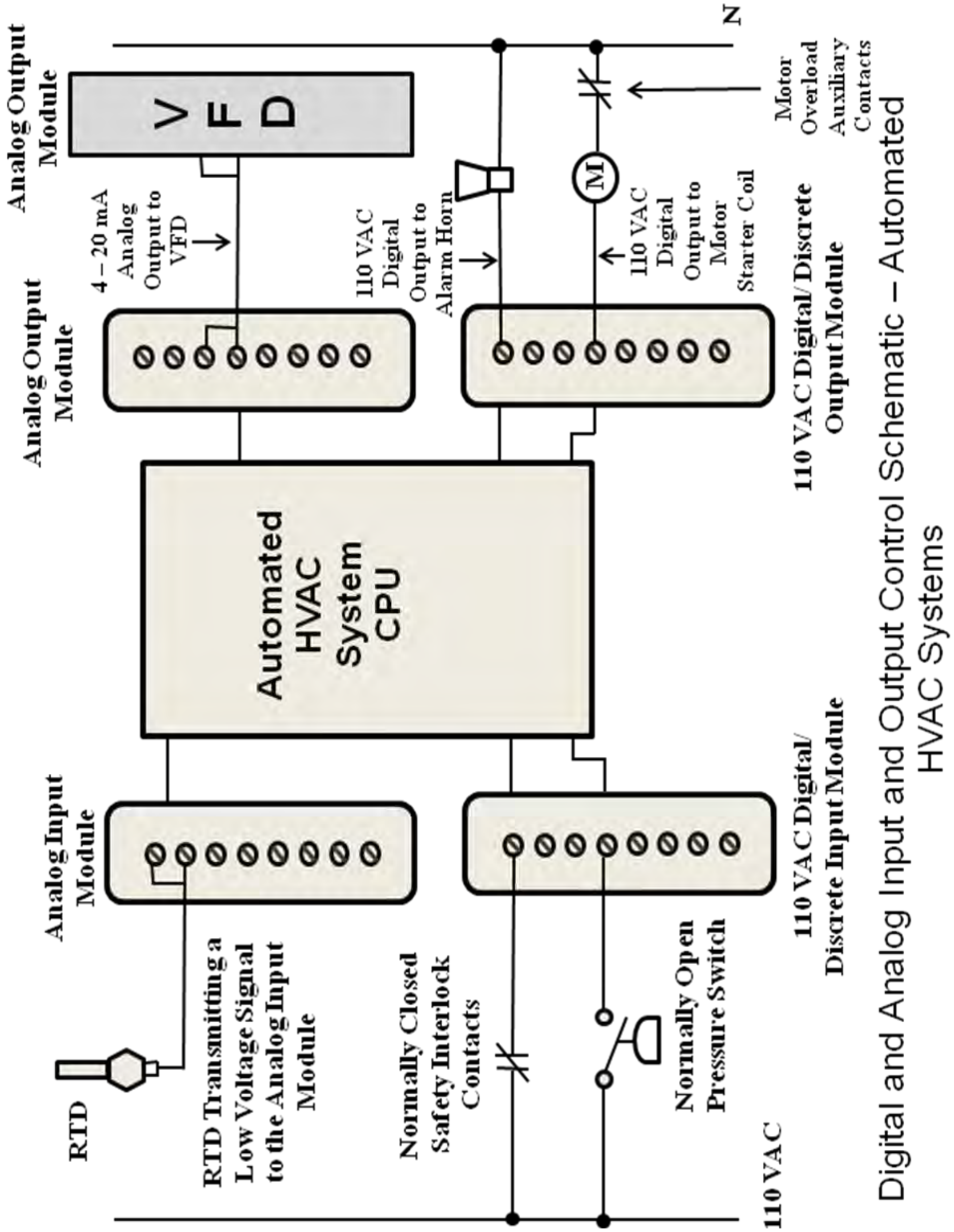

Figure 3.11 Automated HVAC Control System Architecture

Digital or Discrete Outputs: Digital outputs are typically generated by relay contacts based in the discrete output modules. A “1” or a “HIGH” from the CPU is transformed into a 5 volts DC, 10 volts DC, 24 volts AC/DC, or 110 volts AC signal, which can then be used to start and stop equipment, as part of the overall automated HVAC control scheme. For example, a “1” from the CPU is transformed into a 110 volt AC signal by the 110 Volt AC Discrete Output Module in Figure 3.11. This 110 Volt AC output from the output module is fed to the AC motor starter coil to turn on the motor in response to the specific input conditions and the program or algorithm. Another discrete output from the 110 Volt AC Discrete Output Module, in Figure 3.11, is fed to an alarm horn to annunciate an alarm condition. Note that in both of these examples, the second terminal of each controlled field device is connected to the power system neutral, designated as “N.”

Analog Inputs: Analog inputs represent gradually varying parameters or signals. Analog signals, in automated HVAC systems, include temperature, humidity, volume, pressure and even electrical current drawn by fan, blower or pump motors. Common analog current range is 4 to 20 mA. In certain cases, signals representing pressure, temperature, and volume are converted into analog voltage signals. These analog voltage signals range, commonly, from 0 to 10 Volts DC. Analog input modules can also convert signals from thermocouples, RTD’s, pressure sources and other types of analog signal sources, into digital logic numbers for use by the CPU. See the RTD application example shown in Figure 3.11. The analog temperature measurement from the RTD, in the form of low voltage signal, is fed to the Analog Input Module. The analog module converts this voltage signal into an equivalent digital value. This digital value is presented to the CPU for computation or algorithm execution purposes.

Analog Outputs: Operation of analog outputs can, essentially, be explained as the operation of analog inputs in reverse. Analog output begins with a presentation of a digital or binary number by the CPU to the analog output module. This digital or binary number is then transformed into an equivalent analog signal such as 0 – 5 Volt DC, 0 – 10 Volt DC, 0 – 110 Volt AC or 4 – 20 mA DC signal. Any of such analog signals can then be used to command the gradual or analog operation of HVAC control equipment like valves, variable frequency drives, heat sources, electric valve actuators, pneumatic actuators, etc. In Figure 3.11, the Analog Output Module is shown feeding a 4 to 20 mA analog output to a Variable Frequency Drive (VFD). The drive can base its output power frequency on this 4 to 20 mA analog input to control the speed of an HVAC fan or pump motor.

Segment 3

Self Assessment Problems and Questions

1. An air-conditioning system uses HFC-134a refrigerant. The pressure-enthalpy diagram for this refrigerant is presented on the next page. The refrigeration system is cycled between 290 psia and 60 psia.

(a) Draw the refrigeration cycle on the given diagram.
(b) What is the change in ***enthalpy*** during the ***expansion*** process
(c) Determine the percentages of liquid and vapor at the end of the throttling segment of the refrigeration cycle.
(d) How much enthalpy is absorbed by the system (refrigerant) in the evaporation (latent) phase?
(e) How much enthalpy is extracted from the system (refrigerant) in the condensation (condenser) phase of the cycle?
(f) Determine the percentages of liquid and vapor at B.
(g) Assume that the mass flow rate of the refrigerant being cycled in this air conditioning system is 10 lbm/min and the compressor efficiency is 70%. Determine the amount of electrical power demanded by the compressor motor if the compressor motor efficiency is 90%.
(h) Which leg of the refrigeration cycle would be considered isentropic?

I. A-B
II. B-C
III. C-D
IV. D-A

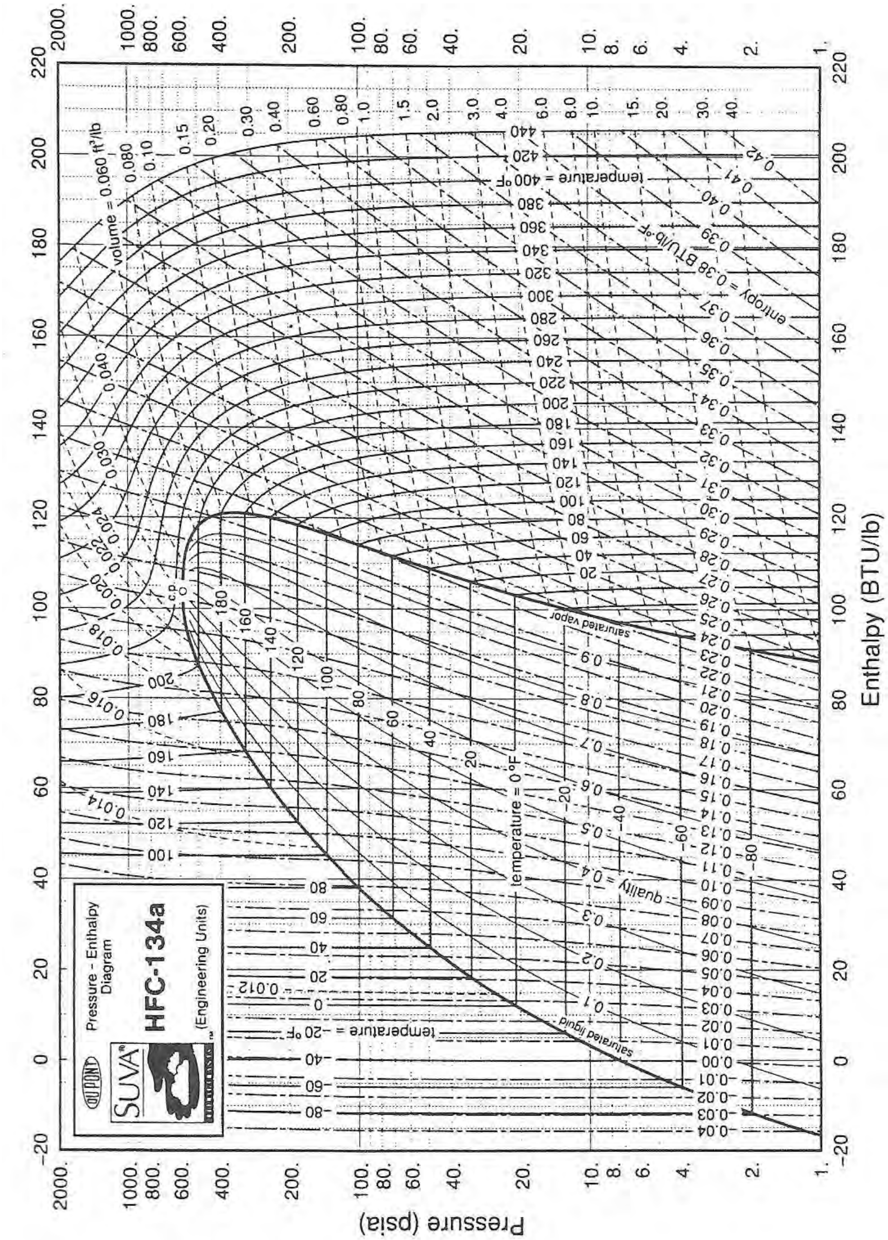

Figure 3.12 Pressure-Enthalpy Diagram, HFC-134a

2. As stated in this segment, 1 tonne (SI/Metric) of refrigeration capacity is equivalent to 3.86 kW of power. Provide the mathematical proof for this equivalence.

3. As an Energy Engineer, you are performing an energy cost assessment for operating a 20,000 BTU/hr air conditioner. Based on the data and specifications provided below, determine the total annual cost of electrical energy consumed and the input power demanded by the air conditioning system:

SEER Rating: 12 BTU/W-h
Air Conditioning System Rating: 20,000 BTU/hr
Total, Annual, Seasonal Operating Period: 200 days, 10 hours per day.
Average, Combined, Electrical Energy Cost Rate: $0.20/kWh

APPENDICES

Appendix A
Solutions for Self-Assessment Problems

This appendix includes the solutions and answers to end of segment self-assessment problems and questions.

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Appendix B
Steam Tables

These steam tables, copyright ASME, published with ASME permission, do not include the heat of evaporation value, $\mathbf{h_{fg}}$, values for the saturation temperature and pressures. The saturated steam tables presented in this text are the compact version. However, the $\mathbf{h_{fg}}$ values can be derived by simply subtracting the available values of $\mathbf{h_L}$ from $\mathbf{h_v}$, for the respective saturation pressures and temperatures. In other words: $\mathbf{h_{fg} = h_v - h_L}$

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Appendix C
Common Units and Unit Conversion Factors

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Appendix D
Common Symbols

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