
ABSTRACT

The *Thermodynamics, Heat Transfer, and Fluid Flow Fundamentals Handbook* was developed to assist nuclear facility operating contractors provide operators, maintenance personnel, and the technical staff with the necessary fundamentals training to ensure a basic understanding of the thermal sciences. The handbook includes information on thermodynamics and the properties of fluids; the three modes of heat transfer - conduction, convection, and radiation; and fluid flow, and the energy relationships in fluid systems. This information will provide personnel with a foundation for understanding the basic operation of various types of DOE nuclear facility fluid systems.

Key Words: Training Material, Thermodynamics, Heat Transfer, Fluid Flow, Bernoulli's Equation

OVERVIEW

The *Department of Energy Fundamentals Handbook* entitled *Thermodynamics, Heat Transfer, and Fluid Flow* was prepared as an information resource for personnel who are responsible for the operation of the Department's nuclear facilities. A basic understanding of the thermal sciences is necessary for DOE nuclear facility operators, maintenance personnel, and the technical staff to safely operate and maintain the facility and facility support systems. The information in the handbook is presented to provide a foundation for applying engineering concepts to the job. This knowledge will help personnel more fully understand the impact that their actions may have on the safe and reliable operation of facility components and systems.

The *Thermodynamics, Heat Transfer, and Fluid Flow* handbook consists of three modules that are contained in three volumes. The following is a brief description of the information presented in each module of the handbook.

Volume 1 of 3

Module 1 - Thermodynamics

This module explains the properties of fluids and how those properties are affected by various processes. The module also explains how energy balances can be performed on facility systems or components and how efficiency can be calculated.

Volume 2 of 3

Module 2 - Heat Transfer

This module describes conduction, convection, and radiation heat transfer. The module also explains how specific parameters can affect the rate of heat transfer.

Volume 3 of 3

Module 3 - Fluid Flow

This module describes the relationship between the different types of energy in a fluid stream through the use of Bernoulli's equation. The module also discusses the causes of head loss in fluid systems and what factors affect head loss.

The information contained in this handbook is by no means all encompassing. An attempt to present the entire subject of thermodynamics, heat transfer, and fluid flow would be impractical. However, the *Thermodynamics, Heat Transfer, and Fluid Flow* handbook does present enough information to provide the reader with a fundamental knowledge level sufficient to understand the advanced theoretical concepts presented in other subject areas, and to better understand basic system and equipment operations.

**Department of Energy
Fundamentals Handbook**

**THERMODYNAMICS, HEAT TRANSFER,
AND FLUID FLOW
Module 1
Thermodynamics**

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TERMINAL OBJECTIVE

- 1.0 Given operating conditions of a system, **EVALUATE** the thermodynamic state of the system.

ENABLING OBJECTIVES

- 1.1 **DEFINE** the following properties:
- Specific volume
 - Density
 - Specific gravity
 - Humidity
- 1.2 **DESCRIBE** the following classifications of thermodynamic properties:
- Intensive properties
 - Extensive properties
- 1.3 **DEFINE** the thermodynamic properties temperature and pressure.
- 1.4 **DESCRIBE** the Fahrenheit, Celsius, Kelvin, and Rankine temperature scales including:
- Absolute zero temperature
 - The freezing point of water at atmospheric pressure
 - The boiling point of water at atmospheric pressure
- 1.5 **CONVERT** temperatures between the Fahrenheit, Celsius, Kelvin, and Rankine scales.
- 1.6 **DESCRIBE** the relationship between absolute pressure, gauge pressure, and vacuum.
- 1.7 **CONVERT** pressures between the following units:
- Pounds per square inch
 - Inches of water
 - Inches of mercury
 - Millimeters of mercury
 - Microns of mercury
- 1.8 **DEFINE** the following:
- Heat
 - Latent heat
 - Sensible heat
 - Unit used to measure heat

ENABLING OBJECTIVES (Cont.)

- 1.9 **DEFINE** the following thermodynamic properties:
- Specific enthalpy
 - Entropy
- 1.10 **DESCRIBE** the following types of thermodynamic systems:
- Isolated system
 - Closed system
 - Open system
- 1.11 **DEFINE** the following terms concerning thermodynamic systems:
- Thermodynamic surroundings
 - Thermodynamic equilibrium
 - Control volume
 - Steady-state
- 1.12 **DESCRIBE** the following terms concerning thermodynamic processes:
- Thermodynamic process
 - Cyclic process
 - Reversible process
 - Irreversible process
 - Adiabatic process
 - Isentropic process
 - Throttling process
 - Polytropic process
- 1.13 **DISTINGUISH** between intensive and extensive properties.
- 1.14 **DEFINE** the following terms:
- Saturation
 - Subcooled liquid
 - Superheated vapor
 - Critical Point
 - Triple Point
 - Vapor pressure curve
 - Quality
 - Moisture content
- 1.15 **DESCRIBE** the processes of sublimation, vaporization, condensation, and fusion.

ENABLING OBJECTIVES (Cont.)

- 1.16 Given a Mollier diagram and sufficient information to indicate the state of the fluid, **DETERMINE** any unknown properties for the fluid.
- 1.17 Given a set of steam tables and sufficient information to indicate the state of the fluid, **DETERMINE** any unknown properties for the fluid.
- 1.18 **DETERMINE** the change in the enthalpy of a fluid as it passes through a system component, given the state of the fluid at the inlet and outlet of the component and either steam tables or a Mollier diagram.
- 1.19 **STATE** the First Law of Thermodynamics.
- 1.20 Using the First Law of Thermodynamics, **ANALYZE** an open system including all energy transfer processes crossing the boundaries.
- 1.21 Using the First Law of Thermodynamics, **ANALYZE** cyclic processes for a thermodynamic system.
- 1.22 Given a defined system, **PERFORM** energy balances on all major components in the system.
- 1.23 Given a heat exchanger, **PERFORM** an energy balance across the two sides of the heat exchanger.
- 1.24 **IDENTIFY** the path(s) on a T-s diagram that represents the thermodynamic processes occurring in a fluid system.
- 1.25 **STATE** the Second Law of Thermodynamics.
- 1.26 Using the Second Law of Thermodynamics, **DETERMINE** the maximum possible efficiency of a system.
- 1.27 Given a thermodynamic system, **CONDUCT** an analysis using the Second Law of Thermodynamics.
- 1.28 Given a thermodynamic system, **DESCRIBE** the method used to determine:
- The maximum efficiency of the system
 - The efficiency of the components within the system

ENABLING OBJECTIVES (Cont.)

- 1.29 **DIFFERENTIATE** between the path for an ideal process and that for a real process on a T-s or h-s diagram.
- 1.30 Given a T-s or h-s diagram for a system **EVALUATE**:
- a. System efficiencies
 - b. Component efficiencies
- 1.31 **DESCRIBE** how individual factors affect system or component efficiency.
- 1.32 Apply the ideal gas laws to **SOLVE** for the unknown pressure, temperature, or volume.
- 1.33 **DESCRIBE** when a fluid may be considered to be incompressible.
- 1.34 **CALCULATE** the work done in constant pressure and constant volume processes.
- 1.35 **DESCRIBE** the effects of pressure changes on confined fluids.
- 1.36 **DESCRIBE** the effects of temperature changes on confined fluids.

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THERMODYNAMIC PROPERTIES

Thermodynamic properties describe measurable characteristics of a substance. A knowledge of these properties is essential to the understanding of thermodynamics.

EO 1.1 DEFINE the following properties:

- a. **Specific volume**
- b. **Density**
- c. **Specific gravity**
- d. **Humidity**

EO 1.2 DESCRIBE the following classifications of thermodynamic properties:

- a. **Intensive properties**
- b. **Extensive properties**

Mass and Weight

The *mass* (*m*) of a body is the measure of the amount of material present in that body. The *weight* (*wt*) of a body is the force exerted by that body when its mass is accelerated in a gravitational field. Mass and weight are related as shown in Equation 1-1.

$$wt = \frac{mg}{g_c} \quad (1-1)$$

where:

wt	=	weight (lbf)
m	=	mass (lbm)
g	=	acceleration of gravity = 32.17 ft/sec ²
g _c	=	gravitational constant = 32.17 lbf-ft/lbf-sec ²

Note that *g_c* has the same numerical value as the acceleration of gravity at sea level, but is not the acceleration of gravity. Rather, it is a dimensional constant employed to facilitate the use of Newton's Second Law of Motion with the English system of units.

The weight of a body is a force produced when the mass of the body is accelerated by a gravitational acceleration. The mass of a certain body will remain constant even if the gravitational acceleration acting upon that body changes.

According to Newton's Second Law of Motion, force (F) = ma , where a is acceleration. For example, on earth an object has a certain mass and a certain weight. When the same object is placed in outer space, away from the earth's gravitational field, its mass is the same, but it is now in a "weightless" condition (that is, gravitational acceleration and, thus, force equal zero).

The English system uses the pound-force (lbf) as the unit of weight. Knowing that acceleration has the units of ft/sec^2 and using Newton's second law, we can determine that the units of mass are $\text{lbf}\cdot\text{sec}^2/\text{ft}$. For simplification, $1 \text{ lbf}\cdot\text{sec}^2/\text{ft}$ is called a slug. The basic unit of mass in the English system is the slug. However, the slug is an almost meaningless unit for the average individual. The unit of mass generally used is the pound-mass (lbm). In order to allow lbm to be used as a unit of mass, we must divide Newton's second law by the gravitational constant (g_c).

$$\left(32.17 \frac{\text{lbm} \cdot \text{ft}}{\text{lbf} \cdot \text{sec}^2} \right) = g_c$$

Newton's second law can be expressed by Equation 1-2.

$$F = \frac{ma}{g_c} \quad (1-2)$$

Use of the gravitational constant, g_c , adapts Newton's second law such that $1 \text{ lbf} = 1 \text{ lbm}$ at the surface of the earth. It is important to note that this relationship is only true at the surface of the earth, where the acceleration due to gravity is $32.17 \text{ ft}/\text{sec}^2$. However, because all of our discussions will be based upon experiences and observations on earth, we will use the lbm as the unit of mass.

NOTE: In Equation 1-2, acceleration "a" is often written as "g" because, in this case, the acceleration is the gravitational acceleration due to the earth's gravitational field ($g = 32.17 \text{ ft}/\text{sec}^2$).

Example:

Using Equation 1-2, prove that $1 \text{ lbf} = 1 \text{ lbm}$ on earth.

Solution:

$$F = \frac{mg}{g_c}$$

$$1 \text{ lbf} = \frac{(1 \text{ lbm}) (32.17 \text{ ft}/\text{sec}^2)}{32.17 \frac{(\text{lbm} \cdot \text{ft})}{(\text{lbf} \cdot \text{sec}^2)}}$$

$$1 \text{ lbf} = 1 \text{ lbf} \text{ (an equality)}$$

Specific Volume

The *specific volume* (v) of a substance is the total volume (V) of that substance divided by the total mass (m) of that substance (volume per unit mass). It has units of cubic feet per pound-mass (ft^3/lbm).

$$v = \frac{V}{m} \quad (1-3)$$

where:

v = specific volume (ft^3/lbm)

V = volume (ft^3)

m = mass (lbm)

Density

The *density* (ρ) of a substance is the total mass (m) of that substance divided by the total volume (V) occupied by that substance (mass per unit volume). It has units of pound-mass per cubic feet (lbm/ft^3). The density (ρ) of a substance is the reciprocal of its specific volume (v).

$$\rho = \frac{m}{V} = \frac{1}{v} \quad (1-4)$$

where:

ρ = density (lbm/ft^3)

m = mass (lbm)

V = volume (ft^3)

v = specific volume (ft^3/lbm)

Specific Gravity

Specific gravity (S.G.) is a measure of the relative density of a substance as compared to the density of water at a standard temperature. Physicists use 39.2°F (4°C) as the standard, but engineers ordinarily use 60°F. In the International System of Units (SI Units), the density of water is 1.00 g/cm³ at the standard temperature. Therefore, the specific gravity (which is dimensionless) for a liquid has the same numerical value as its density in units of g/cm³. Since the density of a fluid varies with temperature, specific gravities must be determined and specified at particular temperatures.

Humidity

Humidity is the amount of moisture (water vapor) in the air. It can be expressed as absolute humidity or relative humidity. *Absolute humidity* is the mass of water vapor divided by a unit volume of air (grams of water/cm³ of air). *Relative humidity* is the amount of water vapor present in the air divided by the maximum amount that the air could contain at that temperature. Relative humidity is expressed as a percentage. The relative humidity is 100% if the air is saturated with water vapor and 0% if no water vapor is present in the air at all.

Intensive and Extensive Properties

Thermodynamic properties can be divided into two general classes, intensive and extensive properties. An *intensive property* is independent of the amount of mass. The value of an *extensive property* varies directly with the mass. Thus, if a quantity of matter in a given state is divided into two equal parts, each part will have the same value of intensive property as the original and half the value of the extensive property. Temperature, pressure, specific volume, and density are examples of intensive properties. Mass and total volume are examples of extensive properties.

Summary

The important information from this chapter is summarized below.

Thermodynamic Properties Summary

The following properties were defined:

- Specific volume (v) is the total volume (V) of a substance divided by the total mass (m) of that substance.
- Density (ρ) is the total mass (m) of a substance divided by the total volume (V) occupied by that substance.
- Specific gravity (S.G.) is a measure of the relative density of a substance as compared to the density of water at a standard temperature.
- Humidity is the amount of moisture (water vapor) in the air. It can be measured in absolute or relative units.

The following classifications of thermodynamic properties were described:

- Intensive properties are those that are independent of the amount of mass.
- Extensive properties are those that vary directly with the mass.

TEMPERATURE AND PRESSURE MEASUREMENTS

Several types of temperature and pressure measurements are used during discussions of thermodynamics. Operators must recognize the different types and their interrelationships in order to understand thermodynamics.

- EO 1.3** **DEFINE** the thermodynamic properties temperature and pressure.
- EO 1.4** **DESCRIBE** the Fahrenheit, Celsius, Kelvin, and Rankine temperature scales including:
- a. **Absolute zero temperature**
 - b. **The freezing point of water at atmospheric pressure**
 - c. **The boiling point of water at atmospheric pressure**
- EO 1.5** **CONVERT** temperatures between the Fahrenheit, Celsius, Kelvin, and Rankine scales.
- EO 1.6** **DESCRIBE** the relationship between absolute pressure, gauge pressure, and vacuum.
- EO 1.7** **CONVERT** pressures between the following units:
- a. **Pounds per square inch**
 - b. **Inches of water**
 - c. **Inches of mercury**
 - d. **Millimeters of mercury**
 - e. **Microns of mercury**
-

Temperature

Temperature is a measure of the molecular activity of a substance. The greater the movement of molecules, the higher the temperature. It is a relative measure of how "hot" or "cold" a substance is and can be used to predict the direction of heat transfer.

Temperature Scales

The two temperature scales normally employed for measurement purposes are the Fahrenheit (F) and Celsius (C) scales. These scales are based on a specification of the number of increments between the freezing point and boiling point of water at standard atmospheric pressure. The Celsius scale has 100 units between these points, and the Fahrenheit scale has 180 units. The zero points on the scales are arbitrary.

The freezing point of water was selected as the zero point of the Celsius scale. The coldest temperature achievable with a mixture of ice and salt water was selected as the zero point of the Fahrenheit scale. The temperature at which water boils was set at 100 on the Celsius scale and 212 on the Fahrenheit scale. The relationship between the scales is represented by the following equations.

$$^{\circ}\text{F} = 32.0 + (9/5)^{\circ}\text{C} \tag{1-5}$$

$$^{\circ}\text{C} = (^{\circ}\text{F} - 32.0)(5/9) \tag{1-6}$$

It is necessary to define an absolute temperature scale having only positive values. The absolute temperature scale that corresponds to the Celsius scale is called the Kelvin (K) scale, and the absolute scale that corresponds to the Fahrenheit scale is called the Rankine (R) scale. The zero points on both absolute scales represent the same physical state. This state is where there is no molecular motion of individual atoms. The relationships between the absolute and relative temperature scales are shown in the following equations.

$$^{\circ}\text{R} = ^{\circ}\text{F} + 460 \tag{1-7}$$

$$^{\circ}\text{K} = ^{\circ}\text{C} + 273 \tag{1-8}$$

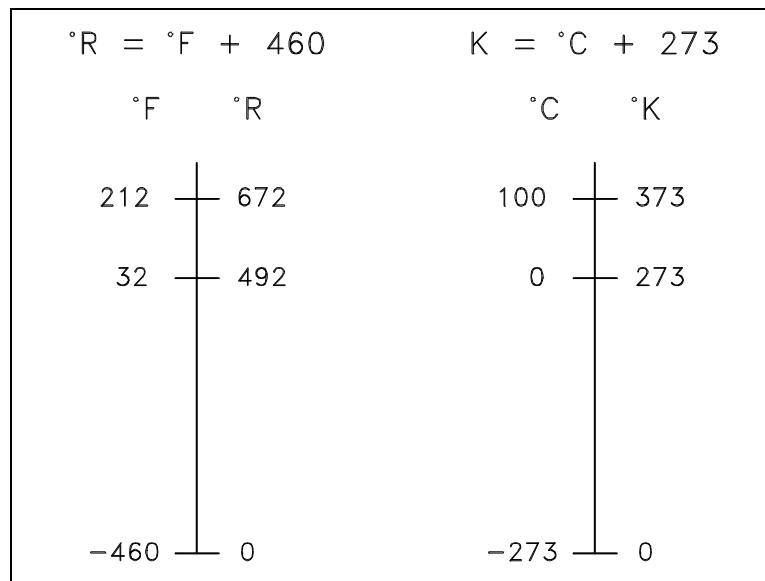


Figure 1 Comparison of Temperature Scales

The conversion of one temperature scale to another is sometimes required at nuclear facilities, and the operator should be acquainted with the process. The following two examples will be helpful.

Example 1: Temperature Scale Conversion

What is the Rankine equivalent of 80°C?

Solution:

$$^{\circ}\text{F} = (9/5) ^{\circ}\text{C} + 32$$

$$= (9/5)(80) + 32$$

$$= 176 ^{\circ}\text{F}$$

$$^{\circ}\text{R} = ^{\circ}\text{F} + 460$$

$$= 176 + 460$$

$$= 636 ^{\circ}\text{R}$$

Example 2: Temperature Scale Conversion

What is the Kelvin equivalent of 80°F?

Solution:

$$^{\circ}\text{C} = (5/9) (^{\circ}\text{F} - 32)$$

$$= (5/9) (80 - 32)$$

$$= 26.7^{\circ}\text{C}$$

$$^{\circ}\text{K} = ^{\circ}\text{C} + 273$$

$$= 26.7 + 273$$

$$= 299.7 ^{\circ}\text{K}$$

Pressure

Pressure is a measure of the force exerted per unit area on the boundaries of a substance (or system). It is caused by the collisions of the molecules of the substance with the boundaries of the system. As molecules hit the walls, they exert forces that try to push the walls outward. The forces resulting from all of these collisions cause the pressure exerted by a system on its surroundings. Pressure is frequently measured in units of lbf/in² (psi).

Pressure Scales

When pressure is measured relative to a perfect vacuum, it is called absolute pressure (psia); when measured relative to atmospheric pressure (14.7 psi), it is called gauge pressure (psig). The latter pressure scale was developed because almost all pressure gauges register zero when open to the atmosphere. Therefore, pressure gauges measure the difference between the pressure of the fluid to which they are connected and that of the surrounding air.

If the pressure is below that of the atmosphere, it is designated as a vacuum. A perfect vacuum would correspond to absolute zero pressure. All values of absolute pressure are positive, because a negative value would indicate tension, which is considered impossible in any fluid. Gauge pressures are positive if they are above atmospheric pressure and negative if they are below atmospheric pressure. Figure 2 shows the relationships between absolute, gauge, vacuum, and atmospheric pressures, as do Equations 1-9 and 1-10.

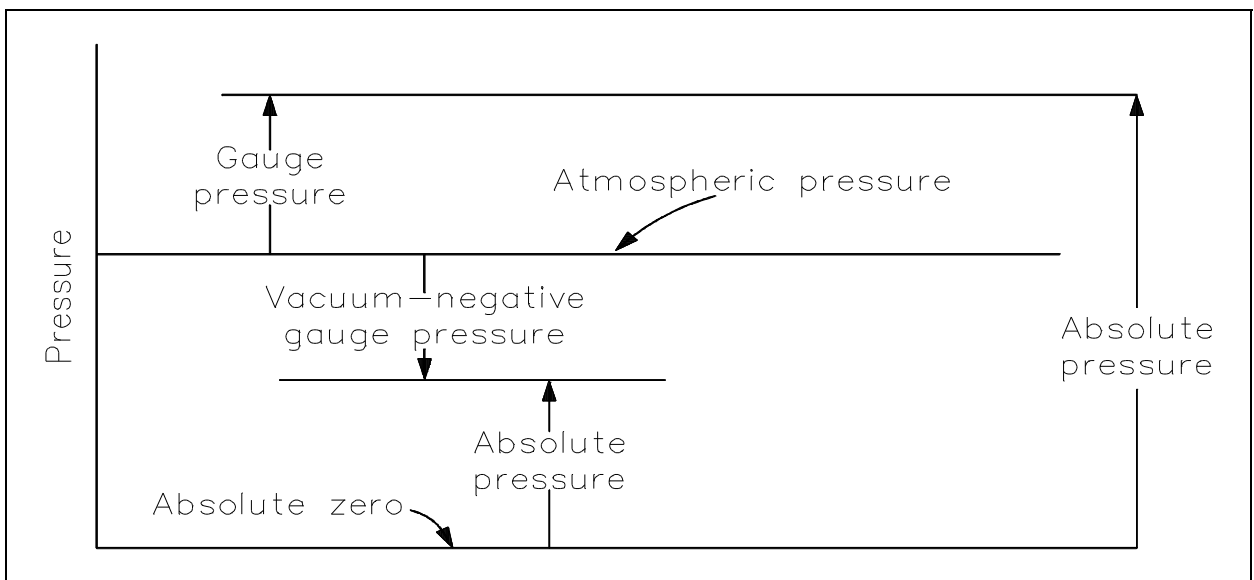


Figure 2 Pressure Relationships

$$P_{\text{abs}} = P_{\text{atm}} + P_{\text{gauge}} \quad (1-9)$$

$$P_{\text{abs}} = P_{\text{atm}} - P_{\text{vac}} \quad (1-10)$$

P_{atm} is atmospheric pressure, which is also called the barometric pressure. P_{gauge} is the gauge pressure, and P_{vac} is vacuum. Once again, the following examples relating the various pressures will be helpful in understanding the idea of gauge versus absolute pressures.

Example 1: Pressure Relationships

How deep can a diver descend in ocean water (density = 64 lbm/ft³) without damaging his watch, which will withstand an absolute pressure of 80 psia? ($P = \text{density} \cdot \text{height}$)

Solution:

$$\text{Assume: } P_{\text{atm}} = 14.7 \text{ psia}$$

$$P_{\text{abs}} = P_{\text{atm}} + P_{\text{gauge}}$$

$$80 \text{ psia} = 14.7 + P_{\text{gauge}}$$

$$P_{\text{gauge}} = (80 - 14.7) = 65.3 \text{ psig}$$

$$P_{\text{gauge}} = \text{density} \cdot \text{height} = \rho H$$

$$(65.3)(144 \text{ in}^2/\text{ft}^2) = (64 \text{ lbm}/\text{ft}^3)H$$

$$H = (65.3)(144)/(64)$$

$$H = 146.9 \text{ ft}$$

Example 2: Pressure Relationships

What is the absolute pressure at the bottom of a swimming pool 6 feet deep that is filled with fresh water? $P_{\text{atm}} = 14.7$ psia

Solution:

$$\begin{aligned}P_{\text{abs}} &= P_{\text{atm}} + P_{\text{gauge}} \\&= 14.7 + \rho H \\&= 14.7 + [(62.4 \text{ lbm/ft}^3)(6 \text{ ft})/(144 \text{ in.}^2/\text{ft}^2)] \\&= 14.7 + 2.6 \\P_{\text{abs}} &= 17.3 \text{ psia}\end{aligned}$$

In addition to pounds per square inch, pressure can be measured with reference to the force that exists in a column of fluid at a certain height. The most common of these are inches of water, inches of mercury, millimeters of mercury, and microns of mercury. Conversion factors are listed below.

$$14.7 \text{ psia} = 408 \text{ inches of water}$$

$$14.7 \text{ psia} = 29.9 \text{ inches of mercury}$$

$$1 \text{ inch of mercury} = 25.4 \text{ millimeters of mercury}$$

$$1 \text{ millimeter of mercury} = 10^3 \text{ microns of mercury}$$

Summary

The important information from this chapter is summarized below.

Temperature and Pressure Scales Summary

The following properties were defined as follows.

- Temperature is a measure of the molecular activity of a substance.
- Pressure is a measure of the force per unit area exerted on the boundaries of a substance (or system).

The relationship between the Fahrenheit, Celsius, Kelvin, and Rankine temperature scales was described.

- Absolute zero = $-460\text{ }^{\circ}\text{F}$ or $-273\text{ }^{\circ}\text{C}$
- Freezing point of water = $32\text{ }^{\circ}\text{F}$ or $0\text{ }^{\circ}\text{C}$
- Boiling point of water = $212\text{ }^{\circ}\text{F}$ or $100\text{ }^{\circ}\text{C}$

Conversions between the different scales can be made using the following formulas.

- $^{\circ}\text{F} = 32 + (9/5)^{\circ}\text{C}$
- $^{\circ}\text{C} = (^{\circ}\text{F} - 32)(5/9)$
- $^{\circ}\text{R} = ^{\circ}\text{F} + 460$
- $^{\circ}\text{K} = ^{\circ}\text{C} + 273$

Relationships between absolute pressure, gauge pressure, and vacuum can be shown using the following formulas.

- $P_{\text{abs}} = P_{\text{atm}} + P_{\text{gauge}}$
- $P_{\text{abs}} = P_{\text{atm}} - P_{\text{vac}}$

Temperature and Pressure Scales Summary (Cont.)

Converting between the different pressure units can be done using the following conversions.

- 14.7 psia = 408 inches of water
- 14.7 psia = 29.9 inches of mercury
- 1 inch of mercury = 25.4 millimeters of mercury
- 1 millimeter of mercury = 10^3 microns of mercury

ENERGY, WORK, AND HEAT

Heat and work are the two ways in which energy can be transferred across the boundary of a system. One of the most important discoveries in thermodynamics was that work could be converted into an equivalent amount of heat and that heat could be converted into work.

- EO 1.8 DEFINE the following:**
- a. Heat**
 - b. Latent heat**
 - c. Sensible heat**
 - d. Units used to measure heat**

- EO 1.9 DEFINE the following thermodynamic properties:**
- a. Specific enthalpy**
 - b. Entropy**
-

Energy

Energy is defined as the capacity of a system to perform work or produce heat.

Potential Energy

Potential energy (PE) is defined as the energy of position. Using English system units, it is defined by Equation 1-11.

$$PE = \frac{mgz}{g_c} \quad (1-11)$$

where:

- PE = potential energy (ft-lbf)
- m = mass (lbm)
- z = height above some reference level (ft)
- g = acceleration due to gravity (ft/sec²)
- g_c = gravitational constant = 32.17 ft-lbm/lbf-sec²

In most practical engineering calculations, the acceleration due to gravity (g) is numerically equal to the gravitational constant (g_c); thus, the potential energy (PE) in foot-pounds-force is numerically equal to the product of the mass (m) in pounds-mass times the height (z) in feet above some reference level.

Example:

Determine the potential energy of 50 lbm of water in a storage tank 100 ft above the ground.

Solution:

Using Equation 1-11

$$PE = \frac{mgz}{g_c}$$

$$PE = \frac{(50 \text{ lbm}) (32.17 \text{ ft/sec}^2) (100 \text{ ft})}{32.17 \text{ ft-lbm/lbf-sec}^2}$$

$$PE = 5000 \text{ ft-lbf}$$

Kinetic Energy

Kinetic energy (KE) is the energy of motion. Using English system units, it is defined by Equation 1-12.

$$KE = \frac{mv^2}{2g_c} \quad (1-12)$$

where:

KE = kinetic energy (ft-lbf)

m = mass (lbm)

v = velocity (ft/sec)

g_c = gravitational constant = 32.17 ft-lbm/lbf-sec²

Example:

Determine the kinetic energy of 7 lbm of steam flowing through a pipe at a velocity of 100 ft/sec.

Solution:

Using Equation 1-12.

$$KE = \frac{mv^2}{2g_c}$$

$$KE = \frac{(7 \text{ lbm}) (100 \text{ ft/sec})^2}{2(32.17 \text{ ft-lbm/lbf-sec}^2)}$$

$$KE = \frac{(7 \text{ lbm}) (10,000 \text{ ft}^2/\text{sec}^2)}{(64.34 \text{ ft-lbm/lbf-sec}^2)}$$

$$KE = 1088 \text{ ft-lbf}$$

Specific Internal Energy

Potential energy and kinetic energy are macroscopic forms of energy. They can be visualized in terms of the position and the velocity of objects. In addition to these macroscopic forms of energy, a substance possesses several microscopic forms of energy. Microscopic forms of energy include those due to the rotation, vibration, translation, and interactions among the molecules of a substance. None of these forms of energy can be measured or evaluated directly, but techniques have been developed to evaluate the change in the total sum of all these microscopic forms of energy. These microscopic forms of energy are collectively called *internal energy*, customarily represented by the symbol U . In engineering applications, the unit of internal energy is the *British thermal unit* (Btu), which is also the unit of heat.

The specific internal energy (u) of a substance is its internal energy per unit mass. It equals the total internal energy (U) divided by the total mass (m).

$$u = \frac{U}{m} \quad (1-13)$$

where:

u = specific internal energy (Btu/lbm)

U = internal energy (Btu)

m = mass (lbm)

Example:

Determine the specific internal energy of 12 lbm of steam if the total internal energy is 23,000 Btu.

Solution:

Using Equation 1-13.

$$u = \frac{U}{m}$$

$$u = \frac{23,000 \text{ Btu}}{12 \text{ lbm}}$$

$$u = 1916.67 \text{ Btu/lbm}$$

Specific P-V Energy

In addition to the internal energy (U), another form of energy exists that is important in understanding energy transfer systems. This form of energy is called P-V energy because it arises from the pressure (P) and the volume (V) of a fluid. It is numerically equal to PV, the product of pressure and volume. Because energy is defined as the capacity of a system to perform work, a system where pressure and volume are permitted to expand performs work on its surroundings. Therefore, a fluid under pressure has the capacity to perform work. In engineering applications, the units of P-V energy, also called flow energy, are the units of pressure times volume (pounds-force per square foot times cubic feet), which equals foot-pounds force (ft-lbf).

The specific P-V energy of a substance is the P-V energy per unit mass. It equals the total P-V divided by the total mass m, or the product of the pressure P and the specific volume v, and is written as Pv.

$$Pv = \frac{PV}{m} \tag{1-14}$$

where:

P = pressure (lbf/ft²)

V = volume (ft³)

v = specific volume (ft³/lbm) = $\frac{V}{m}$

m = mass (lbm)

Example:

Determine the specific P-V energy of 15 lbm of steam at 1000 psi in an 18 ft³ tank.

Solution:

Using Equation 1-14

$$Pv = \frac{PV}{m}$$

$$Pv = \frac{(1000 \text{ lbf/in.}^2) (144 \text{ in.}^2/\text{ft}^2) (18 \text{ ft}^3)}{15 \text{ lbm}}$$

$$Pv = 172,800 \text{ ft-lbf/lbm}$$

Specific Enthalpy

Specific enthalpy (h) is defined as $h = u + Pv$, where u is the specific internal energy (Btu/lbm) of the system being studied, P is the pressure of the system (lbf/ft²), and v is the specific volume (ft³/lbm) of the system. Enthalpy is usually used in connection with an "open" system problem in thermodynamics. Enthalpy is a property of a substance, like pressure, temperature, and volume, but it cannot be measured directly. Normally, the enthalpy of a substance is given with respect to some reference value. For example, the specific enthalpy of water or steam is given using the reference that the specific enthalpy of water is zero at .01°C and normal atmospheric pressure. The fact that the absolute value of specific enthalpy is unknown is not a problem, however, because it is the change in specific enthalpy (Δh) and not the absolute value that is important in practical problems. Steam tables include values of enthalpy as part of the information tabulated.

Work

Kinetic energy, potential energy, internal energy, and P-V energy are forms of energy that are properties of a system. *Work* is a form of energy, but it is energy in transit. Work is not a property of a system. Work is a process done by or on a system, but a system contains no work. This distinction between the forms of energy that are properties of a system and the forms of energy that are transferred to and from a system is important to the understanding of energy transfer systems.

Work is defined for mechanical systems as the action of a force on an object through a distance. It equals the product of the force (F) times the displacement (d).

$$W = Fd \quad (1-15)$$

where:

W = work (ft-lbf)

F = force (lbf)

d = displacement (ft)

Example:

Determine the amount of work done if a force of 150 lbf is applied to an object until it has moved a distance of 30 feet.

Solution:

Using Equation 1-15

$$W = Fd$$

$$W = (150 \text{ lbf})(30 \text{ ft})$$

$$W = 4500 \text{ ft-lbf}$$

In dealing with work in relation to energy transfer systems, it is important to distinguish between work done by the system on its surroundings and work done on the system by its surroundings. Work is done by the system when it is used to turn a turbine and thereby generate electricity in a turbine-generator. Work is done on the system when a pump is used to move the working fluid from one location to another. A positive value for work indicates that work is done by the system on its surroundings; a negative value indicates that work is done on the system by its surroundings.

Heat

Heat, like work, is energy in transit. The transfer of energy as heat, however, occurs at the molecular level as a result of a temperature difference. The symbol Q is used to denote heat. In engineering applications, the unit of heat is the British thermal unit (Btu). Specifically, this is called the 60 degree Btu because it is measured by a one degree temperature change from 59.5 to 60.5°F.

As with work, the amount of heat transferred depends upon the path and not simply on the initial and final conditions of the system. Also, as with work, it is important to distinguish between heat added to a system from its surroundings and heat removed from a system to its surroundings. A positive value for heat indicates that heat is added to the system by its surroundings. This is in contrast to work that is positive when energy is transferred from the system and negative when transferred to the system. The symbol q is sometimes used to indicate the heat added to or removed from a system per unit mass. It equals the total heat (Q) added or removed divided by the mass (m). The term "specific heat" is not used for q since specific heat is used for another parameter. The quantity represented by q is referred to simply as the heat transferred per unit mass.

$$q = \frac{Q}{m} \quad (1-16)$$

where:

q = heat transferred per unit mass (Btu/lbm)

Q = heat transferred (Btu)

m = mass (lbm)

Example:

Determine the heat transferred per unit mass if 1500 Btu's are transferred to 40 lbm of water.

Solution:

Using Equation 1-16

$$q = \frac{Q}{m}$$

$$q = \frac{1500 \text{ Btu}}{40 \text{ lbm}}$$

$$q = 37.5 \text{ Btu/lbm}$$

The best way to quantify the definition of heat is to consider the relationship between the amount of heat added to or removed from a system and the change in the temperature of the system. Everyone is familiar with the physical phenomena that when a substance is heated, its temperature increases, and when it is cooled, its temperature decreases. The heat added to or removed from a substance to produce a change in its temperature is called *sensible heat*. The units of heat are often defined in terms of the changes in temperature it produces.

Another type of heat is called *latent heat*. Latent heat is the amount of heat added to or removed from a substance to produce a change in phase. When latent heat is added, no temperature change occurs. There are two types of latent heat. The first is the *latent heat of fusion*. This is the amount of heat added or removed to change phase between solid and liquid. The second type of latent heat is the *latent heat of vaporization*. This is the amount of heat added or removed to change phase between liquid and vapor. The latent heat of vaporization is sometimes called the latent heat of condensation.

Different substances are affected to different magnitudes by the addition of heat. When a given amount of heat is added to different substances, their temperatures increase by different amounts. The ratio of the heat (Q) added to or removed from a substance to the change in temperature (ΔT) produced is called the *heat capacity* (C_p) of the substance. The heat capacity of a substance per unit mass is called the *specific heat* (c_p) of the substance. The subscript p indicates that the heat capacity and specific heat apply when the heat is added or removed at constant pressure.

$$C_p = \frac{Q}{\Delta T}$$

$$c_p = \frac{Q}{m\Delta T} \quad (1-17)$$

$$c_p = \frac{q}{\Delta T}$$

where:

C_p = heat capacity at constant pressure (Btu/°F)

c_p = specific heat at constant pressure (Btu/lbm-°F)

Q = heat transferred (Btu)

q = heat transferred per unit mass (Btu/lbm)

m = mass (lbm)

ΔT = temperature change (°F)

One lbm of water is raised 1°F and one Btu of heat is added. This implies that the specific heat (c_p) of water is one Btu/lbm-°F. The c_p of water is equal to one Btu/lbm-°F only at 39.1°F.

By rearranging Equation 1-17 we obtain $Q = mc_p\Delta T$, which is used to calculate latent heat. By substituting mass flow rate in lbm/hr, \dot{m} , for m , we obtain $\dot{Q} = \dot{m}c_p\Delta T$. This equation is used to calculate heat transfer in Btu/hr and will be useful in later chapters.

Example:

How much heat is required to raise the temperature of 5 lbm of water from 50°F to 150°F? (Assume the specific heat (c_p) for water is constant at 1.0 Btu/lbm-°F.)

Solution:

$$c_p = \frac{Q}{m\Delta T}$$

$$Q = c_p m \Delta T$$

$$Q = (1.0 \text{ Btu/lbm-}^\circ\text{F})(5 \text{ lbm})(150^\circ\text{F} - 50^\circ\text{F})$$

$$Q = (1.0 \text{ Btu/lbm-}^\circ\text{F})(5 \text{ lbm})(100^\circ\text{F})$$

$$Q = 500 \text{ Btu}$$

From the previous discussions on heat and work, it is evident that there are many similarities between them. Heat and work are both transient phenomena. Systems never possess heat or work, but either or both may occur when a system undergoes a change of energy state. Both heat and work are boundary phenomena in that both are observed at the boundary of the system. Both represent energy crossing the system boundary.

Entropy

Entropy (S) is a property of a substance, as are pressure, temperature, volume, and enthalpy. Because entropy is a property, changes in it can be determined by knowing the initial and final conditions of a substance. Entropy quantifies the energy of a substance that is no longer available to perform useful work. Because entropy tells so much about the usefulness of an amount of heat transferred in performing work, the steam tables include values of specific entropy ($s = S/m$) as part of the information tabulated. Entropy is sometimes referred to as a measure of the inability to do work for a given heat transferred. Entropy is represented by the letter S and can be defined as ΔS in the following relationships.

$$\Delta S = \frac{\Delta Q}{T_{\text{abs}}} \quad (1-18)$$

$$\Delta s = \frac{\Delta q}{T_{\text{abs}}} \quad (1-19)$$

where:

$$\Delta S = \quad \text{the change in entropy of a system during some process (Btu/}^\circ\text{R)}$$

ΔQ	=	the amount of heat transferred to or from the system during the process (Btu)
T_{abs}	=	the absolute temperature at which the heat was transferred ($^{\circ}\text{R}$)
Δs	=	the change in specific entropy of a system during some process (Btu/lbm $^{-\circ}\text{R}$)
Δq	=	the amount of heat transferred to or from the system during the process (Btu/lbm)

Like enthalpy, entropy cannot be measured directly. Also, like enthalpy, the entropy of a substance is given with respect to some reference value. For example, the specific entropy of water or steam is given using the reference that the specific entropy of water is zero at 32°F . The fact that the absolute value of specific entropy is unknown is not a problem because it is the change in specific entropy (Δs) and not the absolute value that is important in practical problems.

Energy and Power Equivalences

The various forms of energy involved in energy transfer systems (such as potential energy, kinetic energy, internal energy, P-V energy, work and heat) may be measured in numerous basic units. In general, three types of units are used to measure energy: (1) mechanical units, such as the foot-pound-force (ft-lbf); (2) thermal units, such as the British thermal unit (Btu); and (3) electrical units, such as the watt-second (W-sec). In the mks and cgs systems, the mechanical units of energy are the joule (j) and the erg, the thermal units are the kilocalorie (kcal) and the calorie (cal), and the electrical units are the watt-second (W-sec) and the erg. Although the units of the various forms of energy are different, they are equivalent.

Some of the most important experiments in science were those conducted by J. P. Joule in 1843, who showed quantitatively that there was a direct correspondence between mechanical and thermal energy. These experiments showed that one kilocalorie equals 4,186 joules. These same experiments, when performed using English system units, show that one British thermal unit (Btu) equals 778.3 ft-lbf. These experiments established the equivalence of mechanical and thermal energy. Other experiments established the equivalence of electrical energy with both mechanical and thermal energy. For engineering applications, these equivalences are expressed by the following relationships.

$$1 \text{ ft-lbf} = 1.286 \times 10^{-3} \text{ Btu} = 3.766 \times 10^{-7} \text{ kW-hr}$$

$$1 \text{ Btu} = 778.3 \text{ ft-lbf} = 2.928 \times 10^{-4} \text{ kW-hr}$$

$$1 \text{ kW-hr} = 3.413 \times 10^3 \text{ Btu} = 2.655 \times 10^6 \text{ ft-lbf}$$

There is one additional unit of energy encountered in engineering applications. It is the horsepower-hour (hp-hr). It is a mechanical unit of energy defined by the following relationship:

$$1 \text{ hp-hr} = 1.980 \times 10^6 \text{ ft-lbf}$$

These relationships can be used to convert between the various English system units for the various forms of energy.

Most computations involving the energy of the working fluid in an energy transfer system are performed in Btu's. Forms of mechanical energy (such as potential energy, kinetic energy, and mechanical work) and other forms of energy (such as P-V energy) are usually given in foot-pounds-force. These are converted to Btu's by using $1 \text{ Btu} = 778.3 \text{ ft-lbf}$.

This conversion factor is often used. In fact, a constant called the mechanical equivalent of heat, usually denoted by the symbol J and sometimes referred to as *Joule's constant*, is defined as:

$$J = 778 \frac{\text{ft-lbf}}{\text{Btu}}.$$

Power is defined as the time rate of doing work. It is equivalent to the rate of the energy transfer. Power has units of energy per unit time. As with energy, power may be measured in numerous basic units, but the units are equivalent. In the English system, the mechanical units of power are foot-pounds-force per second or per hour (ft-lbf/sec or ft-lbf/hr) and horsepower (hp). The thermal units of power are British thermal units per hour (Btu/hr), and the electrical units of power are watts (W) or kilowatts (kW). For engineering applications, the equivalence of these units is expressed by the following relationships.

$$1 \text{ ft-lbf/sec} = 4.6263 \text{ Btu/hr} = 1.356 \times 10^{-3} \text{ kW}$$

$$1 \text{ Btu/hr} = 0.2162 \text{ ft-lbf/sec} = 2.931 \times 10^{-4} \text{ kW}$$

$$1 \text{ kW} = 3.413 \times 10^3 \text{ Btu/hr} = 737.6 \text{ ft-lbf/sec}$$

Horsepower is related to foot-pounds-force per second (ft-lbf/sec) by the following relationship:

$$1 \text{ hp} = 550.0 \text{ ft-lbf/sec}$$

These relationships can be used to convert the English system units for power.

Summary

The important information from this chapter is summarized below.

Energy, Work, and Heat Summary

- Heat is described as energy in transit. This transfer occurs on a molecular level as a result of temperature differences. The unit of heat is the British thermal unit (Btu).

Latent heat = the amount of heat added or removed to produce only a phase change.

Sensible heat = the heat added or removed that causes a temperature change.

- The following properties were defined:

Specific enthalpy (h) is defined as $h = u + Pv$, where u is the specific internal energy (Btu/lbm) of the system being studied, P is the pressure of the system (lbf/ft²), and v is the specific volume (ft³/lbm) of the system.

Entropy is sometimes referred to as a measure of the inability to do work for a given heat transferred.

THERMODYNAMIC SYSTEMS AND PROCESSES

Defining an appropriate system can greatly simplify a thermodynamic analysis. A thermodynamic system is any three-dimensional region of space that is bounded by one or more surfaces. The bounding surfaces may be real or imaginary and may be at rest or in motion. The boundary may change its size or shape. The region of physical space that lies outside the selected boundaries of the system is called the surroundings or the environment.

EO 1.10 **DESCRIBE the following types of thermodynamic systems:**

- a. **Isolated system**
- b. **Closed system**
- c. **Open system**

EO 1.11 **DEFINE the following terms concerning thermodynamic systems:**

- a. **Thermodynamic surroundings**
- b. **Thermodynamic equilibrium**
- c. **Control volume**
- d. **Steady-state**

EO 1.12 **DESCRIBE the following terms concerning thermodynamic processes:**

- a. **Thermodynamic process**
- b. **Cyclic process**
- c. **Reversible process**
- d. **Irreversible process**
- e. **Adiabatic process**
- f. **Isentropic process**
- g. **Throttling process**
- h. **Polytropic process**

Thermodynamic Systems and Surroundings

Thermodynamics involves the study of various systems. A *system* in thermodynamics is nothing more than the collection of matter that is being studied. A system could be the water within one side of a heat exchanger, the fluid inside a length of pipe, or the entire lubricating oil system for a diesel engine. Determining the boundary to solve a thermodynamic problem for a system will depend on what information is known about the system and what question is asked about the system.

Everything external to the system is called the *thermodynamic surroundings*, and the system is separated from the surroundings by the *system boundaries*. These boundaries may either be fixed or movable. In many cases, a thermodynamic analysis must be made of a device, such as a heat exchanger, that involves a flow of mass into and/or out of the device. The procedure that is followed in such an analysis is to specify a control surface, such as the heat exchanger tube walls. Mass, as well as heat and work (and momentum), may flow across the control surface.

Types of Thermodynamic Systems

Systems in thermodynamics are classified as isolated, closed, or open based on the possible transfer of mass and energy across the system boundaries. An *isolated system* is one that is not influenced in any way by the surroundings. This means that no energy in the form of heat or work may cross the boundary of the system. In addition, no mass may cross the boundary of the system.

A thermodynamic system is defined as a quantity of matter of fixed mass and identity upon which attention is focused for study. A *closed system* has no transfer of mass with its surroundings, but may have a transfer of energy (either heat or work) with its surroundings.

An *open system* is one that may have a transfer of both mass and energy with its surroundings.

Thermodynamic Equilibrium

When a system is in equilibrium with regard to all possible changes in state, the system is in *thermodynamic equilibrium*. For example, if the gas that comprises a system is in thermal equilibrium, the temperature will be the same throughout the entire system.

Control Volume

A *control volume* is a fixed region in space chosen for the thermodynamic study of mass and energy balances for flowing systems. The boundary of the control volume may be a real or imaginary envelope. The *control surface* is the boundary of the control volume.

Steady State

Steady state is that circumstance in which there is no accumulation of mass or energy within the control volume, and the properties at any point within the system are independent of time.

Thermodynamic Process

Whenever one or more of the properties of a system change, a change in the state of the system occurs. The path of the succession of states through which the system passes is called the *thermodynamic process*. One example of a thermodynamic process is increasing the temperature of a fluid while maintaining a constant pressure. Another example is increasing the pressure of a confined gas while maintaining a constant temperature. Thermodynamic processes will be discussed in more detail in later chapters.

Cyclic Process

When a system in a given initial state goes through a number of different changes in state (going through various processes) and finally returns to its initial values, the system has undergone a *cyclic process or cycle*. Therefore, at the conclusion of a cycle, all the properties have the same value they had at the beginning. Steam (water) that circulates through a closed cooling loop undergoes a cycle.

Reversible Process

A *reversible process* for a system is defined as a process that, once having taken place, can be reversed, and in so doing leaves no change in either the system or surroundings. In other words the system and surroundings are returned to their original condition before the process took place. In reality, there are no truly reversible processes; however, for analysis purposes, one uses reversible to make the analysis simpler, and to determine maximum theoretical efficiencies. Therefore, the reversible process is an appropriate starting point on which to base engineering study and calculation.

Although the reversible process can be approximated, it can never be matched by real processes. One way to make real processes approximate reversible process is to carry out the process in a series of small or infinitesimal steps. For example, heat transfer may be considered reversible if it occurs due to a small temperature difference between the system and its surroundings. For example, transferring heat across a temperature difference of 0.00001 °F "appears" to be more reversible than for transferring heat across a temperature difference of 100 °F. Therefore, by cooling or heating the system in a number of infinitesimally small steps, we can approximate a reversible process. Although not practical for real processes, this method is beneficial for thermodynamic studies since the rate at which processes occur is not important.

Irreversible Process

An *irreversible process* is a process that cannot return both the system and the surroundings to their original conditions. That is, the system and the surroundings would not return to their

original conditions if the process was reversed. For example, an automobile engine does not give back the fuel it took to drive up a hill as it coasts back down the hill.

There are many factors that make a process irreversible. Four of the most common causes of irreversibility are friction, unrestrained expansion of a fluid, heat transfer through a finite temperature difference, and mixing of two different substances. These factors are present in real, irreversible processes and prevent these processes from being reversible.

Adiabatic Process

An *adiabatic process* is one in which there is no heat transfer into or out of the system. The system can be considered to be perfectly insulated.

Isentropic Process

An *isentropic process* is one in which the entropy of the fluid remains constant. This will be true if the process the system goes through is reversible and adiabatic. An isentropic process can also be called a constant entropy process.

Polytropic Process

When a gas undergoes a reversible process in which there is heat transfer, the process frequently takes place in such a manner that a plot of the Log P (pressure) vs. Log V (volume) is a straight line. Or stated in equation form $PV^n = \text{a constant}$. This type of process is called a *polytropic process*. An example of a polytropic process is the expansion of the combustion gasses in the cylinder of a water-cooled reciprocating engine.

Throttling Process

A *throttling process* is defined as a process in which there is no change in enthalpy from state one to state two, $h_1 = h_2$; no work is done, $W = 0$; and the process is adiabatic, $Q = 0$. To better understand the theory of the ideal throttling process let's compare what we can observe with the above theoretical assumptions.

An example of a throttling process is an ideal gas flowing through a valve in midposition. From experience we can observe that: $P_{in} > P_{out}$, $vel_{in} < vel_{out}$ (where P = pressure and vel = velocity). These observations confirm the theory that $h_{in} = h_{out}$. Remember $h = u + Pv$ (v = specific volume), so if pressure decreases then specific volume must increase if enthalpy is to remain constant (assuming u is constant). Because mass flow is constant, the change in specific volume is observed as an increase in gas velocity, and this is verified by our observations.

The theory also states $W = 0$. Our observations again confirm this to be true as clearly no "work" has been done by the throttling process. Finally, the theory states that an ideal throttling process is adiabatic. This cannot clearly be proven by observation since a "real" throttling process is not ideal and will have some heat transfer.

Summary

The important information from this chapter is summarized below.

Thermodynamic Systems and Processes Summary

- A thermodynamic system is a collection of matter and space with its boundaries defined in such a way that the energy transfer across the boundaries can be best understood.
- Surroundings are everything not in the system being studied.
- Systems are classified into one of three groups:
 - Isolated system - neither mass nor energy can cross the boundaries
 - Closed system - only energy can cross the boundaries
 - Open system - both mass and energy can cross the boundaries
- A control volume is a fixed region of space that is studied as a thermodynamic system.
- Steady state refers to a condition where the properties at any given point within the system are constant over time. Neither mass nor energy are accumulating within the system.
- A thermodynamic process is the succession of states that a system passes through. Processes can be described by any of the following terms:
 - Cyclic process - a series of processes that results in the system returning to its original state
 - Reversible process - a process that can be reversed resulting in no change in the system or surroundings
 - Irreversible process - a process that, if reversed, would result in a change to the system or surroundings
 - Adiabatic process - a process in which there is no heat transfer across the system boundaries
 - Isentropic process - a process in which the entropy of the system remains unchanged
 - Polytropic process - the plot of $\log P$ vs. $\log V$ is a straight line, $PV^n = \text{constant}$
 - Throttling process - a process in which enthalpy is constant $h_1 = h_2$, work = 0, and which is adiabatic, $Q=0$.

CHANGE OF PHASE

The phase change of materials in a system is very important to thermodynamics. It is possible to design systems to take advantage of the phase changes between solid and liquid or between liquid and vapor to enhance the performance of the system.

EO 1.13 DISTINGUISH between intensive and extensive properties.

EO 1.14 DEFINE the following terms:

- a. **Saturation**
- b. **Subcooled liquid**
- c. **Superheated vapor**
- d. **Critical Point**
- e. **Triple Point**
- f. **Vapor pressure curve**
- g. **Quality**
- h. **Moisture Content**

EO 1.15 DESCRIBE the processes of sublimation, vaporization, condensation, and fusion.

Classification of Properties

As discussed earlier in this module, properties are classified as either intensive or extensive. Properties are *intensive* if independent of the amount of mass present and *extensive* if a function of the amount of mass present. Properties such as pressure, temperature, and density are intensive, whereas volume and mass are extensive. An extensive property may be made intensive by dividing the particular property by the total mass. Total volume (V), which is an extensive property, can be changed to specific volume, which is an intensive property, by dividing by the mass of the system, $v = V/m$. Any specific property (specific volume, specific enthalpy, specific entropy), is an intensive property, as indicated in Figure 3.

The use of intensive and extensive properties is demonstrated in the following discussion. Consider as a system 1 lbm of water contained in the piston-cylinder arrangement of Figure 4. Suppose that the piston and weight maintain a pressure of 14.7 psia in the cylinder and that the initial temperature is 60°F, part (a) of Figure 4. As heat is transferred to the water, the temperature increases. The specific volume increases slightly, and the pressure remains constant. When the temperature reaches 212°F, additional heat transfer results in a change in phase (boiling), as indicated in part (b).

That is, some of the liquid becomes vapor and both the temperature and pressure remain constant, but the specific volume increases considerably. When the last drop of liquid is vaporized, further transfer of heat results in an increase in both temperature and specific volume of the vapor, part (c). In this example, temperature and pressure are intensive, and therefore do not depend upon the amount of mass present. By examining the specific volume (an intensive property) of the water in the piston instead of the volume (an extensive property), we can examine how any portion of the water in the piston changes. Volume by itself tells us nothing about the water in the piston. However, by knowing the specific volume we can tell if the water is a liquid or steam.

It is customary to define some intensive properties associated with extensive properties. For example, the volume per unit mass is called the specific volume,

$$v \equiv \frac{V}{M}$$

and the internal energy per unit mass is called the specific internal energy.

$$u \equiv \frac{U}{M}$$

Intensive properties are useful because they can be tabulated or graphed without reference to the amount of material under study.

Figure 3 Intensive Properties

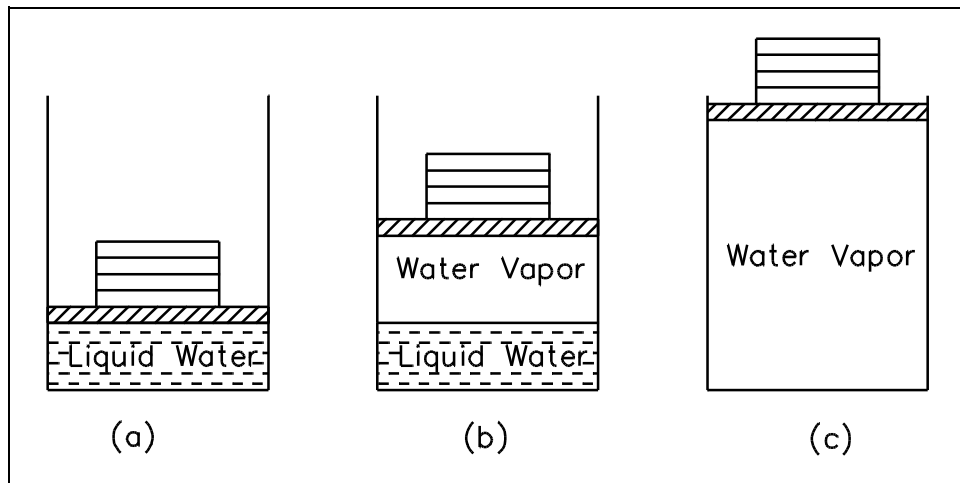


Figure 4 Piston-cylinder Arrangement

Saturation

The term *saturation* defines a condition in which a mixture of vapor and liquid can exist together at a given temperature and pressure. The temperature at which vaporization (boiling) starts to occur for a given pressure is called the *saturation temperature* or *boiling point*. The pressure at which vaporization (boiling) starts to occur for a given temperature is called the *saturation pressure*. For water at 212°F, the saturation pressure is 14.7 psia and, for water at 14.7 psia, the saturation temperature is 212°F. For a pure substance there is a definite relationship between saturation pressure and saturation temperature. The higher the pressure, the higher the saturation temperature. The graphical representation of this relationship between temperature and pressure at saturated conditions is called the *vapor pressure curve*. A typical vapor pressure curve is shown in Figure 5. The vapor/liquid mixture is at saturation when the conditions of pressure and temperature fall on the curve.

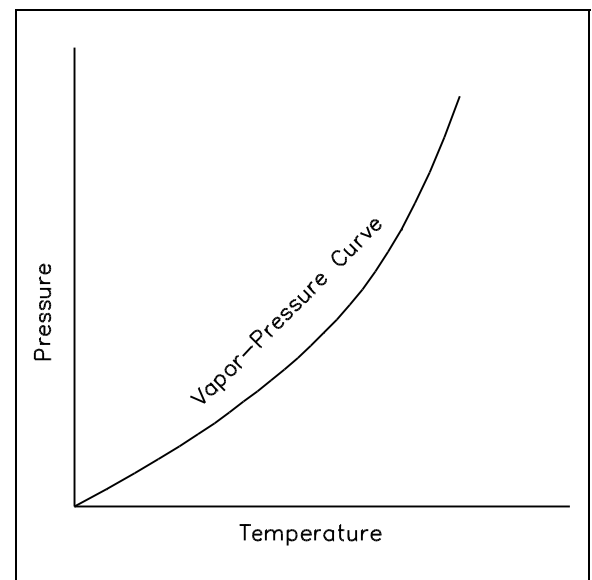


Figure 5 Vapor Pressure Curve

Saturated and Subcooled Liquids

If a substance exists as a liquid at the saturation temperature and pressure, it is called a *saturated liquid*.

If the temperature of the liquid is lower than the saturation temperature for the existing pressure, it is called either a *subcooled liquid* (implying that the temperature is lower than the saturation temperature for the given pressure) or a *compressed liquid* (implying that the pressure is greater than the saturation pressure for the given temperature). Both terms have the same meaning, so either term may be used.

Quality

When a substance exists as part liquid and part vapor at saturation conditions, its *quality* (x) is defined as the ratio of the mass of the vapor to the total mass of both vapor and liquid. Thus, if the mass of vapor is 0.2 lbm and the mass of the liquid is 0.8 lbm, the quality is 0.2 or 20%. Quality is an intensive property. Quality has meaning when the substance is in a saturated state only, at saturation pressure and temperature. The area under the bell-shaped curve on figure 6 shows the region in which quality is important.

$$x = \frac{m_{\text{vapor}}}{(m_{\text{liquid}} + m_{\text{vapor}})} \quad (1-20)$$

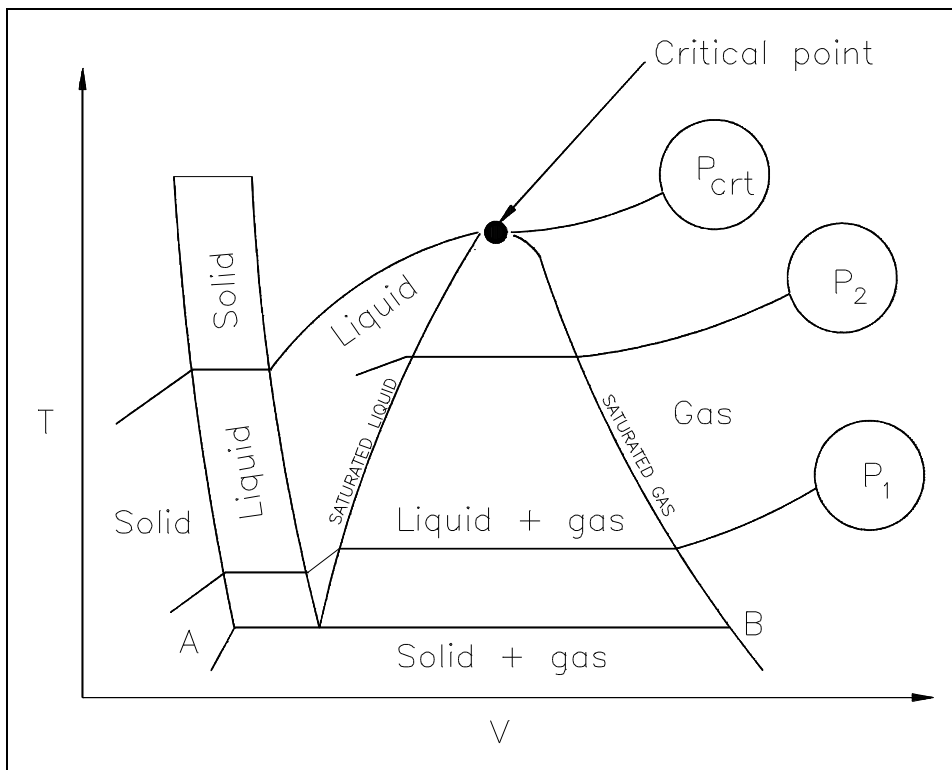


Figure 6 T-V Diagram Showing the Saturation Region

Moisture Content

The *moisture content* of a substance is the opposite of its quality. Moisture (M) is defined as the ratio of the mass of the liquid to the total mass of both liquid and vapor. The moisture of the mixture in the previous paragraph would be 0.8 or 80%. The following equations show how to calculate the moisture of a mixture and the relationship between quality and moisture.

$$M = \frac{m_{\text{liquid}}}{(m_{\text{liquid}} + m_{\text{vapor}})} \quad (1-21)$$

$$M = 1 - x$$

Saturated and Superheated Vapors

If a substance exists entirely as vapor at saturation temperature, it is called *saturated vapor*. Sometimes the term dry saturated vapor is used to emphasize that the quality is 100%. When the vapor is at a temperature greater than the saturation temperature, it is said to exist as *superheated vapor*. The pressure and temperature of superheated vapor are independent properties, since the temperature may increase while the pressure remains constant. Actually, the substances we call gases are highly superheated vapors.

Constant Pressure Heat Addition

Consider the plot on the temperature-volume diagram of Figure 7, viewing the constant-pressure line that represents the states through which the water of the previous discussion passes as it is heated from the initial state of 14.7 psia and 60°F. Let state A represent the initial state and state B represent the start of the saturated liquid line (212°F). Therefore, line AB represents the process in which the liquid is heated from the initial temperature to the saturation temperature.

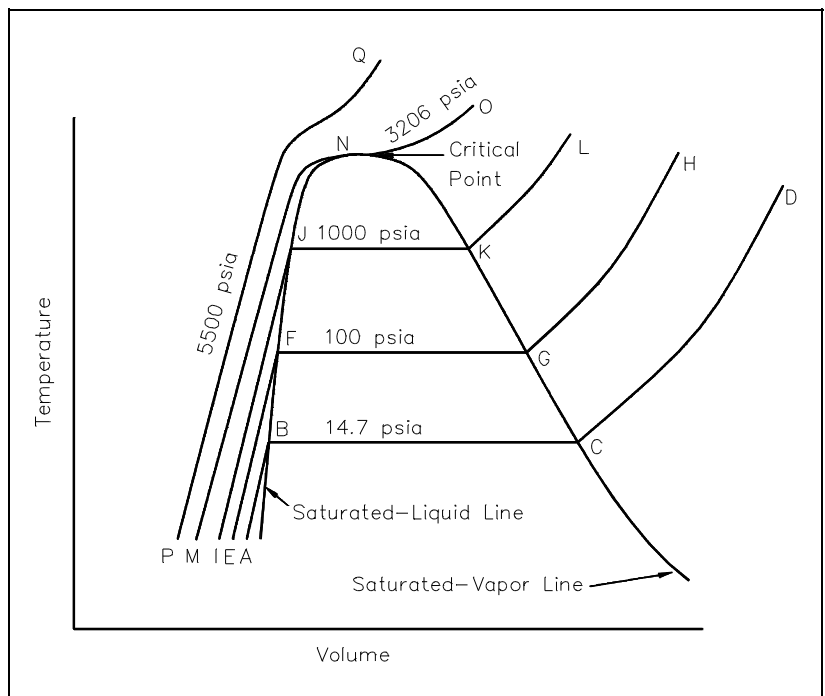


Figure 7 T-V Diagram

Point C is the saturated vapor state, and line BC is the constant-temperature process in which the change of phase from liquid to vapor occurs. Line CD represents the process in which the steam is super-heated at constant pressure. Temperature and volume both increase during the process.

Now let the process take place at a constant pressure of 100 psia, beginning from an initial temperature of 60°F. Point E represents the initial state, the specific volume being slightly less than 14.7 psia and 60°F. Vaporization now begins at point F, where the temperature is 327.8°F. Point G is the saturated-vapor state, and line GH is the constant-pressure process in which the steam is superheated.

In a similar manner, a constant pressure of 1000 psia is represented by line IJKL, the saturation temperature being 544.6°F.

Critical Point

At a pressure of 3206.2 psia, represented by line MNO, there is no constant-temperature vaporization process. Rather, point N is a point of inflection, with the slope being zero. This point is called the *critical point*, and at the critical point the saturated-liquid and saturated-vapor states are identical. The temperature, pressure, and specific volume at the critical point are called the critical temperature, critical pressure, and critical volume.

A constant pressure process greater than the critical pressure is represented by line PQ. There is no definite change in phase from liquid to vapor and no definite point at which there is a change from the liquid phase to the vapor phase. For pressures greater than the critical pressure, the substance is usually called a liquid when the temperature is less than the critical temperature (705.47°F) and a vapor or gas when the temperature is greater than the critical temperature. In the figure, line NJFB represents the saturated liquid line, and the line NKGC represents the saturated vapor line.

Fusion

Consider one further experiment with the piston-cylinder arrangement of Figure 4. Suppose the cylinder contained 1 lbm of ice at 0°F, 14.7 psia. When heat is transferred to the ice, the pressure remains constant, the specific volume increases slightly, and the temperature increases until it reaches 32°F, at which point the ice melts while the temperature remains constant. In this state the ice is called a saturated solid. For most substances, the specific volume increases during this melting process, but for water the specific volume of the liquid is less than the specific volume of the solid. This causes ice to float on water. When all the ice is melted, any further heat transfer causes an increase in temperature of the liquid. The process of melting is also referred to as *fusion*. The heat added to melt ice into a liquid is called the latent heat of fusion.

Sublimation

If the initial pressure of the ice at 0°F is 0.0505 psia, heat transfer to the ice first results in an increase in temperature to 20°F. At this point, however, the ice passes directly from the solid phase to the vapor phase in the process known as *sublimation*. Sublimation is a special term used for cases in which the transition between the solid phase and the vapor phase occurs directly, without passing through the liquid phase. Further heat transfer would result in superheating the vapor.

Triple Point

Finally, consider an initial pressure of the ice of 0.08854 psia. Again, as a result of heat transfer, the temperature will increase until it reaches 32°F. At this point, however, further heat transfer may result in some of the ice becoming vapor and some becoming liquid because it is possible to have the three phases in equilibrium. This is called the *triple point*, defined as the state in which all three phases may be present in equilibrium.

Figure 8 is a pressure-temperature diagram for water that shows how the solid, liquid, and vapor phases may exist together in equilibrium. Along the sublimation line, the solid and vapor phases are in equilibrium, along the fusion line, the solid and liquid phases are in equilibrium; and along the vaporization line, the liquid and vapor phases are in equilibrium. The only point at which all three phases may exist in equilibrium is the triple point. The temperature and pressure for the triple point of water are 32.02°F and 0.08865 psia. The vaporization line ends at the critical point because there is no distinct change from the liquid phase to the vapor phase above the critical point.

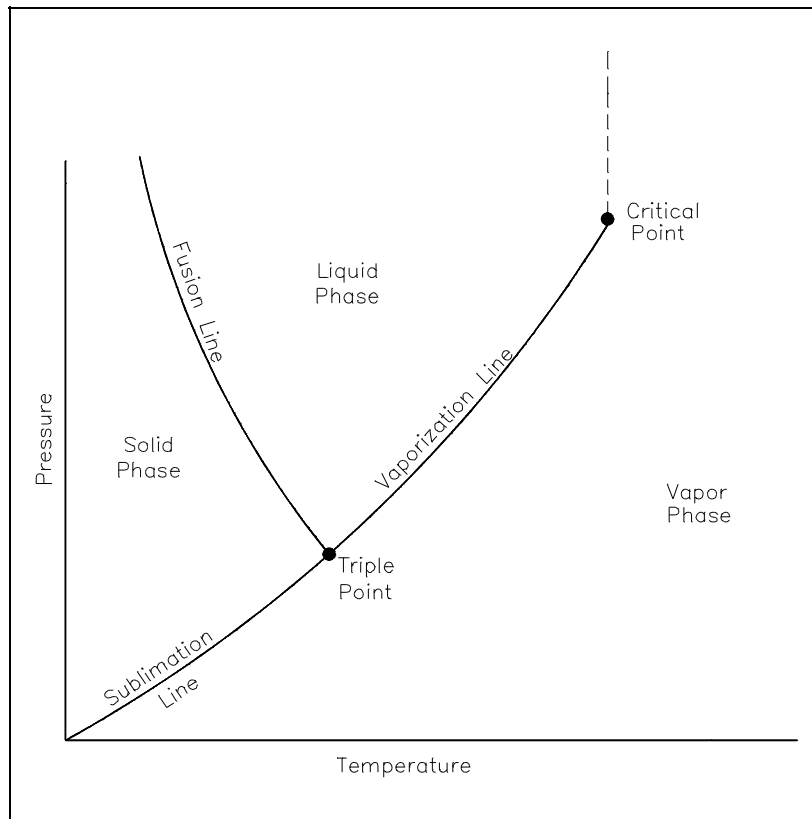


Figure 8 Pressure-Temperature Diagram

Condensation

All the processes discussed on the preceding pages (vaporization, sublimation, and fusion) occur during a heat addition to a substance. If heat is removed from a substance, the opposite of the described processes will occur.

As previously described, a heat addition at a constant pressure to a saturated liquid will cause the liquid to evaporate (change phase from liquid to vapor). If heat is removed at a constant pressure from a saturated vapor, *condensation* will occur and the vapor will change phase to liquid. So the processes of vaporization and condensation are the exact opposite of each other.

Similarly, freezing is the opposite process of melting and fusion. Sublimation also has an opposite process in which a gas goes directly to solid, but this process is not normally referred to with a unique term.

Summary

The important information from this chapter is summarized on the following page.

Change of Phase Summary

- Classification of Properties

Intensive properties are independent of mass (temperature, pressure, or any specific property)

Extensive properties are a function of the mass of the system (mass, volume)

- The following terms were defined in this chapter:

Saturation	-	combination of temperature and pressure at which a mixture of vapor and liquid can exist at equilibrium
Subcooled liquid	-	a liquid at a temperature below saturation temperature for its pressure
Superheated vapor	-	a vapor at a temperature above saturation temperature for its pressure
Critical point	-	the temperature and pressure above which there is no distinction between the liquid and vapor phases
Triple point	-	the temperature and pressure at which all three phases can exist in equilibrium
Vapor pressure curve	-	a graphical representation of the relationship between temperature and pressure at saturated conditions
Quality	-	the fraction of the total mass of a mixture that is in the vapor phase
Moisture Content	-	the fraction of the total mass of a mixture that is in the liquid phase

Change of Phase Summary (Cont.)

- The following terms are labels for processes that occur when a substance changes between the three phases of matter: solid, liquid, and vapor.

Sublimation - change of phase from solid to vapor

Vaporization - change of phase from liquid to vapor

Condensation - change of phase from vapor to liquid

Fusion or melting - change of phase from solid to liquid

PROPERTY DIAGRAMS AND STEAM TABLES

Property diagrams and steam tables are used in studying the theoretical and actual properties and efficiencies of a given system.

- EO 1.16** **Given a Mollier diagram and sufficient information to indicate the state of the fluid, DETERMINE any unknown properties for the fluid.**
- EO 1.17** **Given a set of steam tables and sufficient information to indicate the state of the fluid, DETERMINE any unknown properties for the fluid.**
- EO 1.18** **DETERMINE the change in the enthalpy of a fluid as it passes through a system component, given the state of the fluid at the inlet and outlet of the component and either steam tables or a Mollier diagram.**
-

Property Diagrams

The phases of a substance and the relationships between its properties are most commonly shown on *property diagrams*. A large number of different properties have been defined, and there are some dependencies between properties. For example, at standard atmospheric pressure and temperature above 212°F, water exists as steam and not a liquid; it exists as a liquid at temperatures between 32°F and 212°F; and, it exists as ice at temperatures below 32°F. In addition, the properties of ice, water, and steam are related. Saturated steam at 212°F and standard atmospheric pressure has a specific volume of 26.8 ft³/lbm. At any other temperature and pressure, saturated steam has a different specific volume. For example, at 544°F and 1000 psia pressure, its specific volume is 0.488 ft³/lbm.

There are five basic properties of a substance that are usually shown on property diagrams. These are: pressure (P), temperature (T), specific volume (v), specific enthalpy (h), and specific entropy (s). When a mixture of two phases, such as water and steam, is involved, a sixth property, quality (x), is also used.

There are six different types of commonly encountered property diagrams. These are: Pressure-Temperature (P-T) diagrams, Pressure-Specific Volume (P-v) diagrams, Pressure-Enthalpy (P-h) diagrams, Enthalpy-Temperature (h-T) diagrams, Temperature-entropy (T-s) diagrams, and Enthalpy-Entropy (h-s) or Mollier diagrams.

Pressure-Temperature (P-T) Diagram

A P-T diagram is the most common way to show the phases of a substance. Figure 9 is the P-T diagram for pure water. A P-T diagram can be constructed for any pure substance. The line that separates the solid and vapor phases is called the sublimation line. The line that separates the solid and liquid phases is called the fusion line. The line that separates the liquid and vapor phases is called the vaporization line. The point where the three lines meet is called the triple point. The triple point is the only point at which all three phases can exist in equilibrium. The point where the vaporization line ends is called the critical point. At temperatures and pressures greater than those at the critical point, no substance can exist as a liquid no matter how great a pressure is exerted upon it.

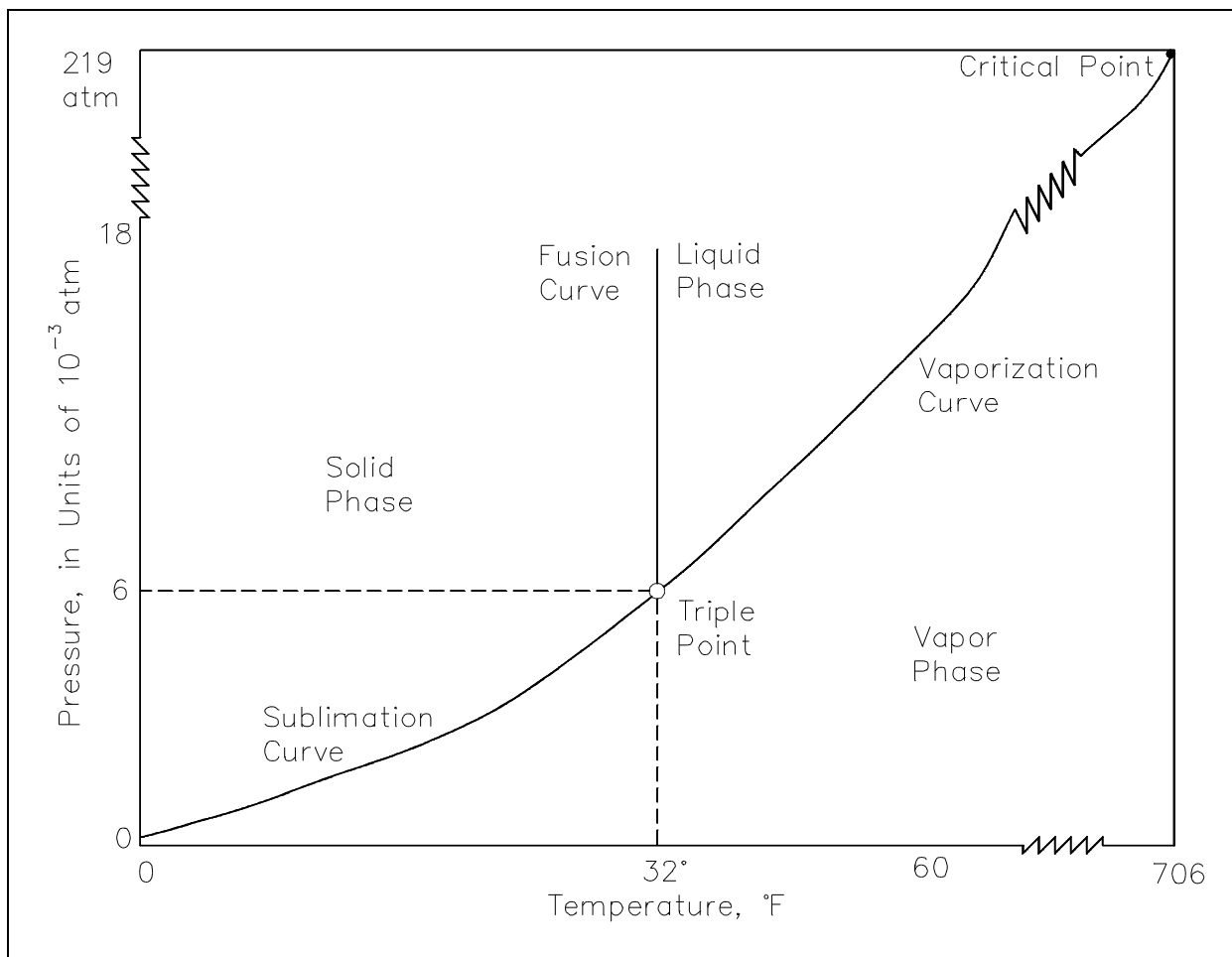


Figure 9 P-T Diagram for Water

Pressure-Specific Volume (P-v) Diagram

A P-v diagram is another common type of property diagram. Figure 10 is the P-v diagram for pure water. A P-v diagram can be constructed for any pure substance. A P-v diagram is different from a P-T diagram in one particularly important way. There are regions on a P-v diagram in which two phases exist together. In the liquid-vapor region in Figure 10, water and steam exist together.

For example, at point A, water with a specific volume (v_f), given by point B, exists together with steam with a specific volume (v_g), given by point C. The dotted lines on Figure 10 are lines of constant temperature. The quality of the mixture at any point in the liquid-vapor region can be found because the specific volumes of water, steam, and the mixture are all known. The quality can be found using the following relationship.

$$v = xv_g + (1 - x)v_f$$

$$x = \frac{v - v_f}{v_g - v_f} = \frac{v - v_f}{v_{fg}}$$

where:

v = specific volume of the mixture (ft^3/lbm)

x = quality of the mixture (no units)

v_g = specific volume of the vapor (ft^3/lbm)

v_f = specific volume of the liquid (ft^3/lbm)

v_{fg} = specific volume change of vaporization (ft^3/lbm) or $v_{fg} = v_g - v_f$

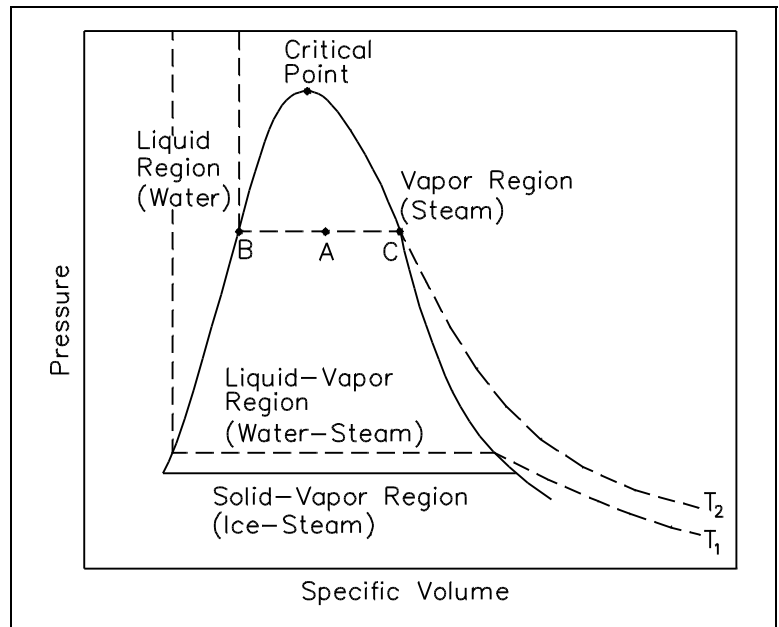


Figure 10 P-v Diagram for Water

Pressure-Enthalpy (P-h) Diagram

A P-h diagram exhibits the same features as a P-v diagram. Figure 11 is the P-h diagram for pure water. A P-h diagram can be constructed for any pure substance. Like the P-v diagram, there are regions on a P-h diagram in which two phases exist together. In the liquid-vapor region in Figure 11, water and steam exist together. For example, at point A, water with an enthalpy (h_f), given by point B, exists together with steam with an enthalpy (h_g), given by point C. The quality of the mixture at any point in the liquid-vapor region can be found using the following relationship.

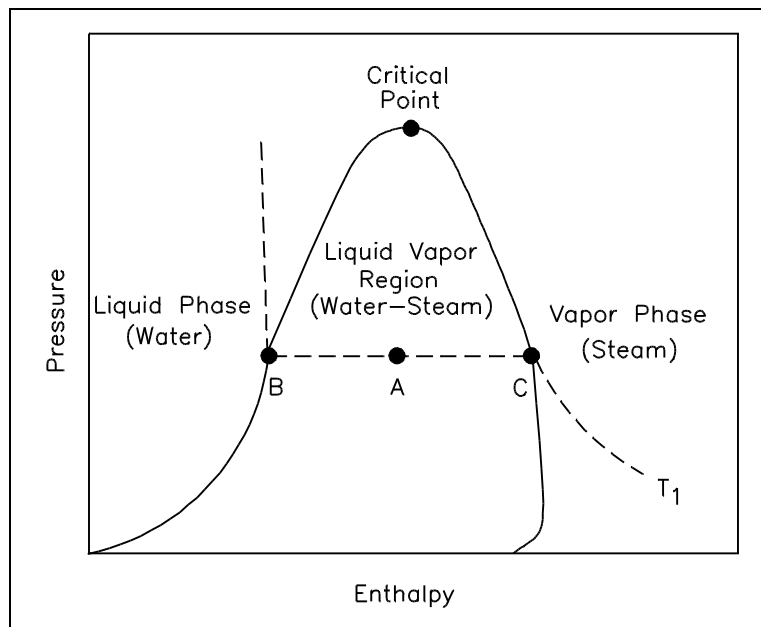


Figure 11 P-h Diagram for Water

$$h = xh_g + (1 - x)h_f$$

$$x = \frac{h - h_f}{h_{fg}}$$

where:

h = specific enthalpy of the mixture (Btu/lbm)

x = quality of the mixture (no units)

h_g = specific enthalpy of the saturated vapor (Btu/lbm)

h_f = specific enthalpy of the saturated liquid (Btu/lbm)

h_{fg} = specific enthalpy change of vaporization (Btu/lbm) or $h_{fg} = h_g - h_f$

Enthalpy-Temperature (h-T) Diagram

An h-T diagram exhibits the same features as on the previous property diagrams. Figure 12 is the h-T diagram for pure water. An h-T diagram can be constructed for any pure substance. As in the previous property diagrams, there are regions on the h-T diagram in which two phases exist together. The region between the saturated liquid line and the saturated vapor line represents the area of two phases existing at the same time. The vertical distance between the two saturation lines represents the latent heat of vaporization. If pure water existed at point A on the saturated liquid line and an amount of heat was added equal to the latent heat of vaporization, then the water would change phase from a saturated liquid to a saturated vapor (point B), while maintaining a constant temperature. As shown in Figure 12, operation outside the saturation lines results in a subcooled liquid or superheated steam.

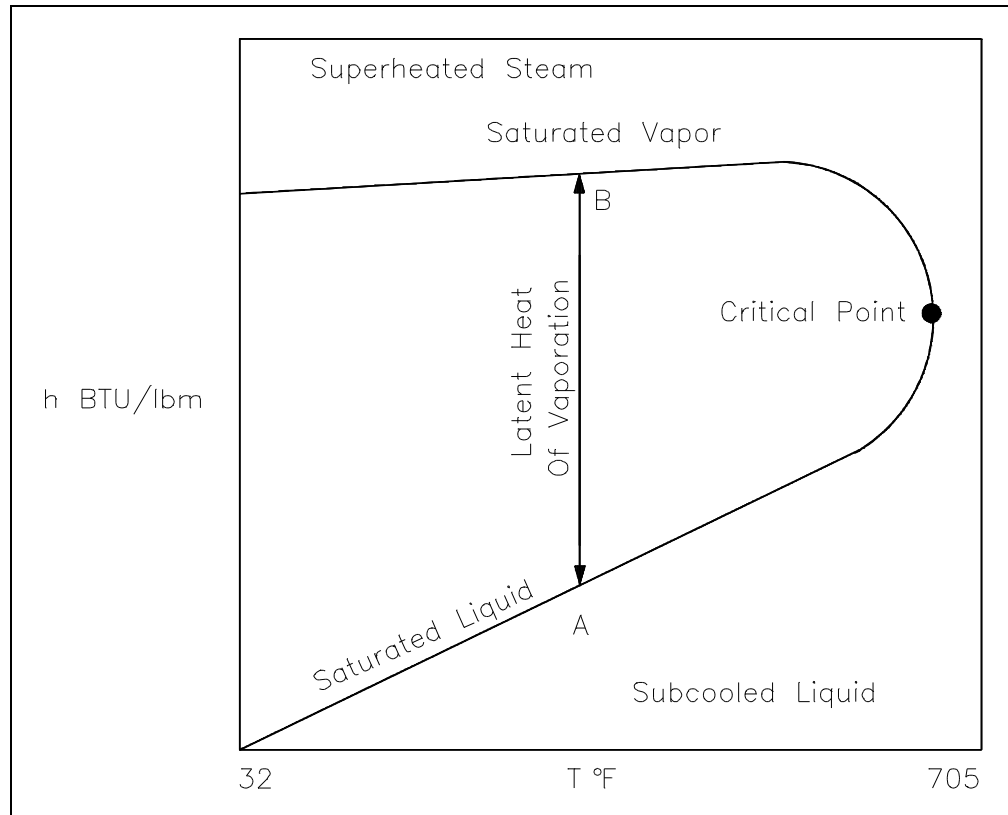


Figure 12 h-T Diagram for Water

The quality of the mixture at any point in the liquid-vapor region can be found using the same relationship as shown for the P-h diagram.

$$x = \frac{h - h_f}{h_{fg}}$$

Temperature-Entropy (T-s) Diagram

A T-s diagram is the type of diagram most frequently used to analyze energy transfer system cycles. This is because the work done by or on the system and the heat added to or removed from the system can be visualized on the T-s diagram. By the definition of entropy, the heat transferred to or from a system equals the area under the T-s curve of the process. Figure 13 is the T-s diagram for pure water. A T-s diagram can be constructed for any pure substance. It exhibits the same features as P-v diagrams.

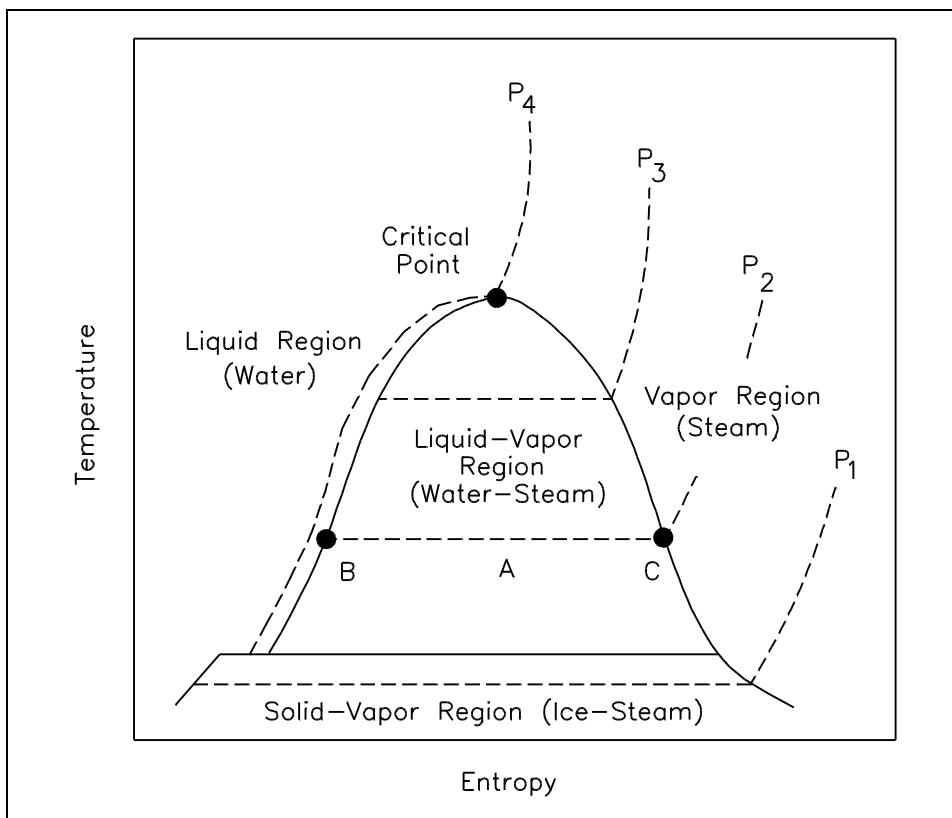


Figure 13 T-s Diagram for Water

In the liquid-vapor region in Figure 13, water and steam exist together. For example, at point A, water with an entropy (s_f) given by point B, exists together with steam with an entropy (s_g) given by point C. The quality of the mixture at any point in the liquid-vapor region can be found using the following relationship.

$$s = xs_g + (1 - x)s_f$$

$$x = \frac{s - s_f}{s_{fg}}$$

where:

s = specific entropy of the mixture (Btu/lbm-°R)

x = quality of the mixture (no units)

s_g = specific entropy of the saturated vapor (Btu/lbm-°R)

s_f = specific entropy of the saturated liquid
(Btu/lbm-°R)

s_{fg} = specific entropy change of vaporization (Btu/lbm-°R) or $s_{fg} = s_g - s_f$

Enthalpy-Entropy (h-s) or Mollier Diagram

The Mollier diagram, shown in Figure A-1 of Appendix A, is a chart on which enthalpy (h) versus entropy (s) is plotted. It is sometimes known as the h - s diagram and has an entirely different shape from the T - s diagrams. The chart contains a series of constant temperature lines, a series of constant pressure lines, a series of constant moisture or quality lines, and a series of constant superheat lines. The Mollier diagram is used only when quality is greater than 50% and for superheated steam.

Steam Tables

Steam tables consist of two sets of tables of the energy transfer properties of water and steam: saturated steam tables and superheated steam tables. Portions of the tables are shown in Figure A-2 of Appendix A. Both sets of tables are tabulations of pressure (P), temperature (T), specific volume (v), specific enthalpy (h), and specific entropy (s). The following notation is used in steam tables. Some tables use v for v (specific volume) because there is little possibility of confusing it with velocity.

T = temperature (°F)

P = pressure (psi)

v = specific volume (ft³/lbm)

v_f = specific volume of saturated liquid (ft³/lbm)

v_g = specific volume of saturated vapor (ft³/lbm)

v_{fg}	=	specific volume change of vaporization (ft ³ /lbm)
h	=	specific enthalpy (Btu/lbm)
h_f	=	specific enthalpy of saturated liquid (Btu/lbm)
h_g	=	specific enthalpy of saturated vapor (Btu/lbm)
h_{fg}	=	specific enthalpy change of vaporization (Btu/lbm)
s	=	specific entropy (Btu/lbm-°R)
s_f	=	specific entropy of saturated liquid (Btu/lbm-°R)
s_g	=	specific entropy of saturated vapor (Btu/lbm-°R)
s_{fg}	=	specific entropy change of vaporization (Btu/lbm-°R)
Sh	=	number of degrees of superheat (°F)

The saturated steam tables give the energy transfer properties of saturated water and saturated steam for temperatures from 32 to 705.47°F (the critical temperature) and for the corresponding pressure from 0.08849 to 3208.2 psi. Normally, the saturated steam tables are divided into two parts: temperature tables, which list the properties according to saturation temperature (T_{sat}); and pressure tables, which list them according to saturation pressure (P_{sat}). Figure A-2 shows a portion of a typical saturated steam temperature table and a portion of a typical saturated steam pressure table. The values of enthalpy and entropy given in these tables are measured relative to the properties of saturated liquid at 32°F. Hence, the enthalpy (h_f) of saturated liquid and the entropy (s_f) of saturated liquid have values of approximately zero at 32°F.

Most practical applications using the saturated steam tables involve steam-water mixtures. The key property of such mixtures is steam quality (x), defined as the mass of steam present per unit mass of steam-water mixture, or steam moisture content (y), defined as the mass of water present per unit mass of steam-water mixture. The following relationships exist between the quality of a liquid-vapor mixture and the specific volumes, enthalpies, or entropies of both phases and of the mixture itself. These relationships are used with the saturated steam tables.

$$v = xv_g + (1 - x)v_f$$

$$x = \frac{v - v_f}{v_{fg}}$$

$$h = xh_g + (1 - x)h_f$$

$$x = \frac{h - h_f}{h_{fg}}$$

$$s = xs_g + (1 - x)s_f$$

$$x = \frac{s - s_f}{s_{fg}}$$

In order to solve problems in Thermodynamics, information concerning the "state" of the substance studied must be obtained. Usually, two properties (for example, v , p , T , h , s) of the substance must be known in order to determine the other needed properties. These other properties are usually obtained utilizing either the Mollier diagram (if the substance is steam) or the saturated and superheated steam tables, as shown in the Figures A-1 and A-2.

The following two examples illustrate the use of the Mollier diagram and the steam tables.

Example 1: Use of Mollier Chart.

Superheated steam at 700 psia and 680°F is expanded at constant entropy to 140 psia. What is the change in enthalpy?

Solution:

Use the Mollier Chart. Locate point 1 at the intersection of the 700 psia and the 680°F line. Read $h = 1333$ Btu/lbm.

Follow the entropy line downward vertically to the 140 psia line and read $h = 1178$ Btu/lbm.

$$\Delta h = 1178 - 1333 = -155 \text{ Btu/lbm}$$

Example 2: Use of steam tables

What are the specific volume, enthalpy, and entropy of steam having a quality of 90% at 400 psia?

Solution:

From the steam tables at 400 psia:

$$\begin{aligned}v_f &= 0.01934 & v_g &= 1.14162 & h_f &= 424.2 \\h_{fg} &= 780.4 & s_f &= 0.6217 & s_{fg} &= 0.8630 \\v &= v_f + x(v_{fg}) \\v &= 0.01934 + (0.9)(1.14162) = 1.0468 \text{ lbm/ft}^3 \\h &= h_f + x(h_{fg}) \\h &= 424.2 + (0.9)(780.4) = 1126.56 \text{ Btu/lbm} \\s &= s_f + x(s_{fg}) \\s &= 0.6217 + (0.9)(0.8630) = 1.3984 \text{ Btu/lbm-}^\circ\text{R}\end{aligned}$$

If the substance is not water vapor, the "state" of the substance is usually obtained through the use of T-s (temperature-entropy) and h-s (enthalpy-entropy) diagrams, available in most thermodynamics texts for common substances. The use of such diagrams is demonstrated by the following two examples.

Example 3: Use of the h-s diagram

Mercury is used in a nuclear facility. What is the enthalpy of the mercury if its pressure is 100 psia and its quality is 70%?

Solution:

From the mercury diagram, Figure A-3 of Appendix A, locate the pressure of 100 psia. Follow that line until reaching a quality of 70%. The intersection of the two lines gives an enthalpy that is equal to $h = 115 \text{ Btu/lbm}$.

Example 4: Use of the T-s diagram

Carbon dioxide is used in a particular process in which the pressure is 100 psia and the temperature is 100°F. What is the enthalpy value of the gas?

Solution:

From the carbon dioxide diagram, Figure A-4 of Appendix A, locate the pressure of 100 psia. Follow that line until reaching a temperature of 100°F. The intersection of the two lines gives an enthalpy that is equal to $h = 316$ Btu/lbm.

Once the various states have been fixed for the particular process the substance has passed through (for example, going from a saturated liquid state to a compressed liquid state across a pump), energy exchanges may be determined as was shown in Example 1. The energy exchanges are never 100 percent efficient, as already discussed. The degree of efficiency obtained by the system depends upon the process through which the system has passed. Generally, the efficiency of a component depends upon how much friction exists in the flow of the substance, the pressure drops within the system, the inlet and outlet temperatures, and various other factors. The properties affecting the efficiency of the system are determined by use of the charts and diagrams mentioned in this section.

When power cycles are utilized for large systems, the efficiency of each component should be maximized in order to have the highest possible overall efficiency for the system. Each component affects the system efficiency in a different manner. To maximize efficiency, the practical approach to large systems is to have multistage expansion with reheat between stages and regenerators in the system where applicable.

Summary

The important information from this chapter is summarized below.

Property Diagrams and Steam Tables Summary

- The Mollier diagram can be used to determine various properties of a fluid.

Mollier diagram is an h versus s plot.

Can only be used when quality is greater than 50% and for superheated steam.

Contains a series of constant temperature, constant pressure, constant moisture content, and constant superheat lines.

- The steam tables can be used to determine various properties of water using the following equations.

$$v = xv_g + (1 - x)v_f$$

$$h = xh_g + (1 - x)h_f$$

$$s = xs_g + (1 - x)s_f$$

$$x = \frac{v - v_f}{v_{fg}}$$

$$x = \frac{h - h_f}{h_{fg}}$$

$$x = \frac{s - s_f}{s_{fg}}$$

- The change in enthalpy of a fluid as it passes through a component can be determined using a Mollier diagram on steam tables.

FIRST LAW OF THERMODYNAMICS

The First Law of Thermodynamics is a balance of the various forms of energy as they pertain to the specified thermodynamic system (control volume) being studied.

- EO 1.19** **STATE** the First Law of Thermodynamics.
- EO 1.20** Using the First Law of Thermodynamics, **ANALYZE** an open system including all energy transfer processes crossing the boundaries.
- EO 1.21** Using the First Law of Thermodynamics, **ANALYZE** cyclic processes for a thermodynamic system.
- EO 1.22** Given a defined system, **PERFORM** energy balances on all major components in the system.
- EO 1.23** Given a heat exchanger, **PERFORM** an energy balance across the two sides of the heat exchanger.
- EO 1.24** **IDENTIFY** the path(s) on a T-s diagram that represents the thermodynamic processes occurring in a fluid system.
-

First Law of Thermodynamics

The First Law of Thermodynamics states:

Energy can neither be created nor destroyed, only altered in form.

For any system, energy transfer is associated with mass and energy crossing the control boundary, external work and/or heat crossing the boundary, and the change of stored energy within the control volume. The mass flow of fluid is associated with the kinetic, potential, internal, and "flow" energies that affect the overall energy balance of the system. The exchange of external work and/or heat complete the energy balance.

The First Law of Thermodynamics is referred to as the Conservation of Energy principle, meaning that energy can neither be created nor destroyed, but rather transformed into various forms as the fluid within the control volume is being studied. The energy balance spoken of here is maintained within the system being studied. The system is a region in space (control volume) through which the fluid passes. The various energies associated with the fluid are then observed as they cross the boundaries of the system and the balance is made.

As discussed in previous chapters of this module, a system may be one of three types: isolated, closed, or open. The open system, the most general of the three, indicates that mass, heat, and external work are allowed to cross the control boundary. The balance is expressed in words as: all energies into the system are equal to all energies leaving the system plus the change in storage of energies within the system. Recall that energy in thermodynamic systems is composed of kinetic energy (KE), potential energy (PE), internal energy (U), and flow energy (P_L); as well as heat and work processes.

$$\Sigma (\text{all energies in}) = \Sigma (\text{all energies out}) + \Delta(\text{energy stored in system})$$

$$\Sigma E_{\text{in}} = \Sigma E_{\text{out}} + \Delta E \text{ storage}$$

For most industrial plant applications that most frequently use cycles, there is no change in storage (i.e. heat exchangers do not swell while in operation).

In equation form, the balance appears as indicated on Figure 14.

where:

$$\dot{Q} = \text{heat flow into the system (Btu/hr)}$$

$$\dot{m}_{\text{in}} = \text{mass flow rate into the system (lbm/hr)}$$

$$u_{\text{in}} = \text{specific internal energy into the system (Btu/lbm)}$$

$$P_{\text{in}} v_{\text{in}} = \text{pressure-specific volume energy into the system (ft-lbf/lbm)}$$

$$\frac{\bar{V}_{\text{in}}^2}{2g_c} = \text{kinetic energy into the system (ft-lbf /lbm) where}$$

$$\bar{V}_{\text{in}} = \text{average velocity of fluid (ft/sec)}$$

$$g_c = \text{the gravitational constant (32.17 ft-lbm/lbf-sec}^2\text{)}$$

$$\frac{g}{g_c} Z_{\text{in}} = \text{potential energy of the fluid entering the system (ft-lbf/lbm) where}$$

$$Z_{\text{in}} = \text{height above reference level (ft)}$$

$$g = \text{acceleration due to gravity (ft/sec}^2\text{)}$$

$$g_c = \text{the gravitational constant (32.17 ft-lbm/lbf-sec}^2\text{)}$$

- \dot{W} = work flow out of the system (ft-lbf/hr)
- \dot{m}_{out} = mass flow rate out of the system (lbm/hr)
- u_{out} = specific internal energy out of the system (Btu/lbm)
- $P_{out} v_{out}$ = pressure-specific volume energy out of the system (ft-lbf/lbm)
- $\frac{\bar{V}_{out}^2}{2g_c}$ = kinetic energy out the system (ft-lbf/lbm)
- $\frac{g}{g_c} Z_{out}$ = potential energy out of the system (ft-lbf/lbm)

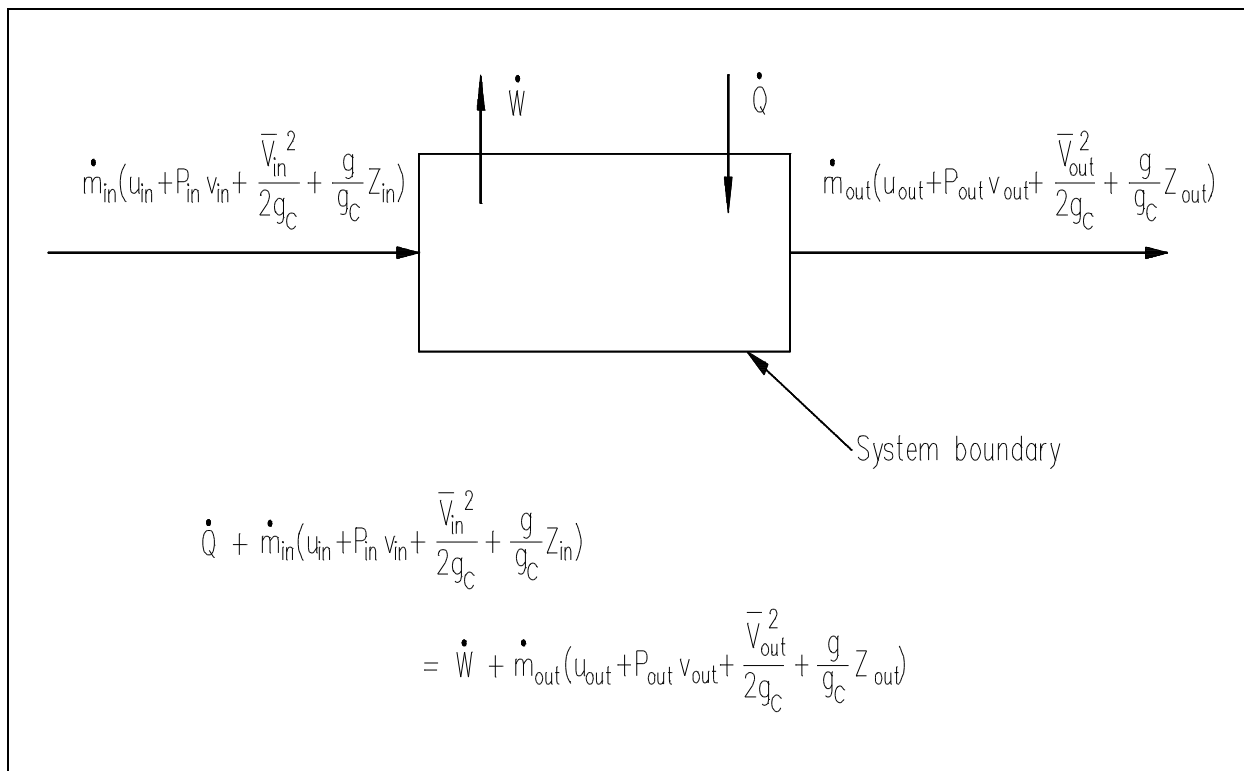


Figure 14 First Law of Thermodynamics

Heat and/or work can be directed into or out of the control volume. But, for convenience and as a standard convention, the net energy exchange is presented here with the net heat exchange assumed to be into the system and the net work assumed to be out of the system. If no mass crosses the boundary, but work and/or heat do, then the system is referred to as a "closed" system. If mass, work and heat do not cross the boundary (that is, the only energy exchanges taking place are within the system), then the system is referred to as an isolated system. Isolated and closed systems are nothing more than specialized cases of the open system. In this text, the open system approach to the First Law of Thermodynamics will be emphasized because it is more general. Also, almost all practical applications of the first law require an open system analysis.

An understanding of the control volume concept is essential in analyzing a thermodynamic problem or constructing an energy balance. Two basic approaches exist in studying Thermodynamics: the control mass approach and the control volume approach. The former is referred to as the LeGrange approach and the latter as the Eulerian approach. In the control mass concept, a "clump" of fluid is studied with its associated energies. The analyzer "rides" with the clump wherever it goes, keeping a balance of all energies affecting the clump.

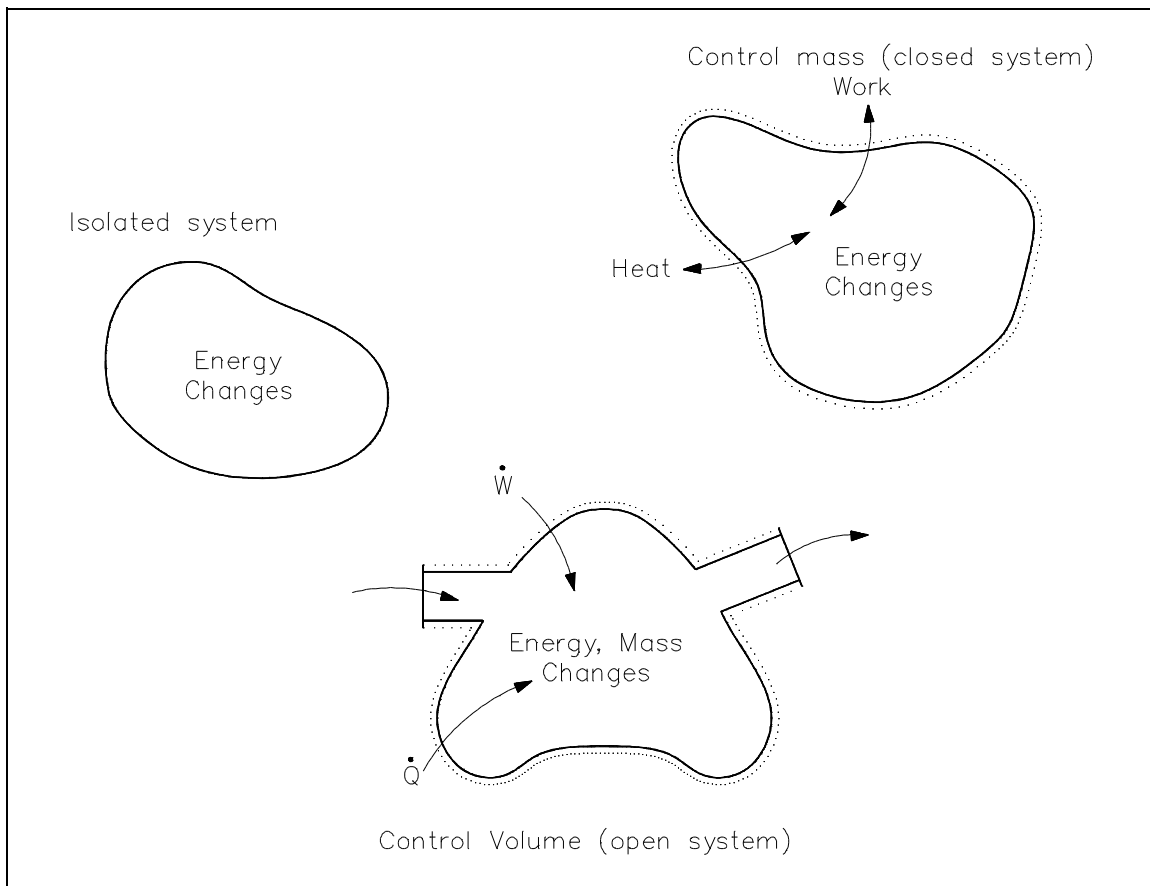


Figure 15 Control Volume Concepts

The control volume approach is one in which a fixed region in space is established with specified control boundaries, as shown in Figure 15. The energies that cross the boundary of this control volume, including those with the mass crossing the boundary, are then studied and the balance performed. The control volume approach is usually used today in analyzing thermodynamic systems. It is more convenient and requires much less work in keeping track of the energy balances. Examples of control volume applications are included in Figures 16-18.

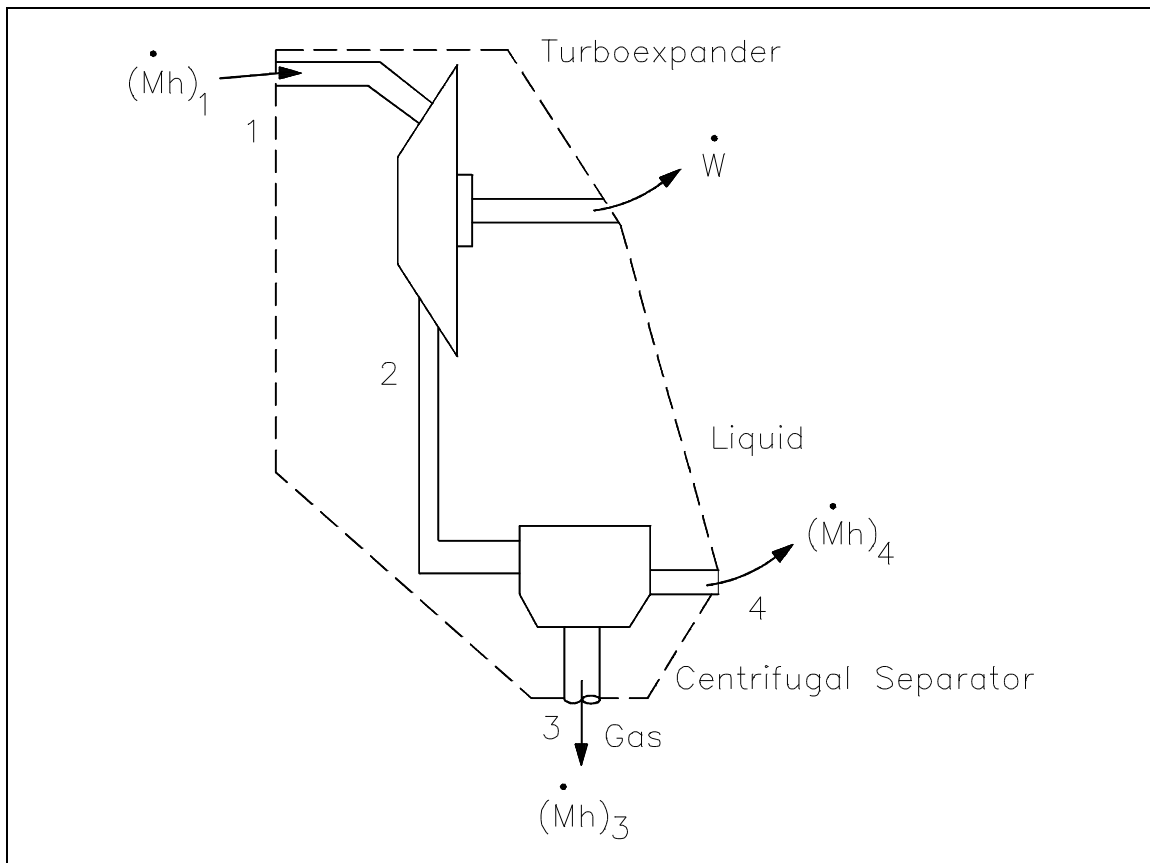


Figure 16 Open System Control Volumes

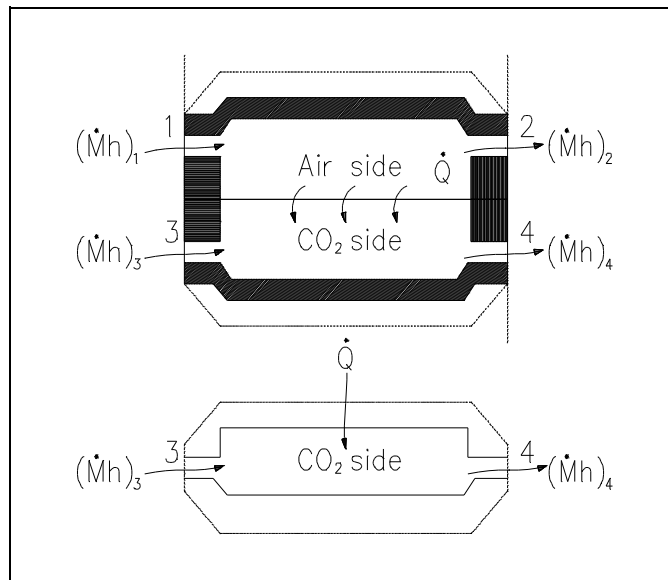


Figure 17 Open System Control Volumes (Cont.)

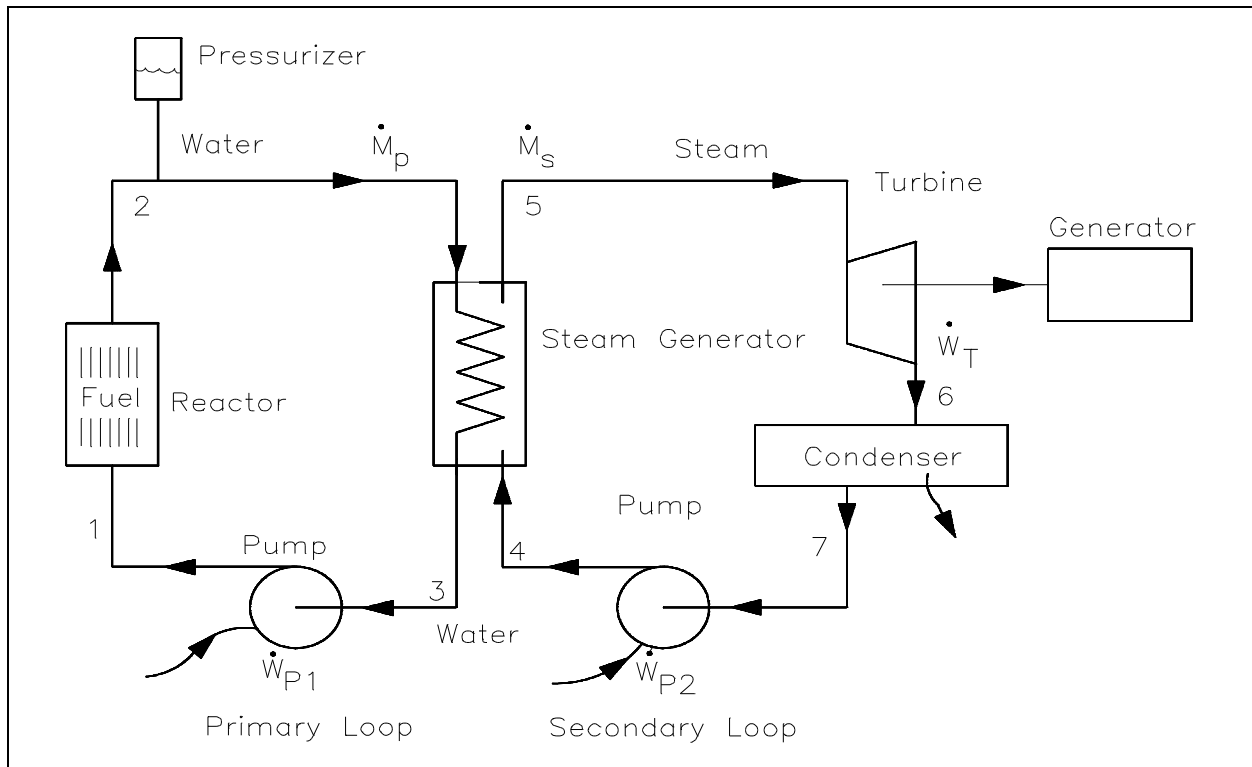


Figure 18 Multiple Control Volumes in Same System

The forms of energy that may cross the control volume boundary include those associated with the mass (m) crossing the boundary. Mass in motion has potential (PE), kinetic (KE), and internal energy (U). In addition, since the flow is normally supplied with some driving power (a pump for example), there is another form of energy associated with the fluid caused by its pressure. This form of energy is referred to as flow energy (Pv-work). The thermodynamic terms thus representing the various forms of energy crossing the control boundary with the mass are given as $m(u + Pv + ke + pe)$.

In open system analysis, the u and Pv terms occur so frequently that another property, enthalpy, has been defined as $h = u + Pv$. This results in the above expression being written as $m(h + ke + pe)$. In addition to the mass and its energies, externally applied work (W), usually designated as shaft work, is another form of energy that may cross the system boundary.

In order to complete and satisfy the conservation of energy relationship, energy that is caused by neither mass nor shaft work is classified as heat energy (Q). Then we can describe the relationship in equation form as follows.

$$\dot{m}(h_{in} + pe_{in} + ke_{in}) + \dot{Q} = \dot{m}(h_{out} + pe_{out} + ke_{out}) + \dot{W} \quad (1-22)$$

where:

\dot{m}	=	mass flow rate of working fluid (lbm/hr)
h_{in}	=	specific enthalpy of the working fluid entering the system (Btu/lbm)
h_{out}	=	specific enthalpy of the working fluid leaving the system (Btu/lbm)
pe_{in}	=	specific potential energy of working fluid entering the system (ft-lbf/lbm)
pe_{out}	=	specific potential energy of working fluid leaving the system (ft-lbf/lbm)
ke_{in}	=	specific kinetic energy of working fluid entering the system (ft-lbf/lbm)
ke_{out}	=	specific kinetic energy of working fluid leaving the system (ft-lbf/lbm)
\dot{W}	=	rate of work done by the system (ft-lbf/hr)
\dot{Q}	=	heat rate into the system (Btu/hr)

Example 1 illustrates the use of the control volume concept while solving a first law problem involving most of the energy terms mentioned previously.

Example 1: Open System Control Volume

The enthalpies of steam entering and leaving a steam turbine are 1349 Btu/lbm and 1100 Btu/lbm, respectively. The estimated heat loss is 5 Btu/lbm of steam. The flow enters the turbine at 164 ft/sec at a point 6.5 ft above the discharge and leaves the turbine at 262 ft/sec. Determine the work of the turbine.

Solution:

$$\dot{m}_{in}(h_{in} + pe_{in} + ke_{in}) + \dot{Q} = \dot{m}_{out}(h_{out} + pe_{out} + ke_{out}) + \dot{W}$$

1) Divide by \dot{m} since, $\dot{m}_{in} = \dot{m}_{out} = \dot{m}$.

$$(h_{in} + pe_{in} + ke_{in}) + q = (h_{out} + pe_{out} + ke_{out}) + w$$

where:

q = heat added to the system per pound (Btu/lbm)

w = work done by the system per pound (ft-lbf/lbm)

2) Use Joule's constant $J = 778 \text{ ft-lbf/Btu}$ for conversions and substitute known values.

$$1349 \text{ Btu/lbm} + (6.5/778) \text{ Btu/lbm} + [(164)^2/2(32.17)(778)] \text{ Btu/lbm} + (-5 \text{ Btu/lbm}) = 1100 \text{ Btu/lbm} + 0 \text{ pe}_{out} + [(262)^2/2(32.17)(778)] \text{ Btu/lbm} + w$$

Note: The minus sign indicates heat out of the turbine.

3) Solve for work, w .

$$1349 \text{ Btu/lbm} + 8.3548 \times 10^{-3} \text{ Btu/lbm} + 0.5368 \text{ Btu/lbm} - 5 \text{ Btu/lbm} = 1100 \text{ Btu/lbm} + 1.37 \text{ Btu/lbm} + w$$

$$1344.54 \text{ Btu/lbm} = 1101.37 \text{ Btu/lbm} + w$$

$$w = 1344.54 \text{ Btu/lbm} - 1101.37 \text{ Btu/lbm}$$

$$w = 243.17 \text{ Btu/lbm}$$

This example demonstrates that potential and kinetic energy terms are insignificant for a turbine, since the Δp_e and Δk_e values are less than 1 Btu/lbm.

When the system (the fluid being studied) changes its properties (temperature, pressure, volume) from one value to another as a consequence of work or heat or internal energy exchange, then it is said that the fluid has gone through a "process." In some processes, the relationships between pressure, temperature, and volume are specified as the fluid goes from one thermodynamic state to another. The most common processes are those in which the temperature, pressure, or volume is held constant during the process. These would be classified as isothermal, isobaric, or isovolumetric processes, respectively. Iso means "constant or one." If the fluid passes through various processes and then eventually returns to the same state it began with, the system is said to have undergone a cyclic process. One such cyclic process used is the Rankine cycle, two examples of which are shown in Figure 19.

The processes that comprise the cycle are described below.

- ab: Liquid is compressed with no change in entropy (by ideal pump).
- bc: Constant pressure transfer of heat in the boiler. Heat is added to the compressed liquid, two-phase, and superheat states.
- cd: Constant entropy expansion with shaft work output (in ideal turbine).
- da: Constant pressure transfer of heat in the sink. Unavailable heat is rejected to the heat sink (condenser).

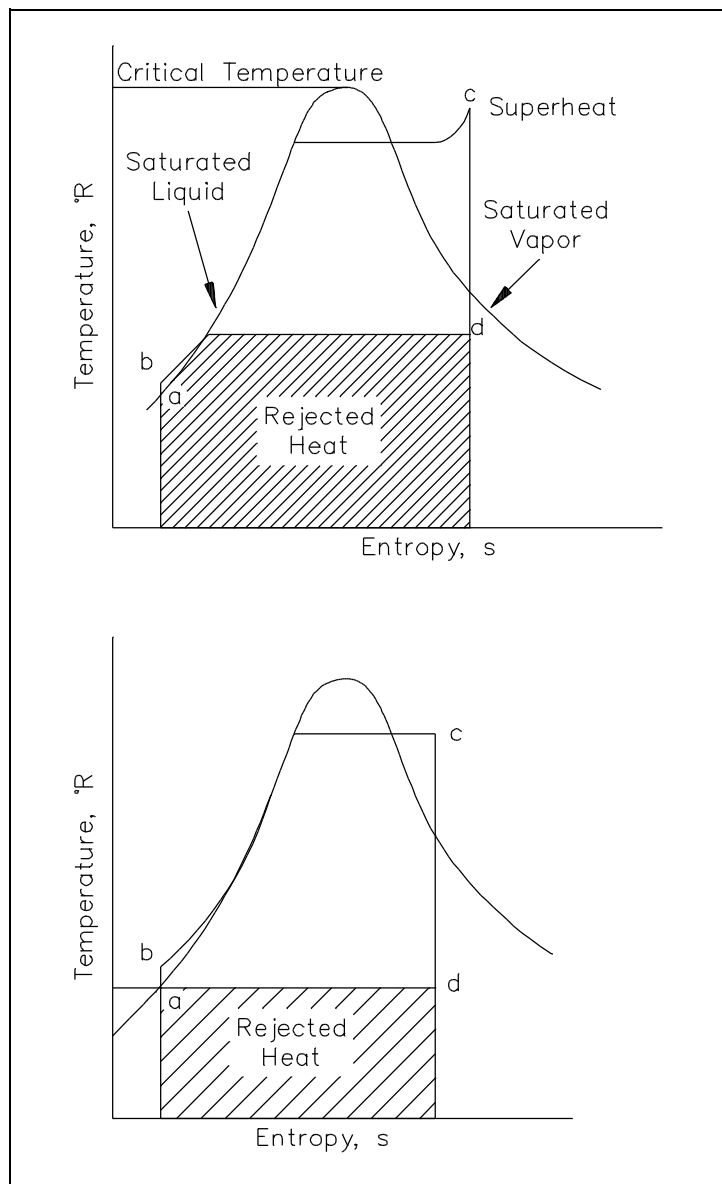


Figure 19 T-s Diagram with Rankine Cycles

Note the individual processes the fluid must go through before completing the complete cycle. Rankine cycles will be discussed in greater detail later in this module. Figure 20 shows a typical steam plant cycle. Heat is supplied to the steam generator (boiler) where liquid is converted to steam or vapor. The vapor is then expanded adiabatically in the turbine to produce a work output. Vapor leaving the turbine then enters the condenser where heat is removed and the vapor is condensed into the liquid state. The condensation process is the heat-rejection mechanism for the cycle. Saturated liquid is delivered to the condensate pump and then the feed pump where its pressure is raised to the saturation pressure corresponding to the steam generator temperature, and the high pressure liquid is delivered to the steam generator where the cycle repeats itself.

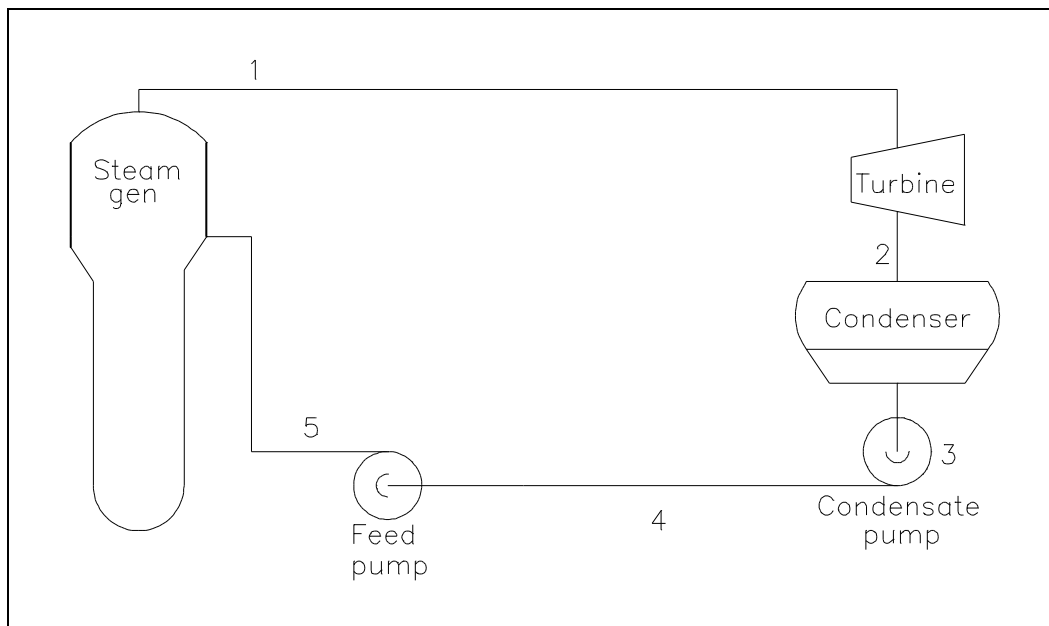


Figure 20 Typical Steam Plant Cycle

With the example complete, it seems appropriate to discuss the various components of a typical steam plant system. Although such a system is extremely complex, only the major components will be discussed. A typical steam plant system consists of: a heat source to produce the thermal energy (e.g. nuclear or fossil fuel); a steam generator to change the thermal energy into steam energy (a complete steam plant usually exists in connection with the steam generator in converting the steam into eventual electrical energy); pumps to transfer the fluid back to the heat source (reactor coolant pumps in a nuclear reactor); a pressurizer to ensure that the primary system maintains its desired pressure; and the necessary piping to ensure the fluid passes through each stage of its cyclic process. Of necessity, the steam plant is a large "closed" system. However, each component of the system is thermodynamically analyzed as an open system as the fluid passes through it. Of primary importance is the process of dissipating the energy created by the heat source. This process takes place in the steam generator, which acts as a giant two-phase heat generator.

The hot fluid from the heat source passes through the primary side of the steam generator where its energy is passed to the secondary side of the heat exchanger in such a manner as to create steam. The fluid, with its energy removed from the primary side, leaves the steam generator at a lower temperature, and is pumped back to the heat source to be "re-heated." Each major component of a steam plant can be treated as a separate open system problem. A thermodynamic analysis, using the various forms of energies discussed, can be applied to the particular component in studying its behavior. A simplified example of the thermodynamics involved in the steam generator is shown below.

Example 2: Open System - Steam Plant Component

Primary fluid enters the heat exchanger of a nuclear facility at 610°F and leaves at 540°F. The flow rate is approximately 1.38×10^8 lbm/hr. If the specific heat of the fluid is taken as 1.5 Btu/lbm-°F, what is the heat transferred out of the steam generator?

Solution:

$$\dot{m}_{in}(h + pe + ke_{in}) + \dot{Q} = \dot{m}_{out}(h + pe + ke_{out}) + \dot{W}$$

- 1) Neglecting pe and ke and assuming no work is done on the system.

$$\dot{m}(h_{in}) + \dot{Q} = \dot{m}(h_{out})$$

$$\dot{Q} = \dot{m}(h_{out} - h_{in})$$

- 2) Substituting $\dot{Q} = \dot{m}c_p\Delta T$ where c_p = specific heat capacity (Btu/lbm-°F).

$$= \dot{m}(c_p)(T_{out} - T_{in})$$

$$= 1.38 \times 10^8 \text{ lbm/hr} (1.5 \text{ Btu/lbm-°F}) (540 - 610^\circ\text{F})$$

$$\dot{Q} = -1.45 \times 10^{10} \text{ Btu/hr}$$

The minus sign indicating heat out of the heat exchanger, which is consistent with the physical case. This example demonstrates that for a heat exchanger, the heat transfer rate can be calculated using the equation $\dot{Q} = \dot{m}(h_{out}-h_{in})$, or $\dot{Q} = \dot{m}c_p\Delta T$. It is important to note that the later equation can only be used when no phase change occurs since $\Delta T = 0$ during a phase change. The first equation can be used for a phase change heat transfer process as well as for latent heat calculations.

The pumps used for returning the fluid to the heat source can be analyzed as a thermodynamic system also. One such example is illustrated in Example 3.

Example 3: Open System - Coolant

A power pump is used to return the fluid from the heat exchanger back to the core. The flow rate through the pump is about 3.0×10^7 lbm/hr with the fluid entering the pump as saturated liquid at 540°F. The pressure rise across the pump is 90 psia. What is the work of the pump, neglecting heat losses and changes in potential and kinetic energy?

Solution:

$$\dot{m}(h_{in} + pe_{in} + ke_{in}) + \dot{Q} = \dot{m}(h_{out} + pe_{out} + ke_{out}) + \dot{W}$$

- 1) Assume $\dot{Q} = 0$ and neglect changes in pe and ke

$$\dot{m}(h_{in}) = \dot{m}(h_{out}) + \dot{W}$$

- 2) $\dot{W} = \dot{m} (h_{in} - h_{out})$ where \dot{W} is the rate of doing work by the pump

$$h_{in} = u_{in} + vP_{in}$$

$$h_{out} = u_{out} + vP_{out}$$

$$(h_{in} - h_{out}) = (u_{in} - u_{out}) + (vP_{in} - vP_{out}) = \Delta u + (vP_{in} - vP_{out})$$

- 3) Since no heat is transferred, $\Delta u = 0$ and the specific volume out of the pump is the same as the specific volume entering since water is incompressible.

$$(h_{in} - h_{out}) = v(P_{in} - P_{out})$$

- 4) Substituting the expression for work, $\dot{W} = \dot{m} (h_{in} - h_{out})$ we have:

$$\dot{W} = \dot{m} v(P_{in} - P_{out}).$$

- 5) Using 0.01246 for specific volume.

$$\dot{W} = 3.0 \times 10^7 \text{ lbm/hr} (0.01246 \text{ ft}^3/\text{lbm}) (-90 \text{ psia}) (144 \text{ in}^2/\text{ft}^2)/778 \text{ ft-lbf/Btu}$$

$$\dot{W} = -6.23 \times 10^6 \text{ Btu/hr or } -2446 \text{ hp}$$

Note: The minus sign indicating work put into the fluid by the pump. 1 hp = 2545 Btu/hr.

A thermodynamic balance across the reactor core gives an indication of the amount of heat removed by the coolant that is given off by the fuel rods.

Example 4: Thermodynamic Balance across Heat Source

In a particular facility, the temperature leaving the reactor core is 612°F, while that entering the core is 542°F. The coolant flow through the heat source is 1.32×10^8 lbm/hr. The c_p of the fluid averages 1.47 Btu/lbm°F. How much heat is being removed from the heat source? The pe and ke energies are small compared to other terms and may be neglected.

Solution:

$$\dot{m}(h + pe + ke)_{in} + \dot{Q} = \dot{m}(h + pe + ke)_{out} + \dot{W}$$

$$\dot{Q} = \dot{m}(h_{out} - h_{in})$$

1) Substituting $\dot{Q} = \dot{m}c_p\Delta T$ where c_p = specific heat capacity.

$$\dot{Q} = \dot{m}(c_p) (T_{out} - T_{in})$$

$$\dot{Q} = 1.32 \times 10^8 \text{ lbm/hr} (1.47 \text{ Btu/lbm } ^\circ\text{F}) (612 - 542^\circ\text{F})$$

$$\dot{Q} = 1.36 \times 10^{10} \text{ Btu/hr}$$

For this example $\dot{Q} = \dot{m}c_p\Delta T$ has been used to calculate the heat transfer rate since no phase change has occurred. However, $\dot{Q} = \dot{m}(h_{out}-h_{in})$ could also have been used had the problem data included inlet and outlet enthalpies.

The individual principal components of a reactor system have been thermodynamically analyzed. If all components were combined into an overall system, the system could be analyzed as a "closed" system problem. Such an analysis is illustrated in the following example.

Example 5: Overall Thermodynamic Balance

A nuclear facility (primary side) is to be studied as a complete system. The heat produced by the heat source is 1.36×10^{10} Btu/hr. The heat removed by the heat exchanger (steam generator) is 1.361×10^{10} Btu/hr. What is the required pump power to maintain a stable temperature?

Solution:

\dot{W}_p = pump work, \dot{Q}_c = heat produced by the heat source, \dot{Q}_s = heat transferred into steam generator

$$\dot{m}(h + pe + ke) + \dot{W}_p + \dot{Q}_c = \dot{Q}_s + \dot{m}(h + pe + ke)$$

- 1) For a closed system, the mass entering and leaving the system is zero, therefore, \dot{m} is constant. The energy entering and leaving the system is zero, and you can assume that the ke and pe are constant so that:

$$\dot{Q}_c + \dot{W}_p = \dot{Q}_s$$

- 2)
$$\begin{aligned} \dot{W}_p &= \dot{Q}_s - \dot{Q}_c \\ &= 1.361 \times 10^{10} \text{ Btu/hr} - 1.36 \times 10^{10} \text{ Btu/hr} \\ &= 1.0 \times 10^7 \text{ Btu/hr} \end{aligned}$$

$$\dot{W}_p = 4007 \text{ hp}$$

Of the examples just completed, emphasis should be placed on the heat exchanger analysis. Both the primary side and the secondary side have their own energy balances as the heat energy is transferred from one fluid to the other. In calculating heat exchanger heat transfer rates, we found that we could use the equations $\dot{Q} = \dot{m}_p \Delta h$.

Perhaps a short analysis of the secondary side of the heat exchanger will aid in understanding the heat exchanger's importance in the energy conversion process.

Example 6: Secondary Side of Heat Exchanger

Steam flows through a condenser at 2.0×10^6 kg/hr, entering as saturated vapor at 40°C ($h = 2574$ kJ/kg), and leaving at the same pressure as subcooled liquid at 30°C ($h = 125.8$ kJ/kg). Cooling water is available at 18°C ($h = 75.6$ kJ/kg). Environmental requirements limit the exit temperature to 25°C ($h = 104.9$ kJ/kg). Determine the required cooling water flow rate.

Solution:

Thermal balance gives the following:

$$\dot{Q}_{\text{stm}} = - \dot{Q}_{\text{cw}}$$

$$\dot{m}_{\text{stm}}(h_{\text{out}} - h_{\text{in}})_{\text{stm}} = - \dot{m}_{\text{cw}}(h_{\text{out}} - h_{\text{in}})_{\text{cw}}$$

$$\dot{m}_{\text{cw}} = - \dot{m}_{\text{stm}}(h_{\text{out}} - h_{\text{in}})_{\text{stm}} / (h_{\text{out}} - h_{\text{in}})_{\text{cw}}$$

$$= 2.0 \times 10^6 \text{ kg/hr} (125.8 - 2574 \text{ kJ/kg}) / (104.9 - 75.6 \text{ kJ/kg})$$

$$\dot{m}_{\text{cw}} = 1.67 \times 10^8 \text{ kg/hr}$$

In this example, we calculated the flow rate using the equation $\dot{Q} = \dot{m}\Delta h$ since a phase change occurred when the steam was condensed to liquid water. $\dot{Q} = \dot{m}c_p\Delta T$ would not have worked since $\Delta T=0$ for a phase change. Had we attempted to solve the problem using $\dot{Q} = \dot{m}c_p\Delta T$, we would have discovered that an error occurs since the $\Delta T = 10^\circ\text{C}$ is the ΔT needed to subcool the liquid from saturation at 40°C to a subcooled value of 30°C . Therefore, the heat transfer process to condense the steam to a saturated liquid has not been taken into account.

Summary

The important information from this chapter is summarized below.

First Law of Thermodynamics Summary

- The First Law of Thermodynamics states that energy can neither be created nor destroyed, only altered in form.
- In analyzing an open system using the First Law of Thermodynamics, the energy into the system is equal to the energy leaving the system.
- If the fluid passes through various processes and then eventually returns to the same state it began with, the system is said to have undergone a cyclic process. The first law is used to analyze a cyclic process.
- The energy entering any component is equal to the energy leaving that component at steady state.
- The amount of energy transferred across a heat exchanger is dependent upon the temperature of the fluid entering the heat exchanger from both sides and the flow rates of these fluids.
- A T-s diagram can be used to represent thermodynamic processes.

SECOND LAW OF THERMODYNAMICS

The Second Law of Thermodynamics is used to determine the maximum efficiency of any process. A comparison can then be made between the maximum possible efficiency and the actual efficiency obtained.

- EO 1.25** **STATE** the Second Law of Thermodynamics.
- EO 1.26** **Using** the Second Law of Thermodynamics, **DETERMINE** the maximum possible efficiency of a system.
- EO 1.27** **Given** a thermodynamic system, **CONDUCT** an analysis using the Second Law of Thermodynamics.
- EO 1.28** **Given** a thermodynamic system, **DESCRIBE** the method used to determine:
- a. **The maximum efficiency of the system**
 - b. **The efficiency of the components within the system**
- EO 1.29** **DIFFERENTIATE** between the path for an ideal process and that for a real process on a T-s or h-s diagram.
- EO 1.30** **Given** a T-s or h-s diagram for a system **EVALUATE**:
- a. **System efficiencies**
 - b. **Component efficiencies**
- EO 1.31** **DESCRIBE** how individual factors affect system or component efficiency.
-

Second Law of Thermodynamics

One of the earliest statements of the Second Law of Thermodynamics was made by R. Clausius in 1850. He stated the following.

It is impossible to construct a device that operates in a cycle and produces no effect other than the removal of heat from a body at one temperature and the absorption of an equal quantity of heat by a body at a higher temperature.

With the Second Law of Thermodynamics, the limitations imposed on any process can be studied to determine the maximum possible efficiencies of such a process and then a comparison can be made between the maximum possible efficiency and the actual efficiency achieved. One of the areas of application of the second law is the study of energy-conversion systems. For example, it is not possible to convert all the energy obtained from a nuclear reactor into electrical energy. There must be losses in the conversion process. The second law can be used to derive an expression for the maximum possible energy conversion efficiency taking those losses into account. Therefore, the second law denies the possibility of completely converting into work all of the heat supplied to a system operating in a cycle, no matter how perfectly designed the system may be. The concept of the second law is best stated using Max Planck's description:

It is impossible to construct an engine that will work in a complete cycle and produce no other effect except the raising of a weight and the cooling of a heat reservoir.

The Second Law of Thermodynamics is needed because the First Law of Thermodynamics does not define the energy conversion process completely. The first law is used to relate and to evaluate the various energies involved in a process. However, no information about the direction of the process can be obtained by the application of the first law. Early in the development of the science of thermodynamics, investigators noted that while work could be converted completely into heat, the converse was never true for a cyclic process. Certain natural processes were also observed always to proceed in a certain direction (e.g., heat transfer occurs from a hot to a cold body). The second law was developed as an explanation of these natural phenomena.

Entropy

One consequence of the second law is the development of the physical property of matter termed entropy (S). Entropy was introduced to help explain the Second Law of Thermodynamics. The change in this property is used to determine the direction in which a given process will proceed. Entropy can also be explained as a measure of the unavailability of heat to perform work in a cycle. This relates to the second law since the second law predicts that not all heat provided to a cycle can be transformed into an equal amount of work, some heat rejection must take place. The change in entropy is defined as the ratio of heat transferred during a reversible process to the absolute temperature of the system.

$$\Delta S = \frac{\Delta Q}{T_{\text{abs}}} \quad (\text{For a reversible process})$$

where

ΔS = the change in entropy of a system during some process (Btu/°R)

ΔQ = the amount of heat added to the system during the process (Btu)

T_{abs} = the absolute temperature at which the heat was transferred (°R)

The second law can also be expressed as $\Delta S \geq 0$ for a closed cycle. In other words, entropy must increase or stay the same for a cyclic system; it can never decrease.

Entropy is a property of a system. It is an extensive property that, like the total internal energy or total enthalpy, may be calculated from specific entropies based on a unit mass quantity of the system, so that $S = ms$. For pure substances, values of the specific entropy may be tabulated along with specific enthalpy, specific volume, and other thermodynamic properties of interest. One place to find this tabulated information is in the steam tables described in a previous chapter (refer back to Figure 19).

Specific entropy, because it is a property, is advantageously used as one of the coordinates when representing a reversible process graphically. The area under a reversible process curve on the T-s diagram represents the quantity of heat transferred during the process.

Thermodynamic problems, processes, and cycles are often investigated by substitution of reversible processes for the actual irreversible process to aid the student in a second law analysis. This substitution is especially helpful because only reversible processes can be depicted on the diagrams (h-s and T-s, for example) used for the analysis. Actual or irreversible processes cannot be drawn since they are not a succession of equilibrium conditions. Only the initial and final conditions of irreversible processes are known; however, some thermodynamics texts represent an irreversible process by dotted lines on the diagrams.

Carnot's Principle

With the practice of using reversible processes, Sadi Carnot in 1824 advanced the study of the second law by disclosing a principle consisting of the following propositions.

1. No engine can be more efficient than a reversible engine operating between the same high temperature and low temperature reservoirs. Here the term heat reservoir is taken to mean either a heat source or a heat sink.
2. The efficiencies of all reversible engines operating between the same constant temperature reservoirs are the same.
3. The efficiency of a reversible engine depends only upon the temperatures of the heat source and heat receiver.

Carnot Cycle

The above principle is best demonstrated with a simple cycle (shown in Figure 21) and an example of a proposed heat power cycle. The cycle consists of the following reversible processes.

- 1-2: adiabatic compression from T_C to T_H due to work performed on fluid.
- 2-3: isothermal expansion as fluid expands when heat is added to the fluid at temperature T_H .
- 3-4: adiabatic expansion as the fluid performs work during the expansion process and temperature drops from T_H to T_C .
- 4-1: isothermal compression as the fluid contracts when heat is removed from the fluid at temperature T_C .

This cycle is known as a Carnot Cycle. The heat input (Q_H) in a Carnot Cycle is graphically represented on Figure 21 as the area under line 2-3. The heat rejected (Q_C) is graphically represented as the area under line 1-4. The difference between the heat added and the heat rejected is the net work (sum of all work processes), which is represented as the area of rectangle 1-2-3-4.

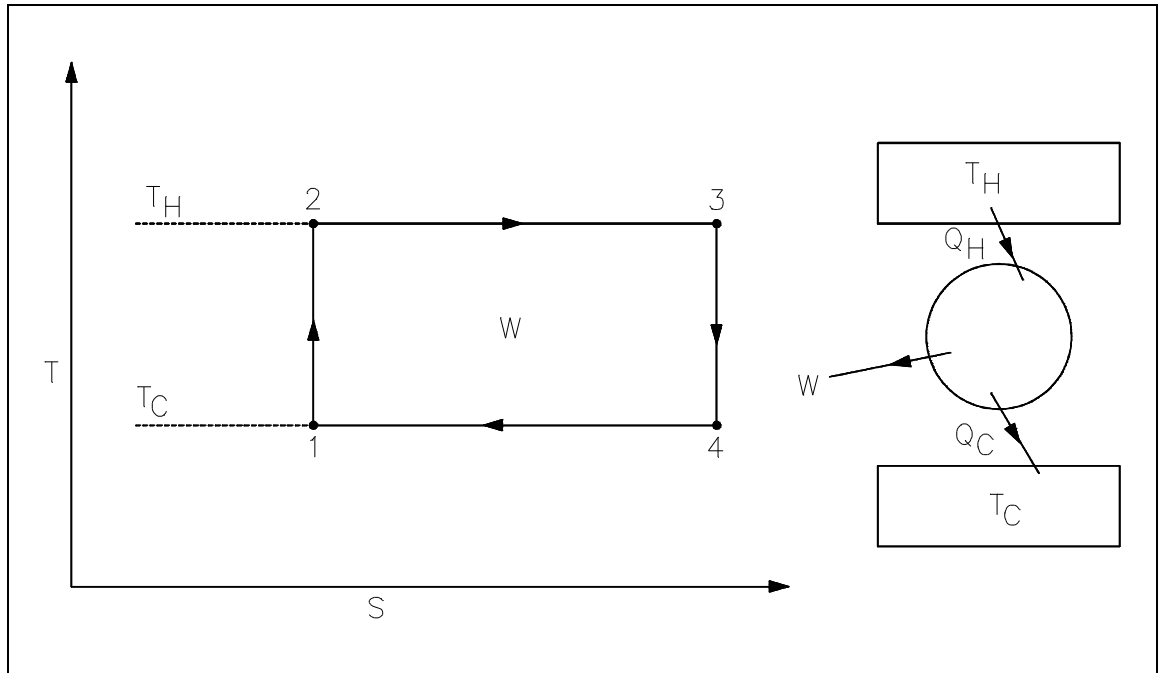


Figure 21 Carnot Cycle Representation

The efficiency (η) of the cycle is the ratio of the net work of the cycle to the heat input to the cycle. This ratio can be expressed by the following equation.

$$\begin{aligned} \eta &= (Q_H - Q_C)/Q_H = (T_H - T_C)/T_H \\ &= 1 - (T_C/T_H) \end{aligned} \tag{1-23}$$

where:

- η = cycle efficiency
- T_C = designates the low-temperature reservoir ($^{\circ}\text{R}$)
- T_H = designates the high-temperature reservoir ($^{\circ}\text{R}$)

Equation 1-23 shows that the maximum possible efficiency exists when T_H is at its largest possible value or when T_C is at its smallest value. Since all practical systems and processes are really irreversible, the above efficiency represents an upper limit of efficiency for any given system operating between the same two temperatures. The system's maximum possible efficiency would be that of a Carnot efficiency, but because Carnot efficiencies represent reversible processes, the actual system will not reach this efficiency value. Thus, the Carnot efficiency serves as an unattainable upper limit for any real system's efficiency. The following example demonstrates the above principles.

Example 1: Carnot Efficiency

An inventor claims to have an engine that receives 100 Btu of heat and produces 25 Btu of useful work when operating between a source at 140°F and a receiver at 0°F. Is the claim a valid claim?

Solution:

$$T_H = 140^\circ\text{F} + 460 = 600^\circ\text{R}$$

$$T_C = 0^\circ\text{F} + 460 = 460^\circ\text{R}$$

$$\eta = (600-460)/600 \times 100 = 23.3\%$$

$$\text{Claimed efficiency} = 25/100 = 25\%$$

Therefore, the claim is invalid.

The most important aspect of the second law for our practical purposes is the determination of maximum possible efficiencies obtained from a power system. Actual efficiencies will always be less than this maximum. The losses (friction, for example) in the system and the fact that systems are not truly reversible preclude us from obtaining the maximum possible efficiency. An illustration of the difference that may exist between the ideal and actual efficiency is presented in Figure 22 and the following example.

Example 2: Actual vs. Ideal Efficiency

The actual efficiency of a steam cycle is 18.0%. The facility operates from a steam source at 340°F and rejects heat to atmosphere at 60°F. Compare the Carnot efficiency to the actual efficiency.

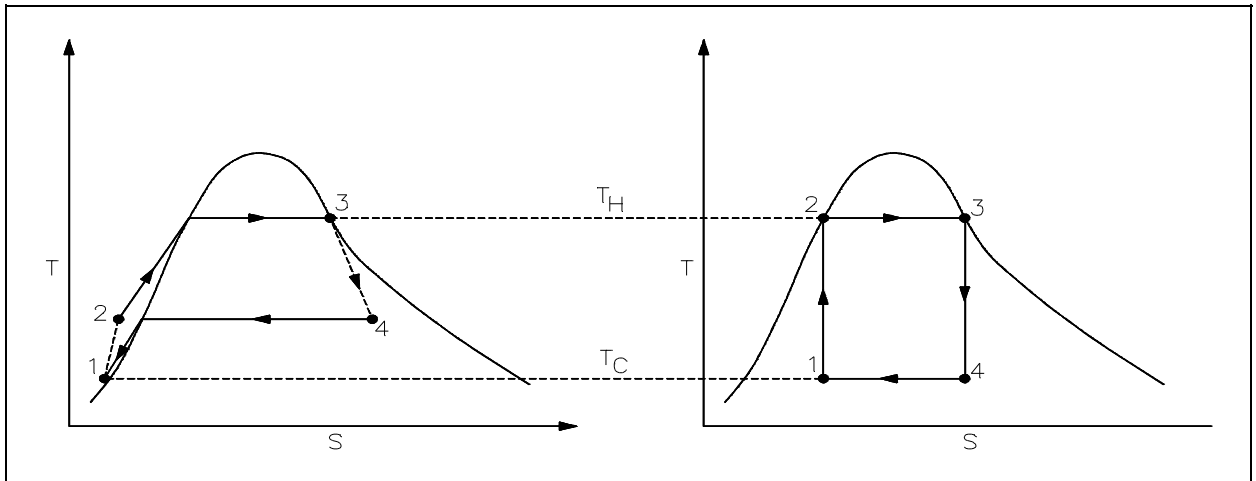


Figure 22 Real Process Cycle Compared to Carnot Cycle

Solution:

$$\begin{aligned} \eta &= 1 - (T_c / T_h) \\ \eta &= 1 - (60 + 460)/(340 + 460) \\ &= 1 - 520/800 \\ &= 35\% \quad \text{as compared to 18.0\% actual efficiency.} \end{aligned}$$

An open system analysis was performed using the First Law of Thermodynamics in the previous chapter. The second law problems are treated in much the same manner; that is, an isolated, closed, or open system is used in the analysis depending upon the types of energy that cross the boundary. As with the first law, the open system analysis using the second law equations is the more general case, with the closed and isolated systems being "special" cases of the open system. The solution to second law problems is very similar to the approach used in the first law analysis.

Figure 23 illustrates the control volume from the viewpoint of the second law. In this diagram, the fluid moves through the control volume from section in to section out while work is delivered external to the control volume. We assume that the boundary of the control volume is at some environmental temperature and that all of the heat transfer (Q) occurs at this boundary. We have already noted that entropy is a property, so it may be transported with the flow of the fluid into and out of the control volume, just like enthalpy or internal energy. The entropy flow into the control volume resulting from mass transport is, therefore, $\dot{m}_{in}s_{in}$, and the entropy flow out of the control volume is $\dot{m}_{out}s_{out}$, assuming that the properties are uniform at

sections in and out. Entropy may also be added to the control volume because of heat transfer at the boundary of the control volume.

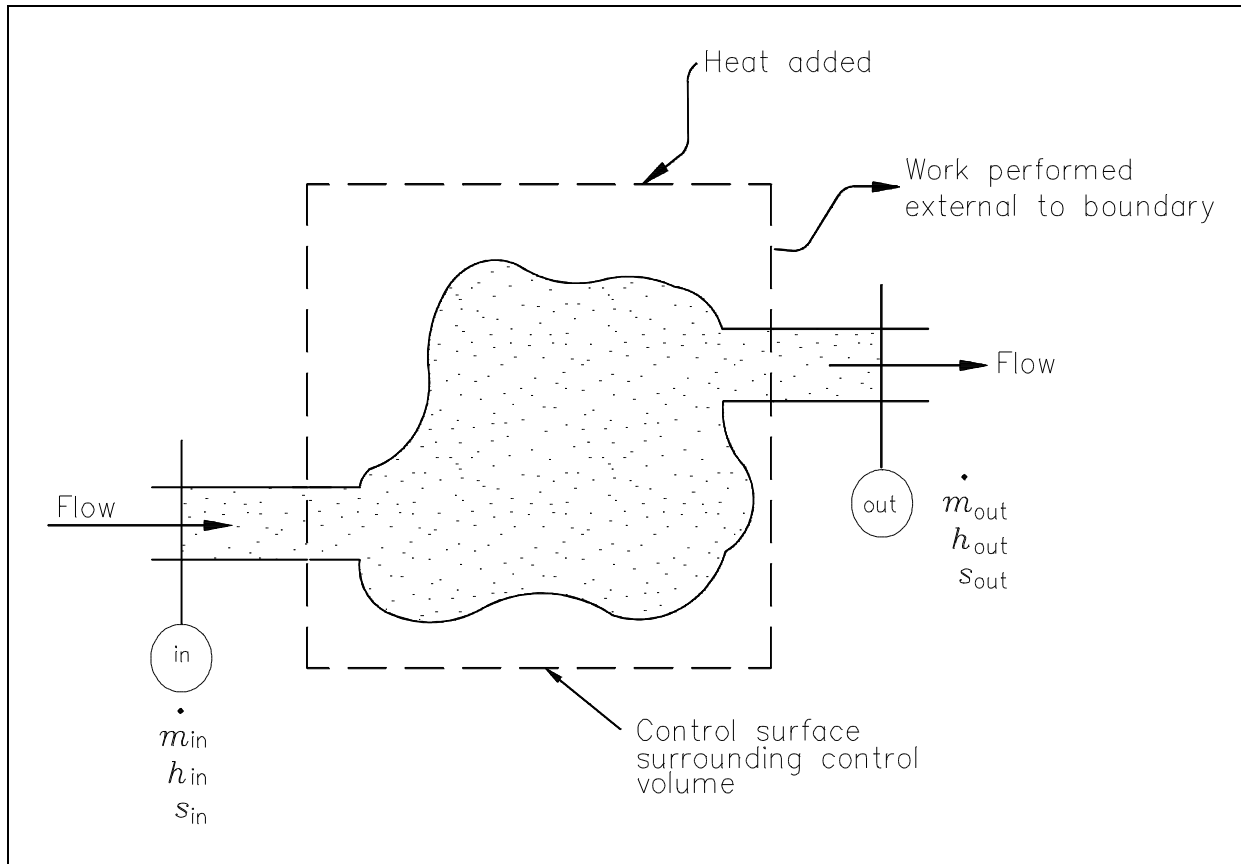


Figure 23 Control Volume for Second Law Analysis

A simple demonstration of the use of this form of system in second law analysis will give the student a better understanding of its use.

Example 3: Open System Second Law

Steam enters the nozzle of a steam turbine with a velocity of 10 ft/sec at a pressure of 100 psia and temperature of 500°F at the nozzle discharge. The pressure and temperature are 1 atm at 300°F. What is the increase in entropy for the system if the mass flow rate is 10,000 lbm/hr?

Solution:

$$\dot{m}s_{\text{in}} + \dot{p} = \dot{m}s_{\text{out}} \quad \text{where } \dot{p} = \text{entropy added to the system}$$

$$\dot{p} = \dot{m} (s_{\text{out}} - s_{\text{in}})$$

$$s_{\text{in}} = 1.7088 \text{ Btu/lbm } ^\circ\text{R} \quad (\text{from steam tables})$$

$$s_{\text{out}} = 1.8158 \text{ Btu/lbm } ^\circ\text{R} \quad (\text{from steam tables})$$

$$\dot{p}/\dot{m} = s_{\text{out}} - s_{\text{in}} = 1.8158 - 1.7088 \text{ Btu/lbm } ^\circ\text{R}$$

$$\dot{p}/\dot{m} = 0.107 \text{ Btu/lbm } ^\circ\text{R}$$

$$\dot{p} = 10,000 (0.107)$$

$$\dot{p} = 1070 \text{ Btu/lbm } ^\circ\text{R}. = \text{entropy added to the system}$$

It should always be kept in mind that the Second Law of Thermodynamics gives an upper limit (which is never reached in physical systems) to how efficiently a thermodynamic system can perform. A determination of that efficiency is as simple as knowing the inlet and exit temperatures of the overall system (one that works in a cycle) and applying Carnot's efficiency equation using these temperatures in absolute degrees.

Diagrams of Ideal and Real Processes

Any ideal thermodynamic process can be drawn as a path on a property diagram, such as a T-s or an h-s diagram. A real process that approximates the ideal process can also be represented on the same diagrams (usually with the use of dashed lines).

In an ideal process involving either a reversible expansion or a reversible compression, the entropy will be constant. These isentropic processes will be represented by vertical lines on either T-s or h-s diagrams, since entropy is on the horizontal axis and its value does not change. A real expansion or compression process operating between the same pressures as the ideal process will look much the same, but the dashed lines representing the real process will slant slightly towards the right since the entropy will increase from the start to the end of the process. Figures 24 and 25 show ideal and real expansion and compression processes on T-s and h-s diagrams.

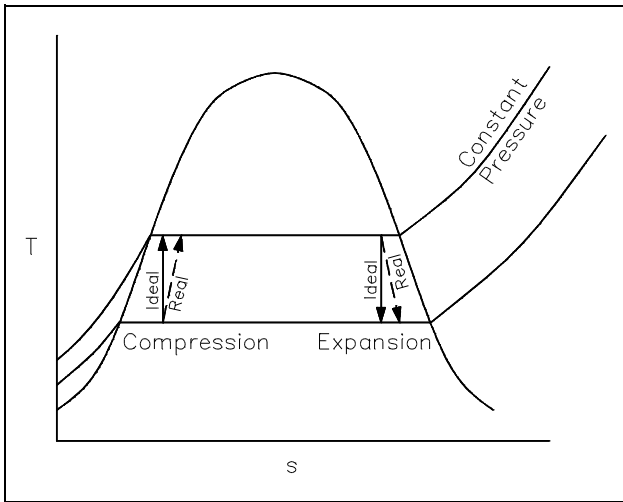


Figure 24 Expansion and Compression Processes on T-s Diagram

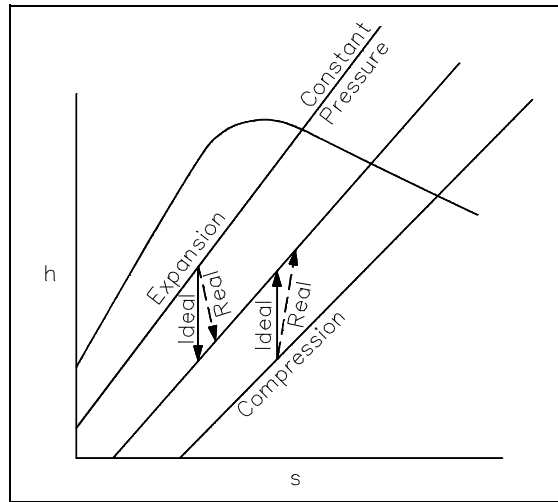


Figure 25 Expansion and Compression Processes on h-s Diagram

Power Plant Components

In order to analyze a complete power plant steam power cycle, it is first necessary to analyze the elements which make up such cycles. (See Figure 26) Although specific designs differ, there are three basic types of elements in power cycles, (1) turbines, (2) pumps and (3) heat exchangers. Associated with each of these three types of elements is a characteristic change in the properties of the working fluid.

Previously we have calculated system efficiency by knowing the temperature of the heat source and the heat sink. It is also possible to calculate the efficiencies of each individual component.

The efficiency of each type of component can be calculated by comparing the actual work produced by the component to the work that would have been produced by an ideal component operating isentropically between the same inlet and outlet conditions.

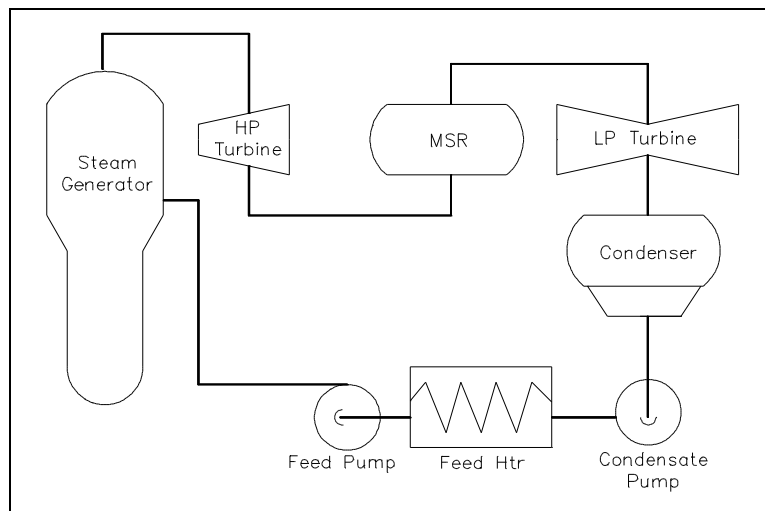


Figure 26 Steam Cycle

A steam turbine is designed to extract energy from the working fluid (steam) and use it to do work in the form of rotating the turbine shaft. The working fluid does work as it expands through the turbine. The shaft work is then converted to electrical energy by the generator. In the application of the first law, general energy equation to a simple turbine under steady flow conditions, it is found that the decrease in the enthalpy of the working fluid $H_{in} - H_{out}$ equals the work done by the working fluid in the turbine (W_t).

$$H_{in} - H_{out} = W_t \quad (1-24)$$

$$\dot{m}(h_{in} - h_{out}) = \dot{w}_t \quad (1-25)$$

where:

- H_{in} = enthalpy of the working fluid entering the turbine (Btu)
- H_{out} = enthalpy of the working fluid leaving the turbine (Btu)
- W_t = work done by the turbine (ft-lb_f)
- \dot{m} = mass flow rate of the working fluid (lb_m/hr)
- h_{in} = specific enthalpy of the working fluid entering the turbine (Btu/lbm)
- h_{out} = specific enthalpy of the working fluid leaving the turbine (Btu/lbm)
- \dot{w}_t = power of turbine (Btu/hr)

These relationships apply when the kinetic and potential energy changes and the heat losses of the working fluid while in the turbine are negligible. For most practical applications, these are valid assumptions. However, to apply these relationships, one additional definition is necessary. The steady flow performance of a turbine is idealized by assuming that in an ideal case the working fluid does work reversibly by expanding at a constant entropy. This defines the so-called ideal turbine. In an ideal turbine, the entropy of the working fluid entering the turbine S_{in} equals the entropy of the working fluid leaving the turbine.

$$S_{in} = S_{out}$$

$$s_{in} = s_{out}$$

where:

- S_{in} = entropy of the working fluid entering the turbine (Btu/°R)
- S_{out} = entropy of the working fluid leaving the turbine (Btu/°R)
- s_{in} = specific entropy of the working fluid entering the turbine (Btu/lbm -°R)
- s_{out} = specific entropy of the working fluid leaving the turbine (Btu/lbm -°R)

The reason for defining an ideal turbine is to provide a basis for analyzing the performance of turbines. An ideal turbine performs the maximum amount of work theoretically possible.

An actual turbine does less work because of friction losses in the blades, leakage past the blades and, to a lesser extent, mechanical friction. Turbine efficiency η_t , sometimes called isentropic turbine efficiency because an ideal turbine is defined as one which operates at constant entropy, is defined as the ratio of the actual work done by the turbine $W_{t, \text{actual}}$ to the work that would be done by the turbine if it were an ideal turbine $W_{t, \text{ideal}}$.

$$\eta_t = \frac{W_{t, \text{actual}}}{W_{t, \text{ideal}}} \quad (1-26)$$

$$\eta = \frac{(h_{\text{in}} - h_{\text{out}})_{\text{actual}}}{(h_{\text{in}} - h_{\text{out}})_{\text{ideal}}} \quad (1-27)$$

where:

- η_t = turbine efficiency (no units)
- $W_{t, \text{actual}}$ = actual work done by the turbine (ft-lbf)
- $W_{t, \text{ideal}}$ = work done by an ideal turbine (ft-lbf)
- $(h_{\text{in}} - h_{\text{out}})_{\text{actual}}$ = actual enthalpy change of the working fluid (Btu/lbm)
- $(h_{\text{in}} - h_{\text{out}})_{\text{ideal}}$ = actual enthalpy change of the working fluid in an ideal turbine (Btu/lbm)

In many cases, the turbine efficiency η_t has been determined independently. This permits the actual work done to be calculated directly by multiplying the turbine efficiency η_t by the work done by an ideal turbine under the same conditions. For small turbines, the turbine efficiency is generally 60% to 80%; for large turbines, it is generally about 90%.

The actual and idealized performances of a turbine may be compared conveniently using a T-s diagram. Figure 27 shows such a comparison. The ideal case is a constant entropy. It is represented by a vertical line on the T-s diagram. The actual turbine involves an increase in entropy. The smaller the increase in entropy, the closer the turbine efficiency η_t is to 1.0 or 100%.

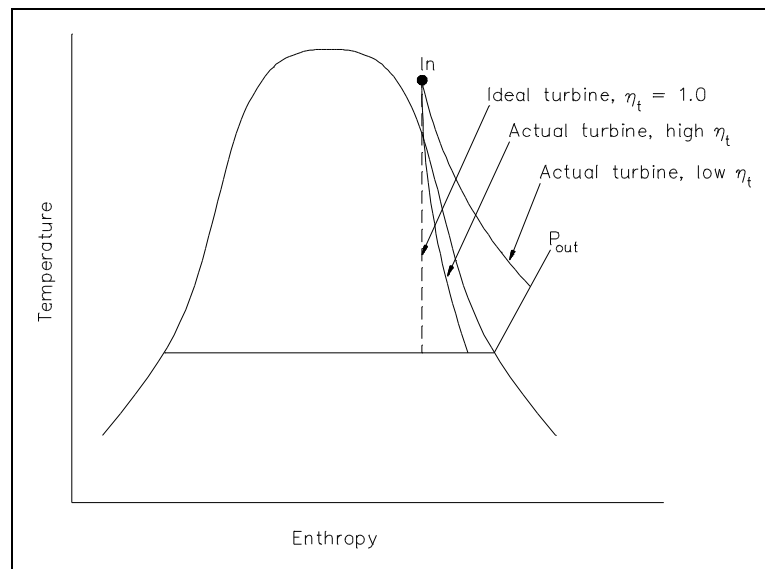


Figure 27 Comparison of Ideal and Actual Turbine Performances

A pump is designed to move the working fluid by doing work on it. In the application of the first law general energy equation to a simple pump under steady flow conditions, it is found that the increase in the enthalpy of the working fluid $H_{out} - H_{in}$ equals the work done by the pump, W_p , on the working fluid.

$$H_{out} - H_{in} = W_p \quad (1-28)$$

$$\dot{m}(h_{out} - h_{in}) = \dot{w}_p \quad (1-29)$$

where:

H_{out} = enthalpy of the working fluid leaving the pump (Btu)

H_{in} = enthalpy of the working fluid entering the pump (Btu)

W_p = work done by the pump on the working fluid (ft-lbf)

\dot{m} = mass flow rate of the working fluid (lbm/hr)

h_{out} = specific enthalpy of the working fluid leaving the pump (Btu/lbm)

h_{in} = specific enthalpy of the working fluid entering the pump (Btu/lbm)

\dot{w}_p = power of pump (Btu/hr)

These relationships apply when the kinetic and potential energy changes and the heat losses of the working fluid while in the pump are negligible. For most practical applications, these are valid assumptions. It is also assumed that the working fluid is incompressible. For the ideal case, it can be shown that the work done by the pump W_p is equal to the change in enthalpy across the ideal pump.

$$W_{p \text{ ideal}} = (H_{out} - H_{in})_{\text{ideal}} \quad (1-30)$$

$$\dot{w}_{p \text{ ideal}} = \dot{m} (h_{out} - h_{in})_{\text{ideal}} \quad (1-31)$$

where:

W_p = work done by the pump on the working fluid (ft-lbf)

H_{out} = enthalpy of the working fluid leaving the pump (Btu)

H_{in} = enthalpy of the working fluid entering the pump (Btu)

\dot{w}_p = power of pump (Btu/hr)

\dot{m} = mass flow rate of the working fluid (lbm/hr)

h_{out} = specific enthalpy of the working fluid leaving the pump (Btu/lbm)

h_{in} = specific enthalpy of the working fluid entering the pump (Btu/lbm)

The reason for defining an ideal pump is to provide a basis for analyzing the performance of actual pumps. A pump requires more work because of unavoidable losses due to friction and fluid turbulence. The work done by a pump W_p is equal to the change in enthalpy across the actual pump.

$$W_{p, \text{ actual}} = (H_{\text{out}} - H_{\text{in}})_{\text{actual}} \quad (1-32)$$

$$\dot{w}_{p, \text{ actual}} = \dot{m} (h_{\text{out}} - h_{\text{in}})_{\text{actual}} \quad (1-33)$$

Pump efficiency, η_p , is defined as the ratio of the work required by the pump if it were an ideal pump $w_{p, \text{ ideal}}$ to the actual work required by the pump $w_{p, \text{ actual}}$.

$$\eta_p = \frac{W_{p, \text{ ideal}}}{W_{p, \text{ actual}}} \quad (1-34)$$

Example:

A pump operating at 75% efficiency has an inlet specific enthalpy of 200 Btu/lbm. The exit specific enthalpy of the ideal pump is 600 Btu/lbm. What is the exit specific enthalpy of the actual pump?

Solution:

Using Equation 1-34:

$$\eta_p = \frac{W_{p, \text{ ideal}}}{W_{p, \text{ actual}}}$$

$$W_{p, \text{ actual}} = \frac{W_{p, \text{ ideal}}}{\eta_p}$$

$$(h_{\text{out}} - h_{\text{in}})_{\text{actual}} = \frac{(h_{\text{out}} - h_{\text{in}})_{\text{ideal}}}{\eta_p}$$

$$h_{\text{out, actual}} = \frac{(h_{\text{out}} - h_{\text{in}})_{\text{ideal}}}{\eta_p} + h_{\text{in, actual}}$$

$$h_{\text{out, actual}} = \frac{(600 \text{ Btu/lbm} - 200 \text{ Btu/lbm})}{.75} + 200 \text{ Btu/lbm}$$

$$h_{\text{out, actual}} = 533.3 \text{ Btu/lbm} + 200 \text{ Btu/lbm}$$

$$h_{\text{out, actual}} = 733.3 \text{ Btu/lbm}$$

Pump efficiency, η_p , relates the work required by an ideal pump to the actual work required by the pump; it relates the minimum amount of work theoretically possible to the actual work required by the pump. However, the work required by a pump is normally only an intermediate form of energy. Normally a motor or turbine is used to run the pump. Pump efficiency does not account for losses in this motor or turbine. An additional efficiency factor, motor efficiency η_m , is defined as the ratio of the actual work required by the pump to the electrical energy input to the pump motor, when both are expressed in the same units.

$$\eta_m = \frac{W_{p, \text{ actual}}}{W_{m, \text{ in}} C}$$

where:

η_m	=	motor efficiency (no units)
$W_{p, \text{ actual}}$	=	actual work required by the pump (ft-lbf)
$W_{m, \text{ in}}$	=	electrical energy input to the pump motor (kw-hr)
C	=	conversion factor = 2.655×10^6 ft-lbf/kw-hr

Like pump efficiency η_p , motor efficiency η_m is always less than 1.0 or 100% for an actual pump motor. The combination of pump efficiency η_p and motor efficiency η_m relates the ideal pump to the electrical energy input to the pump motor.

$$\eta_m \eta_p = \frac{W_{p, \text{ ideal}}}{W_{m, \text{ in}} C} \quad (1-35)$$

where:

η_m	=	motor efficiency (no units)
η_p	=	pump efficiency (no units)
$W_{p, \text{ ideal}}$	=	ideal work required by the pump (ft-lbf)
$W_{m, \text{ in}}$	=	electrical energy input to the pump motor (kw-hr)
C	=	conversion factor = 2.655×10^6 ft-lbf/kw-hr

A heat exchanger is designed to transfer heat between two working fluids. There are several heat exchangers used in power plant steam cycles. In the steam generator or boiler, the heat source (e.g., reactor coolant) is used to heat and vaporize the feedwater. In the condenser, the steam exhausting from the turbine is condensed before being returned to the steam generator. In addition to these two major heat exchangers, numerous smaller heat exchangers are used throughout the steam cycle. Two primary factors determine the rate of heat transfer and the temperature difference between the two fluids passing through the heat exchanger.

In the application of the first law general energy equation to a simple heat exchanger under steady flow conditions, it is found that the mass flow rates and enthalpies of the two fluids are related by the following relationship.

$$\dot{m}_1 (h_{\text{out},1} - h_{\text{in},1}) = -\dot{m}_2 (h_{\text{out},2} - h_{\text{in},2}) \quad (1-36)$$

where:

- \dot{m}_1 = mass flow rate of the working fluid 1 (lbm/hr)
- \dot{m}_2 = mass flow rate of the working fluid 2 (lbm/hr)
- $h_{\text{out},1}$ = specific enthalpy of the working fluid 1 leaving the heat exchanger (Btu/lbm)
- $h_{\text{in},1}$ = specific enthalpy of the working fluid 1 entering the heat exchanger (Btu/lbm)
- $h_{\text{out},2}$ = specific enthalpy of the working fluid 2 leaving the heat exchanger (Btu/lbm)
- $h_{\text{in},2}$ = specific enthalpy of the working fluid 2 entering the heat exchanger (Btu/lbm)

In the preceding sections we have discussed the Carnot cycle, cycle efficiencies, and component efficiencies. In this section we will apply this information to allow us to compare and evaluate various ideal and real cycles. This will allow us to determine how modifying a cycle will affect the cycle's available energy that can be extracted for work.

Since the efficiency of a Carnot cycle is solely dependent on the temperature of the heat source and the temperature of the heat sink, it follows that to improve a cycles' efficiency all we have to do is increase the temperature of the heat source and decrease the temperature of the heat sink. In the real world the ability to do this is limited by the following constraints.

1. For a real cycle the heat sink is limited by the fact that the "earth" is our final heat sink. And therefore, is fixed at about 60°F (520°R).
2. The heat source is limited to the combustion temperatures of the fuel to be burned or the maximum limits placed on nuclear fuels by their structural components (pellets, cladding etc.). In the case of fossil fuel cycles the upper limit is ~3040°F (3500°R). But even this temperature is not attainable due to the metallurgical restraints of the boilers, and therefore they are limited to about 1500°F (1960°R) for a maximum heat source temperature.

Using these limits to calculate the maximum efficiency attainable by an ideal Carnot cycle gives the following.

$$\eta = \frac{T_{\text{SOURCE}} - T_{\text{SINK}}}{T_{\text{SOURCE}}} = \frac{1960^{\circ}\text{R} - 520^{\circ}\text{R}}{1960^{\circ}\text{R}} = 73.5\%$$

This calculation indicates that the Carnot cycle, operating with ideal components under real world constraints, should convert almost 3/4 of the input heat into work. But, as will be shown, this ideal efficiency is well beyond the present capabilities of any real systems.

Heat Rejection

To understand why an efficiency of 73% is not possible we must analyze the Carnot cycle, then compare the cycle using real and ideal components. We will do this by looking at the T-s diagrams of Carnot cycles using both real and ideal components.

The energy added to a working fluid during the Carnot isothermal expansion is given by q_s . Not all of this energy is available for use by the heat engine since a portion of it (q_r) must be rejected to the environment. This is given by:

$$q_r = T_o \Delta s \text{ in units of Btu/lbm,} \tag{1-37}$$

where T_o is the average heat sink temperature of 520°R . The available energy (A.E.) for the Carnot cycle may be given as:

$$\text{A.E.} = q_s - q_r \tag{1-38}$$

Substituting equation 1-37 for q_r gives:

$$\text{A.E.} = q_s - T_o \Delta s \text{ in units of Btu/lbm.} \tag{1-39}$$

and is equal to the area of the shaded region labeled available energy in Figure 28 between the temperatures 1962° and 520°R . From Figure 28 it can be seen that any cycle operating at a temperature of less than 1962°R will be less efficient. Note that by developing materials capable of withstanding the stresses above 1962°R , we could greatly add to the energy available for use by the plant cycle.

From equation 1-37, one can see why the change in entropy can be defined as a measure of the energy unavailable to do work. If the temperature of the heat sink is known, then the change in entropy does correspond to a measure of the heat rejected by the engine.

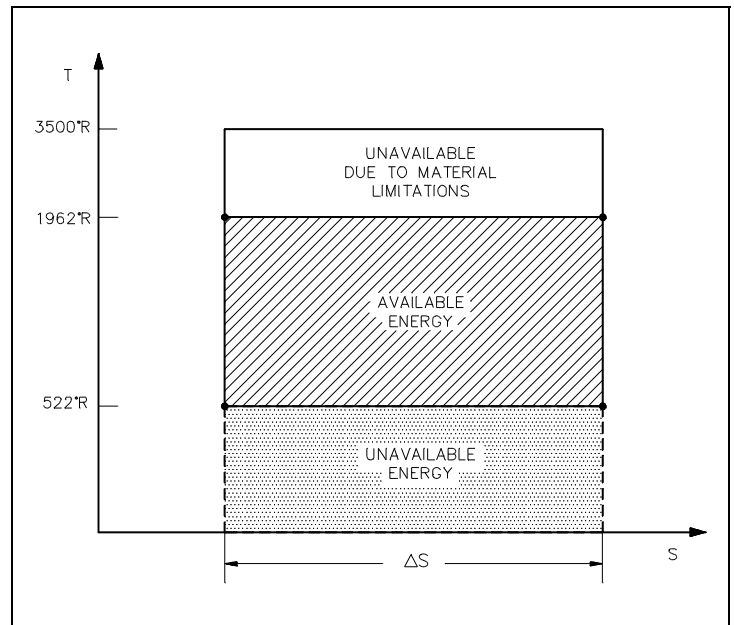


Figure 28 Carnot Cycle

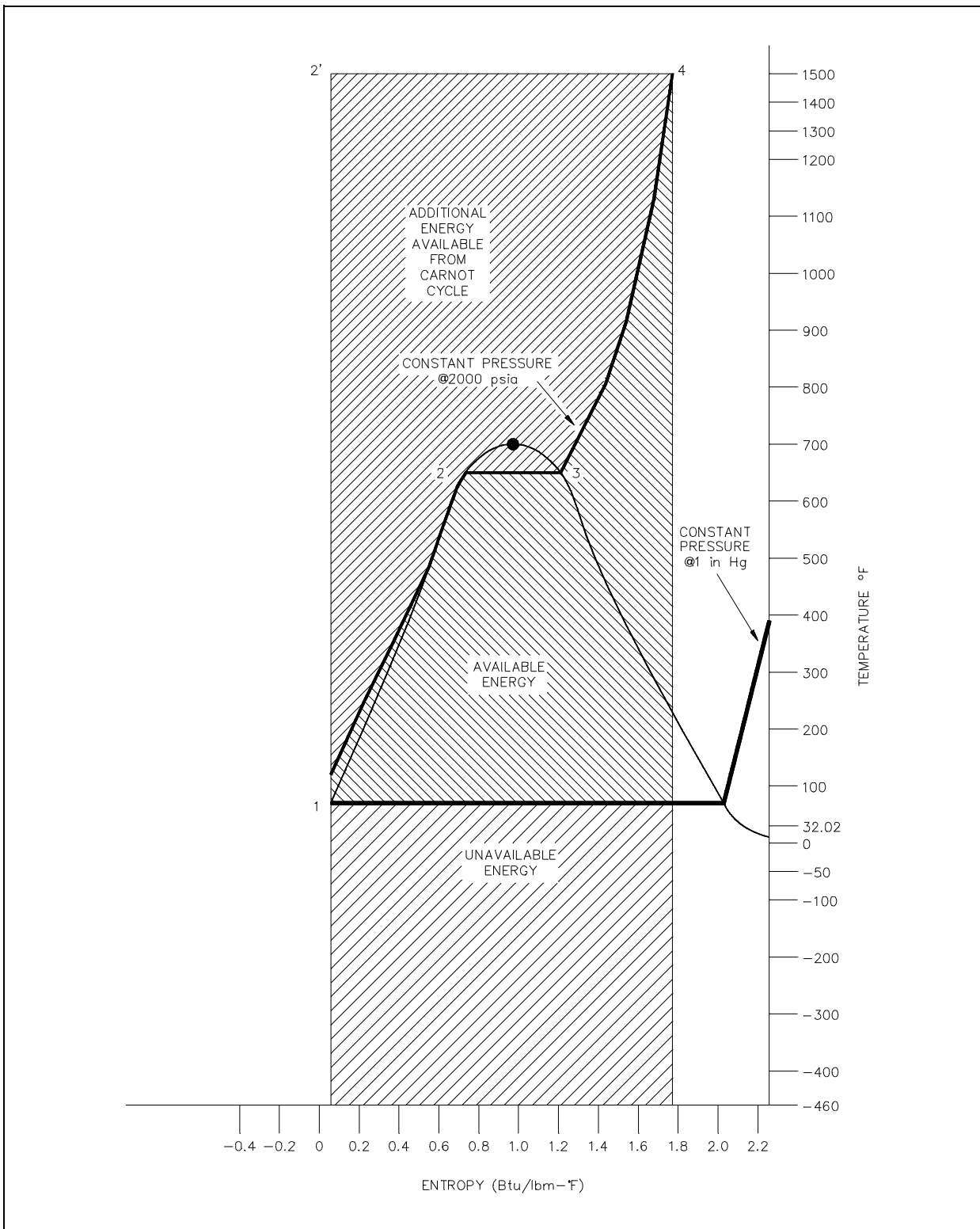


Figure 29 Carnot Cycle vs. Typical Power Cycle Available Energy

Figure 29 is a typical power cycle employed by a fossil fuel plant. The working fluid is water, which places certain restrictions on the cycle. If we wish to limit ourselves to operation at or below 2000 psia, it is readily apparent that constant heat addition at our maximum temperature of 1962°R is not possible (Figure 29, 2' to 4). In reality, the nature of water and certain elements of the process controls require us to add heat in a constant pressure process instead (Figure 29, 1-2-3-4). Because of this, the average temperature at which we are adding heat is far below the maximum allowable material temperature.

As can be seen, the actual available energy (area under the 1-2-3-4 curve, Figure 29) is less than half of what is available from the ideal Carnot cycle (area under 1-2'-4 curve, Figure 29) operating between the same two temperatures. Typical thermal efficiencies for fossil plants are on the order of 40% while nuclear plants have efficiencies of the order of 31%. Note that these numbers are less than 1/2 of the maximum thermal efficiency of the ideal Carnot cycle calculated earlier.

Figure 30 shows a proposed Carnot steam cycle superimposed on a T-s diagram. As shown, it has several problems which make it undesirable as a practical power cycle. First a great deal of pump work is required to compress a two phase mixture of water and steam from point 1 to the saturated liquid state at point 2. Second, this same isentropic compression will probably result in some pump cavitation in the feed system. Finally, a condenser designed to produce a two-phase mixture at the outlet (point 1) would pose technical problems.

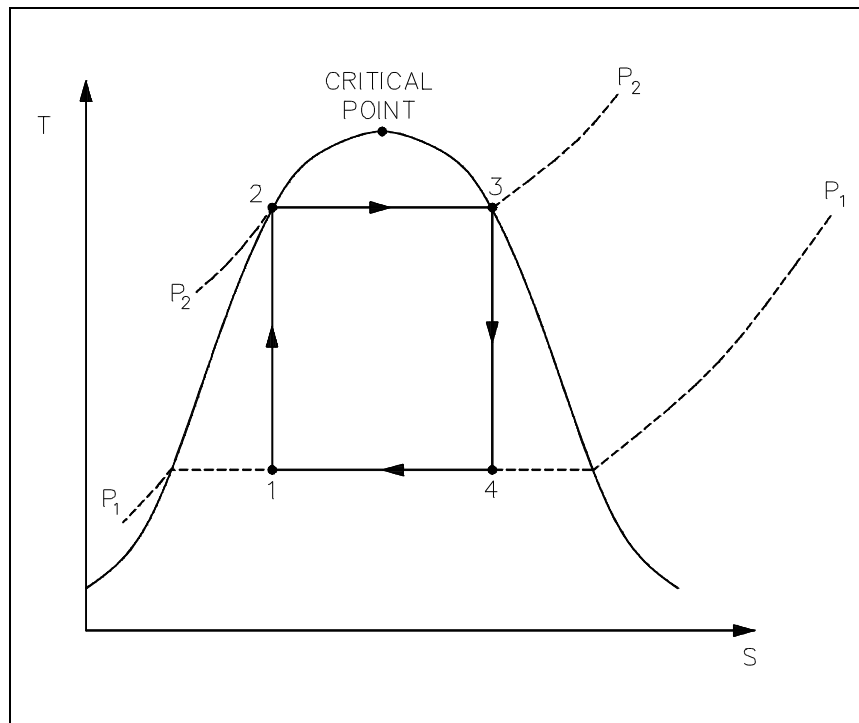


Figure 30 Ideal Carnot Cycle

Early thermodynamic developments were centered around improving the performance of contemporary steam engines. It was desirable to construct a cycle that was as close to being reversible as possible and would better lend itself to the characteristics of steam and process control than the Carnot cycle did. Towards this end, the Rankine cycle was developed.

The main feature of the Rankine cycle, shown in Figure 31, is that it confines the isentropic compression process to the liquid phase only (Figure 31 points 1 to 2). This minimizes the amount of work required to attain operating pressures and avoids the mechanical problems associated with pumping a two-phase mixture. The compression process shown in figure 31 between points 1 and 2 is greatly exaggerated*. In reality, a temperature rise of only 1°F occurs in compressing water from 14.7 psig at a saturation temperature of 212°F to 1000 psig.

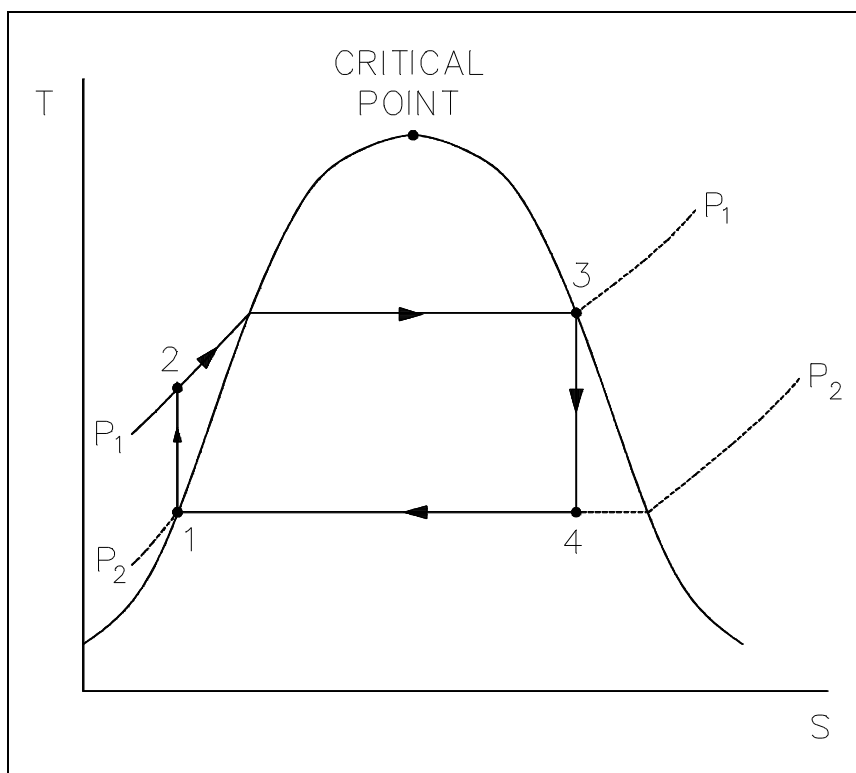


Figure 31 Rankine Cycle

** The constant pressure lines converge rapidly in the subcooled or compressed liquid region and it is difficult to distinguish them from the saturated liquid line without artificially expanding them away from it.*

In a Rankine cycle available and unavailable energy on a T-s diagram, like a T-s diagram of a Carnot cycle, is represented by the areas under the curves. The larger the unavailable energy, the less efficient the cycle.

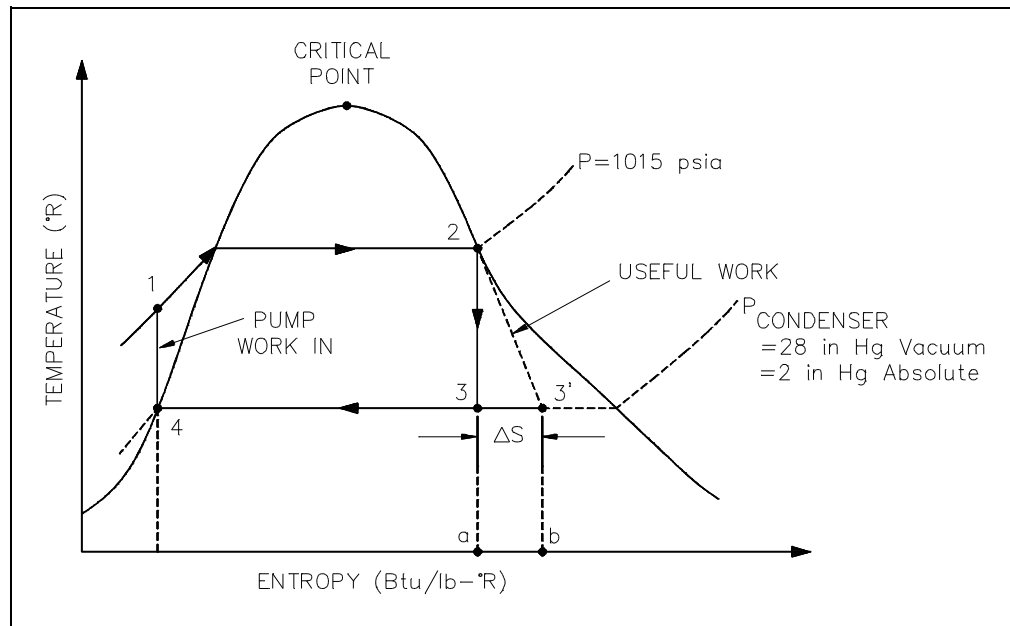


Figure 32 Rankine Cycle With Real v.s. Ideal

From the T-s diagram (Figure 32) it can also be seen that if an ideal component, in this case the turbine, is replaced with a non-ideal component, the efficiency of the cycle will be reduced. This is due to the fact that the non-ideal turbine incurs an increase in entropy which increases the area under the T-s curve for the cycle. But the increase in the area of available energy (3-2-3', Figure 32) is less than the increase in area for unavailable energy (a-3-3'-b, Figure 32).

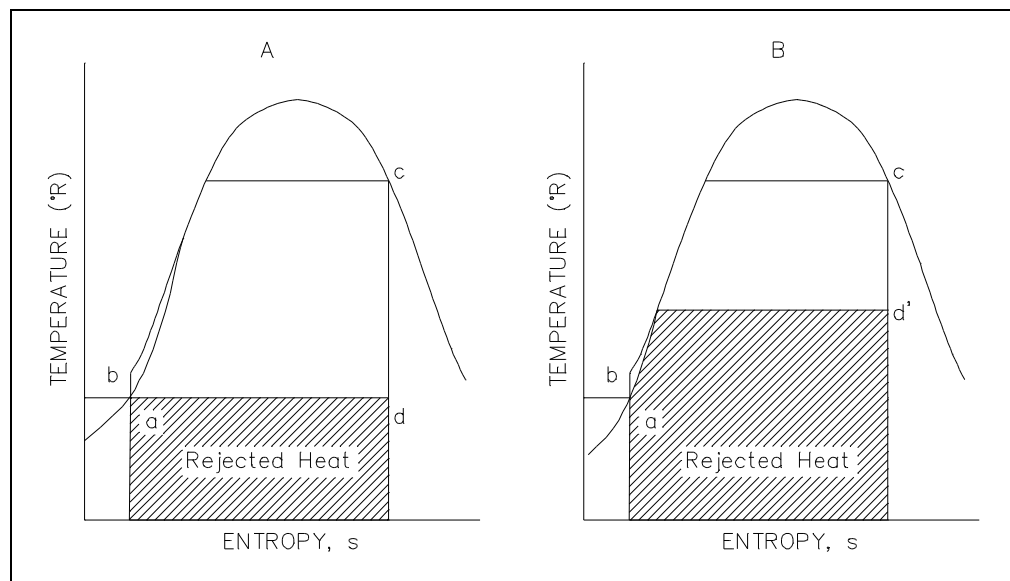


Figure 33 Rankine Cycle Efficiencies T-s

The same loss of cycle efficiency can be seen when two Rankine cycles are compared (see Figure 33). Using this type of comparison, the amount of rejected energy to available energy of one cycle can be compared to another cycle to determine which cycle is the most efficient, i.e. has the least amount of unavailable energy.

An h-s diagram can also be used to compare systems and help determine their efficiencies. Like the T-s diagram, the h-s diagram will show (Figure 34) that substituting non-ideal components in place of ideal components in a cycle, will result in the reduction in the cycles efficiency. This is because a change in enthalpy (h) always occurs when work is done or heat is added or removed in an actual cycle (non-ideal). This deviation from an ideal constant enthalpy (vertical line on the diagram) allows the inefficiencies of the cycle to be easily seen on a h-s diagram.

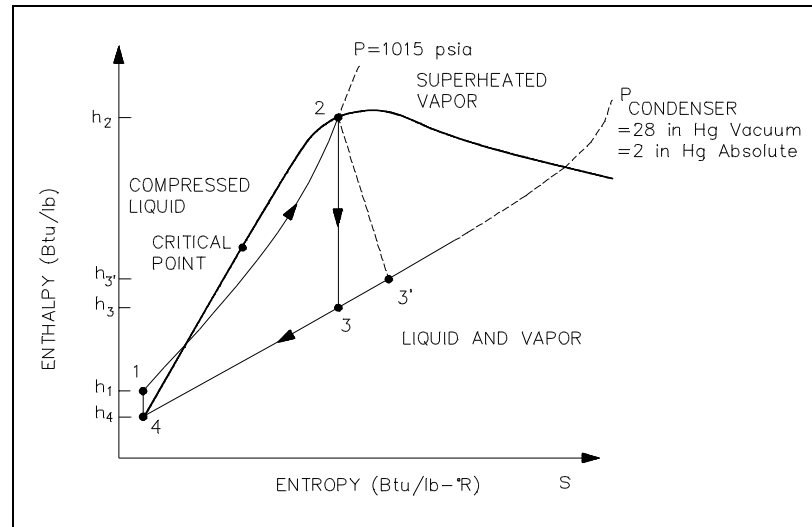


Figure 34 h-s Diagram

Typical Steam Cycle

Figure 35 shows a simplified version of the major components of a typical steam plant cycle. This is a simplified version and does not contain the exact detail that may be found at most power plants. However, for the purpose of understanding the basic operation of a power cycle, further detail is not necessary.

The following are the processes that comprise the cycle:

- 1-2: Saturated steam from the steam generator is expanded in the high pressure (HP) turbine to provide shaft work output at a constant entropy.
- 2-3: The moist steam from the exit of the HP turbine is dried and superheated in the moisture separator reheater (MSR).
- 3-4: Superheated steam from the MSR is expanded in the low pressure (LP) turbine to provide shaft work output at a constant entropy.

- 4-5: Steam exhaust from the turbine is condensed in the condenser in which heat is transferred to the cooling water under a constant vacuum condition.
- 5-6: The feedwater is compressed as a liquid by the condensate and feedwater pump and the feedwater is preheated by the feedwater heaters.
- 6-1: Heat is added to the working fluid in the steam generator under a constant pressure condition.

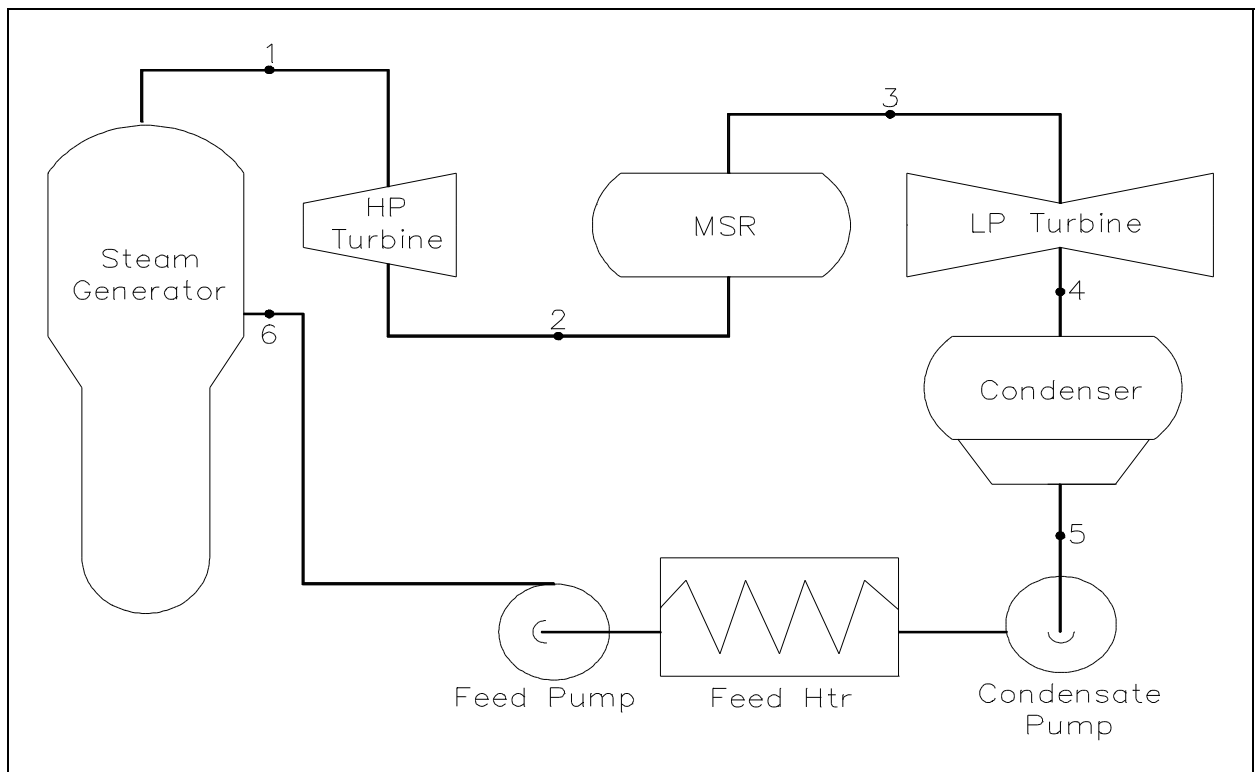


Figure 35 Typical Steam Cycle

The previous cycle can also be represented on a T-s diagram as was done with the ideal Carnot and Rankine cycles. This is shown in Figure 36. The numbered points on the cycle correspond to the numbered points on Figure 36.

It must be pointed out that the cycle we have just shown is an ideal cycle and does not exactly represent the actual processes in the plant. The turbine and pumps in an ideal cycle are ideal pumps and turbines and therefore do not exhibit an increase in entropy across them. Real pumps and turbines would exhibit an entropy increase across them.

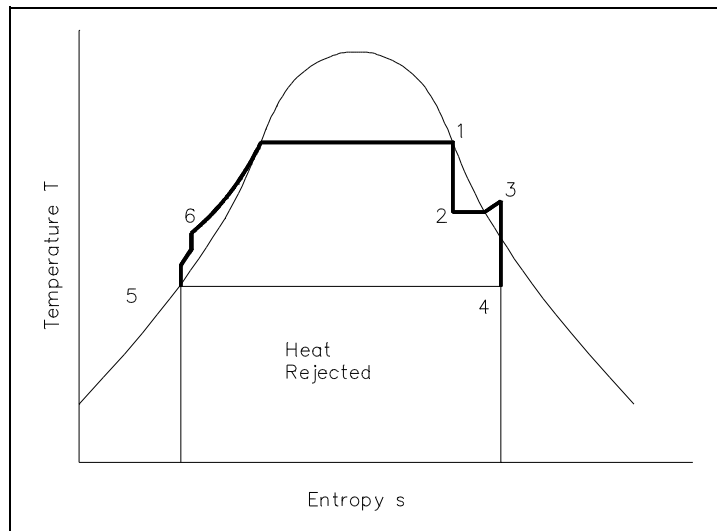


Figure 36 Steam Cycle (Ideal)

Figure 37 is a T-s diagram of a cycle which more closely approximates actual plant processes. The pumps and turbines in this cycle more closely approximate real pumps and turbines and thus exhibit an entropy increase across them. Additionally, in this cycle, a small degree of subcooling is evident in the condenser as shown by the small dip down to point 5. This small amount of subcooling will decrease cycle efficiency since additional heat has been removed from the cycle to the cooling water as heat rejected. This additional heat rejected must then be made up for in the steam generator. Therefore, it can be seen that excessive condenser subcooling will decrease cycle efficiency. By controlling the temperature or flow rate of the cooling water to the condenser, the operator can directly effect the overall cycle efficiency.

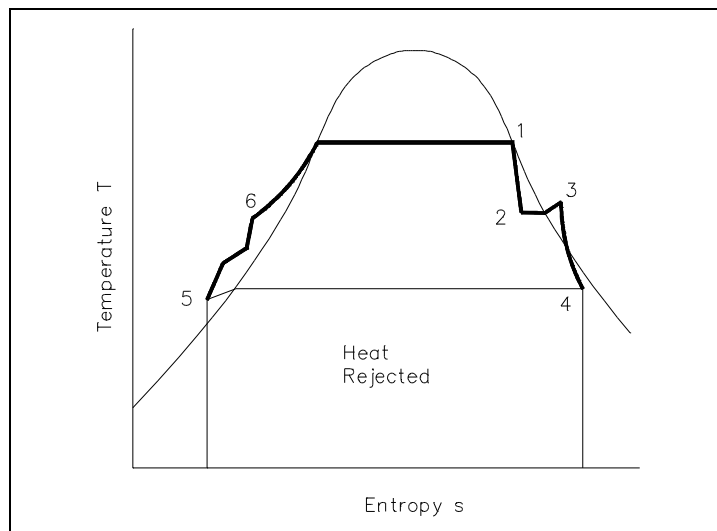


Figure 37 Steam Cycle (Real)

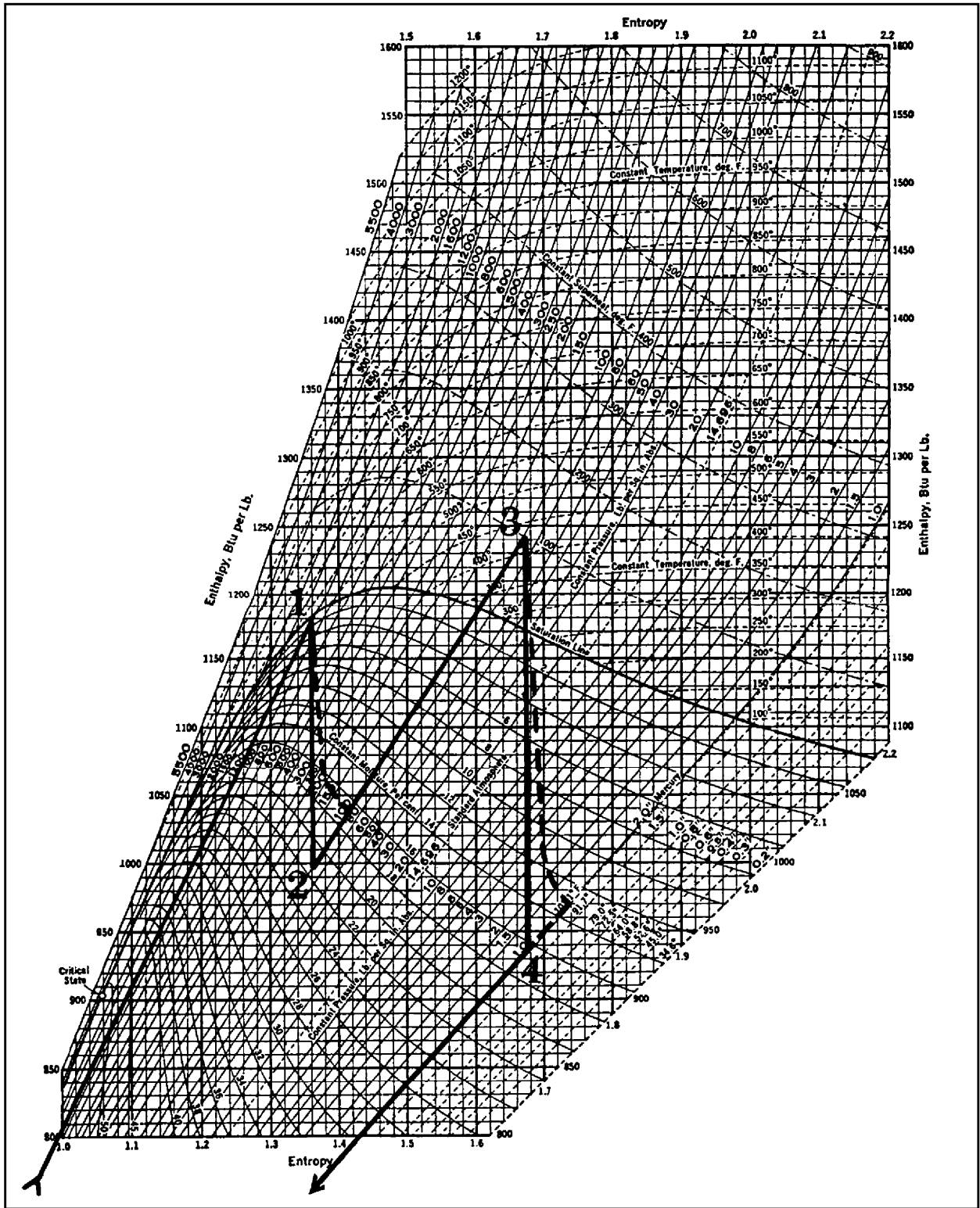


Figure 38 Mollier Diagram

It is sometimes useful to plot on the Mollier diagram the processes that occur during the cycle. This is done on Figure 38. The numbered points on Figure 38 correspond to the numbered points on Figures 35 and 36. Because the Mollier diagram is a plot of the conditions existing for water in vapor form, the portions of the plot which fall into the region of liquid water do not show up on the Mollier diagram. The following conditions were used in plotting the curves on Figure 38.

- Point 1: Saturated steam at 540°F
- Point 2: 82.5% quality at exit of HP turbine
- Point 3: Temperature of superheated steam is 440°F
- Point 4: Condenser vacuum is 1 psia

The solid lines on Figure 38 represent the conditions for a cycle which uses ideal turbines as verified by the fact that no entropy change is shown across the turbines. The dotted lines on Figure 38 represent the path taken if real turbines were considered, in which case an increase in entropy is evident.

Causes of Inefficiency

In the preceding sections, cycle and component efficiencies have been discussed, but the actual causes or reasons for the inefficiencies have not been explained. In this section we will compare some of the types and causes for the inefficiencies of real components and cycles to that of their "ideal" counterparts.

Components

In real systems, a percentage of the overall cycle inefficiency is due to the losses by the individual components. Turbines, pumps, and compressors all behave non-ideally due to heat losses, friction and windage losses. All of these losses contribute to the non-isentropic behavior of real equipment. As explained previously (Figures 24, 25) these losses can be seen as an increase in the system's entropy or amount of energy that is unavailable for use by the cycle.

Cycles

In real systems, a second source of inefficiencies is from the compromises made due to cost and other factors in the design and operation of the cycle. Examples of these types of losses are: In a large power generating station the condensers are designed to subcool the liquid by 8-10°F. This subcooling allows the condensate pumps to pump the water forward without cavitation. But, each degree of subcooling is energy that must be put back by reheating the water, and this heat (energy) does no useful work and therefore increases the inefficiency of the cycle. Another example of a loss due to a system's design is heat loss to the environment, i.e. thin or poor insulation. Again this is energy lost to the system and therefore unavailable to do work. Friction is another real world loss, both resistance to fluid flow and mechanical friction in machines. All of these contribute to the system's inefficiency.

Summary

The important information from this chapter is summarized below.

Second Law of Thermodynamics Summary

- Planck's statement of the Second Law of Thermodynamics is:

It is impossible to construct an engine that will work in a complete cycle and produce no other effect except the raising of a weight and the cooling of a heat reservoir.

- The Second Law of Thermodynamics demonstrates that the maximum possible efficiency of a system is the Carnot efficiency written as:

$$\eta = (T_H - T_C)/T_H$$

- The maximum efficiency of a closed cycle can be determined by calculating the efficiency of a Carnot cycle operating between the same value of high and low temperatures.
- The efficiency of a component can be calculated by comparing the work produced by the component to the work that would have been produced by an ideal component operating isentropically between the same inlet and outlet conditions.
- An isentropic expansion or compression process will be represented as a vertical line on a T-s or h-s diagram. A real expansion or compression process will look similar, but will be slanted slightly to the right.
- Efficiency will be decreased by:

Presence of friction
Heat losses
Cycle inefficiencies

COMPRESSION PROCESSES

Compression and pressurization processes are very common in many types of industrial plants. These processes vary from being the primary function of a piece of equipment, such as an air compressor, to an incidental result of another process, such as filling a tank with water without first opening the valve.

- EO 1.32** **Apply the ideal gas laws to SOLVE for the unknown pressure, temperature, or volume.**
- EO 1.33** **DESCRIBE when a fluid may be considered to be incompressible.**
- EO 1.34** **CALCULATE the work done in constant pressure and constant volume processes.**
- EO 1.35** **DESCRIBE the effects of pressure changes on confined fluids.**
- EO 1.36** **DESCRIBE the effects of temperature changes on confined fluids.**
-

Boyle's and Charles' Laws

The results of certain experiments with gases at relatively low pressure led Robert Boyle to formulate a well-known law. It states that:

the pressure of a gas expanding at constant temperature varies inversely to the volume, or

$$(P_1)(V_1) = (P_2)(V_2) = (P_3)(V_3) = \text{constant.} \quad (1-40)$$

Charles, also as the result of experimentation, concluded that:

the pressure of a gas varies directly with temperature when the volume is held constant, and the volume varies directly with temperature when the pressure is held constant, or

$$\frac{V_1}{V_2} = \frac{T_1}{T_2} \quad \text{or} \quad \frac{P_1}{P_2} = \frac{T_1}{T_2}. \quad (1-41)$$

Ideal Gas Law

By combining the results of Charles' and Boyle's experiments, the relationship

$$\frac{Pv}{T} = \text{constant} \tag{1-42}$$

may be obtained. The constant in the above equation is called the ideal gas constant and is designated by R; thus the ideal gas equation becomes

$$Pv = RT \tag{1-43}$$

where the pressure and temperature are absolute values. The values of the ideal gas constant (R) for several of the more common gases are given in Figure 39.

Gas	Chemical Symbol	Molecular Weight	Gas Constant ft-lbf/lbm·R	Specific Heat Btu/lbm		Specific Heat Ratio
				c_p	c_v	
Air		M	R			k
Carbon dioxide	CO ₂	44.00	35.13	0.160	0.205	1.28
Hydrogen	H ₂	2.016	766.80	2.44	3.42	1.40
Nitrogen	N ₂	28.02	55.16	0.176	0.247	1.40
Oxygen	O ₂	32.0	48.31	0.155	0.217	1.40
Steam	H ₂ O	18.016	85.81	0.36	0.46	1.28

Steam at pressures less than 1 psia behaves very nearly as a perfect gas.

Figure 39 Ideal Gas Constant Values

The individual gas constant (R) may be obtained by dividing the universal gas constant (Ro) by the molecular weight (MW) of the gas, $R = \frac{R_o}{MW}$. The units of R must always be consistent with the units of pressure, temperature, and volume used in the gas equation. No real gases follow the ideal gas law or equation completely. At temperatures near a gases boiling point, increases in pressure will cause condensation to take place and drastic decreases in volume. At very high pressures, the intermolecular forces of a gas are significant. However, most gases are in approximate agreement at pressures and temperatures above their boiling point.

The ideal gas law is utilized by engineers working with gases because it is simple to use and approximates real gas behavior. Most physical conditions of gases used by man fit the above description. Perhaps the most common use of gas behavior studied by engineers is that of the compression process using ideal gas approximations. Such a compression process may occur at constant temperature ($pV = \text{constant}$), constant volume, or adiabatic (no heat transfer). Whatever the process, the amount of work that results from it depends upon the process, as brought out in the discussion on the First Law of Thermodynamics. The compression process using ideal gas considerations results in work performed on the system and is essentially the area under a P-V curve. As can be seen in Figure 40, different amounts of work result from different ideal gas processes such as constant temperature and constant pressure.

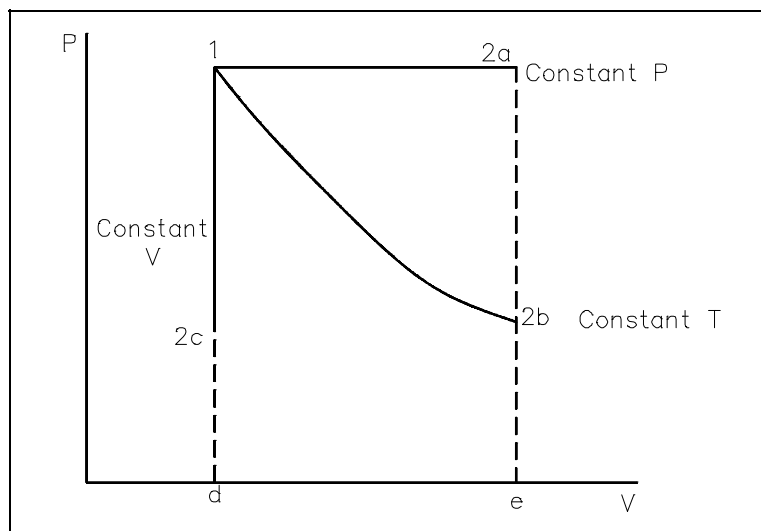


Figure 40 Pressure-Volume Diagram

Fluid

A *fluid* is any substance that conforms to the shape of its container. It may be either a liquid or a gas.

Compressibility of Fluids

Usually a fluid may be considered incompressible when the velocity of the fluid is greater than one-third of the speed of sound for the fluid, or if the fluid is a liquid. The treatment of a fluid that is considered incompressible is easy because the density is assumed to be constant, giving a simple relationship for the state of the substance. The variation of density of the fluid with changes in pressure is the primary factor considered in deciding whether a fluid is incompressible.

Fluids that are compressible have much more complex equations to deal with, due to density changes, and have property relationships that vary more rapidly than incompressible fluids. In addition, fixing the state of a liquid can be done easily by knowing its temperature and pressure. Once the substance becomes a gas, the process becomes more difficult.

Constant Pressure Process

To determine the work done in a constant pressure process, the following equation is used:

$$W_{1-2} = P(\Delta V) \quad (1-44)$$

Constant Volume Process

The solution of Equation 1-45 for a constant volume process is also not difficult. The work done in a constant volume process is the product of the volume and the change in pressure.

$$W_{1-2} = V(\Delta P) \quad (1-45)$$

In addition to gases, Equation 1-45 also applies to liquids. The power requirement for pumps that move incompressible liquids (such as water) can be determined from Equation 1-44. Replacing the volume (V) with the product of the specific volume and the mass yields Equation 1-45.

$$W_{1-2} = mv(\Delta P) \quad (1-46)$$

Taking the time rate of change of both sides of Equation 1-46 determines the power requirements of the pump.

$$\dot{W}_{1-2} = \dot{m}v(\Delta P) \quad (1-47)$$

Effects of Pressure Changes on Fluid Properties

The predominant effect of an increase in pressure in a compressible fluid, such as a gas, is an increase in the density of the fluid. An increase in the pressure of an incompressible fluid will not have a significant effect on the density. For example, increasing the pressure of 100 °F water from 15 psia to 15,000 psia will only increase the density by approximately 6%. Therefore, in engineering calculations, it is assumed that incompressible fluids' density remain constant.

Effects of Temperature Changes on Fluid Properties

An increase in temperature will tend to decrease the density of any fluid. If the fluid is confined in a container of fixed volume, the effect of a temperature change will depend on whether the fluid is compressible.

If the fluid is a gas, it will respond to a temperature change in a manner predicted by the ideal gas laws. A 5% increase in absolute temperature will result in a 5% increase in the absolute pressure.

If the fluid is an incompressible liquid in a closed container, an increase in the temperature will have a tremendously greater and potentially catastrophic effect. As the fluid temperature increases, it tries to expand, but expansion is prevented by the walls of the container. Because the fluid is incompressible, this results in a tremendous increase in pressure for a relatively minor temperature change. The change in specific volume for a given change in temperature is not the same at various beginning temperatures. Resultant pressure changes will vary. A useful thumb rule for water is that pressure in a water-solid system will increase about 100 psi for every 1 °F increase in temperature.

Summary

The important information from this chapter is summarized below.

Compression Processes Summary

- The ideal gas law can be used to determine how the properties of pressure, temperature, and volume will be related during compression processes.

$$Pv = R T$$

- A fluid may be considered incompressible if one of two conditions is true:

The fluid is a liquid.

The fluid is a gas with a velocity greater than one-third of the speed of sound in the gas.

- The work for certain types of processes can be determined as follows:

Constant pressure process $W_{1-2} = P(\Delta V)$

Constant volume process $W_{1-2} = V(\Delta P)$

Appendix A

Thermodynamics

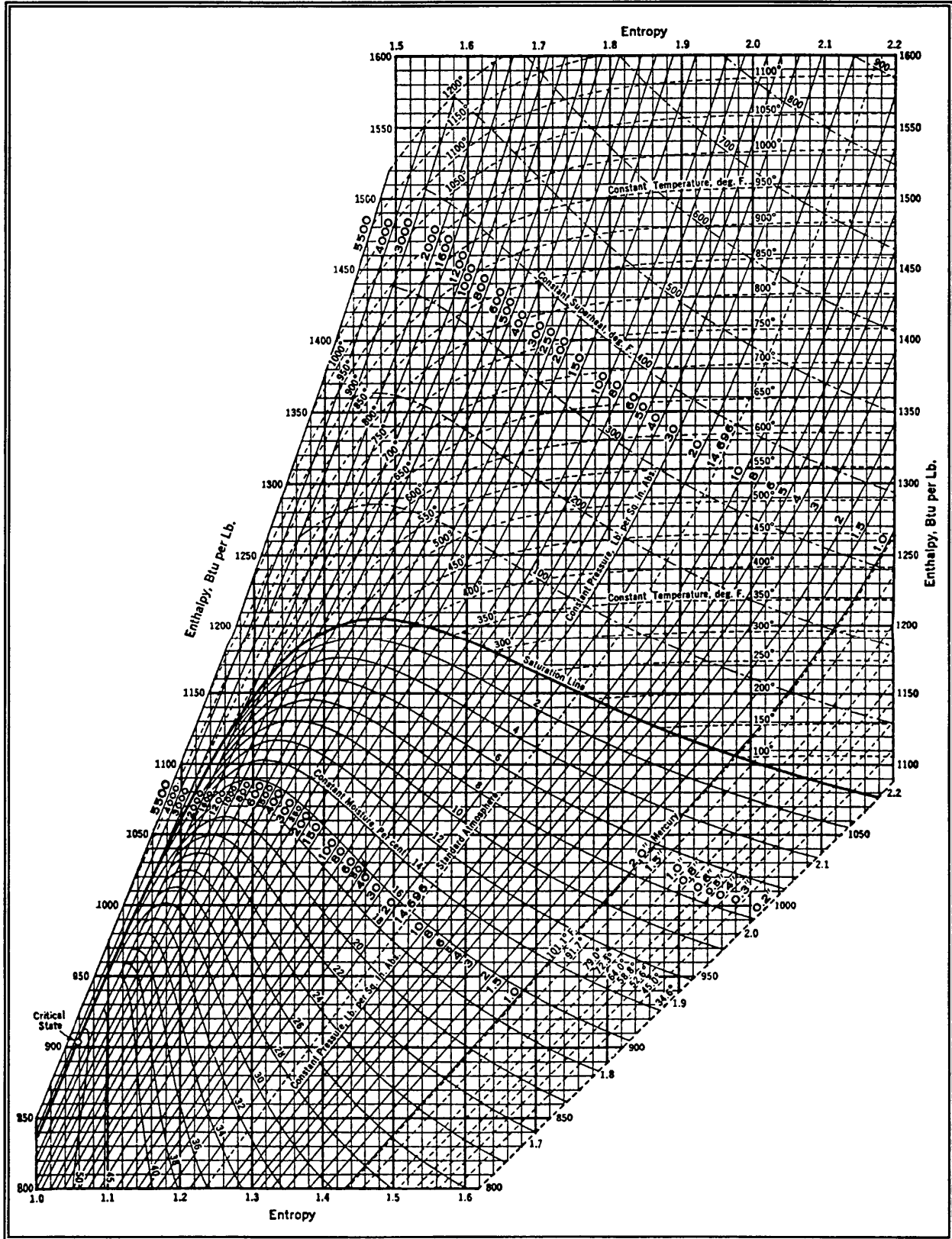


Figure A-1

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Saturated Steam: Pressure Table											
Abs Press. Lb/Sq In. p	Temp Fahr t	Specific Volume			Enthalpy			Entropy			Abs Press. Lb/Sq In. p
		Sat. Liquid v _f	Evap v _{fg}	Sat. Vapor v _g	Sat. Liquid h _f	Evap h _{fg}	Sat. Vapor h _g	Sat. Liquid s _f	Evap s _{fg}	Sat. Vapor s _g	
200.0	381.80	0.01839	2.2689	2.2873	355.5	842.8	1198.3	0.5438	1.0016	1.5454	200.0
210.0	385.91	0.01844	2.16373	2.18217	359.9	839.1	1199.0	0.5490	0.9923	1.5413	210.0
220.0	389.88	0.01850	2.06779	2.08629	364.2	835.4	1199.6	0.5540	0.9834	1.5374	220.0
230.0	393.70	0.01855	1.97991	1.99846	368.3	831.8	1200.1	0.5588	0.9748	1.5336	230.0
240.0	397.39	0.01860	1.89909	1.91769	372.3	828.4	1200.6	0.5634	0.9665	1.5299	240.0
250.0	400.97	0.01865	1.82452	1.84317	376.1	825.0	1201.1	0.5679	0.9585	1.5264	250.0
260.0	404.44	0.01870	1.75548	1.77418	379.9	821.6	1201.5	0.5722	0.9508	1.5230	260.0
270.0	407.80	0.01875	1.69137	1.71013	383.6	818.3	1201.9	0.5764	0.9433	1.5197	270.0
280.0	411.07	0.01880	1.63169	1.65049	387.1	815.1	1202.3	0.5805	0.9361	1.5166	280.0
290.0	414.25	0.01885	1.57597	1.59482	390.6	812.0	1202.6	0.5844	0.9291	1.5135	290.0
300.0	417.35	0.01889	1.52384	1.54274	394.0	808.9	1202.9	0.5882	0.9223	1.5105	300.0
350.0	431.73	0.01912	1.30642	1.32554	409.8	794.2	1204.0	0.6059	0.8909	1.4968	350.0
400.0	444.60	0.01934	1.14162	1.16095	424.2	780.4	1204.6	0.6217	0.8630	1.4847	400.0

Saturated Steam: Temperature Table											
Temp Fahr t	Abs Press Lb per Sq In. p	Specific Volume			Enthalpy			Entropy			Temp Fahr t
		Sat. Liquid v _f	Evap v _{fg}	Sat. Vapor v _g	Sat. Liquid h _f	Evap h _{fg}	Sat. Vapor h _g	Sat. Liquid s _f	Evap s _{fg}	Sat. Vapor s _g	
460.0	466.87	0.01961	0.97463	0.99424	441.5	763.2	1204.8	0.6405	0.8299	1.4704	460.0
464.0	485.56	0.01969	0.93588	0.95557	446.1	758.6	1204.7	0.6454	0.8213	1.4667	464.0
468.0	504.83	0.01976	0.89885	0.91862	450.7	754.0	1204.6	0.6502	0.8127	1.4629	468.0
472.0	524.67	0.01984	0.86345	0.88329	455.2	749.3	1204.5	0.6551	0.8042	1.4592	472.0
476.0	545.11	0.01992	0.82958	0.84950	459.9	744.5	1204.3	0.6599	0.7956	1.4555	476.0
480.0	566.15	0.02000	0.79716	0.81717	464.5	739.6	1204.1	0.6648	0.7871	1.4518	480.0
484.0	587.81	0.02009	0.76613	0.78622	469.1	734.7	1203.8	0.6696	0.7785	1.4481	484.0
488.0	610.10	0.02017	0.73641	0.75658	473.8	729.7	1203.5	0.6745	0.7700	1.4444	488.0
492.0	633.03	0.02026	0.70794	0.72820	478.5	724.6	1203.1	0.6793	0.7614	1.4407	492.0
496.0	656.61	0.02034	0.68065	0.70100	483.2	719.5	1202.7	0.6842	0.7528	1.4370	496.0

Superheated Steam																	
Abs Press. Lb/Sq In. (Sat. Temp)	Sat. Water	Sat. Steam	Temperature - Degrees Fahrenheit														
			450	500	550	600	650	700	800	900	1000	1100	1200	1300	1400	1500	
400 (444.60)	Sh		5.40	55.40	105.40	155.40	205.40	255.40	355.40	455.40	555.40	655.40	755.40	855.40	955.40	1055.40	
	v	0.01934	1.1610	1.1738	1.2841	1.3836	1.4763	1.5646	1.6499	1.8151	1.9759	2.1339	2.2901	2.4450	2.5987	2.7515	2.9037
	h	424.17	1204.6	1208.8	1245.1	1277.5	1307.4	1335.9	1363.4	1417.0	1470.1	1523.3	1576.9	1631.2	1686.2	1741.9	1798.2
	s	0.6217	1.4847	1.4894	1.5282	1.5611	1.5901	1.6163	1.6406	1.6850	1.7255	1.7632	1.7988	1.8325	1.8647	1.8955	1.9250
420 (449.40)	Sh		6.0	50.60	100.60	150.60	200.60	250.60	350.60	450.60	550.60	650.60	750.60	850.60	950.60	1050.60	
	v	0.01942	1.1057	1.1071	1.2148	1.3113	1.4007	1.4856	1.5676	1.7258	1.8795	2.0304	2.1795	2.3273	2.4739	2.6196	2.7647
	h	429.56	1204.7	1205.2	1242.4	1275.4	1305.8	1334.5	1362.3	1416.2	1469.4	1522.7	1576.4	1630.8	1685.8	1741.6	1798.0
	s	0.6276	1.4802	1.4808	1.5206	1.5542	1.5835	1.6100	1.6345	1.6791	1.7197	1.7575	1.7932	1.8269	1.8591	1.8899	1.9195
440 (454.03)	Sh		45.97	95.97	145.97	195.97	245.97	345.97	445.97	545.97	645.97	745.97	845.97	945.97	1045.97		
	v	0.01950	1.0554	1.1517	1.2454	1.3319	1.4138	1.4926	1.6445	1.7918	1.9363	2.0790	2.2203	2.3605	2.4998	2.6384	
	h	434.77	1204.8	1239.7	1273.4	1304.2	1333.2	1361.1	1415.3	1468.7	1522.1	1575.9	1630.4	1685.5	1741.2	1797.7	
	s	0.6332	1.4759	1.5132	1.5474	1.5772	1.6040	1.6286	1.6734	1.7142	1.7521	1.7878	1.8216	1.8538	1.8847	1.9143	

Figure A-2

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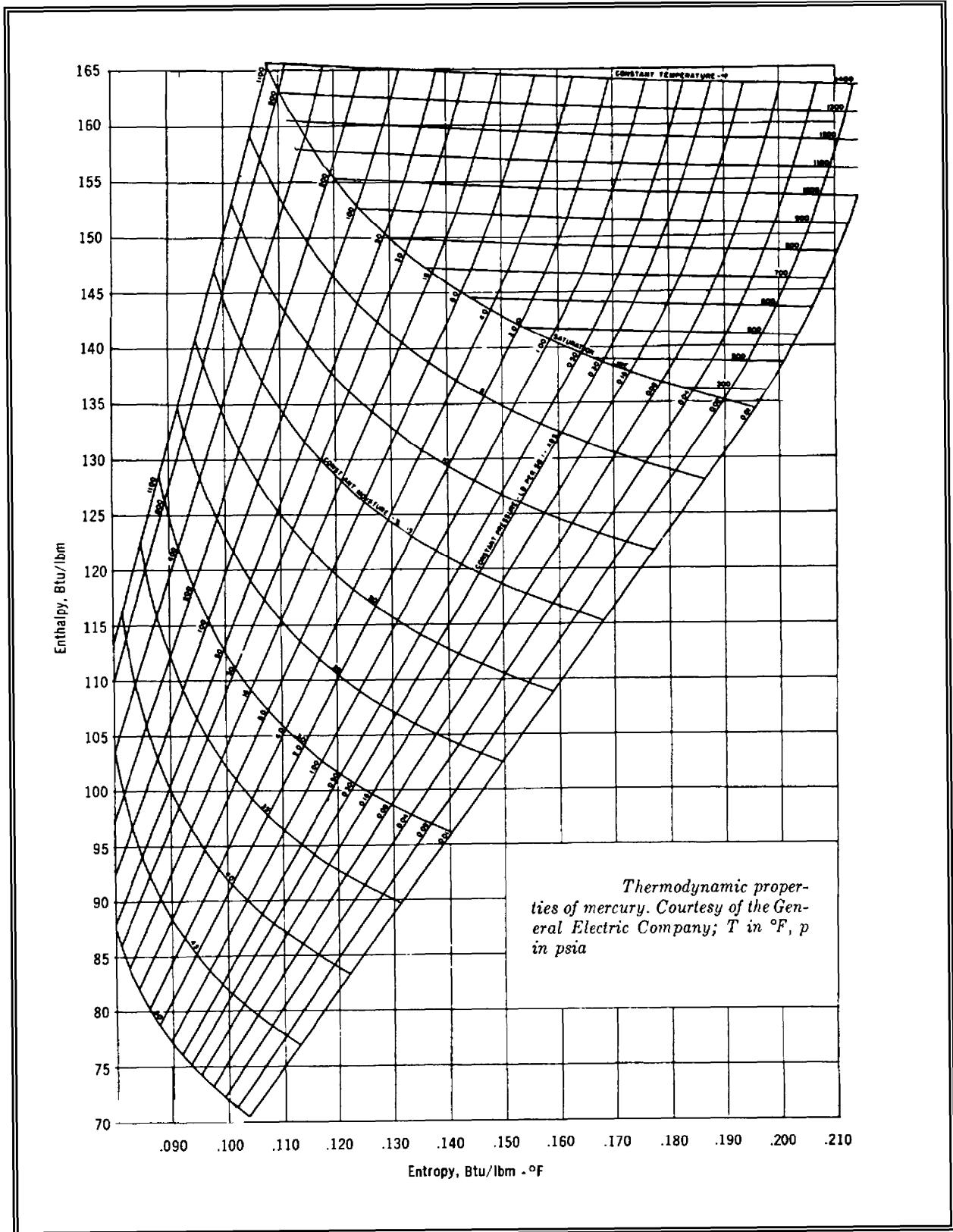
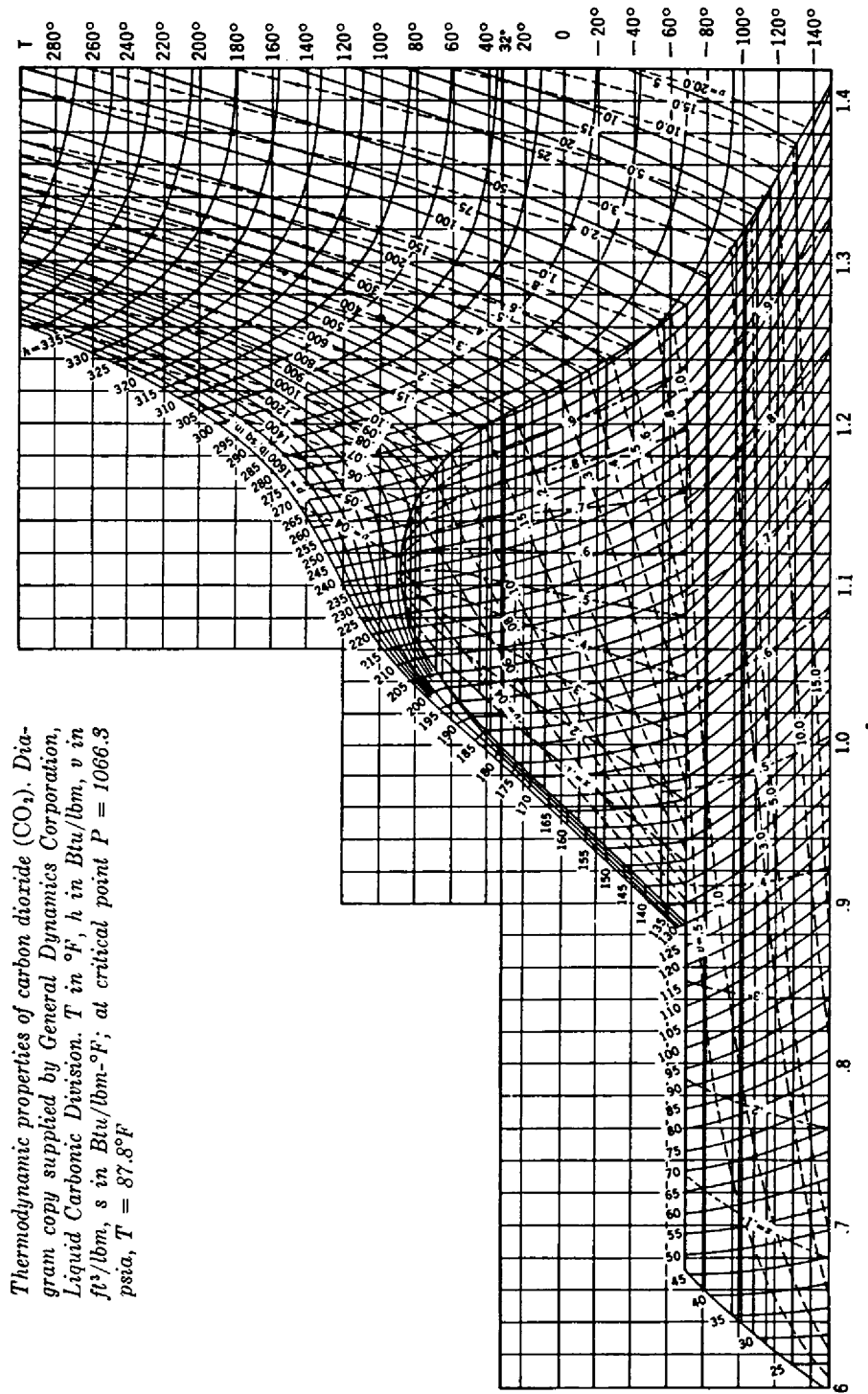


Figure A-3

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Thermodynamic properties of carbon dioxide (CO₂). Diagram copy supplied by General Dynamics Corporation, Liquid Carbonic Division. T in °F, h in Btu/lbm, v in ft³/lbm, s in Btu/lbm-°F; at critical point P = 1066.8 psia, T = 87.8°F

Figure A-4

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**Department of Energy
Fundamentals Handbook**

**THERMODYNAMICS, HEAT TRANSFER,
AND FLUID FLOW,
Module 2
Heat Transfer**

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TERMINAL OBJECTIVE

- 1.0 Given the operating conditions of a thermodynamic system and the necessary formulas, **EVALUATE** the heat transfer processes which are occurring.

ENABLING OBJECTIVES

- 1.1 **DESCRIBE** the difference between heat and temperature.
- 1.2 **DESCRIBE** the difference between heat and work.
- 1.3 **DESCRIBE** the Second Law of Thermodynamics and how it relates to heat transfer.
- 1.4 **DESCRIBE** the three modes of heat transfer.
- 1.5 **DEFINE** the following terms as they relate to heat transfer:
- Heat flux
 - Thermal conductivity
 - Log mean temperature difference
 - Convective heat transfer coefficient
 - Overall heat transfer coefficient
 - Bulk temperature
- 1.6 Given Fourier's Law of Conduction, **CALCULATE** the conduction heat flux in a rectangular coordinate system.
- 1.7 Given the formula and the necessary values, **CALCULATE** the equivalent thermal resistance.
- 1.8 Given Fourier's Law of Conduction, **CALCULATE** the conduction heat flux in a cylindrical coordinate system.
- 1.9 Given the formula for heat transfer and the operating conditions of the system, **CALCULATE** the rate of heat transfer by convection.
- 1.10 **DESCRIBE** how the following terms relate to radiant heat transfer:
- Black body radiation
 - Emissivity
 - Radiation configuration factor

ENABLING OBJECTIVES (Cont.)

- 1.11 **DESCRIBE** the difference in the temperature profiles for counter-flow and parallel flow heat exchangers.
- 1.12 **DESCRIBE** the differences between regenerative and non-regenerative heat exchangers.
- 1.13 Given the temperature changes across a heat exchanger, **CALCULATE** the log mean temperature difference for the heat exchanger.
- 1.14 Given the formulas for calculating the conduction and convection heat transfer coefficients, **CALCULATE** the overall heat transfer coefficient of a system.
- 1.15 **DESCRIBE** the process that occurs in the following regions of the boiling heat transfer curve:
- a. Nucleate boiling
 - b. Partial film boiling
 - c. Film boiling
 - d. Departure from nucleate boiling (DNB)
 - e. Critical heat flux

TERMINAL OBJECTIVE

- 2.0 Given the operating conditions of a typical nuclear reactor, **DESCRIBE** the heat transfer processes which are occurring.

ENABLING OBJECTIVES

- 2.1 **DESCRIBE** the power generation process in a nuclear reactor core and the factors that affect the power generation.
- 2.2 **DESCRIBE** the relationship between temperature, flow, and power during operation of a nuclear reactor.
- 2.3 **DEFINE** the following terms:
- a. Nuclear enthalpy rise hot channel factor
 - b. Average linear power density
 - c. Nuclear heat flux hot channel factor
 - d. Heat generation rate of a core
 - e. Volumetric thermal source strength
- 2.4 **CALCULATE** the average linear power density for an average reactor core fuel rod.
- 2.5 **DESCRIBE** a typical reactor core axial and radial flux profile.
- 2.6 **DESCRIBE** a typical reactor core fuel rod axial and radial temperature profile.
- 2.7 **DEFINE** the term decay heat.
- 2.8 Given the operating conditions of a reactor core and the necessary formulas, **CALCULATE** the core decay heat generation.
- 2.9 **DESCRIBE** two categories of methods for removing decay heat from a reactor core.

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HEAT TRANSFER TERMINOLOGY

To understand and communicate in the thermal science field, certain terms and expressions must be learned in heat transfer.

- EO 1.1** **DESCRIBE the difference between heat and temperature.**
- EO 1.2** **DESCRIBE the difference between heat and work.**
- EO 1.3** **DESCRIBE the Second Law of Thermodynamics and how it relates to heat transfer.**
- EO 1.4** **DESCRIBE the three modes of heat transfer.**
- EO 1.5** **DEFINE the following terms as they relate to heat transfer:**
- a.** **Heat flux**
 - b.** **Thermal conductivity**
 - c.** **Log mean temperature difference**
 - d.** **Convective heat transfer coefficient**
 - e.** **Overall heat transfer coefficient**
 - f.** **Bulk temperature**

Heat and Temperature

In describing heat transfer problems, students often make the mistake of interchangeably using the terms heat and temperature. Actually, there is a distinct difference between the two. *Temperature* is a measure of the amount of energy possessed by the molecules of a substance. It is a relative measure of how hot or cold a substance is and can be used to predict the direction of heat transfer. The symbol for temperature is T. The common scales for measuring temperature are the Fahrenheit, Rankine, Celsius, and Kelvin temperature scales.

Heat is energy in transit. The transfer of energy as heat occurs at the molecular level as a result of a temperature difference. Heat is capable of being transmitted through solids and fluids by conduction, through fluids by convection, and through empty space by radiation. The symbol for heat is Q. Common units for measuring heat are the British Thermal Unit (Btu) in the English system of units and the calorie in the SI system (International System of Units).

Heat and Work

Distinction should also be made between the energy terms *heat* and *work*. Both represent energy in transition. Work is the transfer of energy resulting from a force acting through a distance. Heat is energy transferred as the result of a temperature difference. Neither heat nor work are thermodynamic properties of a system. Heat can be transferred into or out of a system and work can be done on or by a system, but a system cannot contain or store either heat or work. Heat into a system and work out of a system are considered positive quantities.

When a temperature difference exists across a boundary, the Second Law of Thermodynamics indicates the natural flow of energy is from the hotter body to the colder body. The Second Law of Thermodynamics denies the possibility of ever completely converting into work all the heat supplied to a system operating in a cycle. The Second Law of Thermodynamics, described by Max Planck in 1903, states that:

It is impossible to construct an engine that will work in a complete cycle and produce no other effect except the raising of a weight and the cooling of a reservoir.

The second law says that if you draw heat from a reservoir to raise a weight, lowering the weight will not generate enough heat to return the reservoir to its original temperature, and eventually the cycle will stop. If two blocks of metal at different temperatures are thermally insulated from their surroundings and are brought into contact with each other the heat will flow from the hotter to the colder. Eventually the two blocks will reach the same temperature, and heat transfer will cease. Energy has not been lost, but instead some energy has been transferred from one block to another.

Modes of Transferring Heat

Heat is always transferred when a temperature difference exists between two bodies. There are three basic modes of heat transfer:

Conduction involves the transfer of heat by the interactions of atoms or molecules of a material through which the heat is being transferred.

Convection involves the transfer of heat by the mixing and motion of macroscopic portions of a fluid.

Radiation, or radiant heat transfer, involves the transfer of heat by electromagnetic radiation that arises due to the temperature of a body.

The three modes of heat transfer will be discussed in greater detail in the subsequent chapters of this module.

Heat Flux

The rate at which heat is transferred is represented by the symbol \dot{Q} . Common units for heat transfer rate is Btu/hr. Sometimes it is important to determine the heat transfer rate per unit area, or *heat flux*, which has the symbol \dot{Q}'' . Units for heat flux are Btu/hr-ft². The heat flux can be determined by dividing the heat transfer rate by the area through which the heat is being transferred.

$$\dot{Q}'' = \frac{\dot{Q}}{A} \quad (2-1)$$

where:

$$\dot{Q}'' = \text{heat flux (Btu/hr-ft}^2\text{)}$$

$$\dot{Q} = \text{heat transfer rate (Btu/hr)}$$

$$A = \text{area (ft}^2\text{)}$$

Thermal Conductivity

The heat transfer characteristics of a solid material are measured by a property called the *thermal conductivity* (k) measured in Btu/hr-ft-°F. It is a measure of a substance's ability to transfer heat through a solid by conduction. The thermal conductivity of most liquids and solids varies with temperature. For vapors, it depends upon pressure.

Log Mean Temperature Difference

In heat exchanger applications, the inlet and outlet temperatures are commonly specified based on the fluid in the tubes. The temperature change that takes place across the heat exchanger from the entrance to the exit is not linear. A precise temperature change between two fluids across the heat exchanger is best represented by the *log mean temperature difference* (LMTD or ΔT_{lm}), defined in Equation 2-2.

$$\Delta T_{lm} = \frac{(\Delta T_2 - \Delta T_1)}{\ln(\Delta T_2/\Delta T_1)} \quad (2-2)$$

where:

$$\Delta T_2 = \text{the larger temperature difference between the two fluid streams at either the entrance or the exit to the heat exchanger}$$

$$\Delta T_1 = \text{the smaller temperature difference between the two fluid streams at either the entrance or the exit to the heat exchanger}$$

Convective Heat Transfer Coefficient

The convective heat transfer coefficient (h), defines, in part, the heat transfer due to convection. The *convective heat transfer coefficient* is sometimes referred to as a film coefficient and represents the thermal resistance of a relatively stagnant layer of fluid between a heat transfer surface and the fluid medium. Common units used to measure the convective heat transfer coefficient are Btu/hr - ft² - °F.

Overall Heat Transfer Coefficient

In the case of combined heat transfer, it is common practice to relate the total rate of heat transfer (\dot{Q}), the overall cross-sectional area for heat transfer (A_o), and the overall temperature difference (ΔT_o) using the overall heat transfer coefficient (U_o). The *overall heat transfer coefficient* combines the heat transfer coefficient of the two heat exchanger fluids and the thermal conductivity of the heat exchanger tubes. U_o is specific to the heat exchanger and the fluids that are used in the heat exchanger.

$$\dot{Q} = U_o A_o \Delta T_o \quad (2-3)$$

where:

\dot{Q} = the rate heat of transfer (Btu/hr)

U_o = the overall heat transfer coefficient (Btu/hr - ft² - °F)

A_o = the overall cross-sectional area for heat transfer (ft²)

ΔT_o = the overall temperature difference (°F)

Bulk Temperature

The fluid temperature (T_b), referred to as the *bulk temperature*, varies according to the details of the situation. For flow adjacent to a hot or cold surface, T_b is the temperature of the fluid that is "far" from the surface, for instance, the center of the flow channel. For boiling or condensation, T_b is equal to the saturation temperature.

Summary

The important information in this chapter is summarized below.

Heat Transfer Terminology Summary

- Heat is energy transferred as a result of a temperature difference.
- Temperature is a measure of the amount of molecular energy contained in a substance.
- Work is a transfer of energy resulting from a force acting through a distance.
- The Second Law of Thermodynamics implies that heat will not transfer from a colder to a hotter body without some external source of energy.
- Conduction involves the transfer of heat by the interactions of atoms or molecules of a material through which the heat is being transferred.
- Convection involves the transfer of heat by the mixing and motion of macroscopic portions of a fluid.
- Radiation, or radiant heat transfer, involves the transfer of heat by electromagnetic radiation that arises due to the temperature of a body.
- Heat flux is the rate of heat transfer per unit area.
- Thermal conductivity is a measure of a substance's ability to transfer heat through itself.
- Log mean temperature difference is the ΔT that most accurately represents the ΔT for a heat exchanger.
- The local heat transfer coefficient represents a measure of the ability to transfer heat through a stagnant film layer.
- The overall heat transfer coefficient is the measure of the ability of a heat exchanger to transfer heat from one fluid to another.
- The bulk temperature is the temperature of the fluid that best represents the majority of the fluid which is not physically connected to the heat transfer site.

CONDUCTION HEAT TRANSFER

Conduction heat transfer is the transfer of thermal energy by interactions between adjacent atoms and molecules of a solid.

EO 1.6 **Given Fourier's Law of Conduction, CALCULATE the conduction heat flux in a rectangular coordinate system.**

EO 1.7 **Given the formula and the necessary values, CALCULATE the equivalent thermal resistance.**

EO 1.8 **Given Fourier's Law of Conduction, CALCULATE the conduction heat flux in a cylindrical coordinate system.**

Conduction

Conduction involves the transfer of heat by the interaction between adjacent molecules of a material. Heat transfer by conduction is dependent upon the driving "force" of temperature difference and the resistance to heat transfer. The resistance to heat transfer is dependent upon the nature and dimensions of the heat transfer medium. All heat transfer problems involve the temperature difference, the geometry, and the physical properties of the object being studied.

In conduction heat transfer problems, the object being studied is usually a solid. Convection problems involve a fluid medium. Radiation heat transfer problems involve either solid or fluid surfaces, separated by a gas, vapor, or vacuum. There are several ways to correlate the geometry, physical properties, and temperature difference of an object with the rate of heat transfer through the object. In conduction heat transfer, the most common means of correlation is through Fourier's Law of Conduction. The law, in its equation form, is used most often in its rectangular or cylindrical form (pipes and cylinders), both of which are presented below.

$$\text{Rectangular} \quad \dot{Q} = k A \left(\frac{\Delta T}{\Delta x} \right) \quad (2-4)$$

$$\text{Cylindrical} \quad \dot{Q} = k A \left(\frac{\Delta T}{\Delta r} \right) \quad (2-5)$$

where:

- \dot{Q} = rate of heat transfer (Btu/hr)
- A = cross-sectional area of heat transfer (ft^2)
- Δx = thickness of slab (ft)
- Δr = thickness of cylindrical wall (ft)
- ΔT = temperature difference ($^{\circ}\text{F}$)
- k = thermal conductivity of slab (Btu/ft-hr- $^{\circ}\text{F}$)

The use of Equations 2-4 and 2-5 in determining the amount of heat transferred by conduction is demonstrated in the following examples.

Conduction-Rectangular Coordinates

Example:

1000 Btu/hr is conducted through a section of insulating material shown in Figure 1 that measures 1 ft^2 in cross-sectional area. The thickness is 1 in. and the thermal conductivity is $0.12 \text{ Btu/hr-ft-}^{\circ}\text{F}$. Compute the temperature difference across the material.

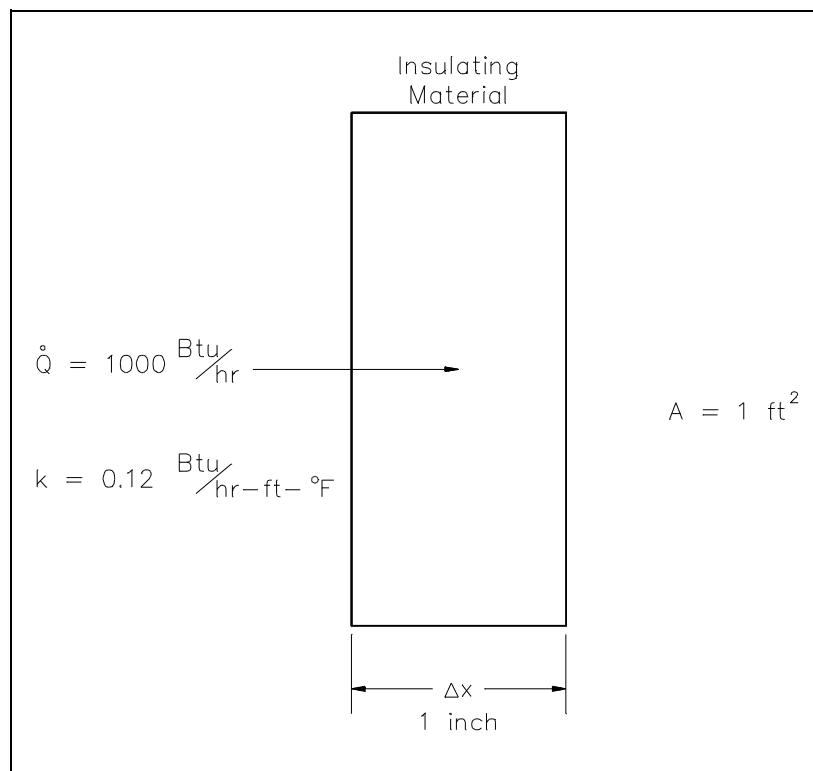


Figure 1 Conduction Through a Slab

Solution:

Using Equation 2-4:

$$\dot{Q} = k A \left(\frac{\Delta T}{\Delta x} \right)$$

Solving for ΔT :

$$\begin{aligned} \Delta T &= \dot{Q} \left(\frac{\Delta x}{k A} \right) \\ &= \frac{\left(1000 \frac{\text{Btu}}{\text{hr}} \right) \left(\frac{1}{12} \text{ ft} \right)}{\left(0.12 \frac{\text{Btu}}{\text{hr-ft-}^\circ\text{F}} \right) (1 \text{ ft}^2)} \end{aligned}$$

$$\Delta T = 694^\circ\text{F}$$

Example:

A concrete floor with a conductivity of 0.8 Btu/hr-ft-°F measures 30 ft by 40 ft with a thickness of 4 inches. The floor has a surface temperature of 70°F and the temperature beneath it is 60°F. What is the heat flux and the heat transfer rate through the floor?

Solution:

Using Equations 2-1 and 2-4:

$$\begin{aligned} \dot{Q}'' &= \frac{\dot{Q}}{A} = k \left(\frac{\Delta T}{\Delta x} \right) \\ &= \left(0.8 \frac{\text{Btu}}{\text{hr-ft-}^\circ\text{F}} \right) \left(\frac{10^\circ\text{F}}{0.333 \text{ ft}} \right) \\ &= 24 \frac{\text{Btu}}{\text{hr-ft}^2} \end{aligned}$$

Using Equation 2-3:

$$\begin{aligned}\dot{Q} &= k A \left(\frac{\Delta T}{\Delta x} \right) = \dot{Q}'' A \\ &= \left(24 \frac{\text{Btu}}{\text{hr-ft}^2} \right) (1200 \text{ ft}^2) \\ &= 28,800 \frac{\text{Btu}}{\text{hr}}\end{aligned}$$

Equivalent Resistance Method

It is possible to compare heat transfer to current flow in electrical circuits. The heat transfer rate may be considered as a current flow and the combination of thermal conductivity, thickness of material, and area as a resistance to this flow. The temperature difference is the potential or driving function for the heat flow, resulting in the Fourier equation being written in a form similar to Ohm's Law of Electrical Circuit Theory. If the thermal resistance term $\Delta x/k$ is written as a resistance term where the resistance is the reciprocal of the thermal conductivity divided by the thickness of the material, the result is the conduction equation being analogous to electrical systems or networks. The electrical analogy may be used to solve complex problems involving both series and parallel thermal resistances. The student is referred to Figure 2, showing the equivalent resistance circuit. A typical conduction problem in its analogous electrical form is given in the following example, where the "electrical" Fourier equation may be written as follows.

$$\dot{Q}'' = \frac{\Delta T}{R_{th}} \quad (2-6)$$

where:

$$\dot{Q}'' = \text{Heat Flux } (\dot{Q}/A) \text{ (Btu/hr-ft}^2\text{)}$$

$$\Delta T = \text{Temperature Difference } (^\circ\text{F)}$$

$$R_{th} = \text{Thermal Resistance } (\Delta x/k) \text{ (hr-ft}^2\text{-}^\circ\text{F/Btu)}$$

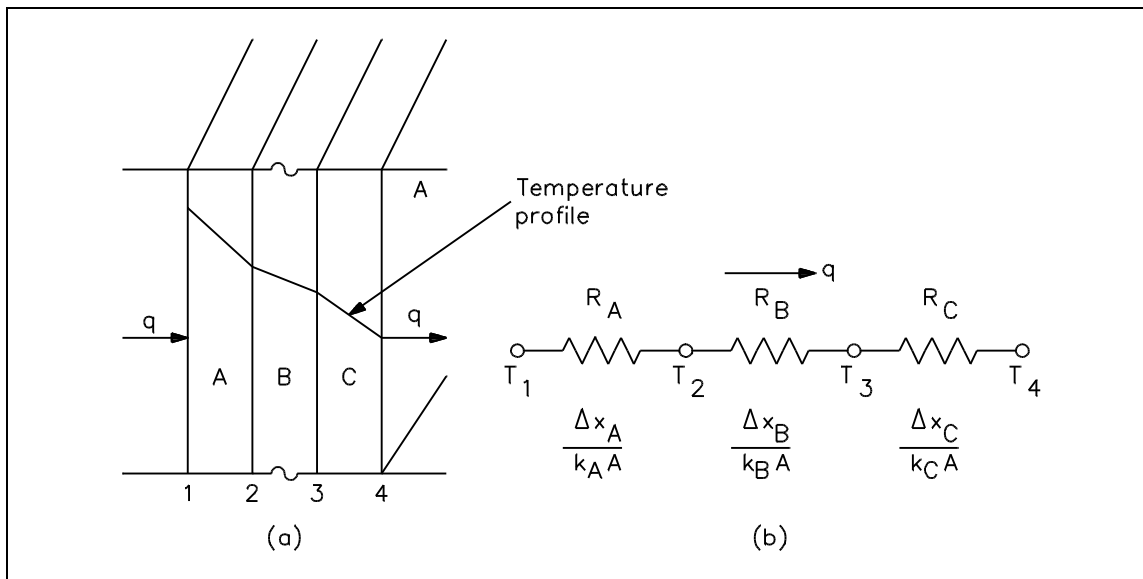


Figure 2 Equivalent Resistance

Electrical Analogy

Example:

A composite protective wall is formed of a 1 in. copper plate, a 1/8 in. layer of asbestos, and a 2 in. layer of fiberglass. The thermal conductivities of the materials in units of Btu/hr-ft-°F are as follows: $k_{Cu} = 240$, $k_{asb} = 0.048$, and $k_{fib} = 0.022$. The overall temperature difference across the wall is 500°F. Calculate the thermal resistance of each layer of the wall and the heat transfer rate per unit area (heat flux) through the composite structure.

Solution:

$$\begin{aligned}
 R_{Cu} &= \frac{\Delta x_{Cu}}{k_{Cu}} & R_{asb} &= \frac{\Delta x_{asb}}{k_{asb}} & R_{fib} &= \frac{\Delta x_{fib}}{k_{fib}} \\
 &= \frac{1 \text{ in} \left(\frac{1 \text{ ft}}{12 \text{ in}} \right)}{240 \frac{\text{Btu}}{\text{hr-ft-}^\circ\text{F}}} & &= \frac{0.125 \text{ in} \left(\frac{1 \text{ ft}}{12 \text{ in}} \right)}{0.048 \frac{\text{Btu}}{\text{hr-ft-}^\circ\text{F}}} & &= \frac{2 \text{ in} \left(\frac{1 \text{ ft}}{12 \text{ in}} \right)}{0.022 \frac{\text{Btu}}{\text{hr-ft-}^\circ\text{F}}} \\
 &= 0.000347 \frac{\text{hr-ft}^2\text{-}^\circ\text{F}}{\text{Btu}} & &= 0.2170 \frac{\text{hr-ft}^2\text{-}^\circ\text{F}}{\text{Btu}} & &= 7.5758 \frac{\text{hr-ft}^2\text{-}^\circ\text{F}}{\text{Btu}}
 \end{aligned}$$

$$\begin{aligned} \frac{\dot{Q}}{A} &= \frac{(T_i - T_o)}{(R_{Cu} + R_{asb} + R_{fib})} \\ &= \frac{500^\circ\text{F}}{(0.000347 + 0.2170 + 7.5758) \frac{\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F}}{\text{Btu}}} \\ &= 64.2 \frac{\text{Btu}}{\text{hr}\cdot\text{ft}^2} \end{aligned}$$

Conduction-Cylindrical Coordinates

Heat transfer across a rectangular solid is the most direct application of Fourier's law. Heat transfer across a pipe or heat exchanger tube wall is more complicated to evaluate. Across a cylindrical wall, the heat transfer surface area is continually increasing or decreasing. Figure 3 is a cross-sectional view of a pipe constructed of a homogeneous material.

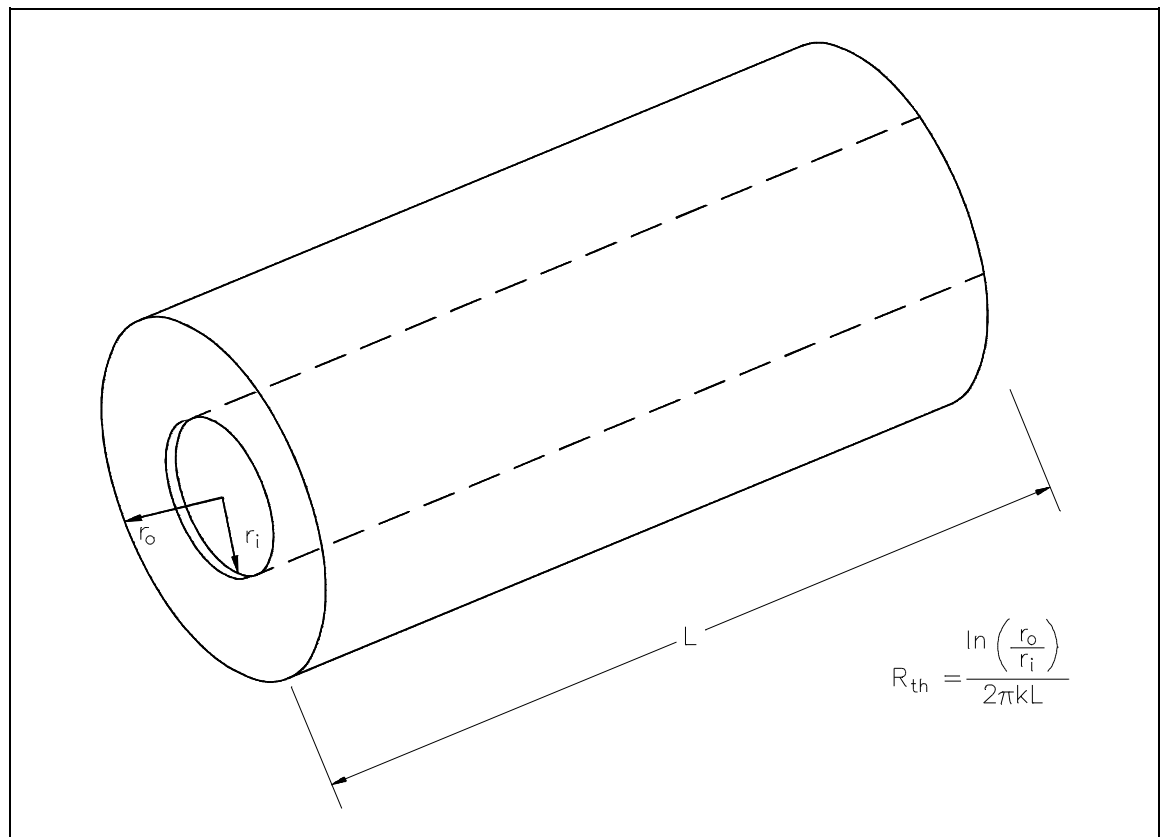


Figure 3 Cross-sectional Surface Area of a Cylindrical Pipe

The surface area (A) for transferring heat through the pipe (neglecting the pipe ends) is directly proportional to the radius (r) of the pipe and the length (L) of the pipe.

$$A = 2\pi rL$$

As the radius increases from the inner wall to the outer wall, the heat transfer area increases.

The development of an equation evaluating heat transfer through an object with cylindrical geometry begins with Fourier's law Equation 2-5.

$$\dot{Q} = k A \left(\frac{\Delta T}{\Delta r} \right)$$

From the discussion above, it is seen that no simple expression for area is accurate. Neither the area of the inner surface nor the area of the outer surface alone can be used in the equation. For a problem involving cylindrical geometry, it is necessary to define a log mean cross-sectional area (A_{lm}).

$$A_{lm} = \frac{A_{outer} - A_{inner}}{\ln \left(\frac{A_{outer}}{A_{inner}} \right)} \quad (2-7)$$

Substituting the expression $2\pi rL$ for area in Equation 2-7 allows the log mean area to be calculated from the inner and outer radius without first calculating the inner and outer area.

$$\begin{aligned} A_{lm} &= \frac{2\pi r_{outer} L - 2\pi r_{inner} L}{\ln \left(\frac{2\pi r_{outer} L}{2\pi r_{inner} L} \right)} \\ &= 2\pi L \left(\frac{r_{outer} - r_{inner}}{\ln \frac{r_{outer}}{r_{inner}}} \right) \end{aligned}$$

This expression for log mean area can be inserted into Equation 2-5, allowing us to calculate the heat transfer rate for cylindrical geometries.

$$\begin{aligned}\dot{Q} &= k A_{lm} \left(\frac{\Delta T}{\Delta r} \right) \\ &= k \left[2 \pi L \left(\frac{r_o - r_i}{\ln \frac{r_o}{r_i}} \right) \right] \left(\frac{T_o - T_i}{r_o - r_i} \right) \\ \dot{Q} &= \frac{2 \pi k L (\Delta T)}{\ln(r_o/r_i)}\end{aligned}\quad (2-8)$$

where:

L = length of pipe (ft)

r_i = inside pipe radius (ft)

r_o = outside pipe radius (ft)

Example:

A stainless steel pipe with a length of 35 ft has an inner diameter of 0.92 ft and an outer diameter of 1.08 ft. The temperature of the inner surface of the pipe is 122°F and the temperature of the outer surface is 118°F. The thermal conductivity of the stainless steel is 108 Btu/hr-ft-°F.

Calculate the heat transfer rate through the pipe.

Calculate the heat flux at the outer surface of the pipe.

Solution:

$$\begin{aligned}\dot{Q} &= \frac{2 \pi k L (T_h - T_c)}{\ln(r_o/r_i)} \\ &= \frac{6.28 \left(108 \frac{\text{Btu}}{\text{hr-ft-}^\circ\text{F}} \right) (35 \text{ ft}) (122^\circ\text{F} - 118^\circ\text{F})}{\ln \frac{0.54 \text{ ft}}{0.46 \text{ ft}}} \\ &= 5.92 \times 10^5 \frac{\text{Btu}}{\text{hr}}\end{aligned}$$

$$\begin{aligned}
 \dot{Q}'' &= \frac{\dot{Q}}{A} \\
 &= \frac{\dot{Q}}{2 \pi r_o L} \\
 &= \frac{5.92 \times 10^5 \frac{\text{Btu}}{\text{hr}}}{2 (3.14) (0.54 \text{ ft}) (35 \text{ ft})} \\
 &= 4985 \frac{\text{Btu}}{\text{hr-ft}^2}
 \end{aligned}$$

Example:

A 10 ft length of pipe with an inner radius of 1 in and an outer radius of 1.25 in has an outer surface temperature of 250°F. The heat transfer rate is 30,000 Btu/hr. Find the interior surface temperature. Assume $k = 25 \text{ Btu/hr-ft-}^\circ\text{F}$.

Solution:

$$\dot{Q} = \frac{2 \pi k L (T_h - T_c)}{\ln(r_o/r_i)}$$

Solving for T_h :

$$\begin{aligned}
 T_h &= \frac{\dot{Q} \ln(r_o/r_i)}{2 \pi k L} + T_c \\
 &= \frac{\left(30,000 \frac{\text{Btu}}{\text{hr}}\right) \left(\ln \frac{1.25 \text{ in}}{1 \text{ in}}\right)}{2 (3.14) \left(25 \frac{\text{Btu}}{\text{hr-ft-}^\circ\text{F}}\right) (10 \text{ ft})} + 250^\circ\text{F} \\
 &= 254^\circ\text{F}
 \end{aligned}$$

The evaluation of heat transfer through a cylindrical wall can be extended to include a composite body composed of several concentric, cylindrical layers, as shown in Figure 4.

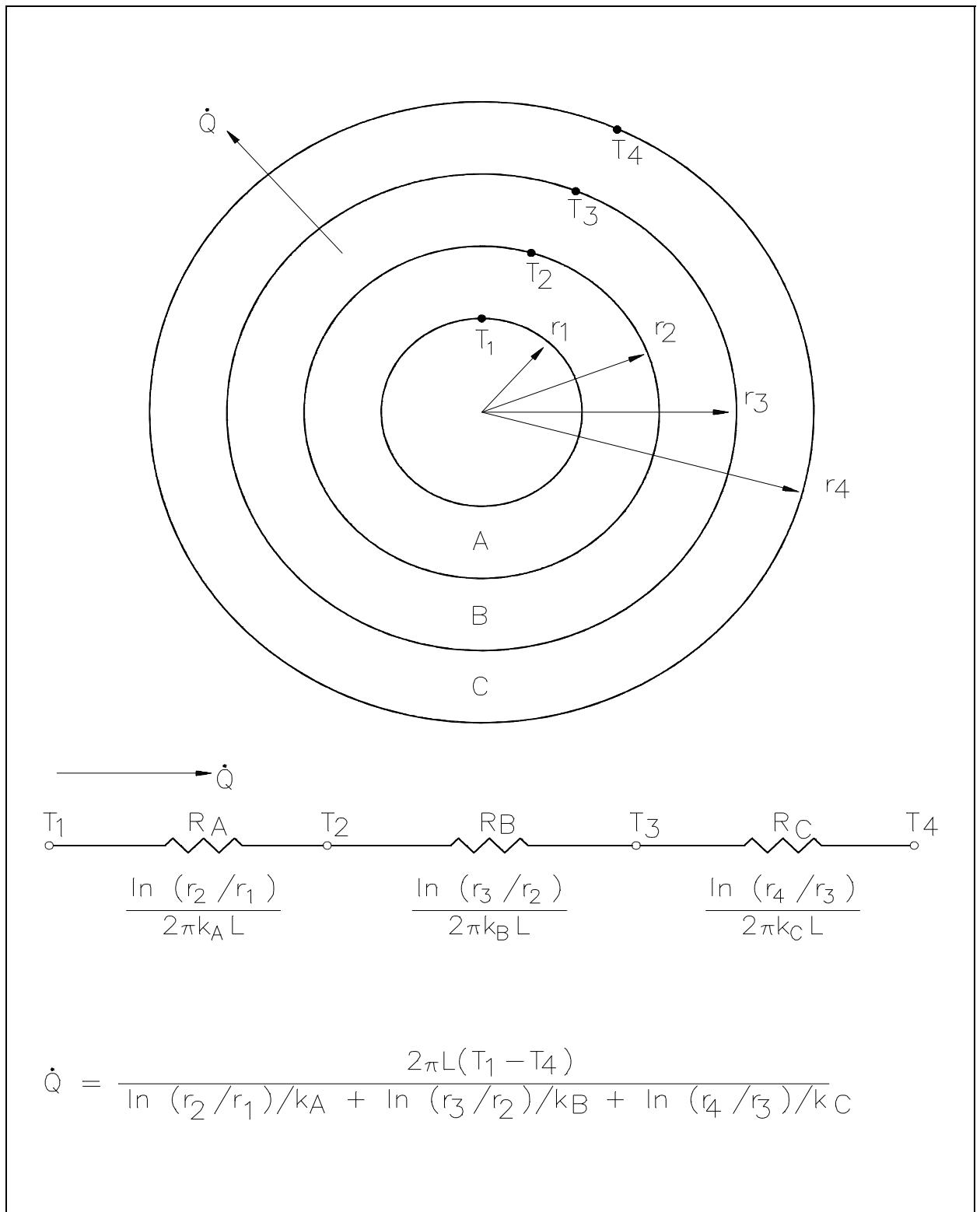


Figure 4 Composite Cylindrical Layers

Example:

A thick-walled nuclear coolant pipe ($k_s = 12.5$ Btu/hr-ft-°F) with 10 in. inside diameter (ID) and 12 in. outside diameter (OD) is covered with a 3 in. layer of asbestos insulation ($k_a = 0.14$ Btu/hr-ft-°F) as shown in Figure 5. If the inside wall temperature of the pipe is maintained at 550°F, calculate the heat loss per foot of length. The outside temperature is 100°F.

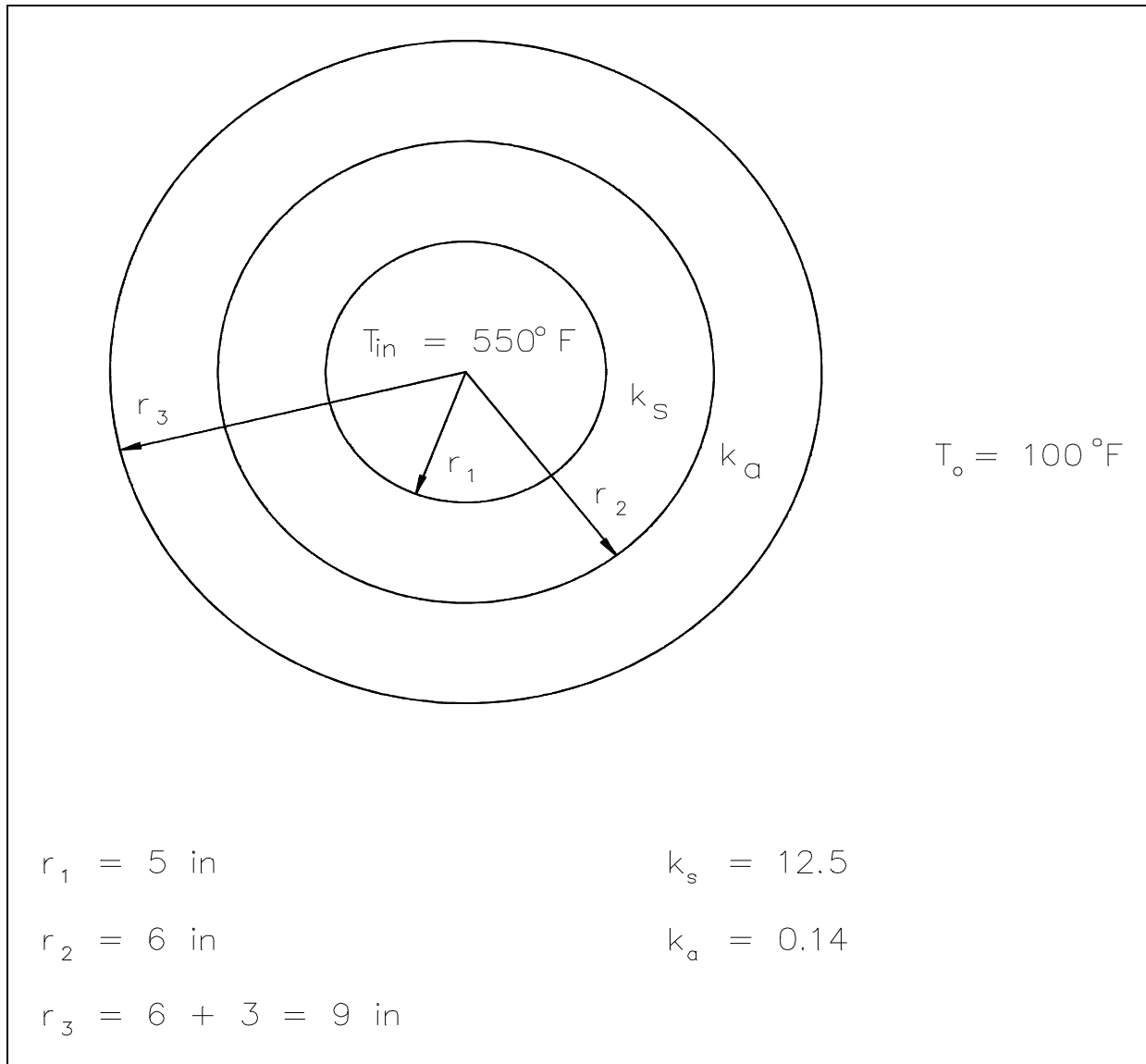


Figure 5 Pipe Insulation Problem

Solution:

$$\begin{aligned} \frac{\dot{Q}}{L} &= \frac{2\pi (T_{in} - T_o)}{\left[\frac{\ln\left(\frac{r_2}{r_1}\right)}{k_s} + \frac{\ln\left(\frac{r_3}{r_2}\right)}{k_a} \right]} \\ &= \frac{2\pi (550^\circ\text{F} - 100^\circ\text{F})}{\left[\frac{\ln\left(\frac{6 \text{ in}}{5 \text{ in}}\right)}{12.5 \frac{\text{Btu}}{\text{hr}\cdot\text{ft}\cdot^\circ\text{F}}} + \frac{\ln\left(\frac{9 \text{ in}}{6 \text{ in}}\right)}{0.14 \frac{\text{Btu}}{\text{hr}\cdot\text{ft}\cdot^\circ\text{F}}} \right]} \\ &= 971 \frac{\text{Btu}}{\text{hr}\cdot\text{ft}} \end{aligned}$$

Summary

The important information in this chapter is summarized below.

Conduction Heat Transfer Summary

- Conduction heat transfer is the transfer of thermal energy by interactions between adjacent molecules of a material.
- Fourier's Law of Conduction can be used to solve for rectangular and cylindrical coordinate problems.
- Heat flux (\dot{Q}'') is the heat transfer rate (\dot{Q}) divided by the area (A).
- Heat conductance problems can be solved using equivalent resistance formulas analogous to electrical circuit problems.

CONVECTION HEAT TRANSFER

Heat transfer by the motion and mixing of the molecules of a liquid or gas is called convection.

EO 1.9 **Given the formula for heat transfer and the operating conditions of the system, CALCULATE the rate of heat transfer by convection.**

Convection

Convection involves the transfer of heat by the motion and mixing of "macroscopic" portions of a fluid (that is, the flow of a fluid past a solid boundary). The term natural convection is used if this motion and mixing is caused by density variations resulting from temperature differences within the fluid. The term forced convection is used if this motion and mixing is caused by an outside force, such as a pump. The transfer of heat from a hot water radiator to a room is an example of heat transfer by natural convection. The transfer of heat from the surface of a heat exchanger to the bulk of a fluid being pumped through the heat exchanger is an example of forced convection.

Heat transfer by convection is more difficult to analyze than heat transfer by conduction because no single property of the heat transfer medium, such as thermal conductivity, can be defined to describe the mechanism. Heat transfer by convection varies from situation to situation (upon the fluid flow conditions), and it is frequently coupled with the mode of fluid flow. In practice, analysis of heat transfer by convection is treated empirically (by direct observation).

Convection heat transfer is treated empirically because of the factors that affect the stagnant film thickness:

- Fluid velocity
- Fluid viscosity
- Heat flux
- Surface roughness
- Type of flow (single-phase/two-phase)

Convection involves the transfer of heat between a surface at a given temperature (T_s) and fluid at a bulk temperature (T_b). The exact definition of the bulk temperature (T_b) varies depending on the details of the situation. For flow adjacent to a hot or cold surface, T_b is the temperature of the fluid "far" from the surface. For boiling or condensation, T_b is the saturation temperature of the fluid. For flow in a pipe, T_b is the average temperature measured at a particular cross-section of the pipe.

The basic relationship for heat transfer by convection has the same form as that for heat transfer by conduction:

$$\dot{Q} = h A \Delta T \quad (2-9)$$

where:

$$\dot{Q} = \text{rate of heat transfer (Btu/hr)}$$

$$h = \text{convective heat transfer coefficient (Btu/hr-ft}^2\text{-}^\circ\text{F)}$$

$$A = \text{surface area for heat transfer (ft}^2\text{)}$$

$$\Delta T = \text{temperature difference (}^\circ\text{F)}$$

The convective heat transfer coefficient (h) is dependent upon the physical properties of the fluid and the physical situation. Typically, the convective heat transfer coefficient for laminar flow is relatively low compared to the convective heat transfer coefficient for turbulent flow. This is due to turbulent flow having a thinner stagnant fluid film layer on the heat transfer surface. Values of h have been measured and tabulated for the commonly encountered fluids and flow situations occurring during heat transfer by convection.

Example:

A 22 foot uninsulated steam line crosses a room. The outer diameter of the steam line is 18 in. and the outer surface temperature is 280°F. The convective heat transfer coefficient for the air is 18 Btu/hr-ft²-°F. Calculate the heat transfer rate from the pipe into the room if the room temperature is 72°F.

Solution:

$$\begin{aligned} \dot{Q} &= h A \Delta T \\ &= h (2 \pi r L) \Delta T \\ &= \left(18 \frac{\text{Btu}}{\text{hr-ft}^2\text{-}^\circ\text{F}} \right) (2 (3.14) (0.75 \text{ ft}) (22 \text{ ft})) (280^\circ\text{F} - 72^\circ\text{F}) \\ &= 3.88 \times 10^5 \frac{\text{Btu}}{\text{hr}} \end{aligned}$$

Many applications involving convective heat transfer take place within pipes, tubes, or some similar cylindrical device. In such circumstances, the surface area of heat transfer normally given in the convection equation ($\dot{Q} = h A \Delta T$) varies as heat passes through the cylinder. In addition, the temperature difference existing between the inside and the outside of the pipe, as well as the temperature differences along the pipe, necessitates the use of some average temperature value in order to analyze the problem. This average temperature difference is called the log mean temperature difference (LMTD), described earlier.

It is the temperature difference at one end of the heat exchanger minus the temperature difference at the other end of the heat exchanger, divided by the natural logarithm of the ratio of these two temperature differences. The above definition for LMTD involves two important assumptions: (1) the fluid specific heats do not vary significantly with temperature, and (2) the convection heat transfer coefficients are relatively constant throughout the heat exchanger.

Overall Heat Transfer Coefficient

Many of the heat transfer processes encountered in nuclear facilities involve a combination of both conduction and convection. For example, heat transfer in a steam generator involves convection from the bulk of the reactor coolant to the steam generator inner tube surface, conduction through the tube wall, and convection from the outer tube surface to the secondary side fluid.

In cases of combined heat transfer for a heat exchanger, there are two values for h . There is the convective heat transfer coefficient (h) for the fluid film inside the tubes and a convective heat transfer coefficient for the fluid film outside the tubes. The thermal conductivity (k) and thickness (Δx) of the tube wall must also be accounted for. An additional term (U_o), called the overall heat transfer coefficient, must be used instead. It is common practice to relate the total rate of heat transfer (\dot{Q}) to the cross-sectional area for heat transfer (A_o) and the overall heat transfer coefficient (U_o). The relationship of the overall heat transfer coefficient to the individual conduction and convection terms is shown in Figure 6.

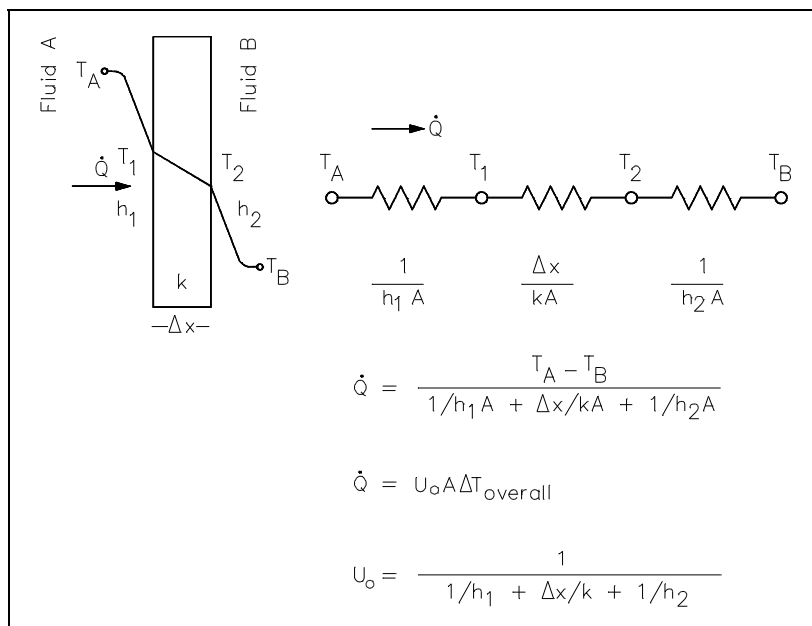


Figure 6 Overall Heat Transfer Coefficient

Recalling Equation 2-3:

$$\dot{Q} = U_o A_o \Delta T_o$$

where U_o is defined in Figure 6.

An example of this concept applied to cylindrical geometry is illustrated by Figure 7, which shows a typical combined heat transfer situation.

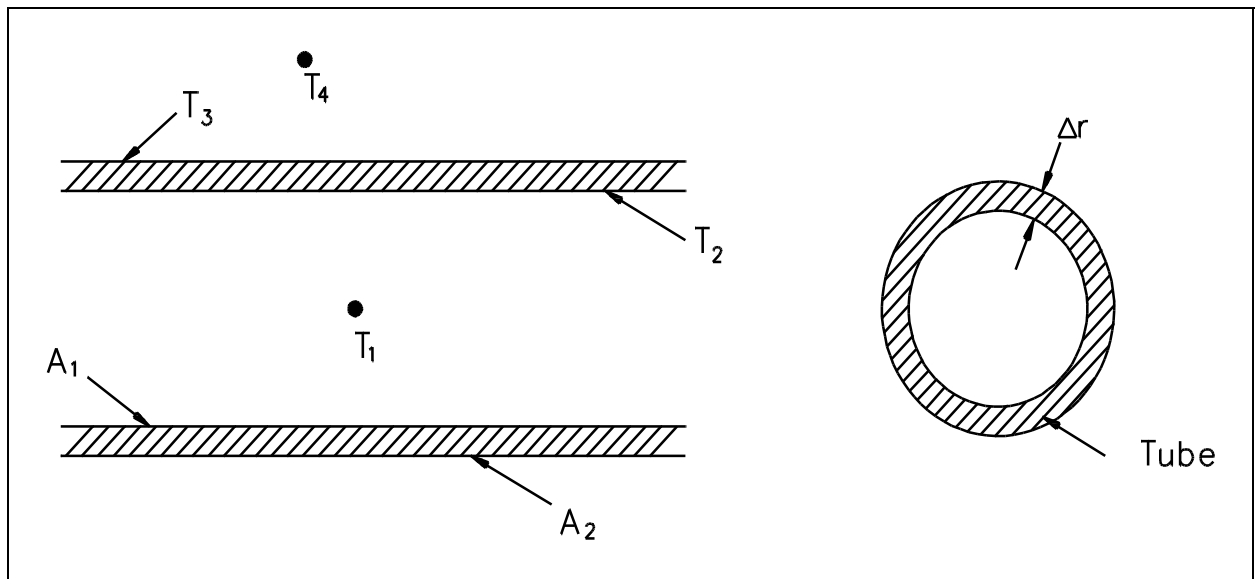


Figure 7 Combined Heat Transfer

Using the figure representing flow in a pipe, heat transfer by convection occurs between temperatures T_1 and T_2 ; heat transfer by conduction occurs between temperatures T_2 and T_3 ; and heat transfer occurs by convection between temperatures T_3 and T_4 . Thus, there are three processes involved. Each has an associated heat transfer coefficient, cross-sectional area for heat transfer, and temperature difference. The basic relationships for these three processes can be expressed using Equations 2-5 and 2-9.

$$\dot{Q} = h_1 A_1 (T_1 - T_2)$$

$$\dot{Q} = \frac{k}{\Delta r} A_{lm} (T_2 - T_3)$$

$$\dot{Q} = h_2 A_2 (T_3 - T_4)$$

ΔT_o can be expressed as the sum of the ΔT of the three individual processes.

$$\Delta T_o = (T_1 - T_2) + (T_2 - T_3) + (T_3 - T_4)$$

If the basic relationship for each process is solved for its associated temperature difference and substituted into the expression for ΔT_o above, the following relationship results.

$$\Delta T_o = \dot{Q} \left(\frac{1}{h_1 A_1} + \frac{\Delta r}{k A_{lm}} + \frac{1}{h_2 A_2} \right)$$

This relationship can be modified by selecting a reference cross-sectional area A_o .

$$\Delta T_o = \frac{\dot{Q}}{A_o} \left(\frac{A_o}{h_1 A_1} + \frac{\Delta r A_o}{k A_{lm}} + \frac{A_o}{h_2 A_2} \right)$$

Solving for \dot{Q} results in an equation in the form $\dot{Q} = U_o A_o \Delta T_o$.

$$\dot{Q} = \frac{1}{\left(\frac{A_o}{h_1 A_1} + \frac{\Delta r A_o}{k A_{lm}} + \frac{A_o}{h_2 A_2} \right)} A_o \Delta T_o$$

where:

$$U_o = \frac{1}{\left(\frac{A_o}{h_1 A_1} + \frac{\Delta r A_o}{k A_{lm}} + \frac{A_o}{h_2 A_2} \right)} \quad (2-10)$$

Equation 2-10 for the overall heat transfer coefficient in cylindrical geometry is relatively difficult to work with. The equation can be simplified without losing much accuracy if the tube that is being analyzed is thin-walled, that is the tube wall thickness is small compared to the tube diameter. For a thin-walled tube, the inner surface area (A_1), outer surface area (A_2), and log mean surface area (A_{lm}), are all very close to being equal. Assuming that A_1 , A_2 , and A_{lm} are equal to each other and also equal to A_o allows us to cancel out all the area terms in the denominator of Equation 2-11.

This results in a much simpler expression that is similar to the one developed for a flat plate heat exchanger in Figure 6.

$$U_o = \frac{1}{\frac{1}{h_1} + \frac{\Delta r}{k} + \frac{1}{h_2}} \quad (2-11)$$

The convection heat transfer process is strongly dependent upon the properties of the fluid being considered. Correspondingly, the convective heat transfer coefficient (h), the overall coefficient (U_o), and the other fluid properties may vary substantially for the fluid if it experiences a large temperature change during its path through the convective heat transfer device. This is especially true if the fluid's properties are strongly temperature dependent. Under such circumstances, the temperature at which the properties are "looked-up" must be some type of average value, rather than using either the inlet or outlet temperature value.

For internal flow, the bulk or average value of temperature is obtained analytically through the use of conservation of energy. For external flow, an average film temperature is normally calculated, which is an average of the free stream temperature and the solid surface temperature. In any case, an average value of temperature is used to obtain the fluid properties to be used in the heat transfer problem. The following example shows the use of such principles by solving a convective heat transfer problem in which the bulk temperature is calculated.

Convection Heat Transfer

Example:

A flat wall is exposed to the environment. The wall is covered with a layer of insulation 1 in. thick whose thermal conductivity is 0.8 Btu/hr-ft-°F. The temperature of the wall on the inside of the insulation is 600°F. The wall loses heat to the environment by convection on the surface of the insulation. The average value of the convection heat transfer coefficient on the insulation surface is 950 Btu/hr-ft²-°F. Compute the bulk temperature of the environment (T_b) if the outer surface of the insulation does not exceed 105°F.

Solution:

- a. Find heat flux (\dot{Q}'') through the insulation.

$$\dot{Q} = k A \left(\frac{\Delta T}{\Delta x} \right)$$
$$\frac{\dot{Q}}{A} = 0.8 \frac{\text{Btu}}{\text{hr-ft-}^\circ\text{F}} \left(\frac{600^\circ\text{F} - 105^\circ\text{F}}{1 \text{ in } \frac{1 \text{ ft}}{12 \text{ in}}} \right)$$
$$= 4752 \frac{\text{Btu}}{\text{hr-ft}^2}$$

- b. Find the bulk temperature of the environment.

$$\dot{Q} = h A (T_{\text{ins}} - T_{\text{b}})$$
$$(T_{\text{ins}} - T_{\text{b}}) = \frac{\dot{Q}}{h A}$$
$$T_{\text{b}} = T_{\text{ins}} - \frac{\dot{Q}''}{h}$$
$$T_{\text{b}} = 105^\circ\text{F} - \frac{4752 \frac{\text{Btu}}{\text{hr-ft}^2}}{950 \frac{\text{Btu}}{\text{hr-ft}^2-^\circ\text{F}}}$$
$$T_{\text{b}} = 100^\circ\text{F}$$

Summary

The important information in this chapter is summarized below.

Convection Heat Transfer Summary

- Convection heat transfer is the transfer of thermal energy by the mixing and motion of a fluid or gas.
- Whether convection is natural or forced is determined by how the medium is placed into motion.
- When both convection and conduction heat transfer occurs, the overall heat transfer coefficient must be used to solve problems.
- The heat transfer equation for convection heat transfer is $\dot{Q} = hA\Delta T$.

RADIANT HEAT TRANSFER

Radiant heat transfer is thermal energy transferred by means of electromagnetic waves or particles.

- EO 1.10** **DESCRIBE** how the following terms relate to radiant heat transfer:
- a. **Black body radiation**
 - b. **Emissivity**
 - c. **Radiation configuration factor**
-

Thermal Radiation

Radiant heat transfer involves the transfer of heat by electromagnetic radiation that arises due to the temperature of a body. Most energy of this type is in the infra-red region of the electromagnetic spectrum although some of it is in the visible region. The term thermal radiation is frequently used to distinguish this form of electromagnetic radiation from other forms, such as radio waves, x-rays, or gamma rays. The transfer of heat from a fireplace across a room in the line of sight is an example of radiant heat transfer.

Radiant heat transfer does not need a medium, such as air or metal, to take place. Any material that has a temperature above absolute zero gives off some radiant energy. When a cloud covers the sun, both its heat and light diminish. This is one of the most familiar examples of heat transfer by thermal radiation.

Black Body Radiation

A body that emits the maximum amount of heat for its absolute temperature is called a black body. Radiant heat transfer rate from a black body to its surroundings can be expressed by the following equation.

$$\dot{Q} = \sigma AT^4 \quad (2-12)$$

where:

\dot{Q} = heat transfer rate (Btu/hr)

σ = Stefan-Boltzman constant (0.174 Btu/hr-ft²-°R⁴)

A = surface area (ft²)

T = temperature (°R)

Two black bodies that radiate toward each other have a net heat flux between them. The net flow rate of heat between them is given by an adaptation of Equation 2-12.

$$\dot{Q} = \sigma A (T_1^4 - T_2^4)$$

where:

A = surface area of the first body (ft²)

T₁ = temperature of the first body (°R)

T₂ = temperature of the second body (°R)

All bodies above absolute zero temperature radiate some heat. The sun and earth both radiate heat toward each other. This seems to violate the Second Law of Thermodynamics, which states that heat cannot flow from a cold body to a hot body. The paradox is resolved by the fact that each body must be in direct line of sight of the other to receive radiation from it. Therefore, whenever the cool body is radiating heat to the hot body, the hot body must also be radiating heat to the cool body. Since the hot body radiates more heat (due to its higher temperature) than the cold body, the net flow of heat is from hot to cold, and the second law is still satisfied.

Emissivity

Real objects do not radiate as much heat as a perfect black body. They radiate less heat than a black body and are called gray bodies. To take into account the fact that real objects are gray bodies, Equation 2-12 is modified to be of the following form.

$$\dot{Q} = \varepsilon \sigma A T^4$$

where:

ε = emissivity of the gray body (dimensionless)

Emissivity is simply a factor by which we multiply the black body heat transfer to take into account that the black body is the ideal case. Emissivity is a dimensionless number and has a maximum value of 1.0.

Radiation Configuration Factor

Radiative heat transfer rate between two gray bodies can be calculated by the equation stated below.

$$\dot{Q} = f_a f_e \sigma A (T_1^4 - T_2^4)$$

where:

$f_a =$ is the shape factor, which depends on the spatial arrangement of the two objects (dimensionless)

$f_e =$ is the emissivity factor, which depends on the emissivities of both objects (dimensionless)

The two separate terms f_a and f_e can be combined and given the symbol f . The heat flow between two gray bodies can now be determined by the following equation:

$$\dot{Q} = f\sigma A(T_1^4 - T_2^4) \quad (2-13)$$

The symbol (f) is a dimensionless factor sometimes called the *radiation configuration factor*, which takes into account the emissivity of both bodies and their relative geometry. The radiation configuration factor is usually found in a text book for the given situation. Once the configuration factor is obtained, the overall net heat flux can be determined. Radiant heat flux should only be included in a problem when it is greater than 20% of the problem.

Example:

Calculate the radiant heat between the floor (15 ft x 15 ft) of a furnace and the roof, if the two are located 10 ft apart. The floor and roof temperatures are 2000°F and 600°F, respectively. Assume that the floor and the roof have black surfaces.

Solution:

$$A_1 = A_2 = (15 \text{ ft}) (15 \text{ ft}) = 225 \text{ ft}^2$$

$$T_1 = 2000^\circ\text{F} + 460 = 2460^\circ\text{R}$$

$$T_2 = 600^\circ\text{F} + 460 = 1060^\circ\text{R}$$

Tables from a reference book, or supplied by the instructor, give:

$$f_{1-2} = f_{2-1} = 0.31$$

$$Q_{1-2} = \sigma A f (T_1^4 - T_2^4)$$

$$= (0.174 \frac{\text{Btu}}{\text{hr-ft}^2-\text{R}^4}) (225 \text{ ft}^2) (0.31) [(2460^\circ\text{R})^4 - (1060^\circ\text{R})^4]$$

$$= 4.29 \times 10^{14} \text{ Btu/hr}$$

Summary

The important information in this chapter is summarized below.

Radiant Heat Transfer Summary

- Black body radiation is the maximum amount of heat that can be transferred from an ideal object.
- Emissivity is a measure of the departure of a body from the ideal black body.
- Radiation configuration factor takes into account the emittance and relative geometry of two objects.

HEAT EXCHANGERS

Heat exchangers are devices that are used to transfer thermal energy from one fluid to another without mixing the two fluids.

- EO 1.11** **DESCRIBE** the difference in the temperature profiles for counter-flow and parallel flow heat exchangers.
- EO 1.12** **DESCRIBE** the differences between regenerative and non-regenerative heat exchangers.
- EO 1.13** **Given** the temperature changes across a heat exchanger, **CALCULATE** the log mean temperature difference for the heat exchanger.
- EO 1.14** **Given** the formulas for calculating the conduction and convection heat transfer coefficients, **CALCULATE** the overall heat transfer coefficient of a system.

Heat Exchangers

The transfer of thermal energy between fluids is one of the most important and frequently used processes in engineering. The transfer of heat is usually accomplished by means of a device known as a heat exchanger. Common applications of heat exchangers in the nuclear field include boilers, fan coolers, cooling water heat exchangers, and condensers.

The basic design of a heat exchanger normally has two fluids of different temperatures separated by some conducting medium. The most common design has one fluid flowing through metal tubes and the other fluid flowing around the tubes. On either side of the tube, heat is transferred by convection. Heat is transferred through the tube wall by conduction.

Heat exchangers may be divided into several categories or classifications. In the most commonly used type of heat exchanger, two fluids of different temperature flow in spaces separated by a tube wall. They transfer heat by convection and by conduction through the wall. This type is referred to as an "ordinary heat exchanger," as compared to the other two types classified as "regenerators" and "cooling towers."

An ordinary heat exchanger is single-phase or two-phase. In a single-phase heat exchanger, both of the fluids (cooled and heated) remain in their initial gaseous or liquid states. In two-phase exchangers, either of the fluids may change its phase during the heat exchange process. The steam generator and main condenser of nuclear facilities are of the two-phase, ordinary heat exchanger classification.

Single-phase heat exchangers are usually of the tube-and-shell type; that is, the exchanger consists of a set of tubes in a container called a shell (Figure 8). At the ends of the heat exchanger, the tube-side fluid is separated from the shell-side fluid by a tube sheet. The design of two-phase exchangers is essentially the same as that of single-phase exchangers.

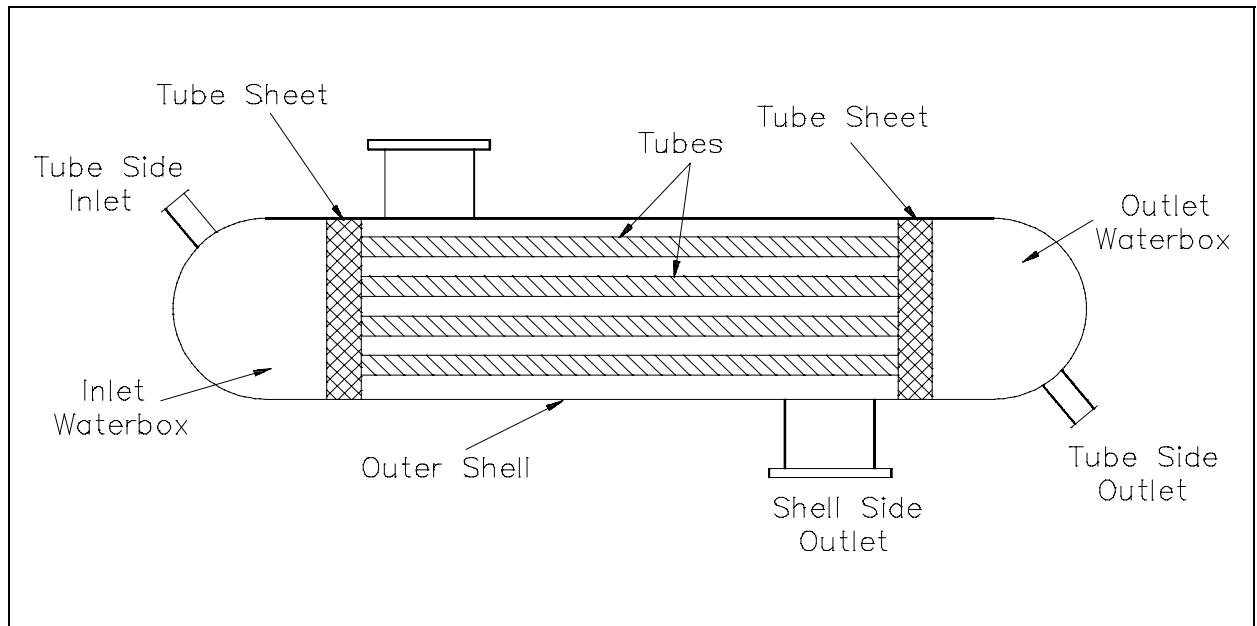


Figure 8 Typical Tube and Shell Heat Exchanger

Parallel and Counter-Flow Designs

Although ordinary heat exchangers may be extremely different in design and construction and may be of the single- or two-phase type, their modes of operation and effectiveness are largely determined by the direction of the fluid flow within the exchanger.

The most common arrangements for flow paths within a heat exchanger are counter-flow and parallel flow. A counter-flow heat exchanger is one in which the direction of the flow of one of the working fluids is opposite to the direction to the flow of the other fluid. In a parallel flow exchanger, both fluids in the heat exchanger flow in the same direction.

Figure 9 represents the directions of fluid flow in the parallel and counter-flow exchangers. Under comparable conditions, more heat is transferred in a counter-flow arrangement than in a parallel flow heat exchanger.

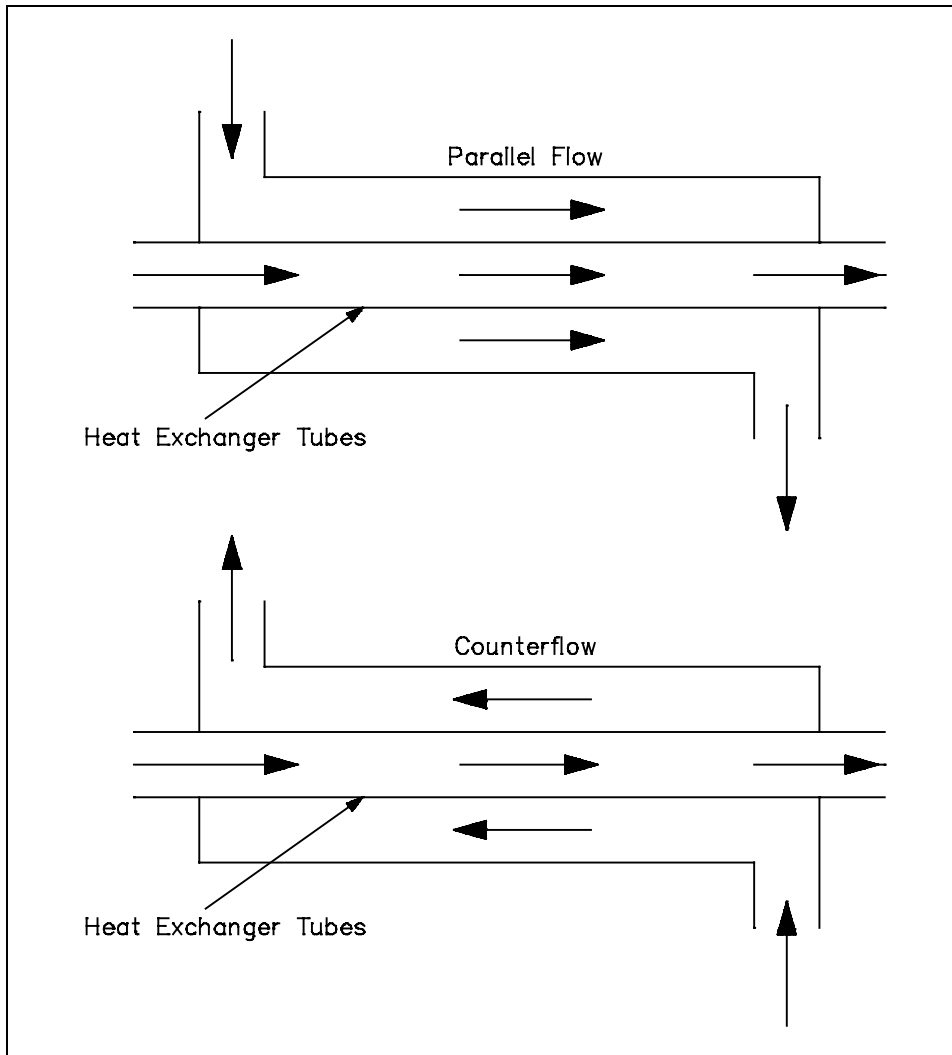


Figure 9 Fluid Flow Direction

The temperature profiles of the two heat exchangers indicate two major disadvantages in the parallel-flow design. First, the large temperature difference at the ends (Figure 10) causes large thermal stresses. The opposing expansion and contraction of the construction materials due to diverse fluid temperatures can lead to eventual material failure. Second, the temperature of the cold fluid exiting the heat exchanger never exceeds the lowest temperature of the hot fluid. This relationship is a distinct disadvantage if the design purpose is to raise the temperature of the cold fluid.

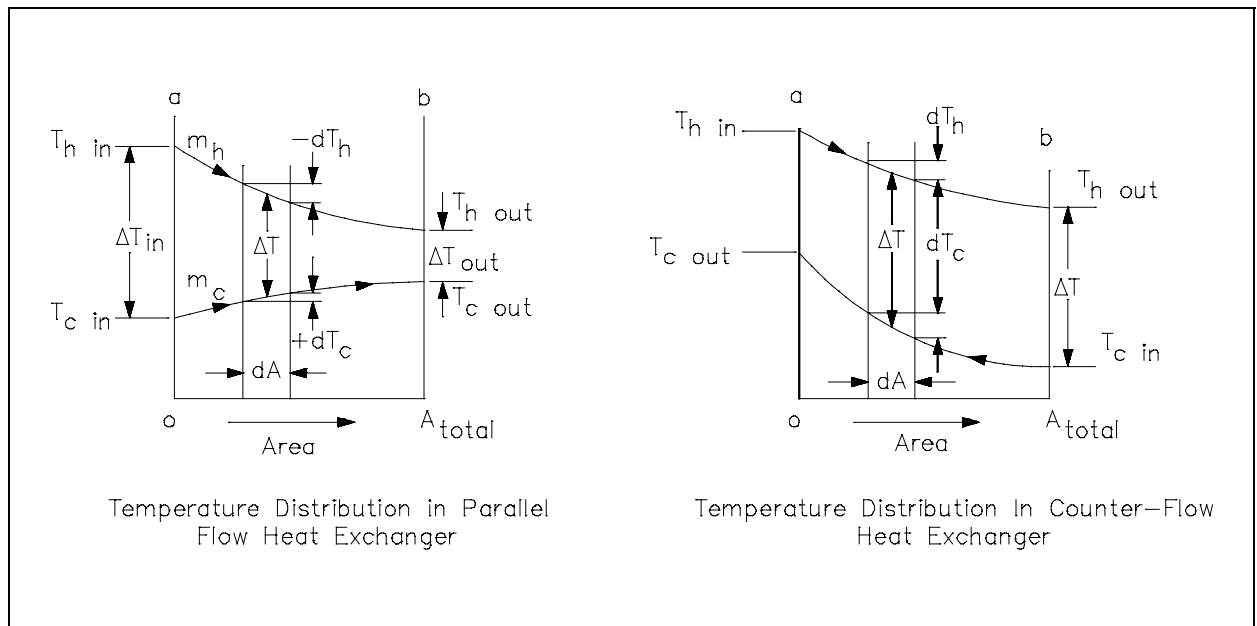


Figure 10 Heat Exchanger Temperature Profiles

The design of a parallel flow heat exchanger is advantageous when two fluids are required to be brought to nearly the same temperature.

The counter-flow heat exchanger has three significant advantages over the parallel flow design. First, the more uniform temperature difference between the two fluids minimizes the thermal stresses throughout the exchanger. Second, the outlet temperature of the cold fluid can approach the highest temperature of the hot fluid (the inlet temperature). Third, the more uniform temperature difference produces a more uniform rate of heat transfer throughout the heat exchanger.

Whether parallel or counter-flow, heat transfer within the heat exchanger involves both conduction and convection. One fluid (hot) convectively transfers heat to the tube wall where conduction takes place across the tube to the opposite wall. The heat is then convectively transferred to the second fluid. Because this process takes place over the entire length of the exchanger, the temperature of the fluids as they flow through the exchanger is not generally constant, but varies over the entire length, as indicated in Figure 10. The rate of heat transfer varies along the length of the exchanger tubes because its value depends upon the temperature difference between the hot and the cold fluid at the point being viewed.

Non-Regenerative Heat Exchanger

Applications of heat exchangers may be classified as either regenerative or non-regenerative. The non-regenerative application is the most frequent and involves two separate fluids. One fluid cools or heats the other with no interconnection between the two fluids. Heat that is removed from the hotter fluid is usually rejected to the environment or some other heat sink (Figure 11).

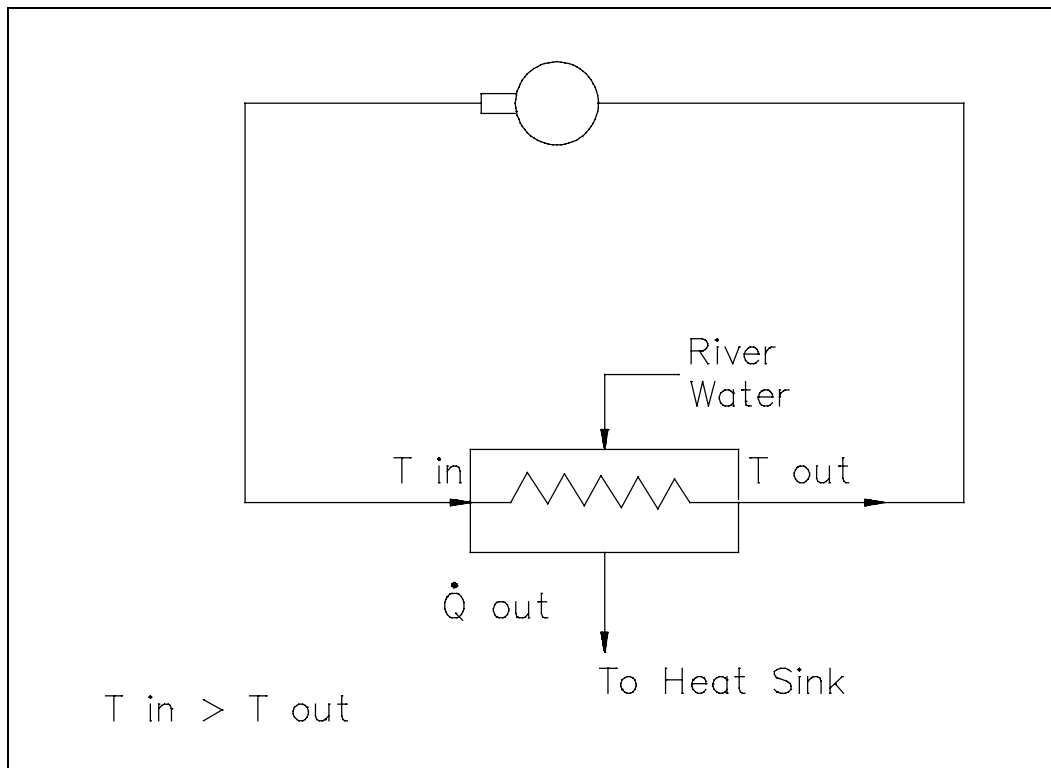


Figure 11 Non-Regenerative Heat Exchanger

Regenerative Heat Exchanger

A regenerative heat exchanger typically uses the fluid from a different area of the same system for both the hot and cold fluids. An example of both regenerative and non-regenerative heat exchangers working in conjunction is commonly found in the purification system of a reactor facility. The primary coolant to be purified is drawn out of the primary system, passed through a regenerative heat exchanger, non-regenerative heat exchanger, demineralizer, back through the regenerative heat exchanger, and returned to the primary system (Figure 12).

In the regenerative heat exchanger, the water returning to the primary system is pre-heated by the water entering the purification system. This accomplishes two objectives. The first is to minimize the thermal stress in the primary system piping due to the cold temperature of the purified coolant being returned to the primary system.

The second is to reduce the temperature of the water entering the purification system prior to reaching the non-regenerative heat exchanger, allowing use of a smaller heat exchanger to achieve the desired temperature for purification. The primary advantage of a regenerative heat exchanger application is conservation of system energy (that is, less loss of system energy due to the cooling of the fluid).

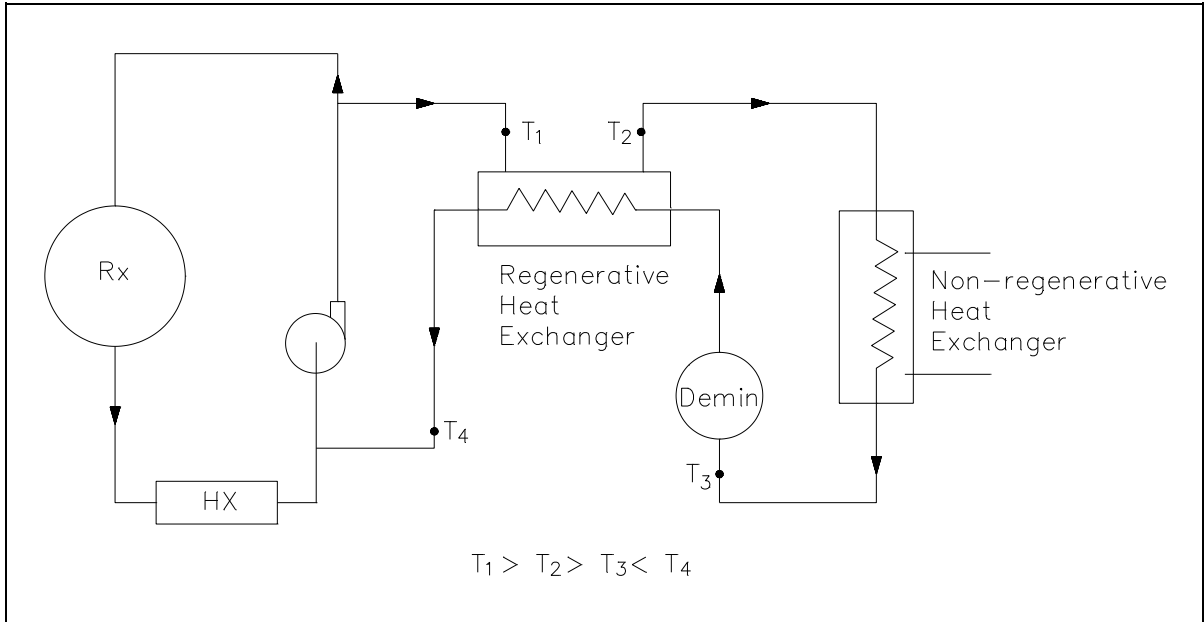


Figure 12 Regenerative Heat Exchanger

Cooling Towers

The typical function of a cooling tower is to cool the water of a steam power plant by air that is brought into direct contact with the water. The water is mixed with vapor that diffuses from the condensate into the air. The formation of the vapor requires a considerable removal of internal energy from the water; the internal energy becomes "latent heat" of the vapor. Heat and mass exchange are coupled in this process, which is a steady-state process like the heat exchange in the ordinary heat exchanger.

Wooden cooling towers are sometimes employed in nuclear facilities and in factories of various industries. They generally consists of large chambers loosely filled with trays or similar wooden elements of construction. The water to be cooled is pumped to the top of the tower where it is distributed by spray or wooden troughs. It then falls through the tower, splashing down from deck to deck. A part of it evaporates into the air that passes through the tower. The enthalpy needed for the evaporation is taken from the water and transferred to the air, which is heated while the water cools. The air flow is either horizontal due to wind currents (cross flow) or vertically upward in counter-flow to the falling water. The counter-flow is caused by the

chimney effect of the warm humid air in the tower or by fans at the bottom (forced draft) or at the top (induced flow) of the tower. Mechanical draft towers are more economical to construct and smaller in size than natural-convection towers of the same cooling capacity.

Log Mean Temperature Difference Application To Heat Exchangers

In order to solve certain heat exchanger problems, a log mean temperature difference (LMTD or ΔT_{lm}) must be evaluated before the heat removal from the heat exchanger is determined. The following example demonstrates such a calculation.

Example:

A liquid-to-liquid counterflow heat exchanger is used as part of an auxiliary system at a nuclear facility. The heat exchanger is used to heat a cold fluid from 120°F to 310°F. Assuming that the hot fluid enters at 500°F and leaves at 400°F, calculate the LMTD for the exchanger.

Solution:

$$\Delta T_2 = 400^\circ\text{F} - 120^\circ\text{F} = 280^\circ\text{F}$$

$$\Delta T_1 = 500^\circ\text{F} - 310^\circ\text{F} = 190^\circ\text{F}$$

$$\begin{aligned} \Delta T_{lm} &= \frac{(\Delta T_2 - \Delta T_1)}{\ln \left(\frac{\Delta T_2}{\Delta T_1} \right)} \\ &= \frac{(280^\circ\text{F} - 190^\circ\text{F})}{\ln \left(\frac{280^\circ\text{F}}{190^\circ\text{F}} \right)} \\ &= 232^\circ\text{F} \end{aligned}$$

The solution to the heat exchanger problem may be simple enough to be represented by a straight-forward overall balance or may be so detailed as to require integral calculus. A steam generator, for example, can be analyzed by an overall energy balance from the feedwater inlet to the steam outlet in which the amount of heat transferred can be expressed simply as $\dot{Q} = \dot{m} \Delta h$, where \dot{m} is the mass flow rate of the secondary coolant and Δh is the change in enthalpy of the fluid. The same steam generator can also be analyzed by an energy balance on the primary flow stream with the equation $\dot{Q} = \dot{m} c_p \Delta T$, where \dot{m} , c_p , and ΔT are the mass flow rate, specific heat capacity, and temperature change of the primary coolant. The heat

transfer rate of the steam generator can also be determined by comparing the temperatures on the primary and secondary sides with the heat transfer characteristics of the steam generator using the equation $\dot{Q} = U_o A_o \Delta T_{lm}$.

Condensers are also examples of components found in nuclear facilities where the concept of LMTD is needed to address certain problems. When the steam enters the condenser, it gives up its latent heat of vaporization to the circulating water and changes phase to a liquid. Because condensation is taking place, it is appropriate to term this the latent heat of condensation. After the steam condenses, the saturated liquid will continue to transfer some heat to the circulating water system as it continues to fall to the bottom (hotwell) of the condenser. This continued cooling is called subcooling and is necessary to prevent cavitation in the condensate pumps.

The solution to condenser problems is approached in the same manner as those for steam generators, as shown in the following example.

Overall Heat Transfer Coefficient

When dealing with heat transfer across heat exchanger tubes, an overall heat transfer coefficient, U_o , must be calculated. Earlier in this module we looked at a method for calculating U_o for both rectangular and cylindrical coordinates. Since the thickness of a condenser tube wall is so small and the cross-sectional area for heat transfer is relatively constant, we can use Equation 2-11 to calculate U_o .

$$U_o = \frac{1}{\frac{1}{h_1} + \frac{\Delta r}{k} + \frac{1}{h_2}}$$

Example:

Referring to the convection section of this manual, calculate the heat rate per foot of tube from a condenser under the following conditions. $\Delta T_{lm} = 232^\circ\text{F}$. The outer diameter of the copper condenser tube is 0.75 in. with a wall thickness of 0.1 in. Assume the inner convective heat transfer coefficient is 2000 Btu/hr-ft²-°F, and the thermal conductivity of copper is 200 Btu/hr-ft-°F. The outer convective heat transfer coefficient is 1500 Btu/hr-ft²-°F.

Solution:

$$\begin{aligned}U_o &= \frac{1}{\frac{1}{h_1} + \frac{\Delta r}{k} + \frac{1}{h_2}} \\&= \frac{1}{\frac{1}{2000} + \frac{0.1 \text{ in}}{200} \frac{1 \text{ ft}}{12 \text{ in}} + \frac{1}{1500}} \\&= 827.6 \frac{\text{Btu}}{\text{hr-ft}^2-\text{°F}}\end{aligned}$$

$$\dot{Q} = U_o A_o \Delta T_{lm}$$

$$\frac{\dot{Q}}{L} = \frac{U_o A_o \Delta T_{lm}}{L}$$

$$= U_o 2\pi r \Delta T_{lm}$$

$$= \left(827.6 \frac{\text{Btu}}{\text{hr-ft}^2-\text{°F}} \right) (2\pi) (0.375 \text{ in}) \left(\frac{1 \text{ ft}}{12 \text{ in}} \right) (232\text{°F})$$

$$= 37,700 \frac{\text{Btu}}{\text{hr-ft}}$$

Summary

The important information in this chapter is summarized below.

Heat Exchangers Summary

- Heat exchangers remove heat from a high-temperature fluid by convection and conduction.
- Counter-flow heat exchangers typically remove more heat than parallel flow heat exchangers.
- Parallel flow heat exchangers have a large temperature difference at the inlet and a small temperature difference at the outlet.
- Counter-flow heat exchangers have an even temperature difference across the heat transfer length.
- Regenerative heat exchangers improve system efficiency by returning energy to the system. A non-regenerative heat exchanger rejects heat to the surroundings.
- The heat transfer rate for a heat exchanger can be calculated using the equation below.

$$\dot{Q} = U_o A_o \Delta T_{lm}$$

BOILING HEAT TRANSFER

The formation of steam bubbles along a heat transfer surface has a significant effect on the overall heat transfer rate.

- EO 1.15** **DESCRIBE** the process that occurs in the following regions of the boiling heat transfer curve:
- a. **Nucleate boiling**
 - b. **Partial film boiling**
 - c. **Film boiling**
 - d. **Departure from nucleate boiling (DNB)**
 - e. **Critical heat flux**
-

Boiling

In a nuclear facility, convective heat transfer is used to remove heat from a heat transfer surface. The liquid used for cooling is usually in a compressed state, (that is, a subcooled fluid) at pressures higher than the normal saturation pressure for the given temperature. Under certain conditions, some type of boiling (usually nucleate boiling) can take place. It is advisable, therefore, to study the process of boiling as it applies to the nuclear field when discussing convection heat transfer.

More than one type of boiling can take place within a nuclear facility, especially if there is a rapid loss of coolant pressure. A discussion of the boiling processes, specifically local and bulk boiling, will help the student understand these processes and provide a clearer picture of why bulk boiling (specifically film boiling) is to be avoided in nuclear facility operations.

Nucleate Boiling

The most common type of local boiling encountered in nuclear facilities is nucleate boiling. In nucleate boiling, steam bubbles form at the heat transfer surface and then break away and are carried into the main stream of the fluid. Such movement enhances heat transfer because the heat generated at the surface is carried directly into the fluid stream. Once in the main fluid stream, the bubbles collapse because the bulk temperature of the fluid is not as high as the heat transfer surface temperature where the bubbles were created. This heat transfer process is sometimes desirable because the energy created at the heat transfer surface is quickly and efficiently "carried" away.

Bulk Boiling

As system temperature increases or system pressure drops, the bulk fluid can reach saturation conditions. At this point, the bubbles entering the coolant channel will not collapse. The bubbles will tend to join together and form bigger steam bubbles. This phenomenon is referred to as bulk boiling. Bulk boiling can provide adequate heat transfer provided that the steam bubbles are carried away from the heat transfer surface and the surface is continually wetted with liquid water. When this cannot occur film boiling results.

Film Boiling

When the pressure of a system drops or the flow decreases, the bubbles cannot escape as quickly from the heat transfer surface. Likewise, if the temperature of the heat transfer surface is increased, more bubbles are created. As the temperature continues to increase, more bubbles are formed than can be efficiently carried away. The bubbles grow and group together, covering small areas of the heat transfer surface with a film of steam. This is known as *partial film boiling*. Since steam has a lower convective heat transfer coefficient than water, the steam patches on the heat transfer surface act to insulate the surface making heat transfer more difficult. As the area of the heat transfer surface covered with steam increases, the temperature of the surface increases dramatically, while the heat flux from the surface decreases. This unstable situation continues until the affected surface is covered by a stable blanket of steam, preventing contact between the heat transfer surface and the liquid in the center of the flow channel. The condition after the stable steam blanket has formed is referred to as *film boiling*.

The process of going from nucleate boiling to film boiling is graphically represented in Figure 13. The figure illustrates the effect of boiling on the relationship between the heat flux and the temperature difference between the heat transfer surface and the fluid passing it.

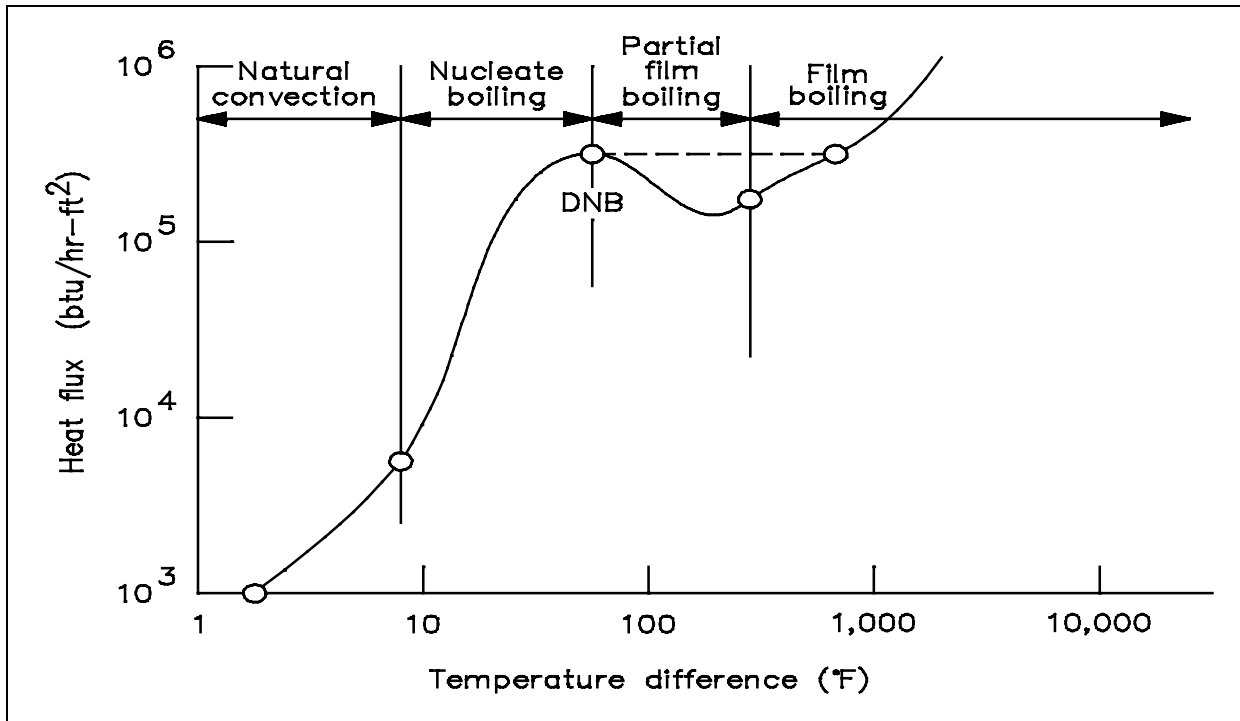


Figure 13 Boiling Heat Transfer Curve

Four regions are represented in Figure 13. The first and second regions show that as heat flux increases, the temperature difference (surface to fluid) does not change very much. Better heat transfer occurs during nucleate boiling than during natural convection. As the heat flux increases, the bubbles become numerous enough that partial film boiling (part of the surface being blanketed with bubbles) occurs. This region is characterized by an increase in temperature difference and a decrease in heat flux. The increase in temperature difference thus causes total film boiling, in which steam completely blankets the heat transfer surface.

Departure from Nucleate Boiling and Critical Heat Flux

In practice, if the heat flux is increased, the transition from nucleate boiling to film boiling occurs suddenly, and the temperature difference increases rapidly, as shown by the dashed line in the figure. The point of transition from nucleate boiling to film boiling is called the point of departure from nucleate boiling, commonly written as DNB. The heat flux associated with DNB is commonly called the critical heat flux (CHF). In many applications, CHF is an important parameter.

For example, in a reactor, if the critical heat flux is exceeded and DNB occurs at any location in the core, the temperature difference required to transfer the heat being produced from the surface of the fuel rod to the reactor coolant increases greatly. If, as could be the case, the temperature increase causes the fuel rod to exceed its design limits, a failure will occur.

The amount of heat transfer by convection can only be determined after the local heat transfer coefficient is determined. Such determination must be based on available experimental data. After experimental data has been correlated by dimensional analysis, it is a general practice to write an equation for the curve that has been drawn through the data and to compare experimental results with those obtained by analytical means. In the application of any empirical equation for forced convection to practical problems, it is important for the student to bear in mind that the predicted values of heat transfer coefficient are not exact. The values of heat transfer coefficients used by students may differ considerably from one student to another, depending on what source "book" the student has used to obtain the information. In turbulent and laminar flow, the accuracy of a heat transfer coefficient predicted from any available equation or graph may be no better than 30%.

Summary

The important information in this chapter is summarized below.

Boiling Heat Transfer Summary

- Nucleate boiling is the formation of small bubbles at a heat transfer surface. The bubbles are swept into the coolant and collapse due to the coolant being a subcooled liquid. Heat transfer is more efficient than for convection.
- Bulk boiling occurs when the bubbles do not collapse due to the coolant being at saturation conditions.
- Film boiling occurs when the heat transfer surface is blanketed with steam bubbles and the heat transfer coefficient rapidly decreases.
- Departure from nucleate boiling (DNB) occurs at the transition from nucleate to film boiling.
- Critical heat flux (CHF) is the heat flux that causes DNB to occur.

HEAT GENERATION

Heat generation and power output in a reactor are related. Reactor power is related to the mass flow rate of the coolant and the temperature difference across the reactor core.

- EO 2.1** **DESCRIBE** the power generation process in a nuclear reactor core and the factors that affect the power generation.
- EO 2.2** **DESCRIBE** the relationship between temperature, flow, and power during operation of a nuclear reactor.
- EO 2.3** **DEFINE** the following terms:
- a. Nuclear enthalpy rise hot channel factor
 - b. Average linear power density
 - c. Nuclear heat flux hot channel factor
 - d. Heat generation rate of a core
 - e. Volumetric thermal source strength
- EO 2.4** **CALCULATE** the average linear power density for an average reactor core fuel rod.
- EO 2.5** **DESCRIBE** a typical reactor core axial and radial flux profile.
- EO 2.6** **DESCRIBE** a typical reactor core fuel rod axial and radial temperature profile.
-

Heat Generation

The heat generation rate in a nuclear core is directly proportional to the fission rate of the fuel and the thermal neutron flux present. On a straight thermodynamic basis, this same heat generation is also related to the fluid temperature difference across the core and the mass flow rate of the fluid passing through the core. Thus, the size of the reactor core is dependent upon and limited by how much liquid can be passed through the core to remove the generated thermal energy. Many other factors affect the amount of heat generated within a reactor core, but its limiting generation rate is based upon how much energy can safely be carried away by the coolant.

The fission rate within a nuclear reactor is controlled by several factors. The density of the fuel, the neutron flux, and the type of fuel all affect the fission rate and, therefore, the heat generation rate. The following equation is presented here to show how the heat generation rate (\dot{Q}) is related to these factors. The terms will be discussed in more detail in the Nuclear Science modules.

$$\dot{Q} = G N \sigma_f \phi V_f \quad (2-14)$$

where:

\dot{Q} = heat generation rate (Btu/sec)

G = energy produced per fission (Btu/fission)

N = number of fissionable fuel nuclei/unit volume (atoms/cm³)

σ_f = microscopic fission cross-section of the fuel (cm²)

ϕ = neutron flux (n/cm²-sec)

V_f = volume of the fuel (cm³)

The thermal power produced by a reactor is directly related to the mass flow rate of the reactor coolant and the temperature difference across the core. The relationship between power, mass flow rate, and temperature is given in Equation 2-14.

$$\dot{Q} = \dot{m} c_p \Delta T \quad (2-15)$$

where:

\dot{Q} = heat generation rate (Btu/hr)

\dot{m} = mass flow rate (lbm/hr)

c_p = specific heat capacity of reactor coolant system (Btu/lbm-°F)

ΔT = temperature difference across core (°F)

For most types of reactors (boiling water reactor excluded), the temperature of the coolant is dependent upon reactor power and coolant flow rate. If flow rate is constant, temperature will vary directly with power. If power is constant, temperature will vary inversely with flow rate.

Flux Profiles

Once the type and amount of fuel is determined, the shape of the neutron flux distribution along the core is established. Both radial and axial flux distributions must be determined. A radial distribution looks at flux from the center of the core out to the edges. An axial distribution looks at flux from the bottom to the top of the core. As seen in Equation 2-14, the fission rate directly affects the heat generation rate within a reactor core. In the core regions of highest flux, the highest heat generation rate will be present.

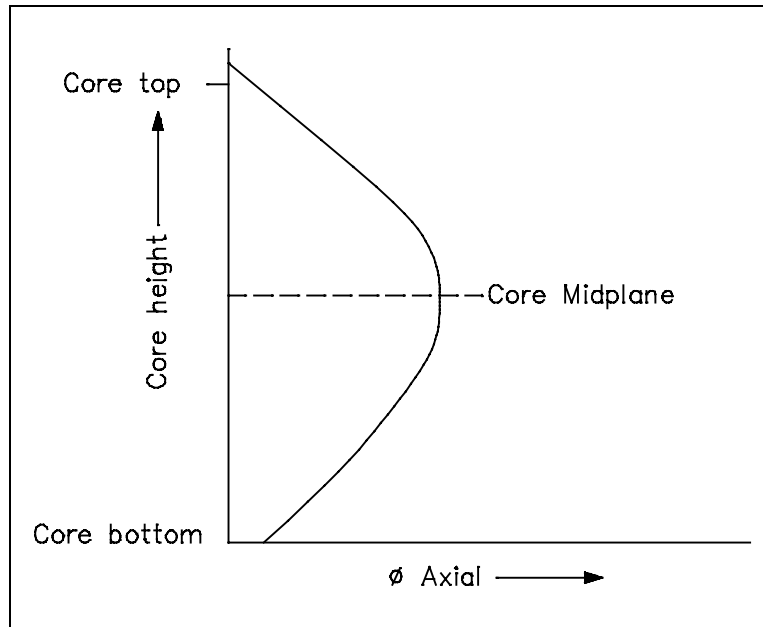


Figure 14 Axial Flux Profile

Many factors affect the axial and radial flux distributions, including the number and type of control rods, the geometry and size of core, the concentration of fission product poisons, and reflector properties. The peak power production regions within each distribution normally occurs near the center of the core, as indicated in Figures 14 and 15, but can vary during transients or as the core ages.

The above figures represent the neutron flux profiles without considering the effects of control rods. Once control rods and reflectors are taken into account, the flux profiles become much flatter although the peak still occurs near the center.

The shape of the profiles can be determined by measuring the ratio of the peak flux to the average flux in the distribution. This peaking factor is referred to as the hot channel factor. A hot channel factor of 1.0 would imply a flat flux profile.

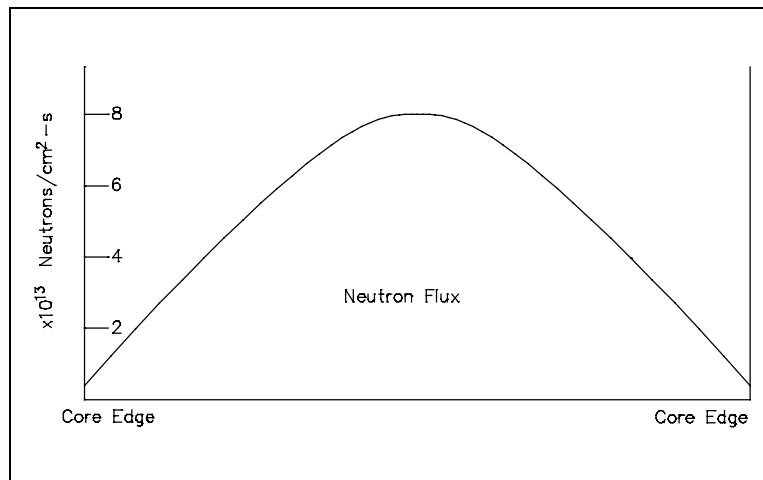


Figure 15 Radial Flux Profile

Thermal Limits

Hot channel factors are calculated values used to take into account various uncertainties in tolerances used in core manufacturing. For example, consider a coolant channel of the minimum acceptable width and length, that happens to be adjacent to a fuel plate with the maximum acceptable fuel loading. In this channel, we would now have less water than in the average channel, receiving more heat than the normal coolant channel. For any given values of core power and flow, this hypothetical channel would be closest to a thermal limit. Therefore, all design considerations are based upon the hot channel factor for each core. The *nuclear heat flux hot channel factor* (HFHCF) is the ratio of the maximum heat flux expected at any area to the average heat flux for the core. The *nuclear enthalpy rise hot channel factor* is the ratio of the total kW heat generation along the fuel rod with the highest total kW to the total kW of the average fuel rod.

Thus the limitation of the peak flux value in a core is directly related to the hot channel factor. However, in discussing flux profiles, "average" values of flux in the core are usually referred to rather than peaks.

Average Linear Power Density

In nuclear reactors, the fuel is usually distributed in individual components which sometimes resemble rods, tubes, or plates. It is possible to determine the average power produced per unit length of fuel component by dividing the total thermal output of the core by the total length of all the fuel components in the core. This quantity is called the *average linear power density*. Common units for measuring average linear power density are kW/ft.

Example:

Calculate the average linear power density for an entire core if a 3400 MW reactor is operating at full power.

Core data is: each fuel rod is 12 ft long
 264 rods/fuel assembly
 193 fuel assemblies in the core

Solution:

$$\begin{aligned} \text{Average linear power density} &= \frac{\text{total thermal power}}{\text{total fuel rod length}} \\ \text{Average linear power density} &= \frac{3.4 \times 10^6 \text{ kW}}{12 (264) (193)} \\ &= 5.56 \text{ kW/ft} \end{aligned}$$

Maximum Local Linear Power Density

The maximum local linear power density when compared to the average linear power density results in the definition of the nuclear heat flux hot channel factor. The nuclear heat flux hot channel factor can be looked at as having axial and radial components that are dependent upon the power densities and, thus, the flux in the radial and axial planes of the core. Once the hot channel factor is known, the maximum local linear power density anywhere in the core can be determined, as demonstrated in the following example.

Example:

If the nuclear heat flux hot channel factor is 1.83, calculate the maximum local linear power density in the core for the previous example (the average linear power density problem).

Solution:

$$\begin{aligned} \text{Maximum linear power density} &= \text{HFHCF} (\text{Av linear power density}) \\ &= 1.83 (5.56) \text{ kW/ft} \\ &= 10.18 \text{ kW/ft} \end{aligned}$$

Normally, nuclear facility operators are provided with the above core power and heat generation distributions, rather than having to calculate them. In addition, various monitoring systems are always employed to provide the operator with a means of monitoring core performance and the proximity of the existing operating conditions to core operational limitations.

Temperature Profiles

Additional areas of interest are the temperature profiles found within the core. A typical axial temperature profile along a coolant channel for a pressurized water reactor (PWR) is shown in Figure 16. As would be expected, the temperature of the coolant will increase throughout the entire length of the channel.

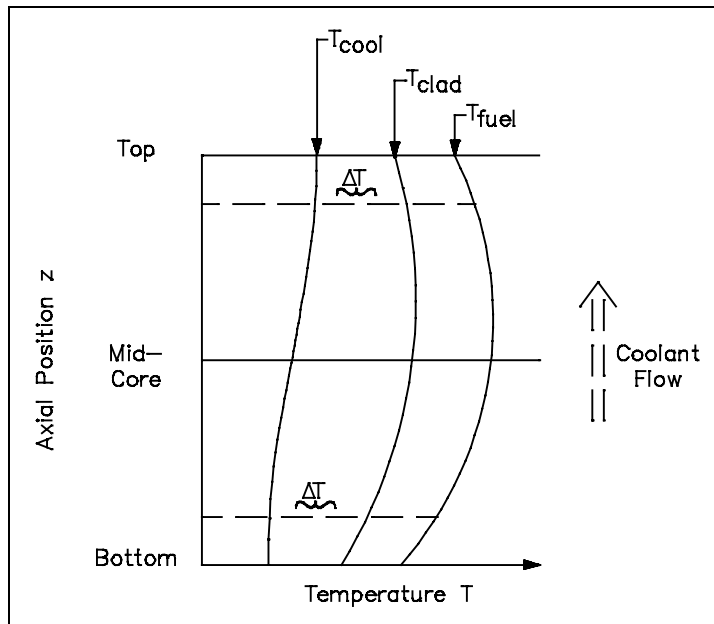


Figure 16 Axial Temperature Profile

However, the rate of increase will vary along with the linear heat flux of the channel. The power density and linear heat rate will follow the neutron flux shape. However, the temperature distributions are skewed by the changing capacity of the coolant to remove the heat energy. Since the coolant increases in temperature as it flows up the channel, the fuel cladding and, thus, the fuel temperatures are higher in the upper axial region of the core.

A radial temperature profile across a reactor core (assuming all channel coolant flows are equal) will basically follow the radial power distribution. The areas with the highest heat generation rate (power) will produce the most heat and have the highest temperatures. A radial temperature profile for an individual fuel rod and coolant channel is shown in Figure 17. The basic shape of the profile will be dependent upon the heat transfer coefficient of the various materials involved. The temperature differential across each material will have to be sufficient to transfer the heat produced. Therefore, if we know the heat transfer coefficient for each material and the heat flux, we can calculate peak fuel temperatures for a given coolant temperature.

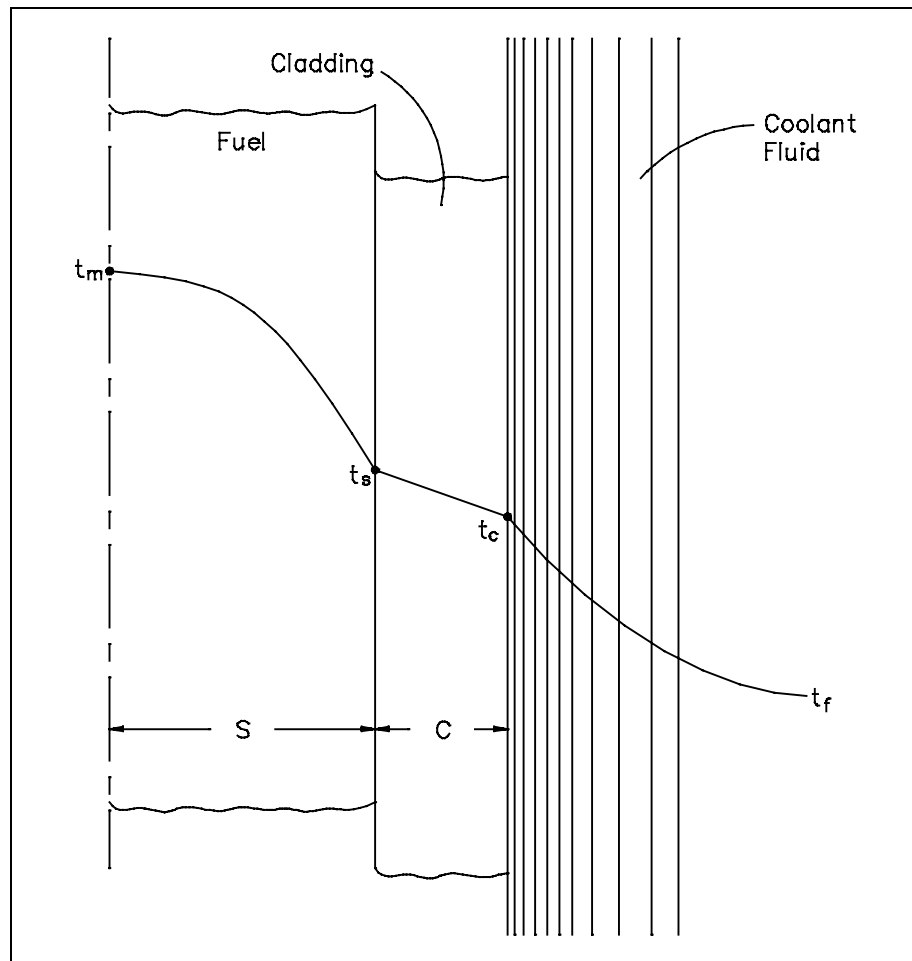


Figure 17 Radial Temperature Profile Across a Fuel Rod and Coolant Channel

Volumetric Thermal Source Strength

The total heat output of a reactor core is called the *heat generation rate*. The heat generation rate divided by the volume of fuel will give the average *volumetric thermal source strength*. The volumetric thermal source strength may be used to calculate the heat output of any section of fuel rod, provided the volume of the section is known.

$$\text{Volumetric Thermal Source Strength} = \frac{\dot{Q}_{\text{core}}}{V_{\text{fuel}}}$$

Fuel Changes During Reactor Operation

During the operation of a nuclear reactor, physical changes occur to the fuel that affect its ability to transfer heat to the coolant. The exact changes that occur are dependant on the type and form of fuel. Some reactors use fuel assemblies that consist of zircalloy tubes containing cylindrical ceramic pellets of uranium dioxide. During manufacture, a small space or gap is left between the fuel pellets and the zircalloy tube (clad). This gap is filled with pressurized helium. As the reactor is operated at power, several physical changes occur in the fuel that affect the gap between the pellets and clad. One change occurs due to high pressure in the coolant outside the clad and the relatively high temperature of the clad during reactor operation. The high temperature and high pressure causes the clad to be pushed in on the pellets by a process referred to as *creep*. Another physical change is caused by the fission process. Each fission event creates two fission product atoms from a fuel atom. Even though each fission product atom is roughly half the mass of the fuel atom, the fission products take up more volume than the original fuel atom. Fission products that are gases can collect together and form small gas bubbles within the fuel pellet. These factors cause the fuel pellets to swell, expanding them out against the clad. So the two processes of pellet swell and clad creep both work to reduce the gap between the fuel and clad.

This change in the gap between the pellet and clad has significant impact on heat transfer from the fuel and operating fuel temperatures. Initially a significant temperature difference exists across the gap to cause heat transfer to take place by convection through the helium gas. As the size of the gap is reduced, a smaller temperature difference can maintain the same heat flux. When the fuel pellets and clad come in contact, heat transfer by conduction replaces convection and the temperature difference between the fuel surface and clad decreases even more. Due to the processes of pellet swell and clad creep, the fuel temperatures of some reactors decrease slightly over time while the heat flux from the fuel and therefore the power of the reactor remain constant.

Not all changes that occur to the fuel during reactor operation work to enhance heat transfer. If the chemistry of the coolant is not carefully controlled within appropriate limits, chemical reactions can take place on the surface of the clad, resulting in the formation of a layer of corrosion products or crud between the metal of the clad and the coolant. Typically, this layer will have a lower thermal conductivity than that of the clad material, so it will act as an insulating blanket, reducing heat transfer.

If this corrosion layer is allowed to form, a larger temperature difference will be required between the coolant and fuel to maintain the same heat flux. Therefore, operation at the same power level will cause higher fuel temperatures after the buildup of corrosion products and crud.

Summary

The important information in this chapter is summarized below:

Heat Generation Summary

- The power generation process in a nuclear core is directly proportional to the fission rate of the fuel and the thermal neutron flux present.
- The thermal power produced by a reactor is directly related to the mass flow rate of the reactor coolant and the temperature difference across the core.
- The nuclear enthalpy rise hot channel factor is the ratio of the total kW heat generation along a fuel rod with the highest total kW, to the total kW of the average fuel rod.
- The average linear power density in the core is the total thermal power divided by the active length of the fuel rods.
- The nuclear heat flux hot channel factor is the ratio of the maximum heat flux expected at any area to the average heat flux for the core.
- The total heat output of a reactor core is called the heat generation rate.
- The heat generation rate divided by the volume of fuel will give the average volumetric thermal source strength.

DECAY HEAT

Decay heat production is a particular problem associated with nuclear reactors. Even though the reactor is shut down, heat is produced from the decay of fission fragments. Limits for each particular reactor are established to prevent damage to fuel assemblies due to decay heat.

- EO 2.7** **DEFINE** the term decay heat.
- EO 2.8** **Given the operating conditions of a reactor core and the necessary formulas, CALCULATE** the core decay heat generation.
- EO 2.9** **DESCRIBE** two categories of methods for removing decay heat from a reactor core.

Reactor Decay Heat Production

A problem peculiar to power generation by nuclear reactors is that of decay heat. In fossil fuel facilities, once the combustion process is halted, there is no further heat generation, and only a relatively small amount of thermal energy is stored in the high temperature of plant components. In a nuclear facility, the fission of heavy atoms such as isotopes of uranium and plutonium results in the formation of highly radioactive fission products. These fission products radioactively decay at a rate determined by the amount and type of radioactive nuclides present. Some radioactive atoms will decay while the reactor is operating and the energy released by their decay will be removed from the core along with the heat produced by the fission process. All radioactive materials that remain in the reactor at the time it is shut down and the fission process halted will continue to decay and release energy. This release of energy by the decay of fission products is called *decay heat*.

The amount of radioactive materials present in the reactor at the time of shutdown is dependent on the power levels at which the reactor operated and the amount of time spent at those power levels. The amount of decay heat is very significant. Typically, the amount of decay heat that will be present in the reactor immediately following shutdown will be roughly 7% of the power level that the reactor operated at prior to shutdown. A reactor operating at 1000 MW will produce 70 MW of decay heat immediately after a shutdown. The amount of decay heat produced in the reactor will decrease as more and more of the radioactive material decays to some stable form. Decay heat may decrease to about 2% of the pre-shutdown power level within the first hour after shutdown and to 1% within the first day. Decay heat will continue to decrease after the first day, but it will decrease at a much slower rate. Decay heat will be significant weeks and even months after the reactor is shutdown.

The design of the reactor must allow for the removal of this decay heat from the core by some means. If adequate heat removal is not available, decay heat will increase the temperatures in the core to the point that fuel melting and core damage will occur. Fuel that has been removed from the reactor will also require some method of removing decay heat if the fuel has been exposed to a significant neutron flux. Each reactor facility will have its own method of removing decay heat from both the reactor core and also any irradiated fuel removed from the core.

Calculation of Decay Heat

The amount of decay heat being generated in a fuel assembly at any time after shutdown can be calculated in two ways. The first way is to calculate the amount of fission products present at the time of shutdown. This is a fairly detailed process and is dependent upon power history. For a given type of fuel, the concentrations, decay energies, and half lives of fission products are known. By starting from a known value, based on power history at shutdown, the decay heat generation rate can be calculated for any time after shutdown.

An exact solution must take into account the fact that there are hundreds of different radionuclides present in the core, each with its own concentration and decay half-life. It is possible to make a rough approximation by using a single half-life that represents the overall decay of the core over a certain period of time. An equation that uses this approximation is Equation 2-16.

$$\dot{Q} = \dot{Q}_o \left(\frac{1}{2} \right)^{\frac{\text{time}}{\text{half-life}}} \quad (2-16)$$

where:

\dot{Q} = decay heat generation rate at some time after shutdown

\dot{Q}_o = initial decay heat immediately after shutdown

time = amount of time since shutdown

half-life = overall decay half-life of the core

Example:

A 250 MW reactor has an unexpected shutdown. From data supplied by the vendor, we know that decay heat at time of shutdown will be 7% of the effective power at time of shutdown and will decrease with a 1 hr half life. Effective power at time of shutdown was calculated to be 120 MW. How much heat removal capability (in units of Btu/hr) will be required 12 hours after shutdown?

Solution:

- (a) First determine the decay heat immediately following shutdown.

$$(120 \text{ MW})(.07) = 8.4 \text{ MW decay heat at shutdown}$$

- (b) Then use Equation 2-15 to determine the decay heat 12 hours later.

$$\begin{aligned} \dot{Q} &= \dot{Q}_o \left(\frac{1}{2} \right)^{\frac{\text{time}}{\text{half-life}}} \\ &= 8.4 \text{ MW} \left(\frac{1}{2} \right)^{\frac{12 \text{ hr}}{1 \text{ hr}}} \\ &= 2.05 \times 10^{-3} \text{ MW} \left(\frac{3.413 \times 10^6 \text{ Btu/hr}}{1 \text{ MW}} \right) \\ &= 7000 \frac{\text{Btu}}{\text{hr}} \end{aligned}$$

The second method is much simpler to use, but is not useful for forecasting heat loads in the future. To calculate the decay heat load at a given point after shutdown, secure any heat removal components from the primary system or spent fuel pool and plot the heatup rate. If the mass of the coolant and the specific heat of the coolant are known, the heat generation rate can be accurately calculated.

$$\dot{Q} = m c_p \frac{\Delta T}{\Delta t} \quad (2-17)$$

where:

$$\dot{Q} = \text{decay heat (Btu/hr)}$$

$$m = \text{mass of coolant (lbm)}$$

$$c_p = \text{specific heat capacity of coolant (Btu/lbm-}^\circ\text{F)}$$

$$\Delta T = \text{temperature change of coolant (}^\circ\text{F)}$$

$$\Delta t = \text{time over which heatup takes place (hr)}$$

Example:

Three days after a planned reactor shutdown, it is desired to perform maintenance on one of two primary heat exchangers. Each heat exchanger is rated at 12,000 Btu/hr. To check the current heat load on the primary system due to decay heat, cooling is secured to both heat exchangers. The primary system heats up at a rate of 0.8°F/hr. The primary system contains 24,000 lbm of coolant with a specific heat capacity of 0.8 Btu/lbm-°F. Will one heat exchanger be sufficient to remove the decay heat?

Solution:

$$\begin{aligned}\dot{Q} &= m c_p \frac{\Delta T}{\Delta t} \\ &= (24,000 \text{ lbm}) \left(0.8 \frac{\text{Btu}}{\text{lbm-}^\circ\text{F}} \right) \left(\frac{0.8^\circ\text{F}}{1 \text{ hr}} \right) \\ &= 15,360 \frac{\text{Btu}}{\text{hr}}\end{aligned}$$

One heat exchanger removes 12,000 Btu/hr.

One heat exchanger will not be sufficient.

Decay Heat Limits

Reactor decay heat can be a major concern. In the worst case scenarios, it can cause melting of and/or damage to the reactor core, as in the case of Three Mile Island. The degree of concern with decay heat will vary according to reactor type and design. There is little concern about core temperature due to decay heat for low power, pool-type reactors.

Each reactor will have some limits during shutdown that are based upon decay heat considerations. These limits may vary because of steam generator pressure, core temperature, or any other parameter that may be related to decay heat generation. Even during refueling processes, heat removal from expended fuel rods is a controlling factor. For each limit developed, there is usually some safety device or protective feature established.

Decay Heat Removal

Methods for removing decay heat from a reactor core can be grouped into two general categories. One category includes methods which circulate fluid through the reactor core in a closed loop, using some type of heat exchanger to transfer heat out of the system. The other category includes methods which operate in an open system, drawing in cool fluid from some source and discharging warmer fluid to some storage area or the environment.

In most reactors, decay heat is normally removed by the same methods used to remove heat generated by fission during reactor operation. Additionally, many reactors are designed such that natural circulation between the core and either its normal heat exchanger or an emergency heat exchanger can remove decay heat. These are examples of the first category of methods for decay heat removal.

If a reactor design is such that decay heat removal is required for core safety, but accidents are possible that will make the closed loop heat transfer methods described above unavailable, then an emergency cooling system of some sort will be included in the reactor design. Generally, emergency cooling systems consist of some reliable source of water that is injected into the core at a relatively low temperature. This water will be heated by the decay heat of the core and exit the reactor via some path where it will either be stored in some structure or released to the environment. Use of this type of system is almost always less desirable than the use of the closed loop systems described above.

Students should research systems, limits, and protective features applicable to their own specific facilities.

Summary

The important information in this chapter is summarized below.

Decay Heat Summary

- Decay heat is the amount of heat generated by decay of fission products after shutdown of the facility.
- The amount of decay heat is dependent on the reactor's power history.
- Methods for removing decay heat usually fall into one of the following categories.
 - Closed loop systems, where coolant is circulated between the reactor and a heat exchanger in a closed loop. The heat exchanger transfers the decay heat to the fluid in the secondary side of the heat exchanger.
 - Once through systems, where coolant from a source is injected into the reactor core. The decay heat is transferred from the fuel assemblies into the coolant, then the coolant leaves the reactor and is either collected in a storage structure or released to the environment.
- The limits for decay heat are calculated to prevent damage to the reactor core.

end of text.

CONCLUDING MATERIAL

Review activities:

DOE - ANL-W, BNL, EG&G Idaho,
EG&G Mound, EG&G Rocky Flats,
LLNL, LANL, MMES, ORAU, REEC_o,
WHC, WINCO, WEMCO, and WSRC.

Preparing activity:

DOE - NE-73
Project Number 6910-0018/2

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**Department of Energy
Fundamentals Handbook**

**THERMODYNAMICS, HEAT TRANSFER,
AND FLUID FLOW,
Module 3
Fluid Flow**

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TERMINAL OBJECTIVE

- 1.0 Given conditions affecting the fluid flow in a system, **EVALUATE** the effects on the operation of the system.

ENABLING OBJECTIVES

- 1.1 **DESCRIBE** how the density of a fluid varies with temperature.
- 1.2 **DEFINE** the term buoyancy.
- 1.3 **DESCRIBE** the relationship between the pressure in a fluid column and the density and depth of the fluid.
- 1.4 **STATE** Pascal's Law.
- 1.5 **DEFINE** the terms mass flow rate and volumetric flow rate.
- 1.6 **CALCULATE** either the mass flow rate or the volumetric flow rate for a fluid system.
- 1.7 **STATE** the principle of conservation of mass.
- 1.8 **CALCULATE** the fluid velocity or flow rate in a specified fluid system using the continuity equation.
- 1.9 **DESCRIBE** the characteristics and flow velocity profiles of laminar flow and turbulent flow.
- 1.10 **DEFINE** the property of viscosity.
- 1.11 **DESCRIBE** how the viscosity of a fluid varies with temperature.
- 1.12 **DESCRIBE** the characteristics of an ideal fluid.
- 1.13 **DESCRIBE** the relationship between the Reynolds number and the degree of turbulence of the flow.
- 1.14 **DESCRIBE** the relationship between Bernoulli's equation and the First Law of Thermodynamics.

ENABLING OBJECTIVES (Cont.)

- 1.15 **DEFINE** the term head with respect to its use in fluid flow.
- 1.16 **EXPLAIN** the energy conversions that take place in a fluid system between the velocity, elevation, and pressure heads as flow continues through a piping system.
- 1.17 Given the initial and final conditions of the system, **CALCULATE** the unknown fluid properties using the simplified Bernoulli equation.
- 1.18 **DESCRIBE** the restrictions applied to Bernoulli's equation when presented in its simplest form.
- 1.19 **EXPLAIN** how to extend the Bernoulli equation to more general applications.
- 1.20 **RELATE** Bernoulli's principle to the operation of a venturi.
- 1.21 **DEFINE** the terms head loss, frictional loss, and minor losses.
- 1.22 **DETERMINE** friction factors for various flow situations using the Moody chart.
- 1.23 **CALCULATE** the head loss in a fluid system due to frictional losses using Darcy's equation.
- 1.24 **CALCULATE** the equivalent length of pipe that would cause the same head loss as the minor losses that occur in individual components.
- 1.25 **DEFINE** natural circulation and forced circulation.
- 1.26 **DEFINE** thermal driving head.
- 1.27 **DESCRIBE** the conditions necessary for natural circulation to exist.
- 1.28 **EXPLAIN** the relationship between flow rate and temperature difference in natural circulation flow.
- 1.29 **DESCRIBE** how the operator can determine whether natural circulation exists in the reactor coolant system and other heat removal systems.
- 1.30 **DESCRIBE** how to enhance natural circulation flow.
- 1.31 **DEFINE** two-phase flow.

ENABLING OBJECTIVES (Cont.)

- 1.32 **DESCRIBE** two-phase flow including such phenomena as bubbly, slug, and annular flow.
- 1.33 **DESCRIBE** the problems associated with core flow oscillations and flow instability.
- 1.34 **DESCRIBE** the conditions that could lead to core flow oscillation and instability.
- 1.35 **DESCRIBE** the phenomenon of pipe whip.
- 1.36 **DESCRIBE** the phenomenon of water hammer.
- 1.37 **DEFINE** the terms net positive suction head and cavitation.
- 1.38 **CALCULATE** the new volumetric flow rate, head, or power for a variable speed centrifugal pump using the pump laws.
- 1.39 **DESCRIBE** the effect on system flow and pump head for the following changes:
 - a. Changing pump speeds
 - b. Adding pumps in parallel
 - c. Adding pumps in series

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CONTINUITY EQUATION

Understanding the quantities measured by the volumetric flow rate and mass flow rate is crucial to understanding other fluid flow topics. The continuity equation expresses the relationship between mass flow rates at different points in a fluid system under steady-state flow conditions.

- EO 1.1** **DESCRIBE** how the density of a fluid varies with temperature.
- EO 1.2** **DEFINE** the term buoyancy.
- EO 1.3** **DESCRIBE** the relationship between the pressure in a fluid column and the density and depth of the fluid.
- EO 1.4** **STATE** Pascal's Law.
- EO 1.5** **DEFINE** the terms mass flow rate and volumetric flow rate.
- EO 1.6** **CALCULATE** either the mass flow rate or the volumetric flow rate for a fluid system.
- EO 1.7** **STATE** the principle of conservation of mass.
- EO 1.8** **CALCULATE** the fluid velocity or flow rate in a specified fluid system using the continuity equation.
-

Introduction

Fluid flow is an important part of most industrial processes; especially those involving the transfer of heat. Frequently, when it is desired to remove heat from the point at which it is generated, some type of fluid is involved in the heat transfer process. Examples of this are the cooling water circulated through a gasoline or diesel engine, the air flow past the windings of a motor, and the flow of water through the core of a nuclear reactor. Fluid flow systems are also commonly used to provide lubrication.

Fluid flow in the nuclear field can be complex and is not always subject to rigorous mathematical analysis. Unlike solids, the particles of fluids move through piping and components at different velocities and are often subjected to different accelerations.

Even though a detailed analysis of fluid flow can be extremely difficult, the basic concepts involved in fluid flow problems are fairly straightforward. These basic concepts can be applied in solving fluid flow problems through the use of simplifying assumptions and average values, where appropriate. Even though this type of analysis would not be sufficient in the engineering design of systems, it is very useful in understanding the operation of systems and predicting the approximate response of fluid systems to changes in operating parameters.

The basic principles of fluid flow include three concepts or principles; the first two of which the student has been exposed to in previous manuals. The first is the principle of momentum (leading to equations of fluid forces) which was covered in the manual on Classical Physics. The second is the conservation of energy (leading to the First Law of Thermodynamics) which was studied in thermodynamics. The third is the conservation of mass (leading to the continuity equation) which will be explained in this module.

Properties of Fluids

A *fluid* is any substance which flows because its particles are not rigidly attached to one another. This includes liquids, gases and even some materials which are normally considered solids, such as glass. Essentially, fluids are materials which have no repeating crystalline structure.

Several properties of fluids were discussed in the Thermodynamics section of this text. These included temperature, pressure, mass, specific volume and density. *Temperature* was defined as the relative measure of how hot or cold a material is. It can be used to predict the direction that heat will be transferred. *Pressure* was defined as the force per unit area. Common units for pressure are pounds force per square inch (psi). *Mass* was defined as the quantity of matter contained in a body and is to be distinguished from weight, which is measured by the pull of gravity on a body. The *specific volume* of a substance is the volume per unit mass of the substance. Typical units are ft³/lbm. *Density*, on the other hand, is the mass of a substance per unit volume. Typical units are lbm/ft³. Density and specific volume are the inverse of one another. Both density and specific volume are dependant on the temperature and somewhat on the pressure of the fluid. As the temperature of the fluid increases, the density decreases and the specific volume increases. Since liquids are considered incompressible, an increase in pressure will result in no change in density or specific volume of the liquid. In actuality, liquids can be slightly compressed at high pressures, resulting in a slight increase in density and a slight decrease in specific volume of the liquid.

Buoyancy

Buoyancy is defined as the tendency of a body to float or rise when submerged in a fluid. We all have had numerous opportunities of observing the buoyant effects of a liquid. When we go swimming, our bodies are held up almost entirely by the water. Wood, ice, and cork float on water. When we lift a rock from a stream bed, it suddenly seems heavier on emerging from the water. Boats rely on this buoyant force to stay afloat. The amount of this buoyant effect was first computed and stated by the Greek philosopher Archimedes. When a body is placed in a fluid, it is buoyed up by a force equal to the weight of the water that it displaces.

If a body weighs more than the liquid it displaces, it sinks but will appear to lose an amount of weight equal to that of the displaced liquid, as our rock. If the body weighs less than that of the displaced liquid, the body will rise to the surface eventually floating at such a depth that will displace a volume of liquid whose weight will just equal its own weight. A floating body displaces its own weight of the fluid in which it floats.

Compressibility

Compressibility is the measure of the change in volume a substance undergoes when a pressure is exerted on the substance. Liquids are generally considered to be incompressible. For instance, a pressure of 16,400 psig will cause a given volume of water to decrease by only 5% from its volume at atmospheric pressure. Gases on the other hand, are very compressible. The volume of a gas can be readily changed by exerting an external pressure on the gas

Relationship Between Depth and Pressure

Anyone who dives under the surface of the water notices that the pressure on his eardrums at a depth of even a few feet is noticeably greater than atmospheric pressure. Careful measurements show that the pressure of a liquid is directly proportional to the depth, and for a given depth the liquid exerts the same pressure in all directions.

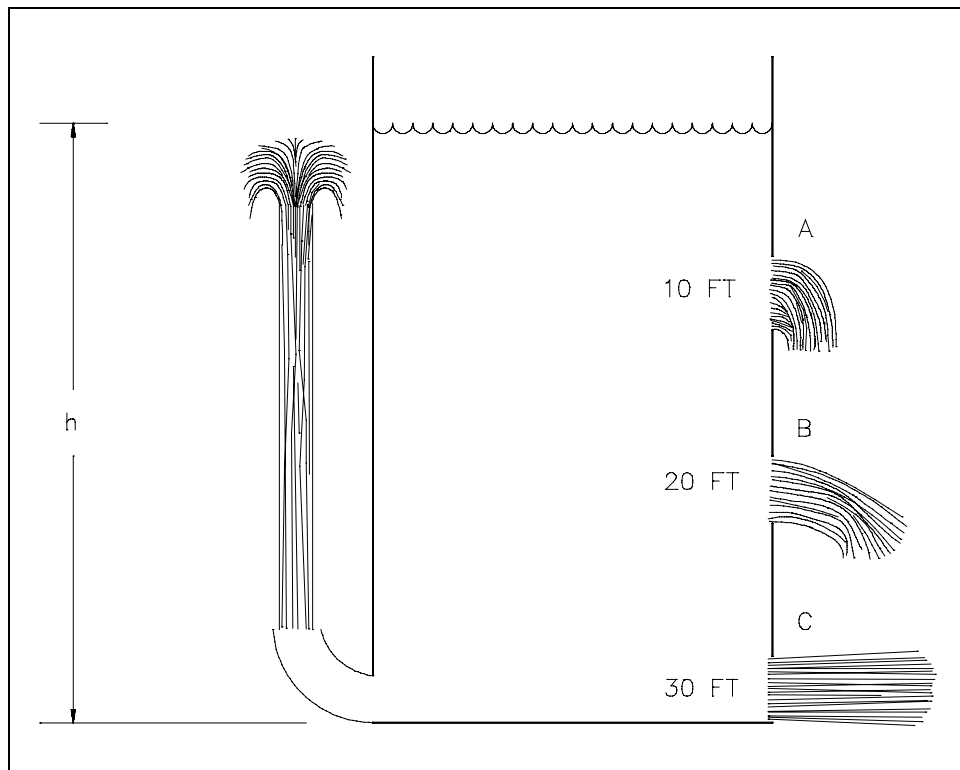


Figure 1 Pressure Versus Depth

As shown in Figure 1 the pressure at different levels in the tank varies and this causes the fluid to leave the tank at varying velocities. Pressure was defined to be force per unit area. In the case of this tank, the force is due to the weight of the water above the point where the pressure is being determined.

Example:

$$\begin{aligned} \text{Pressure} &= \frac{\text{Force}}{\text{Area}} \\ &= \frac{\text{Weight}}{\text{Area}} \\ P &= \frac{m g}{A g_c} \\ &= \frac{\rho V g}{A g_c} \end{aligned}$$

where:

m = mass in lbm

g = acceleration due to earth's gravity $32.17 \frac{\text{ft}}{\text{sec}^2}$

g_c = $32.17 \frac{\text{lbm-ft}}{\text{lbf-sec}^2}$

A = area in ft^2

V = volume in ft^3

ρ = density of fluid in $\frac{\text{lbm}}{\text{ft}^3}$

The volume is equal to the cross-sectional area times the height (h) of liquid. Substituting this in to the above equation yields:

$$P = \frac{\rho A h g}{A g_c}$$

$$P = \frac{\rho h g}{g_c}$$

This equation tells us that the pressure exerted by a column of water is directly proportional to the height of the column and the density of the water and is independent of the cross-sectional area of the column. The pressure thirty feet below the surface of a one inch diameter standpipe is the same as the pressure thirty feet below the surface of a large lake.

Example 1:

If the tank in Figure 1 is filled with water that has a density of 62.4 lbm/ft³, calculate the pressures at depths of 10, 20, and 30 feet.

Solution:

$$\begin{aligned}
 P &= \frac{\rho h g}{g_c} \\
 P_{10 \text{ feet}} &= \left(62.4 \frac{\text{lbm}}{\text{ft}^3}\right) (10 \text{ ft}) \left(\frac{32.17 \frac{\text{ft}}{\text{sec}^2}}{32.17 \frac{\text{lbm-ft}}{\text{lbf-sec}^2}}\right) \\
 &= 624 \frac{\text{lbf}}{\text{ft}^2} \left(\frac{1 \text{ ft}^2}{144 \text{ in}^2}\right) \\
 &= 4.33 \frac{\text{lbf}}{\text{in}^2} \\
 P_{20 \text{ feet}} &= \left(62.4 \frac{\text{lbm}}{\text{ft}^3}\right) (20 \text{ ft}) \left(\frac{32.17 \frac{\text{ft}}{\text{sec}^2}}{32.17 \frac{\text{lbm-ft}}{\text{lbf-sec}^2}}\right) \\
 &= 1248 \frac{\text{lbf}}{\text{ft}^2} \left(\frac{1 \text{ ft}^2}{144 \text{ in}^2}\right) \\
 &= 8.67 \frac{\text{lbf}}{\text{in}^2}
 \end{aligned}$$

$$\begin{aligned}
 P_{30 \text{ feet}} &= \left(62.4 \frac{\text{lbm}}{\text{ft}^3}\right) (30 \text{ ft}) \left(\frac{32.17 \frac{\text{ft}}{\text{sec}^2}}{32.17 \frac{\text{lbm-ft}}{\text{lbf-sec}^2}} \right) \\
 &= 1872 \frac{\text{lbf}}{\text{ft}^2} \left(\frac{1 \text{ ft}^2}{144 \text{ in}^2} \right) \\
 &= 13.00 \frac{\text{lbf}}{\text{in}^2}
 \end{aligned}$$

Example 2:

A cylindrical water tank 40 ft high and 20 ft in diameter is filled with water that has a density of 61.9 lbm/ft³.

- (a) What is the water pressure on the bottom of the tank?
- (b) What is the average force on the bottom?

Solution:

$$\begin{aligned}
 \text{(a)} \quad P &= \frac{\rho h g}{g_c} \\
 P &= \left(61.9 \frac{\text{lbm}}{\text{ft}^3}\right) (40 \text{ ft}) \left(\frac{32.17 \frac{\text{ft}}{\text{sec}^2}}{32.17 \frac{\text{lbm-ft}}{\text{lbf-sec}^2}} \right) \\
 &= 2476 \frac{\text{lbf}}{\text{ft}^2} \left(\frac{1 \text{ ft}^2}{144 \text{ in}^2} \right) \\
 &= 17.2 \frac{\text{lbf}}{\text{in}^2}
 \end{aligned}$$

$$(b) \quad \text{Pressure} = \frac{\text{Force}}{\text{Area}}$$

$$\text{Force} = (\text{Pressure}) (\text{Area})$$

$$\text{Area} = \pi r^2$$

$$\begin{aligned} F &= \left(17.2 \frac{\text{lbf}}{\text{in}^2} \right) \pi (10 \text{ ft})^2 \left(\frac{144 \text{ in}^2}{1 \text{ ft}^2} \right) \\ &= 7.78 \times 10^5 \text{ lbf} \end{aligned}$$

Pascal's Law

The pressure of the liquids in each of the previously cited cases has been due to the weight of the liquid. Liquid pressures may also result from application of external forces on the liquid. Consider the following examples. Figure 2 represents a container completely filled with liquid. A, B, C, D, and E represent pistons of equal cross-sectional areas fitted into the walls of the vessel. There will be forces acting on the pistons C, D, and E due to the pressures caused by the different depths of the liquid. Assume that the forces on the pistons due to the pressure caused by the weight of the liquid are as follows: A = 0 lbf, B = 0 lbf, C = 10 lbf, D = 30 lbf, and E = 25 lbf. Now let an external force of 50 lbf be applied to piston A. This external force will cause the pressure at all points in the container to increase by the same amount. Since the pistons all have the same cross-sectional area, the increase in pressure will result in the forces on the pistons all increasing by 50 lbf. So if an external force of 50 lbf is applied to piston A, the force exerted by the fluid on the other pistons will now be as follows: B = 50 lbf, C = 60 lbf, D = 80 lbf, and E = 75 lbf.

This effect of an external force on a confined fluid was first stated by Pascal in 1653.

Pressure applied to a confined fluid is transmitted undiminished throughout the confining vessel of the system.

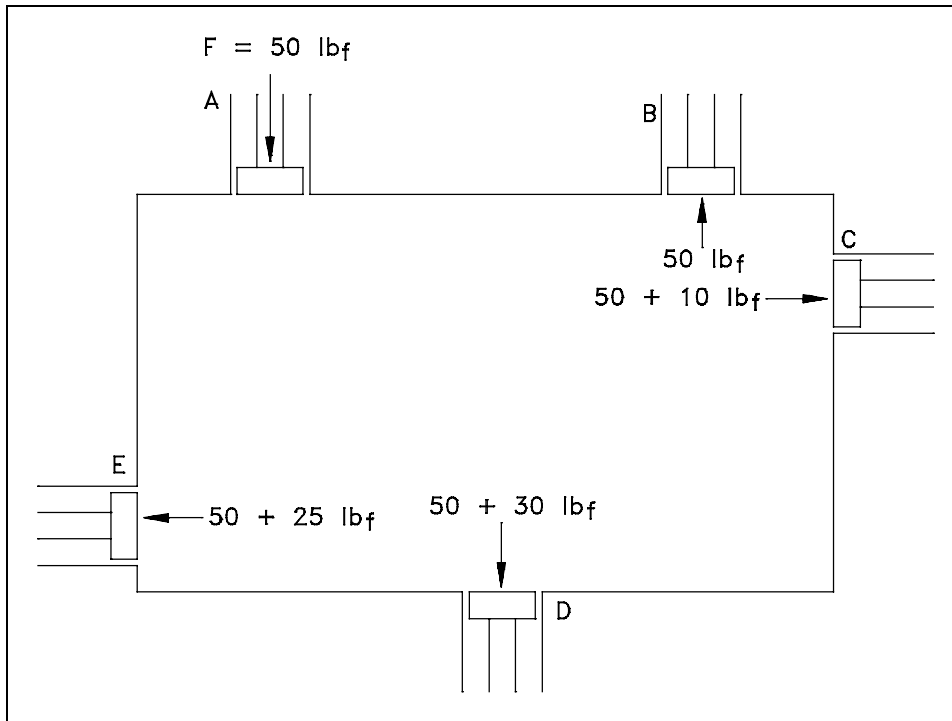


Figure 2 Pascal's Law

Control Volume

In thermodynamics, a *control volume* was defined as a fixed region in space where one studies the masses and energies crossing the boundaries of the region. This concept of a control volume is also very useful in analyzing fluid flow problems. The boundary of a control volume for fluid flow is usually taken as the physical boundary of the part through which the flow is occurring. The control volume concept is used in fluid dynamics applications, utilizing the continuity, momentum, and energy principles mentioned at the beginning of this chapter. Once the control volume and its boundary are established, the various forms of energy crossing the boundary with the fluid can be dealt with in equation form to solve the fluid problem. Since fluid flow problems usually treat a fluid crossing the boundaries of a control volume, the control volume approach is referred to as an "open" system analysis, which is similar to the concepts studied in thermodynamics. There are special cases in the nuclear field where fluid does not cross the control boundary. Such cases are studied utilizing the "closed" system approach.

Regardless of the nature of the flow, all flow situations are found to be subject to the established basic laws of nature that engineers have expressed in equation form. Conservation of mass and conservation of energy are always satisfied in fluid problems, along with Newton's laws of motion. In addition, each problem will have physical constraints, referred to mathematically as boundary conditions, that must be satisfied before a solution to the problem will be consistent with the physical results.

Volumetric Flow Rate

The *volumetric flow rate* (\dot{V}) of a system is a measure of the volume of fluid passing a point in the system per unit time. The volumetric flow rate can be calculated as the product of the cross-sectional area (A) for flow and the average flow velocity (v).

$$\dot{V} = A v \quad (3-1)$$

If area is measured in square feet and velocity in feet per second, Equation 3-1 results in volumetric flow rate measured in cubic feet per second. Other common units for volumetric flow rate include gallons per minute, cubic centimeters per second, liters per minute, and gallons per hour.

Example:

A pipe with an inner diameter of 4 inches contains water that flows at an average velocity of 14 feet per second. Calculate the volumetric flow rate of water in the pipe.

Solution:

Use Equation 3-1 and substitute for the area.

$$\dot{V} = (\pi r^2) v$$

$$\dot{V} = (3.14) \left(\frac{2}{12} \text{ ft}\right)^2 \left(14 \frac{\text{ft}}{\text{sec}}\right)$$

$$\dot{V} = 1.22 \frac{\text{ft}^3}{\text{sec}}$$

Mass Flow Rate

The *mass flow rate* (\dot{m}) of a system is a measure of the mass of fluid passing a point in the system per unit time. The mass flow rate is related to the volumetric flow rate as shown in Equation 3-2 where ρ is the density of the fluid.

$$\dot{m} = \rho \dot{V} \quad (3-2)$$

If the volumetric flow rate is in cubic feet per second and the density is in pounds-mass per cubic foot, Equation 3-2 results in mass flow rate measured in pounds-mass per second. Other common units for measurement of mass flow rate include kilograms per second and pounds-mass per hour.

Replacing \dot{V} in Equation 3-2 with the appropriate terms from Equation 3-1 allows the direct calculation of the mass flow rate.

$$\dot{m} = \rho A v \quad (3-3)$$

Example:

The water in the pipe of the previous example had a density of 62.44 lbm/ft³. Calculate the mass flow rate.

Solution:

$$\dot{m} = \rho \dot{V}$$

$$\dot{m} = (62.44 \frac{\text{lbm}}{\text{ft}^3}) (1.22 \frac{\text{ft}^3}{\text{sec}})$$

$$\dot{m} = 76.2 \frac{\text{lbm}}{\text{sec}}$$

Conservation of Mass

In thermodynamics, you learned that energy can neither be created nor destroyed, only changed in form. The same is true for mass. Conservation of mass is a principle of engineering that states that all mass flow rates into a control volume are equal to all mass flow rates out of the control volume plus the rate of change of mass within the control volume. This principle is expressed mathematically by Equation 3-4.

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}} + \frac{\Delta m}{\Delta t} \quad (3-4)$$

where:

$$\frac{\Delta m}{\Delta t} = \text{the increase or decrease of the mass within the control volume over a (specified time period)}$$

Steady-State Flow

Steady-state flow refers to the condition where the fluid properties at any single point in the system do not change over time. These fluid properties include temperature, pressure, and velocity. One of the most significant properties that is constant in a steady-state flow system is the system mass flow rate. This means that there is no accumulation of mass within any component in the system.

Continuity Equation

The continuity equation is simply a mathematical expression of the principle of conservation of mass. For a control volume that has a single inlet and a single outlet, the principle of conservation of mass states that, for steady-state flow, the mass flow rate into the volume must equal the mass flow rate out. The continuity equation for this situation is expressed by Equation 3-5.

$$\dot{m}_{\text{inlet}} = \dot{m}_{\text{outlet}} \quad (3-5)$$

$$(\rho Av)_{\text{inlet}} = (\rho Av)_{\text{outlet}}$$

For a control volume with multiple inlets and outlets, the principle of conservation of mass requires that the sum of the mass flow rates into the control volume equal the sum of the mass flow rates out of the control volume. The continuity equation for this more general situation is expressed by Equation 3-6.

$$\sum \dot{m}_{\text{inlets}} = \sum \dot{m}_{\text{outlets}} \quad (3-6)$$

One of the simplest applications of the continuity equation is determining the change in fluid velocity due to an expansion or contraction in the diameter of a pipe.

Example: Continuity Equation - Piping Expansion

Steady-state flow exists in a pipe that undergoes a gradual expansion from a diameter of 6 in. to a diameter of 8 in. The density of the fluid in the pipe is constant at 60.8 lbm/ft³. If the flow velocity is 22.4 ft/sec in the 6 in. section, what is the flow velocity in the 8 in. section?

Solution:

From the continuity equation we know that the mass flow rate in the 6 in. section must equal the mass flow rate in the 8 in. section. Letting the subscript 1 represent the 6 in. section and 2 represent the 8 in. section we have the following.

$$\dot{m}_1 = \dot{m}_2$$

$$\rho_1 A_1 v_1 = \rho_2 A_2 v_2$$

$$v_2 = v_1 \frac{\rho_1}{\rho_2} \frac{A_1}{A_2}$$

$$= v_1 \frac{\pi r_1^2}{\pi r_2^2}$$

$$= \left(22.4 \frac{\text{ft}}{\text{sec}} \right) \frac{(3 \text{ in})^2}{(4 \text{ in})^2}$$

$$v_2 = 12.6 \frac{\text{ft}}{\text{sec}}$$

So by using the continuity equation, we find that the increase in pipe diameter from 6 to 8 inches caused a decrease in flow velocity from 22.4 to 12.6 ft/sec.

The continuity equation can also be used to show that a decrease in pipe diameter will cause an increase in flow velocity.

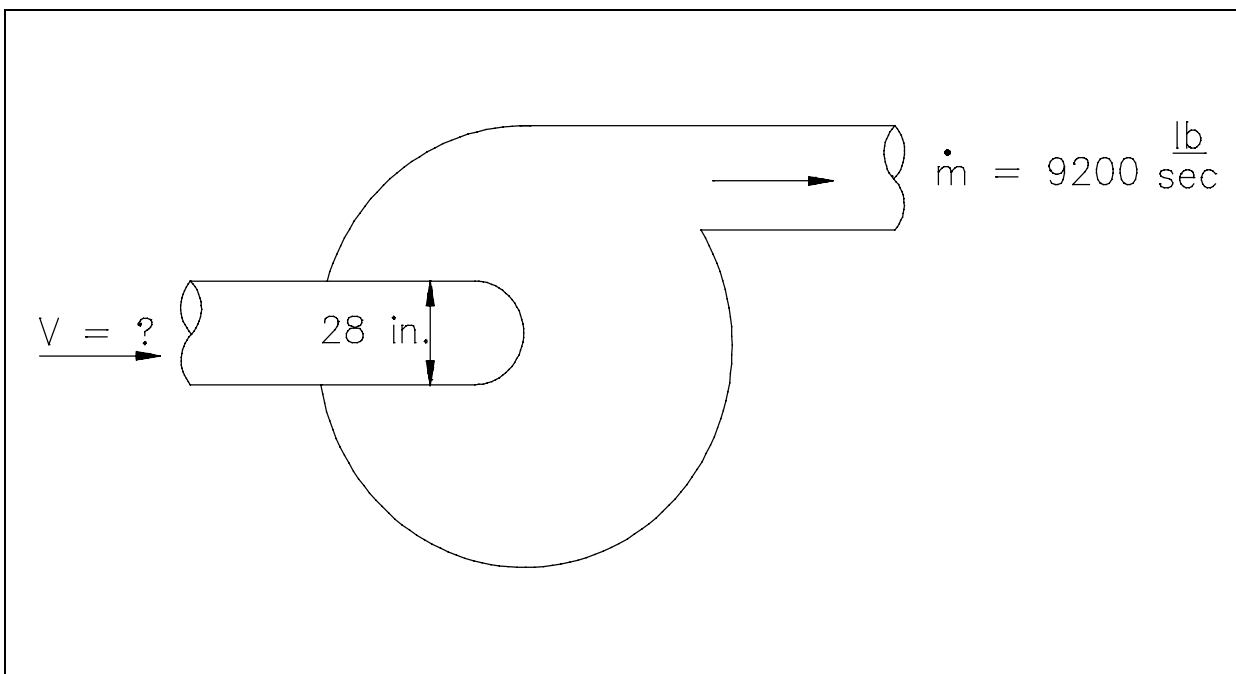


Figure 3 Continuity Equation

Example: Continuity Equation - Centrifugal Pump

The inlet diameter of the reactor coolant pump shown in Figure 3 is 28 in. while the outlet flow through the pump is 9200 lbm/sec. The density of the water is 49 lbm/ft³. What is the velocity at the pump inlet?

Solution:

$$A_{\text{inlet}} = \pi r^2 = (3.14) \left(14 \text{ in} \left(\frac{1 \text{ ft}}{12 \text{ in}} \right) \right)^2$$

$$= 4.28 \text{ ft}^2$$

$$\dot{m}_{\text{inlet}} = \dot{m}_{\text{outlet}} = 9200 \frac{\text{lbm}}{\text{sec}}$$

$$(\rho Av)_{\text{inlet}} = 9200 \frac{\text{lbm}}{\text{sec}}$$

$$v_{\text{inlet}} = \frac{9200 \frac{\text{lbm}}{\text{sec}}}{A\rho}$$

$$= \frac{9200 \frac{\text{lbm}}{\text{sec}}}{(4.28 \text{ ft}^2) \left(49 \frac{\text{lbm}}{\text{ft}^3} \right)}$$

$$v_{\text{inlet}} = 43.9 \frac{\text{ft}}{\text{sec}}$$

The above example indicates that the flow rate into the system is the same as that out of the system. The same concept is true even though more than one flow path may enter or leave the system at the same time. The mass balance is simply adjusted to state that the sum of all flows entering the system is equal to the sum of all the flows leaving the system if steady-state conditions exist. An example of this physical case is included in the following example.

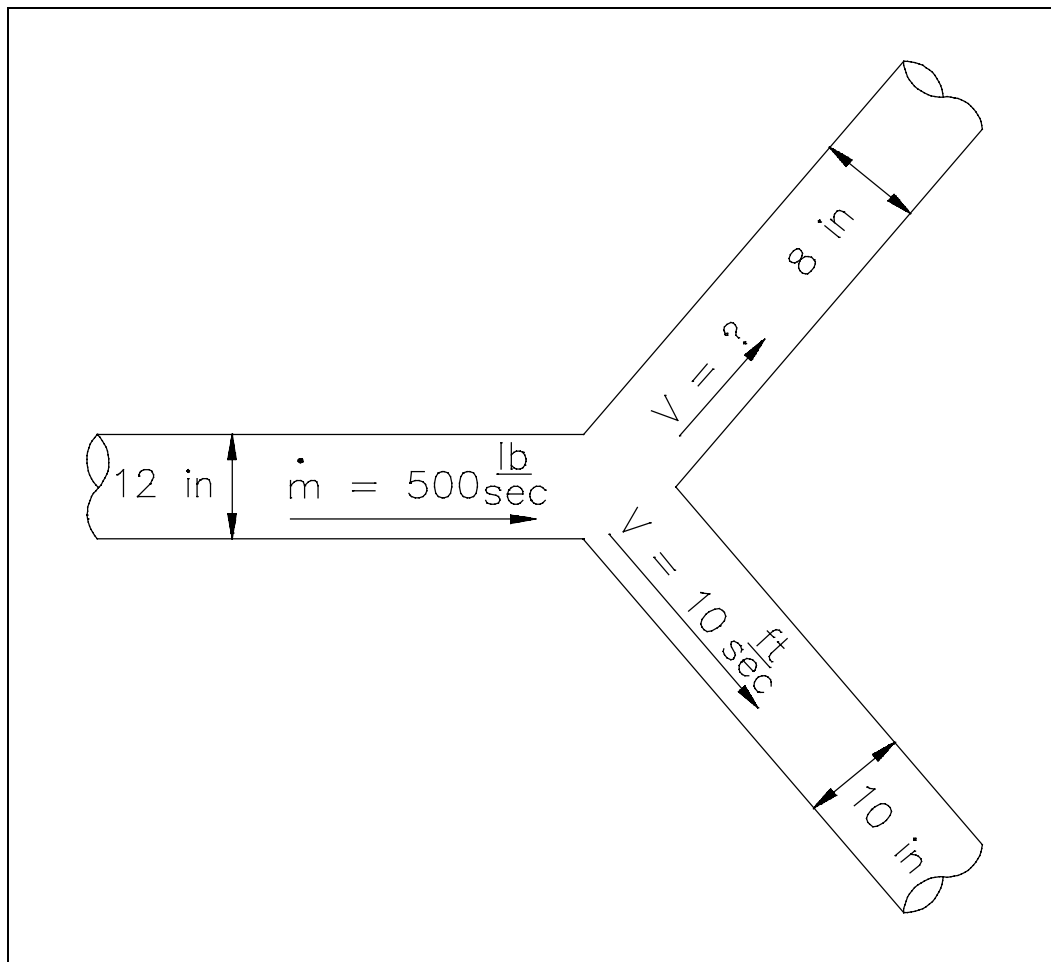


Figure 4 "Y" Configuration for Example Problem

Example: Continuity Equation - Multiple Outlets

A piping system has a "Y" configuration for separating the flow as shown in Figure 4. The diameter of the inlet leg is 12 in., and the diameters of the outlet legs are 8 and 10 in. The velocity in the 10 in. leg is 10 ft/sec. The flow through the main portion is 500 lbm/sec. The density of water is 62.4 lbm/ft^3 . What is the velocity out of the 8 in. pipe section?

Solution:

$$A_8 = \pi \left(4 \text{ in.} \cdot \frac{1 \text{ ft}}{12 \text{ in.}} \right)^2$$

$$= 0.349 \text{ ft}^2$$

$$A_{10} = \pi \left(5 \text{ in.} \cdot \frac{1 \text{ ft}}{12 \text{ in.}} \right)^2$$

$$= 0.545 \text{ ft}^2$$

$$\Sigma \dot{m}_{\text{inlets}} = \Sigma \dot{m}_{\text{outlets}}$$

$$\dot{m}_{12} = \dot{m}_{10} + \dot{m}_8$$

$$\dot{m}_8 = \dot{m}_{12} - \dot{m}_{10}$$

$$(\rho A v)_8 = \dot{m}_{12} - (\rho A v)_{10}$$

$$v_8 = \frac{\dot{m}_{12} - (\rho A v)_{10}}{(\rho A)_8}$$

$$= \frac{500 \frac{\text{lbm}}{\text{sec}} - \left(62.4 \frac{\text{lbm}}{\text{ft}^3} \right) (0.545 \text{ ft}^2) \left(10 \frac{\text{ft}}{\text{sec}} \right)}{\left(62.4 \frac{\text{lbm}}{\text{ft}^3} \right) (0.349 \text{ ft}^2)}$$

$$v_8 = 7.3 \frac{\text{ft}}{\text{sec}}$$

Summary

The main points of this chapter are summarized on the next page.

Continuity Equation Summary

- Density changes in a fluid are inversely proportional to temperature changes.
- Buoyancy is the tendency of a body to float or rise when submerged in a fluid.
- The pressure exerted by a column of water is directly proportional to the height of the column and the density of the water.

$$P = \frac{\rho h g}{g_c}$$

- Pascal's law states that pressure applied to a confined fluid is transmitted undiminished throughout the confining vessel of a system.
- Volumetric flow rate is the volume of fluid per unit time passing a point in a fluid system.
- Mass flow rate is the mass of fluid per unit time passing a point in a fluid system.
- The volumetric flow rate is calculated by the product of the average fluid velocity and the cross-sectional area for flow.

$$\dot{V} = A v$$

- The mass flow rate is calculated by the product of the volumetric flow rate and the fluid density.

$$\dot{m} = \rho A v$$

- The principle of conservation of mass states that all mass flow rates into a control volume are equal to all mass flow rates out of the control volume plus the rate of change of mass within the control volume.
- For a control volume with a single inlet and outlet, the continuity equation can be expressed as follows:

$$\dot{m}_{\text{inlet}} = \dot{m}_{\text{outlet}}$$

- For a control volume with multiple inlets and outlets, the continuity equation is:

$$\sum \dot{m}_{\text{inlets}} = \sum \dot{m}_{\text{outlets}}$$

LAMINAR AND TURBULENT FLOW

The characteristics of laminar and turbulent flow are very different. To understand why turbulent or laminar flow is desirable in the operation of a particular system, it is necessary to understand the characteristics of laminar and turbulent flow.

- EO 1.9** **DESCRIBE** the characteristics and flow velocity profiles of laminar flow and turbulent flow.
- EO 1.10** **DEFINE** the property of viscosity.
- EO 1.11** **DESCRIBE** how the viscosity of a fluid varies with temperature.
- EO 1.12** **DESCRIBE** the characteristics of an ideal fluid.
- EO 1.13** **DESCRIBE** the relationship between the Reynolds number and the degree of turbulence of the flow.
-

Flow Regimes

All fluid flow is classified into one of two broad categories or regimes. These two flow regimes are laminar flow and turbulent flow. The flow regime, whether laminar or turbulent, is important in the design and operation of any fluid system. The amount of fluid friction, which determines the amount of energy required to maintain the desired flow, depends upon the mode of flow. This is also an important consideration in certain applications that involve heat transfer to the fluid.

Laminar Flow

Laminar flow is also referred to as streamline or viscous flow. These terms are descriptive of the flow because, in laminar flow, (1) layers of water flowing over one another at different speeds with virtually no mixing between layers, (2) fluid particles move in definite and observable paths or streamlines, and (3) the flow is characteristic of viscous (thick) fluid or is one in which viscosity of the fluid plays a significant part.

Turbulent Flow

Turbulent flow is characterized by the irregular movement of particles of the fluid. There is no definite frequency as there is in wave motion. The particles travel in irregular paths with no observable pattern and no definite layers.

Flow Velocity Profiles

Not all fluid particles travel at the same velocity within a pipe. The shape of the velocity curve (the velocity profile across any given section of the pipe) depends upon whether the flow is laminar or turbulent. If the flow in a pipe is laminar, the velocity distribution at a cross section will be parabolic in shape with the maximum velocity at the center being about twice the average velocity in the pipe. In turbulent flow, a fairly flat velocity distribution exists across the section of pipe, with the result that the entire fluid flows at a given single value. Figure 5 helps illustrate the above ideas. The velocity of the fluid in contact with the pipe wall is essentially zero and increases the further away from the wall.

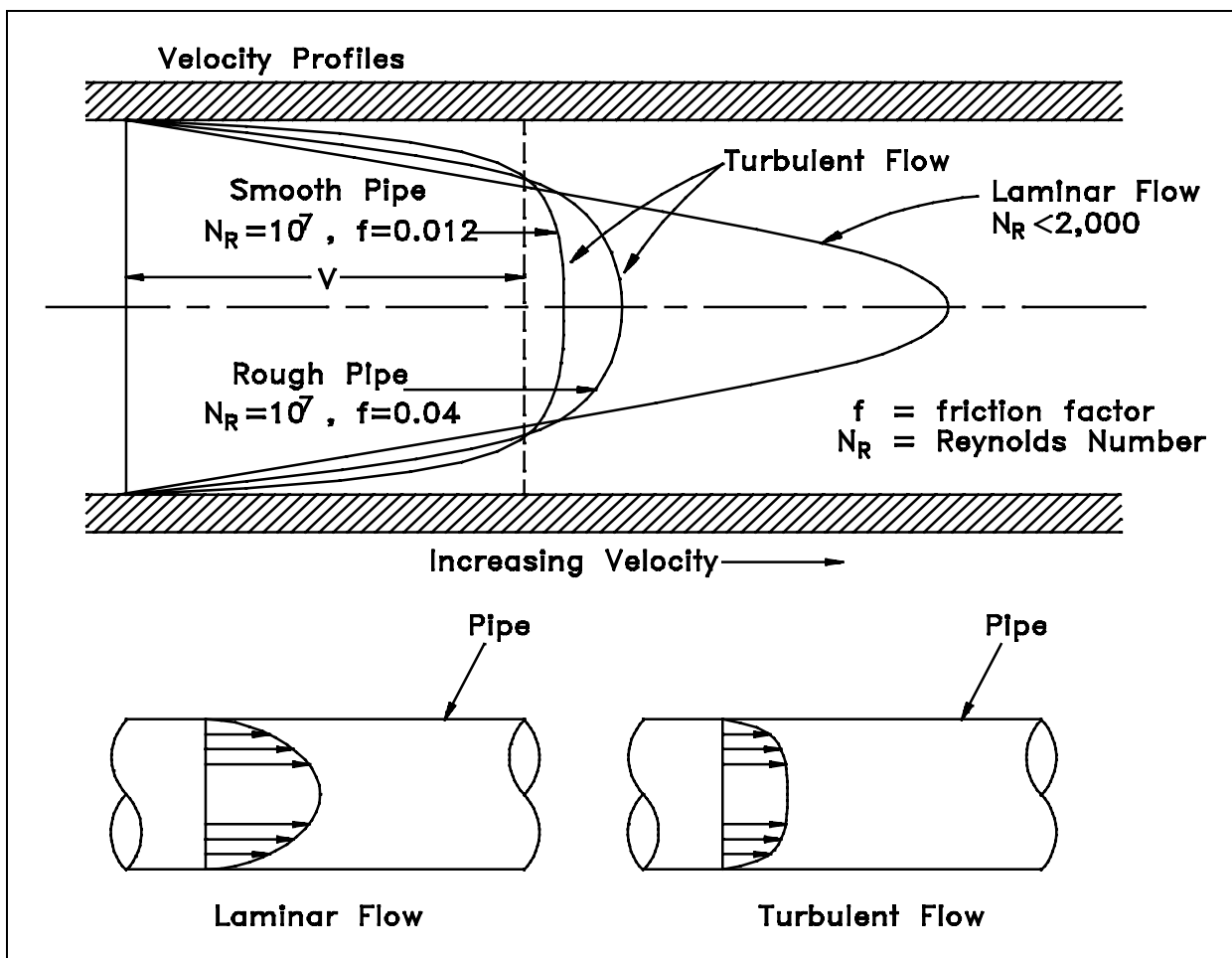


Figure 5 Laminar and Turbulent Flow Velocity Profiles

Note from Figure 5 that the velocity profile depends upon the surface condition of the pipe wall. A smoother wall results in a more uniform velocity profile than a rough pipe wall.

Average (Bulk) Velocity

In many fluid flow problems, instead of determining exact velocities at different locations in the same flow cross-section, it is sufficient to allow a single average velocity to represent the velocity of all fluid at that point in the pipe. This is fairly simple for turbulent flow since the velocity profile is flat over the majority of the pipe cross-section. It is reasonable to assume that the average velocity is the same as the velocity at the center of the pipe.

If the flow regime is laminar (the velocity profile is parabolic), the problem still exists of trying to represent the "average" velocity at any given cross-section since an average value is used in the fluid flow equations. Technically, this is done by means of integral calculus. Practically, the student should use an average value that is half of the center line value.

Viscosity

Viscosity is a fluid property that measures the resistance of the fluid to deforming due to a shear force. Viscosity is the internal friction of a fluid which makes it resist flowing past a solid surface or other layers of the fluid. Viscosity can also be considered to be a measure of the resistance of a fluid to flowing. A thick oil has a high viscosity; water has a low viscosity. The unit of measurement for absolute viscosity is:

$$\mu = \text{absolute viscosity of fluid (lbf-sec/ft}^2\text{)}.$$

The viscosity of a fluid is usually significantly dependent on the temperature of the fluid and relatively independent of the pressure. For most fluids, as the temperature of the fluid increases, the viscosity of the fluid decreases. An example of this can be seen in the lubricating oil of engines. When the engine and its lubricating oil are cold, the oil is very viscous, or thick. After the engine is started and the lubricating oil increases in temperature, the viscosity of the oil decreases significantly and the oil seems much thinner.

Ideal Fluid

An *ideal fluid* is one that is incompressible and has no viscosity. Ideal fluids do not actually exist, but sometimes it is useful to consider what would happen to an ideal fluid in a particular fluid flow problem in order to simplify the problem.

Reynolds Number

The flow regime (either laminar or turbulent) is determined by evaluating the Reynolds number of the flow (refer to figure 5). The *Reynolds number*, based on studies of Osborn Reynolds, is a dimensionless number comprised of the physical characteristics of the flow. Equation 3-7 is used to calculate the Reynolds number (N_R) for fluid flow.

$$N_R = \rho v D / \mu g_c \quad (3-7)$$

where:

- N_R = Reynolds number (unitless)
- v = average velocity (ft/sec)
- D = diameter of pipe (ft)
- μ = absolute viscosity of fluid (lbf-sec/ft²)
- ρ = fluid mass density (lbm/ft³)
- g_c = gravitational constant (32.2 ft-lbm/lbf-sec²)

For practical purposes, if the Reynolds number is less than 2000, the flow is laminar. If it is greater than 3500, the flow is turbulent. Flows with Reynolds numbers between 2000 and 3500 are sometimes referred to as transitional flows. Most fluid systems in nuclear facilities operate with turbulent flow. Reynolds numbers can be conveniently determined using a Moody Chart; an example of which is shown in Appendix B. Additional detail on the use of the Moody Chart is provided in subsequent text.

Summary

The main points of this chapter are summarized below.

Laminar and Turbulent Flow Summary

- **Laminar Flow**
Layers of water flow over one another at different speeds with virtually no mixing between layers.
The flow velocity profile for laminar flow in circular pipes is parabolic in shape, with a maximum flow in the center of the pipe and a minimum flow at the pipe walls.
The average flow velocity is approximately one half of the maximum velocity.
- **Turbulent Flow**
The flow is characterized by the irregular movement of particles of the fluid.
The flow velocity profile for turbulent flow is fairly flat across the center section of a pipe and drops rapidly extremely close to the walls.
The average flow velocity is approximately equal to the velocity at the center of the pipe.
- Viscosity is the fluid property that measures the resistance of the fluid to deforming due to a shear force. For most fluids, temperature and viscosity are inversely proportional.
- An ideal fluid is one that is incompressible and has no viscosity.
- An increasing Reynolds number indicates an increasing turbulence of flow.

BERNOULLI'S EQUATION

Bernoulli's equation is a special case of the general energy equation that is probably the most widely-used tool for solving fluid flow problems. It provides an easy way to relate the elevation head, velocity head, and pressure head of a fluid. It is possible to modify Bernoulli's equation in a manner that accounts for head losses and pump work.

- EO 1.14** **DESCRIBE** the relationship between Bernoulli's equation and the First Law of Thermodynamics.
- EO 1.15** **DEFINE** the term head with respect to its use in fluid flow.
- EO 1.16** **EXPLAIN** the energy conversions that take place in a fluid system between the velocity, elevation, and pressure heads as flow continues through a piping system.
- EO 1.17** **Given** the initial and final conditions of the system, **CALCULATE** the unknown fluid properties using the simplified Bernoulli equation.
- EO 1.18** **DESCRIBE** the restrictions applied to Bernoulli's equation when presented in its simplest form.
- EO 1.19** **EXPLAIN** how to extend the Bernoulli equation to more general applications.
- EO 1.20** **RELATE** Bernoulli's principle to the operation of a venturi.

General Energy Equation

The conservation of energy principle states that energy can be neither created nor destroyed. This is equivalent to the First Law of Thermodynamics, which was used to develop the general energy equation in the module on thermodynamics. Equation 3-8 is a statement of the general energy equation for an open system.

$$Q + (U + PE + KE + PV)_{in} = W + (U + PE + KE + PV)_{out} + (U + PE + KE + PV)_{stored} \quad (3-8)$$

where:

Q	=	heat (Btu)
U	=	internal energy (Btu)
PE	=	potential energy (ft-lbf)
KE	=	kinetic energy (ft-lbf)
P	=	pressure (lbf/ft ²)
V	=	volume (ft ³)
W	=	work (ft-lbf)

Simplified Bernoulli Equation

Bernoulli's equation results from the application of the general energy equation and the first law of thermodynamics to a steady flow system in which no work is done on or by the fluid, no heat is transferred to or from the fluid, and no change occurs in the internal energy (i.e., no temperature change) of the fluid. Under these conditions, the general energy equation is simplified to Equation 3-9.

$$(PE + KE + PV)_1 = (PE + KE + PV)_2 \quad (3-9)$$

Substituting appropriate expressions for the potential energy and kinetic energy, Equation 3-9 can be rewritten as Equation 3-10.

$$\frac{mgz_1}{g_c} + \frac{mv_1^2}{2g_c} + P_1V_1 = \frac{mgz_2}{g_c} + \frac{mv_2^2}{2g_c} + P_2V_2 \quad (3-10)$$

where:

m	=	mass (lbm)
z	=	height above reference (ft)
v	=	average velocity (ft/sec)
g	=	acceleration due to gravity (32.17 ft/sec ²)
g _c	=	gravitational constant, (32.17 ft-lbm/lbf-sec ²)

Note: The factor g_c is only required when the English System of measurement is used and mass is measured in pound mass. It is essentially a conversion factor needed to allow the units to come out directly. No factor is necessary if mass is measured in slugs or if the metric system of measurement is used.

Each term in Equation 3-10 represents a form of energy possessed by a moving fluid (potential, kinetic, and pressure related energies). In essence, the equation physically represents a balance of the KE, PE, PV energies so that if one form of energy increases, one or more of the others will decrease to compensate and vice versa.

Multiplying all terms in Equation 3-10 by the factor g_c/mg results in the form of Bernoulli's equation shown by Equation 3-11.

$$z_1 + \frac{v_1^2}{2g} + P_1 v_1 \frac{g_c}{g} = z_2 + \frac{v_2^2}{2g} + P_2 v_2 \frac{g_c}{g} \quad (3-11)$$

Head

Since the units for all the different forms of energy in Equation 3-11 are measured in units of distance, these terms are sometimes referred to as "heads" (pressure head, velocity head, and elevation head). The term *head* is used by engineers in reference to pressure. It is a reference to the height, typically in feet, of a column of water that a given pressure will support. Each of the energies possessed by a fluid can be expressed in terms of head. The elevation head represents the potential energy of a fluid due to its elevation above a reference level. The velocity head represents the kinetic energy of the fluid. It is the height in feet that a flowing fluid would rise in a column if all of its kinetic energy were converted to potential energy. The pressure head represents the flow energy of a column of fluid whose weight is equivalent to the pressure of the fluid.

The sum of the elevation head, velocity head, and pressure head of a fluid is called the total head. Thus, Bernoulli's equation states that the total head of the fluid is constant.

Energy Conversions in Fluid Systems

Bernoulli's equation makes it easy to examine how energy transfers take place among elevation head, velocity head, and pressure head. It is possible to examine individual components of piping systems and determine what fluid properties are varying and how the energy balance is affected.

If a pipe containing an ideal fluid undergoes a gradual expansion in diameter, the continuity equation tells us that as the diameter and flow area get bigger, the flow velocity must decrease to maintain the same mass flow rate. Since the outlet velocity is less than the inlet velocity, the velocity head of the flow must decrease from the inlet to the outlet. If the pipe lies horizontal, there is no change in elevation head; therefore, the decrease in velocity head must be compensated for by an increase in pressure head. Since we are considering an ideal fluid that is incompressible, the specific volume of the fluid will not change. The only way that the pressure head for an incompressible fluid can increase is for the pressure to increase. So the Bernoulli equation indicates that a decrease in flow velocity in a horizontal pipe will result in an increase in pressure.

If a constant diameter pipe containing an ideal fluid undergoes a decrease in elevation, the same net effect results, but for different reasons. In this case the flow velocity and the velocity head must be constant to satisfy the mass continuity equation.

So the decrease in elevation head can only be compensated for by an increase in pressure head. Again, the fluid is incompressible so the increase in pressure head must result in an increase in pressure.

Although the Bernoulli equation has several restrictions placed upon it, there are many physical fluid problems to which it is applied. As in the case of the conservation of mass, the Bernoulli equation may be applied to problems in which more than one flow may enter or leave the system at the same time. Of particular note is the fact that series and parallel piping system problems are solved using the Bernoulli equation.

Example: Bernoulli's Equation

Assume frictionless flow in a long, horizontal, conical pipe. The diameter is 2.0 ft at one end and 4.0 ft at the other. The pressure head at the smaller end is 16 ft of water. If water flows through this cone at a rate of 125.6 ft³/sec, find the velocities at the two ends and the pressure head at the larger end.

Solution:

$$\dot{V}_1 = A_1 v_1$$

$$v_1 = \frac{\dot{V}_1}{A_1} \qquad v_2 = \frac{\dot{V}_2}{A_2}$$

$$v_1 = \frac{125.6 \frac{\text{ft}^3}{\text{sec}}}{\pi(1 \text{ ft})^2} \qquad v_2 = \frac{125.6 \frac{\text{ft}^3}{\text{sec}}}{\pi(2 \text{ ft})^2}$$

$$v_1 = 40 \frac{\text{ft}}{\text{sec}} \qquad v_2 = 10 \frac{\text{ft}}{\text{sec}}$$

$$z_1 + \frac{v_1^2}{2g} + P_1 v_1 \frac{g_c}{g} = z_2 + \frac{v_2^2}{2g} + P_2 v_2 \frac{g_c}{g}$$

$$P_2 v_2 \frac{g_c}{g} = P_1 v_1 \frac{g_c}{g} + (z_1 - z_2) + \frac{v_1^2 - v_2^2}{2g}$$

$$= 16 \text{ ft} + 0 \text{ ft} + \frac{\left(40 \frac{\text{ft}}{\text{sec}}\right)^2 - \left(10 \frac{\text{ft}}{\text{sec}}\right)^2}{2 \left(32.17 \frac{\text{ft-lbm}}{\text{lbf-sec}^2}\right)}$$

$$= 39.3 \text{ ft}$$

Restrictions on the Simplified Bernoulli Equation

Practical applications of the simplified Bernoulli Equation to real piping systems is not possible due to two restrictions. One serious restriction of the Bernoulli equation in its present form is that no fluid friction is allowed in solving piping problems. Therefore, Equation 3-10 only applies to ideal fluids. However, in reality, the total head possessed by the fluid cannot be transferred completely from one point to another because of friction. Taking these losses of head into account would provide a much more accurate description of what takes place physically. This is especially true because one purpose of a pump in a fluid system is to overcome the losses in pressure due to pipe friction.

The second restriction on Bernoulli's equation is that no work is allowed to be done on or by the fluid. This restriction prevents two points in a fluid stream from being analyzed if a pump exists between the two points. Since most flow systems include pumps, this is a significant limitation. Fortunately, the simplified Bernoulli equation can be modified in a manner that satisfactorily deals with both head losses and pump work.

Extended Bernoulli

The Bernoulli equation can be modified to take into account gains and losses of head. The resulting equation, referred to as the Extended Bernoulli equation, is very useful in solving most fluid flow problems. In fact, the Extended Bernoulli equation is probably used more than any other fluid flow equation. Equation 3-12 is one form of the Extended Bernoulli equation.

$$z_1 + \frac{v_1^2}{2g} + P_1 v_1 \frac{g_c}{g} + H_p = z_2 + \frac{v_2^2}{2g} + P_2 v_2 \frac{g_c}{g} + H_f \quad (3-12)$$

where:

- z = height above reference level (ft)
- v = average velocity of fluid (ft/sec)
- P = pressure of fluid (lbf/ft²)
- v = specific volume of fluid (ft³/lbm)
- H_p = head added by pump (ft)
- H_f = head loss due to fluid friction (ft)
- g = acceleration due to gravity (ft/sec²)

The head loss due to fluid friction (H_f) represents the energy used in overcoming friction caused by the walls of the pipe. Although it represents a loss of energy from the standpoint of fluid flow, it does not normally represent a significant loss of total energy of the fluid. It also does not violate the law of conservation of energy since the head loss due to friction results in an equivalent increase in the internal energy (u) of the fluid. These losses are greatest as the fluid flows through entrances, exits, pumps, valves, fittings, and any other piping with rough inner surfaces.

Most techniques for evaluating head loss due to friction are empirical (based almost exclusively on experimental evidence) and are based on a proportionality constant called the friction factor (f), which will be discussed in the next section.

Example: Extended Bernoulli

Water is pumped from a large reservoir to a point 65 feet higher than the reservoir. How many feet of head must be added by the pump if 8000 lbm/hr flows through a 6-inch pipe and the frictional head loss is 2 feet? The density of the fluid is 62.4 lbm/ft³, and the cross-sectional area of a 6-inch pipe is 0.2006 ft².

Solution:

To use the modified form of Bernoulli's equation, reference points are chosen at the surface of the reservoir (point 1) and at the outlet of the pipe (point 2). The pressure at the surface of the reservoir is the same as the pressure at the exit of the pipe, i.e., atmospheric pressure. The velocity at point 1 will be essentially zero.

Using the equation for the mass flow rate to determine the velocity at point 2:

$$\dot{m}_2 = \rho A_2 v_2$$

$$v_2 = \frac{\dot{m}_2}{\rho A_2}$$

$$v_2 = \frac{8000 \frac{\text{lbm}}{\text{hr}}}{\left(62.4 \frac{\text{lbm}}{\text{ft}^3}\right) (0.2006 \text{ ft}^2)}$$

$$v_2 = 639 \frac{\text{ft}}{\text{hr}} \left(\frac{1 \text{ hr}}{3600 \text{ sec}} \right)$$

$$v_2 = 0.178 \frac{\text{ft}}{\text{sec}}$$

Now we can use the Extended Bernoulli equation to determine the required pump head.

$$z_1 + \frac{v_1^2}{2g} + P_1 v_1 \frac{g_c}{g} + H_p = z_2 + \frac{v_2^2}{2g} + P_2 v_2 \frac{g_c}{g} + H_f$$

$$H_p = (z_2 - z_1) + \left(\frac{v_2^2 - v_1^2}{2g} \right) + (P_2 - P_1) v \frac{g_c}{g} + H_f$$

$$= 65 \text{ ft} + \frac{\left(0.178 \frac{\text{ft}}{\text{sec}} \right)^2 - \left(0 \frac{\text{ft}}{\text{sec}} \right)^2}{2 \left(32.17 \frac{\text{ft-lbm}}{\text{lbf-sec}^2} \right)} + 0 \text{ ft} + 2 \text{ ft}$$

$$H_p = 67 \text{ ft}$$

The student should note that the solution of this example problem has a numerical value that "makes sense" from the data given in the problem. The total head increase of 67 ft. is due primarily to the 65 ft. elevation increase and the 2 ft. of friction head.

Application of Bernoulli's Equation to a Venturi

Many plant components, such as a venturi, may be analyzed using Bernoulli's equation and the continuity equation. A venturi is a flow measuring device that consists of a gradual contraction followed by a gradual expansion. An example of a venturi is shown in Figure 6. By measuring the differential pressure between the inlet of the venturi (point 1) and the throat of the venturi (point 2), the flow velocity and mass flow rate can be determined based on Bernoulli's equation.

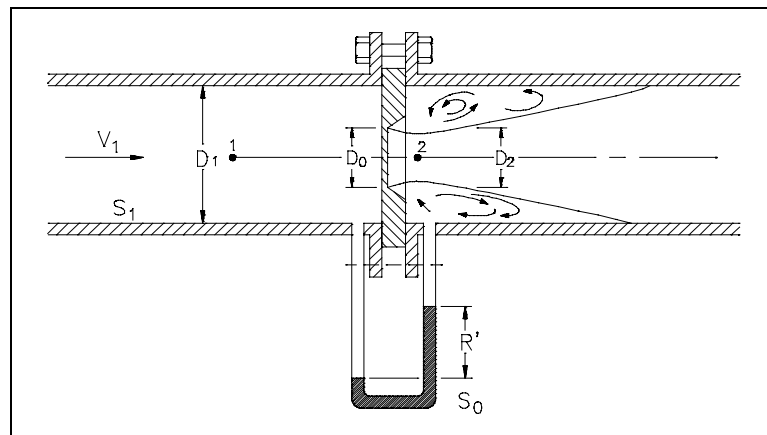


Figure 6 Venturi Meter

Bernoulli's equation states that the total head of the flow must be constant. Since the elevation does not change significantly, if at all, between points 1 and 2, the elevation head at the two points will be essentially the same and will cancel out of the equation. So Bernoulli's equation simplifies to Equation 3-13 for a venturi.

$$\frac{v_1^2}{2g} + P_1 v_1 \frac{g_c}{g} = \frac{v_2^2}{2g} + P_2 v_2 \frac{g_c}{g} \quad (3-13)$$

Applying the continuity equation to points 1 and 2 allows us to express the flow velocity at point 1 as a function of the flow velocity at point 2 and the ratio of the two flow areas.

$$\rho_1 A_1 v_1 = \rho_2 A_2 v_2$$

$$v_1 = \frac{\rho_2 A_2 v_2}{\rho_1 A_1}$$

$$v_1 = v_2 \frac{A_2}{A_1}$$

Using algebra to rearrange Equation 3-13 and substituting the above result for v_1 allows us to solve for v_2 .

$$\frac{v_2^2 - v_1^2}{2g} = (P_1 - P_2) v \frac{g_c}{g}$$

$$v_2^2 - \left(v_2 \frac{A_2}{A_1} \right)^2 = (P_1 - P_2) 2v g_c$$

$$v_2^2 \left(1 - \left(\frac{A_2}{A_1} \right)^2 \right) = (P_1 - P_2) 2v g_c$$

$$v_2^2 = \frac{(P_1 - P_2) 2vg_c}{\left(1 - \left(\frac{A_2}{A_1}\right)^2\right)}$$

$$v_2 = \sqrt{\frac{(P_1 - P_2) 2vg_c}{\left(1 - \left(\frac{A_2}{A_1}\right)^2\right)}}$$

$$v_2 = \sqrt{P_1 - P_2} \sqrt{\frac{2vg_c}{\left(1 - \left(\frac{A_2}{A_1}\right)^2\right)}}$$

Therefore the flow velocity at the throat of the venturi and the volumetric flow rate are directly proportional to the square root of the differential pressure.

The pressures at the upstream section and throat are actual pressures, and velocities from Bernoulli's equation without a loss term are theoretical velocities. When losses are considered in the energy equation, the velocities are actual velocities. First, with the Bernoulli equation (that is, without a head-loss term), the theoretical velocity at the throat is obtained. Then by multiplying this by the venturi factor (C_v), which accounts for friction losses and equals 0.98 for most venturis, the actual velocity is obtained. The actual velocity times the actual area of the throat determines the actual discharge volumetric flow rate.

The pressure drop, $P_1 - P_2$, across the venturi can be used to measure the flow rate using a U-tube manometer as shown in Figure 6. The reading, R , of the manometer is proportional to the pressure drop and thus the velocity of the fluid.

Summary

The main points of this chapter are summarized below.

Bernoulli's Equation Summary

- Bernoulli's equation is an application of the First Law of Thermodynamics.
- Bernoulli's equation is an application of the general energy equation to a steady flow system in which no work is done on or by the fluid, no heat is transferred to or from the fluid, and no change occurs in the internal energy of the fluid.
- Head is the term used to describe pressure exerted on or by a fluid.
- As fluid flows in a piping system, changes in elevation, velocity, and pressure heads must be consistent so that Bernoulli's equation is satisfied.
- Bernoulli's equation can be modified to take into account friction losses and pump work.
- A venturi can be used to determine mass flow rates due to changes in pressure and fluid velocity.
- The volumetric flow rate through a venturi is directly proportional to the square root of the differential pressure between the venturi's inlet and its throat.

HEAD LOSS

The head loss that occurs in pipes is dependent on the flow velocity, pipe length and diameter, and a friction factor based on the roughness of the pipe and the Reynolds number of the flow. The head loss that occurs in the components of a flow path can be correlated to a piping length that would cause an equivalent head loss.

- EO 1.21** **DEFINE** the terms head loss, frictional loss, and minor losses.
- EO 1.22** **DETERMINE** friction factors for various flow situations using the Moody chart.
- EO 1.23** **CALCULATE** the head loss in a fluid system due to frictional losses using Darcy's equation.
- EO 1.24** **CALCULATE** the equivalent length of pipe that would cause the same head loss as the minor losses that occur in individual components.
-

Head Loss

Head loss is a measure of the reduction in the total head (sum of elevation head, velocity head and pressure head) of the fluid as it moves through a fluid system. Head loss is unavoidable in real fluids. It is present because of: the friction between the fluid and the walls of the pipe; the friction between adjacent fluid particles as they move relative to one another; and the turbulence caused whenever the flow is redirected or affected in any way by such components as piping entrances and exits, pumps, valves, flow reducers, and fittings.

Frictional loss is that part of the total head loss that occurs as the fluid flows through straight pipes. The head loss for fluid flow is directly proportional to the length of pipe, the square of the fluid velocity, and a term accounting for fluid friction called the friction factor. The head loss is inversely proportional to the diameter of the pipe.

$$\text{Head Loss} \propto f \frac{Lv^2}{D}$$

Friction Factor

The friction factor has been determined to depend on the Reynolds number for the flow and the degree of roughness of the pipe's inner surface.

The quantity used to measure the roughness of the pipe is called the relative roughness, which equals the average height of surface irregularities (ϵ) divided by the pipe diameter (D).

$$\text{Relative Roughness} = \frac{\epsilon}{D}$$

The value of the friction factor is usually obtained from the Moody Chart (Figure B-1 of Appendix B). The Moody Chart can be used to determine the friction factor based on the Reynolds number and the relative roughness.

Example:

Determine the friction factor (f) for fluid flow in a pipe that has a Reynolds number of 40,000 and a relative roughness of 0.01.

Solution:

Using the Moody Chart, a Reynolds number of 40,000 intersects the curve corresponding to a relative roughness of 0.01 at a friction factor of 0.04.

Darcy's Equation

The frictional head loss can be calculated using a mathematical relationship that is known as Darcy's equation for head loss. The equation takes two distinct forms. The first form of Darcy's equation determines the losses in the system associated with the length of the pipe.

$$H_f = f \frac{L v^2}{D 2 g} \quad (3-14)$$

where:

- f = friction factor (unitless)
- L = length of pipe (ft)
- D = diameter of pipe (ft)
- v = fluid velocity (ft/sec)
- g = gravitational acceleration (ft/sec²)

Example: Darcy's Head Loss Equation

A pipe 100 feet long and 20 inches in diameter contains water at 200°F flowing at a mass flow rate of 700 lbm/sec. The water has a density of 60 lbm/ft³ and a viscosity of 1.978 x 10⁻⁷ lbf-sec/ft². The relative roughness of the pipe is 0.00008. Calculate the head loss for the pipe.

Solution:

The sequence of steps necessary to solve this problem is first to determine the flow velocity. Second, using the flow velocity and the fluid properties given, calculate the Reynolds number. Third, determine the friction factor from the Reynolds number and the relative roughness. Finally, use Darcy's equation to determine the head loss.

$$\dot{m} = \rho Av$$

$$v = \frac{\dot{m}}{\rho A}$$

$$= \frac{700 \frac{\text{lbm}}{\text{sec}}}{\left(60 \frac{\text{lbm}}{\text{ft}^3}\right) \pi (10 \text{ in})^2 \frac{1 \text{ ft}^2}{144 \text{ in}^2}}$$

$$v = 5.35 \frac{\text{ft}}{\text{sec}}$$

$$N_R = \frac{\rho v D}{\mu g_c}$$

$$N_R = \frac{\left(60 \frac{\text{lbm}}{\text{ft}^3}\right) \left(5.35 \frac{\text{ft}}{\text{sec}}\right) \left(20 \text{ in} \frac{1 \text{ ft}}{12 \text{ in}}\right)}{1.978 \times 10^{-7} \frac{\text{lbf-sec}}{\text{ft}^2} 32.17 \frac{\text{ft-lbm}}{\text{lbf-sec}^2}} = 8.4 \times 10^7$$

Use the Moody Chart for a Reynolds number of 8.4×10^7 and a relative roughness of 0.00008.

$$f = 0.012$$

$$H_f = f \frac{L}{D} \frac{v^2}{2g}$$

$$= (0.012) \frac{100 \text{ ft}}{20 \text{ in} \left(\frac{1 \text{ ft}}{12 \text{ in}}\right)} \cdot \frac{\left(5.35 \frac{\text{ft}}{\text{sec}}\right)^2}{2 \left(32.17 \frac{\text{ft}}{\text{sec}^2}\right)}$$

$$H_f = 0.32 \text{ ft}$$

Minor Losses

The losses that occur in pipelines due to bends, elbows, joints, valves, etc. are sometimes called *minor losses*. This is a misnomer because in many cases these losses are more important than the losses due to pipe friction, considered in the preceding section. For all minor losses in turbulent flow, the head loss varies as the square of the velocity. Thus a convenient method of expressing the minor losses in flow is by means of a loss coefficient (k). Values of the loss coefficient (k) for typical situations and fittings is found in standard handbooks. The form of Darcy's equation used to calculate minor losses of individual fluid system components is expressed by Equation 3-15.

$$H_f = k \frac{v^2}{2g} \quad (3-15)$$

Equivalent Piping Length

Minor losses may be expressed in terms of the equivalent length (L_{eq}) of pipe that would have the same head loss for the same discharge flow rate. This relationship can be found by setting the two forms of Darcy's equation equal to each other.

$$f \frac{L}{D} \frac{v^2}{2g} = k \frac{v^2}{2g}$$

This yields two relationships that are useful.

$$L_{eq} = k \frac{D}{f} \quad (3-16)$$

$$k = f \frac{L_{eq}}{D} \quad (3-17)$$

Typical values of L_{eq}/D for common piping system components are listed in Table 1. The equivalent length of piping that will cause the same head loss as a particular component can be determined by multiplying the value of L_{eq}/D for that component by the diameter of the pipe. The higher the value of L_{eq}/D , the longer the equivalent length of pipe.

TABLE 1	
Typical Values of $\frac{L_{eq}}{D}$	
Item	$\frac{L_{eq}}{D}$
<u>Globe Valve</u>	
Conventional	400
Y-Pattern	160
<u>Gate Valve</u>	
Fully Open	10
75% Open	35
50% Open	150
25% Open	900
<u>Standard Tee</u>	
Flow through Run	10
Flow through Branch	60
90° Standard Elbow	30
45° Standard Elbow	16
Return Bend	50

Example:

A fully-open gate valve is in a pipe with a diameter of 10 inches. What equivalent length of pipe would cause the same head loss as the gate valve?

Solution:

From Table 1, we find that the value of L_{eq}/D for a fully-open gate valve is 10.

$$\begin{aligned}
 L_{eq} &= (L/D) D \\
 &= 10 (10 \text{ inches}) \\
 &= 100 \text{ inches}
 \end{aligned}$$

By adding the equivalent lengths of all components to the actual length of pipe in a system we can obtain the L_{eq} value for the entire piping system.

Summary

The main points of this chapter are summarized below.

Head Loss Summary

- Head loss is the reduction in the total head (sum of potential head, velocity head, and pressure head) of a fluid caused by the friction present in the fluid's motion.
- Frictional loss is that part of the total head loss that occurs as the fluid flows through straight pipes.
- Minor losses are the head losses that occur due to bends, elbows, joints, valves, and other components. Any time the flow experiences a change in direction or a change in cross-sectional area, it will experience a head loss.
- The friction factor for fluid flow can be determined using a Moody Chart if the relative roughness of the pipe and the Reynolds number of the flow can be determined.
- Darcy's equation can be used to calculate frictional losses.
- A special form of Darcy's equation can be used to calculate minor losses.
- The length of pipe that would cause the same head loss as a valve or fitting can be determined by multiplying the value of L/D for the component found in handbooks or vendor manuals by the diameter of the pipe.

NATURAL CIRCULATION

Natural circulation is the circulation of fluid within piping systems or open pools that is due to the density changes caused by temperature differences. Natural circulation does not require any mechanical devices to maintain flow.

- EO 1.25** **DEFINE** natural circulation and forced circulation.
- EO 1.26** **DEFINE** thermal driving head.
- EO 1.27** **DESCRIBE** the conditions necessary for natural circulation to exist.
- EO 1.28** **EXPLAIN** the relationship between flow rate and temperature difference in natural circulation flow.
- EO 1.29** **DESCRIBE** how the operator can determine whether natural circulation exists in the reactor coolant system and other heat removal systems.
- EO 1.30** **DESCRIBE** how to enhance natural circulation flow.
-

Forced and Natural Circulation

In the previous chapters on fluid flow, it was explained that any time that fluid flows there is some friction associated with the movement, which will cause head loss. It was pointed out that this head loss is commonly compensated for in piping systems by pumps that do work on the fluid, compensating for the head loss due to friction. Circulation of fluid in systems by pumps is referred to as *forced circulation*.

It is possible to design some fluid systems in a manner that does not require the presence of pumps to provide circulation. The head required to compensate for the head losses is created by density gradients and elevation changes. Flow that occurs under these circumstances is called *natural circulation*.

Thermal Driving Head

Thermal driving head is the force that causes natural circulation to take place. It is caused by the difference in density between two bodies or areas of fluid.

Consider two equal volumes of the same type of fluid. If the two volumes are not at the same temperature, then the volume with the higher temperature will also have a lower density and, therefore, less mass. Since the volume at the higher temperature will have a lower mass, it will also have less force exerted on it by gravity. This difference in the force of gravity exerted on the fluid will tend to cause the hotter fluid to rise and the colder fluid to sink.

This effect is seen in many places. One example of this is a hot air balloon. The force causing a hot air balloon to rise is a result of a difference in density between the hot air inside the balloon and the cooler air surrounding it.

Heat added to the air in the balloon adds energy to the molecules of air. The movement of the air molecules increases and the air molecules take up more space. The air molecules inside the balloon take up more space than the same amount of air molecules outside the balloon. This means the hot air is less dense and lighter than the surrounding air. Since the air in the balloon is less dense, gravity has less effect on it. The result is that the balloon weighs less than the surrounding air. Gravity pulls cooler air down into the space occupied by the balloon. The downward movement of the cooler air forces the balloon out of the space previously occupied, and the balloon rises.

Conditions Required for Natural Circulation

Natural circulation will only occur if the correct conditions exist. Even after natural circulation has begun, removal of any one of these conditions will cause the natural circulation to stop. The conditions for natural circulation are as follows.

1. A temperature difference exists (heat source and heat sink exists).
2. The heat source is at a lower elevation than the heat sink.
3. The fluids must be in contact with each other.

There must be two bodies of fluid at different temperatures. This could also be one body of fluid with areas of different temperatures. The difference in temperature is necessary to cause a density difference in the fluid. The density difference is the driving force for natural circulation flow.

The difference in temperature must be maintained for the natural circulation to continue. Addition of heat by a heat source must exist at the high temperature area. Continuous removal of heat by a heat sink must exist at the low temperature area. Otherwise the temperatures would eventually equalize, and no further circulation would occur.

The heat source must be at a lower elevation than the heat sink. As shown by the example of the balloon, a warmer fluid is less dense and will tend to rise, and a cooler fluid is more dense and will tend to sink. To take advantage of the natural movement of warm and cool fluids, the heat source and heat sink must be at the proper elevations.

The two areas must be in contact so that flow between the areas is possible. If the flow path is obstructed or blocked, then natural circulation cannot occur.

Example of Natural Circulation Cooling

Natural circulation is frequently the primary means of cooling for pool-type reactors and for irradiated fuel assemblies stored in pools of water after removal from the reactor. The heat source is the fuel assembly. The heat sink is the bulk of the water in the pool.

Water at the bottom of a fuel assembly absorbs energy generated by the assembly. The water increases in temperature and decreases in density. Gravity pulls cooler (more dense) water into the bottom of the assembly displacing the warmer water. The warmer (lighter) water is forced to give up its position to the cooler (heavier) water. The warmer (lighter) water rises higher in the assembly. As water travels up the length of the assembly, it absorbs more energy. The water becomes lighter and lighter being continuously forced upward by more dense water moving in below it. In turn, the cooler water absorbs energy from the assembly and is also forced to rise as natural circulation flow continues. Water exiting the top of the fuel assembly gives up its energy as it mixes with the bulk of the water in the pool. The bulk of the water in the pool is commonly cooled by circulation through heat exchangers in a separate process.

Flow Rate and Temperature Difference

The thermal driving head that causes natural circulation is due to the density change caused by a temperature difference. In general, the greater the temperature difference between the hot and cold areas of fluid, the greater the thermal driving head and the resulting flow rate. However, it is good practice to keep the hot fluid subcooled to prevent a change of phase from occurring. It is possible to have natural circulation take place in two-phase flow, but it is usually more difficult to maintain flow.

Various parameters can be used to indicate or verify natural circulation is occurring. This is dependent on plant type. For instance for a pressurized water reactor (PWR) selected Reactor Coolant System (RCS) parameters that would be used are as follows.

1. RCS ΔT ($T_{\text{Hot}} - T_{\text{Cold}}$) should be 25-80% of the full power value and either steady or slowly decreasing. This indicates that the decay heat is being removed from the system at an adequate rate to maintain or reduce core temperatures.
2. RCS Hot and Cold leg temperatures should be steady or slowly decreasing. Again, this indicates that heat is being removed and the decay heat load is decreasing as expected.
3. Steam generator steam pressure (secondary side pressure) should be following RCS temperature. This verifies that the steam generator is removing heat from the RCS coolant.

If natural circulation for a PWR is in progress or is imminent, several actions can be performed to ensure or enhance core cooling capabilities. First, pressurizer level can be maintained greater than 50%. Secondly, maintain the RCS subcooled by 15°F or greater.

Both of these actions will help ensure steam/vapor pockets are not formed in the RCS where they would restrict RCS flow. Thirdly, maintain steam generator water level \geq normal range. This provides an adequate heat sink to ensure heat removal is sufficient to prevent boiling of the RCS.

Summary

The main points of this chapter are listed below.

Natural Circulation Flow Summary

- Natural circulation flow is circulation of a fluid without the use of mechanical devices.
- Forced circulation flow is circulation of a fluid through a system by pumps.
- Thermal driving head is the driving force for natural circulation caused by the difference in density between two areas of fluid.
- Three items are necessary to support natural circulation:
 - There must be a heat sink and a heat source.
 - The heat source must be located below the heat sink.
 - Flowpaths must exist between the warm fluid and the cold fluid.
- Generally, the greater the temperature difference, the higher the natural circulation flow rate.
- Natural circulation in a PWR can be verified by monitoring:
 - RCS ΔT - 25%-80% full power value
 - $T_{\text{Hot}} / T_{\text{Cold}}$ - steady or slowly decreasing
 - S/G steam pressure - tracking RCS temperature
- Natural circulation in a PWR can be enhanced by:
 - maintain pressurizer level $>50\%$
 - maintain RCS $\geq 15^{\circ}\text{F}$ subcooling
 - maintain adequate heat sink, S/G level \geq normal range

TWO-PHASE FLUID FLOW

Water at saturation conditions may exist as both a fluid and a vapor. This mixture of steam and water can cause unusual flow characteristics within fluid systems.

- EO 1.31** **DEFINE** two-phase flow.
- EO 1.32** **DESCRIBE** two-phase flow including such phenomena as bubbly, slug, and annular flow.
- EO 1.33** **DESCRIBE** the problems associated with core flow oscillations and flow instability.
- EO 1.34** **DESCRIBE** the conditions that could lead to core flow oscillation and instability.
- EO 1.35** **DESCRIBE** the phenomenon of pipe whip.
- EO 1.36** **DESCRIBE** the phenomenon of water hammer.
-

Two-Phase Fluid Flow

All of the fluid flow relationships discussed previously are for the flow of a single phase of fluid whether liquid or vapor. At certain important locations in fluid flow systems the simultaneous flow of liquid water and steam occurs, known as *two-phase flow*. These simple relationships used for analyzing single-phase flow are insufficient for analyzing two-phase flow.

There are several techniques used to predict the head loss due to fluid friction for two-phase flow. Two-phase flow friction is greater than single-phase friction for the same conduit dimensions and mass flow rate. The difference appears to be a function of the type of flow and results from increased flow speeds. Two-phase friction losses are experimentally determined by measuring pressure drops across different piping elements. The two-phase losses are generally related to single-phase losses through the same elements.

One accepted technique for determining the two-phase friction loss based on the single-phase loss involves the two-phase friction multiplier (R), which is defined as the ratio of the two-phase head loss divided by the head loss evaluated using saturated liquid properties.

$$R = \frac{H_f, \text{ two-phase}}{H_f, \text{ saturated liquid}} \quad (3-18)$$

where:

R = two-phase friction multiplier (no units)

H_f , two-phase = two-phase head loss due to friction (ft)

H_f , saturated liquid = single-phase head loss due to friction (ft)

The friction multiplier (R) has been found to be much higher at lower pressures than at higher pressures. The two-phase head loss can be many times greater than the single-phase head loss.

Although a wide range of names has been used for two-phase flow patterns, we shall define only three types of flow. The flow patterns to be used are defined as follows:

1. Bubbly flow: there is dispersion of vapor bubbles in a continuum of liquid.
2. Slug flow: in bubbly flow, the bubbles grow by coalescence and ultimately become of the same order of diameter as the tube. This generates the typical bullet-shaped bubbles that are characteristic of the slug-flow regime.
3. Annular flow: the liquid is now distributed between a liquid film flowing up the wall and a dispersion of droplets flowing in the vapor core of the flow.

Flow Instability

Unstable flow can occur in the form of flow oscillations or flow reversals. *Flow oscillations* are variations in flow due to void formations or mechanical obstructions from design and manufacturing. A flow oscillation in one reactor coolant channel sometimes causes flow oscillations in the surrounding coolant channels due to flow redistribution. Flow oscillations are undesirable for several reasons. First, sustained flow oscillations can cause undesirable forced mechanical vibration of components. This can lead to failure of those components due to fatigue. Second, flow oscillations can cause system control problems of particular importance in liquid-cooled nuclear reactors because the coolant is also used as the moderator. Third, flow oscillations affect the local heat transfer characteristics and boiling. It has been found through testing that the critical heat flux (CHF) required for departure from nucleate boiling (DNB) can be lowered by as much as 40% when flow is oscillating. This severely reduces the thermal limit and the power density along the length of the reactor core. Again, it has been found through testing that flow oscillations are not a significant problem for some pressurized water reactors unless power is above 150% for the normal flow conditions. Flow oscillations can be a problem during natural circulation operations because of the low flow rates present.

During natural circulation, the steam bubbles formed during a flow oscillation may have enough of an effect to actually cause complete flow reversal in the affected channel.

Both the flow oscillations and flow reversals lead to a very unstable condition since the steam blankets formed on heated surfaces directly affect the ability to transfer heat away from those surfaces.

Pipe Whip

If a pipe were to rupture, the reaction force created by the high velocity fluid jet could cause the piping to displace and cause extensive damage to components, instrumentation, and equipment in the area of the rupture. This characteristic is similar to an unattended garden hose or fire hose "whipping" about unpredictably. This type of failure is analyzed to minimize damage if pipe whip were to occur in the vicinity of safety-related equipment.

Water Hammer

Water hammer is a liquid shock wave resulting from the sudden starting or stopping of flow. It is affected by the initial system pressure, the density of the fluid, the speed of sound in the fluid, the elasticity of the fluid and pipe, the change in velocity of the fluid, the diameter and thickness of the pipe, and the valve operating time.

During the closing of a valve, kinetic energy of the moving fluid is converted into potential energy. Elasticity of the fluid and pipe wall produces a wave of positive pressure back toward the fluid's source. When this wave reaches the source, the mass of fluid will be at rest, but under tremendous pressure. The compressed liquid and stretched pipe walls will now start to release the liquid in the pipe back to the source and return to the static pressure of the source. This release of energy will form another pressure wave back to the valve. When this shockwave reaches the valve, due to the momentum of the fluid, the pipe wall will begin to contract. This contraction is transmitted back to the source, which places the pressure in the piping below that of the static pressure of the source. These pressure waves will travel back and forth several times until the fluid friction dampens the alternating pressure waves to the static pressure of the source. Normally, the entire hammer process takes place in under one second.

The initial shock of suddenly stopped flow can induce transient pressure changes that exceed the static pressure. If the valve is closed slowly, the loss of kinetic energy is gradual. If it is closed quickly, the loss of kinetic energy is very rapid. A shock wave results because of this rapid loss of kinetic energy. The shock wave caused by water hammer can be of sufficient magnitude to cause physical damage to piping, equipment, and personnel. Water hammer in pipes has been known to pull pipe supports from their mounts, rupture piping, and cause pipe whip.

Pressure Spike

A *pressure spike* is the resulting rapid rise in pressure above static pressure caused by water hammer. The highest pressure spike attained will be at the instant the flow changed and is governed by the following equation.

$$\Delta P = \frac{\rho_c \Delta v}{g_c}$$

where:

$$\Delta P = \text{Pressure spike} \left(\frac{\text{lbf}}{\text{ft}^2} \right)$$

$$\rho = \text{Density of the fluid} \left(\frac{\text{lbf}}{\text{ft}^3} \right)$$

$$c = \text{Velocity of the pressure wave} \\ \text{(Speed of sound in the fluid)} \left(\frac{\text{ft}}{\text{sec}} \right)$$

$$\Delta v = \text{Change in velocity of the fluid} \left(\frac{\text{ft}}{\text{sec}} \right)$$

$$g_c = \text{Gravitational constant } 32.17 \left(\frac{\text{lbf-ft}}{\text{lbf-sec}^2} \right)$$

Example: Pressure spike

Water at a density of 62.4 lbf/ft³ and a pressure of 120 psi is flowing through a pipe at 10 ft/sec. The speed of sound in the water is 4780 ft/sec. A check valve suddenly closed. What is the maximum pressure of the fluid in psi?

Solution:

$$P_{\text{Max}} = P_{\text{Static}} + \Delta P_{\text{Spike}}$$

$$P_{\text{Max}} = 120 \frac{\text{lbf}}{\text{in}^2} + \frac{\rho_c \Delta V}{g_c}$$

$$P_{\text{Max}} = 120 \frac{\text{lbf}}{\text{in}^2} + \frac{62.4 \frac{\text{lbf}}{\text{ft}^3} \cdot 4780 \frac{\text{ft}}{\text{sec}} \cdot 10 \frac{\text{ft}}{\text{sec}}}{32.17 \frac{\text{lbf ft}}{\text{lbf sec}^2}}$$

$$P_{\text{Max}} = 120 \frac{\text{lbf}}{\text{in}^2} + 92,631 \frac{\text{lbf}}{\text{ft}^2} \left(\frac{\text{ft}^2}{144 \text{ in}^2} \right)$$

$$P_{\text{Max}} = 120 \frac{\text{lbf}}{\text{in}^2} + 643.3 \frac{\text{lbf}}{\text{in}^2}$$

$$P_{\text{Max}} = 763.3 \text{ psi}$$

Steam Hammer

Steam hammer is similar to water hammer except it is for a steam system. *Steam hammer* is a gaseous shock wave resulting from the sudden starting or stopping of flow. Steam hammer is not as severe as water hammer for three reasons:

1. The compressibility of the steam dampens the shock wave
2. The speed of sound in steam is approximately one third the speed of sound in water.
3. The density of steam is approximately 1600 times less than that of water.

The items of concern that deal with steam piping are thermal shock and water slugs (i.e., condensation in the steam system) as a result of improper warm up.

Operational Considerations

Water and steam hammer are not uncommon occurrences in industrial plants. Flow changes in piping systems should be done slowly as part of good operator practice. To prevent water and steam hammer, operators should ensure liquid systems are properly vented and ensure gaseous or steam systems are properly drained during start-up. When possible, initiate pump starts against a closed discharge valve, and open the discharge valve slowly to initiate system flow. If possible, start-up smaller capacity pumps before larger capacity pumps. Use warm-up valves around main stream stop valves whenever possible. If possible, close pump discharge valves before stopping pumps. Periodically verify proper function of moisture traps and air traps during operation.

Summary

The main points from this chapter are summarized below.

Two-Phase Fluid Flow Summary

- The combination of liquid and vapor flowing through a pipe is called two-phase flow.
- Types of two-phase flow include:
 - Bubbly flow: there is a dispersion of vapor bubbles in a continuum of liquid.
 - Slug flow: the bubbles grow by coalescence and ultimately become of the same order of diameter as the tube, generating bullet shaped bubbles.
 - Annular flow: the liquid is distributed between a liquid film flowing up the wall and a dispersion of droplets flowing in the vapor core of the flow.
- Core flow oscillations and instabilities can cause:
 - undesirable mechanical vibration of components.
 - a reduction in the heat flux required to cause DNB.
 - interruptions to actual circulation flow.
- Flow oscillations and instabilities can occur during the following conditions:
 - core is outside design conditions, power > 150%
 - mechanical failure, causing flow blockage
 - inadequate core cooling during natural circulation, such that boiling is occurring
- Pipe whip is the displacement of piping created by the reaction forces of a high velocity fluid jet following a pipe rupture.
- Water hammer is a liquid shock wave resulting from a sudden starting or stopping of flow.

CENTRIFUGAL PUMPS

Centrifugal pumps are one of the most common components found in fluid systems. In order to understand how a fluid system containing a centrifugal pump operates, it is necessary to understand the head and flow relationships for a centrifugal pump.

- EO 1.37** **DEFINE** the terms net positive suction head and cavitation.
- EO 1.38** **CALCULATE** the new volumetric flow rate, head, or power for a variable speed centrifugal pump using the pump laws.
- EO 1.39** **DESCRIBE** the effect on system flow and pump head for the following changes:
- a. Changing pump speeds
 - b. Adding pumps in parallel
 - c. Adding pumps in series
-

Energy Conversion in a Centrifugal Pump

Fluid entering a centrifugal pump is immediately directed to the low pressure area at the center or eye of the impeller. As the impeller and blading rotate, they transfer momentum to incoming fluid. A transfer of momentum to the moving fluid increases the fluid's velocity. As the fluid's velocity increases its kinetic energy increases. Fluid of high kinetic energy is forced out of the impeller area and enters the volute.

The volute is a region of continuously increasing cross-sectional area designed to convert the kinetic energy of the fluid into fluid pressure. The mechanism of this energy conversion is the same as that for subsonic flow through the diverging section of a nozzle. The mathematical analysis of flow through the volute is based on the general energy equation, the continuity equation, and the equation relating the internal properties of a system. The key parameters influencing the energy conversion are the expanding cross-sectional area of the volute, the higher system back pressure at the discharge of the volute, and the incompressible, subsonic flow of the fluid. As a result of the interdependence of these parameters, the fluid flow in the volute, similar to subsonic flow in a diverging nozzle, experiences a velocity decrease and a pressure increase.

Operating Characteristics of a Centrifugal Pump

Normally, a centrifugal pump produces a relatively low pressure increase in the fluid. This pressure increase can be anywhere from several dozen to several hundred psid across a centrifugal pump with a single stage impeller. The term PSID (**P**ounds **F**orce **P**er **S**quare **I**nch **D**ifferential) is equivalent to ΔP . In this context, it is the pressure difference between the suction and discharge of a pump. PSID can also be used to describe a pressure drop across a system component (strainers, filters, heat exchangers, valves, demineralizers, etc.). When a centrifugal pump is operating at a constant speed, an increase in the system back pressure on the flowing stream causes a reduction in the magnitude of volumetric flow rate that the centrifugal pump can maintain.

Analysis of the relationship between the volumetric flow rate (\dot{V}) that a centrifugal pump can maintain and the pressure differential across the pump (ΔP_{pump}) is based on various physical characteristics of the pump and the system fluid. Variables evaluated by design engineers to determine this relationship include the pump efficiency, the power supplied to the pump, the rotational speed, the diameter of the impeller and blading, the fluid density, and the fluid viscosity. The result of this complicated analysis for a typical centrifugal pump operating at one particular speed is illustrated by the graph in Figure 7.

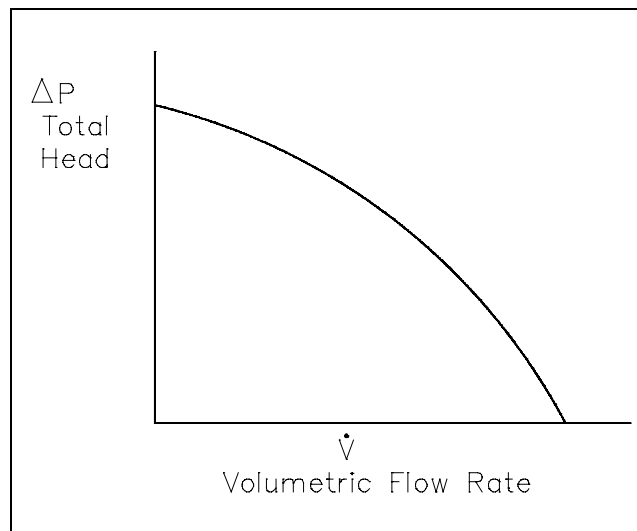


Figure 7 Typical Centrifugal Pump Characteristic Curve

Pump head, on the vertical axis, is the difference between system back pressure and the inlet pressure of the pump (ΔP_{pump}). Volumetric flow rate (\dot{V}), on the horizontal axis, is the rate at which fluid is flowing through the pump. The graph assumes one particular speed (N) for the pump impeller.

Cavitation

When the liquid being pumped enters the eye of a centrifugal pump, the pressure is significantly reduced. The greater the flow velocity through the pump the greater this pressure drop. If the pressure drop is great enough, or if the temperature of the liquid is high enough, the pressure drop may be sufficient to cause the liquid to flash to steam when the local pressure falls below the saturation pressure for the fluid that is being pumped. These vapor bubbles are swept along the pump impeller with the fluid. As the flow velocity decreases the fluid pressure increases. This causes the vapor bubbles to suddenly collapse on the outer portions of the impeller. The formation of these vapor bubbles and their subsequent collapse is *cavitation*.

Cavitation can be a very serious problem for centrifugal pumps. Some pumps can be designed to operate with limited amounts of cavitation. Most centrifugal pumps cannot withstand cavitation for significant periods of time; they are damaged by erosion of the impeller, vibration, or some other cavitation-induced problem.

Net Positive Suction Head

It is possible to ensure that cavitation is avoided during pump operation by monitoring the net positive suction head of the pump. *Net positive suction head* (NPSH) for a pump is the difference between the suction pressure and the saturation pressure of the fluid being pumped. NPSH is used to measure how close a fluid is to saturated conditions. Equation 3-19 can be used to calculate the net positive suction head available for a pump. The units of NPSH are feet of water.

$$\text{NPSH} = P_{\text{suction}} - P_{\text{saturation}} \quad (3-19)$$

where:

$$\begin{aligned} P_{\text{suction}} &= \text{suction pressure of the pump} \\ P_{\text{saturation}} &= \text{saturation pressure for the fluid} \end{aligned}$$

By maintaining the available NPSH at a level greater than the NPSH required by the pump manufacturer, cavitation can be avoided.

Pump Laws

Centrifugal pumps generally obey what are known as the pump laws. These laws state that the flow rate or capacity is directly proportional to the pump speed; the discharge head is directly proportional to the square of the pump speed; and the power required by the pump motor is directly proportional to the cube of the pump speed. These laws are summarized in the following equations.

$$\dot{V} \propto n \quad (3-20)$$

$$H_p \propto n^2 \quad (3-21)$$

$$p \propto n^3 \quad (3-22)$$

where:

n = speed of pump impeller (rpm)

\dot{V} = volumetric flow rate of pump (gpm or ft³/hr)

H_p = head developed by pump (psid or feet)

p = pump power (kW)

Using these proportionalities, it is possible to develop equations relating the condition at one speed to those at a different speed.

$$\dot{V}_1 \left(\frac{n_2}{n_1} \right) = \dot{V}_2 \quad (3-23)$$

$$H_{p_1} \left(\frac{n_2}{n_1} \right)^2 = H_{p_2} \quad (3-24)$$

$$p_1 \left(\frac{n_2}{n_1} \right)^3 = p_2 \quad (3-25)$$

Example: Pump Laws

A cooling water pump is operating at a speed of 1800 rpm. Its flow rate is 400 gpm at a head of 48 ft. The power of the pump is 45 kW. Determine the pump flow rate, head, and power requirements if the pump speed is increased to 3600 rpm.

Solution:

Flow rate

$$\begin{aligned} \dot{V}_2 &= \dot{V}_1 \left(\frac{n_2}{n_1} \right) \\ &= (400 \text{ gpm}) \left(\frac{3600 \text{ rpm}}{1800 \text{ rpm}} \right) \\ &= 800 \text{ gpm} \end{aligned}$$

Head

$$\begin{aligned} H_{P_2} &= H_{P_1} \left(\frac{n_2}{n_1} \right)^2 \\ &= 48 \text{ ft} \left(\frac{3600 \text{ rpm}}{1800 \text{ rpm}} \right)^2 \\ &= 192 \text{ ft} \end{aligned}$$

Power

$$\begin{aligned} P_2 &= P_1 \left(\frac{n_2}{n_1} \right)^3 \\ &= 45 \text{ kW} \left(\frac{3600 \text{ rpm}}{1800 \text{ rpm}} \right)^3 \\ &= 360 \text{ kW} \end{aligned}$$

It is possible to develop the characteristic curve for the new speed of a pump based on the curve for its original speed. The technique is to take several points on the original curve and apply the pump laws to determine the new head and flow at the new speed. The pump head versus flow rate curve that results from a change in pump speed is graphically illustrated in Figure 8.

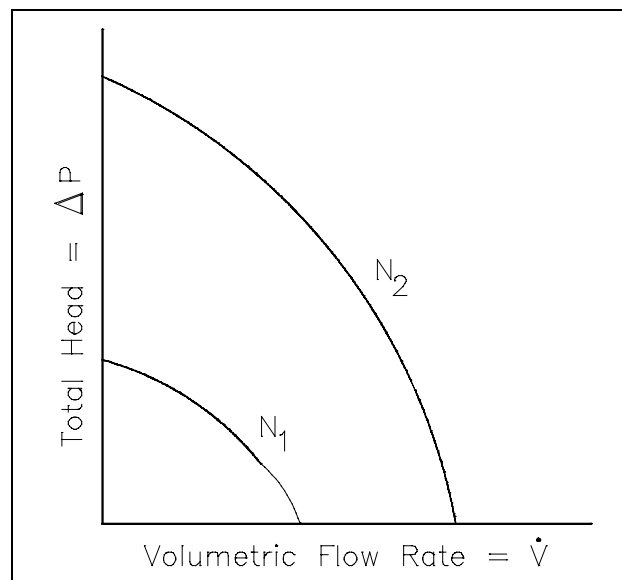


Figure 8 Changing Speeds for Centrifugal Pump

System Characteristic Curve

In the chapter on head loss, it was determined that both frictional losses and minor losses in piping systems were proportional to the square of the flow velocity. Since flow velocity is directly proportional to the volumetric flow rate, the system head loss must be directly proportional to the square of the volumetric flow rate. From this relationship, it is possible to develop a curve of system head loss versus volumetric flow rate. The head loss curve for a typical piping system is in the shape of a parabola as shown in Figure 9.

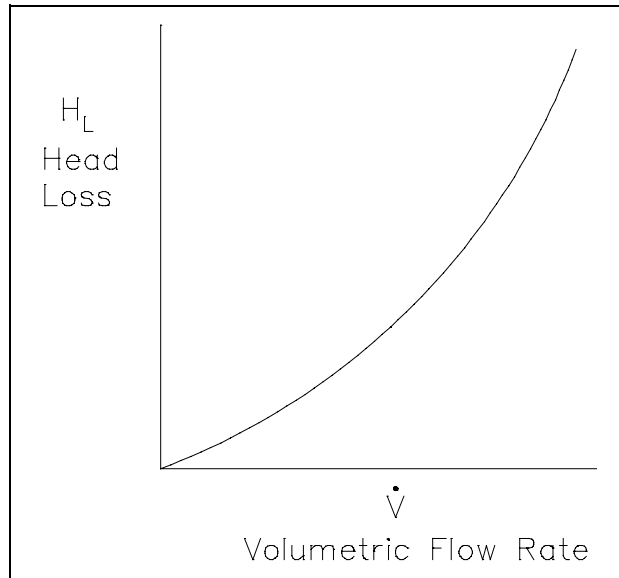


Figure 9 Typical System Head Loss Curve

System Operating Point

The point at which a pump operates in a given piping system depends on the flow rate and head loss of that system. For a given system, volumetric flow rate is compared to system head loss on a system characteristic curve. By graphing a system characteristic curve and the pump characteristic curve on the same coordinate system, the point at which the pump must operate is identified. For example, in Figure 10, the operating point for the centrifugal pump in the original system is designated by the intersection of the pump curve and the system curve (h_{L_0}).

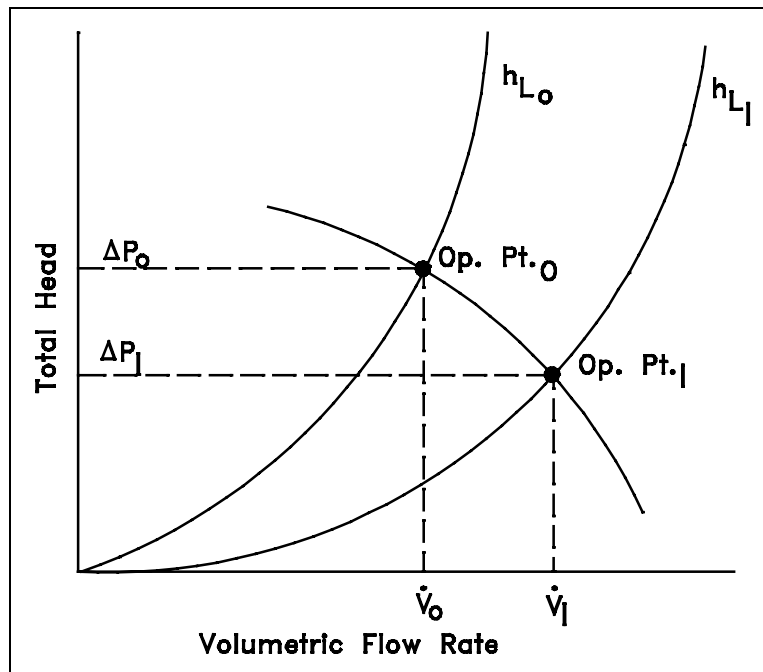


Figure 10 Operating Point for a Centrifugal Pump

The system has a flow rate equal to \dot{V}_0 and a total system head loss equal to ΔP_0 . In order to maintain the flow rate (\dot{V}_0), the pump head must be equal to ΔP_0 . In the system described by the system curve (h_{L1}), a valve has been opened in the system to reduce the system's resistance to flow. For this system, the pump maintains a large flow rate (\dot{V}_1) at a smaller pump head (ΔP_1).

System Use of Multiple Centrifugal Pumps

A typical centrifugal pump has a relatively low number of moving parts and can be easily adapted to a variety of prime movers. These prime movers include AC and DC electric motors, diesel engines, steam turbines, and air motors. Centrifugal pumps are typically small in size and can usually be built for a relatively low cost. In addition, centrifugal pumps provide a high volumetric flow rate with a relatively low pressure.

In order to increase the volumetric flow rate in a system or to compensate for large flow resistances, centrifugal pumps are often used in parallel or in series. Figure 11 depicts two identical centrifugal pumps operating at the same speed in parallel.

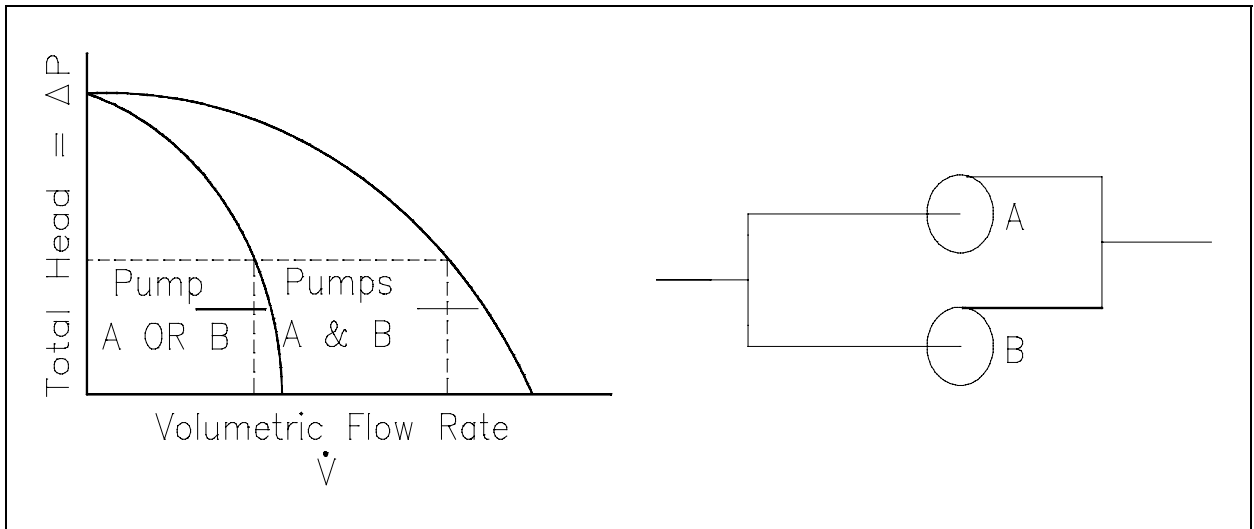


Figure 11 Pump Characteristic Curve for Two Identical Centrifugal Pumps Used in Parallel

Centrifugal Pumps in Parallel

Since the inlet and the outlet of each pump shown in Figure 11 are at identical points in the system, each pump must produce the same pump head. The total flow rate in the system, however, is the sum of the individual flow rates for each pump.

When the system characteristic curve is considered with the curve for pumps in parallel, the operating point at the intersection of the two curves represents a higher volumetric flow rate than for a single pump and a greater system head loss. As shown in Figure 12, a greater system head loss occurs with the increased fluid velocity resulting from the increased volumetric flow rate. Because of the greater system head, the volumetric flow rate is actually less than twice the flow rate achieved by using a single pump.

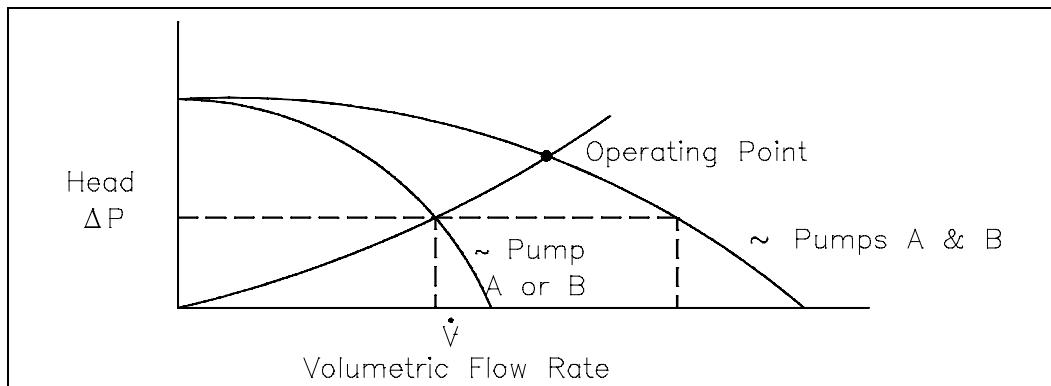


Figure 12 Operating Point for Two Parallel Centrifugal Pumps

Centrifugal Pumps in Series

Centrifugal pumps are used in series to overcome a larger system head loss than one pump can compensate for individually. As illustrated in Figure 13, two identical centrifugal pumps operating at the same speed with the same volumetric flow rate contribute the same pump head. Since the inlet to the second pump is the outlet of the first pump, the head produced by both pumps is the sum of the individual heads. The volumetric flow rate from the inlet of the first pump to the outlet of the second remains the same.

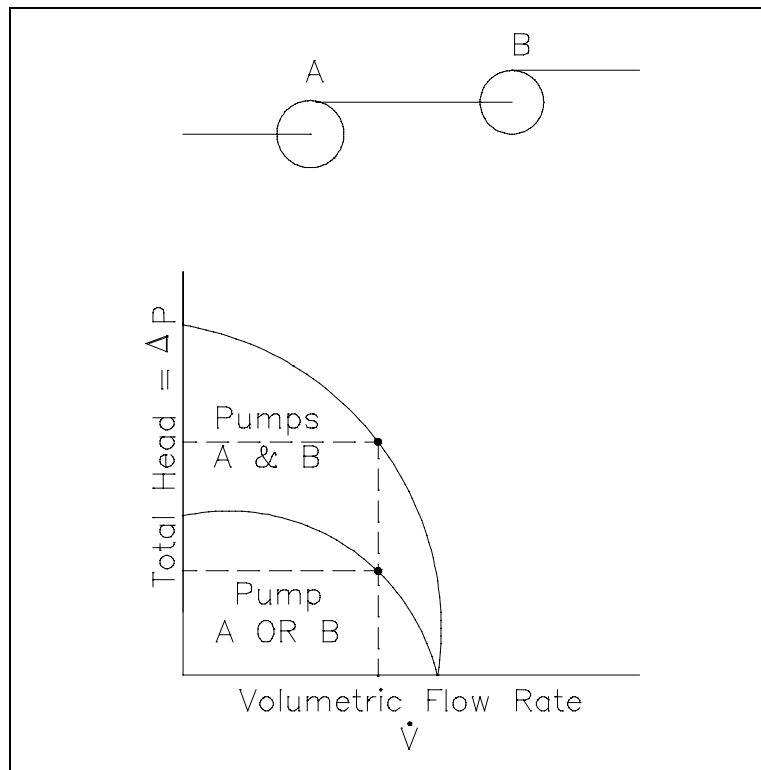


Figure 13 Pump Characteristic Curve for Two Identical Centrifugal Pumps Used in Series

As shown in Figure 14, using two pumps in series does not actually double the resistance to flow in the system. The two pumps provide adequate pump head for the new system and also maintain a slightly higher volumetric flow rate.

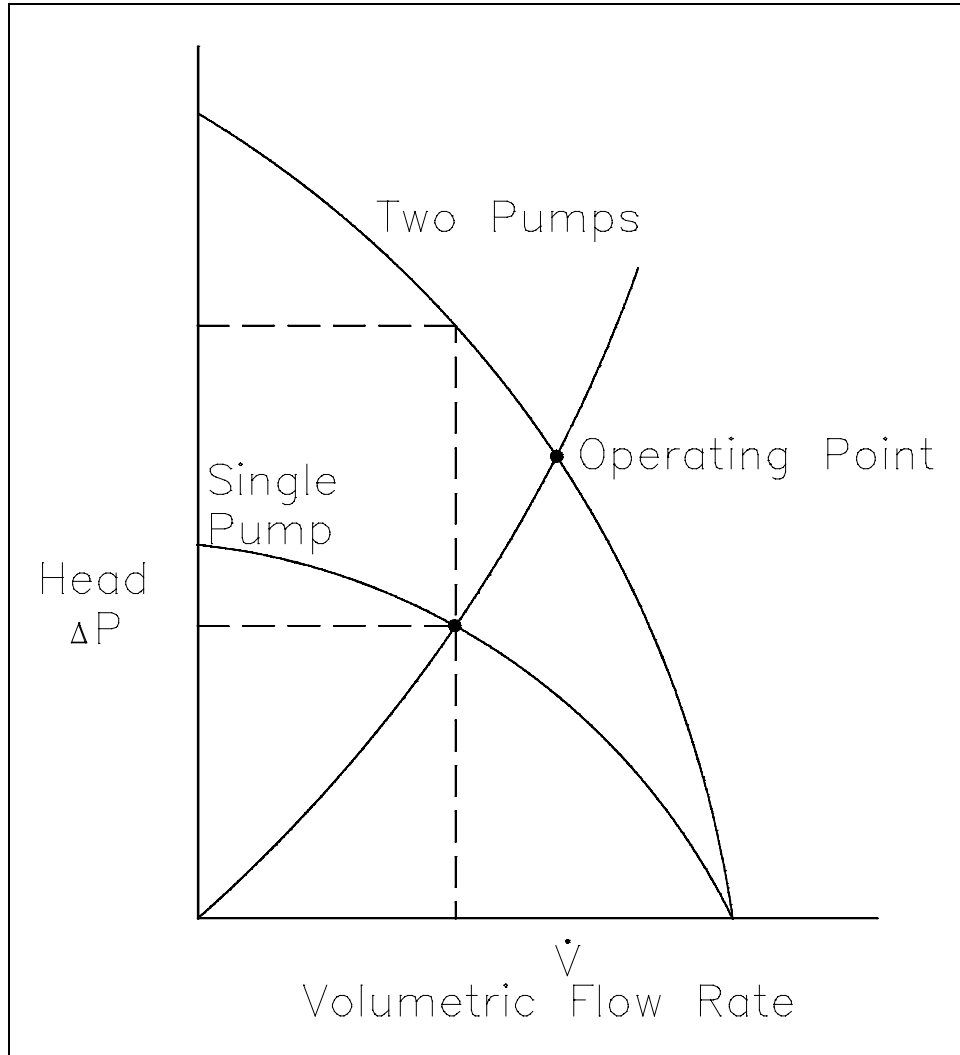


Figure 14 Operating Point for Two Centrifugal Pumps in Series

Summary

The main points from this chapter are summarized below.

Centrifugal Pumps Summary

- Net positive suction head is the difference between the pump suction pressure and the saturation pressure for the fluid.
- Cavitation is the formation and subsequent collapse of vapor bubbles on the impeller of a pump as the local pressure falls below and then rises above the saturation pressure of the fluid being pumped.
- The pump laws can be used to determine the effect of varying the speed of a centrifugal pump on the flow, head, and power.

$$\dot{V}_1 \left(\frac{n_2}{n_1} \right) = \dot{V}_2$$

$$H_{P_1} \left(\frac{n_2}{n_1} \right)^2 = H_{P_2}$$

$$P_1 \left(\frac{n_2}{n_1} \right)^3 = P_2$$

- The combined pump curve for two centrifugal pumps in parallel can be determined by adding the individual flows for any given head.
- The combined pump curve for two centrifugal pumps in series can be determined by adding the individual heads for any given flow.
- The operating point (head and flow) of a system can be determined by plotting the pump curve and the system head loss curve on the same axes. The system will operate at the intersection of the two curves.

Appendix B
Fluid Flow

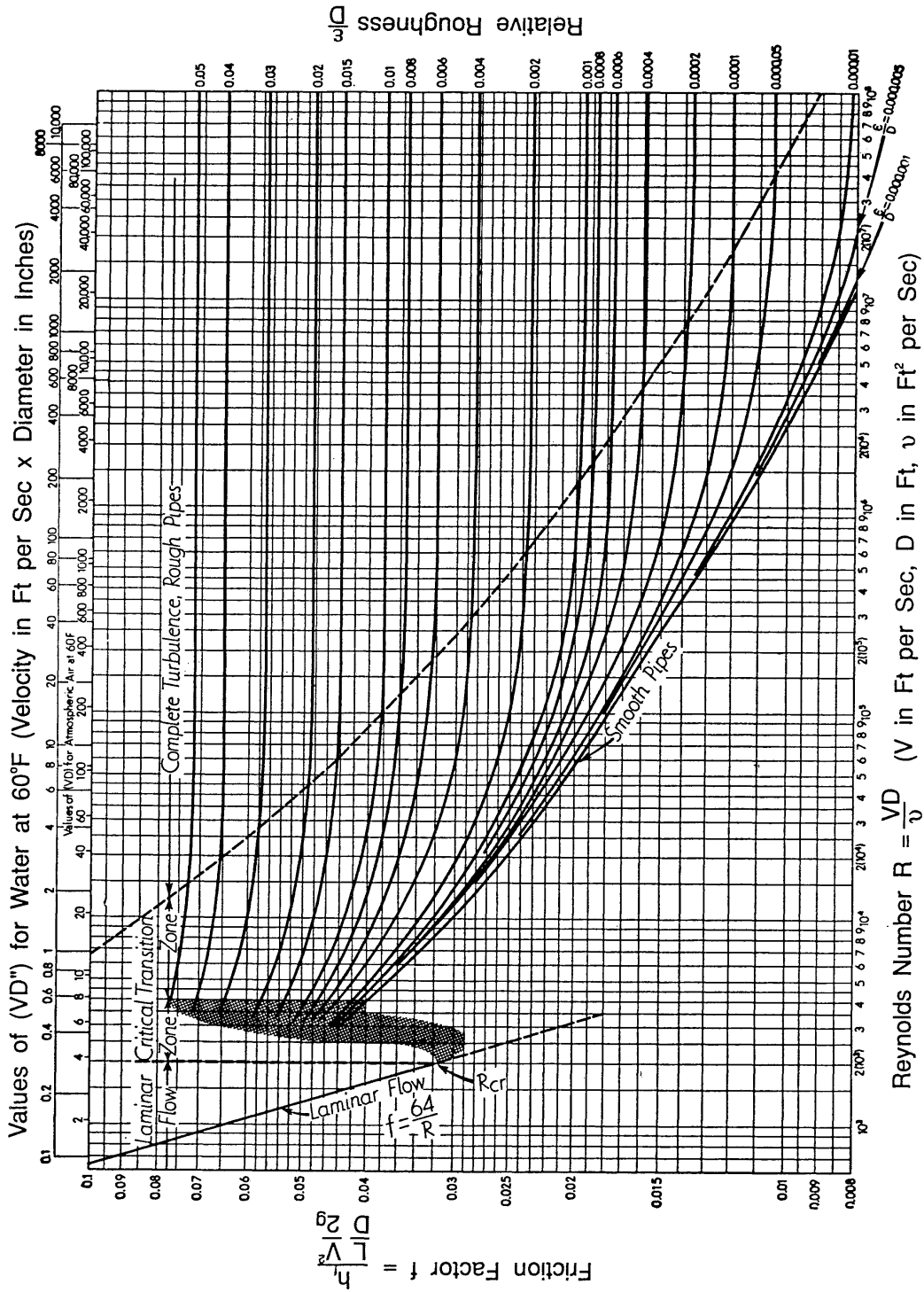


Figure B-1 Moody Chart

end of text.

CONCLUDING MATERIAL

Review activities:

Preparing activity:

DOE - ANL-W, BNL, EG&G Idaho,
EG&G Mound, EG&G Rocky Flats,
LLNL, LANL, MMES, ORAU, REEC_o,
WHC, WINCO, WEMCO, and WSRC.

DOE - NE-73
Project Number 6910-0018/3