CHAPTER

ONE

A THERMODYNAMICS REVIEW

1-1 INTRODUCTION

The design, operation, and performance of electric-generating powerplants are largely dependent upon the science of thermodynamics. It is assumed that the reader is familiar with the basics of that science. The material in this chapter, then, will not [illegible] a comprehensive treatment of thermodynamics but will instead review some fundamentals and present some aspects that are of particular relevance to power-generation systems.

The material in this chapter will include parts of both basic and applied thermodynamics. The first and second laws will be reviewed as well as processes, property relations, and cycles. The concepts of reversibility and entropy and the role they play in the performance of power-generating cycles will be discussed.

Chapter 2 will be devoted to the analysis of the Rankine cycle, which plays the major role in power systems today. Other cycles and aspects, such as the Brayton gas cycle, heat transfer, and fluid flow in powerplant components, will be presented where appropriate and applicable throughout the book.

1-2 THE FIRST LAW AND THE OPEN SYSTEM

The first law of thermodynamics is the law of *conservation of energy*, which states that energy can neither be created nor destroyed. The energy of a system undergoing change (process) can be increased or decreased by exchange with the surroundings and converted from one form to another within that system. This is, therefore, simply a "bookkeeping" law that declares the exchange and convertibility of energy and sees to it that all energy is accounted for when a change occurs. The first law does *not* indicate whether conversions of energy from one form to another are or are not performed perfectly or whether some forms may be completely converted to others. Such limitations are left to the second law.

The most general mathematical expression of the first law is that for an open system undergoing change in a transient state. A *system* is a specified region, not necessarily of constant volume or fixed boundaries, where transfers and conversions of energy and mass are to be studied. An *open system* is one where energy and mass cross the boundaries of that system. An *open system in the transient state* is one in which the mass inflow and outflow are not equal or vary with time and in which the mass within the system changes with time. The transient system will not be covered here, although an example (a pressurizer in a nuclear pressurized-water reactor) will be presented in Chap. 10.

We will now discuss the *steady-state open system*, also called the *steady-state steady-flow (SSSF) system*. This is one in which mass and energy flows across the boundaries do *not* vary with time and in which the mass within the system remains constant. The SSSF system, applicable to mechanical energy (i.e., ignoring electrical, magnetic, chemical, and other effects) is shown schematically in Fig. 1-1. The first-law equation for that system is

$$\begin{aligned} \mathrm{PE}_1 + \mathrm{KE}_1 + \mathrm{IE}_1 + \mathrm{FE}_1 + \Delta Q \\ = \mathrm{PE}_2 + \mathrm{KE}_2 + \mathrm{IE}_2 + \mathrm{FE}_2 + \Delta W_{sf} \qquad (1\text{-}1a) \end{aligned}$$

where the subscripts 1 and 2 indicate the inlet and exit stations of the open system. Equation (1-1*a*) assumes for simplicity that only one inlet and one exit exist, although the SSSF equation can be written easily enough for multiple inlets, exits, or both. In Eq. (1-1*a*)

$$\mathrm{PE} = \text{potential energy} = mz\frac{g}{g_c} \qquad (1\text{-}2)$$

where m is the mass of a quantity of matter or fluid entering and leaving the system (both equal in SSSF), z is the elevation of stations 1 or 2 above a common datum, g is the gravitational acceleration, and g_c is a conversion factor numerically equal to 32.2 $\mathrm{lb}_m \cdot \mathrm{ft/(lb}_f \cdot \mathrm{s}^2)$ or 1.0 $\mathrm{kg \cdot m/(N \cdot s^2)}$

$$\mathrm{KE} = \text{kinetic energy} = m\frac{V_i^2}{2g_c} \qquad (1\text{-}3)$$

where V_i is the velocity of the mass at 1 or 2.

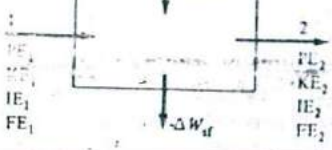

Figure 1-1 Schematic of a steady-state steady-flow (SSSF) system with one inlet and one outlet.

$$\mathrm{IE} = \text{internal energy} = U \tag{1-4}$$

Internal energy is a sole function of temperature for perfect gases and a strong function of temperature and weak function of pressure for nonperfect gases, vapors, and liquids. It is a measure of the internal (molecular) activity and interaction of the fluid.

$$\mathrm{FE} = \text{flow energy} = PV = Pmv \tag{1-5}$$

Flow energy, or flow work, is the work done by the flowing fluid to push the quantity represented by mass m into, and out of, the system. Mathematically it is equal to the product of pressure P and volume V.

$$\Delta Q = \text{net heat added} = Q_A - |Q_R| \tag{1-6}$$

where Q_A is heat *added* to (entering) and Q_R is the heat *rejected* by (leaving) the system across its boundaries. It is convenient to consider that heat added is positive and heat rejected is negative. Mathematically

$$\Delta Q = mc_n(T_2 - T_1) \tag{1-7}$$

where c_n is a specific heat that depends upon the process taking place between 1 and 2. Values for c_n for various processes are given in Table 1-1.

$$\begin{aligned} \Delta W_{sf} &= \text{net steady-flow mechanical work done by the system} \\ &= W_{by} - |W_{on}| \end{aligned} \tag{1-8}$$

where W_{by} is the work done by the system and W_{on} is the work done on the system. The convention is that the work done by the system is positive and the work done on the system is negative. Mathematically the steady-flow work is given by

$$\Delta W_{sf} = -\int_1^2 V\,dP \tag{1-9}$$

Equation (1-9) requires a relationship between pressure P and volume V for evaluation. The most general relationship is given by

$$PV^n = \text{constant} \tag{1-10}$$

where n is called the *polytropic exponent* and varies from zero to infinity. Its value for certain processes is given in Table 1-1.

Table 1-1 Values of c_n and n for various processes

Process	c_n	n
Constant pressure	c_p	0
Constant temperature	∞	1
Adiabatic reversible	0	$k = \dfrac{c_p}{c_v}$
Constant volume	c_v	∞
Polytropic	$c_v \dfrac{k - n}{1 - n}$	$0 - \infty$

Equation (1-1a) may now be written for mass m entering and leaving the system as

$$mz_1\frac{g}{g_c} + m\frac{V_{s1}^2}{2g_c} + U_1 + P_1V_1 + \Delta Q$$
$$= mz_2\frac{g}{g_c} + m\frac{V_{s2}^2}{2g_c} + U_2 + P_2V_2 + \Delta W_{sf} \qquad (1\text{-}1b)$$

and for a unit mass

$$z_1\frac{g}{g_c} + \frac{V_{s1}^2}{2g_c} + u_1 + P_1v_1 + \Delta q$$
$$= z_2\frac{g}{g_c} + \frac{V_{s2}^2}{2g_c} + u_2 + P_2v_2 + \Delta w_{sf} \qquad (1\text{-}1c)$$

where the lowercase symbols represent specific values of the uppercase ones, i.e., per unit mass. Thus u = specific internal energy = U/m, v = specific volume = V/m, etc.

Table 1-2 Some common thermodynamic symbols

c_p = specific heat at constant pressure, Btu/(lb$_m$ · °F) or J(kg · K)
c_v = specific heat at constant volume, Btu/(lb$_m$ · °F) or J(kg · K)
h = specific enthalpy, Btu lb$_m$ or J/kg
H = total enthalpy, Btu or J
J = energy conversion factor = 778.16 ft · lb/Btu or 1.0 N m/J
M = molecular mass, lb$_m$/lb·mol or kg/kg·mol
n = polytropic exponent, dimensionless
P = absolute pressure (gauge pressure + barometric pressure), lb$_f$/ft^2; unit may be lb$_f$/in^2, commonly written psia; or Pa
Q = heat transferred to or from system, Btu or J, or Btu/cycle or J/cycle
R = gas constant, lb$_f$ · ft/(lb$_m$ · °R) or J(kg · K) = $\overline{R}/M$
$\overline{R}$ = universal gas constant = 1.545.33, lb$_f$ · ft/(lb·mol · °R) or 8.31434 × 10^3 J/(kg·mol · K)
s = specific entropy, Btu/(lb$_m$ · °R) or J/(kg · K)
S = total entropy, Btu/°R or J/kg
t = temperature, °F or °C
T = temperature on absolute scale, °R or K
u = specific internal energy, Btu/lb$_m$ or J/kg
U = total internal energy, Btu or J
v = specific volume, ft^3/lb$_m$ or m^3/kg
V = total volume, ft^3 or m^3
W = work done by or on system, lb$_f$ · ft or J, or Btu/cycle or J/cycle
x = quality of a two-phase mixture = mass of vapor divided by total mass, dimensionless
k = ratio of specific heats, c_p/c_v, dimensionless
η = efficiency, as dimensionless fraction or percent

Subscripts used in vapor tables

f refers to saturated liquid
g refers to saturated vapor
fg refers to change in property because of change from saturated liquid to saturated vapor

The units of all the terms in Eqs. (1-1*a*) and (1-1*b*) are those of energy, such as Btu, ft · lb$_f$, or Joule (J). The units in Eq. (1-1*c*) are Btu/lb$_m$, ft · lb$_f$/lb$_m$ or J/kg. A list of these and other common thermodynamic symbols is given in Table 1-2.

The Enthalpy

The sums $U + PV$ and $u + Pv$ appear together very frequently in thermodynamics. The combination, therefore, has been given the name *enthalpy* (with the stress on the middle syllable) and the symbols H and h, where h = specific enthalpy = H/m. Thus

$$H = U + PV \tag{1-11a}$$

and

$$h = u + Pv \tag{1-11b}$$

Equation (1-1*b*) can now be written, using H, as

$$mz_1\frac{g}{g_c} + m\frac{V_{s1}^2}{2g_c} + H_1 + \Delta Q$$

$$= mz_2\frac{g}{g_c} + m\frac{V_{s2}^2}{2g_c} + H_2 + \Delta W_{sf} \tag{1-1d}$$

and Eq. (1-1*c*) can be similarly written using h.

Enthalpies and internal energy are *properties* of the fluid, which means that each would have a single value at any given state of the fluid. They are defined as

$$c_v \equiv \left(\frac{\partial u}{\partial T}\right)_v \tag{1-12}$$

and

$$c_p \equiv \left(\frac{\partial h}{\partial T}\right)_P \tag{1-13}$$

where c_p and c_v are the specific heats at constant pressure and constant volume, respectively. They have units of Btu/(lb$_m$ · °R) or J/(kg · K). They are related by

$$c_p - c_v = R \tag{1-14}$$

where R is the gas constant. For ideal gases

$$du = c_v\,dT \tag{1-15a}$$

and

$$dh = c_p\,dT \tag{1-16a}$$

where c_v and c_p are constant and independent of temperature for monatomic gases such as helium but increase with temperature for diatomic gases such as air and more so for triatomic gases such as CO$_2$, etc. For helium c_v = 0.753 Btu/(lb$_m$ · °R) = 3.153 kJ/(kg · K), c_p = 1.250 Btu/(lb$_m$ · °R) = 5.234 kJ/(kg · K), both independent of temperature. For air at low temperatures, c_v = 0.171 Btu/(lb$_m$ · °R) = 0.716 kJ/(kg · K), c_p = 0.240 Btu/(lb$_m$ · °R) = 1.005 kJ/(kg · K). For constant specific heats, or for small changes in temperature, Eqs. (1-15*a*) and (1-16*a*) may be written as

$$\Delta u = c_v\,\Delta T \tag{1-15b}$$

and
$$\Delta h = c_p \, \Delta T \tag{1-16b}$$

Where there is a large increase in temperature

$$\Delta u = \int_1^2 c_v(T)\, dT \tag{1-17}$$

and
$$\Delta h = \int_1^2 c_p(T)\, dT \tag{1-18}$$

Expressions for $c_p(T)$ in terms of temperatures for various gases (other than monatomic) may be found in the literature [4, 5]. $c_v(T)$ is found from $c_p(T)$ by subtracting the gas constant R for the particular gas, Eq. (1-14). For pure air

$$c_p(T) = 0.219 + 3.42 \times 10^{-5}T - 2.93 \times 10^{-9}T^2 \tag{1-19}$$

where $c_p(T)$ is in Btu/(lb$_m$ · °R) and T in °R. $c_v(T)$ for pure air is obtained by subtracting R for air or $53.34/778.16 = 0.0685$ Btu/(lb$_m$ · °R) from Eq. (1-19).

It should be added here that the combination $U + PV$ is enthalpy H whether the system is open or closed (below). In the open system PV is the flow energy. In the closed system PV is simply the product of pressure times volume.

Equations (1-1a) through (1-1d) are used to solve problems of open systems. In considering a particular problem it is often found that some terms drop out, some are unchanged between stations 1 and 2, or that the change is negligibly small compared with those of other terms. Some examples of open systems are

1. A steam generator

$$\Delta W_{sf} = 0$$
$$\mathrm{PE}_2 - \mathrm{PE}_1 = \text{negligible}$$
$$\mathrm{KE}_2 - \mathrm{KE}_1 = \text{negligible}$$

Thus
$$\Delta Q = H_2 - H_1 \tag{1-20a}$$
and
$$\Delta q = h_2 - h_1 \tag{1-20b}$$

2. A gas or steam turbine

$$\Delta Q = \text{negligible}$$
$$\mathrm{PE}_2 - \mathrm{PE}_1 = \text{negligible}$$
$$\mathrm{KE}_2 - \mathrm{KE}_1 = \text{negligible}$$

thus
$$\Delta W_{sf} = H_1 - H_2 \tag{1-21a}$$
and
$$\Delta w_{sf} = h_1 - h_2 \tag{1-21b}$$

3. A water (or incompressible fluid) pump

$$\Delta Q = \text{negligible}$$
$$\mathrm{PE}_2 - \mathrm{PE}_1 = 0 \qquad \text{(considering immediate inlet and exit)}$$

$$\mathrm{KE}_2 - \mathrm{KE}_1 = \text{negligible}$$

$$U_2 = U_1$$

And, because water is essentially incompressible and little or no change in temperature or volume takes place,

$$V_2 \approx V_1 = V$$

Thus $$\Delta W_{sf} = \mathrm{FE}_1 - \mathrm{FE}_2 = V(P_1 - P_2) \tag{1-22a}$$

and $$\Delta w_{sf} = v(P_1 - P_2) \tag{1-22b}$$

Both should be negative.

4. A nozzle

$$\Delta Q = 0$$

$$\Delta W_{sf} = 0$$

$$\mathrm{PE}_2 - \mathrm{PE}_1 = 0 \quad \text{(considering immediate inlet and exit)}$$

$$\mathrm{KE}_1 = \text{usually negligible compared with } \mathrm{KE}_2$$

Thus $$V_{s2} = \sqrt{2g_c J(h_1 - h_2)} \tag{1-23a}$$

$$= \sqrt{2g_c J c_p (T_1 - T_2)} \quad \text{ideal gas} \tag{1-23b}$$

$$= \sqrt{2g_c v(P_1 - P_2)} \quad \text{incompressible fluid} \tag{1-23c}$$

5. Throttling

$$\Delta Q = 0$$

$$\Delta W_{sf} = 0$$

$$\mathrm{PE}_2 - \mathrm{PE}_1 = 0 \quad \text{or negligible}$$

$$\mathrm{KE}_2 - \mathrm{KE}_1 = \text{negligible}$$

Thus $$H_1 = H_2 \tag{1-24a}$$

and $$h_1 = h_2 \tag{1-24b}$$

1-3 THE FIRST LAW AND THE CLOSED SYSTEM

The open system, discussed above, is one in which mass crosses the boundaries. A *closed system*, by contrast, is one in which only energy and not mass may cross the boundaries. A third system of some interest is the *isolated system*, a special instance of the closed system. It is one in which neither mass or energy cross the boundaries but in which energy transformations may take place within the boundaries.

Because mass does not cross the boundaries in a closed system, the potential, kinetic, and flow energy terms of Eqs. (1-1*a*) through (1-1*d*) drop out and the first-law equation for the closed system is simplified to

$$U_1 + \Delta Q = U_2 + \Delta W_{nf} \tag{1-25a}$$

or

$$\Delta Q = \Delta U + \Delta W_{nf} \tag{1-25b}$$

and

$$\Delta q = \Delta u + \Delta w_{nf} \tag{1-25c}$$

In the case of the open system, Eqs. (1-1*a*) through (1-1*d*), 1 and 2 refer to positions in space with respect to the system. In the case of the closed system they instead refer to differences in the time domain, i.e., before and after the process in question has taken place.

ΔW_{nf} is called the *nonflow work*. Mathematically it is given by

$$\Delta W_{nf} = \int_1^2 P\,dV \tag{1-26}$$

which, as does Eq. (1-9), requires a relationship between P and V for evaluation, such as that given by Eq. (1-10).

Some examples of the closed system are:

1. A closed rigid tank

$$\Delta W_{nf} = 0$$

Thus

$$\Delta Q = U_2 - U_1 = \Delta U \tag{1-27}$$

2. An insulated cylinder in which a fluid expands behind a piston (or is compressed by it)

$$\Delta Q = 0$$

Thus

$$\Delta W_{nf} = U_1 - U_2 = -\Delta U \tag{1-28}$$

1-4 THE CYCLE

In order to convert forms of energy, particularly heat, to work on an extended or continuous basis (our main objective), one needs to operate on a cycle. A *process* begins at one state of the working fluid and ends at another, and that is that. A *cycle*, on the other hand, is a series of processes that begins and ends at the same state and thus can repeat indefinitely, or as long as needed. An example is the ideal diesel cycle,[*] shown on the $P - V$ and $T - s$ diagrams in Fig. 1-2. It is composed of an

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[*] Rudolf Diesel (1858–1913) was born in Paris of German parents and moved to London in 1870 because of the Franco-German War. He was educated in Germany, where in 1893 he obtained a patent on the engine that bears his name. Originally his idea was to inject coal dust instead of liquid fuel into compressed air at high enough temperatures for the dust to ignite. He narrowly escaped death when his first attempt resulted in the engine's blowing up at the first injection of fuel. A successful engine was produced after some 4 years of tedious and costly work. Diesel disappeared and presumably drowned while crossing the English Channel during a storm.

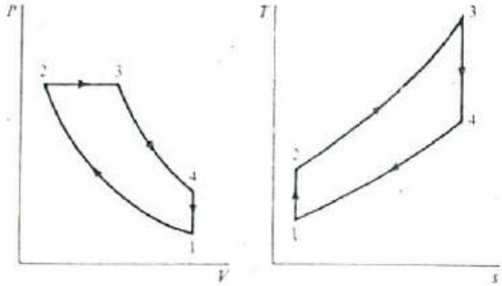


Figure 1-2 Pressure-volume and temperature-entropy diagrams of an ideal diesel cycle.

ideal and adiabatic (no heat exchanged) compression process 1-2, a constant-pressure heat addition process 2-3, an ideal and adiabatic expansion process 3-4, and a constant-volume heat rejection process 4-1, which returns the cycle back to 1. Because the beginning and the end of the cycle is 1 (or any other point), a thermodynamic cycle is a closed system where

$$\Delta U = U_1 - U_1 = 0$$

and the first law for this, and all other cycles, becomes

$$\Delta Q_{\text{net}} = Q_A - |Q_R| = \Delta W_{\text{net}} \qquad \text{(for a cycle)} \tag{1-29}$$

1-5 PROPERTY RELATIONSHIPS

Perfect Gases

Property relationships for perfect gases for different processes are given in Table 1-3. A *perfect (or ideal) gas* is one which, at any state, obeys the *equation of state for perfect gases*

$$PV = mRT \tag{1-30a}$$

$$Pv = RT \tag{1-30b}$$

$$pV = nR_oT \tag{1-30c}$$

where R = specific gas constant. Different gases have different values of R; for air $R = 53.3$ ft · lb$_f$/(lb$_m$ · °R), 286.8 J/(kg · K)

n = number of moles = m/M, where M is the molecular mass of the gas = 28.97 for air

R_o = universal gas constant = RM, the same for all perfect gases = 1545.33 ft · lb$_f$/(lb · mol · °R) = 8314.34 J/(kg · mol · K)

T = absolute temperature in degrees Rankine or Kelvin

Table 1-3 Perfect-gas relationships (constant specific heats)

Process	P, v, T relationships	$u_2 - u_1$	$h_2 - h_1$	$s_2 - s_1$	w (nonflow)	w (flow)	Q
Isothermal	T = constant $P_1/P_2 = v_2/v_1$	0	0	$(R/J)\ln(v_2/v_1)$	$(P_1v_1/J)\ln(v_2/v_1)$	$(P_1v_1/J)\ln(v_2/v_1)$	$(P_1v_1/J)\ln(v_2/v_1)$
Constant pressure	P = constant $T_2/T_1 = v_2/v_1$	$c_v(T_2 - T_1)$	$c_p(T_2 - T_1)$	$c_p\ln(T_2/T_1)$	$P(v_2 - v_1)/J$	0	$c_p(T_2 - T_1)$
Constant volume	v = constant $T_2/T_1 = P_2/P_1$	$c_v(T_2 - T_1)$	$c_p(T_2 - T_1)$	$c_v\ln(T_2/T_1)$	0	$v(P_1 - P_2)/J$	$c_v(T_2 - T_1)$
Isentropic (Adiabatic reversible)	s = constant $P_1v_1^k = P_2v_2^k$ $T_2/T_1 = (v_1/v_2)^{k-1}$ $T_2/T_1 = (P_2/P_1)^{(k-1)/k}$	$c_v(T_2 - T_1)$	$c_p(T_2 - T_1)$	0	$\dfrac{(P_2v_2 - P_1v_1)}{J(1 - k)}$	$\dfrac{k(P_2v_2 - P_1v_1)}{J(1 - k)}$	0
Throttling	h = constant T = constant $P_1/P_2 = v_2/v_1$	0	0	$(R/J)\ln(v_2/v_1)$	0	0	0
Polytropic	$P_1v_1^n = P_2v_2^n$ $T_2/T_1 = (v_1/v_2)^{n-1}$ $T_2/T_1 = (P_2/P_1)^{(n-1)/n}$	$c_v(T_2 - T_1)$	$c_p(T_2 - T_1)$	$c_v\ln(P_2/P_1)$ $+ c_p\ln(v_2/v_1)$	$\dfrac{(P_2v_2 - P_1v_1)}{J(1 - n)}$	$\dfrac{n(P_2v_2 - P_1v_1)}{J(1 - n)}$	$c_v\left(\dfrac{k - n}{1 - n}\right)(T_2 - T_1)$

Imperfect Gases

Property relationships for nonperfect gases and vapors are more complex than those for perfect gases. Property values for these fluids are facilitated by the use of charts and tables [4, 5, 6].

A *nonperfect gas* is one in which the molecules are close enough to exert forces on each other, as when a perfect gas is highly compressed and/or highly cooled with respect to its critical conditions. It is often described by modifying Eqs. (1-30*a*, *b*, and *c*) to the form

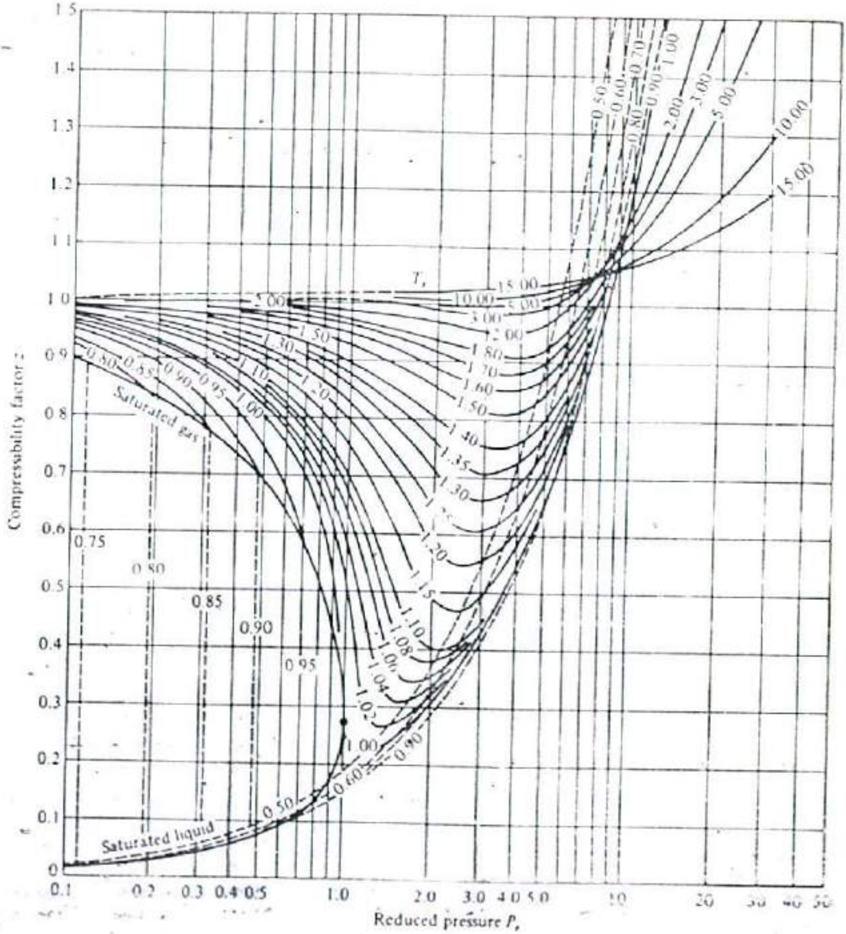


Figure 1-3 Generalized compressibility factor chart.

$$PV = mZRT \tag{1-31}$$

where Z is a compressibility factor that depends upon P, T, and the gas itself. Z is given in various compressibility charts. A generalized compressibility chart (see Fig. 1-3) gives Z for all gases as a function of the *reduced pressure* P_r and *reduced temperature* T_r, where

$$P_r = \frac{P}{P_c} \qquad \text{and} \qquad T_r = \frac{T}{T_c} \tag{1-32}$$

where P_c and T_c are the pressure and temperature, respectively, at the thermodynamic critical point for each gas. Note that at $P_c = 1$ and $T_c = 1$, the critical point for all fluids, $Z = 0.27$.

The critical constants for some fluids of interest are given in Table 1-4.

Example 1-1 Nitrogen is stored in a 10-ft^3 rigid tank at 1000 psia and 70°F. Find the mass of nitrogen in the tank and the error if the perfect gas law were used.

SOLUTION For nitrogen, $P_c = 492.91$ psia, $T_c = 227.16$°R, and $R = 55.15$ ft · lb$_f$/(lb$_m$ · °R). Therefore

$$P_r = \frac{1000}{492.91} = 2.029 \qquad T_r = \frac{70 + 460}{227.16} = 2.52$$

From Fig. 1-3: $Z = 0.98$. Using Eq. (1-19*d*)

$$m = \frac{PV}{ZRT} = \frac{(1000 \times 144)10}{0.98 \times 5515(70 + 460)} = 50.27 \text{ lb}_m$$

Table 1-4 Constants for some fluids*

Fluid	M	R, ft · lb$_f$/(lb$_m$ · °R)	P_c		T_c	
			psia	bar	°R	K
Air	28.967	53.34	547.43	37.744	557.1	309.50
Ammonia	17.032	90.77	1635.67	112.803	238.34	132.41
Carbon dioxide	44.011	35.12	1071.34	73.884	547.56	304.20
Carbon monoxide	28.011	55.19	507.44	34.995	239.24	132.91
Freon-12	120.925	12.78	596.66	41.148	693.29	385.16
Helium	4.003	386.33	33.22	2.291	9.34	5.19
Hydrogen	2.016	766.53	188.07	12.970	59.83	33.24
Methane	16.043	96.40	67.31	46.418	343.26	190.70
Nitrogen	28.016	55.15	492.91	33.993	227.16	126.20
Octane	114.232	13.54	362.11	24.973	1024.92	569.40
Oxygen	32.000	48.29	736.86	50.817	278.60	154.78
Sulfur dioxide	64.066	24.12	1143.34	78.850	775.26	430.70
Water	18.016	85.80	3206.18	221.112	1165.09	647.27

*Data from Ref. 4.

Using the perfect gas law, $m = PV/RT = 49.265\ \text{lb}_m$ with an error of about -2 percent.

Vapors

Property values for vapors, such as steam, are more complex than those for nonperfect gases and are given in the tables in App. A through F and in charts such as the T-s, P-v, and h-s (Mollier) diagrams (s is entropy, Sec. 1-8). They include data for the saturated liquid, the saturated vapor, the difference between these two, and for the superheated vapor. By the use of the familiar parameter *quality* x, which is the ratio of mass of vapor to mass of vapor and liquid in a two-phase mixture, one is able to obtain the properties of such a mixture. Recall that a two-phase mixture in equilibrium includes saturated liquid and saturated vapor and that neither can be subcooled or superheated. Thus the specific enthalpy of a two-phase mixture is given by

$$h = h_f + xh_{fg} \tag{1-33}$$

where h_f is the enthalpy of the saturated liquid and h_{fg} is the difference between the enthalpy of the saturated vapor, h_g, and h_f, that is, $h_{fg} = h_g - h_f$, all obtained at the pressure of the system. Similar equations are written for specific volume and entropy as

$$v = v_f + xv_{fg} \tag{1-34}$$

and

$$s = s_f + xs_{fg} \tag{1-35}$$

Subcooled Liquids

The subcooled liquid is the least understood. A *subcooled liquid* is one at a temperature below the saturation temperture at the given pressure. Examples are water at 70°F and 14.696 psia where the saturation temperature is 212°F, or water at 640°F and 2500 psia where the saturation temperature is 668.11°F. The term *compressed liquid* is synonymous with subcooled liquid. If we change our point of view from pressure to temperature and use the same examples above, we find that water at 70°F, if saturated, would have a pressure of 0.36292 psia. Because it is at 14.696 psia, it is said to be compressed. Likewise, water at 600°F has a saturation pressure of 2059.9 psia. Being at 2500 psia, it is said to be compressed. Hence, subcooled and compressed liquid have the same meaning.

Property data for the subcooled liquid may be found in tables [6], charts (Fig. 1-4), or by approximation from the saturated temperature steam tables. This works as follows. Data for a subcooled (or compressed) liquid are obtained as if the liquid were saturated at its given temperature, and the pressure is ignored. This approximation is reasonable, provided the actual and saturation pressures are not too far apart, within a few hundred psia for water, for example.

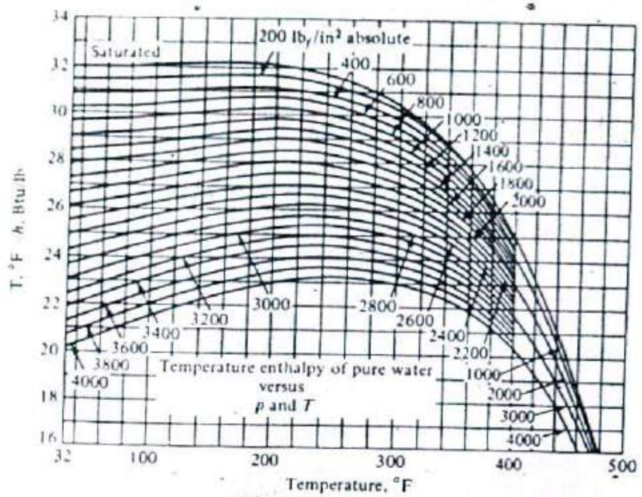


Figure 1-4 Enthalpy of subcooled water.

Example 1-2 Find the enthalpy of water at (1) 70°F and 1 atm and at (2) 640°F and 2500 psia.

SOLUTION

1. $h = h_f$ at 70°F $= 38.052$ Btu/lb$_m$
2. $h = h_f$ at 640°F $= 679.1$ Btu/lb$_m$

The American Society of Mechanical Engineers (ASME) tables for the compressed liquid give enthalpies of 38.09 and 673.47 Btu/lb$_m$, respectively. The approximations have resulted in errors of less than -0.1 percent in condition 1 and less than $+1$ percent in condition 2. Some subcooled water data is included in the superheat steam table, A-3, Appendix.

1-6 THE SECOND LAW OF THERMODYNAMICS

Whereas the first law of thermodynamics was one of conservation of energy, declaring that all forms of energy are convertible to one another, the second law puts a limitation on the conversion of some forms of energy to others. We are most concerned with two forms, heat and work. The second law does not negate the equivalence of conversion of these two, only the extent. Work is the more valuable commodity. It can

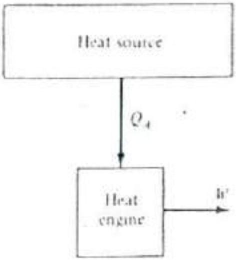


Figure 1-5 A device that violates the Kelvin-Planck statement of the second law.

be completely and continuously converted to heat. The opposite is not true. Heat cannot be completely and continuously converted to work. In other words, heat is not entirely *available* to do work on a continuous basis, i.e., in a cycle (though it may be in a process).

The portion of heat that cannot thus be converted to work, called *unavailable energy*, has to be rejected as low-grade heat after the work has been done. Thus, while energy is conserved, availability is not. The availability of a system always decreases. Another way of phrasing the second law is that the thermal efficiency of continuously converting heat to work, in a heat engine, must be less than 100 percent. The Carnot cycle (Sec. 1-9) represents an ideal heat engine that gives us an upper value of that efficiency between any two temperature limits.

There are a few historical statements of the second law that convey the above thoughts. Two are:

1. The Kelvin-Planck statement.* *It is impossible to construct a device which will operate in a cycle and produce no effect other than the raising of a weight and the exchange of heat with a single reservoir.* In this statement "operate in a cycle" means operate continuously, "raising of a weight" means doing work, and "exchange of heat with a single reservoir" means heat is only added, not rejected, and that there is a thermal efficiency of 100 percent. Figure 1-5 shows a device that violates the Kelvin-Planck statement.

---

* This statement is credited to both William Thompson, later Lord Kelvin (1824–1907), and Max Planck (1858–1947), though each stated it in a somewhat different way. Kelvin, who was knighted for helping lay the first transatlantic cable, was a professor of physics at Glasgow University, an excellent mathematician, an inventor, and a designer and was interested in athletics, the arts, and music. He contributed most to thermodynamics, establishing the thermodynamic temperature scale which is independent of the properties of matter. He also helped establish the first law on a firm foundation and contributed to the statement of the second law. Max Karl Ernst Ludwig Planck was a German professor who studied in Munich and taught in the universities of Munich, Kiel, and Berlin. One of his great contributions was in wavelength radiation and the definition of the "black body." He postulated the quantum theory and Planck's constant h, which has continued to influence physics and related sciences to a degree well beyond its original intent. Planck had wide interests in fields other than physics, including philosophy, religion, and social and political matters.

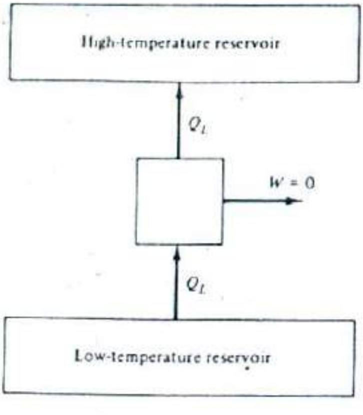


Figure 1-6 A device that violates the Clausius statement of the second law.

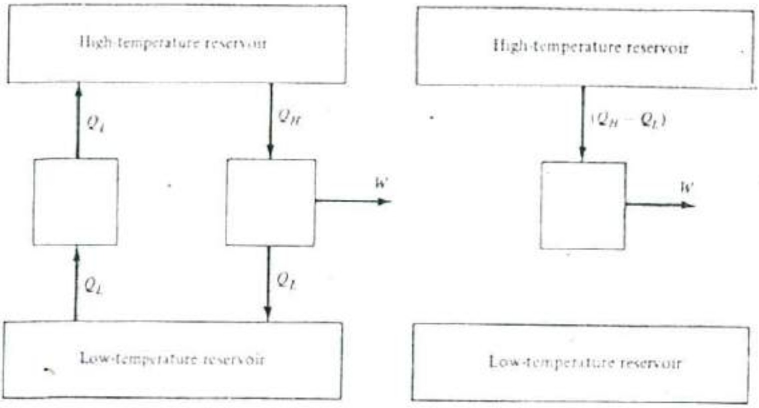
2. The Clausius statement.* *It is impossible to construct a device which operates in a cycle and produces no effect other than the transfer of heat from a cooler to a hotter body.*

These two statements, though they say different things, can be shown to be equivalent. Figure 1-6 shows a device that operates between a high-temperature reservoir and a low-temperature reservoir. A *reservoir* is a source of heat or a heat sink large enough that it does not undergo a change in temperature when heat is added or subtracted from it. The device in Fig. 1-6 produces no effect other than the transfer of heat from the low-temperature reservoir to the high-temperature reservoir; hence, it produces (or absorbs) zero work. By the first law, with this device as a system, the heat Q_L received from the low-temperature reservoir is equal to that delivered to the high-temperature reservoir.

Let us now add to the above a device that does not violate the Kelvin-Planck statement (Fig. 1-7*a*). Let us also choose that second device so that it rejects to the low-temperature reservoir the same Q_L as the first device. When we combine both devices (Fig. 1-7*b*), the result will be a device that receives $Q_H - Q_L$ from the single reservoir, the high-temperature one, and produces work W. This violates the Kelvin-Planck statement.

---

* Rudolf Julius Emmanuel Clausius (1822–1888) was a German professor of physics and a mathematical genius who worked in optics, electricity, and electrolysis and is credited with founding the kinetic theory of gases. Clausius elaborated and restated the work of Carnot (Sec. 1-9) and thus deduced his famous principle of the second law. He wrote an exhaustive treatise on the steam engine in which he emphasized the then new concept of entropy.



(a) (b)

Figure 1-7 Conversion of device in Fig. 1-6 to device in Fig. 1-5.

1-7 THE CONCEPT OF REVERSIBILITY

The concept of reversibility was first introduced in 1824 by Sadi Carnot, who laid the foundations of the second law and introduced the concept of a cycle. Reversibility applies to processes. A cycle can be reversible, but only if *all* its processes are reversible. One irreversible process in a cycle renders the whole cycle irreversible.

A *reversible process*, also called an *ideal process*, is one which can reverse itself exactly by following the same path it undertook in the first place and thus restore to the system or the surroundings the same heat and work previously exchanged. For example, process 1-2 in Fig. 1-2, which was called ideal and is now called reversible, is so only when it can trace its exact path back from 2 to 1 and thus give back the exact work of compression done upon it from 1 to 2.

Needless to say there are no ideal, and hence no reversible, processes in the real world. Real processes are *irreversible*, although the degree of irreversibility varies between processes. There are many sources of irreversibility in nature. For our purposes, four are most important. These are ***friction, heat transfer, throttling,*** and ***mixing***.

Friction *Mechanical friction* is one in which mechanical work is dissipated into a heating effect, such as in the case of a shaft rotating in a bearing. It is not possible to add the same heat to the bearing and expect the shaft to rotate. Another example is the rubbing of one's hands. Thus friction is the dissipation of energy that otherwise could have been transformed to useful work.

Fluid friction is similar to mechanical. A fluid expanding behind a piston or through a turbine undergoes internal friction, resulting in the dissipation of part of its energy into heating itself at the expense of useful work. The fluid then does less work and exhausts at a higher temperature (or enthalpy) than it would had this fluid friction not existed. The more irreversible the process, the more the heating effect and the less the work.

Heat transfer *Heat transfer* in any of its forms, conduction, convection, or radiation, occurs from a higher temperature to a lower temperature. It cannot reverse itself without the help of an external aid (a heat pump). If it could, it would violate the Clausius statement of the second law.

Heat transfer causes a loss of availability because no work is done between the high- and low-temperature bodies. It also follows that the larger the temperature difference between the two bodies, the larger the loss of availability (i.e., the loss of potential work), and the larger the irreversibility.

All power systems employ heat-transfer processes from a primary source, such as combustion gases or nuclear-reactor primary coolant, to a working fluid. It is thus important to reduce the temperature differences across the heat exchanger to the minimum practicable to reduce this source of irreversibility, which is a primary cause of reduction in efficiency of real systems.

Throttling *Throttling* is an uncontrolled expansion process of a fluid from a high-pressure (and occasionally high-temperature) region to a low-pressure region. Examples are leakage from a steam pipe or the flow through a valve. No work is done and heat transfer across the narrow opening is negligible. The resulting high kinetic energy is dissipated in fluid friction to restore the enthalpy of the fluid to its original value. In Eq. (1-1a) all terms drop out except enthalpy, resulting in Eq. (1-24*a*), which is repeated here:

$$H_1 = H_2 \tag{1-24a}$$

A throttling process, therefore, is a form of fluid friction, one in which the enthalpy is constant. Such a process is also called a *Joule-Thompson expansion*. Throttling is irreversible because flow cannot be reversed from the low-pressure region to the high-pressure region. It results in the loss of availability, i.e., the loss of work that could have been otherwise obtained if expansion between the two regions occurred ideally behind a piston or through a turbine.

It follows that the greater the pressure difference in throttling, the greater the irreversibility. Throttling should be avoided or minimized in power systems, though it is necessary in some applications.

Mixing When two or more separate fluids or gases are made to mix or diffuse into each other, they cannot unmix without external aid. Hence *mixing* is an irreversible process that results in the loss of availability of the constituent fluids. Mixing is unavoidable in many cases, such as when fuels and air are prepared for combustion or when steam and colder water mix in certain devices, such as open feedwater heaters. It is not, however, as major a concern in power systems as fluid friction or heat transfer.

External and Internal Irreversibilities In power systems, irreversibilities are sometimes classified as external and internal.

External irreversibilities are those that occur across the boundaries of the system. The primary source of external irreversibility in power systems is heat transfer, both at the high-temperature end, the heat source, and the low-temperature end, the heat sink. Another source of external irreversibility in power systems is mechanical friction in bearings of rotary machines such as turbines, compressors, pumps, and generators and electrical losses in the latter and in distribution systems.

Internal irreversibilities are those that occur within the boundaries of the system. The primary source of internal irreversibility in power systems is fluid friction in rotary machines such as turbines, compressors, and pumps and in pipes and valves. Other sources are throttling and mixing.

1-8 THE CONCEPT OF ENTROPY

Entropy, first introduced by Clausius in 1865, is a property, as are pressure, temperature, internal energy, and enthalpy. It is given the symbol S (and sometimes Φ) and has the units Btu per degree Rankine (Btu/°R) or the units joule per kelvin (J/K). Specific entropy s has the units Btu/(lb$_m$ · °R) or J/(kg · K). The physical meaning of entropy will be apparent later. It is convenient to introduce it first as a mathematical convenience.

An analog is made with nonflow work

$$\Delta W_{nf} = \int_1^2 P\,dV \tag{1-26}$$

This expression strictly gives the nonflow work for a reversible or ideal process because it is the only case where there is pressure equilibrium, i.e., when the pressure at the face of the piston is the same as in the bulk of the fluid, a situation that does not happen in nature, especially for high-speed machines. Its use, however, is extended to real processes because the pressure differential is small.

Equation (1-26) shows that the nonflow work is, graphically, the area under the process when plotted on a P-V diagram (Fig. 1-8a). Because heat, like work, is an

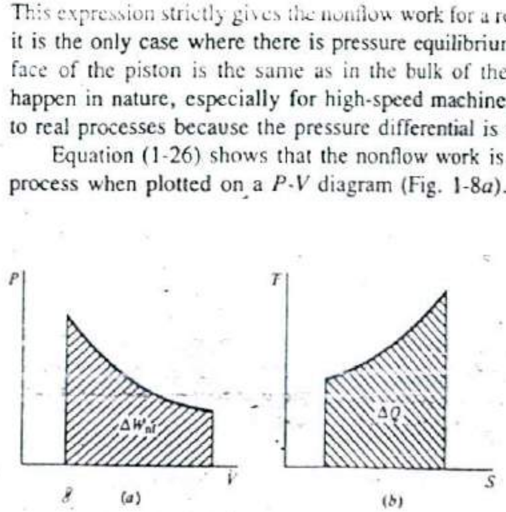


Figure 1-8 Areas under process 1-2 on P-V and T-S diagrams.

important energy term in energy systems, we ask ourselves: Wouldn't it be convenient to have a similar graph where areas represent heat? It is natural to have temperature as one of the ordinates. We invent the second and call it *entropy*. As $\Delta W_{nf} = \int_1^2 P\, dV$ strictly for a reversible process, then we can write

$$\Delta Q = \int_1^2 T\, dS \qquad \text{(reversible process only)} \tag{1-36}$$

and the area under process 1-2 on the T-S diagram equals ΔQ (Fig. 1-8*b*). As we shall see, although we can waive the reversibility requirement for Eq. (1-26) with little error, we cannot do the same for Eq. (1-36), which is true strictly for reversible processes.

In both Eqs. (1-26) and (1-36), if the integral is positive, i.e., we move from left to right, work is done by and heat is added to the process. In cycles, the net work and net heat of the cycles are represented by the enclosed areas on the P-V and T-S diagrams, provided the cycle is reversible (i.e., all its processes are reversible) in the case of the T-S diagram.

It also follows that the net work of a cycle, such as the diesel cycle (Fig. 1-2), is equal to the enclosed area of the cycle on the P-V diagram. A cycle is a power cycle, i.e., the net work is positive, if the cycle is clockwise. It is a reversed cycle or a heat pump if the net work is negative and the cycle is counterclockwise on the P-V diagram. It also follows from Eq. (1-29) that the enclosed area on the T-S diagram is also equal to the net work if the cycle is reversible. The enclosed areas on the P-V and T-S diagrams for the same reversible cycle are therefore equal in sign and magnitude, taking scales and conversion factors into account, of course.

We now examine (1-36) further. A *reversible adiabatic process* (already mentioned in the case of the ideal diesel cycle), while not existing in nature, is a most important process in cycle analysis. For such a process, being adiabatic, $\Delta Q = 0$, and being reversible, it obeys Eq. (1-36). Thus, for an adiabatic reversible process,

$$\int T\, dS = 0 \tag{1-37a}$$

and because T cannot equal zero, $dS = 0$

$$\begin{aligned} &\text{or} \quad S = \text{constant} \\ &\text{and} \quad s = \text{constant} \qquad \text{(adiabatic reversible process)} \end{aligned} \tag{1-37b}$$

Thus an adiabatic reversible process is one of constant entropy, i.e., vertical on the T-S diagram, as represented by the expansion line 1-2, (Fig. 1-9*a*).

All this leads to one physical meaning for entropy: *Entropy is the property that remains constant in an adiabatic reversible process.* This is much like temperature being the property that stays constant in an isothermal process; pressure in a constant-pressure process, etc. A more important physical meaning will become apparent shortly.

Let us now assume that the expanding fluid is a perfect gas (though we can easily come to the same conclusions using a vapor or a mixture of liquid and vapor.) Lines P_1 and P_2 (Fig. 1-9*a*) are constant-pressure lines for a perfect gas on the T-s plane where $P_1 > P_2$. Their shape can be ascertained from the perfect-gas relationship

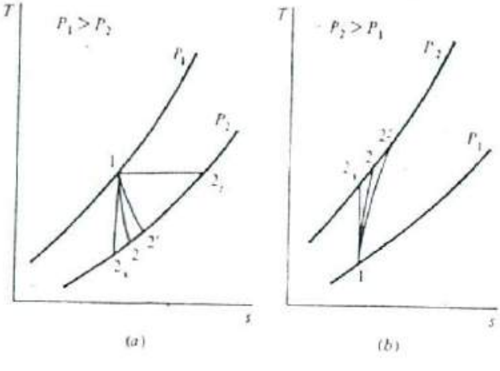


Figure 1-9 Expansion (*a*) and compression (*b*) of a gas from P_1 to P_2 on the T-S diagram. $1\text{-}2_s$ adiabatic reversible, 1-2 adiabatic irreversible, $1\text{-}2_t$ throttling.

$s_2 - s_1 = c_p \ln (T_2/T_1)$ (Table 1-3). Let us further assume that the gas starts at 1 with T_1, s_1 (and corresponding P_1 and v_1) and expands *adiabatically* to pressure P_2. We have already seen that if the process is reversible as well as adiabatic, entropy remains constant and the process is represented by $1\text{-}2_s$.

The question is: What does the process look like if it were adiabatic but irreversible? We have already noted that irreversibility would manifest itself in an increase in temperature of the gas leaving at P_2, point 2, beyond that for the adiabatic reversible process. Thus $T_2 > T_{2_s}$. On the P_2 line, which curves upwards, this can only occur if the entropy at 2 is greater than at 2_s. A more irreversible expansion results in greater self-heating of the gas and the process would be 1-2′, etc. In other words: *The greater the irreversibility, the greater the increase in entropy in an adiabatic process.*

If the expansion were to occur in a turbine, the work is obtained from the first law (which applies equally to reversible and irreversible processes) for the three processes as $H_1 - H_{2_s}$, $H_1 - H_2$, and $H_1 - H_{2'}$, respectively. Because the $T_{2'} > T_2 > T_{2_s}$ and $dh = c_p\,dT$ for gases, then $H_{2'} > H_2 > H_{2_s}$. Thus the work produced by the turbine W_T for three cases is given by

$$H_1 - H_{2_s} > H_1 - H_2 > H_1 - H_{2'}$$

The adiabatic reversible turbine produces the most work. The greater the irreversibility, the less the work.

The process $1\text{-}2_t$ is one of constant temperature and hence, for a gas, constant enthalpy. This is a throttling process [Eq. (1-11)], where the work ΔH is zero and all energy is dissipated in fluid friction. This is the most irreversible process and the one with the most increase in entropy (Fig. 1-9*a*).

The degree of irreversibility is given by an expansion or turbine efficiency called the *polytropic turbine efficiency* η_T (sometimes called the isentropic or adiabatic turbine efficiency), which is equal to the ratio of actual work to ideal work and is given by

$$\eta_T = \frac{H_1 - H_2}{H_1 - H_{2_s}} = \frac{h_1 - h_2}{h_1 - h_{2_s}} \tag{1-38}$$

and for constant specific heats

$$\eta_T = \frac{T_1 - T_2}{T_1 - T_{2s}} \tag{1-39}$$

In the case of compression from P_1 to P_2 where $P_2 > P_1$ (Fig. 1-9*b*), an adiabatic reversible compression follows the constant entropy path 1-2_s. In the case of adiabatic irreversible compression the gas leaves at a higher temperature T_2, here because the fluid absorbs some work input which is dissipated in fluid friction. The greater the irreversibility, the greater the exit temperature $T_{2'} > T_2 > T_{2s}$ and the greater the increase in entropy. Again, since $dh = c_p\, dT$ for gases, then $H_{2'} > H_2 > H_{2s}$ and the work absorbed in compression $|W_c|$ increases with irreversibility or

$$H_{2'} - H_1 > H_2 - H_1 > H_{2s} - H_1$$

The degree of irreversibility here is given by a compressor efficiency, called the *polytropic compressor efficiency* η_c (and sometimes the isentropic or adiabatic compressor efficiency) equal to the ratio of ideal work to actual work (the reverse of that for expansion) and given by

$$\eta_c = \frac{H_{2s} - H_1}{H_2 - H_1} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{1-40}$$

and for constant specific heats

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1} \tag{1-41}$$

We can now state that *the change of entropy is a measure of the unavailable energy*, which leads us to an important physical meaning of entropy, namely that entropy is a measure of irreversibility, or more generally: *Entropy is a measure of disorder*. This is a concept that is used in sciences other than thermodynamics, such as social sciences that deal with society disorders.

Because the universe is an isolated system it does not exchange energy across its boundaries; i.e., it is adiabatic. Because it is full of irreversible processes, it follows that: *The entropy of the universe is continually on the increase* and the end of the universe would occur when entropy is at maximum, i.e., when all energy has been dissipated to a bottom state, when all availability is lost, when all matter is at the same temperature and no life, as we know it, is possible.

In the case of vapors, when expansion between P_1 and P_2 ends in the two-phase region, shown in Fig. 1-10 on both *T-s* and *h-s* (Mollier) charts, the same observation as for the gases applies except that the exit temperature is the same for the adiabatic reversible and adiabatic irreversible processes because they are both in the two-phase region. The exit enthalpy, however, is greater in the case of the irreversible process $h_2 > h_{2s}$ and the work is less: $h_1 - h_2 < h_1 - h_{2s}$. The degree of irreversibility here is given by a turbine efficiency, the same as in Eq. (1-38).

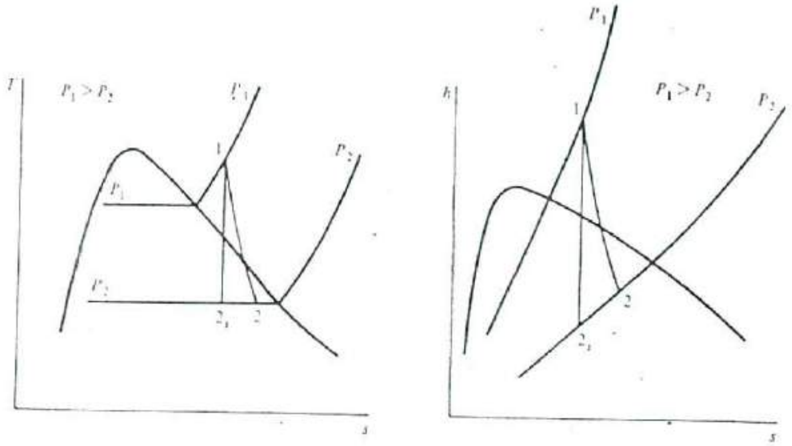


Figure 1-10 Expansion of a vapor from P_1 to P_2 on the T-s and Mollier (h-s) diagrams. 1-2_s adiabatic reversible, 1-2 adiabatic irreversible.

In the case of pumping liquids, a pump efficiency η_p, given by the ratio of ideal work to actual work, is used to obtain the actual work. Using Eq. (1-22a), actual work

$$|W_p| = \frac{H_{2s} - H_1}{\eta_p} \cong \frac{V(P_1 - P_2)}{\eta_p} \tag{1-42}$$

1-9 THE CARNOT CYCLE

Sadi Carnot* laid the foundations of the second law of thermodynamics, introduced the concepts of reversibility and cycles, and introduced the principle that the temperatures of the heat source and heat sink determined the thermal efficiency of a reversible cycle. He also postulated that because all such cycles must reject heat to the heat sink, efficiency is never 100 percent [7]. To show this, and to show that the effect of the working fluid on the thermal efficiency of a reversible cycle is nonexistent, Carnot invented his famous, though hypothetical (one cannot build a reversible engine), *Carnot cycle*.

The Carnot cycle, shown in Fig. 1-11 on the P-V and T-S diagrams, is composed of four processes:

---

* Nicolas Leonard Sadi Carnot (1796–1832), despite his profound and lasting effect on the science of thermodynamics, was a quiet, unassuming Frenchman who lived during the turbulent Napoleonic years and had an unspectacular life. One of his mottoes, "Speak little of what you know and not at all of what you do not know," reflects something of his demeanor.

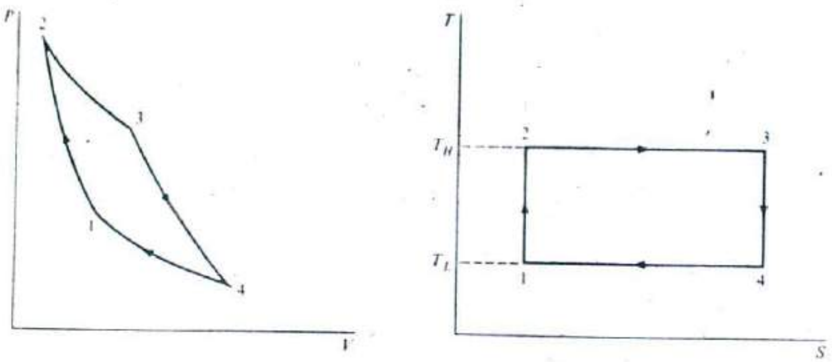


Figure 1-11 Carnot cycle on the P-V and T-S diagrams.

1. *1-2:* reversible adiabatic compression
2. *2-3:* reversible constant-temperature heat addition
3. *3-4:* reversible adiabatic expansion
4. *4-1:* reversible adiabatic heat rejection

The thermal efficiency of the Carnot cycle η_C can now be easily obtained, noting that the change in entropy during heat addition and rejection are equal in magnitude. Thus

$$Q_A = T_H (S_3 - S_2)$$

$$Q_R = T_L (S_1 - S_4)$$

or

$$|Q_R| = T_L (S_4 - S_1) = T_L (S_3 - S_2)$$

where T_H and T_L are the heat source and heat sink absolute temperatures, respectively. Note that the Carnot cycle is both externally and internally reversible. Thus heat transfer between the heat source and the working fluid occurs, hypothetically, across a zero temperature difference. Hence $T_H = T_2$. Similarly $T_L = T_1$.

For *all* power cycles the net work and the thermal efficiency are defined by

$$\Delta W_{net} = Q_A - |Q_R| \tag{1-43}$$

and

$$\eta_{th} = \frac{\Delta W_{net}}{Q_A} \tag{1-44}$$

Thus, the thermal efficiency of the Carnot cycle η_C is given by

$$\eta_C = \frac{T_H - T_L}{T_L} \tag{1-45}$$

This relationship can also be easily obtained from the net work of the different processes using the appropriate relationships in Table 1-1 (Prob. 1-11).

Equation (1-45) shows that the thermal efficiency of the Carnot cycle is only a function of the heat source and heat sink temperatures and is independent of the working fluid. Indeed this conclusion is true of other reversible cycles (Stirling, Ericsson) that receive and reject heat at constant temperature.

Because the Carnot cycle is reversible, it produces the maximum work possible between the given temperature limits T_H and T_L and, hence, a reversible cycle operating between given temperature limits has the highest possible thermal efficiency of all cycles operating between these same temperature limits.

The Carnot efficiency, therefore, is to be considered an upper limit, an ideal, a goal to strive toward when building real cycles, but one that is impossible to exceed or equal.

Another observation of importance is that the trends predicted by the Carnot cycle apply to real cycles. Thus real cycles will have higher efficiencies, the higher the temperature at which they receive heat and the lower the temperature at which they reject heat.

Example 1-3 Air expands in a gas turbine from 10 atm and 2000°F to 1 atm. The exhaust temperature is 1050°F. Assuming a constant specific heat of 0.240 Btu/(lb$_m$ · °F), find the turbine work in Btu/lb$_m$ and the turbine adiabatic efficiency.

SOLUTION
$$W_T \doteq h_1 - h_2 = c_p(T_1 - T_2) = 0.24(2000 - 1050)$$
$$= 228 \text{ Btu/lb}_m$$

Referring to Table 1-3 and Fig. 1-9*a*

$$\frac{T_{2s}}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} \qquad \text{or} \qquad \frac{T_{2s}}{(2000 + 460)} = \left(\frac{1}{10}\right)^{(1.4-1)/0.4}$$

Therefore $T_{2s} = 1274.2°\text{R} = 814.2°\text{F}$

$$\eta_T = \frac{h_1 - h_{2s}}{h_1 - h_2} = \frac{T_1 - T_{2s}}{T_1 - T_2} \qquad (\text{for constant } c_p)$$

$$= \frac{2000 - 1050}{2000 - 814.2} = 0.801 = 80.1\%$$

Example 1-4 Find the work of the turbine of Example 1-1 but use variable specific heats.

SOLUTION Using Eq. (1-19)

$$W_T = \int_1^2 c_p\, dT = \int_{1510}^{2460} (0.219 + 3.42 \times 10^{-5}T - 2.93 \times 10^{-9} T^2)\, dT$$

$$\doteq \left[0.219T + \frac{3.42 \times 10^{-5}}{2} T^2 - \frac{2.93 \times 10^{-9}}{3} T^3\right]_{1510}^{2460}$$

$$= 0.219(2460 - 1510) + \frac{3.42 \times 10^{-5}}{2}(2460^2 - 1510^2) - \frac{2.93 \times 10^{-9}}{3}$$

$$\cdot (2460^3 - 1510^3)$$

$$= 208.05 + 64.49 - 14.54$$

$$= 258 \text{ Btu/lb}_m$$

as compared with $W_T = 228 \text{ Btu/lb}_m$ when a constant specific heat at low temperature $[0.24 \text{ Btu/(lb}_m \cdot {}^\circ\text{R)}]$ was used.

To get the adiabatic efficiency, it is necessary to calculate T_{2s} and h_{2s} using average values of c_p and k between T_1 and the unknown T_{2s}. The solution may be obtained by trial and error, keeping in mind that because both c_p and c_v increase with temperature, and the difference between them is a constant, $k = c_p/c_v$ decreases with temperature. Thus one expects that T_{2s} would be greater than in Example 1-1. Assuming it to be 950°F, find an average $\overline{c_p}$ between 2000 and 950°F using Eq. (1-19)

$$\overline{c_p} = \frac{\int_{1410}^{2460} c_p \, dT}{2000 - 950} = \frac{287.64}{1050} = 0.2739 \text{ Btu/(lb}_m \cdot {}^\circ\text{R)}$$

$$\overline{c_v} = \overline{c_p} - R = 0.2739 - \frac{53.34}{778.16} = 0.2739 - 0.0685 = 0.2054 \text{ Btu/(lb}_m \cdot {}^\circ\text{R)}$$

Therefore $\bar{k} = \dfrac{0.2739}{0.2054} = 1.3335$

and $T_{2s} = T_1 \left(\dfrac{1}{10}\right)^{\frac{0.3335}{1.3335}} = 2460 \times 0.5622 = 1383^\circ\text{R} = 923^\circ\text{F}$

A second trial and error shows that $\overline{c_p} = 0.2736$ and $T_{2s} = 1382^\circ\text{R} = 922^\circ\text{F}$ compared with 814.2°F for the adiabatic reversible work.

The ideal work is

$$h_1 - h_{2s} = \int_{1382}^{2460} c_p \, dT = \overline{c_p}(T_1 - T_{2s}) = 295.4 \text{ Btu/lb}_m$$

The turbine adiabatic efficiency is

$$\frac{258.0}{295.4} = 0.873 = 87.3\%$$

Compared with 80.1 percent when calculated using $c_p = 0.24 \text{ Btu/(lb}_m \cdot {}^\circ\text{R)}$. In the case of air with products of combustion, as in the case with gas turbines and other devices, there are available property tables [8] that take into account the temperature, the fuel-air ratios, and the effects of chemical equilibrium.

Example 1-5 A water pump compresses saturated water at 1 to 2500 psia. The pump efficiency is 0.70. Find the ideal and actual pump and the water exit temperature.

SOLUTION We will use two methods.

1. Assume water to be incompressible so that v = constant = v_1 at 1 psia = 0.016136 ft^3/lb_m, and no temperature rise across the pump.

$$\text{Ideal pump work } |W_{p,s}| = v_1(P_2 - P_1) = \frac{0.016136(2500 - 1)144}{778.16}$$

$$= 7.46 \text{ Btu/lb}_m$$

$$\text{Actual pump work } |W_p| = \frac{\Delta W_{p,s}}{\eta_p} = \frac{7.46}{0.7} = 10.66 \text{ Btu/lb}_m$$

2. Use the ASME steam tables [6]

$$T_1 = 101.74°\text{F} \qquad s_1 = 0.1326 \text{ Btu/lb}_m°\text{R} \qquad h_1 = 69.73 \text{ Btu/lb}_m$$

Using the compressed-liquid tables (a portion of which is reproduced in Table 1-5), for the ideal case $s_{2s} = s_1 = 0.1326$. By interpolation $T_{2s} = 102.63°\text{F}$, $v_{2s} = 0.01602$ ft^3/lb_m, $h_{2s} = 77.17$ Btu/lb_m. Thus, the temperature rises by 0.89°F and, in the ideal case, the specific volume decreases by 0.72 percent.

$$\text{Ideal work } |W_{p,s}| = 77.17 - 69.73 = 7.44 \text{ Btu/lb}_m$$

$$\text{Actual work } |W_p| = \frac{7.44}{0.7} = 10.63 \text{ Btu/lb}_m$$

The actual temperature and specific volume changes may be obtained by interpolation at $h_2 = 69.73 + 10.63 = 80.36$ Btu/lb_m, as 105.86°F and 0.01603 ft^3/lb_m.

It can be seen that the two methods are fairly comparable even for the large pressure difference chosen for this example. Thus method 1 is sufficiently accurate for most calculations, with the possible exception of exit water temperature.

Table 1-5 Liquid data used in calculation of example 1-5*

Pressure, psia	Temperature, °F	Specific volume, ft^3/lb_m	Enthalpy, Btu/lb_m	Entropy, $\text{Btu/(lb}_m \cdot °\text{R)}$
1.0	101.74 (saturated)	0.016136	69.73	0.1326
2500	100	0.01601	77.57	0.1280
	110	0.01605	84.45	0.1455

*Data from Ref. 6.

PROBLEMS

1-1 A perfect gas that has a constant specific heat at constant pressure $c_p = 0.26$ Btu/lb$_m$ undergoes an expansion process in a steady-flow machine with a mass flow rate of 100 lb$_m$/h. The machine is water-cooled. The water mass flow rate is 10 lb$_m$/h. During the process the gas temperature drops from 200 to 100°F and the water temperature rises from 70 to 100°F. Ignoring changes in potential and kinetic energies for the gas, calculate the work of the gas in Btus per hour and watts. [c_p for water = 1.0 Btu/(lb$_m$·°R).]

1-2 Air at 140°F and 100 psia is confined in an uninsulated 10-ft^3 vessel. A propeller is driven inside the vessel by a 50-W electric motor. After a period of 1 h the air temperature dropped to 100°F. Find the heat transfer in Btu per hour.

1-3 A gas has a molecular mass of 30 and a specific heat at constant pressure of 0.25 Btu/(lb$_m$·°R). It undergoes a nonflow polytropic compression during which its temperature increases from 100 to 200°F. The polytropic exponent is 1.3. Calculate the work done and the heat transfer in Btus per pound mass.

1-4 1000 lb$_m$/h of pure air enters a gas turbine at 2000°F and leaves at 1000°F. Find the work in kilowatts using variable specific heat for air, and the error if a constant specific heat equal to that at low temperatures is used.

1-5 Calculate the average specific heats for air in Btus per pound mass per degree Rankine between 1000 and 2000°F.

1-6 Steam is confined in a 100-ft^3 rigid vessel at 5000 psia and 1000°F. Find its mass in pounds mass using (*a*) data from the steam tables and (*b*) the compressibility chart.

1-7 A 10-m^3 rigid tank contains steam at 30 bar and 400°C. It is left to cool down until its pressure drops to 5 bar. Find (*a*) the final condition of the steam and (*b*) the heat transfer, in kilojoules.

1-8 A rigid 10-ft^3 vessel contains air at 15 psia and 1000°F. Heat is added until the air temperature reaches 2000°F. Assuming variable specific heats, calculate (a) the heat added, in Btus, and (*b*) the final pressure, pounds force per square inch absolute.

1-9 Steam at 30 bar and 400°C expands behind a piston in an insulated cylinder to 5 bar and 20 ft^3. Find the work done in kilojoules.

1-10 10 lb$_m$/h of liquid ammonia at 87°F and 250 psia is throttled into a flash tank to 100 psia. Ammonia vapor is drawn out at the top of the tank while liquid is discharged at the bottom. What are the temperature in degrees Fahrenheit, mass and volume flow rates in pounds mass per hour and cubic feet per hour of the two streams?

1-11 Derive the expression for the Carnot cycle efficiency [Eq. (1-45)] using the appropriate work relation for gases from Table 1-3.

1-12 An inventor claims to have built an engine that operates on a cycle, receives 1000 kJ at 500°C, produces work, and rejects 350 kJ at 50°C. Is this claim valid? Why?

1-13 The Carnot cycle is rectangular on the *T-s* diagram. Consider another cycle that is rectangular, but on the *P-v* diagram. Draw that cycle on both the *T-s* and *P-v* diagrams, labeling corners correspondingly, and name all its processes.

1-14 Derive expressions for the efficiency of the cycle in Prob. 1-13 in terms of its high and low temperatures T_H and T_L, and constant specific heats c_p and c_v of a gas working fluid for the cases of equal (*a*) temperature rises and (*b*) heats added at constant volume and pressure. Is this a good cycle? Why?

1-15 Using the expression for the change in entropy of gases for a polytropic process in Table 1-3, derive similar expressions in terms of changes in (*a*) pressure and temperature and (*b*) volume and temperature.

1-16 A reversible cycle consists of an isentropic compression from an initial temperature T_1 to 1000°R, a constant-volume process from 1000 to 1500°R a reversible adiabatic expansion to 1000°R a constant-pressure expansion from 1000 to 1500°R, and a constant-volume process to the initial temperature. Draw the cycle on the *P-v* and *T-s* diagrams, and calculate the initial temperature if the working fluid is a gas with k = 1.40.

1-17 Air expands from 10 bar and 1000°C to 1 bar and 500°C in an insulated turbine. Calculate (*a*) the turbine polytropic efficiency, (*b*) the change in entropy, in kilojoules per kilogram per Kelvin, and (*c*) the work, kilojoules per kilogram, and (*d*) the polytropic exponent n. Assume a constant specific of 1.005 kJ/kg·K.

1-18 Helium is compressed from 15 psia and 40°F to 60 psia. The compressor adiabatic efficiency is 0.70. Find (*a*) the helium exit temperature, in degrees Fahrenheit, (b) the work done in Btu per pound mass, and (c) the change in entropy, Btu/lb_m·°R.

1-19 1000 kg/h of water at 60°C and 1 bar are pumped to 100 bar. The pump efficiency is 0.65. Find the work in kilowatts.

1-20 Saturated Freon-12 vapor at 215 psia expands in a nozzle to 72.433 psia. The nozzle has an efficiency of 0.95 and an exit area of 10 in^2. Find the mass flow rate in pounds mass per hour.

1-21 10^6 lb_m/h of steam at 2500 psia and 1000°F expand in a turbine to 1 psia. The turbine has adiabatic and mechanical efficiencies of 0.90 and 0.95, respectively. It drives an electric generator that has an efficiency of 0.96. Calculate the output power of the generator in megawatts.

CHAPTER

TWO

THE RANKINE CYCLE

2-1 INTRODUCTION

When the Rankine* cycle was devised, it was readily accepted as the standard for steam powerplants and remains so today. Whereas the ideal diesel cycle (Fig. 1-2) is a gas cycle and the Carnot cycle (Fig. 1-11) is a cycle for all fluids, the Rankine cycle is a vapor-and-liquid cycle.

The real Rankine cycle used in powerplants is much more complex than the original, simple ideal Rankine cycle. It is by far the most widely used cycle for electric-power generation today and will most certainly continue to be so in the future. It is the backbone of much of the work presented in this book.

This chapter is devoted exclusively to the Rankine cycle, from its simplest ideal form to its more complex nonideal form with modifications and additions that render it one of the most efficient means of generating electricity today.

2-2 THE IDEAL RANKINE CYCLE

Because Rankine is a vapor-liquid cycle, it is most convenient to draw it on both the P-V and T-S diagrams with respect to the saturated-liquid and vapor lines of the working fluid, which usually, but not always, is H_2O. Figure 2-1 shows a simplified flow

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* William John M. Rankine (1820–1872) was a professor of civil engineering at Glasgow University. He was an engineer and scientist of many talents which, besides civil engineering, included shipbuilding, waterworks, singing, and music composition. He was one of the giants of thermodynamics and the first to write formally on the subject.

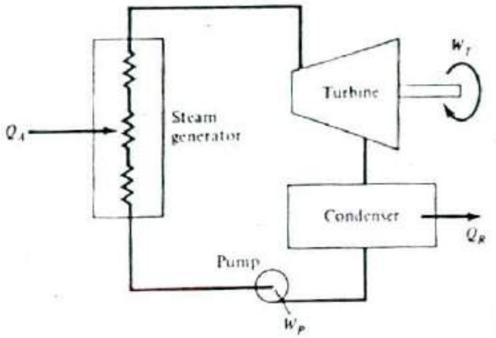
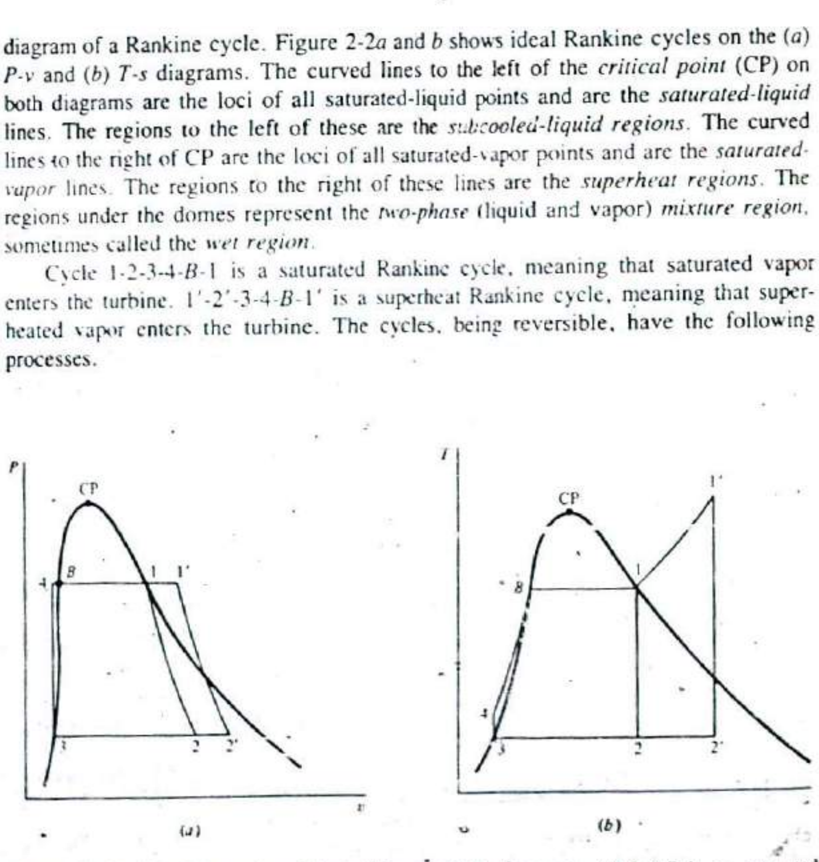


Figure 2-1 Schematic flow diagram of a Rankine cycle.

diagram of a Rankine cycle. Figure 2-2*a* and *b* shows ideal Rankine cycles on the (*a*) *P-v* and (*b*) *T-s* diagrams. The curved lines to the left of the *critical point* (CP) on both diagrams are the loci of all saturated-liquid points and are the *saturated-liquid* lines. The regions to the left of these are the *subcooled-liquid regions*. The curved lines to the right of CP are the loci of all saturated-vapor points and are the *saturated-vapor* lines. The regions to the right of these lines are the *superheat regions*. The regions under the domes represent the *two-phase* (liquid and vapor) *mixture region*, sometimes called the *wet region*.

Cycle 1-2-3-4-*B*-1 is a saturated Rankine cycle, meaning that saturated vapor enters the turbine. 1'-2'-3-4-*B*-1' is a superheat Rankine cycle, meaning that superheated vapor enters the turbine. The cycles, being reversible, have the following processes.



(*a*) (*b*)

Figure 2-2 Ideal Rankine cycles of the (*a*) *P-v* and (*b*) *T-s* diagrams. 1-2-3-4-*B*-1 = saturated cycle. 1'-2'-3-4-*B*-1' = superheated cycle. CP = critical point.

1-2 or 1′-2′: adiabatic reversible expansion through the *turbine*. The exhaust vapor at 2 or 2′ is usually in the two-phase region.

2-3 or 2′-3: constant temperature and, being a two-phase mixture process, constant-pressure heat rejection in the condenser.

3-4: adiabatic reversible compression by the *pump* of saturated liquid at the condenser pressure, 3, to subcooled liquid at the steam-generator pressure, 4. Line 3-4 is vertical on both the *P-V* and *T-S* diagrams because the liquid is essentially incompressible and the pump is adiabatic reversible.

4-1 or 4-1′: constant-pressure heat addition in the *steam generator*. Line 4-*B*-1-1′ is a constant-pressure line on both diagrams. The portion 4-*B* represents bringing the subcooled liquid, 4, to saturated liquid at *B*. The section 4-*B* in the steam generator is called an *economizer*. The portion *B*-1 represents heating the saturated liquid to saturated vapor at constant pressure and temperature (being a two-phase mixture), and section *B*-1 in the steam generator is called the *boiler* or *evaporator*. Portion 1-1′, in the superheat cycle, represents heating the saturated vapor at 1 to 1′. Section 1-1′ in the steam generator is called a *superheater*.

The cycles as shown are internally reversible so that the turbine and pump are adiabatic reversible and hence vertical on the *T-S* diagram; no pressure losses occur in the piping so that line 4-*B*-1-1′ is a constant-pressure line.

The analysis of either cycle is straightforward. Based on a unit mass of vapor in the saturated cycle

$$\left.\begin{aligned}
&\text{Heat added } q_A = h_1 - h_4 \qquad \text{Btu/lb}_m \text{ or J/kg}\\
&\text{Turbine work } w_T = h_1 - h_2 \qquad \text{Btu/lb}_m \text{ or J/kg}\\
&\text{Heat rejected } |q_R| = h_2 - h_3 \qquad \text{Btu/lb}_m \text{ or J/kg}\\
&\text{Pump work } |w_p| = h_4 - h_3\\
&\text{Net work } \Delta w_{\text{net}} = (h_1 - h_2) - (h_4 - h_3) \qquad \text{Btu/lb}_m \text{ or J/kg}\\
&\text{Thermal efficiency } \eta_{\text{th}} = \frac{\Delta w_{\text{net}}}{q_A} = \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_4)}
\end{aligned}\right\} \qquad (2\text{-}1)$$

For small units where P_4 is not too large compared with P_3, $h_4 \approx h_3$, the pump work is negligible compared with the turbine work, and the thermal efficiency may be simplified with little error to $(h_1 - h_2)/(h_1 - h_3)$. This is not true for modern steam powerplants where P_4 is 1000 lb_f/in^2 (about 70 bar) or higher, while P_3 is about 1 lb_f/in^2 (0.07 bar). The pump work in this case may be obtained by finding h_3 as the saturated enthalpy of liquid at P_3 from the steam (or other vapor) tables given in Apps. A to F. h_4 is found from subcooled liquid tables at T_4 and P_4. T_4 is nearly equal to T_3, and the latter is usually used in lieu of T_4, which is difficult to obtain (see Sec. 1-5). Finally, a good approximation for the pump work may be obtained from the change in flow work (Example 3, Sec. 1-2). Thus

$$|w_p| \approx v_3(P_4 - P_3) \qquad (2\text{-}2)$$

which should be converted to the same units as in Eq. (2-1) by the use of proper conversion factors, such as multiply by 144 to convert psia (pounds force per square inch absolute) to pounds force per square foot absolute and divide by 778.16 to convert foot pounds force to Btu.

Another parameter of interest in cycle analysis is the *work ratio* WR, which is defined as the ratio of net work to gross work. For the simple Rankine cycle the work ratio is simply $\Delta w_{net}/w_T$.

The superheat cycle 1′-2′-3-4-*B*-1′ is analyzed by use of Eqs. (2-1) and (2-2), except 1′ is to be substituted for 1.

Because of the information it readily gives regarding the turbine and pump processes, the *T-S* diagram is more useful than the *P-V* diagram and is usually preferred when only one is used. The Mollier, or enthalpy-entropy, diagram is another useful diagram. Its utility, however, is restricted to processes involving the turbine because it gives little or no information of the liquid region.

2-3 THE EXTERNALLY IRREVERSIBLE RANKINE CYCLE

External irreversibility, we are reminded, is primarily the result of the temperature differences between the primary heat source, such as the combustion gases from the steam generator furnace or the primary coolant from a nuclear reactor, and the working fluid, and the temperature differences between condensing working fluid and the heat sink fluid, usually the condenser cooling water.

In Fig. 2-3, line *ab* represents the primary coolant in a counterflow heat exchanger with the working fluid 4-*B*-1 in a saturated Rankine cycle. Line *cd* represents the heat sink fluid (condenser cooling water) in a counterflow or parallel-flow heat exchanger with the condensing working fluid 2-3; both types are the same because the latter is at constant temperature.

As can be seen, the temperature differences between line *ab* and 4-*B*-1-1′ and

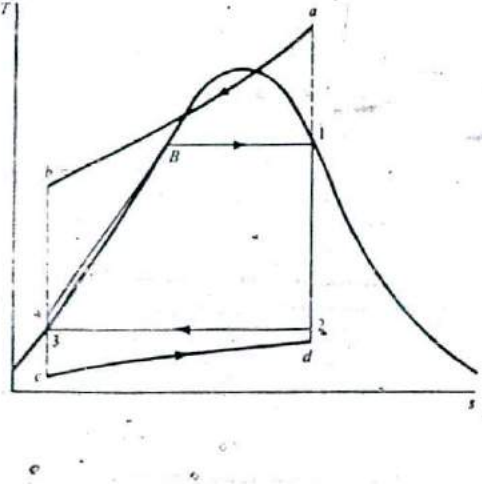
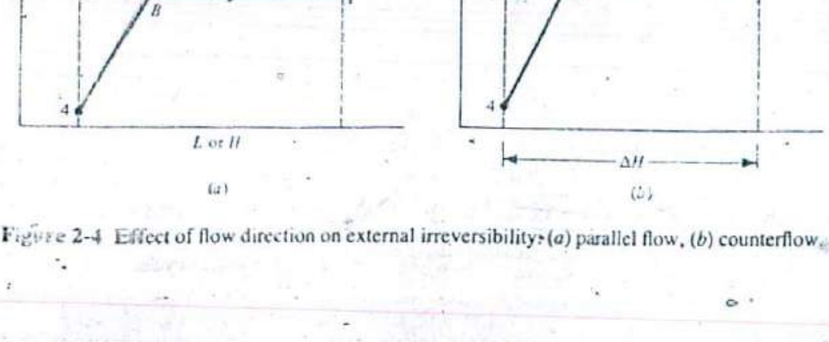


Figure 2-3 External irreversibility with Rankine cycle.

between 2-3 and line *cd* are not constant. We shall evaluate the effects of these differences beginning with the upper end. Figure 2-4 shows temperature-heat exchanger path length diagrams for (*a*) parallel-flow and (*b*) counterflow heat exchangers (steam generators) and the effect of flow directions in the heat exchanger. The minimum approach point between the two lines, called the *pinch point*, represented by *b*-1 and *c*-*B*, must be finite. Too small a pinch-point temperature difference results in low overall temperature differences and, hence, lower irreversibilities, but in a large and costly steam generator; too large a pinch-point temperature difference results in a small, inexpensive steam generator but large overall temperature differences and irreversibilities and, hence, reduction in plant efficiency. The most economical pinch-point temperature difference is obtained by optimization that takes into account both fixed charges (based on capital costs) and operating costs (based on efficiency and, hence, fuel costs).

Figure 2-4, in addition, clearly shows that the overall temperature differences between the heat source and the working fluids are greater in the case of the parallel-flow than counterflow heat exchangers; the result is a less efficient plant if parallel flow is used. Heat-transfer considerations also favor counterflow, resulting in higher overall heat-transfer coefficients and hence small heat exchanger. Thus counterflow is favored over parallel flow from both thermodynamic and heat-transfer considerations.

We will now examine the effect of the type of heat source fluid. Such a fluid may be a gas, such as the combustion gases in a fossil-fueled powerplant, the primary coolant in a gas-cooled reactor, such as CO_2 or He (Sec. 10-11), the water from a pressurized-water reactor (Sec. 10-2), or the molten sodium from a liquid-metal fast-breeder reactor (Chap. 11). This variety of fluids has different specific heats and mass-flow rates. Water from a pressurized-water reactor has a higher specific heat c_p than gases but also a higher mass-flow rate $\dot{m}$ because an effort is made to limit the temperature rise of water through the reactor to maintain nearly even moderation of



(*a*)

(*b*)

Figure 2-4 Effect of flow direction on external irreversibility: (*a*) parallel flow, (*b*) counterflow.

the neutrons (Sec. 9-8). Thus the product $\dot{m}c_p$ is greater in the case of water than in the case of gases.

Assuming that a differential amount of heat dQ exchanged between the two fluids is proportional to a path length dL and that $dQ = \dot{m}c_p\, dT$, where dT is the change in primary-fluid temperature in dL, the slope of line ab is then proportional to the reciprocal of $\dot{m}c_p$ or

$$\frac{dT}{dL} \propto \frac{1}{\dot{m}c_p} \qquad \text{(primary fluid)} \tag{2-3}$$

Hence the slope of line ab for water is much less than that for gases. Liquid sodium falls in between, though closer to gases than to water. This state of affairs is shown in Fig. 2-5 for a counterflow heat exchanger. It can be seen that for a given pinch-point temperature difference, the overall temperature differences between the primary and working fluids are greater in the case of gases than water, in particular in the boiler section, between ae and B-1.

This brings us to an important deduction, namely the determination of whether or not superheat (and reheat) is advantageous. We note that there are two distinct regions where the external irreversibility exists at the higher-temperature end of the cycle. These are: (1) between the primary fluid and the working fluid in the boiler section, i.e., between ae and B-1, and (2) between the primary fluid and the working fluid in the economizer section, i.e., between be and 4-B. We shall deal with these in turn in the next two sections.

There is little that can be done to improve things in the low-temperature end of the cycle, i.e., between 2-3 and cd in the condenser (Fig. 2-3), short of optimizing the condenser to obtain the lowest temperature differences between the two lines. Remember, however, that the lower the temperature of the cooling water at c, the lower the condenser steam temperature and the higher the cycle efficiency.

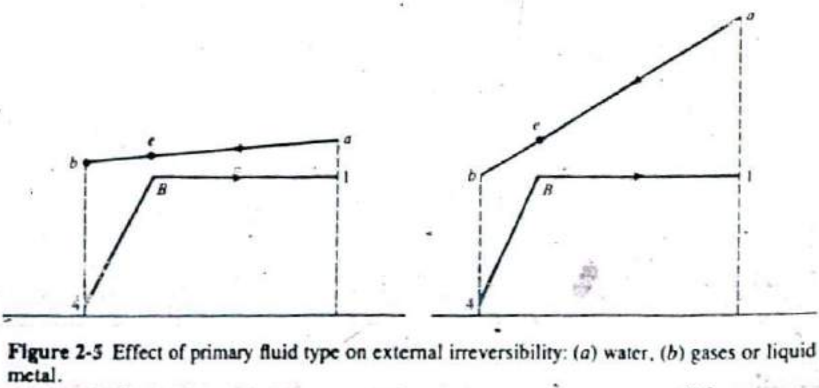


Figure 2-5 Effect of primary fluid type on external irreversibility: (*a*) water, (*b*) gases or liquid metal.

2-4 SUPERHEAT

In this section we will deal with the temperature differences between *ae* and β-1 (Fig. 2-5). It can be seen that these for a given pinch-point temperature difference ΔT_{e-B}, gases (and liquid metals) exhibit larger and increasing temperature differences as the working fluid boils from B-1 than is the case of water where the slope of line *ae* is much lower.

Although the temperature levels are not the same in the two cases, the gases are

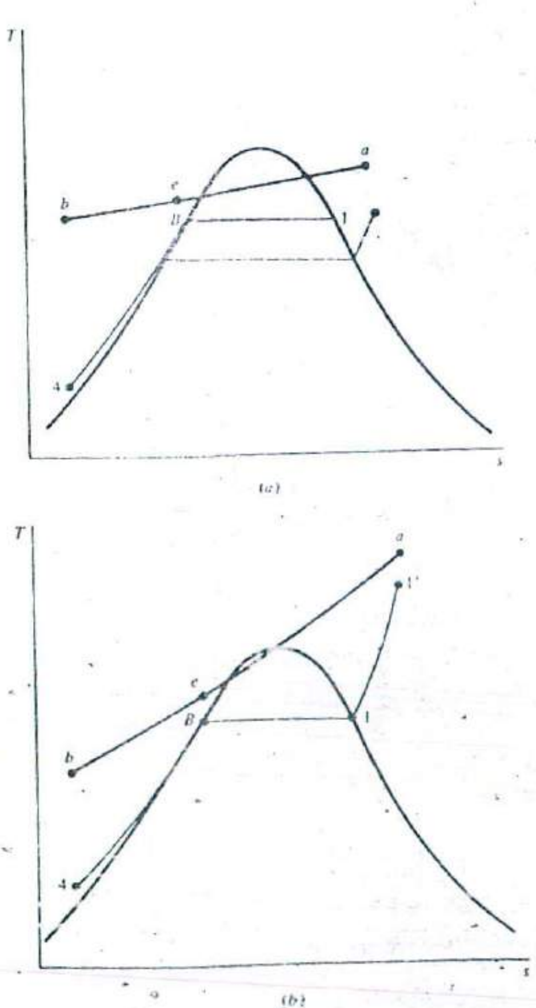


Figure 2-6 Superheat with (*a*) water as primary fluid, (*b*) gases or liquid metal as primary fluid.

usually at higher temperatures, the irreversibility in the case of gases can be reduced by the use of superheat (Fig. 2-6) by bringing the two lines back together again at *a* and 1′ and thus reducing the overall temperature differences between *ae* and *B*-1-1′ (line 4-*B*-1-1′ is a constant-pressure line). Thus superheat would improve the cycle thermal efficiency. Looking at it another way, superheat allows heat addition at an average temperature higher than using saturated steam only. From the Carnot analogy, this should result in higher cycle efficiency.

In the case of water, superheat is not practical because the differences between *ae* and *B*-1 vary little. Actually, if we were to fix the temperature at 1 and use superheat, we would need to lower the boiling temperature (and hence pressure) in *B*-1, as seen by the dashed line in Fig. 2-6*a*. This increases rather than decreases the overall temperature differences and results in reducing rather than increasing cycle efficiency. This is the reason why fossil-fuel and gas-cooled and liquid-metal-cooled nuclear powerplants employ superheat, while pressurized-water-cooled reactors do not. (A boiling-water reactor, Sec. 10-7, produces only saturated steam within the reactor vessel.)

Superheat has an additional beneficial effect. It results in drier steam at turbine exhaust 2′ as compared with 2 for saturated steam (Fig. 2-2 and Example 2-1). A turbine operating with less moisture is more efficient and less prone to blade damage.

Example 2-1 Consider three Rankine steam cycles, all exhausting to 1 psia. Cycle *A* operates at 2500 psia and 1000°F; cycle *B* operates with 2500 psia saturated steam; and cycle *C* operates with superheated steam at a temperature equal to that of cycle *B* but with a pressure of 1000 psia. Calculate the efficiencies and exhaust steam qualities of the three cycles.

SOLUTION Using Eqs. (2-1) and (2-2), and the steam tables, and referring to Fig. 2-2, calculations for cycle *A* are

$$h_{1'} = 1457.5 \text{ Btu/lb}_m \qquad s_{1'} = 1.5269 \text{ Btu/(lb}_m \cdot {}^\circ\text{R)}$$

Because the turbine is reversible adiabatic, its expansion line is isentropic, or $s_{2'} = s_{1'}$. Thus

$$s_{2'} = (s_f + x_2 s_{fg})_{1\text{ psia}}$$

$$1.5269 = 0.1326 + x_2(1.8455)$$

From which quality of turbine exhaust $x_2 = 0.7555$

$$h_{2'} = (h_f + x_2 h_{fg})_{1\text{ psia}} = 69.73 + 0.7555 \times 1036.1$$

$$= 852.5 \text{ Btu/lb}_m$$

$$h_3 = 69.73 \text{ Btu/lb}_m$$

$$|w_p| = h_4 - h_3 = v_3(P_4 - P_3) = \frac{0.016136(2500 - 1) \times 144}{778.16}$$

$$= 7.46 \text{ Btu/lb}_m$$

$$h_4 = 69.73 + 7.46 = 77.19 \text{ Btu/lb}_m$$

$$w_T = h_{1'} - h_{2'} = 1457.5 - 852.5 = 604.98 \text{ Btu/lb}_m$$

$$\Delta w_{\text{net}} = w_T - |w_P| = 604.98 - 7.46 = 597.52 \text{ Btu/lb}_m$$

$$q_A = h_{1'} - h_4 = 1457.5 - 77.19 = 1380.31 \text{ Btu/lb}_m$$

$$|q_R| = h_{2'} - h_3 = 852.5 - 69.73 - 782.77 \text{ Btu/lb}_m$$

$$\eta_{\text{th}} = \frac{\Delta w_{\text{net}}}{q_A} = \frac{597.52}{1380.31} = 0.4329 = 43.29\%$$

$$\text{WR} = \frac{\Delta w_{\text{net}}}{w_T} = \frac{597.52}{604.98} = 0.9877$$

Table 2-1 lists the results for cycle *A* and, using a similar procedure, for cycles *B* and *C*. Cycle *D* is a superheat-reheat cycle that will be discussed in Sec. 2-5. Cycle *E* is a nonideal cycle that will be discussed in Sec. 2-7.

Note that cycle *C* is actually less efficient than cycle *B*, which proves that superheat is not beneficial if the upper temperature is limited.

2-5 REHEAT

An additional improvement in cycle efficiency with gaseous primary fluids as in fossil-fueled and gas-cooled powerplants is achieved by the use of *reheat*.

Figures 2-7 and 2-8 show simplified flow and T-*s* diagrams of an internally reversible Rankine cycle (i.e., one with adiabatic reversible turbine and pump and no pressure drops) that superheats and reheats the vapor.

Table 2-1 Solutions for Examples 2-1, 2-2, and 2-3

	Cycle				
Data	*A* Superheat 2500/1000	*B* 2500 Saturated	*C* Superheat 1000/668.11	*D* 2500/ 1000/1000	*E* 2500/1000 Nonideal
Turbine inlet pressure, psia	2500	2500	1000	2500	2500
Turbine inlet temperature, °F	1000	668.11	668.11	1000	1000
Condenser pressure, psia	1	1	1	1	1
Inlet steam enthalpy, Btu/lb$_m$	1457.5	1093.3	1303.1	1457.5	1457.5
Exhaust steam enthalpy, Btu/lb$_m$	852.52	688.36	834.44	970.5	913.02
Turbine work, Btu/lb$_m$	604.98	404.94	468.66	741.8	544.48
Pump work, Btu/lb$_m$	7.46	7.46	2.98	7.46	11.52
Net work, Btu/lb$_m$	597.52	397.48	465.68	734.34	532.96
Heat added, Btu/lb$_m$	1380.31	1061.11	1230.39	1635.10	1376.25
Exhaust steam quality	0.7555	0.5971	0.7381	0.8694	0.8139
Cycle efficiency, %	43.29	39.42	37.85	44.91	38.73

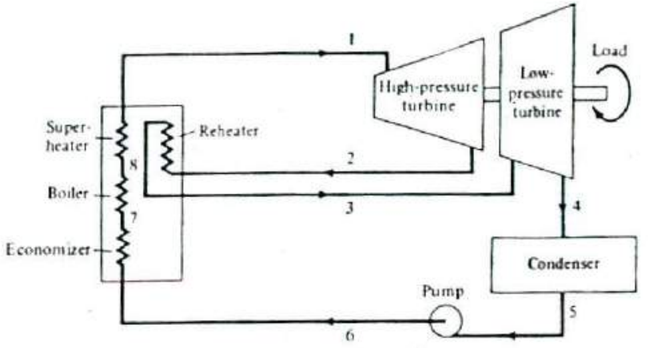


Figure 2-7 Schematic of a Rankine cycle with superheat and reheat.

In the reheat cycle, the vapor at 1 is expanded part of the way in a high-pressure section of the turbine to 2, after which it is returned back to the steam generator, where it is reheated at constant pressure (ideally) to a temperature near that at 1. The reheated steam now expands in the low-pressure section of the turbine to the condenser pressure.

As can be seen reheat allows heat addition twice: from 6 to 1 and from 2 to 3. It results in increasing the average temperature at which heat is added and keeps the boiler-superheat-reheat portion from 7 to 3 close to the primary fluid line *ae*, which

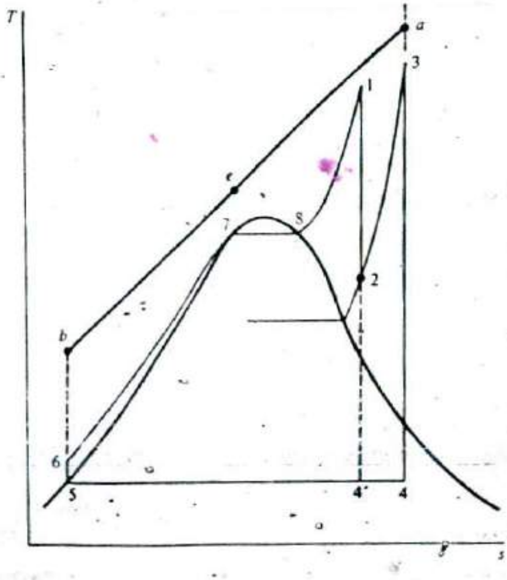


Figure 2-8 *T-s* diagram of Rankine cycle of Fig. 2-7.

results in improvement in cycle efficiency. Reheat also results in drier steam at turbine exhaust (4 instead of 4'), which is beneficial for real cycles.

Modern fossil-fueled powerplants employ superheat and at least one stage of reheat. Some employ two. More than two stages, however, results in cycle complication and increased capital costs that are not justified by improvements in efficiency. Gas-cooled nuclear-reactor powerplants often employ one stage of reheat. Water-cooled and sodium-cooled nuclear-reactor powerplants often employ one stage of reheat, except that the steam to be reheated is not returned to the steam generator. Instead, a separate heat exchanger that employs a portion of the original steam at 1 is used to reheat the steam at 2. That portion condenses and is sent to a feed water heater (Sec. 2-6). Examples of this will be presented in Chaps. 10 and 11.

The analysis of a reheat cycle involves two turbine work terms as well as two heat addition terms. Referring to Fig. 2-8

$$\left.\begin{aligned} w_T &= (h_1 - h_2) + (h_3 - h_4) \\ |w_p| &= h_6 - h_5 \\ \Delta w_{\text{net}} &= (h_1 - h_2) + (h_3 - h_4) - (h_6 - h_5) \\ q_A &= (h_1 - h_6) + (h_3 - h_2) \\ \eta_{\text{th}} &= \frac{\Delta w_{\text{net}}}{q_A} \end{aligned}\right\} \qquad (2\text{-}4)$$

The pressure P_2 at which the steam is reheated affects the cycle efficiency. Figure 2-9 shows the change in cycle efficiency $\Delta\eta$ percent as a function of the ratio of reheat pressure to initial pressure P_2/P_1, for $P_1 = 2500$ psia, $T_1 = 1000$°F, and $T_3 = 1000$°F. $P_2/P_1 = 1.0$ is the case where no reheat is used and hence $\Delta\eta = 0$. A reheat pressure too close to the initial pressure results in little improvement in cycle efficiency because only a small portion of additional heat is added at high temperature. The efficiency improves as the reheat pressure P_2 is lowered and reaches a peak at a pressure ratio P_2/P_1 between 20 and 25 percent. Lowering the reheat pressure further causes the temperature differences between the primary and the working fluids to increase and begin to offset the addition of heat at high temperature, thus causing the efficiency to decrease again. Too low a reheat pressure, in the above case at a pressure ratio of about 0.025, actually results in a negative $\Delta\eta$, i.e., an efficiency below the case of no reheat. The optimum at a pressure ratio of 0.2 to 0.25, calculated for the above conditions, actually holds for most modern powerplants. Figure 2-9 also shows the value of T_2 and x_4. Note that reheat results in drier exhaust steam. Too low a pressure ratio may even result in superheated exhaust steam, an unfavorable situation for condenser operation.

A superheat-reheat powerplant is often designated by $P_1/T_1/T_3$ in pounds force per square inch absolute and degrees Fahrenheit. The above case, for example, is 2500/1000/1000, whereas a double-reheat plant may be designated 2400/1000/1025/1050. The following example shows a sample of the calculations conducted for Fig. 2-8, near the optimum pressure ratio.

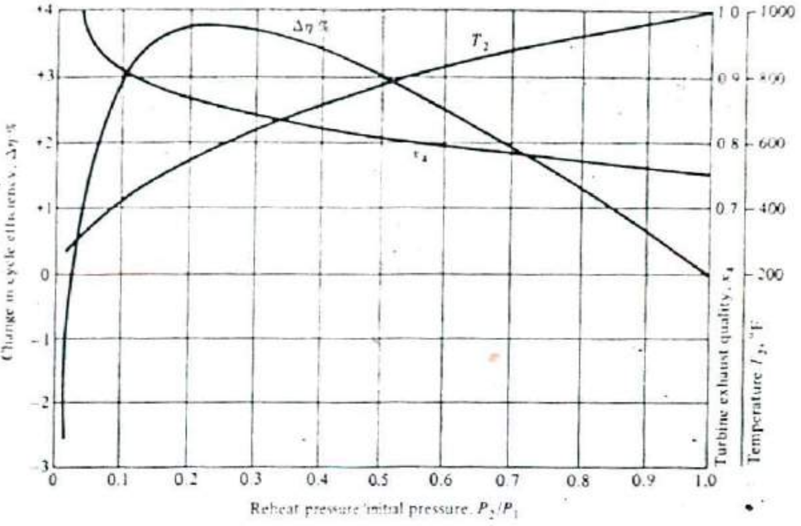


Figure 2-9 Effect of reheat-to-initial pressure ratio on efficiency, high-pressure turbine exit temperature, and low-pressure turbine exit quality. Data for cycle of Fig. 2-7 with initial steam at 2500 psia and 1000°F, and steam reheat to 1000°F (2500/1000/1000)

Example 2-2 Calculate the efficiency and exhaust steam quality of a 2500 psia 1000°F/1000°F internally reversible steam Rankine cycle (cycle *D*, Table 2-1). The reheat pressure is 500 psia. The condenser pressure is 1 psia.

SOLUTION Referring to Fig. 2-8

$$h_1 = 1457.5 \text{ Btu/lb}_m \qquad s_1 = 1.5269 = s_2 > s_g \text{ at 500 psia}$$

Therefore point 2 is in the superheat region. By interpolation

$$T_2 = 547.8°\text{F} \qquad h_2 = 1265.6 \text{ Btu/lb}_m$$

At 500 psia and 1000°F

$$h_3 = 1520.3 \text{ Btu/lb}_m \qquad s_3 = 1.7371 = s_4$$

Therefore $\qquad x_4 = 0.8694 \qquad h_4 = 970.5 \text{ Btu/lb}_m$

As in Example 2-1, $|w_p| = 7.46$ Btu/lb$_m$ and $h_6 = 77.19$ Btu/lb$_m$. Using Eqs. (2-4) gives

$$w_T = 191.5 + 549.8 = 741.7 \text{ Btu/lb}_m$$

$$\Delta w_{\text{net}} = 741.7 - 7.46 = 734.24 \text{ Btu/lb}_m$$

$$q_A = 1380.3 + 254.7 = 1635.0 \text{ Btu/lb}_m$$

and $$\eta_{th} = \frac{734.24}{1635.0} = 0.4491 = 44.91\%$$

This cycle is compared with the previous cycles in Table 2-1. It shows the highest efficiency and driest exhaust steam of all in that table.

2-6 REGENERATION

We have so far discussed means of reducing the external irreversibility caused by the heat transfer between the primary fluid and the working fluid beyond the point of boiling of the latter (point *B*, Figs. 2-3 and 2-4*b*). An examination of these figures shows that a great deal of such irreversibility occurs prior to the point of boiling, i.e., in the economizer section of the steam generator where the temperature differences between *bd* and 4-*B* are the greatest of all during the entire process of heat addition. The slope of the primary-fluid temperature line is of less concern here than in the boiler section because it has a relatively minor effect on the temperature differences in the economizer. Hence, all types of powerplants, fossil-fuel, liquid-metal, gas- or water-cooled nuclear-reactor powerplants, suffer nearly equally from this irreversibility.

This irreversibility can be eliminated if the liquid is added to the steam generator at *B* rather than at 4. This can be done by the process of *regeneration*, in which internal heat is exchanged between the expanding fluid in the turbine and the compressed fluid before heat addition. A well-known gas cycle that uses regeneration is the Stirling cycle, shown on the T-*s* diagram of Fig. 2-10. The ideal Stirling cycle is composed of heat addition at constant temperature 2-3 and heat rejection at constant temperature 4-1. Regeneration or heat exchange occurs reversibly between the constant volume processes 3-4 and 1-2, i.e., between portions of each curve that are at the same temperature. This heat exchange does not figure in the cycle efficiency because it is not obtained from an external source. The areas under 3-4 and 1-2 denoting heat lost by the expanded fluid and gained by the compressed fluid are equal in magnitude, though not in sign. The ideal Stirling cycle has the same efficiency as the Carnot cycle operating between the same temperature limits. This would not have been the case had heat been added from an external source during 1-2 and 2-3 and rejected to an external sink between 3-4 and 4-1.

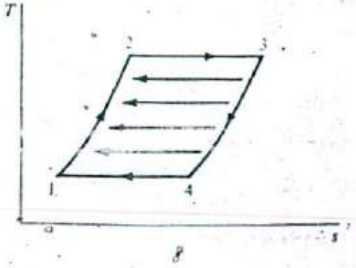


Figure 2-10 *T*-*s* diagram of Stirling cycle. Regeneration occurs between 3-4 and 1-2. Arrows indicate heat exchange.

Adopting the same procedure to a Rankine cycle, i.e., internal and reversible heat exchange from the expanding working fluid in the turbine and the fluid in the economizer section, would necessitate flow and T-s diagrams as shown in Fig. 2-11 for a saturated Rankine cycle. The compressed liquid at 4 would have to be carefully passed around the turbine to receive heat from the expanding vapor in the turbine reversibly at all times (i.e., with zero temperature difference) until it enters the steam generator at B. The steam generator would have no economizer and the irreversibility during heat addition to the economizer would be eliminated. The resulting Rankine cycle would receive and reject heat at constant temperature and, in the absence of other external irreversibilities, would also have the same efficiency as the Carnot cycle operating between the same temperature limits. Hence the great need for eliminating or minimizing the economizer irreversibility.

The ideal procedure of Fig. 2-11 is not practically possible. The vapor making its way through blade passages cannot be made to have adequate heat-transfer surface between it and the compressed liquid, which by necessity would have to be wrapped around the external turbine casing. Even if an adequate surface were possible, the mass-flow rates are so large that the effectiveness of such a heat exchanger would be low. Further, the vapor leaving the turbine would have an unacceptably high moisture content (low quality) for proper turbine operation and efficiency.

Feedwater Heating

A compromise that would reduce rather than eliminate the economizer irreversibility is accomplished by the use of *feedwater heating* (the more general term feed liquid heating that would apply to fluids other than H_2O is seldom used). Feedwater heating involves normal adiabatic (and ideally also reversible) expansion in the turbine. The compressed liquid at 4 is heated in a number of finite steps, rather than continuously, by vapor bled from the turbine at selected stages. Heating of the liquid takes place in heat exchangers called *feedwater heaters*. Feedwater heating dates back to the early 1920s, around the same time that steam temperatures reached about 725°F. Modern

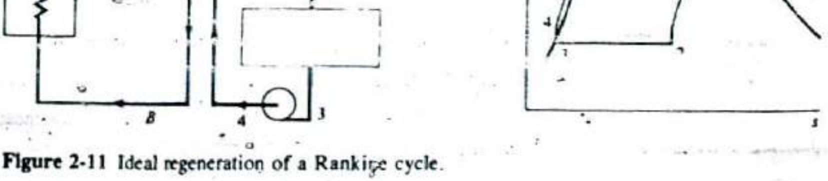


Figure 2-11 Ideal regeneration of a Rankine cycle.

large steam powerplants use between five and eight feedwater heating stages. None is built without feedwater heating.

Because of the finite number of feedwater heating stages, the liquid enters the steam generator at a point below *B*, necessitating an economizer section, though one that is much smaller than if no feedwater heating were used. Because of this, and because the feedwater heaters have irreversibilities of their own, the ideal situation of Fig. 2-11 is not attained and the Rankine cycle cannot attain a Carnot efficiency. A well-designed Rankine cycle, however, is the closest practical cycle to Carnot, and hence its wide acceptance for most powerplants.

There are three types of feedwater heaters in use. These are:

1. Open or direct-contact type
2. Closed type with drains cascaded backward
3. Closed type with drains pumped forward

These types will be discussed and analyzed in detail in this chapter beginning with Sec. 2-8. Their physical design will be described in Chap. 6.

2-7 THE INTERNALLY IRREVERSIBLE RANKINE CYCLE

Internal irreversibility is primarily the result of fluid friction, throttling, and mixing. The most important of these are the irreversibilities in turbines and pumps and pressure losses in heat exchangers, pipes, bends, valves, etc.

In the turbine and pumps, the assumption of adiabatic flow is still valid because the flow rates are so large that the heat losses per unit mass is negligible. However, they are no longer adiabatic reversible, and the entropy, in both, increases. This is shown in Fig. 2-12.

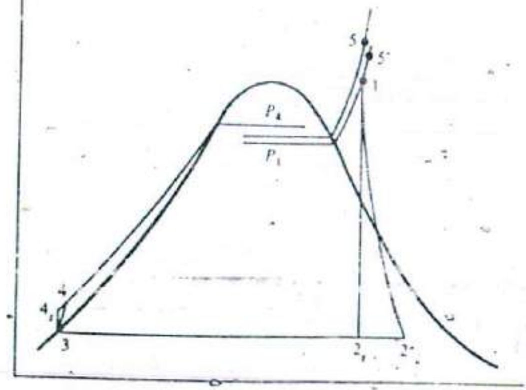


Figure 2-12 A *T-s* diagram of an internally irreversible superheat Rankine cycle.

The entropy increase in the turbine, unlike that in a gas turbine (Fig. 1-8), does not result in a temperature increase if exhaust is to the two-phase region, the usual case. Instead it results in an increase in enthalpy. Thus the ideal expansion, if the turbine were adiabatic reversible, is 1-2_s, but the actual expansion is 1-2. The irreversible losses in the turbine are represented by a turbine efficiency η_T, called the *turbine polytropic efficiency* (and sometimes the adiabatic or isentropic efficiency). This is not to be confused with the cycle thermal efficiency. η_T is given by the ratio of the turbine actual work to the ideal, adiabatic reversible work. Hence

$$\eta_T = \frac{h_1 - h_2}{h_1 - h_{2s}} \tag{2-5}$$

Well-designed turbines have high polytropic efficiencies, around 90 percent. η_T usually increases with turbine size and suffers from moisture in the steam. η_T as given above is an overall polytropic efficiency. However, individual turbine stages have different efficiencies, being higher for early stages where the steam is drier. There will be more on turbines in Chap. 5.

No pressure losses are encountered in the condenser process 2-3 (Fig. 2-12) because it is a two-phase condensation process.

The pump process, being adiabatic and irreversible, also results in an increase in entropy. A single-phase (liquid) process, it results in an increase in temperature and enthalpy. Thus the actual work $h_4 - h_3$ is greater than the adiabatic reversible work $h_{4s} - h_3$. In other words, one pays a penalty for irreversibility: the turbine produces less work, the pump absorbs more work. The pump irreversibility is also represented by a pump efficiency η_p, also called a *pump polytropic efficiency* (and sometimes adiabatic or isentropic efficiency). η_p is given by the ratio of the ideal work to the actual work, the reverse of that for the turbine. Thus

$$\eta_p = \frac{h_{4s} - h_3}{h_4 - h_3} \tag{2-6}$$

In both Eqs. (2-5) and (2-6), the smaller quantity is in the numerator. The actual pump work may now be obtained by modifying Eq. (2-6) to

$$|w_p| = \frac{h_{4s} - h_3}{\eta_p} \approx \frac{v_3(P_4 - P_3)}{\eta_p} \tag{2-7}$$

The liquid leaving the pump must be at a higher pressure than at the turbine inlet because of the friction drops in heat exchangers, feedwater heaters, pipes, bends, valves, etc. Thus P_4 represents the exit pump pressure, P_1 represents the turbine inlet pressure, and P_5 represents the steam-generator exit pressure. The steam leaves the generator at 5 and enters the turbine at 1. The path 5-1 is the result of the combined effects of friction and heat losses. Point 5′ at pressure P_1 represents frictional effects in the pipe connecting steam generator and turbine, including turbine throttle valve, if any. Heat losses from that pipe cause a decrease in entropy to 1. Pressure losses between 4 and 1 could be of the order of a few hundred pounds force per square inch.

Example 2-3 A superheat steam Rankine cycle has turbine inlet conditions of 2500 psia and 1000°F. The turbine and pump polytropic efficiencies are 0.9 and 0.7, respectively. Pressure losses between pump and turbine inlet are 200 psi. Calculate the turbine exhaust steam quality and cycle efficiency.

SOLUTION Referring to Fig. 2-12

$$h_1 = 1457.5 \qquad h_{2s} = 852.52 \text{ Btu/lb}_m \text{ (as in Example 2-1)}$$

$$w_T = \eta_T(h_1 - h_{2s}) = 0.9 \times 604.98 = 544.48 \text{ Btu/lb}_m$$

Therefore
$$h_2 = h_1 - w_T = 913.02 \text{ Btu/lb}_m$$

At 1 psia
$$913.02 = 69.73 + x_2(1036.1)$$

$$\therefore x_2 = 0.8139$$

$$P_4 = P_1 + 200 = 2700 \text{ psia}$$

Thus
$$|w_P| \approx \frac{v_3(P_4 - P_3)}{\eta_P} = \frac{0.016136(2700 - 1) \times 144}{778 \times 0.7}$$

$$\doteq 11.52 \text{ Btu/lb}_m$$

$$h_4 = h_3 + |w_P| = 69.73 + 11.52 = 81.25 \text{ Btu/lb}_m$$

$$\Delta w_{\text{net}} = w_T - |w_P| = 532.96 \text{ Btu/lb}_m$$

$$q_A = h_1 - h_4 = 1376.25$$

Therefore
$$\eta_{\text{th}} = \frac{\Delta w_{\text{net}}}{q_A} = 0.3873 = 38.73\%$$

Thus the internal irreversibilities have resulted in reducing the cycle efficiency from 43.29 percent (Example 2-1) to 38.73 percent, but in an increase in exhaust steam quality from 0.7555 to 0.8139, one beneficial effect of an imperfect turbine. This example is listed as cycle *E* in Table 2-1.

2-8 OPEN OR DIRECT-CONTACT FEEDWATER HEATERS

In the open- or direct-contact-type of feedwater heater the extraction steam is mixed directly with the incoming subcooled feedwater to produce saturated water at the extraction steam pressure. Figure 2-13*a* and *b* shows a schematic flow diagram, and the corresponding T-*s* diagram for a Rankine cycle using, for simplicity of illustration, two such feedwater heaters, one low-pressure and one high-pressure (normally one open-type feedwater heater and between four and seven other heaters are used in modern large powerplants). The physical construction of such a feedwater heater is covered in Chap. 7. A typical open-type feedwater heater is shown in Fig. 6-15.

The condensate water leaves the condenser saturated at 5 and is pumped to 6 to

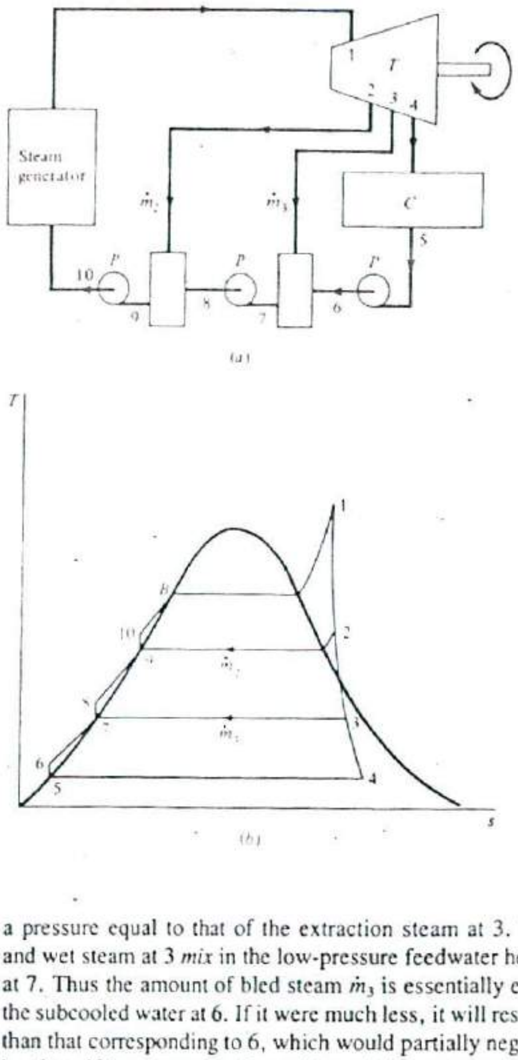


Figure 2-13 Schematic flow and T-s diagrams of a nonideal superheat Rankine cycle with two open-type feedwater heaters.

a pressure equal to that of the extraction steam at 3. The now-subcooled water at 6 and wet steam at 3 *mix* in the low-pressure feedwater heater to produce saturated water at 7. Thus the amount of bled steam $\dot{m}_3$ is essentially equal to that that would saturate the subcooled water at 6. If it were much less, it will result in a much lower temperature than that corresponding to 6, which would partially negate the advantages of feedwater heating. If it were more, it would result in unnecessary loss of turbine work and in a two-phase mixture that would be difficult to pump.

Line 6-7 in Fig. 2-13*b* is a constant-pressure line. (In practice some pressure drop is encountered.) The difference between it and the saturated liquid line 5-*B* is exaggerated for illustration purposes.

The pressure at 6-7 can be no higher than the extraction steam pressure at 3 (or else reverse flow of condensate water would enter the turbine at 3). A second pump must therefore be used to pressurize the saturated water from 7 to a subcooled condition at 8, which is at the pressure of extraction steam at 2. In the high-pressure feedwater heater, superheated steam at 2 mixes with subcooled water at 8 to produce saturated water at 9. This now must be pressurized to 10 in order to enter the steam generator at its pressure.

Because the extracted steam, at 2 or 3, loses a large amount of energy, roughly equal to its latent heat of vaporization, while water, at 6 or 8, gains sensible heat, the amount of extracted steam $\dot{m}_2$ or $\dot{m}_3$ is only a small fraction of the steam passing through the turbine. Note, however, that the mass-flow rate through the turbine is a variable quantity, highest between 1 and 2 and lowest between 3 and 4.

It can also be seen that besides the condensate pump 5-6, one additional pump per open feedwater heater is required.

Open-type feedwater heaters also double as deaerators because the breakup of water in the mixing process helps increase the surface area and liberates noncondensible gases (such as air, O_2, H_2, CO_2) that can be vented to the atmosphere (Sec. 6-7). Hence they are sometimes called *deaerating heaters*, or DA.

In order to analyze the system shown in Fig. 2-13, both a mass balance and an energy balance must be considered. The mass balance, based on a unit-flow rate (1 lb_m/h or kg/s) at throttle (point 1) is given, clockwise, by

$$\left.\begin{aligned}
&\text{Mass flow between 1 and 2} = 1\\
&\text{Mass flow between 2 and 9} = \dot{m}_2\\
&\text{Mass flow between 2 and 3} = 1 - \dot{m}_2\\
&\text{Mass flow between 3 and 7} = \dot{m}_3\\
&\text{Mass flow between 4 and 7} = 1 - \dot{m}_2 - \dot{m}_3\\
&\text{Mass flow between 7 and 9} = 1 - \dot{m}_2\\
&\text{Mass flow between 9 and 1} = 1
\end{aligned}\right\} \qquad (2\text{-}8)$$

where $\dot{m}_2$ and $\dot{m}_3$ are small fractions of 1. Energy balances are now done on the high- and low-pressure feedwater heaters, respectively

$$m_2(h_2 - h_9) = (1 - \dot{m}_2)(h_9 - h_8) \qquad (2\text{-}9)$$

and

$$\dot{m}_3(h_3 - h_7) = (1 - \dot{m}_2 - \dot{m}_3)(h_3 - h_7) \qquad (2\text{-}10)$$

where h is the enthalpy per unit mass at the point of interest. Equations (2-9) and (2-10) show that there are two equations and only two unknowns, $\dot{m}_2$ and $\dot{m}_3$, if the pressures at which steam is bled from the turbine (Sec. 2-13), and therefore the enthalpies, are all known. For any number of feedwater heaters there will be as many equations as there are unknowns, so solutions are always possible. A large number of feedwater heaters would, of course, require the solution of an equal number of simultaneous linear algebraic equations on a digital computer. The pertinent cycle parameters are now obtained, as energy per unit mass-flow rate at turbine inlet (point 1)

$$\left.\begin{aligned}
\text{Heat added } q_A &= (h_1 - h_{10}) \\
\text{Turbine work } w_T &= (h_1 - h_2) + (1 - \dot{m}_2)(h_2 - h_3) \\
&\quad + (1 - \dot{m}_2 - \dot{m}_3)(h_3 - h_4) \\
\text{Pump work } |\Sigma w_p| &= (1 - \dot{m}_2 - \dot{m}_3)(h_6 - h_5) + (1 - \dot{m}_2)(h_8 - h_7) \\
&\quad + (h_{10} - h_9) = (1 - \dot{m}_2 - \dot{m}_3)\frac{v_5(P_6 - P_5)}{\eta_p J} \\
&\quad + (1 - \dot{m}_2)\frac{v_7(P_8 - P_7)}{\eta_p J} + \frac{v_9(P_{10} - P_9)}{\eta_p J} \\
\text{Heat rejected } |q_R| &= (1 - \dot{m}_2 - \dot{m}_3)(h_4 - h_5) \\
\text{Net cycle work } \Delta w_{\text{net}} &= w_T - |w_p| \\
\text{Cycle thermal efficiency } \eta_{\text{th}} &= \frac{\Delta w_{\text{net}}}{q_A} \\
\text{Work ratio WR} &= \frac{w_{\text{net}}}{w_T}
\end{aligned}\right\} \qquad (2\text{-}11)$$

where η_p is the pump efficiency and $J = 778.16$ ft-lb$_f$/Btu.

Example 2-4 An ideal Rankine cycle operates between 2500 psia and 1000°F at throttle and 1 psia in the condenser. One open-type feedwater heater is placed at 200 psia. Assuming 1 lb$_m$/h flow at turbine throttle and no flow pressure drops, calculate the mass-flow rate in the heater and the pertinent parameters for the cycle and compare them with those of the cycle in Example 2-1, which has the same conditions except that no feedwater heater was used.

SOLUTION Referring to Fig. 2-14 and the steam tables

$$h_1 = 1457.5 \text{ Btu/lb}_m \qquad s_1 = 1.5269 \text{ Btu/(lb}_m \cdot {}^\circ\text{F)}$$

At 200 psia

$$s_2 = s_1 = 1.5269 = 0.5438 + x_2(1.0016)$$

Therefore

$$x_2 = 0.9815 \qquad h_2 = 355.5 + 0.9815\,(842.8) = 1182.7 \text{ Btu/lb}_m$$

At 1 psia

$$s_3 = s_1 = 1.5269 = 0.1326 + x_3(1.8455)$$

Thus

$$x_3 = 0.7555 \qquad h_3 = 69.73 + 0.7555(1036.1) = 852.2 \text{ Btu/lb}_m$$

$$h_4 = 69.73 \text{ Btu/lb}_m \qquad v_4 = 0.016136 \text{ ft}^3\text{/lb}_m$$

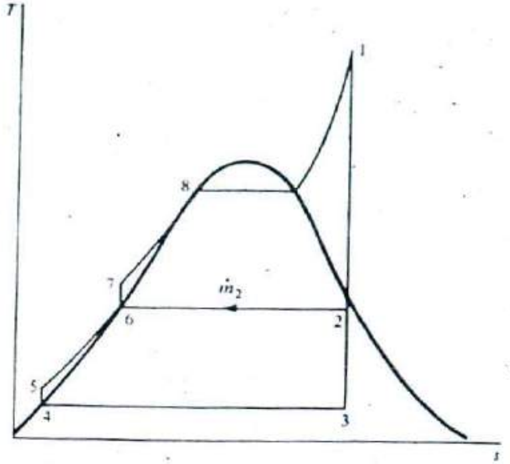


Figure 2-14 *T-s* diagram for Example 2-4

$$h_5 = 69.73 + \frac{0.016136 \times (200 - 1) \times 144}{778.16} = 69.73 + 0.59$$

$$= 70.32 \text{ Btu/lb}_m$$

$$h_6 = 355.5 \text{ Btu/lb}_m \qquad v_6 = 0.01839 \text{ ft}^3\text{/lb}_m$$

$$h_7 = 355.5 + 0.01839 + \frac{(2500 - 200) \times 144}{778.16} = 355.5 + 7.83$$

$$= 363.3 \text{ Btu/lb}_m$$

$$\dot{m}_2(h_2 - h_6) = (1 - \dot{m}_2)(h_6 - h_5)$$

$$\dot{m}_2(1182.7 - 355.5) = (1 - \dot{m}_2)(355.5 - 70.32)$$

$$\therefore \dot{m}_2 = 0.2564$$

$$w_T = (h_1 - h_2) + (1 - \dot{m}_2)(h_2 - h_3)$$

$$= (1457.5 - 1182.7) + (1 - 0.2564)(1182.7 - 852.5)$$

$$= 274.77 + 245.57 = 520.34 \text{ Btu/lb}_m$$

$$|\Sigma w_p| = (1 - \dot{m}_2)(h_5 - h_4) + (h_7 - h_6)$$

$$= (1 - 0.2564)(0.59 + 7.83) = 8.27 \text{ Btu/lb}_m$$

$$\Delta w_{\text{net}} = w_T - |\Sigma w_p| = 520.34 - 8.27 = 512.07 \text{ Btu/lb}_m$$

$$q_A = h_1 - h_7 = 1457.5 - 363.3 = 1094.2 \text{ Btu/lb}_m$$

$$|q_R| = (1 - \dot{m}_2)(h_3 - h_4) = (1 - 0.2564)(852.5 - 69.73)$$

$$= 582.1 \text{ Btu/lb}_m$$

$$\eta_{th} = \frac{\Delta w_{net}}{q_A} = \frac{512.07}{1094.2} = 0.468 = 46.8\%$$

$$\mathrm{WR} = \frac{\Delta w_{net}}{w_T} = \frac{512.07}{520.34} = 0.984$$

Compare this with cycle *A* (Table 2-1), which had no feedwater heater. Note that the turbine work is decreased for the same mass-flow rate at throttle because of reduced turbine mass-flow rate after bleeding and that the pump work is increased. Note also the greater decrease in heat added, which more than makes up for the loss of net work, resulting in a marked improvement in cycle efficiency. This improvement increases as the number of feedwater heaters is increased. The number of feedwater heaters can be as high as seven or eight. An increase beyond that causes little increase in efficiency but adds complications and increased capital costs and thus diminishes returns.

As seen above, open feedwater heaters require, in addition to the condensate pump, as many additional pumps as there are feedwater heaters. Each of these pumps carries nearly full flow, or more accurately full flow minus the bled steam following it. For example, pump 7-8 (Fig. 2-13) carries $(1 - \dot{m}_2)$ $\mathrm{lb}_m/\mathrm{lb}_m$ at throttle. In powerplants such large flow pumps are the source of operational, service, and noise problems and increase plant complexity and cost. In general only one open-type feedwater heater is used, which doubles up as a deaerating heater, followed by a pump called the *boiler feed pump*. (In some nuclear powerplants no open feedwater heaters are used and degassing is done elsewhere.) Other feedwater heaters in the system are therefore of the closed type.

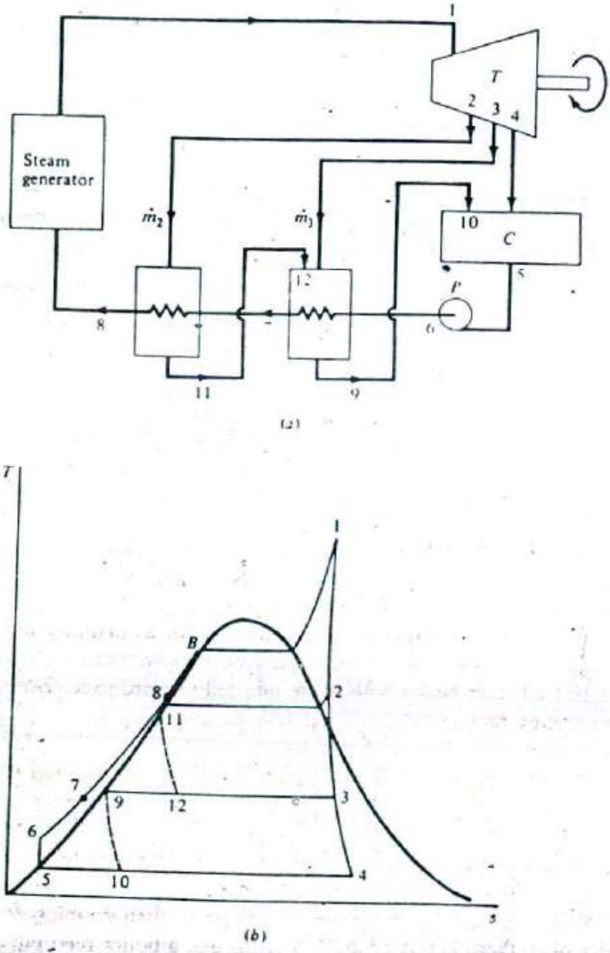
2-9 CLOSED-TYPE FEEDWATER HEATERS WITH DRAINS CASCADED BACKWARD

This type of feedwater heater, though it results in a greater loss of availability than the open type, is the simplest and most commonly used type in powerplants. As in the case of the closed-type feedwater heater with drains pumped forward (Sec. 2-10), it too is a shell-and-tube heat exchanger but differs because of the lack of any moving equipment.

In a closed-type feedwater heater (of either type), feedwater passes through the tubes, and the bled steam, on the shell side, transfers its energy to it and condenses. Thus they are, in essence, small condensers that operate at pressures more elevated than those of the main plant condenser. Because the feedwater goes through the tubes in successive closed feedwater heaters, it does not mix with bled steam and therefore can be pressurized only once by the first condensate pump, which then doubles as a boiler feed pump, though often there is one condensate pump and a boiler feed pump placed downstream to reduce the pressure rise in each pump. A boiler feed pump is automatically required and placed after the deaerating heater if one is used in the plant.

Figure 2-15 shows a simplified flow diagram and corresponding T-s diagram of a nonideal superheat Rankine cycle showing, for simplicity, two feedwater heaters of this type. One pump, 5-6, pressurizes the condensate to a pressure sufficient to pass through the two feedwater heaters and enter the steam generator at 8$'$. Again the difference between the high-pressure line 6-B and the saturated-liquid line 5-B is exaggerated for illustration purposes.

As the bled steam condenses in each feedwater heater, it cannot, of course,



(a)

(b)

Figure 2-15 Schematic flow and T-s diagrams of a nonideal superheat Rankine cycle with two closed-type feedwater heaters with drains cascaded backward.

accumulate there and must be removed and fed back to the system. In this type of feedwater heater, the condensate is fed back to the next lower-pressure feedwater heater. The condensate of the lowest-pressure feedwater heater is (though not always) led back to the main condenser. One can imagine, then, a cascade from higher-pressure to lower-pressure heaters; hence, the name of this type of feedwater heater.

Again starting with the low-pressure feedwater heater, wet steam at 3 is admitted and transfers its energy to high-pressure subcooled water at 6. The events in that heater can be represented by the temperature-length diagram shown in Fig. 2-16*a*. The water exit temperature at 7 cannot reach the inlet bled steam temperature at 3. A difference called the *terminal temperature difference* (TTD, sometimes simply TD) is defined for all closed feedwater heaters as

$$\text{TTD} = \textit{saturation}\text{ temperature of bled steam} - \text{exit water temperature} \qquad (2\text{-}12)$$

The value of TTD varies with heater pressure. In the case of low-pressure heaters, which receive wet or at most saturated bled steam, the TTD is positive and often of the order of 5°F. This difference is obtained by proper heat-transfer design of the heater. Too small a value, although good for plant efficiency, would require a larger heater than can be justified economically. Too large a value would hurt cycle efficiency. In some heaters, the drain at 9 is slightly subcooled. This will be shown later.

The drain from the low-pressure heater is now led to the condenser and enters it as a two-phase mixture at 10. This is a throttling process from the pressure corresponding to 9 to that of the main condenser, and hence there is loss of some availability, as alluded to earlier. There is also some loss of availability as a result of heat transfer. Process 9-10 is a throttling process and hence is a constant enthalpy one.

A closed feedwater heater that receives saturated or wet steam can have a drain cooler and thus be physically composed of a condensing section and a drain cooler section (Fig. 2-16*b*).

Returning to the system of Fig. 2-15, the high-pressure feedwater heater receives superheated steam bled from the turbine at 2 that flows on the shell side at the rate $\dot{m}_2$ and transfers its energy to subcooled liquid entering the tubes at 7. The events

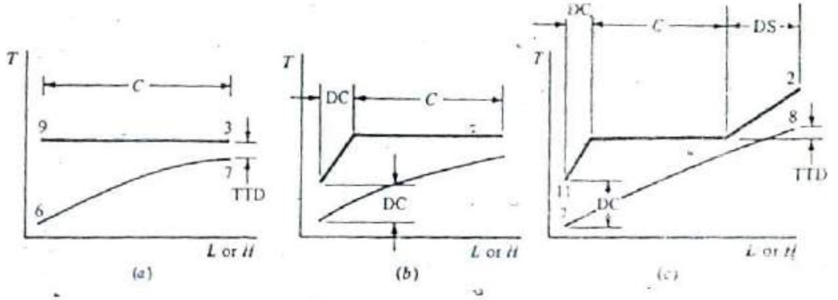


Figure 2-16 Temperature-enthalpy diagrams of (*a*) and (*b*) low-pressure and (*c*) high-pressure feedwater heaters of Fig. 2-15. TTD = terminal temperature difference, DS = desuperheater, C = condenser, DC = drain cooler.

there are shown by the temperature-path length diagram in Fig. 2-16c. Note here that because the inlet steam is superheated at 2, the exit water temperature at 8 can be higher than the saturation temperature of that steam and the TTD, defined, by Eq. (2-12), can be negative. The TTD values for high-pressure heaters, therefore, range between 0 and −5°F, being more negative the higher the pressure, and hence the greater the degree of superheat of the entering steam.

Note also that the drain in this heater is slightly subcooled and hence imparts more energy to the water and thus reduces the loss of availability due to its throttling to the low-pressure heater. The heater is physically composed of a desuperheating section, a condensing section, and a drain cooler section (Fig. 2-16c).

Thus there are four physical possibilities of closed feedwater heaters composed of the following sections or zones (Sec. 6-5):

1. Condenser
2. Condenser, drain cooler
3. Desuperheater, condenser, drain cooler
4. Desuperheater, condenser

The drain at 11 is now throttled to the low-pressure heater entering it at 12 as a two-phase mixture where it joins with the steam bled at 3 and thus aids in the heating of the water in the low-pressure heater. The combined $\dot{m}_2 + \dot{m}_3$ constitutes the low-pressure heater drain, which is throttled to the main condenser at 10. The high-pressure heater exit water at 8 is led into the steam generator. Again, to analyze the system, both a mass and an energy balance are required. A mass balance, also based on a unit-flow rate at turbine inlet, point 1, is given, clockwise, by

$$\left.\begin{aligned}
&\text{Mass flow between 1 and 2} &&= 1\\
&\text{Mass flow between 2 and 3} &&= 1 - \dot{m}_2\\
&\text{Mass flow between 3 and 10} &&= 1 - \dot{m}_2 - \dot{m}_3\\
&\text{Mass flow between 10 and 1} &&= 1\\
&\text{Mass flow between 2 and 12} &&= \dot{m}_2\\
&\text{Mass flow between 3 and 12} &&= \dot{m}_3\\
&\text{Mass flow between 12 and 10} &&= \dot{m}_2 + \dot{m}_3
\end{aligned}\right\} \qquad (2\text{-}13)$$

The energy balances on the high- and low-pressure heaters are now given, respectively, by

$$\dot{m}_2(h_2 - h_{11}) = h_8 - h_7 \qquad (2\text{-}14)$$

and

$$\dot{m}_3(h_3 - h_9) + \dot{m}_2(h_{12} - h_9) = h_7 - h_6 \qquad (2\text{-}15)$$

Recalling that a throttling process is a constant enthalpy process so that

$$h_{12} = h_{11} \qquad \text{and} \qquad h_{10} = h_9$$

and knowing the pressures at which steam is bled from the turbine (Sec. 2-13) so that the enthalpies in Eqs. (2-14) and (2-15) are all known, we again have two equations and two unknowns, $\dot{m}_2$ and $\dot{m}_3$. Or, in general, we will have as many equations as there are unknowns making a solution possible. The pertinent cycle parameters are now obtained, again as energy per unit mass flow rate at turbine inlet (point 1)

$$\left.\begin{aligned}
&\text{Heat added } q_A = h_1 - h_8 \\
&\text{Turbine work } w_T = (h_1 - h_2) + (1 - \dot{m}_2)(h_2 - h_3) \\
&\qquad\qquad + (1 - \dot{m}_2 - \dot{m}_3)(h_3 - h_4) \\
&\text{Pump work } |w_p| = h_6 - h_5 \approx \frac{v_5(P_6 - P_5)}{\eta_p J} \\
&\text{Heat rejected } |q_R| = (1 - \dot{m}_2 - \dot{m}_3)(h_4 - h_5) + (\dot{m}_2 + \dot{m}_3)(h_{10} - h_5) \\
&\text{Net cycle work } \Delta w_{\text{net}} = w_T - |w_p| \\
&\text{Cycle thermal efficiency } \eta_{\text{th}} = \frac{\Delta w_{\text{net}}}{q_A} \\
&\text{Work ratio WR} = \frac{\Delta w_{\text{net}}}{w_T}
\end{aligned}\right\} \qquad (2\text{-}16)$$

Example 2-5 An ideal Rankine cycle operates with 1000 psia, 1000°F steam. It has one closed feedwater heater with drain cascaded backward placed at 100 psia. The condenser pressure is 1 psia. Use TTD = 5°F. The heater has a drain cooler resulting in DC (drain cooler temperature difference) = 10°F.

SOLUTION Referring to Fig. 2-17, the enthalpies, all in Btu/lb$_m$, found by the usual procedure are

$h_1 = 1505.4 \qquad h_2 = 1228.6 \qquad h_3 = 923.31 \qquad h_4 = 69.73 \qquad h_7 = 298.5$

$h_5 = h_4 + v_4(P_5 - P_4) = 69.73 + 2.98 = 72.71$ corresponding to 104.72°F

For TTD = 5°F

$$t_6 = t_7 - 5 = 327.82 - 5 = 322.82°\text{F}$$

Therefore

$$h_6 = 293.36 \qquad \text{(by interpolation)}$$

For DC = 10°F

$$t_8 = t_5 + 10 = 104.72 + 10 = 114.72°\text{F}$$

Thus $h_8 = 82.69$ (by interpolation)

$$\dot{m}_2(h_2 - h_8) = h_6 - h_5$$

$$\dot{m}_2 = \frac{393.36 - 72.71}{1228.6 - 82.69} = 0.1926$$

$$
\begin{aligned}
w_T &= (h_1 - h_2) + (1 - \dot{m}_2)(h_2 - h_3) \\
&= (1505.4 - 1228.6) + (1 - 0.1926)(1228.6 - 923.31) \\
&= 276.8 + 246.49 = 523.29 \\
|w_p| &= (h_5 - h_4) = 2.98 \\
\Delta w_{net} &= 520.31 \\
q_A &= h_1 - h_6 = 1505.4 - 293.36 = 1212.04 \\
|q_R| &= (1 - \dot{m}_2)(h_3 - h_4) + \dot{m}_2(h_9 - h_4) \\
&= 689.18 + 2.50 = 691.68 \\
\eta_{cycle} &= \frac{520.31}{1212.04} = 0.4293 = 42.93\% \\
\text{WR} &= \frac{\Delta w_{net}}{w_T} = \frac{520.31}{523.29} = 0.9943
\end{aligned}
$$

Table 2-2 contains other solutions for ideal Rankine cycles with 1000 psia steam. The cycle in Example 2-5 is cycle *D* in that table. Again note the reduction in work but the improvement in η_{th} over the cycle with no feedwater heating. As stated for the open feedwater heaters, this improvement increases with the number of feedwater heaters until increases in complexity and capital cost make the addition of further heaters, beyond about seven or eight, unprofitable.

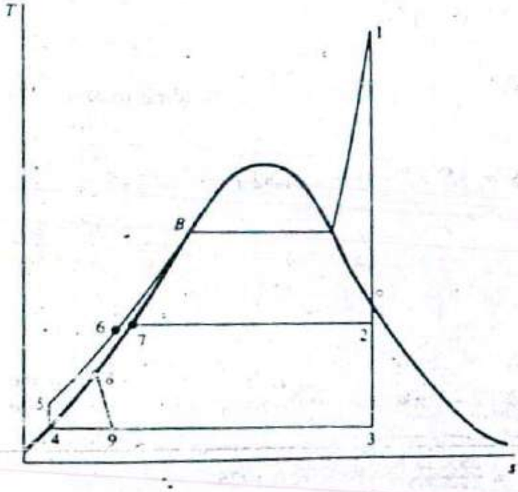


Figure 2-17 *T-s* diagram of Example 2-5.

Table 2-2 Results of example calculations for ideal Rankine cycles*

Cycle	Particulars	Δw_{net}	q_A	η_{th}, %	$\lvert q_R \rvert$	WR
A	No superheat; no fwh†	413.72	1120.19	36.93	706.49	0.9928
B	Superheat; no fwh	579.11	1432.69	40.42	853.58	0.9949
C	Superheat; one open fwh	519.3	1203.95	43.13	685.25	0.9939
D	Superheat; one closed fwh, drains cascaded, DC	520.31	1212.04	42.93	691.68	0.9943
E	Superheat; one closed fwh, drains pumped, DC	529.85	1245.63	42.54	715.73	0.9945
F	Superheat; one closed fwh; drains pumped; no DC	520.59	1210.48	43.01	689.95	0.9943
G	Superheat; reheat; one open fwh	641.59	1447.44	44.33	805.83	0.9951
H	Superheat; reheat; two closed fwh; drains cascaded	609.83	1351.0	45.14	727.62	0.9952
I	Supercritical; double reheat; no fwh; 3500/1000/1025/1050	861.95	1831.92	47.05	969.97	0.9880

* All values in Btu/lb$_m$; all examples, except for cycle *A* which is saturated, and cycle I, at 1000 psia/1000°F. All at 1 psia condenser pressure.

† fwh = feedwater heater.

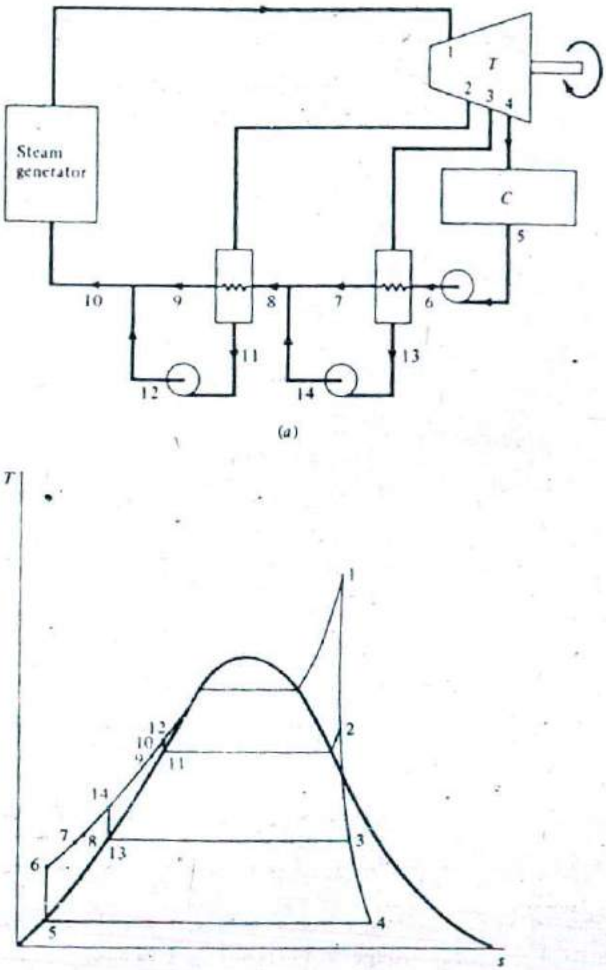
Although this type of feedwater heater is the most common, it causes some loss of availability because of throttling and, to a lesser extent, heat transfer.

2-10 CLOSED-TYPE FEEDWATER HEATERS WITH DRAINS PUMPED FORWARD

This second closed-type feedwater heater avoids throttling but at the expense of some added complexity because of the inclusion of a small pump. It also allows some flexibility to the plant cycle designer who prefers a mix of feedwater heater types that would be deemed most suitable.

As with the previous closed-type feedwater heater, it is a shell-and-tube heat exchanger in which the feedwater passes through the tubes and the bled steam, on the shell side, transfers its energy to it and condenses. They do not mix and the feedwater may be pressurized only once, although a deaerating heater followed by boiler feed pump are usually inserted into the system.

The drain from this type of heater, instead of being cascaded backward, is pumped forward into the main feedwater line. Figure 2-18 shows a simplified flow diagram and corresponding T-*s* diagram for a nonideal superheat Rankine cycle showing, for simplicity, two heaters of this type. Although this system requires one additional pump per heater, it differs from the system using open-type feedwater heaters in that the pumps this time are small and, rather than nearly full feedwater flow, carry only fractional flows corresponding to the bled steam $\dot{m}_2$ and $\dot{m}_3$.



(a)

(b)

Figure 2-18 Schematic flow and T-s diagrams of nonideal superheat Rankine cycle with two closed-type feedwater heaters with drains pumped forward.

Starting with the low-pressure heater, the drain at 13 is pumped forward to the main feedwater line, enters it at 14, and mixes with the exit water from that heater at 7, resulting in a mixture at 8. Point 8 is closer to 7 than 14 on the T-s diagram because the main feedwater flow at 7 is greater than the drain flow $\dot{m}_3$.

The water at 8 enters the high-pressure heater and is heated to 9. The drain leaves the heater at 11, is pumped to 12, and mixes with the feedwater at 9, resulting in full feedwater flow at 10 which now goes to the steam generator.

A mass balance, based on a unit mass-flow rate at turbine inlet, point 1, is given, clockwise, on the T-s diagram by

$$\left.\begin{aligned}
&\text{Mass flow between 1 and 2} &&= 1\\
&\text{Mass flow between 2 and 12} &&= \dot{m}_2\\
&\text{Mass flow between 2 and 3} &&= 1 - \dot{m}_2\\
&\text{Mass flow between 3 and 14} &&= \dot{m}_3\\
&\text{Mass flow between 3 and 7} &&= 1 - \dot{m}_2 - \dot{m}_3\\
&\text{Mass flow at 14} &&= \dot{m}_3\\
&\text{Mass flow between 8 and 9} &&= 1 - \dot{m}_2\\
&\text{Mass flow at 12} &&= \dot{m}_2\\
&\text{Mass flow between 10 and 1} &&= 1
\end{aligned}\right\} \qquad (2\text{-}17)$$

The energy balances on the high- and low-pressure heaters are given, respectively, by

$$\dot{m}_2(h_2 - h_{11}) = (1 - \dot{m}_2)(h_9 - h_8) \qquad (2\text{-}18)$$

and

$$\dot{m}_3(h_3 - h_{13}) = (1 - \dot{m}_2 - \dot{m}_3)(h_7 - h_6) \qquad (2\text{-}19)$$

The values of h_9 and h_7 are obtained from the temperatures t_9 and t_7, which are equal to the saturation temperature of the steam in each heater minus its terminal temperature difference or

$$t_9 = t_{11} - \text{TTD} \qquad \text{hp heater} \qquad (2\text{-}20a)$$

$$\text{and } t_7 = t_{13} - \text{TTD} \qquad \text{lp heater} \qquad (2\text{-}20b)$$

h_{10}, needed for q_A, and h_8, to be used in Eq. (2-18), are obtained from h_{12} and h_{14}, respectively. The latter are given by

$$h_{12} = h_{11} + v_{11}\frac{P_{12} - P_{11}}{\eta_p} \qquad (2\text{-}21a)$$

and

$$h_{14} = h_{13} + v_{13}\frac{P_{14} - P_{13}}{\eta_p} \qquad (2\text{-}21b)$$

Thus

$$h_{10} = \dot{m}_2 h_{12} + (1 - \dot{m}_2)h_9 \qquad (2\text{-}22a)$$

and

$$(1 - \dot{m}_2)h_8 = \dot{m}_3 h_{14} + (1 - \dot{m}_2 - \dot{m}_3)h_7 \qquad (2\text{-}22b)$$

The turbine work

$$w_T = (h_1 - h_2) + (1 - \dot{m}_2)(h_2 - h_3) + (1 - \dot{m}_2 - \dot{m}_3)(h_3 - h_4) \qquad (2\text{-}23)$$

Pump work $|\Sigma w_p| = (1 - \dot{m}_2 - \dot{m}_3)(h_6 - h_5) + \dot{m}_3(h_{14} - h_{13}) + \dot{m}_2(h_{12} - h_{11})$ (2-24)

Heat added $q_A = h_1 - h_{10}$ (2-25)

Thermal efficiency $\eta_{th} = \dfrac{w_T - |\Sigma w_p|}{q_A}$ (2-26)

Example 2-6 Repeat Example 2-5 but for one closed-type feedwater heater with drain pumped forward. TTD = 5°F.

SOLUTION Refer to Fig. 2-19. h_1, h_2, h_3, h_4, h_5, h_6, h_7 are all the same as in Example 2-5

$$h_8 = h_7 + v_7 \frac{(P_8 - P_7) \times 144}{778.13} = 298.5 + 0.017740 \frac{(1000 - 100) \times 144}{778.17}$$

$$= 298.5 + 2.95 = 301.45 \text{ Btu/lb}_m$$

$$h_6 \text{ (as before)} = 293.36 \text{ Btu/lb}_m$$

$$\dot{m}_2(h_2 - h_7) = (1 - \dot{m}_2)(h_6 - h_5)$$

$$\dot{m}_2(1228.6 - 298.5) = (1 - \dot{m}_2)(293.36 - 72.71)$$

$$\therefore \dot{m}_2 = 0.1917$$

$$h_9 = \dot{m}_2 h_8 + (1 - \dot{m}_2)h_6 = 57.79 + 237.12 = 294.91 \text{ Btu/lb}_m$$

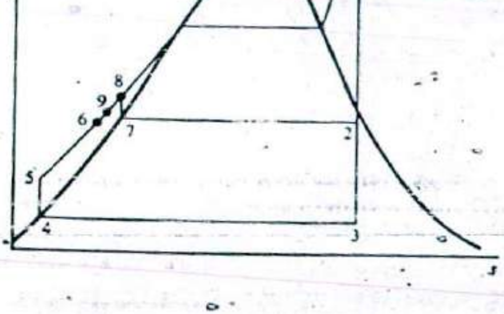


Figure 2-19 *T-s* diagram of Example 2-6.

$$w_T = (h_1 - h_2) + (1 - \dot{m}_2)(h_2 - h_3) = 276.8 + 246.77$$

$$= 523.57 \text{ Btu/lb}_m$$

$$\Sigma w_p = (1 - \dot{m}_2)(h_5 - h_4) + \dot{m}_2(h_8 - h_7) = 2.41 + 0.57$$

$$= 2.98 \text{ Btu/lb}_m$$

$$\Delta w_{\text{net}} = 520.59 \text{ Btu/lb}_m$$

$$q_A = h_1 - h_{10} = 1505.4 - 294.92 = 1210.48 \text{ Btu/lb}_m$$

$$|q_R| = (1 - \dot{m}_2)(h_3 - h_4) = 689.95 \text{ Btu/lb}_m$$

$$\eta_{\text{cycle}} = \frac{520.59}{1210.48} = 0.4301 = 43.01\%$$

$$\text{WR} = \frac{520.59}{523.57} = 0.9943$$

This example is listed as cycle *F* in Table 2-2.

As indicated earlier, the type of closed feedwater heater that has drains pumped forward avoids the loss of availability due to throttling inherent in the previous closed feedwater heater with drains cascaded backward. This, however, is done at the expense of the complexity of adding a drain pump following each heater. Note, however, that unlike the open feedwater heater the drain pump is a low-capacity one because its flow is only that of the bled steam being condensed in the heater. It must however pressurize that condensate to the full feedwater line pressure.

This type of feedwater heater results in a slightly better cycle efficiency if used without a drain cooler because energy transferred from the heater drain in the drain cooler lowers the point in the feedwater line at which energy is to be added from the primary heat source or from a higher pressure feedwater heater. Compare cycle *F* in Table 2-2 with cycle *E*, which is identical except that there is a drain cooler with DC = 10°F.

One other advantage of pumped drains is that, when used as the lowest-pressure feedwater heater in an otherwise all-cascaded system, or with all-cascaded feedwater heaters between it and an open feedwater heater, it prevents the throttling of the combined cascaded flows to the condenser pressure where the energy left in that combined flow is lost to the environment.

2-11 THE CHOICE OF FEEDWATER HEATERS

In general the choice of feedwater heater type depends upon many factors, including designer optimization and preference, practical considerations, cost, and so on, and one sees a variety of cycle designs. There are, however, features that are rather common.

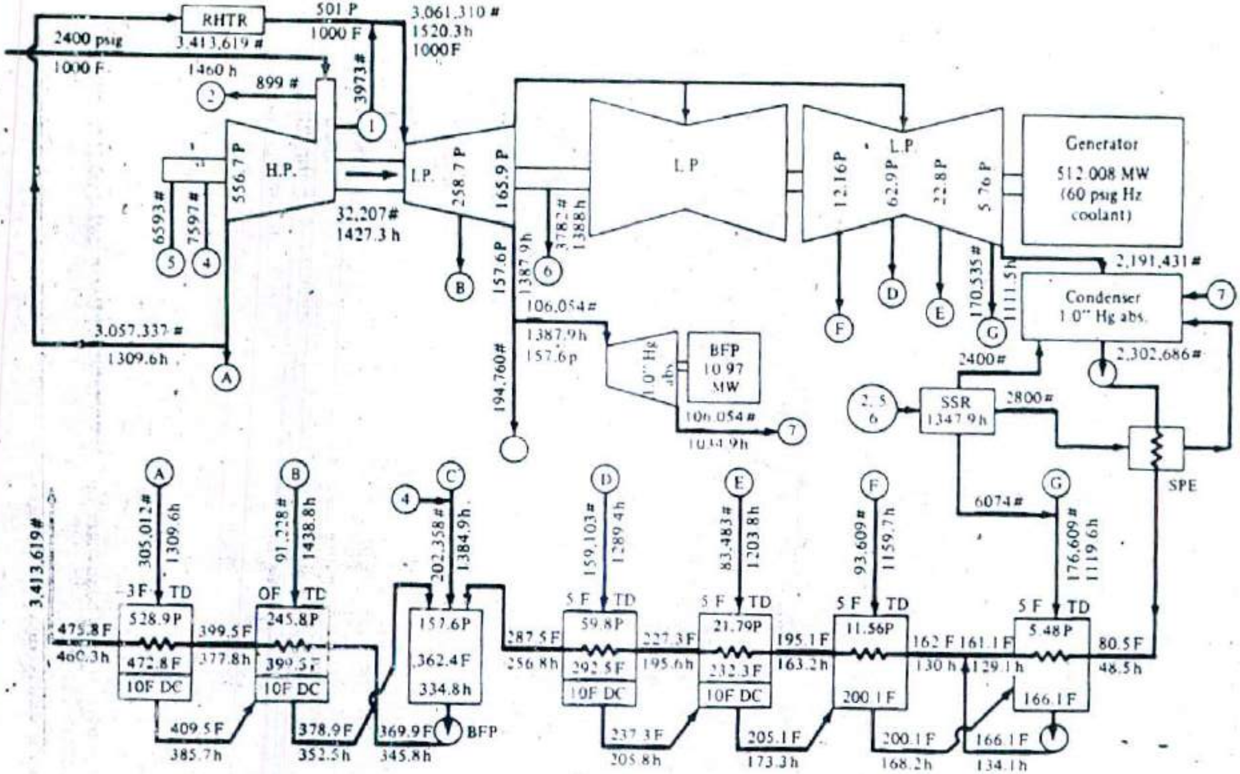


Figure 2-20 Flow diagram of an actual 512-MW 2400 psig/1000°F/1000°F reheat powerplant with seven feedwater heaters. *(Courtesy Wisconsin Power & Light Co.)*

1. One open-type feedwater heater, which doubles as a deaerator and is thus called the DA (deaerating) heater, is used in fossil-fueled powerplants. It is not yet the practice to use it in water-cooled-and-moderated nuclear powerplants because of the concern regarding radioactivity release with deaeration. This type of heater is usually placed near the middle of the feedwater system, where the temperature is most conducive to the release of noncondensables.
2. The closed-type feedwater heater with drains cascaded backward is the most common type, used both before and after the DA heater. It usually has integral desuperheating and drain cooler sections in the high-pressure stages but no superheating section in the very low-pressure stages because the bled steam is saturated or wet. A separate drain cooler is sometimes used for the lowest-pressure heater.
3. One closed feedwater heater with drains pumped forward is often used as the lowest-pressure feedwater heater to pump all accumulating drains back into the feedwater line, as indicated above. Occasionally one encounters one more feedwater of this type at a higher-pressure stage.

Table 2-2 is a compilation of the results of calculations similar to and including those in the previous examples. They all have 1000 psia, 1000°F steam at turbine inlet, except for cycle *A*, which is saturated. Cycles *G* and *H* have reheat to 1000°F. Cycles *A*, *B*, and *I* have no feedwater heaters. The rest have one feedwater of various types except for cycle *H*, which has two. All cycles are ideal, meaning that they are internally reversible with adiabatic reversible turbines and pumps.

Comparison shows large efficiency increases as a result of superheat, reheat, and the use of even one feedwater heater. The differences between different types of feedwater heaters are small. It is to be noted, however, that even a fraction of a percent difference in efficiency can mean a very large difference in annual fuel costs, especially in a fossil powerplant, where the fuel cycle costs are a large portion of the total cost of electricity. (Other costs are the fixed charges on the capital cost and the operation and maintenance cost, O & M.) Differences in efficiency also mean differences in plant size (heat exchangers, etc.) for a given plant output and hence differences in capital cost. Although the cycles summarized in Table 2-2 are ideal, the trends they exhibit are applicable to nonideal cycles, so one should expect the same relative standings in both cases.

Figure 2-20 shows a flow diagram of an actual 512-MW powerplant with superheat, reheat, and seven feedwaters: one DA, five closed with drains cascaded backward, and one, the lowest pressure, closed with drains pumped forward. In such diagrams, there are standard notations (not all to be found in Fig. 2-20), such as

AE	Available energy or isentropic enthalpy difference, Btu/lb_m
BFP	Boiler feed pump
DC	Drain cooler terminal temperature difference (Fig. 2-16*b* and *c*), °F
EL	Exhaust loss, Btu/lb_m
ELEP	Expansion line end-point enthalpy, Btu/lb_m
h	Enthalpy, Btu/lb_m

P	Pressure, psia
RHTR	Reheater
SGFP	Steam generator feed pump
SJAE	Steam-jet air ejector condenser
SPE	Steam packing exhaust condenser
SSR	Steam seal regulator
TD or TTD	Terminal temperature difference (Fig. 2-16), °F
UEEP	Used energy end point, Btu/lb_m
#	Mass-flow rate, lb_m/h

2-12 EFFICIENCY AND HEAT RATE

In the thermodynamic analysis of cycles and powerplants, the thermal efficiency and the power output are of prime importance. The *thermal efficiency* is the ratio of the net work to the heat added to the cycle or powerplant. The thermal efficiencies of powerplants are less than those computed for cycles as above because the analyses above failed to take into account the various auxiliaries used in a powerplant and the various irreversibilities associated with them. A complete analysis of a powerplant must take into account all these auxiliaries, the nonidealities in turbines, pumps, friction, heat transfer, throttling, etc., as well as the differences between full-load and partial-load operation. Such analyses are quite complex and require the use of high-capacity computers.

The *gross efficiency* is the one calculated based on the gross work or power of the turbine-generator. This is the work or power, MW gross, produced before power is tapped for the internal functioning of the powerplant, such as that needed to operate pumps, compressors, fuel-handling equipment, and other auxiliaries, labs, computers, heating systems, lighting, etc. (Fig. 2-21). The *net efficiency* is calculated based on

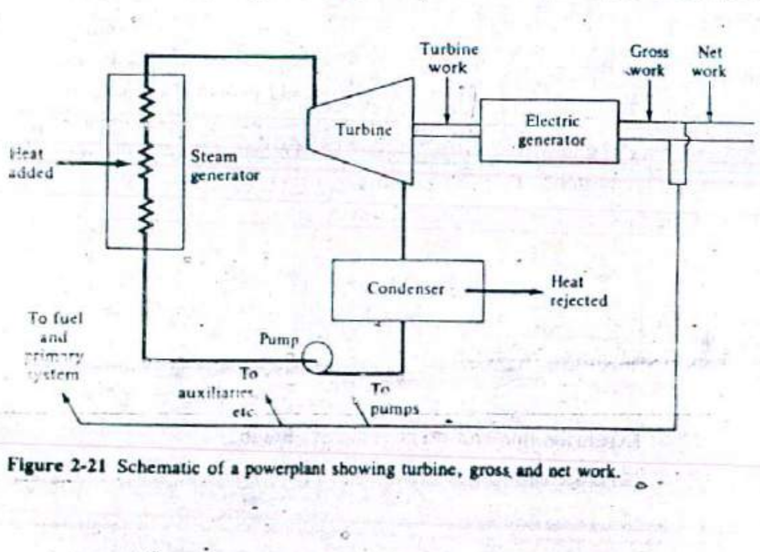


Figure 2-21 Schematic of a powerplant showing turbine, gross and net work.

the net work or power of the plant, i.e., the gross power minus the tapped power, above, or the power leaving at the station bus bars.

Powerplant designers and operators are interested in efficiency as a measure of the economy of the powerplant because it affects capital, fuel, and operating costs. They use in addition another parameter that more readily reflects the fuel economies. That parameter is called a *heat rate* (HR). It is the amount of heat added, usually in Btu, to produce a unit amount of work, usually in kilowatt hours (kWh). Heat rate thus has the units Btu/kWh. The HR is inversely proportional to the efficiency, and hence the lower its value, the better. There are various heat rates corresponding to the work used in the denominator. For example

$$\text{Net cycle HR} = \frac{\text{heat added to cycle, Btu}}{\text{net cycle work kWh}}$$

$$= \frac{\text{rate of heat added to cycle, Btu/h}}{\text{net cycle power, kW}}$$

$$\text{Gross cycle HR} = \frac{\text{rate of heat added to cycle, Btu/h}}{\text{turbine power output, kW}}$$

$$\text{Net station HR} = \frac{\text{rate of heat added to steam generator, Btu/h}}{\text{net station power, kW}}$$

$$\text{Gross station HR} = \frac{\text{rate of heat added to steam generator, Btu/h}}{\text{gross turbine-generator power, kW}}$$

and there are as many thermal efficiencies as there are heat rates. Because 1 kWh = 3412 Btu, the heat rate of any kind is related to the corresponding thermal efficiency by

$$\text{HR} = \frac{3412}{\eta_{th}} \tag{2-27}$$

Example 2-7 A coal-fired powerplant has a turbine-generator rated at 1000 MW gross. The plant requires about 9 percent of this power for its internal operations. It uses 9800 tons of coal per day. This coal has a heating value of 11,500 Btu/lb$_m$, and the steam generator efficiency is 86 percent. Calculate the gross station, net station, and the net steam cycle heat rates.

SOLUTION

$$\text{Rate of coal burned} = 9800 \times \frac{2000}{24} = 816{,}667 \ \text{lb}_m/\text{h}$$

$$\text{Gross station HR} = \frac{816{,}667 \times 11{,}500}{1000 \times 1000} = 9391.67 \ \text{Btu/kWh}$$

$$\text{Station net power output} = (1 - 0.9) \times 1000 = 910 \ \text{MW}$$

$$\text{Net station HR} = \frac{816{,}667 \times 11{,}500}{910 \times 1000} = 10{,}320.5 \text{ Btu/kWh}$$

$$\text{Heat added to steam generator} = 816{,}667 \times 11{,}500 \times 0.86$$

$$= 8.07683 \times 10^9 \text{ Btu/h}$$

$$\text{Net steam cycle HR} = \frac{8.07683 \times 10^9}{0.91 \times 10^6} = 8875.64 \text{ Btu/kWh}$$

The corresponding thermal efficiencies are

$$\text{Gross station efficiency} = \frac{3412}{9391.67} = 36.33\%$$

$$\text{Net station efficiency} = \frac{3412}{10{,}320.5} = 33.06\%$$

$$\text{Net cycle efficiency} = \frac{3412}{8875.64} = 38.44\%$$

When the efficiency and heat rate of a powerplant are quoted without specification, it is usually the net station efficiency and heat rate that are meant. A convenient numerical value to remember for heat rate is 10,000 Btu/kWh. Usually large modern and efficient powerplants have values less than 10,000, while older plants, gas-turbine plants, and alternative power systems such as solar, geothermal, and others, exceed this value.

Figure 2-22, originally published in 1954 [9], contains a history of steam cycles since 1915 and an interesting prediction of things to come, up to 1980. It gives the average overall (net) HR range or band as a function of steam conditions, shown above the band. The heat rates are in turn dependent upon metallurgical constraints and development. The available materials are shown below the band. A landmark station was the 325-MW Eddystone unit 1 of the Philadelphia Electric Company, a double-reheat plant designed for operation with supercritical steam (Sec. 2-14) at 5000 psig/1200°F/1050°F/1050°F (about 345 bar, 650°C/565°C/565°C). Its actual operation was at 4700 psig and 1130°F turbine inlet (325 bar, 610°C). Built in 1959, it had the highest steam conditions and lowest HR of any plant in the world, and its power output was equal to the largest commercially available plant at the time.

Figure 2-22 is shown to predict conditions far beyond what has been achieved to date. The material *X* needed to raise the pressures and temperatures to the 7500 psig and 1400°F level, for example, remains to be developed. The most common steam conditions remain at 2400 to 3500 psia (165 to 240 bar) and 1000 to 1050°F (540 to 565°C). The 1960s and 1970s saw little improvements because there was no motivation to lower heat rates with the then-cheap fossil fuels and the advent of nuclear power. In fact, recent years have seen a rise in heat rates as a result of environmental restrictions on cooling and the increased use of devices to reduce the environmental impact of power generation (cooling towers, electrostatic precipitators, desulfurization, etc.).

Figure 2-22, however, correctly predicts advancements such as single and double

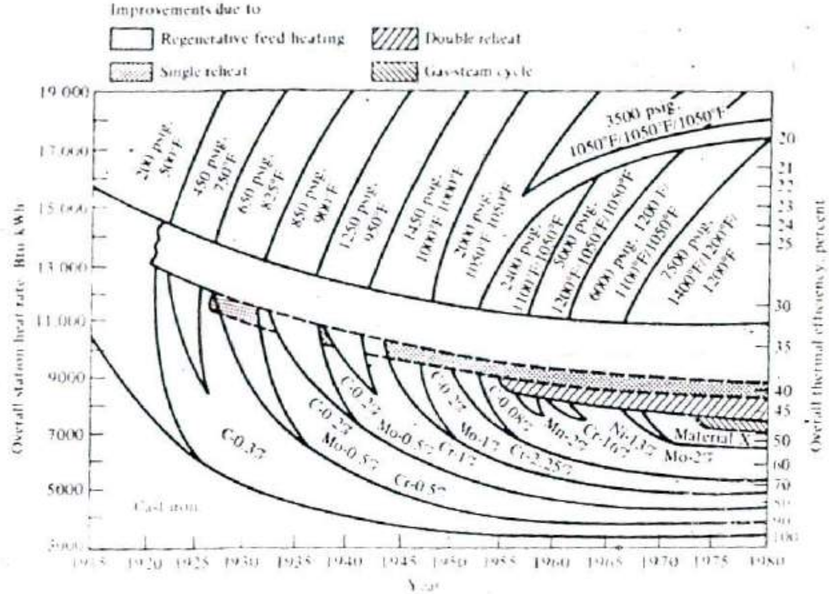


Figure 2-22 The evolution of the steam cycle as predicted in 1954 [9].

reheat and combined gas-turbine-steam-turbine cycles (Sec. 8-10). An advanced 773-MW plant design utilizing double reheat, supercritical steam at 4500 psig/1100/1050/1050°F (310 bar, 593/565/565°C), 10 feedwater heaters, and other novel features, and yielding a heat rate of 8335 Btu/kWh, has recently been proposed [10].

2-13 THE PLACEMENT OF FEEDWATER HEATERS

A natural question arises as to where to place the feedwater heaters (of any kind) in the cycle. In other words: What are the pressures at which steam is to be bled from the turbine that will result in the maximum increase in efficiency (or maximum reduction in heat rate)? It is expected that the answer to this question can be obtained most accurately by a complete optimization of the cycle, a job that entails large, complex, and usually not readily available computer programs.

There is, however, a simple answer based on physical reasoning. As indicated previously, the role of feedwater heaters is to bring the temperature of the feedwater as close as possible to that of the steam generator before the feedwater enters that steam generator. If we were to assume first for simplicity that only one feedwater heater (the type is not important for this discussion) is to be used, we may consider placing it in positions 1, 2, or 3 with respect to the cycle (Fig. 2-23). In position 1

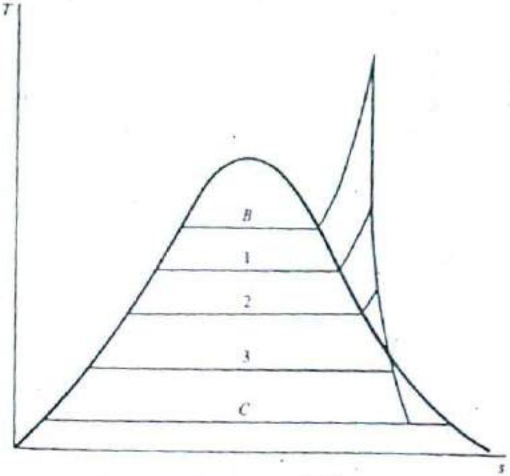


Figure 2-23 One feedwater heater in three possible positions.

we see that heat transfers to the feedwater are caused by ΔT_{B-1} and ΔT_{1-C}, where T_B and T_C are the boiler and condenser temperatures, respectively. In position 3 the corresponding heat transfers are the result of $T_B - T_3$ and $T_3 - T_C$. It is obvious that in both these cases one of these ΔT's is very large. The one position that would minimize both temperature differences is in the middle, position 2, where $T_B - T_2 = T_2 - T_C$. Thus the optimum, from an efficiency point of view, of the *pressure* at which the one feedwater heater is to be placed is obtained by finding the *temperature* that is half way between T_B and T_C and then obtaining the *saturation pressure* corresponding to that temperature. Note that the temperature at which steam is actually bled from the turbine may be in the superheat region at that pressure and thus higher than T_2.

If two feedwater heaters are to be used, the optimum placement is at temperatures that would divide $T_B - T_C$ into three equal parts. In general, then, for *n* feedwater heaters (Fig. 2-24), the optimum temperature rise per heater would be given by

$$\Delta T_{\text{opt}} = \frac{T_B - T_C}{n + 1} \tag{2-28}$$

Example 2-8 The Rankine cycle shown in Fig. 2-24 has an ideal turbine that operates between 1000 psia and 1000°F, and 1 psia. It has seven feedwater heaters. Find the optimum pressure and inlet temperature for the high- and the low-pressure feedwater heaters.

SOLUTION Referring to Fig. 2-24 and the steam tables

$$T_B = 544.58°\text{F} \qquad T_C = 101.74°\text{F} \qquad s_1 = 1.6530$$

$$\Delta T_{\text{opt}} = \frac{544.58 - 101.74}{7 + 1} = 55.36°\text{F}$$

8

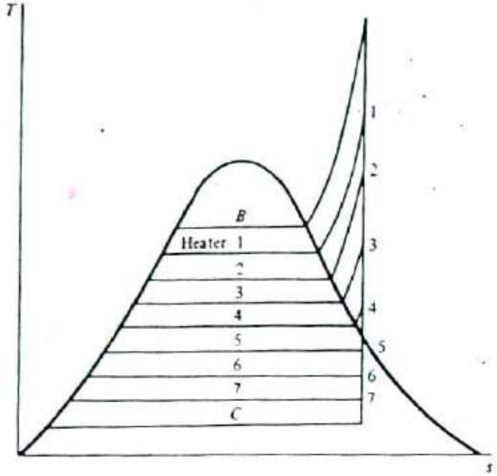


Figure 2-24 T-s diagram of Example 2-8

The low-pressure heater

$$T_7 = T_C + \Delta T_{opt} = 101.74 + 55.36$$
$$= 157.10°\text{F, corresponding to } P_7 = 4.422 \text{ psia}$$

Because s_g at $P_7 = 1.806 > s_1$, the bled steam to heater 7 is, as expected, in the two-phase region, for which

$$s_7 = s_1 = 1.6530 = (s_f + x_7 s_{fg})_{4.422 \text{ psia}}$$
$$= 0.2266 + x_7(1.6277)$$

Thus $\qquad x_7 = 0.876$

and $\qquad h_7 = 125.05 + 0.876 \times 1003.9 = 1004.5 \text{ Btu/lb}_m$

The high-pressure heater

$$T_{sat,1} = T_B - \Delta T_{opt} = 544.58 - 55.36$$
$$= 489.22°\text{F, corresponding to } P_1 = 617.04 \text{ psia}$$

Because at P_1 $s_g = 1.4433 < s_1$, the bled steam to heater 1 is superheated. The inlet temperature, found by interpolation from the steam tables, is 850.0°F with a degree of superheat of 360.8°F, corresponding to an enthalpy of 1435.05 Btu/lb$_m$.

Heater 1, the high-pressure heater, receives highly superheated steam and thus would be constructed with a desuperheater zone, a condensing zone, and most likely, a drain cooler. Its TTD is most likely negative. Heater 7, the low-pressure heater, on the other hand, receives wet steam and will have no desuperheating zone. It will have a condensing section and may not have an integral drain cooler. If not, its drain may be cascaded to the condenser either directly or via a separate drain cooler, or it may be pumped forward into the feedwater line.

The temperatures, pressures, and inlet conditions of the other five feedwater heaters are found in a like manner. They are then used in the appropriate equations for determining the mass-flow rates in the particular type of heater, or mix of heaters, and the various cycle parameters. If the turbine in Example 2-8 were not ideal, the exact turbine expansion line must first be determined in order to find the bled steam inlet temperatures and enthalpies. Here the use of the Mollier diagram may be more useful than the *T*-*s* diagram.

It is now instructive to show the effect of varying ΔT between feedwater heaters from ΔT_{opt} on cycle efficiency. Figure 2-25 shows the effect of varying the total feedwater temperature rise (above the condenser temperature) for a saturated internally reversible steam cycle operating between 1000 and 1 psia, corresponding to saturation temperatures of 544.58°F and 101.74°F, respectively. The curve shows the percent decrease in cycle heat rate (corresponding to increase in cycle efficiency) for 1, 2, 3, 4, and 10 feedwater heaters versus the total temperature rise above the condenser temperature.

It can be seen, as expected, that the curve for a single feedwater heater peaks at a temperature rise halfway between the above saturation temperatures; i.e., it peaks at ΔT of 0.5(544.58 − 101.74), or about 222°F. For two feedwater heaters, the peak occurs at $\frac{2}{3}$(544.58 − 101.740), or about 295°F. It can also be seen that the curves are relatively flat about the optimum values, which indicates that small departures from these optimum values have no serious effect on heat rate. In actual powerplants, the feedwater heaters are not positioned necessarily at their optimum positions. Other considerations may dictate the exact positions. These considerations include the placement of the deaerating heater for best deaeration and the relative positions of the closed heaters before and after it, the existence of a convenient point at which steam is bled such as the crossover between turbine sections or at the steam outlet to the reheater, the design of the turbine casings, and others.

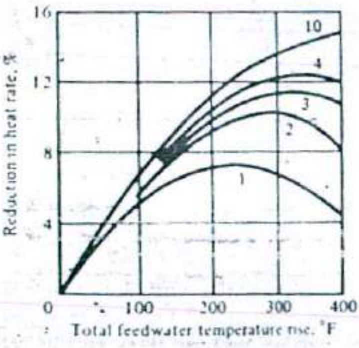


Figure 2-25 Effect of ΔT between feedwater heaters on cycle heat rate.

2-14 THE SUPERCRITICAL-PRESSURE CYCLE

In Fig. 2-26 the feedwater is pressurized at 8 to a pressure beyond the critical pressure of the vapor (3208 psia for steam). The feedwater heating curve shows a gradual change in temperature and density but not in phase to the steam temperature at 1. Such heating can be made to be closer to the heat source temperature than a subcritical cycle with the same steam temperature that shows an abrupt change in temperature within the two-phase region. Looking at it another way, the supercritical-pressure cycle receives more of its heat at higher temperatures than a subcritical cycle with the same turbine inlet steam temperature.

Because of the gradual change in density, supercritical-pressure cycles use once-through steam generators instead of the more common drum-type steam generators (Chap. 3).

A disadvantage of the supercritical-pressure cycle, however, is that expansion from point 1 to the condenser pressure would result in very wet vapor in the latter stages of the turbine. Hence, supercritical-pressure cycles invariably use reheat and often double reheat. A popular base design for a supercritical powerplant used 3500 psia and initial 1000°F steam with reheats to 1025°F and 1050°F (3500/1000/1025/1050). The higher temperatures after reheat were tolerated by the reheater tubes because of the much lower pressures in them.

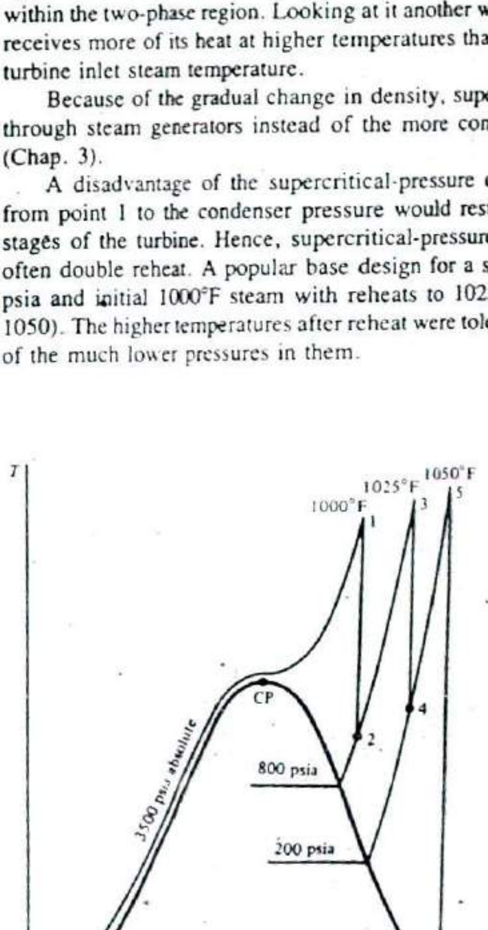


Figure 2-26 *T-s* diagram of an ideal supercritical, double-reheat 3500/1000/1025/1050 steam cycle.

Example 2-9 Calculate the net work, heat added, efficiency, and work ratio of an internally reversible supercritical double-reheat 3500/1000/1025/1050 cycle. Reheats occur at 800 and 200 psia. Condensing is at 1 psia.

SOLUTION Referring to Fig. 2-26 and the steam tables with h values in Btu/lb$_m$ and s values in Btu/(lb$_m$ · °R)

$h_1 = 1422.2 \qquad s_1 = 1.4709$

$s_2 = 1.4709 \qquad h_2 = 1254.5$

$h_3 = 1525.3 \qquad s_3 = 1.69015$

$s_4 = 1.69015 \qquad h_4 = 16336.3$

$h_5 = 1555.4 \qquad s_5 = 1.8603$

$s_6 = 1.8603 \qquad x_6 = 0.936 \qquad h_6 = 1039.7$

$h_7 = 69.73$

$$h_8 = 69.73 + \frac{0.016136(3500 - 1)144}{778.16} = 69.73 + 10.45 = 80.18$$

$$\begin{aligned}\Delta w_{net} &= (1422.20 - 254.5) + (1525.3 - 1336.3) \\ &\quad + (1555.4 - 1039.7) - 10.45 \\ &= 167.7 + 189 + 515.7 - 10.45 \\ &= 872.4 - 10.45 = 861.95 \text{ Btu/lb}_m\end{aligned}$$

$$\begin{aligned}q_A &= (h_1 - h_8) + (h_3 - h_2) + (h_5 - h_4) \\ &= 1342.02 + 270.8 + 219.3 = 1831.92 \text{ Btu/lb}_m\end{aligned}$$

Therefore

$$\eta_{th} = \frac{861.95}{1831.92} = 0.4705$$

and

$$\text{WR} = \frac{861.95}{872.41} = 0.9880$$

The efficiency, of course, would be further improved by the addition of feedwater heaters. This example is listed as cycle *I* in Table 2-2.

2-15 Cogeneration

Cogeneration is the simultaneous generation of electricity and steam (or heat) in a single powerplant. It has long been used by industries and municipalities that need process steam (or heat) as well as electricity. Examples are chemical industries, paper mills, and places that use district heating. Cogeneration is not usually used by large utilities which tend to produce electricity only. Cogeneration is advisable for industries and municipalities if they can produce electricity cheaper, or more conveniently, than that brought from a utility.

From an energy resource point of view, cogeneration is beneficial only if it saves primary energy when compared with separate generation of electricity and steam (or heat). The *cogeneration plant efficiency* η_{co} is given by

$$\eta_{co} = \frac{E + \Delta H_s}{Q_A} \tag{2-29}$$

where E = electric energy generated

ΔH_s = heat energy, or heat energy in process steam

= (enthalpy of steam entering the process)

− (enthalpy of process condensate returning to plant)

Q_A = heat added to plant (in coal, nuclear fuel, etc.)

For separate generation of electricity and steam, the heat added per unit *total* energy output is

$$\frac{e}{\eta_e} + \frac{(1-e)}{\eta_h}$$

where e = electrical fraction of total energy output = $\dfrac{E}{(E+\Delta H_s)}$

η_e = electric plant efficiency

η_h = steam (or heat) generator efficiency

The *combined efficiency* η_c *for separate generation* is therefore given by

$$\eta_c = \frac{1}{(e/\eta_e) + [(1-e)/\eta_h]} \tag{2-30}$$

and cogeneration is beneficial if the efficiency of the cogeneration plant Eq. (2-29) exceeds that of separate generation, Eq. (2-30).

Types of Cogeneration

There are two broad categories of cogeneration:

1. *The topping cycle*, in which primary heat at the higher temperature end of the Rankine cycle is used to generate high-pressure and -temperature steam and electricity in the usual manner. Dependng on process requirements, process steam at low-pressure and temperature is either (*a*) extracted from the turbine at an intermediate stage, much as for feedwater heating, or (*b*) taken at the turbine exhaust,

in which case it is called a *back pressure* turbine. Process steam pressure requirements vary widely, between 0.5 and 40 bar.

2. *The bottoming cycle*, in which primary heat is used at high temperature directly for process requirements. An example is the high-temperature cement kiln. The process low-grade (low temperature and availability) waste heat is then used to generate electricity, obviously at low efficiency. The bottoming cycle thus has a combined efficiency that most certainly lies below that given by Eq. (2-30), and therefore is of little thermodynamic or economic interest.

Only the topping cycle, therefore, can provide true savings in primary energy. In addition, most process applications require low grade (temperature, availability) steam. Such steam is conveniently produced in a topping cycle. There are several arrangements for cogeneration in a topping cycle. Some are:

(*a*) Steam-electric powerplant with a back-pressure turbine.
(*b*) Steam-electric powerplant with steam extraction from a condensing turbine (Fig. 2-27).
(*c*) Gas-turbine powerplant with a heat-recovery boiler (using the gas turbine exhaust to generate steam).
(*d*) Combined steam-gas-turbine cycle powerplant (Secs. 8-8 and 8-9). The steam turbine is either of the back-pressure type (*a*) or of the extraction-condensing type (*b*), above.

The most suitable electric-to-heat generation ratios vary from type to type. The back-pressure steam turbine plant (*a*) is most suitable only when the electric demand is low compared with the heat demand. The combined-cycle plant (*d*) is most suitable only when the electric demand is high, about comparable to the heat demand or higher, though its range is wider with an extraction-condensing steam turbine than with a back-pressure turbine. The gas-turbine cycle (*c*) lies in between. Only the extraction-condensing plant (*b*) is suitable over a wide range of ratios.

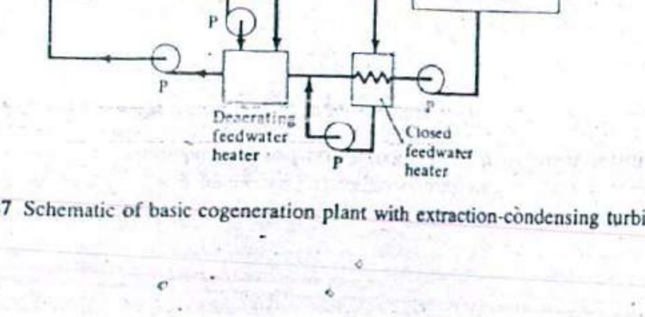


Figure 2-27 Schematic of basic cogeneration plant with extraction-condensing turbine.

Economics of Cogeneration

A privately or municipally owned cogeneration plant is advisable from an economic point of view if the cost of electricity generated by it is less than if purchased from a utility. (If a utility is not available, cogeneration becomes necessary, irrespective of economics.) In general, very low fractions of electric to total energy are not considered economical for cogeneration.

Since the main incentive of cogeneration is process steam (or heat), the economics of cogeneration are sharply influenced by the additional cost of generating electricity.

Powerplant costs are of two kinds: capital costs and production costs. *Capital costs* are given in total dollars or as *unit capital costs* in dollars per kilowatt net. Production costs are calculated annually, or more frequently if desired, and given in mills per kilowatt hour. A mill is one one-thousandth of a United States dollar. Capital costs determine whether a given utility or industry is sound enough to obtain financing and thus able to pay the fixed charges against these costs. *Production costs* are the true measure of the cost of power generated. They are composed of:

a. The fixed charges against the capital costs
b. The fuel costs
c. Operation and maintenance costs

all in mills per kilowatt hour. They are therefore given by:

$$\text{Production costs} = \frac{\text{total } (a+b+c)\ \$ \text{ spent per period} \times 10^3}{\text{KWh (net) generated during same period}} \tag{2-31}$$

where the period is usually taken as one year.

For a congeneration plant, it is important to calculate the production costs of electricity as an excess over the generating cost of steam alone, and to compare it with the cost of electricity when purchased from a utility. It is now necessary to introduce the *plant operating factor* POF, defined for all plants as

$$\text{POF} = \frac{\text{total net energy generated by plant during a period of time}}{\text{rated net energy capacity of plant during same period}} \tag{2-32}$$

where the period is again usually taken as one year. For estimation purposes, it is common to take POF = 0.80. A plant operating with POF = 0.8 is the same as if it operated only at rated capacity for 80 percent of the time or for 0.8(365 × 24) = 7008 h/yr, which is usually rounded out to 7000 h/yr.

The excess cost of electricity for a cogeneration plant may now be obtained from

$$\text{Electric cost} = [(C_{co} - C_h)r + (\text{OM}_{co} - \text{OM}_h) + (F_{co} - F_h)] \frac{10^3}{7000\,P} \text{ mills/kWh} \tag{2-33}$$

where C = capital costs, \$

r = annual fixed charges against the capital cost, fraction of C

OM = annual operation and maintenance costs, \$/yr

F = annual fuel costs, \$/yr

P = electric plant net power rating, kW

and the subscripts *co* and *h* indicate cogeneration and process heat plants, respectively.

Cogeneration plants, built mostly by industries or municipalities, are smaller than utility electric-generating plants and therefore tend to have higher unit capital and operating costs. They have not usually been considered for operation with coal or nuclear energy as a primary heat source, though this picture is slowly changing.

PROBLEMS

2-1 A simple ideal saturated Rankine cycle turbine receives 125 kg/s of steam at 300°C and condenses at 40°C. Calculate (*a*) the net cycle power, in megawatts, and (*b*) the cycle efficiency.

2-2 A simple nonideal saturated Rankine cycle turbine receives 125 kg/s of steam at 300°C and condenses at 40°C (same conditions as Prob. 2-1). This cycle has turbine and pump polytropic efficiencies of 0.88 and 0.75, respectively, and a total pressure drop in the feedwater line and steam generator of 10 bar. Calculate (*a*) the net cycle power, in megawatts, and (*b*) the cycle efficiency.

2-3 Analyze the ideal Rankine cycle *C* in Table 2-2 if the feedwater heater is placed at 100 psia.

2-4 Compare the inlet steam mass and volume flow rates in pound mass per second and cubic feet per second of (*a*) a fossil-fuel powerplant turbine having a polytropic efficiency of 0.90 and receiving steam at 2400 psia and 1000°F and (*b*) a nuclear powerplant turbine having a polytropic efficiency of 0.88 and receiving saturated steam at 1000 psia. Each turbine produces 1000 megawatts, and exhausts to 1 psia.

2-5 To reduce the volume flow rate and hence turbine physical size, powerplants that operate with low initial temperature water as a heat source, such as some types of geothermal (Chap. 12) and ocean temperature energy conversion, OTEC (Chap. 15), powerplants, use working fluids other than steam, such as Freon-12, ammonia, and propane. Compare the mass flow rates, pound mass per hour, volume flow rates, cubic feet per second, and boiler and condenser pressures of (*a*) Freon-12, (*b*) propane, and (*c*) steam, if all cycles operate with adiabatic reversible turbines that receive saturated vapor at 200°F and condense at 70°F, and each produces 100 kW.

2-6 In Prob. 2-5, why do the cycles operate with saturated vapor?

2-7 Consider three nonideal saturated Rankine cycles operating between 200 and 70°F using Freon-12, propane, and steam as working fluids. Each has turbine and pump polytropic efficiencies of 85 and 65 percent, respectively, and produces net work of 100 kW. Calculate (*a*) the mass flow rate in pound mass per hour, (*b*) the volume flow rate in cubic feet per second, (*c*) the heat added, in Btus per hour; and (*d*) the cycle efficiency.

2-8 Consider an ideal saturated steam Rankine cycle with perfect regeneration (Fig. 2-11) operating between 1000 and 1.0 psia. Neglecting pump work, calculate (*a*) the quality of the turbine exhaust steam, (*b*) the turbine work in Btus per pound mass, (*c*) the heat added in Btus per pound mass, and (*d*) the cycle efficiency. Compare that efficiency to that of a similar cycle but without regeneration, and a Carnot cycle, all operating between the same temperature limits.

2-9 Compare the net works, in Btus per pound mass, and efficiencies of two ideal saturated Rankine cycles using Freon-12 as a working fluid and operating between 200 and 72°F. One cycle has no feed heaters and the other has one open-type feed heater placed optimally. Why is feed heating not usually resorted to in such cycles?

2-10 A Rankine cycle with inlet steam at 90 bar and 500°C and condensation at 40°C produces 500 MW. It has one stage of reheat, optimally placed, back to 500°C. One feedwater of the closed type with drains cascaded back to the condenser receives bled steam at the reheat pressure. The high- and low-pressure turbine sections have polytropic efficiencies of 92 and 90 percent, respectively. The pump has a polytropic efficiency of 0.75. Calculate (*a*) the mass flow rate of steam at turbine inlet in kilograms per second, (*b*) the cycle efficiency, and (*c*) the cycle work ratio. Use TDD = − 1.6°C.

2-11 An ideal Rankine cycle operates with turbine inlet steam at 90 bar and 500°C, and a condenser temperature of 40°C. Calculate the efficiency and work ratio of this cycle for the following cases: (*a*) no feedwater heating, (*b*) one open-type feedwater heater, (*c*) one closed-type feedwater heater with drains cascaded back to the condenser, and (*d*) one closed feedwater heater with drains pumped forward. In each case the feedwater heater is optimally placed. Use TDD = 2.5°C.

2-12 A superheated nonideal steam cycle operates with inlet steam at 2400 psia and 1000°F and condenses at 1 psia. It has five feedwater heaters, all optimally placed. Assume the polytropic efficiencies of the turbine sections before, between, and after the bleed points to be all the same and equal to 0.90. Calculate (*a*) the specific enthalpies of the extraction steam to each feedwater heater, in Btus per pound mass and (*b*) the turbine overall polytropic efficiency; and (*c*) estimate the terminal temperature difference for each feedwater heater.

2-13 An 850-MW Rankine cycle operates with turbine inlet steam at 1200 psia and 1000°F and condenser pressure at 1 psia. There are three feedwater heaters placed optimally as follows: (*a*) the high-pressure heater is of the closed type with drains cascaded backward; (*b*) the intermediate-pressure heater is of the open type; (*c*) the low-pressure heater is of the closed type with drains pumped forward. Each of the turbine sections have the same polytropic efficiency of 90 percent. The pumps have polytropic efficiencies of 80 percent. Calculate (*a*) the mass flow rate at the turbine inlet in pound mass per hour, (*b*) the mass flow rate to the condenser, (*c*) the mass flow rate of the condenser cooling water, in pound mass per hour, if it undergoes a 25°F temperature rise, (*d*) the cycle efficiency, and (*e*) the cycle heat rate, in Btus per kilowatt hour.

2-14 If the Rankine cycle is to be used in outer space, heat rejection can be done only by thermal radiation to space which has an effective temperature of 0 absolute. To reduce the size and mass and hence lifting weight of the condenser, condensation has to be at temperatures much higher than those used in land-based Rankine cycles. Condensing temperatures of 1000 to 1500°F are considered. These are higher than the critical temperature of water. This also means a much higher turbine inlet temperature. Thus a liquid metal such as sodium must be used as the working fluid. Consider a 100-kW (thermal) Rankine cycle using sodium, operating with 24.692 psia and 2400°R sodium vapor at turbine inlet and condensing at 1500°R. The turbine and pump polytropic efficiencies are 0.85 and 0.65, respectively. For no feed heaters and ignoring pressure drops, calculate (*a*) the cycle efficiency and (*b*) the heat transfer area of the condenser-radiator if it has an overall heat transfer coefficient of 5 Btu/ft^2 · h · °F.

2-15 Calculate the gross heat rate, in Btus per kilowatt hour, and the gross efficiency of the powerplant shown in Fig. 2-20.

2-16 A 100-MW (thermal) binary-vapor cycle uses saturated mercury vapor at 1600°R at the top turbine inlet. The mercury condenses at 1000°R in a mercury-condenser–steam-boiler in which saturated steam is generated at 400 psia. It is further superheated to 1160°R in the mercury-boiler–steam superheater. The steam condenses at 1 psia. Assume both mercury and steam cycles to be ideal, and ignoring the pump work (*a*) draw flow and *T-s* diagrams of the binary cycle numbering points correspondingly, (*b*) calculate the mass flow rates of mercury and steam, and (*c*) calculate the heat added and heat rejected, in Btus per hour, and the cycle efficiency.

2-17 An advanced-type supercritical powerplant has turbine inlet steam at 7000 psia and 1400°F, double reheat at 1600 psia and 400 psia, both to 1200°F, and condenser at 1 psia. The three turbine sections have polytropic efficiencies of 0.93, 0.91, and 0.89 in order of descending pressures. The pump has a polytropic efficiency of 0.75. The plant receives one unit train of coal daily, which is composed of 100 cars carrying 110 short tons each. The coal has a heating value of 11,000 Btus/lb$_m$. The turbine-generator combined mechanical and electrical efficiency is 0.90. The steam-generator efficiency is 0.87. 8 percent of the gross output is used to run plant auxiliaries. Ignoring, for simplicity, all steam-line pressure drops and all feedwater heaters, calculate (*a*) the plant gross and net outputs, in megawatts, (*b*) the plant cycle, gross and net efficiencies, and (*c*) the cycle, and station gross and net heat rates, in Btus per kilowatt hour.

2-18 Draw flow diagrams of cogeneration plants of (*a*) the topping back-pressure steam turbine type and (*b*) the bottoming-condensing steam turbine type.

2-19 A cogeneration steam plant of the extraction-condensing steam turbine type has turbine inlet flow of 12.5 kg/s at 50 bar and 400°C, extraction for process steam at 220°C and condensation for both turbine and process steam at 40°C. Assuming ideal turbine and pump, no feedwater heating and 50 percent fraction of electricity to total energy output, calculate (*a*) the cogeneration plant efficiency, (*b*) the combined efficiency if separate electric and steam generation plants producing the same outputs as above are used. For all cases take the steam generator efficiency as 85 percent.

CHAPTER
THREE

FOSSIL-FUEL STEAM GENERATORS

3-1 INTRODUCTION

Steam generators are used in both fossil- and nuclear-fuel electric-generating powerplants. The most modern steam generators produce high-pressure (2400 to 3500 psia, 165 to 240 bar) superheated steam, the exception being pressurized-water reactor steam generators, which produce lower-pressure (1000 psia, 70 bar) saturated steam. The steam is invariably used in a Rankine cycle. Steam generators represent by far the greatest energy source for powerplants in the world today.

This chapter will cover fossil-fuel steam generators. Nuclear-fuel steam generators are of radically different design and will be covered in examples in Chaps. 10 and 11.

A *steam generator* is a complex combination of economizer, boiler, superheater, reheater, and air preheater. In addition, it has various auxiliaries, such as stokers, pulverizers, burners, fans, emission control equipment, stack, and ash-handling equipment. A *boiler* is that portion of the steam generator where saturated liquid is converted to saturated steam, although it may be difficult to separate it, physically, from the economizer. The term "boiler" is often used to mean the whole steam generator in the literature, however. Steam generators are classified in different ways. They may, for example, be classified as either (1) utility or (2) industrial steam generators.

Utility steam generators are those used by utilities for electric-power generating plants and are our main concern in this book. Modern utility steam generators are essentially of two basic kinds: (1) the subcritical water-tube drum type and (2) the supercritical once-through type. The supercritical units usually operate at about 3500 psia (240 bar) and higher, above the steam critical pressure of 3208.2 psia. The subcritical drum group usually operate at either 1900 psig (about 130 bar) or 2600 psig (180 bar). The majority of utility steam generators purchased in the 1970s and 1980s are of the 2600 psig water-tube drum variety, which produce superheated steam at about 1000°F (540°C) with one or two stages of reheat. They have the ability to burn coal in pulverized form or oil, although oil is being gradually retired as a fuel because of rising costs and supply problems. Natural gas, although still used in certain parts of the world, is also costly and is now being conserved for domestic uses in the United States. Gas, however, is a clean burning, relatively pollution-free fuel. The steam capacities of modern utility steam generators are high, ranging from 1 to 10 million lb_m/h (125 to 1250 kg/s). They power electric powerplants ranging in output from 125 to 1300 megawatts (MW).

Industrial steam generators, on the other hand, are those used by industrial and institutional concerns and are of many types. These include water-tube pulverized-coal units similar to those used by utilities, but they also may burn stoker (lump) coal, oil, or natural gas, often in combination, as well as municipal refuse and process wastes or by-products. Some even use electric heating. Some are heat-recovery types that use waste heat from industrial processes. They may also be of the fire-tube variety. Industrial steam generators usually do not produce superheated steam. Rather, they usually produce saturated steam, or even only hot water (in which case they should not be called steam generators). They operate at pressures ranging from a few psig to as much as 1500 psig (105 bar) and steam (or hot water) capacities ranging from a few thousand to 1 million lb_m/h (125 kg/s).

Fossil-fueled steam generators are more broadly classified [12, 13] as those having the following components or characteristics.

1. Fire-tube boilers
2. Water-tube boilers
3. Natural-circulation boilers
4. Controlled-circulation boilers
5. Once-through flow
6. Subcritical pressure
7. Supercritical pressure

3-2 THE FIRE-TUBE BOILER

Fire-tube boilers have been used in various early forms to produce steam for industrial purposes since the late eighteenth century. They are no longer used in large utility powerplants. They are covered here, however, for historical reasons and, by contrast, to emphasize the modern water-tube variety. Fire-tube boilers are still used in industrial plants to produce saturated steam at the upper limits of 250 psig (about 18 bar) pressure and 50,000 lb_m/h (6.3 kg/s) capacity. Although their size has increased, their general design has not changed appreciably in the past 25 years.

The fire-tube boiler is a special form of the shell-type boiler. A *shell-type boiler* is a closed, usually cylindrical, vessel or shell that contains water. A portion of the

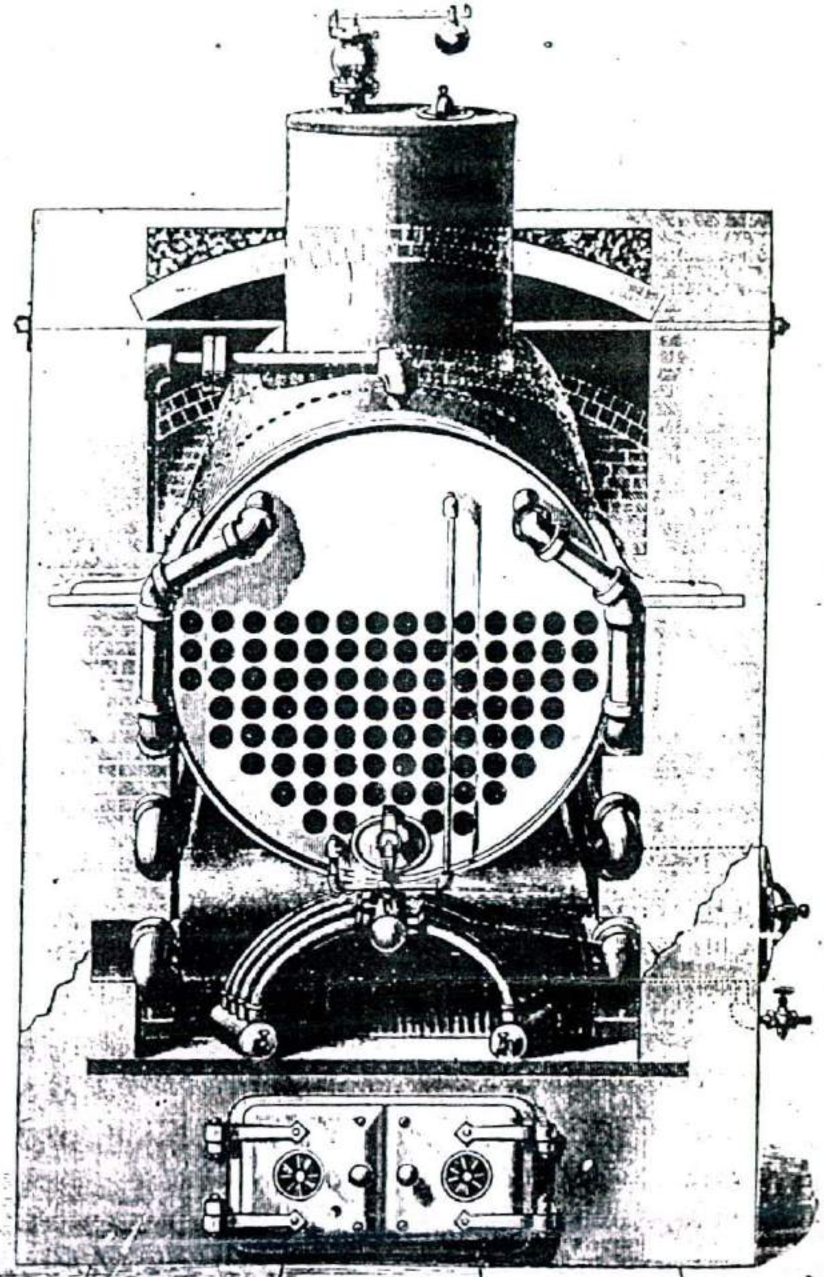


Figure 3-1 The Stead fire-tube steam generator [11].

shell, such as its underside, is simply exposed to heat, such as gases from an externally fired flame. The shell boiler evolved into more modern forms such as the *electric boiler*, in which heat is supplied by electrodes embedded in the water, or the *accumulator*, in which heat is supplied by steam from an outside source passing through tubes within the shell. In both cases the shell itself is no longer exposed to heat.

The shell boiler evolved into the *fire-tube boiler*. Hot gases, instead of steam, were now made to pass through the tubes. Because of improved heat transfer the fire-tube boiler is much more efficient than the original shell boiler and can reach efficiencies of about 70 percent.

The fire tubes were placed in horizontal, vertical, or inclined positions. The most common was the horizontal-tube boiler. Figure 3-1 shows such an early fire-tube boiler. Figure 3-2 shows a simplified sketch of such a boiler. The furnace and grates are located underneath the front end of the shell. The gases pass horizontally along its underside to the rear, reverse direction, and pass through the horizontal tubes to the stack at the front.

There are two types of fire-tube boilers: (1) the fire-box and (2) the scotch marine. In the *fire box boiler*, the furnace, or fire box, is located within the shell, together with the fire tubes. In the *scotch marine boiler* (Fig. 3-3), combustion takes place within one or more cylindrical chambers that are usually situated inside and near the bottom of the main shell. The gases leave these chambers at the rear, reverse direction, and return through the fire tubes to the front and out through the stack. Scotch marine boilers are usually specified with liquid or gas fuels.

Because boiling occurs in the same compartment where water is, fire-tube boilers are limited to saturated-steam production. They are presently confined to relatively small capacities and low steam pressures, such as supplying steam for space heating

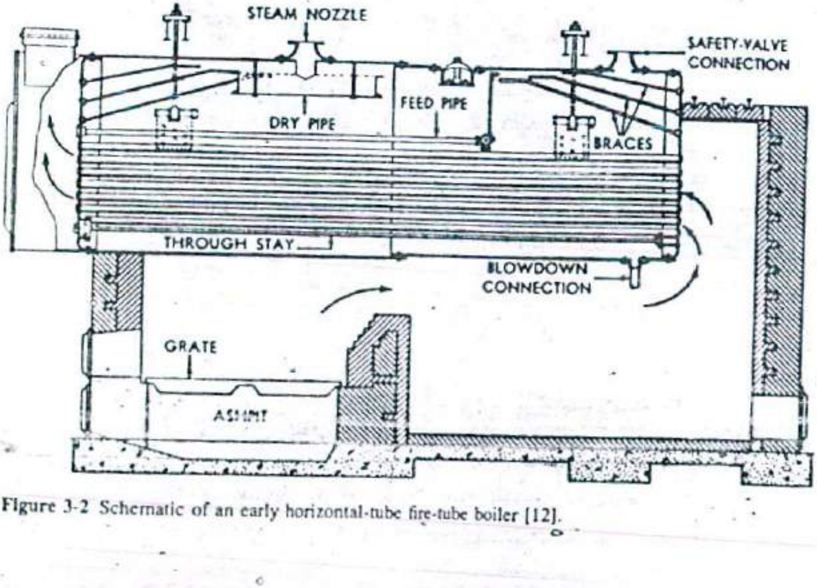
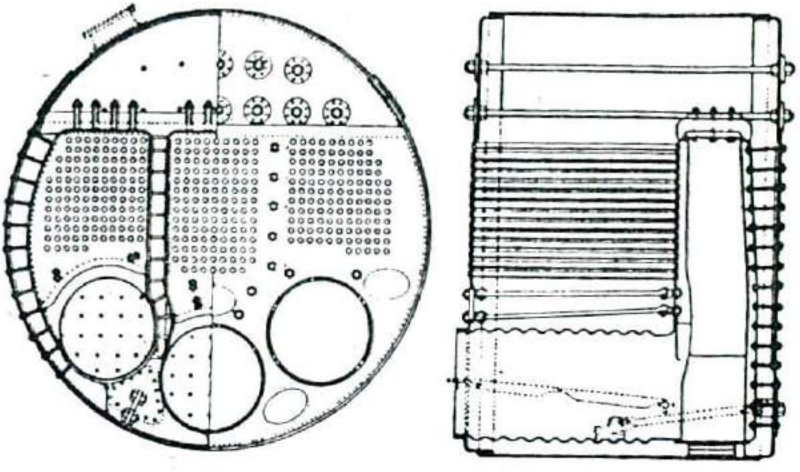


Figure 3-2 Schematic of an early horizontal-tube fire-tube boiler [12].



(a) (b)

Figure 3-3 Schematic of an early scotch marine boiler.

and, in decreasing numbers, for railroad locomotive service. The largest scotch marine boiler offered in the United States today is rated at 2000 boiler horsepower (blhp),* contains two combustion chambers within a 13-ft diameter, 30-ft-long shell.

3-3 THE WATER-TUBE BOILER: EARLY DEVELOPMENTS

The forerunner of the modern steam generator was a water-tube boiler developed by George Babcock and Stephen Wilcox in 1867. Babcock and Wilcox called it the "nonexplosive" water-tube boiler, an allusion to disastrous boiler explosions that were frequent at the time. It was not, however, until early in the twentieth century, with the advent of the steam turbine and its requirement for large steam pressures and flows, that commercial development of the water-tube boiler became a reality.

With higher steam pressures and capacities, fire-tube boilers would need large-diameter shells. With such large diameters, the shells would have to operate under such extreme pressure and temperature stresses that their thicknesses would have been

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* Boiler horsepower was originally used to indicate the size of the boiler. One blhp was defined as 10 ft^2 of boiler heating surface. This was later changed to the amount of heat required to evaporate 30 lb_m/h of water at 100°F to saturated steam at 70 psia. Thus, using enthalpies from the steam tables, this was equal to $30(1180.6 - 68) = 33{,}378$ Btu/h. Later on this was changed to the amount of heat necessary to evaporate 34.5 lb_m/h of saturated water to saturated steam at 1 atm or $34.5 \times 970.3 = 33475.35$ Btu/h. This is now rounded so that the modern value is 1 blhp = 33480 Btu/h.

too large. In addition, they were subjected to scale deposits and boiler explosions and became intolerably costly.

The water-tube boiler puts the pressure instead in tubes and relatively small-diameter drums that are capable of withstanding the extreme pressures of the modern steam generator. In general appearance, the early water-tube boilers looked much like the fire-tube boiler except that the higher-pressure water and steam were inside the tubes and the combustion gases were on the outside. The water-tube boiler went through several stages of development.

The Straight-Tube Boiler

The first water-tube boiler was the *straight-tube boiler* (Fig. 3-4), in which straight tubes, 3 to 4 in OD, inclined at about 15° and staggered with 7- to 8-in spacings, connected two vertical headers. One header was a *downcomer*, or *downtake*, which supplied nearly saturated water to the tubes. The water partially boiled in the tubes. The other header was a *riser*, or *uptake*, which received the water-steam mixture. The water density in the downcomer was larger than the two-phase density in the riser, which caused natural circulation in a clockwise direction (Sec. 3-5). As capacity increased, more than one header each and more than one tube "deck" were used. The two-phase mixture went into an upper drum that was arranged either parallel to the tubes (the *longitudinal drum*, Fig. 3-4*a*) or perpendicular to them (the *cross drum*, Fig. 3-4*b*). These drums received the feedwater from the last feedwater heater and supplied saturated steam to the superheater through a steam separator within the drum which separated steam from the bubbling water. The lower end of the downcomer was connected to a *mud drum*, which collected sediments from the circulating water.

A single longitudinal drum, usually 4 ft in diameter, can allow only a limited number of tubes and hence a limited heating surface. Longitudinal-drum boilers were built with one or more than one parallel drums, depending upon capacity. They were built with heating surfaces of 1000 to 10,000 ft^2 (93 to 930 m^2) and were limited to low pressures of 175 to 340 psia (12 to 23 bar) and steam capacities from 5000 to 80,000 lb_m/h (0.63 to 10 kg/s).

Cross-drum boilers, because of geometry, could accommodate many more tubes than longitudinal-drum boilers and were built with heating surfaces of 1000 to 25,000 ft^2 (93 to 2300 m^2), pressures of 175 to 1465 psia (12 to 100 bar), and steam capacities of 5000 to 500,000 lb_m/h (0.63 to 63 kg/s).

Baffles were installed across the tubes in both kinds to allow for up to three gas passes to ensure maximum exposure of the tubes to the hot combustion gases and minimal gas dead spots.

The Bent-Tube Boiler

There were many versions of the bent-tube boiler. In general, a *bent-tube boiler* used bent, rather than straight, tubes between several drums or drum and headers. The tubes were bent so that they entered and left the drums radially. The number of drums usually

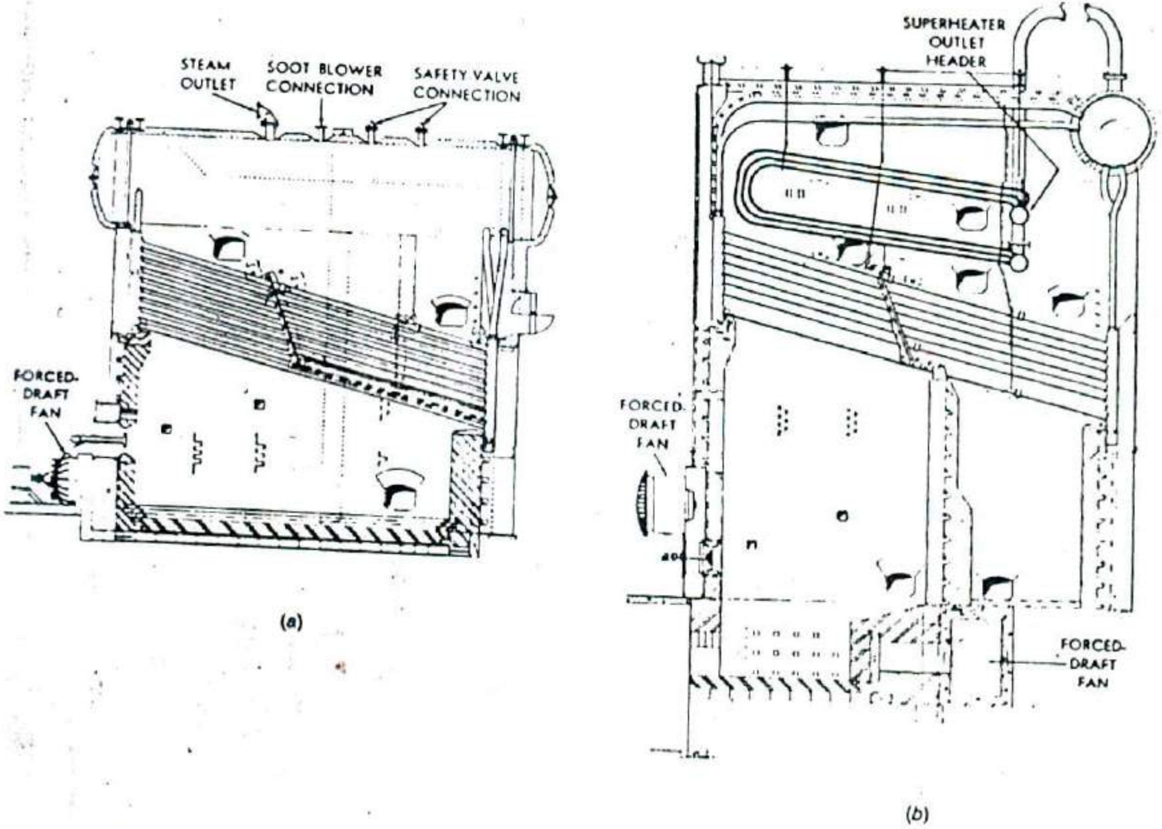

Figure 3-4 Early water-tube boilers: (*a*) longitudinal and (*b*) cross-drum [12].

varied from two to four. Gas baffles were installed to allow for one or more gas passages, as above.

It will suffice to show one example of bent-tube boilers, the so-called four-drum Stirling boiler (Fig. 3-5), which was conceived in the early 1890s and has changed little since. Unlike other bent-tube boilers, this one had three top drums, all containing a two-phase mixture, and one bottom drum (also called a mud drum) that was filled with water.

The four-drum Stirling boiler worked as follows. The combustion gases flowed upward from the furnace at bottom right, through the first bank of tubes connecting the water and front steam drum, through the superheater, and by proper baffling through the second and third tube banks connecting to the center and rear steam drums. The gases then left in counterflow fashion through a straight-tube economizer. Feedwater from the economizer entered the rear steam drum, which may be at a slightly higher level than the other two. Water circulated from the rear to the lower drum through the rear bank of tubes (the downcomer tubes) and then up through both the center and front banks of tubes (the riser tubes) to the center and front drums. All three drums had their steam and water regions interconnected at top and bottom.

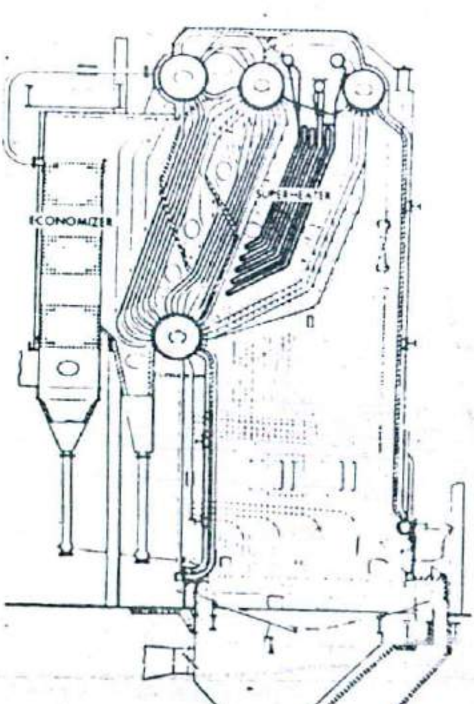

Figure 3-5 A four-drum early Stirling boiler [12].

The tubes were typically 3 to 3½ in OD and spaced 5 and 7 in on centers, with back spacing decreased to maintain gas velocity of the cooler and denser gases. The spacings allowed for replacement of defective tubes without removing neighboring tubes.

The four-drum Stirling boiler was superseded by a simpler two-drum design in which the steam drum was directly above the water drum with one bank of bent tubes to the front, i.e., on the side of the incoming gas and another to the rear. Later designs of the two-drum Stirling boiler used a single gas path. More recent designs of Stirling boilers used cooled furnace walls by lining the interior of the walls with tubes carrying the same boiler water from the plant as seen in Fig. 3-5. These added to the heat-absorbing surface and protected the refractory lining of the walls from excessive temperature. The result was higher rates of combustion and higher steam-flow rates.

The Stirling boiler was generally capable of meeting conditions of rapidly varying loads, was useful where high water quality was difficult to maintain, and was adaptable to various fuels. It found use in both stationary and marine applications.

3-4 THE WATER-TUBE BOILER: RECENT DEVELOPMENTS

The advent of the water-cooled furnace walls, called *water walls*, eventually led to the integration of furnace, economizer, boiler, superheater, reheater, and air preheater into the modern steam generator. Water cooling is also used for superheater and economizer compartment walls and various other components, such as screens, dividing walls, etc. The use of a large number of feedwater heaters (up to seven or eight) means a smaller economizer, and the high pressure means a smaller boiler surface because the latent heat of vaporization decreases rapidly with pressure. Thus a modern high-pressure steam generator requires more superheating and reheating surface and less boiler surface than older units. Beyond about 1500 psia, the water tubes represent the entire boiler surface and no other tubes, such as those seen in the earlier designs of the previous two sections, are required.

Figure 3-6 shows a schematic flow diagram of a common steam-generator system. Water at 450 to 500°F from the plant high-pressure feedwater heater enters the economizer and leaves saturated or as a two-phase mixture of low quality. It then enters the steam drum at midpoint. Water from the steam drum flows through insulated downcomers, which are situated outside the furnace, to a header. The header connects to the water tubes that line the furnace walls and act as risers. The water in the tubes receives heat from the combustion gases and boils further. The density differential between the water in the downcomer and that in the water tubes helps circulation. Steam is separated from the bubbling water in the drum and goes to the superheater and the high-pressure section of the turbine. The exhaust from that turbine returns to the reheater, after which it goes to the low-pressure section of the turbine.

Atmospheric air from a *forced-draft (FD) fan* is preheated by the flue gases just before they are exhausted to the atmosphere. From there it flows into the furnace, where it mixes with the fuel and burns to some 3000°F. The combustion gases impart portions of some of their energy to the water tubes and then the superheater, reheater,

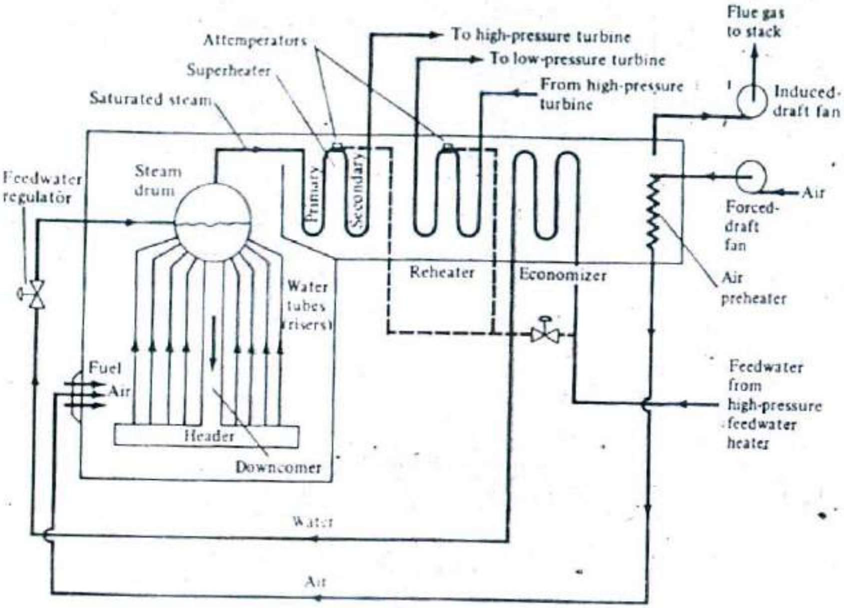

Figure 3-6 Schematic flow diagram of a modern steam generator.

and economizer, and leave the latter at about 600°F. From there they reheat the incoming atmospheric air in the air preheater, leaving it at about 300°F. An *induced-draft (ID) fan* draws the flue gases from the system and sends them up the stack. The temperature of about 300°F of the exiting flue gas represents an availability loss to the plant. This, however, is deemed acceptable because (1) the gas temperature should be kept well above the dew point of the water vapor in the gases (equal to the saturation temperature of water at the partial pressure of the water vapor) to prevent condensation which would form acids that would corrode metal components in its path, and (2) the flue gases must have enough buoyancy to rise in a high plume above the stack for proper atmospheric dispersion.

Example 3-1 A steam generator burns fuel oil with 20 percent excess air. The fuel oil may be represented by $C_{12}H_{26}$. Determine the minimum stack temperature needed to avoid condensation. Assume the flue gas pressure is leaving the air preheater at 45 psia.

SOLUTION Recalling that there are 3.76 mol N_2/mol O_2 in atmospheric air, the combustion equation for stoichiometric (chemically correct mixture) is

$$C_{12}H_{26} + 18.5O_2 + 69.56N_2 \rightarrow 12CO_2 + 13H_2O + 69.56N_2$$

With 20 percent excess air

$$C_{12}H_{24} + 22.2N_2 + 83.473N_2 \rightarrow 12CO_2 + 13H_2O + 3.7O_2 + 83.472N_2$$

The partial pressure of any component in a gas mixture is equal to the total pressure times the mole fraction of that component. Thus partial pressure of H_2O in products is

$$45 \times \frac{13}{12 + 13 + 3.7 + 83.472} = 5.215 \text{ psia}$$

This corresponds to a saturation temperature of 164°F. The average temperature of the gases is kept much higher to avoid local cool spots that might cause condensation.

The Boiler Walls

The water tubes that cool the water walls are closely spaced for maximum heat absorption. Tube construction has varied over the years (Fig. 3-7) from *bare* tubes (*a*) tangent to or (*b*) embedded in the refractory, to (*c*) *studded* tubes, to the now-common *membrane* design (*d*). The membrane design consists of tubes spaced on centers slightly wider than their diameter, connected by bars or membranes welded to the tubes at their centerlines. The membranes act as fins to increase the heat transfer as well as to afford a continuous rigid and pressure-tight construction for the furnace. No additional inner casing is required to contain the combustion gases. Insulation and metal lagging to protect it are provided on the outer side of the wall. One manufacturer has standardized its design on 3-in-diameter tubes on 3.75 in between centers, another on 3 in on 4 in, and yet a third on 2.75 in on 3.75 in.

The Radiant Boiler

Heat is transferred from the combustion gas to the water walls by both radiation and convection. A *radiant boiler*, as the name implies, receives most of its heat by radiation.

The combustion gases have characteristics that depend upon the fuel used, the combustion process, and the air-fuel ratio. They may be *luminous*, i.e., emit all wavelengths and hence strong visible radiation if there are particulates such as soot particles during the combustion process. This is the case with coal and oil. They may be *nonluminous*, in which case they burn cleanly without particulates, as is the case of gaseous fuels. No combustion gases are truly nonluminous because the heavier gases in the combustion products, in particular the triatomic CO_2 and H_2O, but also SO_2, ammonia, and sulfur dioxide, are selective radiators that emit (and absorb) radiation in certain wavelengths, mostly outside the visible range. The portion of radiation within the visible range is small but gives the combustion gases a green-blue appearance. Lighter gases such as the monatomic and diatomic gases are poor radiators.

The radiant energy emitted by the combustion gases depends upon the gas temperature (to the fourth power), the partial pressures of the individual constituent ra-

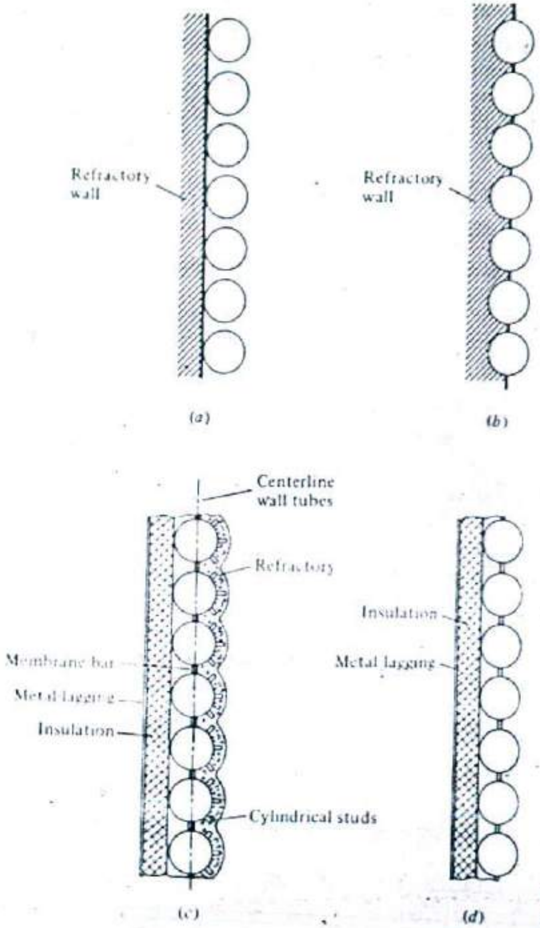

Figure 3-7 Plan views of different types of water tubes.

diating gases, the shape and size of the gases, their proximity to the absorbing body, and the temperature of that body (to the fourth power) [14].

The convective portion of the heat transfer follows the usual Nusselt-Reynolds turbulent forced-convection relationship. It is smaller than the radiant portion because radiation is caused by a thick body of gas, whereas convection is localized near the tube surface.

Heat received by the water walls is conducted through the membranes and tube walls and is then convected to the two-phase mixture inside the tubes by forced-convection nucleate-boiling heat transfer. The heat-transfer resistance of the latter is much smaller than the others so that it may be neglected in design calculations with little error.

Radiant boilers are designed for electric-generating stations to use coal or lignite for pulverized or cyclone furnace applications, oil, or natural gas. They are built to supply a wide range of steam pressures and temperatures, but usually around 1000°F (about 540°C) and steam capacities up to 10×10^6 lb_m/h (1260 kg/s). They are limited to subcritical pressures, usually 1800 to 2500 psig (about 125 to 170 bar).

3-5 WATER CIRCULATION

Water circulates from the steam drum via downcomer pipes to a bottom header, up the water tubes (which act as risers), where it partially boils, and back to the steam drum. Full boiling to 100 percent quality in the tubes is avoided because it would lead to tube burnout or failure as a result of *departure from nucleate boiling* (DNB). The density of the saturated water in the downcomers is greater than the average density of the two-phase mixture in the risers.

Natural circulation is dependent upon the difference between these two densities and the height of the drum above the bottom headers. Most large steam-generator boilers have sufficient natural-circulation driving force and are called *natural-circulation boilers*. Some require additional help by pumping the single-phase flow and are called *controlled-* or *forced-circulation boilers*. For a number of years, forced circulation was favored over natural circulation for relatively high subcritical pressures of about 2300 psig (160 bar) and higher. At such pressures the difference between water and steam densities rapidly decreases (becoming zero at the critical point, 3208 psia, 221 bar). This reduced natural circulation and required a margin of safety against tube failure and burnout due to DNB, i.e., reaching the critical heat flux (also a major concern in water-cooled nuclear-reactor design), if a particular tube receives reduced flow. A pump assist supplied this margin of safety.

More recently the water tubes in high-heat-absorbing areas of the furnace were provided with such devices as internal twisters and springs that would break the vapor film and thus inhibit or retard the onset of DNB. Another approach was to use tubes that were grooved, corrugated, or ribbed on their inside surface. The most recent and satisfactory answer has been tubes that were ribbed (rifled) helically on their inside surface. This ribbing creates a centrifugal action that directs water droplets to the vapor film clinging to the surface and provides a much greater margin of safety against DNB problems than do smooth tubes. Modern large steam-generator boilers can now be designed with natural circulation.

Natural-circulation driving forces are explained with the simplified flow diagram of Fig. 3-8. The *driving pressure* caused by natural circulation, Δp_d, is given by

$$\Delta p_d = (\rho_{dc} - \bar{\rho}_r) H \frac{g}{g_c} \tag{3-1}$$

where Δp_d = driving pressure, lb_f/ft^2 or N/m^2 (Pa)

ρ_{dc} = density of water in the downcomer, nearly saturated at the system pressure, lb_m/ft^3 or kg/m^3

$\bar{\rho}_r$ = average density of steam-water mixture in the riser, $\mathrm{lb_m/ft^3}$ or $\mathrm{kg/m^3}$

H = height of drum-water level above bottom header, ft or m

g = gravitational acceleration $\mathrm{ft/s^2}$ or $\mathrm{m/s^2}$

g_c = conversion factor 32.2 $\mathrm{lb_m \cdot ft/(lb_f \cdot s^2)}$ or 1 $\mathrm{kg/(N \cdot s^2)}$

The most difficult of the parameters in Eq. (3-1) to obtain is $\bar{\rho}_r$, the average density in the riser. It is a function of the *void fraction* distribution along the riser height. The void fraction α of a two-phase mixture is a volumetric quality defined as

$$\alpha = \frac{\text{volume of vapor}}{\text{volume of vapor + liquid}} \tag{3-2}$$

as opposed to the quality x, which is a mass quality, α and x are related by [2]

$$\alpha = \frac{1}{1 + [(1 - x)/x]\psi} \tag{3-3a}$$

and

$$x = \frac{1}{1 + [(1 - \alpha)/\alpha]\dfrac{1}{\psi}} \tag{3-3b}$$

where

$$\psi = \frac{v_f}{v_g} S \tag{3-3c}$$

where v_f and v_g are the specific volumes of the saturated liquid and vapor, respectively, at the system pressure, and S is the *slip ratio* of the two-phase mixture. The two phases of that mixture do not travel at the same speed. Instead there is a slip between them, which causes the vapor to move faster than the liquid. S is a dimensionless number, greater than 1, defined as

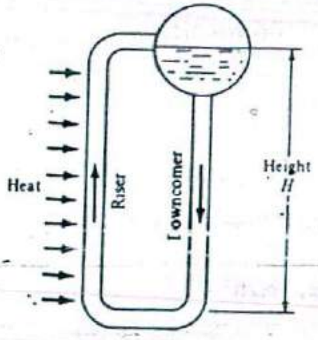

Figure 3-8 A natural circulation loop.

$$S = \frac{\overline{V}_{r,g}}{\overline{V}_{r,f}} \tag{3-4}$$

where $\overline{V}_{r,g}$ and $\overline{V}_{r,f}$ are the average vapor and average liquid velocities at any one cross section of the riser. S has been measured experimentally and found to vary between 1 to less than 10 in most systems, approaching 1 at high pressures (where the liquid and vapor densities approach each other). It is, in general, fairly constant along the path length.

The axial heat flux distribution to the riser determines the quality distribution. In turn, using a reasonable value for S (between 1 and 2), a void fraction distribution is obtained. A mixture density distribution ρ_m is now found from

$$\rho_m = (1 - \alpha)\rho_f + \alpha\rho_g \tag{3-5}$$

where ρ_f and ρ_g are densities (reciprocals of the specific volumes) of the saturated liquid and vapor, respectively. The average mixture distribution in the riser $\overline{\rho}_r$ is now obtained from

$$\overline{\rho}_r = \frac{\int_o^H \rho_m(z)\,dz}{H} \tag{3-6}$$

where z is the axial distance from the bottom of the riser. In the case of uniform axial heating, the solution of the above integral is [2]

$$\overline{\rho}_r = \rho_f - \frac{\rho_f - \rho_g}{1 - \psi}\left\{1 - \left[\frac{1}{\alpha_e(1 - \psi)} - 1\right]\ln\frac{1}{1 - \alpha_e(1 - \psi)}\right\} \tag{3-7}$$

where α_e is the riser exit void fraction.

The driving pressure given by Eq. (3-1) should balance the pressure losses of the single- and two-phase fluids in the loop, which are proportional to their mass-flow rate to the second power. The system seeks its own equilibrium; that is, if the driving pressure is greater than the losses, more fluid is pushed through, causing more losses but also causing x and α to become lower (for the same heat flux) and $\rho_{dc} - \overline{\rho}_r$ to be reduced until equilibrium is reached. The reverse is also true. If the driving pressure is too low for the desired flow rate, a pump is added to assist in circulation.

Example 3-2 A 40-ft-high downcomer-riser system operates at 2500 psia. The riser receives uniform heat flux and saturated water. The exit quality is 50 percent. Calculate the driving pressure. Take $S = 1.2$.

SOLUTION Using Eq. (3-3*c*)

$$\psi = \frac{v_f}{v_g}S = \frac{0.02859}{0.13068} \times 1.2 = 0.2625$$

Using Eq. (3-3*a*)

$$\alpha_e = \frac{1}{1 + [(1 - x_e)/x_e]\psi} = \frac{1}{1 + (0.5/0.5) \times 0.2625} = 0.8879$$

Using Eq. (3-7)

$$\bar{\rho}_r = 34.977 - \frac{34.977 - 7.652}{1 - 0.2625}\left\{1 - \left[\frac{1}{0.8879(1 - 0.2625)} - 1\right] \ln \frac{1}{1 - 0.8879(1 - 0.2625)}\right\} = 18.701 \text{ lb}_m/\text{ft}^3$$

$$\rho_{dc} = \rho_f = 34.977 \text{ lb}_m/\text{ft}^3$$

Use Eq. (3-1) to find the driving pressure

$$\Delta p_d = (34.977 - 18.701)(40)\left(\frac{32.2}{32.2}\right) = 651.06 \text{ lb}_f/\text{ft}^2$$

$$= 4.521 \text{ psi}$$

3-6 THE STEAM DRUM

The steam drum provided in all modern steam generators except once-through types (Sec. 3-8) is where feedwater from the economizer is fed, saturated steam is separated from the boiling water, and the remaining water is recirculated as discussed above. The drum may also be used for chemical water treatment and blowdown to reduce solids in the water. The drum also must be of sufficient volume to accommodate water level changes caused by load changes (assisted by proper steam generator controls) and to prevent a dangerously low level or the "carryover" of water toward the superheater. This would cause deposits of entrained solids in the superheater tubes and thus materially increase their temperature, which would lead to their distortion or burnout. Carryover of solids with the steam has also been known to have far-reaching effects, such as deposit problems on turbine blades (the most troublesome being silica deposits, which are not easily removed by water washing).

The most important steam-drum function is separating the steam from the boiling water. The simplest method is *gravity separation* (Fig. 3-9*a*). If the steam velocity

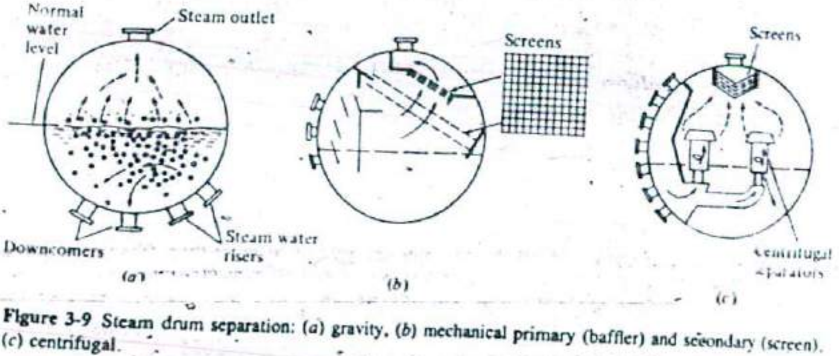

Figure 3-9 Steam drum separation: (*a*) gravity, (*b*) mechanical primary (baffler) and secondary (screen), (*c*) centrifugal.

leaving the water surface is low enough (below about 3 ft/s), the steam bubbles separate naturally without entraining water droplets and the solids they carry (carryover), and without being drawn with the recirculating water into the downcomers (carryunder). Factors other than velocity that affect this process are the positions of the downcomer and riser nozzles with respect to the steam outlet, usually at the top of the drum, and the operating pressure. Gravity separation is strongly affected by the difference in steam and water densities. The higher the pressure, the less the difference between these densities and the less effective the separation. Thus gravity separation, while requiring a simple drum, is economical only for low-steam-capacity, low-pressure service.

In modern high-capacity, high-pressure boilers, *mechanical separation* that assists or supplements gravity separation takes place in two steps: primary and secondary. *Primary separation* removes most of the water from the steam and prevents the carryunder of steam with the recirculating water to downcomers and risers. *Secondary separation*, also called *steam scrubbing* or *drying*, removes remaining mist or fine droplets and the solids they carry from the steam, which results in pure, or "dry and saturated," steam going to the superheater. Mechanical separation is accomplished by fitting the drums with baffles, screens, bent or corrugated plates, and centrifugal separators.

Baffle plates act as primary separators. They change or reverse the steam-flow direction (Fig. 3-9*b*), thus assisting gravity separation, and act as impact plates that cause the water to drain off. Screens made of wire mesh act as secondary separators where the individual wires attract and intercept the fine droplets, much like fabric filters attract dust from gases (Chap. 17). The accumulating drops then fall by gravity back to the main body of water. Bent or corrugated plates are used for both primary and secondary separation. Their effectiveness derives from their large ratio of surface to projected areas. The above plates and screens are used in many other configurations to maximize gravity separation [12, 13].

At high pressures, where the density differential between water and steam diminishes, centrifugal forces, much greater than the gravity forces, are used. *Centrifugal separation* devices are also called *cyclone* or *turbo separators*. They provide separation at pressures nearing critical. In a typical centrifugal separator (Fig. 3-9*c*), the mixture coming in from the risers is deflected tangentially downward into the main body of water. It then enters the separators, which are arranged along the length of the main steam drum. Guide vanes within the separators impart a spinning motion to the mixture, which causes the heavier water droplets to move radially through the lighter steam, to impinge on the separator wall, and to discharge downward below the water surface through an outer concentric cylinder. The separator may be equipped with a corrugated plate at its exit to provide further separation. Finally, screens, just under the drum exit, provide the final drying action.

Typical utility steam drums range in length to more than 100 ft, in diameter to more than 15 ft, and in mass to a few hundred tons. They contain as many as 30 outlet nozzles and many more riser and downcomer nozzles. The larger drums are usually constricted in cylindrical sections, called *courses*, which are welded together, and two hemispherical heads which are welded to the ends.

3-7 SUPERHEATERS AND REHEATERS

Superheaters and reheaters in utility steam generators are made of tubes of 2 to 3 in OD (smaller sizes, about half these dimensions, are used in marine service). The smaller diameters have lower pressure stresses and withstand them better. The larger diameters have lower steam-flow pressure drops and are easier to align. Finning on the outside surface of the tubes is avoided because it increases thermal stresses and makes cleaning difficult. Internal ribbing, like that used in boiler water tubes, is unnecessary because no DNB problems arise here. Adequate heat-transfer design is based on gas flow inside the tubes, which has a much lower conductance than nucleate boiling in the boiler tubes. Because the tubes are subjected to high temperatures, pressures, and thermal stresses, their materials of construction must be carefully selected. Below 850°F carbon steel is adequate. Modern superheaters and reheaters operating at about 1000°F, however, are usually made of special high-strength alloy steels chosen for both strength and corrosion resistance [15]. The exact alloy depends upon steam conditions and the types of fuel, especially if it contains undesirable impurities. The allowable stresses for materials drop drastically as the temperature increases.[*]

Convection Superheater

Early superheater designs placed them above or behind banks of water tubes to protect them from combustion flames and high temperatures. The main mode of heat transfer between the combustion gases and the superheater tubes, therefore, was convection, and that type of superheater became known as the *convection superheater*. The main distinguishing characteristic is its response to load changes. As demand for steam increases, fuel- and airflow, and hence combustion-gas flow, are increased. The convective heat-transfer coefficients increase both inside and outside the tubes, increasing the overall heat-transfer coefficient between gas and steam faster than the increase in mass-flow rate of the steam alone. (The combustion temperatures do not materially change with load.) Thus the steam receives greater heat transfer per unit mass-flow rate and its temperature increases with load (Fig. 3-10).

Radiant Superheater

Because of the need for greater heat absorption, superheaters were eventually placed nearer higher-temperature, in view of the combustion flames. Steam-flow velocities were increased to increase the overall heat-transfer coefficients, and overall superheater designs were improved to overcome expected higher metal temperatures.

This placement of superheater results in the main heat transfer between the hot gases and flame, and the tube outer walls, to be accomplished by radiation. This design

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[*] For example, a material called Croloy 21, specification number SA213, Grade T22, has maximum allowable design stresses of 13,100, 11,000 and 7,800 psi at 900, 950, and 1000°F, respectively. Carbon steel-SA210-A1 has allowable stresses of only 5000 and 3000 psi at 900 and 950°F, respectively.

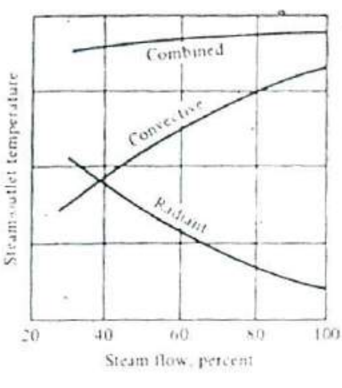


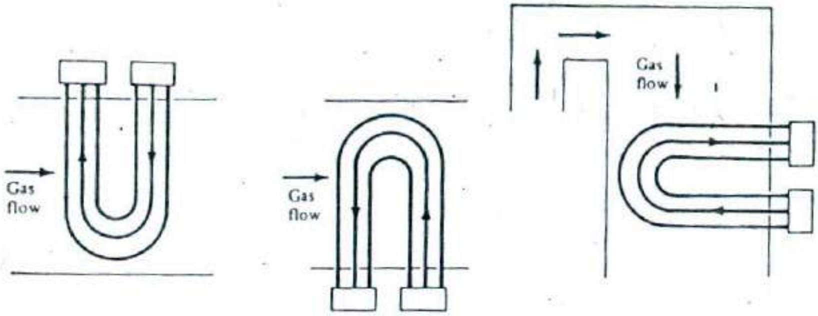
Figure 3-10 Exit-temperature response of convective, radiant, and combined (in-series) superheaters.

has come to be known as a *radiant superheater*. Radiation heat transfer is proportional to $T_f^4 - T_w^4$, where T_f and T_w are the flame and tube wall absolute temperatures, respectively. Because T_f is much greater than T_w, the heat transfer is essentially proportional to T_f^4. Because T_f is not strongly dependent on load, the heat transfer per unit mass flow of steam decreases as the steam flow increases. Thus an increase in steam flow due to an increased load demand would result in a reduction in exit steam temperature, the opposite effect of a convection superheater (Fig. 3-10).

Design considerations for reheaters are similar to those for superheaters except that, although the steam outlet temperatures are about the same, the overall temperatures are lower and the steam pressures are about 20 to 25 percent of those in the superheaters. The pressure stresses are therefore lower and a lower grade steel alloy is tolerated. In addition, larger tubing with higher stresses may be used, which has the additional beneficial effect of reducing the pressure losses in the reheater.

Convection superheaters alone are used with low-temperature steam generators. Radiant and convection superheaters and reheaters are used for high-temperature service. The radiant units are arranged in flat panels or platen sections with wide spacings of several feet to permit radiation through. These are usually followed downstream by sections on a narrower spacing that permit both radiation and convection. Mechanical construction of the sections are of three kinds: *pendant*, *inverted*, and *horizontal*.

Pendant-type superheaters and reheaters are those that are hung from above (Fig. 3-11*a*). They have the advantage of firm structural support but the disadvantage of flow blockage by condensed steam after a cold shutdown, which necessitates slow restart to purge the water that accumulates in the bottom. *Inverted-type* units, on the other hand, are supported from below (Fig. 3-11*b*). They have proper drainage of the condensed steam but lack the structural rigidity of the pendant type, especially in high-speed gas flow. The inverted type is not commonly used. *Horizontal-type* units (Fig. 3-11*c*) are usually supported horizontally in the vertical gas ducts parallel to the main furnace and receive the hot gases after a U-turn at the top. They do not view the flame directly and hence are mainly of the convection type. They have both proper drainage and good structural rigidity. Figure 3-12 shows a typical arrangement of superheaters



(a) (b) (c)

Figure 3-11 Schematic diagram showing (*a*) pendant, (*b*) inverted, and (*c*) horizontal superheaters and reheaters.

Figure 3-12 Superheater (SH), reheater (RH), economizer, and air preheater arrangements in a drum-type steam generator with cyclone furnace. (*Courtesy Babcock and Wilcox.*)

and reheaters. Superheaters and reheaters are often split into primary and secondary units for control purposes (Sec. 3-13).

3-8 ONCE-THROUGH BOILERS

The *once-through* boiler, or steam generator, is also called the *forced-circulation*, *Benson*, or *universal-pressure* boiler. The latter name is because it is applicable to all temperatures and pressures, although economically it is suited to large sizes and pressures in the high subcritical and supercritical range. In contrast to the drum type (Fig. 3-13*a*), the feedwater goes through the economizer, furnace walls, and superheater sections, changing sequentially to saturated water, saturated steam, and superheated steam in one continuous pass (Fig. 3-13*b*). No steam drum is required to separate saturated steam from boiling water and no water recirculation takes place. Reheat of steam after it is expanded in the high-pressure turbine is accomplished by a reheater in the usual manner. Because of the once-through mode of operation, very high purity feedwater is a requirement. Figure 3-14 shows a typical once-through steam generator.

The once-through boiler is the only type suited to supercritical-pressure operation (above 3208 psia; 221 bar, for steam) because the latent heat of vaporization at and beyond the critical pressure is zero and liquid and vapor are one and the same, so no separation in a drum is possible or necessary. While particularly applicable to supercritical pressures, once-through steam generators are used economically for high-

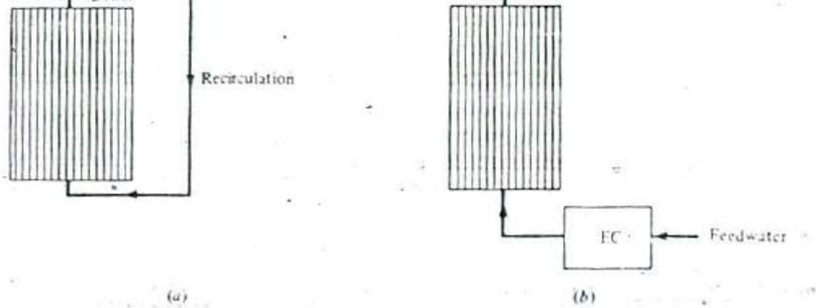


Figure 3-13 Schematic flow diagrams of (*a*) drum type and (*b*) once-through steam generators. SU = superheater, EC = economizer.

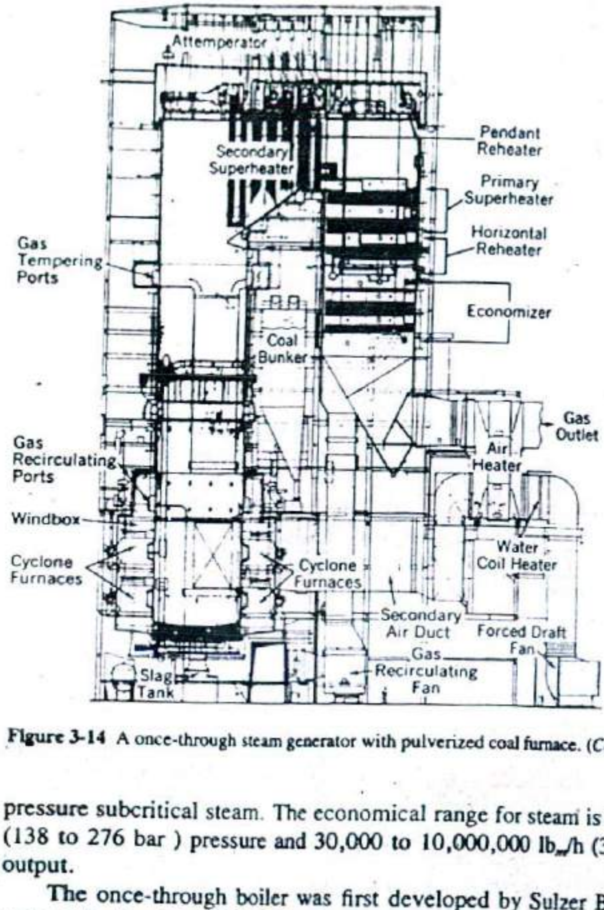


Figure 3-14 A once-through steam generator with pulverized coal furnace. (*Courtesy Babcock and Wilcox.*)

pressure subcritical steam. The economical range for steam is from 2000 to 4000 psia (138 to 276 bar) pressure and 30,000 to 10,000,000 lb_m/h (3.8 to 1260 kg/s) steam output.

The once-through boiler was first developed by Sulzer Brothers, Ltd., of Switzerland in the late 1920s. The first commercial unit was installed in 1932. It was a subcritical unit and was followed by many units whose range was 1200 to 2400 psig (84 to 167 bar).

A large number of pioneering supercritical-pressure once-through steam generators were built for the utility industry, many with double reheat, especially during the 1940s and 1950s. Some used advanced steam pressures of 4500 to 5000 psig (310 to 345 bar) and steam temperatures of 1150 to 1200°F (620 to 650°C) and cross-compound (two-shaft) turbines. Operational and economic considerations, however, led to the use of more moderate steam conditions of 3500 psig and 1000 to 1050°F. Further optimization by the late 1960s led to the development and use of 3500/1000/1025/1050 plants with single-shaft four-casing steam turbines. During the 1970s however, such units fell out of favor because of equipment unreliability, significantly mostly in the low-pressure components of the plants, as well as operational complexity. It is

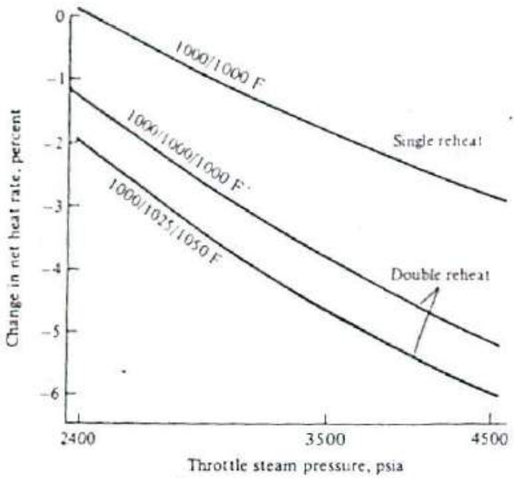


Figure 3-15 Effect of steam conditions on heat rate [16].

now believed that newly developed sophisticated schemes for boiler-turbine operation will largely overcome these difficulties [16] and that the large improvements in efficiency and reduction in heat rate possible with supercritical-double-reheat plants (Fig. 3-15) may reverse the downward trend of the 1970s. Under development, for example, is a 900-MW plant with steam conditions of 4500/1000/1025/1050 that will have a 6 percent improvement in heat rate over the currently popular 2400/1000/1000 plant.

Capital costs for a supercritical steam generator are a few percent higher than those of a drum-type subcritical one of the same capacity, but because of the increased efficiency of the powerplant the capital costs of the turbogenerator as well as the balance of plant (condenser, feedwater heaters, cooling towers, etc.) are lower. The net effect is that the total production costs of electricity in mills per kilowatt hour are lower for the supercritical cycle, particularly one with double reheat.

A *combined-circulation* boiler, developed by Combustion Engineering, Inc., combines once-through flow with circulation for use in supercritical-pressure operation [13]. Recirculation is used only during start-up and low loads in a spherical mixing vessel to protect the furnace walls. A stop-check valve assures once-through flow alone at high loads. A typical unit is 805 MW single reheat 3590/1005/1005.

3-9 ECONOMIZERS

The *economizer* (EC) is the heat exchanger that raises the temperature of the water leaving the highest-pressure feedwater heater to the saturation temperature corresponding to the boiler pressure. This is done by gases leaving the last superheater or reheater. These gases, at high enough temperatures to transfer heat to the superheater-reheaters, enter the economizer at 700 to 1000°F. Part of their energy is used to heat the feedwater. The term "economizer" historically was used because the discharge of such high-temperature gases would have caused a large loss in availability and efficiency and hence loss in economy of operation.

Economizers were introduced before feedwater heating, which meant very low inlet water temperatures to the economizers and consequently low tube outer wall temperatures, below the dew point of the flue gases. This caused condensation and corrosion because of the presence of SO_2 and SO_3 in the gases. The moisture also aided in the collection of ash, thus fouling the tube outer surfaces that reduced heat transfer. Economizers were made of cast iron and had mechanical scrapers for cleaning. Early in its use, steel suffered from corrosive attack of the insides of the tubes by oxygen freed from the feedwater as its temperature rose in the economizer.

Modern steam generators receive heated feedwater and their economizers operate above the dew point of the gases, thus eliminating external corrosion and fouling. Chemical cleaning of internal surfaces is also used [12]. Also, much of the feedwater oxygen is removed in the deaerating feedwater heater (Sec. 2-8) at or above 212°F, which reduces internal corrosion. This is also aided by maintaining the water in the economizer at a pH of 8 to 9. These advances permitted the use of steel, which in turn is suitable for the high pressures encountered in modern economizers.

Modern economizers are designed to allow some boiling of the feedwater in the outlet sections, up to 20 percent quality at full power, less at part loads.

Economizer tubes are commonly 1.75 to 2.75 in OD and are made in vertical sections of continuous tubes, between inlet to outlet headers, with each section formed into several horizontal paths connected by 180° vertical bends for proper draining. Sections are placed side by side on 1.75- to 2-in minimum spacings (edge to edge). The exact spacing depends upon the type of fuel and ash characteristics, which are smaller the cleaner the fuel, such as natural gas. When high-ash fuels are used, the water-soluble ashes accumulating on the economizer are dissolved and washed off during plant shutdown. In that case the economizer is usually located above a hopper, which receives the dissolved deposits. Steam or air-jet cleaning is also used in addition to washing.

Economizers have been built with plain or extended surface tubes. Extended surface tubes with fins or studs on their outer surface have higher heat-transfer characteristics and thus require smaller space. On balance, they have lower capital cost. They are, however, more suited to clean-burning gaseous fuels and situations in which no air preheaters are used, such as in combined steam-gas-turbine cycles (Chap. 8).

Economizers are generally placed between the last superheater-reheater and the air preheater. In some cases, a low-temperature economizer is placed after the air preheater. Such an economizer is called a *stack cooler* and acts as a low-pressure feedwater heater except that the heating medium is the flue gas instead of steam bled from the turbine.

3-10 AIR PREHEATERS

Like economizers, *air preheaters* (or simply air heaters) utilize some of the energy left in the flue gases before exhausting them to the atmosphere. They receive 600 to

800°F gases. As indicated previously, these gases are cooled to only 275 to 350°F to avoid gas condensation and corrosion problems and to allow for proper dispersion in the atmosphere. The air is heated from forced-draft outlet temperatures, not far from atmospheric to 500 to 650°F and sometimes higher. Preheating the air saves fuel that would otherwise be used for that heating. The fuel savings (and hence increase in plant efficiency) are nearly directly proportional to the air temperature rise in the preheater. Typical fuel savings are 4 percent for a 200°F air temperature rise and about 11 percent for a 500°F air temperature rise in the preheater.

In addition to fuel savings, preheated air is a requirement for the operation of pulverized-coal furnaces. Air in the 300 to 600°F range is needed for drying that fuel. Air is also used for transporting it to the furnace and then burning it there. Small stoker-fired units do not require preheated air. However, large stoker-fired bituminous-coal steam generators benefit from preheated air, but only up to about 350°F, to prevent damage to the stoker moving parts.

Air preheaters, like economizers, were first developed in Europe. The first unit commercially built in the United States was a flat-plate heat exchanger in which adjacent steel plates formed alternate air and gas passages. Present day higher air and gas pressures, however, use tubular or regenerative designs that better withstand these pressures. There are two general types of air preheaters: *recuperative* and *regenerative*.

Recuperative air preheaters have heat transferred directly from the hot gases to the air across the heat-exchange surface. They are commonly tubular, although some plate types are still used. Tubular units are essentially counterflow shell-and-tube heat exchangers in which the hot gases flow inside vertical or horizontal straight tubes and the air flows outside. Figure 3-16 shows a vertical preheater. Baffles are provided to maximize air contact with the hot tubes. The tubes are mechanically expanded into top and bottom tube sheets. Thermal expansion is provided by a bellows-type expansion unit shown at the bottom sheet. A hopper is provided below the tubes to collect soot and dust that deposit on the inside tube surfaces. Care is taken to avoid leakage between the air and the hot gases. Leakage would occur from the air that is at a higher pressure to the gas. A leak would result in short-circuiting of combustion air, which would increase both forced- and induced-draft-fan power consumption in direct proportion to the leakage. It also reduces the heater effectiveness.

Tubular preheaters are built in a variety of designs to suit particular steam generators spaces and duct layout. They may provide for one or more passes for both air and gas in counter- or crossflow, in vertical or horizontal arrangements. Tube sizes vary from 1.5 to 4 in OD. The smaller the diameter, the larger the number of tubes and the greater the surface area for a given overall size. Thus smaller diameters result in more compact heaters of a given heat load. Cost, cleaning requirements, and fuel type determine the diameter to be used in a given situation. Modern large steam generators use 2 to 2.5 in OD heater tubes. Diameters of 1.5 to 2 in are used in marine service where space and weight limitations are important.

Regenerative air preheaters are those in which heat is transferred from the hot flue gases, first to an intermediate heat-storage medium, then to the air. The most common is the rotary air preheater, known as the *Ljungstrom* preheater, first developed in Europe in 1920 and first installed in the United States in 1923. It is composed of

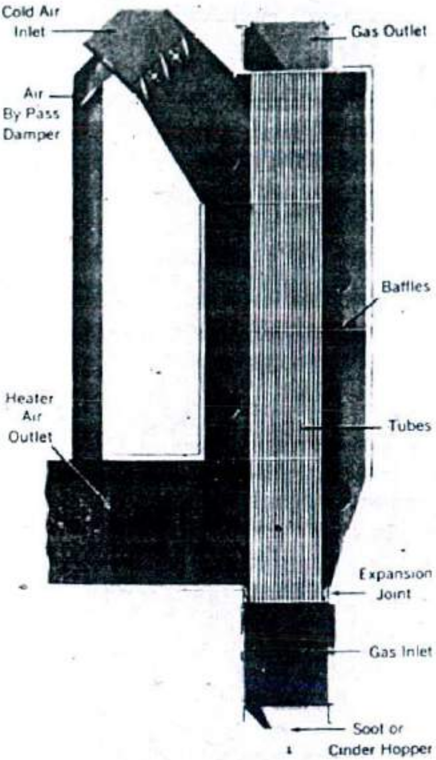


Figure 3-16 a tubular counterflow air preheater. Air bypass is used to control metal temperatures at air inlet end. (*Courtesy Babcock and Wilcox.*)

a rotor driven by an electric motor through reduction gearing so that it rotates slowly and continuously within a housing at 1 to 3 r/min, depending on diameter. The rotor has between 12 and 24 radial members that form sectors (Fig. 3-17). The sectors are filled with a heating surface composed of steel sheets that are flat or form-pressed with corrugated, notched, or undulated ribbing and formed into baskets. They constitute the heat-storage medium of the preheater. A stationary seal covers the equivalent of two opposite sectors. Half the remaining sectors are exposed at any one instant to the hot gases, which are moving in one direction, the other half are exposed to the air, which is moving in the opposite direction.

As the rotating sectors enter the hot-gas zone, they are progressively heated by the gas. They store that heat as sensible heat. When they enter the air zone, they progressively give up this heat to the air. The seal system reduces leakage.

Rotary air preheaters are designed with a vertical or horizontal shaft, depending upon layout and ducting. They are also designed as *laminar-* or *turbulent-surface* types. In the laminar type, the heat-storage elements are compactly spaced so that flow through them is laminar. They are used with clean-burning gaseous fuels. The turbulent

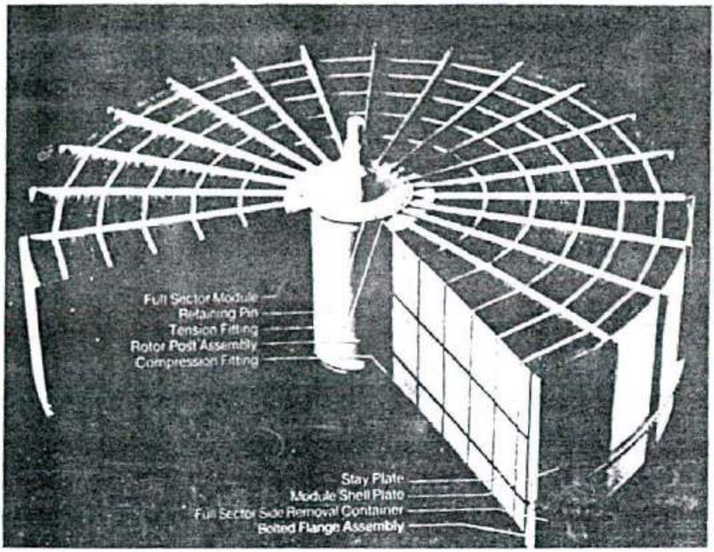


Figure 3-17 A vertical shaft Ljungstrom air preheater (*Courtesy Combustion Engineering, Inc.*)

type has elements with wider spacings (to facilitate cleaning of ash deposits) and turbulent flow in between. It is more suited to coal- and oil-fired systems. To get the same mass of heat-storage elements, the turbulent-type rotor is several times longer than the laminar-type and is, in general, vertically mounted, whereas the laminar type is usually horizontally mounted. Cleaning is accomplished by air or steam jets.

Another type of rotary air preheater, recently developed in Europe, is one in which the storage elements are stationary and the gas and air ducts are connected to two rotating segments each.

3-11 FANS

Early and small-capacity steam generators relied upon natural draft for the combustion gases. Proper design of gas passages must be provided to have the driving pressure between the atmosphere and the gases within the generator, caused by the density difference between atmospheric air and the average gas density, overcome the various pressure losses within the generator and supply the required air for combustion. Analogy here may be made with natural circulation in the boiler (Sec. 3-5).

Large steam generators require an assist to push the air in, pull the gas out, or both. For this, they use large fans [17, 18]. There are two types of fans in use today: *forced-draft (FD)* and *induced-draft (ID)* fans. When either one is used alone, it should overcome the total air and gas pressure losses within the generator.

Forced-draft fans, used alone as in many large steam generators and practically all marine applications, are placed at the air entrance to the air preheater and put the entire system up to the stack entrance under positive gage pressure. Because they handle only cold air, they have several advantages over induced-draft fans. Some advantages:

1. They have lower maintenance problems.
2. They consume much less power because the cold air has the lowest specific volume in air-gas path. Recall (Sec. 1-2) that a fan is a steady-flow thermodynamic system so that the work w_{sf} per unit mass-flow rate is given by

$$w_{sf} = \int v\,dP \tag{1-9}$$

   where v is the specific volume, being lowest for the cold air entering.
3. Their load is reduced by the absence of the additional gas equivalent of the fuel added.
4. As a consequence of the above, their capital and operating costs are lower.

Disadvantages result from the fact that they put the furnace under pressure, in which case it is called a *pressure furnace:*

1. Leakage of noxious gases from the furnace walls would be to the outside, necessitating a gas-tight furnace construction.
2. Special attention must be given to the design of inspection doors, soot blower boxes, and fuel-igniter openings.

For good reliability two forced-draft fans, operating in parallel are usually used, each one capable of at least 60 percent of full load flow when the other is out of service.

Induced-draft fans are located in the gas stream between the air preheater and the stack, either before or after the dust collector (Chap 18). They discharge essentially at atmospheric pressure and place the entire system under negative gage pressure. They must handle hot gases, including the original air, the gas equivalent of the fuel added, and leakages into the system. Their power requirements are therefore greater than forced-draft fans. In addition they must cope with corrosive combustion products and ash. Induced-draft fans are seldom used alone.

When both forced- and induced-draft fans are used in a steam generator, the FD fans push atmospheric air through the air preheater, dampers, various air ducts, and burners into the furnace. The ID fans pull the combustion gases from the furnace, through the heat-transfer surfaces in the superheaters, reheaters, economizer, and gas side air preheater and into the stack. (Sometimes they are built into the stack base.) The stack, because of its height, adds a natural driving pressure of its own (Sec. 3-12). The furnace in this case is said to operate with *balanced draft;* meaning that the pressure in it is approximately atmospheric. Actually, it is kept at a slightly negative gage pressure to ensure that any leakage would be inward.

Steam generators that use low-ash fuels such as gas or oil are usually designed with pressurized firing. Coal-fired generators may be designed with either pressurized- or balanced-draft firing. Modern systems seem to favor the latter.

Fans used in electric-generating plants are among the largest made; capacities of 1.5 million ft³/min (700 m³/s) and 60-in water static pressures (about 2.2 psi, 0.15 bar) are common. Because they operate continuously for long periods (up to 1 or 1½ years), the fans must be well designed, ruggedly constructed, well balanced, and highly efficient over a wide range of outputs. There are two types of fans in common use: *centrifugal* and *axial*. In the centrifugal fan, the gases are accelerated radially through curved or flat impeller blades from rotor to a spiral or volute housing. In the axial fan, gases are accelerated parallel to the rotor axis. This is similar to a desk fan, but here the fan is housed in a casing to develop static pressure. Axial fans (with variable-pitch moving blades) maintain high efficiencies over a wider range of loads than constant-speed centrifugal fans but have higher capital costs.

In general, centrifugal fans with backward-curved blading are used for FD fans and with flat or forward-curved blading (Fig. 3-18) for ID fans. (Occasionally backward-curved blading but with curvature less than that for FD fans is used.) The lesser curvature results in lower tip speed and allows less dirt to cling to the backside of the blades, thus minimizing the erosive effect of ash. Low-speed fans with flat blades are used with particularly dirty or corrosive gases.

Because the pressure differential ΔP across a fan is usually small, the airflow or gas flow may be considered incompressible, that is v = constant, and Eq. (1-9) may be modified to

$$|w_{sf}| = \frac{v\,\Delta P}{\eta_f} \qquad \text{ft}\cdot\text{lb}_f/\text{lb}_m \text{ or J/kg} \tag{3-8}$$

The power would be given by

$$\dot{W}_{sf} = \frac{\dot{m}v\,\Delta P}{\eta_f} \qquad \text{ft}\cdot\text{lb}_f/\text{s or W} \tag{3-9}$$

where v = specific volume of air or gas, obtained from the perfect gas equations (1-30), ft³/lb$_m$ or m³/kg

ΔP = pressure rise across fan, lb$_f$/ft² or N/m² (Pa)

η_f = fan efficiency, dimensionless

$\dot{m}$ = mass-flow rate of air or gas, lb$_m$/s or kg/s

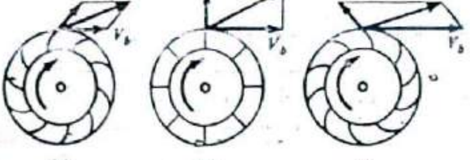


Figure 3-18 Centrifugal blading: (*a*) forward (*b*) flat, and (*c*) backward-curved. Vector diagrams show blade tip velocity V_b, air velocity relative to blade, and V absolute velocity of air leaving blade. V is the same in all cases.

The horsepower may be obtained by dividing $\dot{W}_{sf}$ in ft · lb$_f$/s by 550 or in W by 745.7. The assumption of incompressibility results in errors less than 3 percent for pressure ratios across the fan up to 1.10. Higher pressure ratios would require the use of a P-V relationship of the type $PV^n = C$ (Table 1-3) to evaluate the exact work and power.

Figure 3-19 shows typical characteristics of a centrifugal fan with backward-curved blades. The characteristics usually show pressure and other parameters based on static pressure. *Static pressure* is the force per unit area exerted on a wall by an adjacent fluid that is at rest with respect to the wall (the velocity at the wall in a boundary layer is zero). A fluid in motion, in addition, has a *velocity pressure*. The sum of the static and velocity pressures is called the *total pressure*. Static and velocity pressures are obtained correspondingly from the kinetic and flow energy terms in the general energy equation rewritten for no heat, work, or friction and for incompressible flow (the *Bernoulli equation*). Ignoring the potential energy term

$$Pv + \frac{V_s^2}{2g_c} = \text{constant} \tag{3-10}$$

where each term has units of energy. Multiplying by the density $\rho = 1/v$ gives

$$P + \rho \frac{V_s^2}{2g_c} = \text{constant} \tag{3-11}$$

where each term has units of pressure, P is the static pressure, and $\rho V_s^2/2g_c$ is the velocity (or kinetic) pressure. Dividing Eq. (3-11) by the weight density $\rho g/g_c$ gives

$$h + \frac{V_s^2}{2g} = \text{constant} \tag{3-12}$$

where the terms have the units of length, h here is called the *static head*, $V_s^2/2g$ is called the *velocity head*. The term head, in feet (ft) or meters (m) is often used in fluid technology. Static horsepower is that calculated from Eq. (3-9), where ΔP is the difference between static pressures.

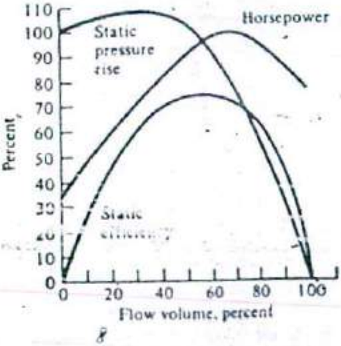


Figure 3-19 Typical constant-speed characteristics of a centrifugal fan with backward-curved blades.

Fan control There are two common methods of controlling the output of fans: *damper control* and *variable-speed* control.

Damper control has the advantage of low capital costs for the damper mechanism itself and for the fan drive motor, which would be a simple constant-speed induction ac motor. It is easily adaptable to automation and allows continuous rather than stepwise control. It suffers, however, from the disadvantage that it adds additional flow resistance that the fan must overcome by increasing its power input. Dampers are usually put on the outlet side of the fan, although inlet dampers, called *inlet vanes*, are sometimes used. They consume less power than outlet dampers but are only effective for moderate load changes near full load. When used with FD fans, they are usually used in combination with outlet dampers. Another combination that results in power savings is the use of a two-speed ac drive motor in conjunction with damper control. Two-speed motors are less expensive than variable-speed ac drives.

Variable-speed control has the advantage of reduced power consumption and is the most efficient method of fan control. The effect of speed on fan performance is that flow, pressure, and power input are directly proportional to N, N^2, and N^3, respectively, where N is the speed of the fan in revolutions per minute (r/min). Thus, reducing speed by say 70 percent reduces the capacity to 70 percent, the pressure to about 50 percent, and the power input to about 35 percent. The actual relationships are dependent upon changes in the effectiveness of the variable-speed drive used. Figure 3-20 shows the effect of speed on pressure and power for a centrifugal fan. The types of drives are (1) variable-speed steam turbine, (2) hydraulic coupling, (3) magnetic coupling, (4) variable-speed dc motor, (5) multiple-speed ac motor, and (6) electronically adjustable motor drive. The main disadvantage of variable-speed control is that all these methods involve higher capital costs than damper control, though the latter is less efficient.

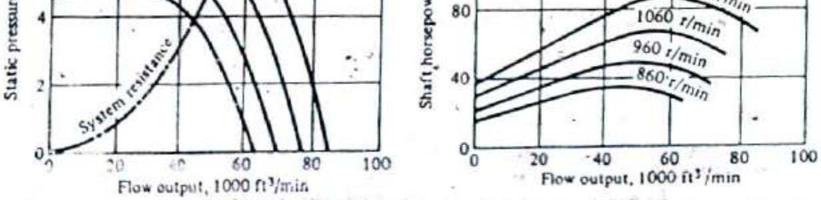


Figure 3-20 Typical centrifugal fan performance curves showing effect of varying speed, r/min [12].

Two other types of fans are used in powerplants: primary-air fans and gas-recirculation fans. *Primary-air fans* supply air to dry and transport pulverized coal to the furnace or to a storage bunker. *Gas-recirculation fans* recirculate gas from a point between the economizer and air preheater back to the bottom of the furnace as part of a steam-temperature control system.

Fans are a major source of noise in powerplants. To reduce that noise, they are often housed in thick masonry acoustical enclosures or equipped with inlet silencers (FD fans), or both [19].

3-12 THE STACK

Tall and conspicuous from a distance, stacks are used in nearly all powerplants. Early steam generators relied solely on stacks to meet the total pressure losses (draft) at the required gas flows. Modern ones have high gas-flow requirements and, because of the various heat exchangers (superheaters, reheaters, economizers, air preheaters), have such high pressure losses that stacks alone are insufficient and fans are added. Stacks have two functions: (1) to assist the fans in overcoming the pressure losses and (2) to help disperse the gas effluent into the atmosphere.

Driving pressure The *driving pressure* Δp_d, in lb_f/ft^2 or N/m^2, supplied by a stack is given by an equation similar to Eq. (3-1)

$$\Delta p_d = (\rho_a - \bar{\rho}_s)H\frac{g}{g_c} \tag{3-13}$$

where ρ_a = atmospheric air density, lb_m/ft^3 or kg/m^3

$\bar{\rho}_s$ = average stack gas density, lb_m/ft^3 or kg/m^3

H = height of the stack, ft or m

Because both air and gas obey the perfect-gas law, Eq. (1-19*a*), from which $\rho = m/V = P/RT$, Eq. (3-13) becomes

$$\Delta p_d = \left(\frac{P_a}{R_a T_a} - \frac{P_s}{R_s \bar{T}_s}\right)H\frac{g}{g_c} \tag{3-14}$$

where P_a and P_s = absolute pressures of atmospheric air (barometric pressure) and stack gas, respectively, lb_f/ft^2 or N/m^2

R_a and R_s = gas constants for air and stack gas, respectively, $ft \cdot lb_f/(lb_m \cdot {}^\circ R)$ or $J/(kg \cdot K)$

T_a and $\bar{T}_s$ = air and average stack gas temperatures respectively, °R or K

Table 3-1 The variation of barometric pressure with altitude

	Altitude, ft									
P_a,	0	1000	2000	3000	4000	5000	6000	7000	8000	10,000
inHg	29.92	28.86	27.82	26.82	25.84	24.90	23.98	23.09	22.22	20.58
psia	14.70	14.17	13.66	13.17	12.69	12.23	11.78	11.34	10.91	10.11
bar	1.013	0.977	0.942	0.908	0.875	0.843	0.812	0.782	0.752	0.697

P_a and P_s differ only slightly. $R_a = 53.34$ ft · lb$_f$/(lb$_m$ · °R). R_s depends upon the gas composition* and hence upon the fuel used and is about 52.2 for bituminous coal, 53.2 for fuel oil, and 55.6 for natural gas. Equation (3-14) can then be rewritten with good accuracy into the form

$$\Delta P_d = \frac{P_a}{R_a}\left(\frac{1}{T_a} - \frac{1}{\bar{T}_s}\right)H\frac{g}{g_c} \tag{3-15}$$

In any one location P_a varies daily according to weather conditions. It is also a function of the altitude of the location. Standard values at different altitudes are given in Table 3-1. T_a, of course, varies seasonally, daily, hourly, etc. $\bar{T}_s$ is dependent upon the temperature variation of the gas along the stack, which is dependent upon the heat losses through the stack walls and any infiltration of cold outside air. The exact value of $\bar{T}_s$ is obtained by integrating the stack local temperature as a function of height and dividing by H, in the same manner as finding the average density in a two-phase channel [Eq. (3-6)]. This function is difficult to evaluate exactly, and arithmetic averaging, implying nearly constant heat loss along the stack, is accepted. Thus

$$\bar{T}_s = \frac{T_o + T_H}{2} \tag{3-16}$$

where T_o and T_H are the stack inlet and exit temperatures, respectively, in °R or K. Values of T_H depend upon T_o, the stack height H and internal diameter D, and the outside atmospheric conditions such as temperature and wind velocity. As expected, T_H increases as T_o and D increase and decreases as H increases.

Stacks, of course, introduce pressure drops of their own. These are caused by wall friction and the pressure equivalent to the kinetic energy of the gases leaving the stack. The latter is usually a few times greater than the former. The total is only a few percent of the driving pressure.

Dispersion The second important function of a stack, *dispersion* of the flue gases into the atmosphere, is defined as the movement of the flue gases horizontally and vertically

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* R for a mixture is equal to the sum of the products of the mass fraction in the mixture times the gas constant for each individual constituent.

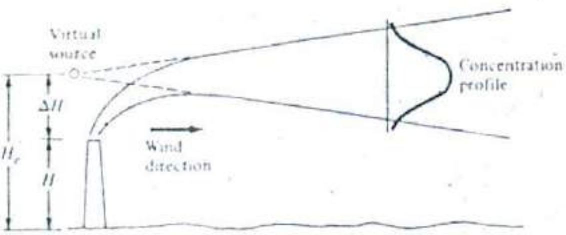

Figure 3-21 Dispersion model forms a stack of height H and plume ΔH.

and their dilution by the atmosphere. The horizontal motion is the result of existing wind. The vertical motion results from the upward motion of high-velocity warm stack-exit gases to much higher elevations. The stack-exit velocity results in a *plume* rise ΔH above the actual stack (Fig. 3-21). The gases bend in the direction of wind flow. According to most models, the *plume height* ΔH is the height of a virtual point source above the stack, obtained by extending the lines of dispersion backward. ΔH is therefore obtained some distance downwind from the stack, where the plume has reached its maximum height. There, thus, is an *effective stack height* H_e given by

$$H_e = H + \Delta H \tag{3-17}$$

In specific situations the proper design of a stack is dependent upon the local topography and airflow patterns. Valleys, for example, can concentrate emissions to unacceptable levels. Model studies are often necessary.

One of the most severe hindrances to dispersion is atmospheric *inversion*, which occurs when the temperature of the atmosphere increases with elevation. It is a condition of little wind and strong stability that results in the reduction of vertical dispersion and therefore the trapping and concentration of local emissions.*

There are several analytical methods of calculating ΔH [20]. Most of them utilize a momentum term that accounts for the vertical momentum of the gas caused by the stack exit velocity and a buoyancy term that accounts for the difference between stack gas and atmospheric densities. They yield different correlations and widely differing values for ΔH. Some of these correlations, giving ΔH in meters, are:

1. Carson and Moses [21]

$$\Delta H = -0.029\frac{V_s D}{V_w} + 2.62\frac{(Q_e)^{0.5}}{V_w} \tag{3-18}$$

where

V_s = stack-gas exit velocity, m/s

D = stack diameter, m

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* There are two kinds of inversion. The first, called *subsidence inversion*, is due to the descent of a layer of air within a high-pressure air mass. The second, called *radiation inversion*, is due to thermal radiation from the earth's surface to the atmosphere above at night.

V_w = wind velocity at stack exit, m/s

Q_e = heat emission, J/s, given by

$$Q_e = \dot{m}c_p(T_s - T_a) \tag{3-19}$$

where

$\dot{m}$ = gas mass-flow rate, kg/s

c_p = specific heat of gas = 1005 J/(kg · K) for dry air at low temperature)

T_s = gas temperature at stack exit, K

T_a = air temperature at stack exit, K.

2. Briggs [22]

$$\Delta H = \frac{114CF^{1/3}}{V_w} \tag{3-20}$$

where

C = dimensionless temperature gradient parameter = $1.58 - 41.4(\Delta\theta/\Delta z)$

$\Delta\theta/\Delta z$ = air potential temperature gradient, K/m = 0 for neutral atmospheric stability conditions*

F = buoyancy flux = $gV_sD^2(T_s - T_a)/4T_a$, m^4/s^3

g = gravitational acceleration = 9.8 m/s^2

3. TVA model [24]

$$\Delta H = \frac{173F^{1/3}}{V_w} e^{-(64\,\Delta\theta/\Delta z)} \tag{3-21}$$

Equation (3-18) is based on many observations and is applicable to all atmospheric stability conditions. Equation (3-20) takes account of the various stability conditions via C and was based on values of $\Delta\theta/\Delta z$ between -0.001 to $+0.013$ K/m. Equation (3-21) was proposed to yield a value of ΔH some 2 km away from the stack where

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* *Stability* is defined as the tendency to resist vertical motion or to suppress turbulence. When the atmosphere is considered to be an adiabatic system, free of frictional or inertial effects, it can be shown [23] that $dT_a/dz = -0.01$ K/m for dry air. The negative of this temperature gradient is called the *adiabatic lapse rate* $\Gamma = 0.01$ K/m. Because the standard atmosphere does not meet the above restrictions, a value, based on meterological data, of $dT_a/dz = -0.0066$ K/m has been adopted as an international standard. The air average temperature in the earth's middle latitudes decreases linearly with elevation in the troposphere from sea level to nearly 10 760 m (35 300 ft), above which it remains constant at 216.7 K (−67°F) in the stratosphere. A *stable atmosphere* is one that does not exhibit much vertical motion or mixing, and hot gases emitted near the earth's surface tend to remain there. A *strongly stable atmosphere* is one where dT_a/dz is positive. This is the case of *inversion* referred to earlier. A *weakly stable atmosphere* is one where dT_a/dz is negative but greater than the environmental standard. A *neutral atmosphere* is one where dT_a/dz is negative and equal to the environmental standard. In neutral conditions, air that is carried rapidly upward

the maximum plume rise was believed to have taken place. It takes into account the stability conditions via $\Delta\theta/\Delta z$, directly built into the equation. There are numerous other correlations for ΔH, some including the horizontal distance from the stack as a parameter. Finally, there are theories based on the assumption that the concentration of dispersed material in the wake is a three-dimensional gaussian distribution around the axis of the wake (Fig. 3-21). A formulation by Cramer [25] is presently accepted for predictions of particulate distributions of the atmosphere.

The nature and environmental effects of pollutants from powerplants are covered in Chap. 17.

3-13 STEAM-GENERATOR CONTROL

Steam-generator, and indeed total, powerplant control is a rather broad subject that includes instrumentation, data processing and controls for combustion, steam flow, temperature and pressure, drum level, burner sequencing, desulfurization precipitators, ash handling, system integration, start-up and shutdown, and automation. It obviously cannot be covered fully in this textbook. In this section, therefore, we will cover, in a simplified manner, only a few basic control systems that apply to the steam generator. These are: feedwater and drum-level control, steam-pressure control, and steam-temperature control.

Feedwater and Drum-Level Control

Feedwater (and therefore steam) flow is controlled to meet load demand by the turbine and at the same time maintain the level of water in the steam drum within relatively narrow limits. It is common to maintain the normal drum level at "half-a-glass,"

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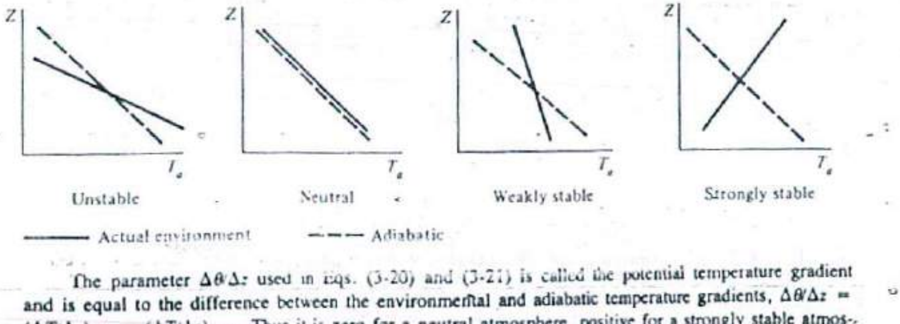
will have the same temperature as the environment and there is no further movement. An *unstable atmosphere* is one where dT_a/dz is negative and less than the environmental standard. It is the case where motion in the vertical direction is enhanced.



The parameter $\Delta\theta/\Delta z$ used in Eqs. (3-20) and (3-21) is called the potential temperature gradient and is equal to the difference between the environmental and adiabatic temperature gradients, $\Delta\theta/\Delta z = (\Delta T/\Delta z)_{env} - (\Delta T/\Delta z)_{adiab}$. Thus it is zero for a neutral atmosphere, positive for a strongly stable atmosphere, and negative for an unstable atmosphere.

meaning half full, in reference to sight glass tubes used outside the drum. A high steam consumption by the turbine, combined with low feedwater supply, for example, would lower the water level in the drum. Figure 3-22 shows a three-element automatic control system, of which drum level is one element.

The drum-level sensor responds to the error between actual drum and its set point, such as in the case of high steam consumption and low feedwater supply, and acts on the controller to increase the feedwater valve opening to meet the steam-flow demand. This action may be too slow and is supplemented by sensors for feedwater and steam flow. The difference between the signals from these two sensors anticipates changes in drum level and sends a signal to the controller to actuate the valve in the proper direction.

Steam-Pressure Control

The steam-pressure control system (Fig. 3-23), sometimes called the "boiler master," maintains steam pressure by adjusting fuel and combustion airflows to meet the desired pressure. When pressure drops, the flows are increased. A steam-pressure sensor acts directly on fuel- and airflow controls, such as the pulverized-coal power drives and the forced-draft fan, to affect the desired changes. A trimming signal from fuel- and airflow sensors maintains the proper fuel-air ratio. Because it is often difficult to obtain accurate fuel flows, a steam-flow sensor is sometimes substituted for the fuel-flow sensor.

Usually about a 5-s delay is allowed when changing coal flow and airflow to ensure the prevention of a momentary rich mixture (high fuel-air ratio) and thus assure smoke-free combustion.

Steam-Temperature Control

Steam-generator outlet temperature control within narrow limits is important to powerplant operation. Steam temperatures may fluctuate because of the buildup of slag or

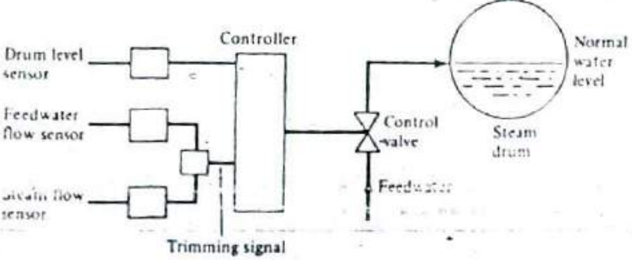


Figure 3-22 Schematic of a three-element feedwater control system.

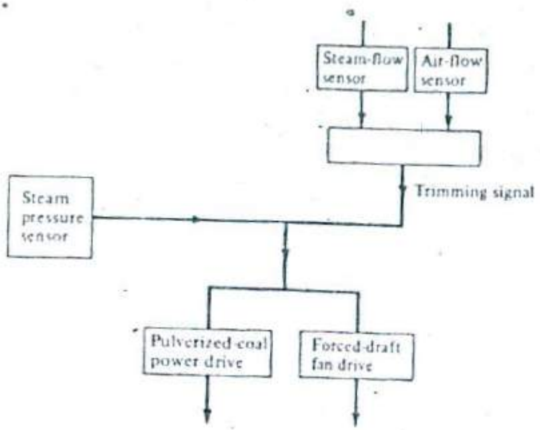


Figure 3-23 Schematic of a steam-pressure control system.

ash on heat-transfer surfaces. The main fluctuations, however, occur because of changes in load. Recall that radiant-type superheaters and reheaters have a drooping temperature-load characteristic, whereas convection-type units have a rising temperature-load characteristic (Fig. 3-10).

A reduction in steam temperature results in loss in plant efficiency. For example, a drop of 35 to 40°F results in about 1 percent increase in heat rate. On the other hand, an increase in steam temperature above design may result in overheating and failure of superheater and reheater tubes and turbine blades.

The temperature of the saturated steam leaving the drum corresponds to the system pressure and thus remains constant as long as the steam-pressure controls are in working order. It is the superheater-reheater responses to load changes that have to be corrected. There are several ways of adjusting the temperature, some of which are discussed below.

Combined radiant-convective superheaters In certain cases radiant and convective superheaters are arranged in series to yield a relatively flat final steam temperature over a wide load range (Fig. 3-10).

Attemperation Attemperation is the reduction of steam temperature by one of two methods. The first uses a *surface attemperator*, which removes heat from the steam in a heat exchanger. One form of the latter, called the *shell type*, has a portion of the steam taken out through tubes from between the primary and secondary superheaters by an automatic valve and diverted to a shell-and-tube heat exchanger containing some of the boiled water. The steam gives up some of its heat to that water and then remixes with the primary steam upon entering the secondary superheater. Temperature control is accomplished by controlling the amount of diverted steam. Another version of the surface attemperator, called the *drum type*, has the heat exchange between the diverted steam and the boiler water occur within the main steam drum, which must now be made larger to accommodate the attemperator tubes.

The second method of attemperation uses a device called the *spray*, or *direct contact, attemperator* (Fig. 3-24). It reduces the steam temperature by spraying low-temperature water from the boiler or economizer exit into the line between the primary and secondary superheaters, Fig. 3-6. The spray nozzle injects water into the throat of a mixing venturi, where the water mixes with high-velocity steam in the throat, vaporizes, and cools the steam. The venturi and a thermal sleeve also protect the main steam pipe from thermal shock caused by any unvaporized water droplets that might otherwise impact on the pipe. The water used must be of high purity to avoid adding deposits on the superheater tubes, pipes, and turbine blades. The spray attemperator has been satisfactory in service. It provides a rapid and sensitive means of temperature control. Steam temperature is controlled by regulating the amount of spray water to produce a flat temperature curve beyond point *a* (Fig. 3-25). A simple energy balance illustrates the principle. For no work, heat, or changes in potential and kinetic energies, the enthalpy entering the system equals that leaving it. Thus

$$\dot{m}_s h_{s1} + \dot{m}_w h_w = (\dot{m}_s + \dot{m}_w) h_{s2} \tag{3-22}$$

where $\dot{m}_s$ and $\dot{m}_w$ = mass-flow rates of steam and water, respectively, lb_m/h or kg/s

h_s and h_w = specific enthalpies of steam and water, respectively, Btu/lb_m or J/kg

subscripts 1 and 2 refer to steam inlet and exit, respectively

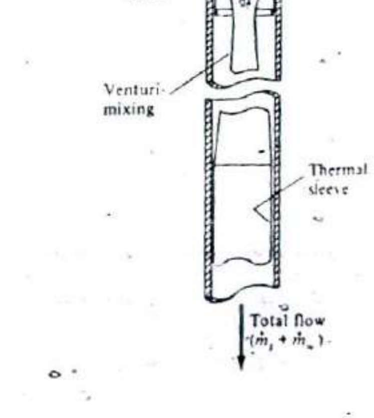


Figure 3-24 A spray attemperator for steam temperature control.

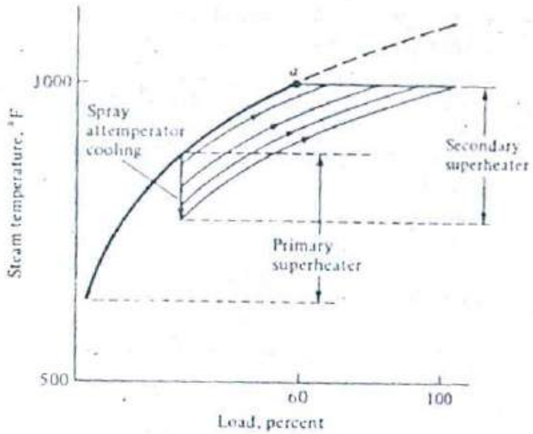


Figure 3-25 Attemperation with oversized primary and secondary superheaters.

Example 3-2 Steam enters a spray attemperator at 2500 psia and 950°F. The spray water comes from the boiler drum, which operates at 2600 psia. Calculate the mass of spray water that must be added per unit mass of steam to reduce its temperature to 900°F.

SOLUTION From the steam tables: $h_{s1} = 1423.1$, $h_{s2} = 1386.7$, $h_w = h_f$ at 2600 psia $= 744.47$, all in Btu/lb$_m$. Using Eq. (3-22)

$$1423.1 + \frac{\dot{m}_w}{\dot{m}_s} \times 744.47 = \left(1 + \frac{\dot{m}_w}{\dot{m}_s}\right) \times 1386.7$$

Therefore

$$\frac{\dot{m}_w}{\dot{m}_s} = 0.0567$$

The attemperator, of either type, may be located before, between, or after the superheater. The first choice will cause condensation of the saturated steam from the boiler before it enters the superheater. In the last choice, the steam temperature exceeds the final desired steam temperature before attemperation. The midlocation between the primary and secondary sections of the superheater is therefore the preferred one.

Attemperation is sometimes used in series with gas recirculation (below), as shown in Fig. 3-26.

Separately fired superheater A superheater with its own burner, fans, combustion chamber, controls, etc., all independent of the steam generator, is sometimes used and may serve more than one steam drum. The rate of firing is adjusted to yield a flat steam temperature-load curve. This system is not generally economical for large electric generating systems and is usually used in the chemical process industry.

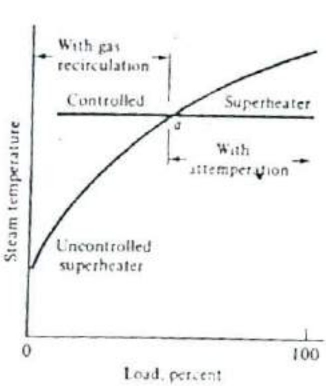


Figure 3-26 Gas recirculation and attemperation in series.

Gas recirculation In this system, gas from some point downstream of the superheater-reheaters, mostly from the economizer outlet but sometimes from the air preheater outlet, is recirculated back to the furnace by means of a gas-recirculation fan (mentioned at the end of Sec. 3-10). The term gas recirculation is restricted to the case where the gas is introduced back to the burning zone, such as before the burners, at the bottom hopper, etc. Gas recirculation to a point downstream of the burning zone is called *gas tempering*. Varying the percentage of recirculated gas alters the heat-absorbing characteristics of the various heat-absorbing surfaces in the steam generators to yield the desired effect and is taken into account in the initial design of the system.

Other types of steam-temperature control are *selecting burners* that give the desired gas temperature, using *tiltable burners* to shift the flame zone in the furnace, *bypassing* a portion of the hot gas around the superheater by dampers, and others.

Regulating reheat outlet steam temperature is necessary for the same reasons as those for regulating superheater outlet temperatures, and the methods used are generally the same.

PROBLEMS

3-1 Calculate the percent energy distribution in a steam generator that goes into steam, losses to flue gases, refuse and heat losses. The steam generator receives 480 short tons of coal per day. The heating value of coal is 13,000 Btu/lb$_m$. 375,000 lb$_m$/h of water enters the steam generator at 2500 psia and 450°F and leaves as steam at 2400 psia and 1000°F. Combustion air enters at 80°F and the flue gases leave at 350°F. The refuse, generated at the rate of 45 short tons per day, has an internal energy of 800 Btu/lb$_m$. The air/fuel ratio by mass is 20:1. Assume that the flue gases have the same variable specific heat as air.

3-2 A steam generator burns a fuel oil that has the following chemical analysis by mass: Carbon 85.3%, hydrogen 14.1%, sulfur 0.5%, and nitrogen 0.1%. Combustion takes place in 125% theoretical air. The flue gases leave the air preheater at 25 psia. Find (*a*) the minimum stack temperature to avoid condensation and (*b*) the maximum mass of sulfurous acid (H_2SO_3) that may be deposited per unit mass of fuel oil burned.

3-3 A West Virginia coal has the following mass analysis: Carbon 75.0%, oxygen 6.7%, hydrogen 5.0%, water 2.5%, sulfur 2.3%, nitrogen 1.5%, and ash 7.0%. It burns in a steam generator with 125 percent theoretical air. Calculate (*a*) the minimum stack temperature to avoid condensation if the flue gas pressure leaving the air preheater is 12 psia and (*b*) the mass of SO_2 produced per unit mass of coal burned.

3-4 A gaseous fuel has the following volumetric analysis: 90% methane (CH_4), 5.0% ethane (C_2H_6), and 5% nitrogen. It burns in a steam generator with 115 percent theoretical air. Calculate the minimum stack temperature if the flue gases leaving the air preheater are at 15 psia.

3-5 A water-tube steam generator operating at 2400 psia has 50-ft-high 3-in ID tubes. The pressure drop is given by $3.25 \times 10^{-6}\,\dot m^2$, where $\dot m$ is the mass flow rate, in pound mass per hour. Assuming saturated water in the downcomer, a water-steam mixture in the tubes with an average density of 18 lb_m/ft^3, and an inlet tube velocity of 3.0 ft/s, calculate the pump pressure necessary for the given flow.

3-6 A drum-type steam generator operates at 160 bars. 1260 kg/s of water goes down the downcomer, 7.32°C subcooled. The downcomer and risers are all 12 m high. The average water-steam mixture density in the risers is 350 kg/m³. The pressure losses in the downcomer and risers total 0.5 bar. Calculate the power, in kilowatts, needed to drive a forced circulation pump if the pump efficiency is 0.70.

3-7 A 40-ft-long 3-in-diameter steam generator water tube receives saturated water at a velocity of 2 feet per second and a pressure of 2400 psia. Heat is added to it uniformly. The slip ratio is 1.8. Calculate the maximum heat added to the tube in Btus per foot if the exit void fraction is not to exceed 0.80.

3-8 Steam enters a convective-type superheater saturated at 160 bars. At a given load it leaves at 480°C. The convective heat transfer coefficients inside h_i and outside h_o of the tubes are proportional to $\dot m_s^{0.8}$ and [illegible] respectively, where $\dot m_s$ and $\dot m_g$ are the mass flow rates of steam and gas. Find the exist gas temperature in degrees centigrade if both steam and gas flow are doubled. The gas temperature T_g remains constant at 2000°C. Assume for simplicity that heat transferred between gas and steam is proportional to the product of an overall heat transfer coefficient U and $(T_g - \bar T_s)$, where $\bar T_s$ is the average steam temperature, and also proportional to $(T_{s2} - T_{s1})$, where T_{s2} and T_{s1} are the exit and inlet steam temperatures. Ignore the conductive resistance in the tube walls and recall from heat transfer that U is inversely proportional to $(1/h_o + 1/h_i)$.

3-9 Steam enters a radiant-type superheater in a dry and saturated condition at 2400 psia. The convective heat-transfer coefficient inside the tubes is proportional to $\dot m^{0.8}$, where $\dot m$ is the steam mass flow rate. The radiative heat transfer between the combustion gases and the tubes outer surface is proportional to $(T_f^4 - T_w^4)$, where T_f and T_w are the flame and tube wall temperatures, respectively. At a given load, the superheater exit steam temperature is 1000°F, and the external heat-transfer coefficient is five times the internal heat-transfer coefficient. Ignoring the conductive resistance in the tube walls and the tube thickness, estimate the temperature of the steam leaving the superheater if the steam mass flow rate is doubled.

3-10 A 24-sector Ljungstrom air preheater contains 1/16-in-thick corrugated steel sheets and rotates at 2 r/min. The hot gases enter at 600°F and leave at 350°F. The air enters at 70°F. For simplicity assume that the gases and air have equal mass flow rates of 100,000 lb_m/h each, and the same specific heat of 0.243 Btu/lb_m·°R. The preheater has an effectiveness of 0.80 (ratio of actual air temperature rise to theoretical maximum temperature rise). Ignore short-circuiting in the preheater and assume perfect heat absorption and release by the steel sheets. Steel has a density of 490 lb_m/ft^3 and a specific heat of 0.1 Btu/lb_m·°R. Calculate (*a*) the air exit temperature and (*b*) the total preheater steel sheet area, in square feet.

3-11 A powerplant receives 92,000 lb_m/h of coal which burns in 125 percent theoretical air. The coal is assumed, for simplicity, to be composed of 100 percent carbon. The air-flue gas pressure losses in the steam generator total 60 in water. Find the power, in megawatts and horsepower, necessary to drive (*a*) a forced-draft fan with air at 60°F and (*b*) an induced-draft fan with flue gases at 300°F if either is used alone. The fans have efficiencies of 0.7. Use c_p for cold air and flue gases of 0.24 and 0.243 Btu/lb_m·°R, respectively. Assume a 2 percent inleakage in the case of the induced-draft fan, and negligible change in specific volume across the fans.

3-12 Find the total power consumed, in megawatts and horsepower, by forced- and induced-draft fans if both are used on a steam generator operating on a balanced draft. Assume that the pressure losses before and after the furnace are 20 and 40 in water, respectively. The steam generator uses 800,000 lb_m/h of fuel oil with air-fuel mass ratio of 18.75. Take leakage into the steam generator at 1 percent of total flow. The outside air is at 40°F and the flue gases are at 325°F. The FD and ID fan efficiencies are 0.8 and 0.7, respectively.

3-13 A centrifugal fan equipped with damper control has the damper wide open at full flow. The damper is partly closed to reduce flow to 80 percent of full load. Estimate the changes, in percent, in static pressure, and the horsepower requirement.

3-14 A centrifugal fan equipped with variable speed control produces a flow of 20,000 ft^3/min at 860 r/min. Its speed is increased at 1160 r/min. Estimate the flow in cubic feet per minute, shaft horsepower and static pressure rise, in H_2O, if its speed is increased to 1160 r/min.

3-15 Combustion gases at 800°F enter a cross-flow-type economizer of a steam generator and leave at 600°F. The steam pressure is 1200 psia. 10^6 lb$_m$/h of feedwater leave the high pressure feedwater heater at 400°F and leave the economizer 7.2°F subcooled. The overall heat transfer coefficient U (which takes into account convection of both gases and water and radiation between the gases and the tubes) is 13.5 Btu/h·ft^2·°F. Calculate the total tube length required if the tubes are 2 in OD.

3-16 A powerplant is situated at an altitude of 1000 ft above sea level. Flue gases enter the stack at 285°F and leave at 230°F. (*a*) Find the height of stack, in feet, necessary for a driving pressure of 0.1 psia if the outside air is at 50°F and (*b*) the change in flue gas flow if the outside air temperature changes to 60°F.

3-17 A powerplant is situated at sea level. 1650 kg/s of flue gases enter a 5-m-diameter stack at 140°C and leave at 110°C. The outside air temperature is at 10°C. The stack is designed for a driving pressure of 0.007 bar. Using the Carson and Moses correlation, calculate the flue gas plume height if the prevailing winds are at 80 km/h.

3-18 A 200-m-high 4-m-diameter stack emits 1000 kg/s of 100°C gases into 5°C air. The prevailing wind velocity is 50 km/h. The atmosphere is in a condition of neutral stability. Calculate the height of the gas plume by two methods.

3-19 A 200-m-high 4-m-diameter stack emits 1000 kg/s of 100°C gases into 5°C air. The prevailing wind velocity is 50 km/h. Using the TVA model, calculate the height of the gas plumes if the atmosphere is (*a*) unstable where $dT/dz = -0.010$ K/m, (*b*) weakly stable where $dT/dz = 0.0050$ K/m, and (*c*) strongly stable where $dT/dz = +0.010$ K/m.

3-20 The steam outlet temperature from a superheater may be given by $t(°F) = 850 + 25L^{1/2}$, where L = load, percent. That temperature should not be allowed to exceed 1000°F. A spray-type attemperator is used to regulate that temperature, using feedwater at 400°F. Calculate (*a*) the percent load at which the attemperator is activated, and (*b*) the pound mass of water injected per pound mass of steam entering the attemperator at half load and at full load. Steam pressure is assumed constant at 1000 psia.

3-21 A spray attemperator receives 125 kg/s of steam at 80 bar and 540°C. It uses spray water from the feedwater line at 200°C. The attemperator exit is at 78 bar and 480°C. How much feedwater is used in kilograms per second?

3-22 A superheat-reheat Rankine cycle has attemperators on both the superheater and reheater which are of the convective type. Flows in the superheater and reheater are 1.1×10^6 and 1.0×10^6 lb$_m$/h, respectively. The maximum pressure is 1000 psia and the reheater is placed optimally in the cycle. At 25 percent load the exit steam from both is 1000°F. As load is increased to 50 percent, these temperatures (without attemperators) would become 1200 and 1150°F. How much water must be added in pounds mass per hour in the attemperators to maintain 1000°F exit temperatures in both if water is available at 400°F?

CHAPTER

FOUR

FUELS AND COMBUSTION

4-1 INTRODUCTION

In Chap. 3 we covered those components of fossil-fueled steam generators that dealt with the working fluid (water and steam) and with the air and flue gases. We deferred discussing the fuel aspects because they require independent treatment. There is a rather wide variety of fuels. Their preparation and feeding, often outside the steam generators, and their methods of firing deserve special attention.

The increasing worldwide demand for energy has focused attention on fuels, their availability (Table 4-1), and environmental effects. The fuels available to utility industry are largely nuclear and fossil, both essentially nonrenewable. Nuclear fuels originated with the universe, and it takes nature millions of years to manufacture fossil fuels.

Fossil fuels originate from the earth as a result of the slow decomposition and chemical conversion of organic material. They come in three basic forms: *solid* (coal), *liquid* (oil), and *natural gas*. Coal represents the largest fossil-fuel energy resource in the world. In the United States today (1983), it is responsible for about 50 percent of electric-power generation. Oil and natural gas are responsible for another 30 percent. The remaining percentage is mostly due to nuclear and hydraulic generation. Natural gas, however, is being phased out of the picture in the United States because it must be conserved for essential industrial and domestic uses.

New combustible-fuel options include the so-called synthetic fuels, or synfuels, which are liquids and gases derived largely from coal, oil shale, and tar sands. A tiny fraction of fuels used today are industrial by-products, industrial and domestic wastes, and biomass.

Table 4-1 World energy reserves

Fuel	Type	Energy, Q*
Fossil	Coal	32
	Oil and gas	6
Fissile	Uranium and thorium	600
Fusile	Deuterium	10^{10}

* 1 Q = 10^{18} Btu or roughly 10^{21} J, not to be confused with a quad q = 10^{15} Btu

This chapter will cover the combustible fuels available to the utility industry, both natural (fossil) and synthetic, and their preparation and firing systems. Nuclear fuels and renewable energy sources, and the environmental aspects of power generation in general, will be covered later in this text.

4-2 COAL

Coal is a general term that encompasses a large number of solid organic minerals with widely differing compositions and properties, although all are essentially rich in amorphous (without regular structure) elemental carbon. It is found in stratified deposits at different and often great depths, although sometimes near the surface. It is estimated that in the United States there are 270,000 million tons of recoverable reserves (those that can be mined economically within the foreseeable future) in 36 of the 50 states. This accounts for about 30 percent of the world's total.

There are many ways of classifying coal according to its chemical and physical properties. The most accepted system is the one used by the American Society for Testing and Materials (ASTM), which classifies coals by grade or rank according to the degree of metamorphism (change in form and structure under the influences of heat, pressure, and water), ranging from the lowest state, lignite, to the highest, anthracite (ASTM D 388). These classifications are briefly described below in a descending order.

Anthracite This is the highest grade of coal. It cotains a high content, 86 to 98 mass percent, of fixed carbon (the carbon content in the elemental state) on a dry, mineral-matter-free basis and a low content of volatile matter, less than 2 to 14 mass percent (chiefly methane, CH_4). Anthracite is a shiny black, dense, hard, brittle coal that borders on graphite* at the upper end of fixed carbon. It is slow-burning and has a heating value just below that of the highest for bituminous coal (see below). Its use

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* Graphite is a moderately soft allotropic form of carbon. Carbon crystallizes perfectly into diamond, imperfectly into graphite, and is amorphous (having no regular structure, noncrystalline) in anthracite and charcoal.

in steam generators is largely confined to burning on stokers, and rarely in pulverized form. In the United States it is mostly found in Pennsylvania.

The anthracite rank of coal is subdivided into three groups. In descending order of fixed-carbon percent, they are *meta-anthracite*, greater than 98 percent; *anthracite*, 92 to 98 percent; and *semianthracite*, 86 to 92 percent.

Bituminous coal The largest group, bituminous coal is a broad class of coals containing 46 to 86 mass percent of fixed carbon and 20 to 40 percent of volatile matter of more complex content than that found in anthracite. It derives its name from *bitumen*, an asphaltic residue obtained in the distillation of some fuels. Bituminous coals range in heating value from 11,000 to more than 14,000 Btu/lb$_m$ (about 25,600 to 32,600 kJ/kg). Bituminous coals usually burn easily, especially in pulverized form.

The bituminous rank is subdivided into five groups: *low-volatile*, *medium-volatile*, and *high-volatile A*, *B*, and *C*. The lower the volatility, the higher the heating value. The low-volatility group is grayish black and granular in structure, while the high-volatility groups are homogeneous or laminar.

Subbituminous coal This is a class of coal with generally lower heating values than bituminous coal, between 8300 to 11,500 Btu/lb$_m$ (about 19,300 to 26,750 kJ/kg). It is relatively high in inherent moisture content, as much as 15 to 30 percent, but often low in sulfur content. It is brownish black or black and mostly homogeneous in structure. Subbituminous coals are usually burned in pulverized form. The subbituminous rank is divided into three groups: *A*, *B*, and *C*.

Lignite The lowest grade of coal, lignite derives its name from the Latin *lignum*, which means "wood." It is brown and laminar in structure, and remnants of wood fiber are often visible in it. It originates mostly from resin-rich plants and is therefore high in both inherent moisture, as high as 30 percent, and volatile matter. Its heating value ranges between less than 6300 to 8300 Btu/lb$_m$ (about 14,650 to 19,300 kJ/kg). Because of the high moisture content and low heating value, lignite it is not economical to transport over long distances and it is usually burned by utilities at the mine site. The lignite rank is subdivided into two groups: *A* and *B*.

Peat Peat is not an ASTM rank of coal. It is, however, considered the first geological step in coal's formation. Peat is a heterogeneous material consisting of decomposed plant matter and inorganic minerals. It contains up to 90 percent moisture. Although not attractive as a utility fuel, it is abundant in many parts of the world. Several states in the United States have large deposits. Because of its abundance, it is used in a few countries (Ireland, Finland, the USSR) in some electric generating plants and in district heating.

4-3 COAL ANALYSIS

There are two types of coal analysis: *proximate* and *ultimate*, both done on a mass-percent basis. Both of these methods may be based on: an *as-received* basis, useful for combustion calculations; a *moisture-free* basis, which avoids variations of the moisture content even in the same shipment and certainly in the different stages of pulverization; and a *dry mineral–matter-free* basis, which circumvents the problem of the ash content's not being the same as the mineral matter in the coal.

Proximate Analysis

This is the easier of two types of coal analysis and the one which supplies readily meaningful information for coal's use in steam generators. The basic method for proximate analysis is given by ANSI/ASTM* Standards D 3172. It determines the mass percentages of fixed carbon, volatile matter, moisture, and ash. Sulfur is obtained in a separate determination.

Fixed carbon is the elemental carbon that exists in coal. In proximate analysis, its determination is approximated by assuming it to be the difference between the original sample and the sum of volatile matter, moisture, and ash.

The *volatile matter* is that portion of coal, other than water vapor, which is driven off when the sample is heated in the absence of oxygen in a standard test (up to 1750°F for 7 min). It consists of hydrocarbon and other gases that result from distillation and decomposition.

Moisture is determined by a standard procedure of drying in an oven. This does not account for all the water present, which includes combined water and water of hydration. There are several other terms for moisture in coal. One, *inherent moisture*, is that existing in the natural state of coal and considered to be part of the deposit, excluding surface water.

Ash is the inorganic salts contained in coal. It is determined in practice as the noncombustible residue after the combustion of dried coal in a standard test (at 1380°F).

Sulfur is determined separately in a standard test, given by ANSI/ASTM Standards D 2492. Being combustible, it contributes to the heating value of the coal. It forms oxides which combine with water to form acids. These cause corrosion problems in the back end of steam generators if the gases are cooled below the dew point, as well as environmental problems (Chap. 17).

Ultimate Analysis

A more scientific test than proximate analysis, ultimate analysis gives the mass percentages of the chemical elements that constitute the coal. These include carbon, hydrogen, nitrogen, oxygen, and sulfur. Ash is determined as a whole, sometimes in a separate analysis. Ultimate analysis is given by ASTM Standards D 3176.

Heating Value

The heating value, Btu/lb_m or J/kg of fuel, may be determined on as-received, dry, or dry-and-ash-free basis. It is the heat transferred when the products of complete

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* American National Standards Institute/American Society for Testing and Materials.

combustion of a sample of coal or other fuel are cooled to the initial temperature of air and fuel. It is determined in a standard test in a *bomb calorimeter* given by ASTM Standards D 2015. There are two determinations: the *higher* (or *gross*) *heating value* (HHV) assumes that the water vapor in the products condenses and thus includes the latent heat of vaporization of the water vapor in the products; the *lower heating value* (LHV) does not. The difference between the two is given by

$$\text{LHV} = \text{HHV} - m_w h_{fg} \tag{4-1a}$$

or

$$\text{LHV} = \text{HHV} - 9m_{H_2} h_{fg} \tag{4-1b}$$

where m_w = mass of water vapor in products of combustion per unit mass of fuel (due to the combustion of H_2 in the fuel, i.e., not including initial H_2O in fuel)

m_{H_2} = mass of original hydrogen per unit mass of fuel, known from ultimate analysis

h_{fg} = latent heat of vaporization of water vapor at its partial pressure in the combustion products, Btu/lb$_m$ H_2O or J/kg H_2O

The partial pressure of water vapor in the products of combustion is obtained by multiplying the mole faction of H_2O in the products, which is obtained from the combustion equation in the usual manner, by the total pressure of the products. The 9 in Eq. (4-1*b*) is the ratio of the molecular masses of H_2O and H_2 and represents the mass of H_2O vapor obtained from a unit mass of H_2.

Because gases are not usually cooled down below the dew point in steam generators (or engines), it does not seem fair to charge them with the higher heating value in calculating energy balances and efficiencies of cycles or engines. Some, however, argue that they should be charged with the total energy content of the fuel. A uniform standard had to be agreed upon, whereupon everybody uses the HHV in energy balances and efficiency calculations. (The LHV is the standard used in European practice, however.)

As indicated above, heating values are obtained by testing. However, a formula of the *Dulong* type (which does not include the effects of dissociation) is used to give approximate higher heating values of anthracite and bituminous coals in Btu/lb$_m$.

$$\text{HHV} = 14{,}600\text{C} + 62{,}000\left(\text{H} - \frac{\text{O}}{8}\right) + 4050\text{S} \tag{4-2}$$

where C, H, O, and S are the mass fractions of carbon, hydrogen, oxygen, and sulfur, respectively, in the coal. For lower-rank fuels, the above formula usually underestimates the HHV.

Table 4-2 gives the proximate and ultimate analyses of some typical U.S. coals.

Calculations of heating value for fuels of known composition will be covered in Sec. 4-14.

Table 4-2 Proximate and ultimate analyses of some U.S. coals

Analysis, mass percent	Anthracite	Bituminous, medium volatility	Subbituminous	Lignite
		Proximate		
Fixed carbon	83.8	70.0	45.9	30.8
Volatile matter	5.7	20.5	30.5	28.2
Moisture	2.5	3.3	19.6	34.8
Ash	8.0	6.2	4.0	6.2
		Ultimate		
C	83.9	80.7	58.8	42.4
H_2	2.9	4.5	3.8	2.8
S	0.7	1.8	0.3	0.7
O_2	0.7	2.4	12.2	12.4
N_2	1.3	1.1	1.3	0.7
H_2O	2.5	3.3	19.6	34.8
		HHV		
Btu/lb_m	13,720	14,310	10,130	7,210

Example 4-1 Write the complete combustion equation, calculate the HHV and LHV, Btu/lb_m, of the medium volatility subbituminous coal in Table 4-2 by using the Dulong-type formula and find the dew point, °F. Assume stoichiometric, i.e. chemically correct, combustion air. Total pressure = 1 atm.

SOLUTION

$$\text{HHV} = 14{,}600 \times 0.807 + 62{,}000\left(0.045 - \frac{0.024}{8}\right) + 4050 \times 0.018$$

$$= 11{,}782 + 2976 + 73 = 14{,}831 \text{ Btu/lb}_m$$

Note that the actual value, 14,310, is slightly lower by about 3.5 percent because dissociation and other effects are not taken into account by the Dulong-type formula.

The relative mole fractions of the fuel constituents from the ultimate analysis are obtained by dividing the mass fractions by their molecular masses. The equivalent molecule of the coal, therefore, is

$$\frac{80.7}{12}\text{C} + \frac{4.5}{2}\text{H}_2 + \frac{1.8}{32}\text{S} + \frac{2.4}{32}\text{O}_2 + \frac{1.1}{28}\text{N}_2 + \frac{3.3}{18}\text{H}_2\text{O}$$

or $0.725\text{C} + 2.250\text{H}_2 + 0.05625\text{S} + 0.075\text{O}_2 + 0.03929\text{N}_2 + 0.1833\text{H}_2\text{O}$

A hydrogen balance gives 2.250H_2O from the H_2 in the fuel + 0.1833H_2O originally in the fuel = 2.4333H_2O in the products.

A sulfur balance gives 0.05625SO_2 in the products.

An oxygen balance thus requires $0.725 + (2.250/2) + 0.05625 = 1.90625O_2$ for combustion, but only $1.90625 - 0.075$ originally in the fuel $= 1.83125O_2$ from the atmosphere.

A nitrogen balance (the atmosphere contains 3.76 mol N_2 per mol O_2) gives $3.76 \times 1.83125 = 6.8855N_2$ from the atmosphere $+ 0.03929$ originally in the fuel $= 6.92479N_2$ in the products.

The complete combustion equation, therefore, is

$$(0.725C + 2.250H_2 + 0.05625S + 0.075O_2 + 0.03929N_2 + 0.1833H_2O) + 1.83125O_2 + 6.8855N_2 \rightarrow 0.725CO_2 + 2.4333H_2O + 0.05625SO_2 + 6.92479N_2$$

The latent heat of vaporization, necessary for the calculation of the LHV, is due to the partial pressure of the H_2O that was formed in the combustion process only, because the moisture originally in the fuel receives and gives off the same heat upon vaporization and condensation. The partial pressure of the combustion H_2O is equal to its mole fraction in the products

$$\frac{2.250}{0.725 + 2.4333 + 0.05625 + 6.92479} \times 14.696 = 3.705 \text{ psia}$$

From the steam tables, h_{fg} at 3.705 psia (by interpolation) is 1008.6 Btu/lb$_m$. Using Eq. (4-1*b*)

$$\text{LHV} = 14{,}831 - 9 \times 0.045 \times 1008.6 = 14{,}422.5 \text{ Btu/lb}_m$$

Note that the differences between HHV and LHV for coals are small (less than 3 percent in the above example) because of their low H_2 content. This is not the case with liquid or gaseous hydrocarbons, which contain a large portion of hydrogen in the C-H molecules.

The dew point of the products is the saturation temperature corresponding to the partial pressure of the total H_2O in the products, or

$$\frac{2.4333}{0.725 + 2.4333 + 0.05625 + 6.92479} \times 14.696 = 3.5268 \text{ psia}$$

This corresponds to a saturation temperature of about 148°F, which is the dew point.

There will be more discussion of heating value in Sec. 4-14.

4-4 COAL FIRING

Since the old days of feeding coal into a furnace by hand, several major advances have been made that permit increasingly higher rates of combustion.

The earliest in the history of steam boilers were *mechanical stokers*, and several types are still being used for small- and medium-sized boilers. All such stokers are designed to continuously feed coal into the furnace by moving it on a grate within the furnace and also to remove ash from the furnace.

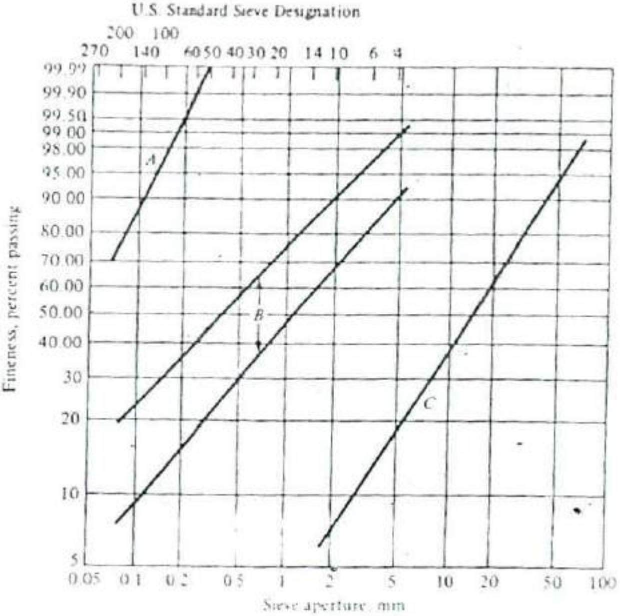


Figure 4-1 Coal sieve analysis. (*A*) pulverized-coal sample; (*B*) coal range for cyclone firing; (*C*) coal as fired. Plot based on graphical system proposed in Ref. 26.

Pulverized-coal firing was introduced in the 1920s and represented a major increase in combustion rates over mechanical stokers. It is widely used today. To prepare the coal for use in pulverized firing, it is crushed and then ground to such a fine powder that approximately 70 percent of it will pass a 200-mesh sieve* (Fig. 4-1). It is suitable for a wide variety of coal, particularly the higher-grade ones. Advantages of pulverized-coal firing are the ability to use any size coal; good variable-load response; a lower requirement for excess air for combustion, resulting in lower fan power consumption; lower carbon loss; higher combustion temperatures and improved thermal efficiency; lower operation and maintenance costs; and the possibility of design for multiple-fuel combustion (oil, gas, and coal).

* There are some seven screen, or sieve, standards in the United States and Europe. The one used here is the U.S. Standard Sieve, in which the number of openings per linear inch designates the mesh. A 100-mesh screen has 100 openings to the inch, or 10,000 openings per square inch. The higher the mesh, the finer the screen. The diameter of the wire determines the opening size. The U.S. Standard Sieve mesh and opening size in inches and millimeters are given below.

Mesh	20	30	40	50	60	100	140	200	325	400
in	1.0331	1.0234	0.0165	0.0117	0.0098	0.0059	0.0041	0.0029	0.0017	0.0015
mm	0.840	0.595	0.420	0.297	0.250	0.149	0.105	0.074	0.044	0.037

In the late 1930s *cyclone-furnace firing* was introduced and became the third major advance in coal firing. It is now also widely used though for a lesser variety of uses than is pulverized coal. In addition to those advantages already mentioned for pulverized-coal firing, cyclone firing provides several other advantages. These are the obvious savings in pulverizing equipment because coal need only be crushed, reduction in furnace size, and reduction in fly ash content of the flue gases. Coal size for cyclone-furnace firing is accomplished in a simple crusher and covers a wide band, with approximately 95 percent of it passing a 4-mesh sieve (Fig. 4-1).

Most recently, *fluidized-bed combustion* has been introduced. In this type of firing, crushed particles of coal are injected into the fluidized bed so that they spread across an air distribution grid. The combustion air, blown through the grid, has an upward velocity sufficient to cause the coal particles to become fluidized, i.e., held in suspension as they burn. Unburned carbon leaving the bed is collected in a cyclone separator and returned back to the bed for another go at combustion. The main advantage of fluidized-bed combustion is the ability to desulfurize the fuel during combustion in order to meet air quality standards for sulfur dioxide emissions. (Other methods are the use of low-sulfur coal, desulfurization of coal before it is burned, and removal of SO_2 from the flue gases by the use of scrubbers, Chap. 17.) Desulfurization is accomplished by the addition of limestone directly to the bed. Fluidized-bed combustion is still undergoing development and has other attractive features (Sec. 4-8).

4-5 MECHANICAL STOKERS

Almost all kinds of coal can be fired on stokers. Stoker firing, however, is the least efficient of all types of firing except hand firing. Partly because of the low efficiency, stoker firing is limited to relatively low capacities, usually for boilers producing less than 400,000 lb_m/h (50 kg/s) of steam, though designers are limiting stoker use to around 100,000 lb_m/h (12.6 kg/s). These capacities are the result of the practical limitations of stoker physical sizes and relatively low burning rates which require a large furnace width for a given steam output. Pulverized and cyclone firing, on the other hand, have higher burning rates and are flexible enough in design to meet the millions of pounds per hour of steam requirements of modern steam generators with narrower and higher furnaces. Stokers, however, remain an important part of steam-generator systems in their size range.

Mechanical stokers are usually classified into four major groups, depending upon the method of introducing the coal into the furnace. These are *spreader stokers, underfed stokers, vibrating-grate stokers*, and *traveling-grate stokers*.

The *spreader stoker* is the most widely used for steam capacities of 75,000 to 400,000 lb_m/h (9.5 to 50 kg/s). It can burn a wide variety of coals from high-rank bituminous to lignite and even some by-product waste fuels such as wood wastes, pulpwood, bark, and others and is responsive to rapid load changes. In the spreader stoker coal is fed from a hopper to a number of feeder-distributor units, each of which has a reciprocating feed plate that transports the coal from the hopper over an adjustable spill plate to an overthrow rotor equipped with curved blades. There are a number of such feeder-distributor mechanisms that inject the coal into the furnace in a wide uniform projectile over the stoker grate (Fig. 4-2). Air is primarily fed upward through the grate from an air plenum below it. This is called *undergrate air*. The finer coal particles, between 25 and 50 percent of the injected coal, are supported by the upward airflow and are burned while in suspension. The larger ones fall to the grate and burn in a relatively thin layer. Some air, called *overfire air*, is blown into the furnace just above the coal projectile. Forced-draft fans are used for both undergrate and overfire air. The unit has equipment for collecting and reinjecting dust and controls for coal flow and airflow to suit load demand on the steam generator.

The problem with stationary spreader stokers was the removal of ash, which was first done manually and then by shutting off individual sections of grates and their air supply for ash removal without affecting other sections of the stoker. The spreader stoker became widely accepted only after the introduction of the *continuous-ash-discharge traveling-grate* stoker in the late 1930s. Traveling-grate stokers, as a class, also include the so-called *chain-grate* stoker. They have grates, links, or keys joined in an endless belt that is driven by a motorized sprocket drive at one end and over an idle shaft sprocket mechanism at the other. Coal may be injected in the above manner or fed directly from a hopper onto the moving grate through an adjustable gate that regulates the thickness of the coal layer. Ash is discharged into an ash pit at either end, depending upon the direction of motion of the traveling grate.

Continuous-cleaning grates that use reciprocating or vibrating designs have also been developed, as have underfeed stokers that are suitable for burning special types

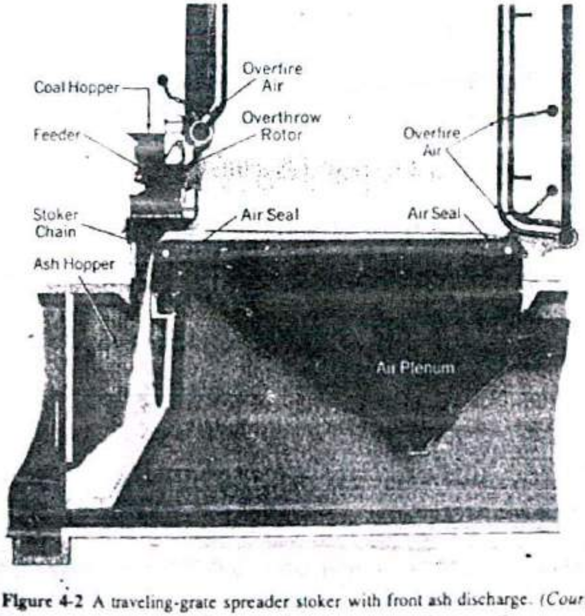


Figure 4-2 A traveling-grate spreader stoker with front ash discharge. *(Courtesy Babcock and Wilcox.)*

of coals. The continuous-ash-removal traveling-grate stoker, however, has high burning rates and remains the preferred type of stoker.

Ignition of the fresh coal in stokers, as well as its combustible volatile matter, driven off by distillation, is started by radiation heat transfer from the burning gases above. The fuel bed continues to burn and grows thinner as the stoker travels to the far end over the bend, where ash is discharged to the ash pit. Arches are sometimes built into the furnace to improve combustion by reflecting heat onto the coal bed.

4-6 PULVERIZED-COAL FIRING

The commercial development of methods for firing coal in pulverized form is a landmark in the history of steam generation. It made possible the construction of large, efficient, and reliable steam generators and powerplants. The concept of firing "powdered" coal, as it was called in earlier times, dates back to Carnot [7], whose idea envisaged its use for the Carnot cycle; to Diesel, who used it in his first experiments on the engine that now bears his name [27]; to Thomas Edison, who improved its firing in cement kilns, thus improving their efficiency and production; and to many others. It was not, however, until the pioneering efforts of John Anderson and his associates and the forerunner of the present Wisconsin Electric Power Company that pulverized coal was used successfully in electric generating powerplants at their Oneida Street and Lakeside Stations, Milwaukee, Wisconsin [28].

The impetus for the early work on coal pulverization stemmed from the belief that, if coal were made fine enough, it would burn as easily and efficiently as a gas. Further inducements came from an increase in oil prices and the wide availability of coal, which makes the present situation sound rather like history repeating itself. Much theoretical work on the mechanism of pulverized-coal combustion began in the early 1920s [29,30]. The mechanism of crushing and pulverizing has not been well understood theoretically and remains a matter of controversy even today. Probably the most accepted law is one published in 1867 in Germany, called *Rittinger's law*, that states that the work needed to reduce a material of a given size to a smaller size is proportional to the surface area of the reduced size. This, and other laws, however, do not take into account many of the processes involved in coal pulverization, and much of the progress in developing pulverized-coal furnaces relies heavily on empirical correlations and designs.

To burn pulverized coal successfully in a furnace, two requirements must be met: (1) the existence of large quantities of very fine particles of coal, usually those that would pass a 200-mesh screen, to ensure ready ignition because of their large surface-to-volume ratios and (2) the existence of a minimum quantity of coarser particles to ensure high combustion efficiency. These larger coarse particles should contain a very small amount larger than a given size, usually that which would be retained on a 50-mesh screen, because they cause slagging and loss of combustion efficiency. Line *A* in Fig. 4-1 represents a typical range for pulverized coal. It shows about 80 percent of the coal passing a 200-mesh screen that corresponds to a 0.074-mm opening and about 99.99 percent passing a 50-mesh screen that corresponds to a 0.297-mm opening, i.e., only 0.1 percent larger than 0.297 mm.

The size of bituminous coal that is shipped as it comes from the mine, called *run-of-mine coal*, is about 8 in. Oversized lumps are broken up but the coal is not screened. Other sizes are given names like *lump* (5 in), which is used in hand firing and domestic applications, *egg* (5 × 2 in), *nut* (2 × 1¼ in), *stoker* (1¼ × ¾ in), and *slack* (¾ × 0 in, meaning ¾ in or less). [Anthracite coal has similar designations, ranging from *broken* (4⅜ × 3 in) to *buckwheat* ($\frac{9}{16}$ × $\frac{5}{16}$ in) and *rice* ($\frac{5}{16}$ × $\frac{3}{16}$ in), ASTM D 310].

Coal is usually delivered to a plant site already sized to meet the feed size required by the pulverizing mill or the cyclone furnace (Sec. 4-7). If the coal is too large, however, it must go through *crushers*, which are part of the plant coal-handling system and are usually located in a crusher house at a convenient transfer point in the coal-conveyor system. The feed size required in pulverizing mills is designated at 1¼ × 0 in; that required for cyclone furnaces is ¼ × 0 in.

Crushers

Although there are several types of commercially available coal crushers, a few stand out for particular uses. To prepare coal for pulverization, the *ring crusher*, or *granulator*, (Fig. 4-3) and the *hammermill* (Fig. 4-4) are preferred. The coal is fed at the top and is crushed by the action of rings that pivot off center on a rotor or by swinging hammers attached to it. Adjustable screen bars determine the maximum size of the discharged coal. Wood and other foreign material is also crushed, but a trap is usually provided to collect tramp iron (metal and other hard-to-crush matter.) Ring crushers and hammermills are used off or on plant site. They reduce run-of-mine coals down to sizes such as ¾ × 0 in. Thus they discharge a large amount of fines suitable for

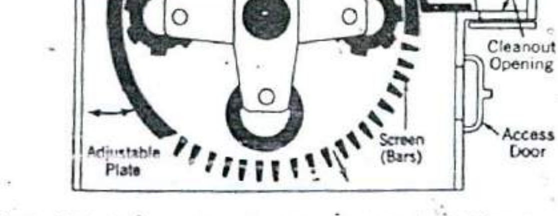


Figure 4-3 A ring-type coal crusher. *(Courtesy Babcock and Wilcox.)*

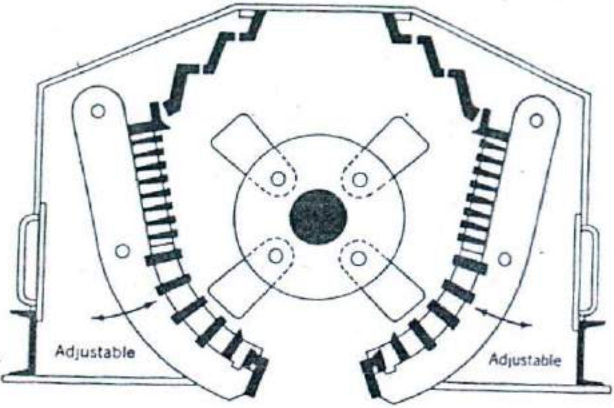


Figure 4-4 A hammer-mill coal crusher. *(Courtesy Babcock and Wilcox.)*

further pulverization, but not for cyclone-furnace firing. For the latter, a crusher type called the *reversible hammermill* is preferred.

A third type, the *Bradford breaker* (Fig. 4-5), is used for large-capacity work. It is composed of a large cylinder consisting of perforated steel or screen plates to which lifting shelves are attached on the inside. The cylinder rotating slowly at about 20 r/min receives feed at one end. The shelves lift the coal, and the breaking action is accomplished by the repeated dropping of the coal until its size permits it to be discharged through the perforations, whose size determines the size of the discharged coal. The quantity of fines is limited because the crushing force, due to gravity, is not large. Bradford breakers easily reject foreign matter and produce relatively uniform size coals. They are usually used at the mine but may also be used at the plant.

Other simple devices called *roll crushers*, which have single or double rolls or rotors equipped with teeth, have been used but have not proven very satisfactory because of their inability to produce coal of uniform size.

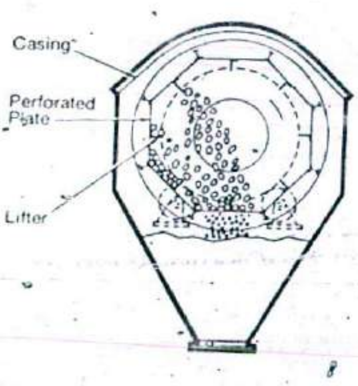


Figure 4-5 A Bradford breaker. *(Courtesy Combustion Engineering, Inc.)*

Pulverizers

The pulverizing process is composed of several stages. The first is the *feeding* system, which must automatically control the fuel-feed rate according to the boiler demand and the air rates required for drying (below) and transporting pulverized fuel to the burner (primary air). The next stage is *drying*. One important property of coal being prepared for pulverization is that it be dry and dusty. Because coals have varying quantities of moisture and in order that lower-rank coals can be used, *dryers* are an integral part of pulverizing equipment. Part of the air from the steam-generator air preheater, the primary air, is forced into the pulverizer at 650°F or more by the primary-air fan. There it is mixed with the coal as it is being circulated and ground.

The heart of the equipment is the *pulverizer*, also called grinding mill. Grinding is accomplished by impact, attrition, crushing, or combinations of these. There are several commonly used pulverizers, classified by speed: (1) low-speed (below 75 r/min): the ball-tube mill; (2) medium-speed (75 to 225 r/min): the ball-and-race and roll-and-race mill; and (3) high-speed (above 225 r/min): the impact or hammermill, and the attrition mill.

The low-speed *ball-tube mill*, one of the oldest on the market, is basically a hollow cylinder with conical ends and heavy-cast wear-resistant liners, less than half-filled with forged steel balls of mixed size. Pulverization is accomplished by attrition and impact as the balls and coal ascend and fall with cylinder rotation. Primary air is circulated over the charge to carry the pulverized coal to classifiers (below). The balltube mill is dependable and requires low maintenance, but it is larger and heavier in construction, consumes more power than others, and because of poor air circulation, works less efficiently with wet coals. It has now been replaced with more efficient types.

The medium-speed *ball-and-race* and *roll-and race* pulverizers are the type in most use nowadays. They operate on the principles of crushing and attrition. Pulverization takes place between two surfaces, one rolling on top of the other. The rolling elements may be balls or ring-shaped rolls that roll between two races, in the manner of a ball bearing. Figure 4-6 shows an example of the former. The balls are between a top stationary race or ring and a rotating bottom ring, which is driven by the vertical shaft of the pulverizer. Primary air causes coal feed to circulate between the grinding elements, and when it becomes fine enough, it becomes suspended in the air and is carried to the classifier. Grinding pressure is varied for the most efficient grinding of various coals by externally adjustable springs on top of the stationary ring. The ball-and-race pulverizer has ball circle diameters varying between 17 and 76 in and capacities between 1½ to 20 tons/h. The roll-and-race pulverizer is operated at lower speeds and larger sizes. A typical one has an 89-in ball circle diameter, a 12-ft diameter, and a 22.5-ft height overall, weighs 150 tons, and is driven by a 700-hp motor [12]. Both types are suitable for direct-firing systems (see below).

High-speed pulverizers use *hammer beaters* that revolve in a chamber equipped with high-wear-resistant liners. They are mostly used with low-rank coals with high-moisture content and use flue gas for drying. They are not widely used for pulverized coal systems.

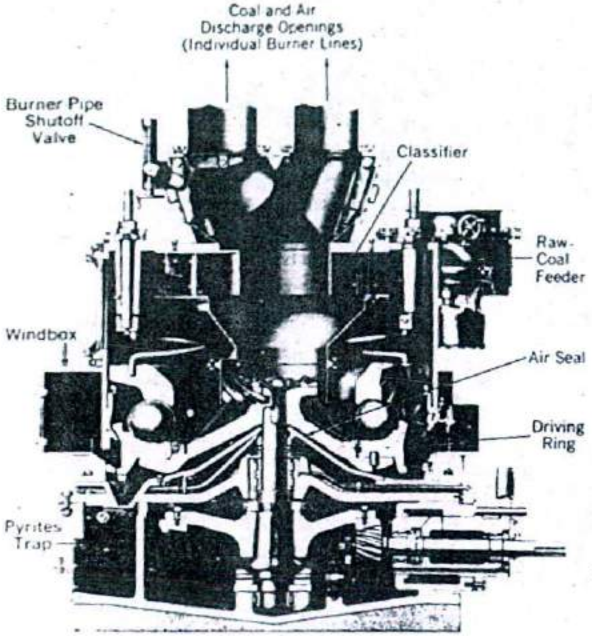


Figure 4-6 A single-row ball-and-race coal pulverizer. *(Courtesy Babcock and Wilcox.)*

The *classifier* referred to above is located at the pulverizer exit. It is usually a cyclone with adjustable inlet vanes. The classifier separates oversized coal and returns it to the grinders to maintain the proper fineness for the particular application and coal used. Adjustment is obtained by varying the gas-suspension velocity in the classifier by adjusting the inlet vanes.

The Pulverized-Coal System

A total pulverized-coal system comprises pulverizing, delivery, and burning equipment. It must be capable of both continuous operation and rapid change as required by load demands. There are two main systems: the bin or storage system and the direct-firing system.

The *bin system* is essentially a batch system by which the pulverized coal is prepared away from the furnace and the resulting pulverized-coal–primary-air mixture goes to a cyclone separator and fabric bag filter that separate and exhaust the moisture-laden air to the atmosphere and discharge the pulverized coal to storage bins (Fig. 4-7). From there, the coal is pneumatically conveyed through pipelines to utilization bins near the furnace for use as required. The bin system was widely used before pulverizing equipment became reliable enough for continuous steady operation. Be-

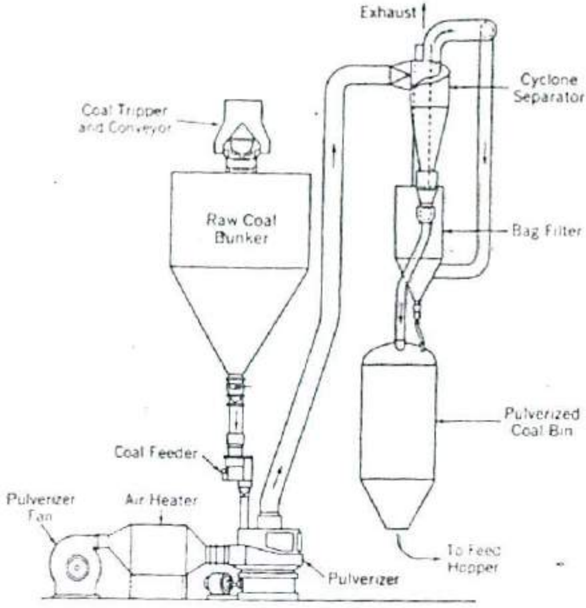


Figure 4-7 Pulverized-coal bin system. *(Courtesy Babcock and Wilcox.)*

cause of the many stages of drying, storing, transporting, etc., the bin system is subject to fire hazards. Nevertheless, it is still in use in many older plants. It has, however, given way to the direct-firing system, which is used exclusively in modern plants.

Compared with the bin system the *direct-firing system* has greater simplicity and hence greater safety, lower space requirements, lower capital and operating costs, and greater plant cleanliness. As its name implies, it continuously processes the coal from the storage receiving bunker through a feeder, pulverizer, and primary-air fan, to the furnace burners (Fig. 4-8). (Another version of this system, less used, places the fan on the outlet side of the pulverizer.) Fuel flow is suited to load demand by a combination of controls on the feeder and on the primary-air fan in order to give air-fuel ratios suitable for the various steam-generator loads. The control operating range on any one direct-firing pulverizer system is only about 3 to 1. Large steam generators are provided with more than one pulverizer system, each feeding a number of burners, so that a wide control range is possible by varying the number of pulverizers and the load on each.

Burners A pulverized-coal *burner* is not too dissimilar to an oil burner. The latter must atomize the liquid fuel to give a large surface-to-volume ratio of fuel for proper interaction with the combustion air. A pulverized-coal burner already receives dried pulverized coal in suspension in the primary air and mixes it with the main combustion

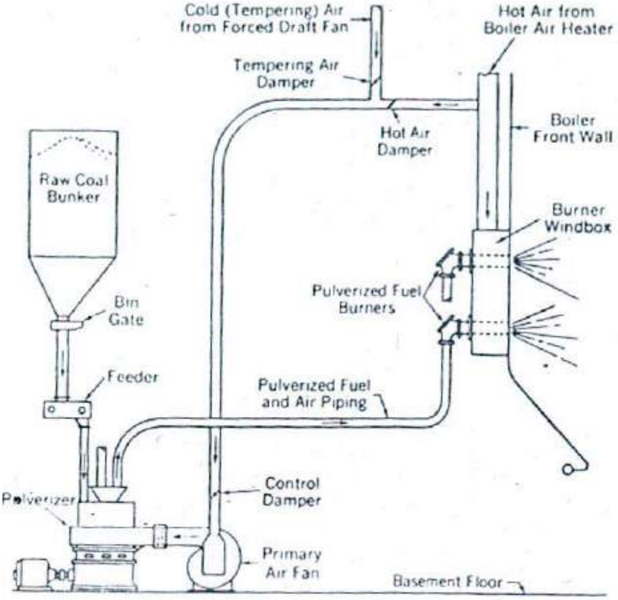


Figure 4-8 Pulverized-coal direct-firing system. *(Courtesy Babcock and Wilcox.)*

air from the steam-generator air preheater. The surface-to-volume ratio of pulverized coal or fineness requirements vary, though not too greatly, from coal to coal (the higher the fixed carbon, the finer the coal). For example, pulverized coal with 80 percent passing a 200-mesh screen and 99.5 percent passing a 50-mesh screen possesses a surface area of approximately 1500 cm^2/g with more than 97 percent of that surface area passing the 200-mesh screen.

The fuel burners may be arranged in one of two configurations. In the first, individual burners, usually arranged horizontally from one or opposite walls, are independent of each other and provide separate flame envelopes. In the second, the burners are arranged so that the fuel and air injected by them interact and produce a single flame envelope. In this configuration the burners are such that fuel and air are injected from the four corners of the furnace along lines that are tangents to an imaginary horizontal circle within the furnace, thus causing a rotative motion and intensive mixing and a flame envelope that fills the furnace area. Vertical firing is also used but is more complex and used only for hard-to-ignite fuels.

The burners themselves can be used to burn pulverized coal only (Fig. 4-9) or all three primary fuels, i.e., pulverized coal, oil, or gas (Fig. 4-10). In Fig. 4-9, the coal impeller promotes the mixing of fuel with the primary air and the tangential doors built into the windbox provide turbulence of the main combustion, or secondary, air to help mix it with the fuel–primary-air mixture leaving the impeller. The total air-

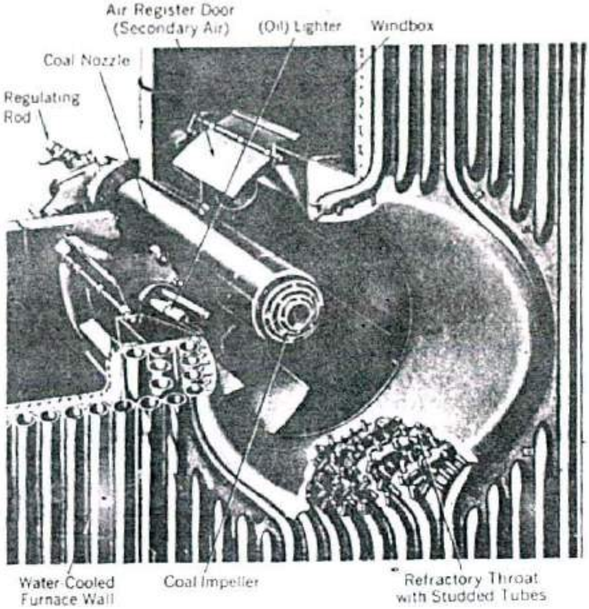


Figure 4-9 A pulverized-coal burner. *(Courtesy Babcock and Wilcox.)*

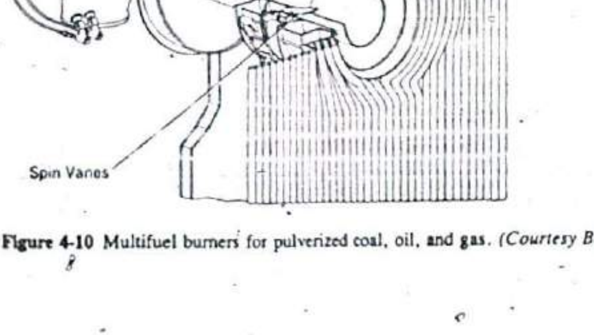


Figure 4-10 Multifuel burners for pulverized coal, oil, and gas. *(Courtesy Babcock and Wilcox.)*

Table 4-3 Excess air required by some fuel systems

Fuel	System	Excess air, %
Coal:	Pulverized, completely water-cooled furnace	15–20
	Pulverized, partially water-cooled furnace	15–40
	Spreader stoker	30–60
	Chain grate and traveling stoker	15–50
	Crushed, cyclone furnace	10–15
Fuel oil:	Oil burners	5–10
	Multifuel burners	10–20
Gas:	Gas burners	5–10
	Multifuel burners	7–12

fuel ratio is greater than stoichrometric (chemically correct) but just enough to ensure complete combustion without wasting energy by adding too much sensible heat to the air. Table 4-3 gives the range of excess air, percent of theoretical, necessary for good combustion of some fuels.

Initial ignition of the burners is accomplished in a variety of ways including a light-fuel oil jet, itself spark-ignited (Fig. 4-9). This igniter is usually energized long enough to ensure a self-sustaining flame. The control equipment ranges from manual to a remotely operated programmed sequence. The igniters may be kept only for seconds in the case of fuel oil or gas. In the case of pulverized coal, however, they are usually kept much longer, sometimes for hours, until the combustion-zone temperature is high enough to ensure a self-sustaining flame. It may also be necessary to activate the igniter at very light loads, especially for coals of low volatility. The impeller is the part of the burner that is subject to severe maintenance problems and is usually replaced once a year or so.

4-7 CYCLONE FURNACES

Cyclone-furnace firing, developed in the 1940s, represents the most significant step in coal firing since the introduction of pulverized-coal firing in the 1920s. It is now widely used to burn poorer grades of coal that contain a high ash content with a minimum of 6 percent to as high as 25 percent, and a high volatile matter, more than 15 percent, to obtain the necessary high rates of combustion. A wide range of moisture is allowable with predrying. One limitation is that ash should not contain a high sulfur content or a high $Fe_2O_3/(CaO + MgO)$ ratio. Such a coal has a tendency to form high ash-fusion temperature materials such as iron and iron sulfide in the slag, which negates the main advantage of cyclone firing.

The main advantage is the removal of much of the ash, about 60 percent, as molten slag that is collected on the cyclone walls by centrifugal action and drained off the bottom to a slag-disintegrating tank below. Thus only 40 percent ash leaves with the flue gases, compared with about 80 percent for pulverized-coal firing. This materially reduces erosion and fouling of steam-generator surfaces as well as the size of dust-removal precipitators or bag houses (Chap. 17) at steam-generator exit. Other advantages are that only crushed coal is used and no pulverization equipment is needed and that the boiler size is reduced. Cyclone-furnace firing uses a range of coal sizes averaging 95 percent passing a 4-mesh screen (region B, Fig. 4-1).

The disadvantages are higher forced-draft fan pressures and therefore higher power requirements, the inability to use the coals mentioned above, and the formation of relatively more oxides of nitrogen, NO_x, which are air pollutants, in the combustion process.

The cyclone is essentially a water-cooled horizontal cylinder (Fig. 4-11) located outside the main boiler furnace, in which the crushed coal is fed and fired with very high rates of heat release. Combustion of the coal is completed before the resulting hot gases enter the boiler furnace. The crushed coal is fed into the cyclone burner at left along with primary air, which is about 20 percent of combustion or secondary air. The primary air enters the burner tangentially, thus imparting a centrifugal motion to the coal. The secondary air is also admitted tangentially at the top of the cyclone at high speed, imparting further centrifugal motion. A small quantity of air, called *tertiary air*, is admitted at the center.

The whirling motion of air and coal results in large heat-release-rate volumetric densities, between 450,000 and 800,000 Btu/(h · ft^3) (about 4700 to 8300 kW/m^3), and high combustion temperatures, more than 3000°F (1650°C). These high temperatures melt the ash into a liquid slag that covers the surface of the cyclone and eventually drains through the slag-tap opening to a slag tank at the bottom of the boiler furnace,

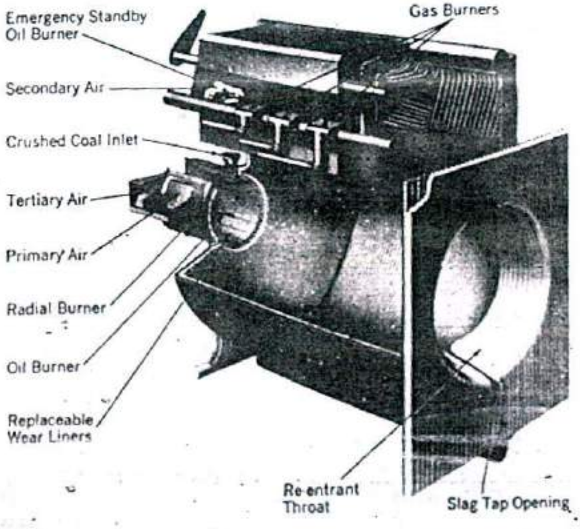


Figure 4-11 A cyclone furnace. *(Courtesy Babcock and Wilcox.)*

where it is solidified and broken for removal. The slag layer that forms on the walls of the cyclone provides insulation against too much heat loss through the walls and contributes to the efficiency of cyclone firing. The high temperatures also explain the large production of NO_x in the gaseous combustion products. These gases leave the cyclone through the throat at right and enter the main boiler furnace. Thus combustion takes place in the relatively small cyclone, and the main boiler furnace has the sole function of heat transfer from the gases to the water-tube walls. Cyclone furnaces are also suitable for fuel-oil and gaseous-fuel firing.

Initial ignition is done by small retractable oil or gas burners in the secondary air ports.

Like pulverized-coal systems, cyclone firing systems can be of the bin, or storage, or direct-firing types, though the bin type is more widely used, especially for most bituminous coals, than in the case of pulverized coal. The cyclone system uses either one-wall, or opposed-wall, firing, the latter being preferred for large steam generators. The size and number of cyclones per boiler depend upon the boiler size and the desired load response because the usual load range for good performance of any one cyclone is from 50 to 100 percent of its rated capacity. Cyclones vary in size from 6 to 10 ft in diameter with heat inputs between 160 to 425 million Btu/h (about 47,000 to 125,000 kW), respectively [12].

The cyclone component requiring the most maintenance is the burner, which is subjected to erosion by the high velocity of the coal. Erosion is minimized by the use of tungsten carbide and other erosion-resistant materials for the burner liners, which are usually replaced once a year or so.

4-8 FLUIDIZED-BED COMBUSTION

We have already noted various attempts at reducing pollutants in the flue gas of powerplant steam generators. The most common are the postcombustion processes that utilize such devices as electrostatic precipitators and baghouses for particulate matter and scrubbers for sulfur dioxide (Chap. 17). Others are processes concurrent with combustion such as cyclone-furnace combustion, in which much of the ash, and hence the particulate matter, is removed during the combustion process. Others, still, are precombustion processes in which "clean" gaseous or liquid fuels, produced from coal by gasification or liquefaction and free of sulfur and ash, are used as steam-generator fuels (Sec. 4-11).

Fluidized-bed combustion is of the concurrent type. It differs from the cyclone furnace in that sulfur is removed during the combustion process. In addition, fluidized-bed combustion occurs at lower temperatures, resulting in lower production of NO_x as well as the avoidance of slagging problems with some coals.

The fluidized-bed combustor has been under development since the 1950s with the aim of perfecting the process and comparing its reliability and economy with SO_2 postcombustion scrubbing. Fluidized beds have been in use for many decades in chemical-industry applications where intimate mixing and contact between reactants are desired. Such contact in a fluidized turbulent state increases heat and mass transfer and reduces time of reaction, plant size, and power requirement. Fluidized beds have even been proposed for use in nuclear-power reactors [1,3]. When used for coal, fluidized-bed combustion results in high combustion efficiency and low combustion temperatures.

Fluidization

A *fluidized bed* is one that contains solid particles which are in intimate contact with a fluid passing through at a velocity sufficiently high to cause the particles to separate and become freely supported by the fluid (Fig. 4-12). A fixed bed, on the other hand, is one in which the velocity of the fluid is too slow to cause fluidization. The minimum fluid velocity necessary for fluidization may be calculated by equating the drag force on a particle due to the motion of the fluid to the weight of the particle. Thus

$$C_D A_c \rho_f \frac{V_t^2}{2g_c} = V \rho_s \frac{g}{g_c} \qquad (4\text{-}3)$$

where C_D = drag coefficient, a function of the particle shape and the Reynolds number, dimensionless

A_c = cross-sectional area of the particle = πr^2 for a spherical particle, ft^2 or m^2

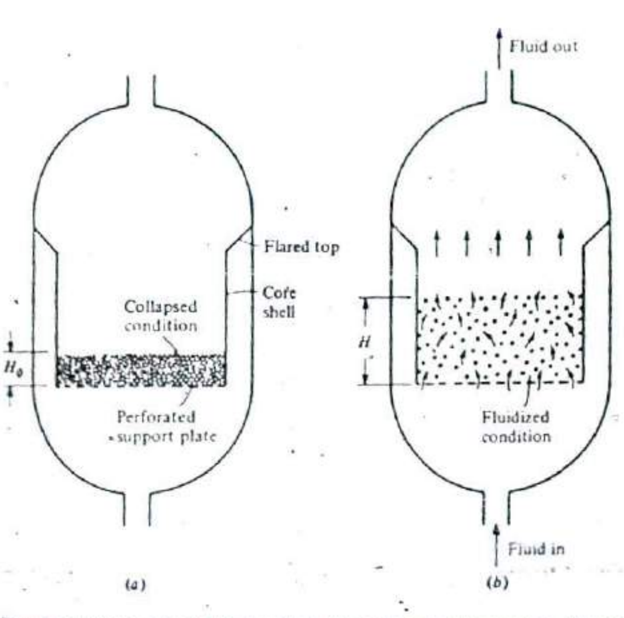


Figure 4-12 Schematic of a fluidized bed: (*a*) particles in collapsed state, (*b*) particles in fluidized state.

r = radius of the particle, if considered spherical, ft or m

ρ_f = density of the fluid, lb_m/ft^3 or kg/m^3

V_s = velocity of the fluid, ft/h or m/s

V = volume of the particle = $\frac{4}{3}\pi r^3$ if spherical, ft^3 or m^3

ρ_s = density of the solid particle, lb_m/ft^3 or kg/m^3

g = gravitational acceleration, ft/h^2 or m/s^2

g_c = conversion factor, $4.17 \times 10^8\ lb_m ft/(lb_f \cdot h^2)$ or $1\ kg/(N \cdot s^2)$

For a spherical particle, Eq. (4-3) yields

$$V_s = \sqrt{\frac{8}{3C_D}\frac{\rho_s}{\rho_f}rg} \tag{4-4}$$

The total pressure drop ΔP, lb_f/ft^2 or Pa, in a fluidized bed is composed of three components

$$\Delta P = \Delta P_w + \Delta P_s + \Delta P_f \tag{4-5}$$

where

ΔP_w = pressure drop due to friction at the wall

ΔP_s = pressure drop due to static weight of solids in bed

ΔP_f = pressure drop due to static weight (or hydrostatic head) of fluid in bed

Fluidized beds usually have large wall diameters, so ΔP_w is relatively small. The fluid in a fluidized-bed combustor varies in a complex manner from air to hot gaseous combustion products. Their average density is much smaller than that of the solids, and ΔP_f, therefore, is also relatively small. Equation (4-5) thus reduces to

$$\Delta P = \Delta P_s = H(1 - \alpha)\rho_s\frac{g}{g_c} \tag{4-6}$$

where

H = height of bed in fluidized state, ft or m

α = average porosity or void fraction of bed in fluidized state = fraction not occupied by solids, dimensionless

The value of α is obtained from

$$\frac{1 - \alpha}{1 - \alpha_o} = \frac{H_o}{H} \tag{4-7}$$

where H_o = height of the bed in the collapsed state, i.e., when all solids are randomly packed at bottom

and α_o = porosity in the collapsed state; the value of α_o of a bed of spheres of equal diameter, randomly packed is very nearly equal to 0.40 [31] but lower for spheres of different diameters

In fluidized-bed combustion, crushed coal, between 1/4 and 3/4 in maximum (6 to 20 mm), is injected into the bed just above an air-distribution grid in the bottom of the bed (Fig. 4-13). The air flows upwards through the grid from the air plenum into the bed, which now becomes the furnace where combustion of the swirling mixture takes place. The products of combustion leaving the bed contain a large proportion of unburned carbon particles that are collected in a cyclone separator, which separates these particles from the gas by imparting a centrifugal acceleration on the mixture. They are then returned back to the fluidized bed to complete their combustion. The boiler water tubes are located in the furnace.

As indicated previously, the most important advantage of fluidized-bed combustion is the concurrent removal of the sulfur dioxide that results normally from the combustion of the sulfur content of the coal. Desulfurization is accomplished by the addition of limestone directly to the bed together with the crushed coal. Limestone is a sedimentary rock composed mostly of calcium carbonate ($CaCO_3$) and sometimes some magnesium carbonate ($MgCO_3$). Limestone absorbs the sulfur dioxide with the help of some oxygen from the excess air, according to

$$CaCO_3 + SO_2 + \frac{1}{2}O_2 \rightarrow CaSO_4 + CO_2 \tag{4-8}$$

The rate of this reaction is maximum at bed temperatures between 1500 and 1600°F (815 to 870°C), though a practical range of operation of fluidized beds of 1380 to 1740°F (750 to 950°C) is common.

The calcium sulfate $CaSO_4$ produced in this process is a dry waste product that is either regenerated or disposed of. Technical problems for handling this calcium sulfate are under study. Reductions in sulfur dioxide emissions of up to 90 percent have been achieved in fluidized-bed pilot plants.

There are other advantages as a result of the low combustion temperatures of fluidized-bed combustors. They allow inferior grades of coal to be used without slagging problems; the carbon and ash carryover in the flue gases does not reach temperatures at which they become soft and foul heat transfer surfaces; the low temperature combustion results in substantial reductions in the emission of oxides of nitrogen NO_x, a serious air pollutant; and cheaper alloy materials are possible, resulting in economy of construction.

Another advantage is the absence of pulverization equipment, resulting in further economies. Also, a fluidized-bed combustor can be designed to incorporate the boiler within the bed, resulting in volumetric heat-transfer rates that are 10 to 15 times higher and surface heat-transfer rates that are 2 to 3 times higher than a conventional boiler.

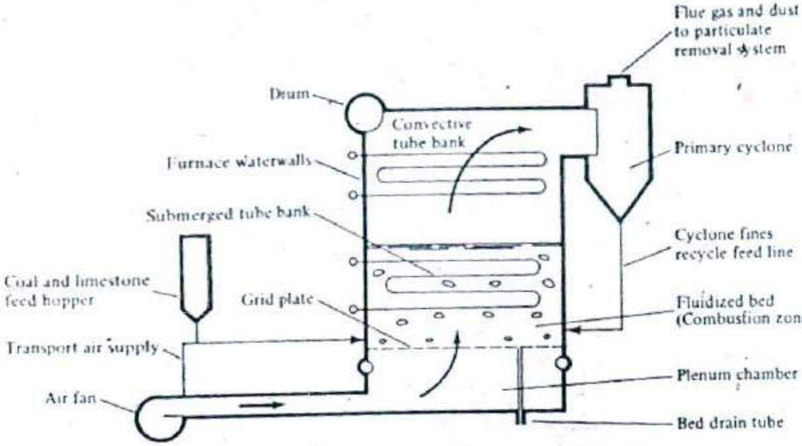


Figure 4-13 Schematic of a fluidized-bed-combustion steam generator. *(Courtesy Combustion Engineering, Inc.)*

A fluidized-bed steam generator is therefore much more compact than a conventional one of the same capacity.

In addition, reductions in sulfur dioxide (and trioxide) in the flue gas means that lower stack-gas temperatures can be tolerated because less acid is formed as a result of the condensation of water vapor. Lower stack-gas temperatures result in an increase in overall plant efficiency.

The problems facing the development of fluidized-bed combustors include those that are associated with feeding the coal and limestone into the bed, the control of carbon carryover with the flue gases, the regeneration or disposal of calcium sulfate, the quenching of combustion by the cooler water tubes within the bed, and variable-load operation.

Fluidized-bed combustors may be used with either a conventional steam powerplant (Rankine cycle) or a combined gas-steam powerplant (Brayton-Rankine) cycle (Sec. 8-6). In the conventional powerplant application, the bed can be of the *atmospheric*, or *pressurized*, type. Atmospheric beds utilize both forced- and induced-draft fans. Pressurized beds use compressors to supply combustion air at pressures of up to 10 bar, which results in slightly higher heat-transfer rates. Electrostatic precipitators, fly-ash removal, air preheaters, and other auxiliaries as are found in conventional steam generators are used. In the combined gas-steam cycle pressurized beds are used. They receive compressed air from the gas-turbine cycle compressor.

By the end of 1981 more than 20 fluidized-bed-combustion water-tube steam generators were contracted for worldwide. Their ratings extended up to 330,000 lb_m/h (about 42 kg/s) capacity, 2450 psig (about 170 bar), and 1000°F (about 540°C) steam conditions. They now are considered by some as competitive with conventional steam generators with gas scrubbing when high-sulfur coals or waste fuels are burned.

4-9 LIQUID FUELS

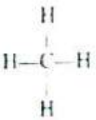
Technically, liquid fuels are an excellent energy source. They are easy to handle, store, and burn and have nearly constant heating values. They are usually a mixture of hydrocarbons that may be represented by the molecule C_nH_m, where m is a function of n that depends upon the "family" of the hydrocarbon. Table 4-4 gives the families of hydrocarbons found in crude and refined oils.

The number of carbon atoms in a hydrocarbon molecule is identified by

1 meth
2 eth
3 prop
4 but
5 pent
6 hex
7 hept
8 oct
9 non
10 dec
11 undec
12 dodec
13 tridec
14 tetradec
15 pentadec
16 hexadec
17 heptadec
18 octadec
19 nonadec
20 eicos
21 heneicos
22 docos
23 tricos
30 triacont
100 hect

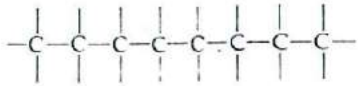
and so on. Straight chain molecules are often called *normal* and are preceded by the letter *n*. Some chain molecules, called *isomers* of the original molecule, are branched and use the prefix *iso*. Some examples are:

1. Methane, CH_4:



$$H-\overset{\displaystyle H}{\underset{\displaystyle H}{\overset{|}{\underset{|}{C}}}}-H$$

2. n-Octane, C_8H_{18}:



$$-\overset{|}{\underset{|}{C}}-\overset{|}{\underset{|}{C}}-\overset{|}{\underset{|}{C}}-\overset{|}{\underset{|}{C}}-\overset{|}{\underset{|}{C}}-\overset{|}{\underset{|}{C}}-\overset{|}{\underset{|}{C}}-\overset{|}{\underset{|}{C}}-$$

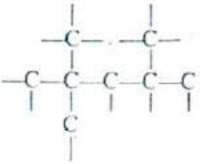
where the H symbol has been dropped for simplicity.

Table 4-4 Hydrocarbon families in oils

Family	General formula	Prefix/suffix	Structure
Paraffin* (alkanes)	C_nH_{2n+2}	-ane	Chain, saturated
Olefin	C_nH_{2n}	-ene, -ylene	Chain, unsaturated, one double bond
Diolefin	C_nH_{2n-2}	-diene	Chain, unsaturated, two double bonds
Naphthene* (cycloparaffin)	C_nH_{2n}	Cyclo-ane	Ring, saturated
Aromatic*			
Benzene	C_nH_{2n-6}		Ring, unsaturated
Naphthalene	C_nH_{2n-12}		Double ring, unsaturated

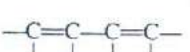
* The primary hydrocarbons found in crude oil

3. Isooctane, C_8H_{18}:



The latter molecule is also called 2-2-4-trimethyl pentane because it has the alkyl (C_nH_{2n+1}) methyl radical (CH_3) attached to the numbers 2, 2, and 4 carbon atoms on a pentane base. It still has the formula C_8H_{18}. This hydrocarbon, because of its structure, is hard to "break" and resists detonation in a gasoline engine. It is this isooctane, rather than normal octane, that is used as a standard with a scale of 100 for detonation-resistant gasoline fuels. Other examples are:

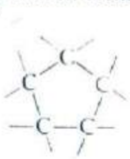
4. Butadiene, C_4H_6:



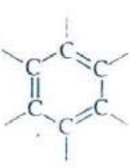
$-C{=}C{-}C{=}C-$

where the double bonds indicate unsaturation, meaning only one hydrogen atom is attached to each of the adjacent carbon atoms.

5. Cyclopentane, C_5H_{10}:



6. Benzene, C_6H_6:



Crude oil is a mixture of an almost infinite number of hydrocarbons, ranging from light gaseous (low n) to heavy (high n) tarlike liquids and waxes of complex molecular structure. They average 83 to 87 percent carbon and 11 to 14 percent hydrogen. Crude oil also contains varying amounts of sulfur, oxygen, nitrogen, particulates, and water. It is refined, fractionally distilled, or cracked into products of narrower range suitable for various applications, such as gasoline, aviation fuels, diesel fuel, fuel oil, lubricating oil, etc.

The fuel most suitable for utility powerplants and industrial uses is fuel oil. It comes in various grades, from light to heavy. They are:

Distillates: No. 1, No. 2, and No. 4
Residual oils: No. 5 (light), No. 5 (heavy), and No. 6

No. 4 could be either a distillate or a mixture of refinery products. The latter grades require preheating for burning and handling. Like coals, oils are classified according

Table 4-5 Average characteristics of fuel oils

Grade	1	2	4	5	6
Analysis, mass %					
C	86.50	86.40	86.10	85.55	85.70
H_2	13.20	12.70	11.90	11.70	10.50
O_2 and N_2	0.20	0.20	0.48	0.70	0.92
S	0.10	0.4–0.7	0.4–1.5	2.0 maximum	2.8 maximum
Sediment and water	Trace	Trace	0.5 maximum	1.0 maximum	2.0 maximum
Ash	Trace	Trace	0.02	0.05	0.08
Density (60°F)					
lb_m/ft^3	51.46	53.98	57.87	59.43	61.50
lb_m/U.S. gal	6.870	7.206	7.727	7.935	8.212
Viscosity					
cSt (100°F)	1.60	2.68	15.0	50.0	360
Pour point, °F	< 0	<0	10	30	65
Atomizing temperature, °F	Atm	Atm	25 minimum	130	200
Higher heating value					
Btu/lb_m	19,940	19,570	18,900	18,650	18,260
Btu/U.S. gallon	137,000	141,000	146,000	148,000	150,000

to physical characteristics by an ASTM specification (ASTM Standards D 396), see for example Table 4-5.

Crude oil has been burned in some boilers and gas turbines. However, because it contains light fractions, its flash point* is low and presents a fire hazard and thus requires special handling procedures. The crude most suitable for direct burning is the so-called *sweet* or low-sulfur crude because it is a low air pollutant. However, it is usually reserved for use as refinery feedstock rather than as a fuel.

In many countries, including the United States, oil is becoming a scarce or valuable commodity and often results in extremely high plant fuel costs. In such countries, new powerplants are often nuclear or coal-fired. For existing oil-fired plants or where no large coal reserves are available, alternatives to straight oil burning are being sought. Some of these are described below.

Emulsion firing An *emulsion* is a suspension of a finely divided fluid in another, in this case water emulsified in heavy oil. Research has shown that, when atomized, the drops of such an emulsion undergo microexplosions of the entrained water as they enter the hot combustion chamber. This causes additional atomization and increased fuel surface-to-volume ratio. In turn, this reduces carbon loss to the combustion process, thus reducing soot and preventing deposit buildup on heat-transfer surfaces. It also reduces excess air requirements and improves combustion efficiency, thus reducing

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* The *flash point* is the lowest temperature that allows inflammable vapors to be formed. It is found by heating the fuel slowly, sweeping a flame across the liquid surface, and noting the temperature at which a distinct flash occurs.

oil requirements. This reduction, although technically meaningful, is, however, much less than needed for oil conservation on a national scale.

Coal-oil and coal-water mixtures (COM and CWM) The main advantage of COMs and CWMs is their ability to reduce or replace oil as fuel in oil-fired utility steam generators. They are currently receiving increasing attention as the cost differential between coal and oil increases.

COMs usually contain about 50 percent coal on a heating-value basis. Their main attraction is that the technology for producing and burning them exists (the first tests on COMs were run as far back as 1880), and several fuel suppliers are offering them commercially, though in limited quantities.

CWMs, preferred by some over COMs because of their greater ability to replace oil, are 70 to 80 percent coal, the rest water, plus a fraction of a percent of a stabilizer. CWMs are used in the form of a slurry, that is, a relatively fine coal suspended in the water. They can thus be handled, stored, transported, and fired like oil. CWMs are expected to result in derating of the steam generators. Limited testing of CWMs as fuel is taking place in the time being.

4-10 LIQUID, GAS, AND SOLID BY-PRODUCTS

Combustible industrial wastes have received increased attention as steam-generator fuels, beginning in the decade of the 1970s. So used, they serve the double purpose of disposing of them and reducing the use of oil. The liquids include solvents, waste oil and oil sludges, oil-water emulsions, polymers, resins, chlorinated hydrocarbons, phenols, tars, combustible chemicals, greases, and fats. The obvious main disadvantage of such liquids is that they vary considerably in heating value, flash and fire points, viscosity, and moisture content. They are not expected to make a big dent in the oil picture.

Gaseous by-products are more attractive, with refinery gas and coke-oven gas the most suitable. *Refinery gas* is generated in the conversion of crude oil to gasoline and other refinery products. It has a high heating value and is often blended with lower-heating-value gas by-products from the refinery prior to combustion. *Coke-oven gas* results from the manufacture of coke from raw coal in a coke oven where the volatile matter is distilled off and the coke-oven gas separated from liquids and solids in the volatile matter by cooling and extraction. It consists of about half H_2, about a third methane, and the remainder of various other gases. Its heating value ranges from 400 to 600 Btu/ft^3 (14,200 to 21,300 kJ/m^3).

Other less attractive gases include *regenerator gas*, which is produced in refinery catalytic cracking processes. It includes less than 10 percent CO and high inert gas and solids contents, and is low in heating value. Its one redeeming factor is its availability from the process at high temperatures (more than 1000°F, 540°C), so that its sensible heat is significant when burned on site. Another, *blast-furnace gas*, a product of iron reduction processes, has a higher CO content (about 30 percent) but has a high dust loading that would cause burner plugging and heat-transfer surface fouling if burned directly. It thus requires cleaning prior to burning.

There are many solid wastes that are available as fuels. The ones that have seen most use are wood waste and sugar-cane waste. Both, naturally, are confined in use to those industries where they are available as waste products. *Wood waste* is found in sawmills, where it represents some 50 percent of the mass of the logs, and in pulp, paper, furniture, plywood, and similar industries. It appears as refuse, bark, sawdust, chips, shavings, and slabs. Most woods have about the same chemical composition, having a proximate analysis of about 70 to 75 percent volatile matter, 25 percent fixed carbon, and 0.5 to 5 percent ash on a dry basis, and a heating value ranging between 8300 to slightly over 9000 Btu/lb$_m$ (about 19,300 to 21,000 kJ/kg), but they vary in density and moisture content. To burn wood efficiently, it must be cut down to chip size in a hogger or chipper to permit continuous feeding and should not have a moisture content greater than 60 or 65 percent. Wood-burning furnaces burn the wood in piles, thin beds on traveling-grate spreader stokers or cyclones. Environmentally, wood burns more cleanly than oil or coal. A relatively small 10-MW powerplant that burns wood chips from forest residue has been in operation since 1977 in Burlington, Vermont. The city has plans for a 50-MW wood-burning station. Other plants are in Wisconsin, Montana, and Oregon.

Sugar-cane waste, also called *bagasse*, is that portion of sugar cane that remains after the sugar is extracted. It consists of matted cellulose fibers and fine particles. Its proximate analysis is about 84 percent volatile matter, 12.5 percent fixed carbon, and 3.5 percent ash on a dry basis, but it contains more than 50 percent moisture. To burn it, it is shredded to short fibers and fines. It has a heating value of 3600 to 4200 Btu/lb$_m$ (about 8400 to 9770 kJ/kg). Sugar mills generate enough bagasse to satisfy much of their demand for cogeneration of both process steam and electricity. Bagasse has been burned most successfully in the so-called *Ward furnace*, which is a refractory furnace that contains individual cells. Piles of bagasse are fed via chutes and burn incompletely in the cells, resulting in partially drying the fuel. A secondary furnace above the cells completes the combustion. Spreader-stoker furnaces have been recently introduced but have not yet proved as simple and reliable as the Ward furnace. Numerous bagasse-fired steam- and electric-generating plants have been in operation at sugar mills throughout the world for decades.

Other solid fuels include food industry by-products, solid wastes, and biomass.

Solid Wastes

Solid wastes, or refuse, are generated by industrial and domestic processes (garbage). Industrial wastes include paper, wood, metal scraps, and agricultural waste products. Domestic wastes include paper, containers, tin, aluminum, food scraps, and sewage. In the United States, solid-waste production is at the rate of about 1 metric ton per person per year and growing at about 5 percent per year. Most of this waste is currently disposed of in land-fill sites near industrial and metropolitan areas. A typical composition of municipal waste in the United States is given in Table 4-6. Some of these wastes, such as paper, some metals, and woods, can be recycled for reuse. Much of it can be burned, since about 50 to 60 percent of it is combustible. Thus there is a potential of burning some 100 million tons of refuse each year for heating or generating steam for electric powerplants.

Table 4-6 Typical composition of municipal waste in the United States

Material	%	Material	%
Paper	50.7	Leather, rubber, plastic	3.3
Food	19.1	Wood	2.9
Metal	10.0	Textiles	2.6
Glass	9.7	Miscellaneous	1.7

The heat content of refuse varies widely, sometimes up to 100 percent, but averages about half that of coal on a mass basis. Thus the energy potential of refuse is not too great, being about 1 percent of the total U.S. needs. The attractive feature, however, is getting rid of a good portion of the total refuse by burning it rather than by dumping it in land-fill sites that are becoming scarce. Another advantage of refuse as fuel is its much lower sulfur content as compared with that of coal or oil.

The main problems are the wide assortment of constituents, a high moisture content, dangers of explosions due to careless volatile-fuel dumping and metal sparks during processing, yet-unknown effects on the operation of large powerplants, and of course, the wide variations in heat content. Because of the latter problem, a practice now receiving wide acceptance is to burn a mixture of solid waste and fossil fuel, with the refuse supplying 10 to 20 percent of the required heat input to the boilers.

Refuse must be carefully prepared for burning. First it is discharged from collecting trucks into a raw-refuse receiving building. It is then shredded in a hammermill and conveyed to a storage bin. The head end of the conveyor is equipped with a magnetic separator that removes ferrous material which may be sent for recycling. An air-classification system removes most of the remaining noncombustibles and heavy particles that cause abrasion. The costs here are in collecting, transporting, and processing the refuse and in the dual-fuel system.

Refuse burning in incinerators has been a wide practice in many parts of the world. One of the first successful refuse-burning electric-generating powerplants, in Bern, Switzerland, burns about 200 tons/day and produces heated water, steam, and electricity. Munich, Germany, has a notable installation, and other cities around the world are following suit.

Some attention and research are directed toward the conversion of organic wastes to synthetic fuels, such as those produced from coal and biomass (below). These processes are largely in the developmental stages.

4-11 SYNTHETIC FUELS

Expected future shortages and supply problems of naturally found liquid and gaseous fuels, as well as environmental problems that restrict the burning of coals, are responsible for the large developmental efforts going on around the world toward the production of synthetic fuels. *Synthetic fuels,* also called *synfuels,* are gaseous and liquid fuels produced largely from coal but also from various wastes and biomass (Sec. 4-12). This production must, among other things, be economical and environmentally acceptable. In this section we shall discuss coal conversion to these two fuels by (1) *gasification* and (2) *liquefaction*. As indicated earlier, the modern use of the available vast resources of coal requires the use of low-sulfur coal, leaving the bulk (about 66 percent) of it unused, cleaning it prior to combustion (washing, froth floating, microwave, and magnetic separation), during combustion (fluidized bed), or after combustion (precipitators, scrubbers). Some of these methods are as yet commercially unproven, and some are costly.

Synthetic fuels in the form of coal-converted clean gaseous or liquid fuels are another alternative. Besides use for steam generation in powerplants, they can be used for domestic, industrial, and transportation purposes.

The basic idea of coal gasification is not new. Gas was manufactured from coal around 1800 when wood and charcoal were growing scarce for smelting of iron, and coal was carbonized to coke by removing its volatiles and using them as a by-product gas. It had a heating value of 550 to 600 Btu/scf (*s*tandard *c*ubic *f*oot), was distributed to urban areas, and was called *town gas*. Coal liquefaction is more recent, initial research having been done in Germany in the 1920s and 1930s. During World War II the Germans and the Japanese, denied access to much of their fuel needs, developed several liquefaction processes that ran their entire war machines, including fuel oil, lubricants, and motor and aviation fuels. One of the German processes, called the Fischer-Tropsch process, is in use commercially in oil-short South Africa today.

Of course, these operations were and are the result of necessity, in one case need in a war economy, in the other political and economic independence. Where such circumstances do not exist, the processes involved must be able to compete economically and environmentally against available fuels or must show potential improvements in expected future shortages to be pursued actively. Because prices of natural fossil fuels have moderated in the early 1980s, the United States government has withdrawn its support from many synthetic-fuel projects. However, it is believed that several such fuels will appear in commercial quantities before the end of this century, and certainly in the next.

Coal Gasification

As stated above, town gas was first produced from coal more than a 100 years ago and had a heating value of 550 to 600 Btu/scf. The next step was *water gas*, similar to our present-day synthetic gas in that it is mainly hydrogen and carbon monoxide, made at about 1800°F (1000°C) by the action of water vapor on coal. It has a heating value of 250 to 325 Btu/scf. This value is increased to 500 to 550 Btu/scf by carburetion with oil cracked at a higher temperature. *Producer gas* obtained from the partial combustion of coal, coke, or wood with added water vapor has a heating value of only 100 to 180 Btu/scf. It is used locally and is not suitable for distribution because distribution costs per Btu are inversely proportional to heating value. It was also used in "gasogens" on motor vehicles in some countries during World War II.

Several coal-gasification processes exist, including three that survived World War

II: the German Lurgi, the Koppers-Totzek, and the Wellman-Galusha processes. These have now evolved into processes that differ slightly, depending upon the particular rank of coal used.

The basic process involves several steps. First, the coal feedstock is crushed by usual methods (Sec. 4-6). If a *caking coal*,* such as certain bituminous coals, is used, the feedstock will have to be pretreated by oxidizing its surface to avoid plugging the gasifier. The following step occurs in the gasifier where the coal undergoes chemical reaction with a mixture of air and steam or oxygen and steam. The reactions with air or oxygen are partial combustion because a rich mixture is used, i.e., one having a fuel-to-air ratio greater than chemically correct, or stoichiometric, which also means that the oxygen in either case is not sufficient to convert all the carbon to carbon dioxide. Coal gasification results in three gas mixtures, classified according to their heating value. They are called *low-Btu*, *medium-Btu*, and *high-Btu* gas. These are obtained in the following ways.

Low-Btu gas The feedstock is reacted with a mixture of air and steam. The air may be enriched in oxygen (i.e., oxygen-to-nitrogen ratio greater than atmospheric) but will be less than stoichiometric. The reactions taking place are

In air
$$C + O_2 + 3.76N_2 \rightarrow CO_2 + 3.76N_2 \tag{4-9}$$

CO_2 from this reaction reacts further with additional C in the rich mixture to give

$$C + CO_2 + 3.76N_2 \rightarrow 2CO + 3.76N_2 \tag{4-10}$$

In steam
$$C + H_2O \rightarrow CO + H_2 \tag{4-11}$$

The result is a gas composed principally of CO, H_2, N_2, and some CO_2. The N_2 may be less than shown if the air is oxygen-enriched. The CO_2 appears if the air is increased beyond that shown or because of stratification (imperfect mixing) in the gasifier. It may also contain small amounts of H_2O, CH_4, and C_2H_6. The gas has a heating value range of 180 to 350 Btu/scf, depending upon the composition of reactants and resulting composition of products.

Medium-Btu gas The feedstock is burned with a mixture of oxygen (rather than air) and steam in the same reactions as given by Eqs. (4-9) to (4-11) but with the nitrogen removed. The result is a gas, composed principally of CO and H_2, that has a heating value range of 250 to 500 Btu/scf, again depending upon the composition of reactants and resulting products. The increase in heating value is a result of the absence of the diluting effect of nitrogen.

In the literature, the term *low-Btu gas* is sometimes used to refer to both low- and medium-Btu gases.

The next step in processing low- and medium-Btu gases is *quenching* to condense the tars and heavy oils that come with the feedstock and did not burn. This is followed by a *purification process* in which the hydrogen sulfide in the gas, formed by the

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* A caking coal is one which softens and agglomerates as a result of the application of heat.

combination of the sulfur in the coal with hydrogen gas, is converted to elemental sulfur by an absorption process* and, simultaneously, a *cleaning process* in which char (solid coal residue, mainly fixed carbon and ash), dust, and ash are removed. These processes occur at low temperature with aqueous solutions at 100 to 220°F so that energy in the form of sensible heat of the product gases is lost to the environment.

High-Btu gas Purified medium-Btu gas may be converted to a high-Btu gas by two additional steps. The first is *shift conversion*, in which CO from the CO-rich gas is saturated with steam and passed through a catalytic reactor thus producing more hydrogen and carbon dioxide

$$CO + H_2O \rightarrow H_2 + CO_2 \qquad (4\text{-}12)$$

The ratio of H_2 to CO_2 in the products can be changed by changing the composition of the reactants. The CO_2 is removed in a wash plant.

The next step is that of *methanation*, which is the production of methane, CH_4, from the available mixture of CO and H_2 in a catalytic reactor

$$3H_2 + CO \rightarrow CH_4 + H_2O \qquad (4\text{-}13)$$

and the H_2O is removed. High-Btu gas is thus largely methane, with a heating value of 950 to 1000 Btu/scf, approaching that of natural gas, which is 950–1100 Btu/scf. It has all the characteristics of natural gas but without the sulfur and other pollutants.

Various methods under development for the production of low- and high-Btu gas may be found in the literature [32]. Because the production of high-Btu gas is more complex and expensive than low-Btu gas, it is intended for use in lieu of natural gas in domestic and industrial applications. Low- and medium-Btu gases are considered for use as utility fuels.

Purified low-Btu gas can be fired directly in a conventional steam generator. It has the advantages of being sulfur and ash free, thus eliminating the need for precipitators and scrubbers (Chap. 17). The lack of SO_2 in the flue gases also permits lower stack temperatures, which results in improved efficiency. In addition, its content of the CO_2 and H_2O inerts reduces combustion temperatures and hence the formation of nitrogen oxides. The disadvantages are the large volumetric rate of flow of the gas for a given heat input to the steam generator, the high costs, and the large demand for water as a source of steam and hydrogen and for cooling in the coal gasification process.

A Combined-Cycle Powerplant

An attractive utilization of low-Btu gas for electric generation is as a fuel for a combined-cycle powerplant. Such cycles are covered more fully in Chap. 8. However, a brief description of such a cycle integrated with a coal gasifier is appropriate at this

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* In contrast to removal of SO_2 from the steam-generator flue gases, the removal of H_2S at this stage is easier and is accomplished by any of several commercially available and proven processes.

time. A *combined-cycle* is one which uses a gas turbine at the high-temperature end and a steam turbine at the low-temperature end.

Low-Btu gasifiers operate at different pressures and temperatures, depending upon the process used. Some operate at high pressures, up to 35 bar, and at exit temperatures between 1000 and 2000°F (540 to 1100°C). As indicated above, the exit gas must be cooled for purification and cleaning. Normally this cooling represents a large loss of energy to the environment. A combined cycle (Fig. 4-14) takes advantage of the high gasifier pressure, and recovers much of that heat loss by a gas-to-gas heat exchanger.

In one proposed design [33] the gas leaves the gasifier at 1 at about 1000°F (540°C) and 300 psia (20 bar), gives up some of its heat in a regenerative heat exchanger leaving at 2, where it is further cooled to 3 by an external heat exchanger to temperatures suitable for purification and cleaning from 3 to 4. The gas then recovers the heat it gave up in the regenerative heat exchanger leaving it at 5. It then enters a combustion

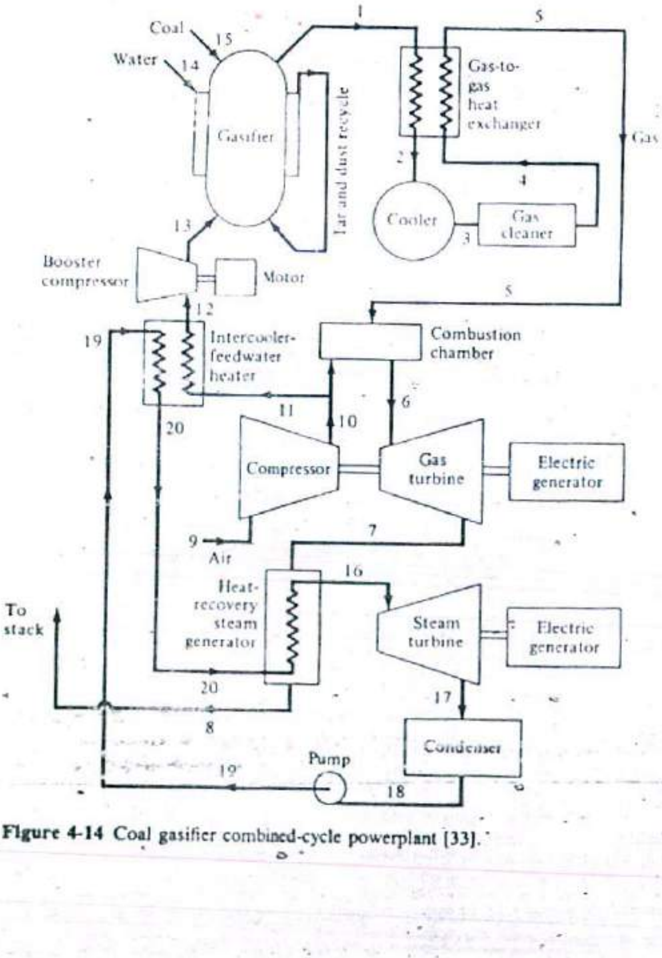

Figure 4-14 Coal gasifier combined-cycle powerplant [33].

chamber where it mixes with compressed air from the gas turbine–driven compressor and leaves at 6 at about 1800°F (980°C). It expands in the gas turbine, leaving it at 7 at about 965°F (520°C). It then enters a heat-recovery steam generator (HRSG) where it generates steam and leaves to the stack at 8 at about 260°F (125°C).

The gas turbine drives one of two electric generators and the compressor. The compressor receives atmospheric air at 9 at about 60°F (15°C) and compresses it to about 600°F (315°C). The compressor has a dual role: it supplies combustion air to the combustion chamber at 10 and gasifier air at 11. The latter is first cooled in a steam-cycle feedwater heater to 12, then boosted by a motor-driven compressor to the gasifier pressure at 13. The gasifier is designed to generate its own steam from feedwater 14. The feedstock 15 reacts there with the steam-air mixture to generate the low-Btu gas at 1.

The steam cycle is fairly standard. Superheated steam at 300 psia (20 bar) and 900°F (480°C) is generated in the HRSG at 16, expands through a steam turbine that drives the second electric generator, and exhausts at 17 to a condenser. The condensate at 18 is pumped, 19, to the feedwater heater which receives its heat from the compressed gasifier air. No bled turbine steam is used in this design, although that form of feedwater heating can be used. The feedwater then enters the HRSG at 20, completing the cycle.

The above cycle (with irreversibilities taken into account) shows an overall efficiency of 34.3 percent, which compares favorably with modern 2400 psi/1000°F/1000°F conventional steam powerplants with scrubbers. An advanced conceptual design with gas turbine inlet at 2800°F (not yet developed) and 2400 psi/1000°F steam shows an efficiency of 43.3 percent [33].

Coal Liquefaction

Coal liquefaction is the conversion of coal into a liquid fuel for direct energy production or a liquid substitute for refinery feedstock from which other liquid fuels may be obtained.

Coal-liquefaction technology, first researched in the 1920s and 1930s, was stimulated in both Germany and Japan by World War II. Japan produced aviation gasoline in a large plant in North Korea which converted coke from coal to calcium carbide in electric furnaces, then to acetylene, acetaldehyde, butyraldehyde, octanol, and finally octane. In a plant in Taiwan, the Japanese also used starch from root vegetables which they fermented to butanol, which was then converted to butyraldehyde, octanol, and octane. The most important German processes were the *Bergius*, which is no longer in use, and the *Fischer-Tropsch* process, which is still used commercially by the Union of South Africa's SASOL Corporation. South Africa has no indigenous oil of its own but has coal. Thanks to coal liquefaction, it is completely independent of foreign oil.

Long, complex hydrocarbon molecular chains have a lower hydrogen-carbon atomic ratio than shorter molecules, like that of octane. In coal liquefaction, the long molecules are shortened by adding hydrogen. The needed hydrogen is generated, and desulfurization is accomplished, in the same manner as for coal gasification. The Fischer-Tropsch process first produces a mixture of CO and H_2 from coal and steam. This is followed by catalytic reactions at about 300°F (150°C) and 150 bar, which yield a range of hydrocarbons from gaseous methane to higher liquid hydrocarbons. These are then separated with methane going as pipeline gas and the rest going to different liquid fuels.

Some half-dozen new processes are currently under development [34] in pilot plants. Scaling these up to commercial sizes is one of the major problems. Another problem, as in coal gasification, is the large demand for water that restricts the use of the large western coal reserves in the United States, which are not located near large supplies of water.

Oil Shale

Shale is a fine-grained rock, formed by the hardening of clay, that splits into thin layers when broken. *Oil shale* is *not* oil-impregnated shale but rather is a shalelike rock impregnated with a waxy organic material called *kerogen*, a substance which originated from vegetation that degraded over millions of years to oil that got absorbed in inorganic matter. Stratification under pressure produced the oil shale. Kerogen can be decomposed to yield synthetic crude oil, called *syncrude* or *shale oil*, by heating in retorts or by underground combustion. One ton of oil shale can produce 25 to 35 gal (95 to 130 L) of low-sulfur oil by retorting at about 950°F (500°C). This oil makes a good refinery feedstock that can be processed further to various useful liquid and gaseous fuels, although one low in fuel oil.

There are vast amounts of oil-shale deposits throughout the world, the largest being in the United States in Colorado, Wyoming, and Utah. It is said that the United States deposits can, by present technology, produce 50 times as much syncrude as all the natural crude produced in the United States to date. The first production was started in France around 1840. The only commercial (publicly owned) facilities nowadays exist in the Soviet Union and China.

Several pilot plants have been successfully operated in the United States, although commercial operations have run into major economic and practical difficulties, resulting in one major cancellation in 1982. Some of the current problems are (1) the large demand for water (3 times as much water is needed as oil produced) in the largely desert areas where oil shale is found in the United States; (2) environmental concerns arising from surface mining; (3) disposal of the spent rock (10 times the mass of the syncrude produced), which has increased in volume due to "puffing" upon heating; and (4) the large amounts of energy required for mining, transportation, processing, and disposal. Doubtless to say, such problems will be less restrictive in future decades when the cost and availability problems of natural crude make oil-shale plants competitive.

Tar Sands

Another potential and very rich source of oil, but one which is even less attractive than oil shale, is found in tar sands in such places as Alberta, Canada, about 10 degrees from the arctic circles. *Tar sand* is a thick, extremely viscous bitumen locked in sands and silt to form a sodden, sticky semiplastic material. It is believed that it contains 2 to 3 times the oil reserves of all the Middle East. Although small pilot plants have operated in the inhospitable terrain, the problems of commercial exploitation seem, at present, to be insurmountable.

One last word regarding synthetic fuels: Coal and other conversion plants, while they are designed to produce clean sulfur- and ash-free fuels, are *not* themselves pollution free. The plants generate enormous amounts of air pollutants such as CO_2, H_2S, SO_2, and NO_x; liquid effluents such as phenols, cresols, xylenols, thiocyanates, and ammonium sulfides; and solid wastes such as ash, slag, and sludge. In addition, carcinogenic compounds in the form of polycyclic aromatic hydrocarbons and amines may be produced from coal tars and coal-derived oils. It is obvious that very careful disposal schemes must be designed at the plant sites.

4-12 BIOMASS

Biomass is organic matter produced by plants, both terrestrial (those grown on land) and aquatic (those grown in water) and their derivatives. It includes forest crops and residues, crops grown especially for their energy content on "energy farms," and animal manure [35]. Unlike coal, oil, and natural gas, which take millions of years to form, biomass can be considered a renewable energy source because plant life renews and adds to itself every year. It can also be considered a form of solar energy as the latter is used indirectly to grow these plants by photosynthesis.

Biomass includes wood waste and bagasse, which have already been covered above, plus other matter. All are highly dispersed and bulky and contain large amounts of water (50 to 90 percent). Thus, it is not economical to transport them over long distances, and conversion into usable energy must take place close to the source, which is limited to particular regions. However, biomass can be converted to liquid or gaseous fuels, thereby increasing its energy density and making feasible transportation over long distances.

Terrestrial crops include (1) sugar crops such as sugar cane and sweet sorghum; (2) herbaceous crops, which are nonwoody plants that are easily converted into liquid or gaseous fuels; and (3) silviculture (forestry) plants such as cultured hybrid poplar, sycamore, sweetgum, alder, eucalyptus, and other hardwoods. Current research focuses on the screening and idenification of species that are suitable for short-rotation growing and on the optimum techniques for planting, fertilization, harvesting, and conversion.

Animal and human waste are an indirect terrestrial crop from which methane for combustion and ethylene (used in the plastics industry) can be produced while retaining the fertilizer value of the manure.

Aquatic crops are grown in fresh, sea, and brackish waters. Both submerged and emergent plants are considered, including seaweeds, marine algae, and of particular interest, the giant California kelp.

Biomass Conversion

Biomass conversion, or simply *bioconversion*, can take many forms: (1) direct combustion, such as wood waste and bagaase (above), (2) thermochemical conversion, and (3) biochemical conversion.

Thermochemical conversion takes two forms: gasification and liquefaction. *Gasification* takes place by heating the biomass with limited oxygen to produce low-Btu gas or by reacting it with steam and oxygen at high pressure and temperature to produce medium-Btu gas. The latter may be used as fuel directly or used in liquefaction by converting it to methanol (methyl alcohol CH_3OH) or ethanol (ethylalcohol CH_3CH_2OH), or it may be converted to high-Btu gas.

Biochemical conversion takes two forms: Anaerobic digestion and fermentation. *Anaerobic digestion* involves the microbial digestion of biomass. (An *anaerobe* is a microorganism that can live and grow without air or oxygen. It gets its oxygen by the decomposition of matter containing it.) It has already been used on animal manure but is also possible with other biomass. The process takes place at low temperatures, up to 65°C, and requires a moisture content of at least 80 percent. It generates a gas consisting mostly of carbon dioxide and methane with minimal impurities such as hydrogen sulfide. The gas can be burned directly or upgraded to synthetic natural gas by removing the CO_2 and the impurities. The residue may consist of protein-rich sludge that can be used as animal feed and liquid effluents that are biologically treated by standard techniques and returned to the soil.

Fermentation is the breakdown of complex molecules in organic compounds under the influence of a ferment such as yeast, bacteria, enzymes, etc. Fermentation is a well-established and widely used technology for the conversion of grains and sugar crops into ethanol. Some 60 million gal were produced in the United States in 1979, with capacity projected to 500 million gal per year by 1985 by the use of surplus grain. It is intended for mixing with gasoline to produce *gasohol* (90 percent gasoline, 10 percent ethanol). By the early 1980s the scheme had not met with great success because of the high cost and the high energy required in the process. One scheme considered for cutting costs of ethanol production by fermentation is in finding less expensive grains or sugars and a process that requires less energy. Glucose produced by hydrolysis of an abundant carbohydrate polymer called *lignocellulose* is being considered for the former.

Biomass energy concepts under study are resulting in the cultivation of large forests in areas not suitable for food production that may yield 10 to 20 tons/acre per year. The energy forest would perhaps be 50 to 200 mi². The trees are to be harvested by automated means, then chipped and pulverized for burning in a powerplant that would be located in the middle of the forest. Sycamore is a promising tree that yields up to 16 tons/acre per year. All of it is used except the foliage, which contains the nutrients and is returned to the soil. A harvested sycamore produces a number of sprouts that are themselves ready for harvesting in 2 to 3 years. Thus no replanting and little fertilization are necessary. It is estimated that an energy farm 350 mi² in area could produce 400 MW of electricity. The electricity costs are uncertain at this time and the possibilities depend upon the costs from other sources, such as oil or coal.

Other schemes envision aquatic farms growing algae, tropical grasses, floating kelp, water hyacinth (one of the most pernicious weeds in the world, one that shelters disease-carrying organisms, causes floods, disrupts hydroelectric plants, and interferes with traffic on major waterways such as the Nile and the Congo), and others. In controlled environments they could yield several hundred tons/acre year. One interesting idea is to use hot condenser cooling water to grow algae in large quantities or increase the yield of other crops.

Finally, while the efficiency of solar energy use in the growth of crops, the photosynthesis efficiency, is rather low, about 3–5%, means of increasing it to 10–11% are under study.

4-13 THE HEAT OF COMBUSTION

In Chap. 1 we treated the first law of thermodynamics from a mechanical engineering point of view, ignoring such energy terms as chemical, electrical, and magnetic. When dealing with combustion systems, we can no longer ignore the chemical energy in the fluid.

The Open System

The first law equation for the steady-state steady-flow system (SSSF) Eq. (1-1*d*) will now be written but for a chemically reactive system (Fig. 4-15), with changes in kinetic and potential energies ignored, as

$$H_R + \Delta Q = H_p + \Delta W_{sf} \tag{4-14}$$

where H_R and H_p are the enthalpies of the reactants and products, respectively, evaluated for their constituents at their respective pressures and temperatures; ΔQ is the net heat added to the system from the surroundings (which is usually negative because in combustion heat is usually rejected to the surroundings); and ΔW_{sf} is the net work done by the system, if any.

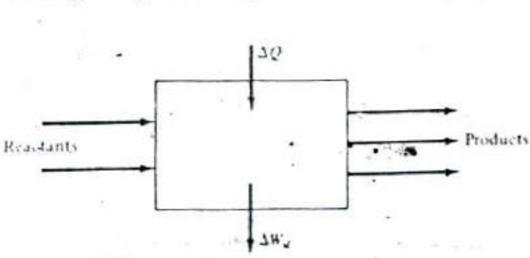

Figure 4-15 A steady-state steady-flow chemically reactive system.

Because the reactants and products are usually composed of several constituents each, Eq. (4-14) is written in the form

$$\sum_{R} (mh) + \Delta Q = \sum_{P} (mh) + \Delta W_{sf} \tag{4-15}$$

where m is the mass and h the specific enthalpy of each constituent. To define the enthalpy, consider the complete combustion of ethane in oxygen

$$C_2H_6 + 3.5O_2 \rightarrow 2CO_2 + 3H_2O \tag{4-16}$$

The enthalpies of the various reactants and products are those that start at the same datum of composition and temperature, in this case the elemental substances and the datum temperature commonly chosen as 25°C (77°F). For example, C_2H_6 is formed from elemental carbon C and hydrogen H_2, CO_2 from C and O_2, and so on. These are exothermic reactions that, when they begin at 25°C and are cooled back to 25°C after the reaction takes place, yield 1211.3 and 3846.7 Btu/lb$_m$ of product, respectively. In other words, in steady flow at 25°C, the formation reactions are

$$2C + 3H_2 \rightarrow C_2H_6 + 1211.38 \text{ Btu/lb}_m\ C_2H_6$$

and

$$C + O_2 \rightarrow CO_2 + 3846.7 \text{ Btu/lb}_m\ CO_2$$

Table 4-7 Enthalpies of formation h_f at 25°C (77°F) and 1 atm pressure*

Substance	Formula	M	State	h_f Btu/lb$_m$	h_f kJ/kg
Carbon	C	12.011	solid	0	0
Oxygen	O_2	32.000	gas	0	0
Hydrogen	H_2	2.016	gas	0	0
Nitrogen	N_2	28.016	gas	0	0
Sulfur	S	32.060	solid	0	0
Carbon monoxide	CO	28.011	gas	−1697.6	−3948.3
Carbon dioxide	CO_2	44.011	gas	−3846.7	−8946.8
Water	H_2O	18.016	liquid	−6825.7	−15,875.5
			vapor	−5774.6	−13,430.8
Methane	CH_4	16.043	gas	−2007.8	−4669.8
Ethane	C_2H_6	30.070	gas	−1211.3	−2817.3
Propane	C_3H_8	44.097	gas	−1013.1	−2356.3
Butane	C_4H_{10}	58.124	gas	−933.7	−2171.6
Octane	C_8H_{18}	114.230	liquid	−941.4	−2189.5
			vapor	−785.1	−1826.0
Nitric oxide	NO	30.008	gas	−1298.8	−3020.8
Nitrogen dioxide	NO_2	46.008	gas	−315.3	−733.3
Sulfur dioxide	SO_2	64.060	gas	−1992	−4632.8

* Based on data from Ref. 36.

The quantities 1211.3 and 3846.7 leave the system and hence are negative. They are given the name *enthalpies of formation*, h_f. Table 4-7 gives values of h_f for various substances at 25°C. Note that the elemental substances, C, O_2, etc., have zero enthalpies of formation. Also note that if the combustion equation involves a liquid, such as octane, or water there will be two enthalpies of formation, depending upon whether they start or end the reaction in a liquid or a vaporized state.

At temperatures other than the datum of 25°C, the enthalpies of formation must include the sensible heat that is a product of the temperature difference and $c_p(T)$, the specific heat at constant pressure which varies with temperature (Sec. 1-2). Table 4-8 gives values of the enthalpies of formation of several substances as a function of temperature.

Because chemical equations, such as Eq. (4-16), are balanced in terms of moles and not masses, and because $m = nM$, Eq. (4-15) is now written in the form

$$\sum_R (nMh_f) + \Delta Q = \sum_P (nMh_f) + \Delta W_{sf} \tag{4-17}$$

where n and M are the number of moles and molecular mass of each constituent, respectively.

Table 4-8 Enthalpies of formation h_f at different temperatures and 1 atm pressure, Btu/lb$_m$*†

Temperature, K	CO_2	CO	H_2O	O_2	H_2	N_2
298	−3846.7	−1697.6	−5774.6	0	0	0
400	−3807.5	−1651.9	−5692.2	40.7	631.4	52.8
500	−3765.4	−1606.5	−5609.4	81.8	1255.5	90.8
600	−3720.4	−1560.2	−5524.0	124.3	1880.4	136.5
700	−3673.1	−1513.0	−5435.9	168.1	2506.9	183.3
800	−3623.7	−1464.5	−5345.0	213.0	3137.4	231.0
900	−3572.6	−1415.0	−5251.1	258.8	3773.3	279.8
1000	−3520.2	−1364.5	−5154.3	305.3	4414.2	329.5
1100	−3466.5	−1313.1	−5054.3	352.5	5062.5	380.2
1200	−3411.9	−1261.0	−4951.4	400.2	5717.8	431.6
1300	−3356.4	−1208.2	−4845.7	448.4	6381.9	483.7
1400	−3300.2	−1154.8	−4737.2	497.0	7055.6	536.5
1500	−3243.4	−1100.9	−4626.2	546.0	7739.1	590.8
1600	−3186.1	−1046.6	−4512.8	595.3	8434.0	643.5
1700	−3128.3	− 992.0	−4397.1	646.0	9136.4	697.6
1800	−3070.1	− 937.0	−4279.6	694.9	9848.2	752.2
1900	−3011.6	− 881.7	−4160.1	745.3	10568	807.0
2000	−2952.8	− 826.1	−4038.9	795.9	11296	862.1
2100	−2893.6	− 770.3	−3916.1	846.8	12031	917.5
2200	−2834.2	− 714.3	−3791.8	929.3	12774	973.1
2300	−2774.6	− 658.2	−3666.2	949.6	13523	1028.9
2400	−2713.1	− 601.8	−3539.4	1001.5	14279	1084.9
2500	−2654.9	− 545.3	−3411.5	1053.7	15043	1141.1

* To convert to kJ/kg multiply by 2.32584.

† Based on data from [36].

Example 4-2 Find the useful heat generated by the combustion of 1 lb_m and 1 scf (standard cubic foot) of ethane in a furnace in a 20 percent deficient air if the reactants are at 25°C and the products at 1500 K. Assume that hydrogen, being more reactive than carbon, satisfies itself first with the oxygen it needs and burns completely to H_2O. Five percent of the heat of combustion is lost to the furnace exterior.

SOLUTION The stoichiometric equation for ethane in air is

$$C_2H_6 + 3.5O_2 + 13.16N_2 \rightarrow 2CO_2 + 3H_2O + 13.16N_2$$

(where there are 3.76 mol N_2/mol O_2 in atmospheric air, thus $13.16 = 3.5 \times 3.76$). With 20 percent deficient air multiply the O_2 and N_2 mol by 0.8. H_2 will burn completely to H_2O and C will burn partially to CO_2 and partially to CO:

$$C_2H_6 + 2.8O_2 + 10.528N_2 \rightarrow aCO_2 + bCO + 3H_2O + 10.528N_2$$

Carbon balance: $$a + b = 2$$

Oxygen balance: $$a + \frac{b}{2} + \frac{3}{2} = 2.8$$

Thus $a = 0.6$, $b = 1.4$, and the combustion equation is

$$C_2H_6 + 2.8O_2 + 10.528N_2 \rightarrow 0.6CO_2 + 1.4CO + 3H_2O + 10.528N_2$$

As there is no work done in a furnace, Eq. (4-17) is written as

$$\Delta Q = \sum_P (nMh_f)_{1500\text{K}} - \sum_R (nMh_f)_{25^\circ\text{C}}$$

$$\begin{aligned}\sum_P (nMh_f)_{1500\text{K}} &= 0.6 \times 44.011 \times -3243.4 + 1.4 \times 28.011 \times -1100.9 \\ &\quad + 3 \times 18.016 \times -4626.2 + 10.528 \times 28.016 \times +590.8 \\ &= -85647.2 + -43172.2 + -250036.9 + -174257.9 \\ &= -204{,}598.4 \text{ Btu/(lb·mol } C_2H_6)\end{aligned}$$

$$\sum_R (nMH_f)_{25^\circ\text{C}} = -1211.3 + 0 + 0 = -1211.3 \text{ Btu/(lb·mol } C_2H_6)$$

$$\begin{aligned}\text{Thus}\quad \Delta Q &= -204{,}598.4 - -1211.3 = -203387.1 \text{ Btu/(lb·mol } C_2H_6) \\ &= \frac{-203{,}387.1}{30.070} = -6763.8 \text{ Btu/lb}_m\ C_2H_6 \\ &= -6763.8 \times 2.32584 = -15731.5 \text{ kJ/kg } C_2H_6\end{aligned}$$

$$\begin{aligned}\Delta Q_{\text{useful}} &= 0.95 \times -6763.8 = -6425.6 \text{ Btu/lb}_m\ C_2H_6 \\ &= -14{,}944.9 \text{kJ/kg } C_2H_6\end{aligned}$$

A standard cubic foot is obtained at 1 atm and 60°F. Since C_2H_6 is a gas, its density at these conditions is obtained from $PV = mRT$

or
$$\rho = \frac{m}{V} = \frac{P}{RT} = \frac{14.696 \times 144}{(1545/30.70) \times (60 + 460)} = 0.0792\ \text{lb}_m/\text{scf}$$

Thus
$$\Delta Q_{\text{useful}} = -6425.6 \times 0.0792 = -508.9\ \text{Btu/scf}$$

The Closed System

Combustion equations for fuels burning in a closed system, such as a cylinder or bomb, may be obtained by writing the first-law equation for the closed system. Thus, by analogy with Eq. (4-14)

$$U_R + \Delta Q = U_P + \Delta W_{nf} \tag{4-18}$$

when ΔW_{nf} is the nonflow work. For gases

$$H = U + PV = U + nR_oT$$

where R_o is the universal gas constant. Thus combustion calculations for a closed system may be carried out using the enthalpies in Tables 4-6 and 4-7 by modifying Eq. 4-18 to

$$\sum_R (nMh_f - nR_oT) + \Delta Q = \sum_P (nMh_f - nR_oT) + \Delta W_{nf} \tag{4-19}$$

4-14 HEATING VALUES

We have repeatedly used the term *heating value* in this chapter to indicate the useful energy content of different fuels. Actually there is more than one heating value for each fuel and these should be carefully differentiated. We have already explained the difference between the *higher heating value* (HHV) and the *lower heating value* (LHV). The former is the heat released when water vapor in the products due to the combustion of hydrogen in the fuel condenses, the latter when it stays in the vapor state (Sec. 4-3). Heating values are commonly tabulated in the literature as the heat released when complete combustion begins at a standard temperature, such as 77°F or 25°C, and the products are cooled to the *same* temperature, in a steady-flow adiabatic system without work. Thus it can be calculated from Eq. (4-17) by putting $\Delta W_{sf} = 0$ and replacing ΔQ by HV for heating value

$$\text{HV} = \left[\sum_P (nMh_f) - \left(\sum_R nMh_f\right)\right]_{T_1} \tag{4-20}$$

where T_1 represents the standard temperature.

The heating value given by Eq. (4-18) is for an open system and is thus sometimes called the *enthalpy of combustion*. It depends upon T_1; upon whether the water vapor formed in combustion condenses (in which case it is the HHV), or not (in which it is the LHV); and upon whether the reactant fuels were in a liquid or vapor state, as it takes a certain amount of energy to vaporize a liquid fuel such as octane.

Example 4-3 Find the higher and lower heating values of ethane at 25°C (77°F).

SOLUTION Using Eqs. (4-16) and (4-20) and Table 4-6, HHV is calculated by assuming that H_2O in the products has condensed.

$$\sum_P (nMh_f)_{25^\circ C} = 2 \times 44.011 \times -3846.7 + 3 \times 18.016 \times -6825.7$$

$$= -707{,}509.6 \text{ Btu/(lb·mol } C_2H_6)$$

$$\sum_R (nMh_f)_{25^\circ C} = 1 \times 30.070 \times -1211.3 + 3.5 \times 32.00 \times 0$$

$$= -36{,}423.8 \text{ Btu/(lb·mol } C_2H_6).$$

$$\text{Thus HHV} = -707{,}509.6 - -36{,}4238 = -671{,}085.8 \text{ Btu/(lb·mol } C_2H_6)$$

$$= \frac{-671{,}085.5}{30.070} = -22{,}317.5 \text{ Btu/lb}_m\ C_2H_6$$

$$= -22{,}317.5 \times 2.32584 = -51{,}906.8 \text{ kJ/kg } C_2H_6$$

LHV is calculated by using $h_f = -5774.6$ Btu/lb$_m$ for H_2O vapor instead of -6825.7 Btu/lb$_m$ for H_2O liquid, resulting in

$$\text{LHV} = -614{,}276.0 \text{ Btu/(lb·mol } C_2H_6)$$

$$= -20{,}428.2 \text{ Btu/lb}_m = -47{,}512.7 \text{ kJ/kg}$$

Note that in calculating the heating values, writing the complete combustion equation in oxygen alone suffices because the addition of nitrogen on both sides of the equation with h_f the same at T_1 (0 at 25°C) does not alter the results. Note also that the same results are obtained with stoichiometric and lean mixtures (excess oxygen or air), so long as complete combustion (C to CO_2, H_2 to H_2O, etc.) is assured.

Heating values for complex fuels such as coal and gas mixtures are obtained by writing the stoichiometric equation for the fuel and following a similar procedure as above.

Heating values for a closed system are obtained by modifying Eq. (4-19) to

$$\text{HV}_{nf} = \sum_P (nMh_f - nR_oT) - \sum_R (nMh_f - nR_oT)$$

and since the temperature of the products is brought back to that of the reactants T_1

$$\text{HV}_{nf} = \text{HV} - (n_P - n_R)R_oT_1 \tag{4-21}$$

where HV_{nf} = nonflow, a closed system heating value

HV = heating value as defined for a steady-flow system

n_P and n_R = total number of moles of products and reactants, respectively

HV_{nf} is sometimes called the *internal energy of combustion*. Needless to say, there are as many HV_{nf}s as there are HVs; that is, they both depend upon T_1, whether the water vapor formed in the combustion process condenses or not, so there are HHV_{nf} and LHV_{nf}, and whether fuels begin the reaction in a liquid or vapor state.

In the literature, heating values of different fuels are usually listed for an open system and for a given T_1, which unfortunately varies from source to source. For example, 60°F and 77°F (25°C) are used, though the numerical differences between them are fairly small.

4-15 COMBUSTION TEMPERATURES

If the heat of combustion or a portion of it is kept in the gases, it will have the effect of increasing their enthalpy (internal energy in a closed system) and hence their temperature. The high temperature is then normally used to effect heat transfer from the gases to the heat-transfer surfaces (water tubes, superheaters, and reheaters in a steam generator) by radiation and/or convection. The same basic first-law equation used to calculate heats of combustion and heating values of fuels can be used to calculate combustion temperatures.

Example 4-4 A low-Btu gas from a coal-gasification plant has the following volumetric analysis: CO 22.9 percent; H_2 10 percent, CO_2 4.4 percent, N_2 62.7 percent. It enters the furnace at 1 atm and 800 K and burns in 100 percent air at 25°C in a steady-flow combustion process during which it loses 10.72 Btu/ft³ at inlet conditions to the surroundings. Find the temperature of the products.

SOLUTION Recalling that volumetric and molal analyses are equal for gases, the stoichiometric combustion equation (for 100 percent air) is

$$0.229CO + 0.100H_2 + 0.044CO_2 + 0.627N_2 + \left(0.229 + \frac{0.100}{2}\right)O_2$$
$$+ \left(0.229 + \frac{0.100}{2}\right)3.76N_2 \rightarrow (0.229 + 0.044)CO_2 + 0.100H_2O$$
$$+ \left[\left(0.229 + \frac{0.100}{2}\right)3.76 + 0.627\right]N_2$$

or

$$(0.229CO + 0.100H_2 + 0.044CO_2 + 0.627N_2) + 0.279O_2 + 1.049N_2$$
$$\rightarrow 0.273CO_2 + 0.100H_2O + 1.676N_2$$

Using Eq. (4-17)

$$\sum_P (nMh_f)_{T_2} = \sum_R (nMh_f) + \Delta Q$$

Using $PV = nR_uT$. 1 mol of gas, between parentheses in the combustion equation above, occupies

$$V = \frac{R_uT}{P} = \frac{1545\ (800 \times 1.8)}{14.696 \times 144} = 1051.3\ \text{ft}^3$$

Thus $\Delta Q = -10.7 \times 1051.3 = 11{,}270$ Btu/mol gas.

$\sum_R (nMh_f)$ are figured at 800 K for the fuel and 25°C for the air. Thus

$$\sum_R (nMh_f) = 0.229 \times 28.011 \times -1464.5 + 0.100 \times 2.016 \times 3137.4$$
$$+ 0.044 \times 44.011 \times -3623.7 + 0.627 \times 28.016 \times 231.0$$
$$+ 0 + 0 = -11{,}721\ \text{Btu/(lb·mol fuel)}$$

Therefore

$$\sum_P (nMh_f) = -11{,}721 - 11{,}270 = -22{,}991\ \text{Btu/(lb·mol fuel)}$$

It is now necessary to solve the problem by finding a value for the combustion temperature that results in a $\sum_P (nMh_f)$ of $-22{,}991$ Btu/(lb·mol fuel). This value is found by trial and error to be 1400 K. To confirm

$$\sum_P (nMh_f)_{1400\ \text{K}} = 0.273 \times 44.011 \times -3300.2 + 0.1 \times 18.016 \times -4737.2$$
$$+ 1.676 \times 28.016 \times 536.5 = -22{,}995\ \text{Btu/(lb·mol fuel)}$$

Thus the combustion temperature is 1400 K = 1127°C = 2520°R = 2060°F.

The Adiabatic-Combustion Temperature

It can be seen from Example 4-4 that a lean mixture, e.g., 120 percent combustion air, would yield a lower combustion temperature because of the dilution effect of the excess air. A rich mixture would also yield a lower temperature because of incomplete combustion.

A fuel burning with no heat exchange with the surroundings and no work done will result in the *adiabatic-combustion* or *adiabatic-flame temperature*. It is greater if the fuel is burned in oxygen than in air because of the dilution effect of nitrogen and greater for a stoichiometric than either a lean or rich mixture because, as above, a lean mixture has a dilution effect, whereas a rich mixture results in incomplete combustion.

At the adiabatic-flame temperature, $\Delta Q = 0$ and

$$\sum_P (nMh) = \sum_R (nMh) \qquad (4\text{-}22)$$

Example 4-5 A desulfurized coal has a moisture-ash-free ultimate analysis of C 81.1 percent, H_2 4.4 percent, O_2 2.7 percent, N_2 1.8 percent on a mass basis. It burns in 120 percent air. The reactants were at 25°C. Find the adiabatic-flame temperature.

SOLUTION The relative number of moles of each constituent is found by dividing its mass percent by its molecular mass M. Thus the molal composition of the coal would be

$$\frac{0.811}{12.011}C + \frac{0.044}{2.016}H_2 + \frac{0.027}{32}O_2 + \frac{0.018}{28.016}N_2$$

or

$$0.06752C + 0.02183H_2 + 0.00084O_2 + 0.00064N_2$$

Normalizing to 1 mol of coal gives

$$0.7434C + 0.2403H_2 + 0.0092O_2 + 0.0071N_2$$

The stoichiometric combustion equation is

$$0.7434C + 0.2403H_2 + 0.0092O_2 + 0.0071N_2 + \left(0.7434 + \frac{0.2403}{2} - 0.0092\right)O_2 + \left(0.7434 + \frac{0.2403}{2} - 0.0092\right)3.76N_2 \rightarrow 0.7434CO_2 + 0.2403H_2O + \left(0.7434 + \frac{0.2403}{2} - 0.0092 + 0.0071\right)N_2$$

or

$$0.7434C + 0.2403H_2 + 0.0092O_2 + 0.0071N_2 + 0.85435O_2 + 3.21236N_2 \rightarrow 0.7434CO_2 + 0.2403H_2O + 3.21946N_2$$

120 percent air means that atmospheric air is increased by 20 percent. The combustion equation is therefore

$$0.7434C + 0.2403H_2 + 0.0092O_2 + 0.0071N_2 + (1.20 \times 0.85435)O_2 + (1.2 \times 3.21236)N_2 \rightarrow 0.734CO_2 + 0.2403H_2O + (0.2 \times 0.85435)O_2 + (1.2 \times 3.21236 + 0.0071)N_2$$

or

$$0.7434C + 0.2403H_2 + 0.0092O_2 + 0.0071N_2 + 1.02522O_2 + 3.85483N_2 \rightarrow 0.7434CO_2 + 0.2403H_2O + 0.17087O_2 + 3.86193N_2$$

All reactants in this particular case are made up of elemental substances (not true if the fuel contained complex molecules such as a hydrocarbon), so that at 25°C

$$\sum_R (nMh_f) = 0$$

The temperature of the products that yields $\sum_P (nMh_f) = 0$ is now obtained by trial and error. At 2100 K

$$\begin{aligned}\sum_P (nMh_f) &= 0.7434 \times 44.011 \times -2893.6 \\ &+ 0.2403 \times 18.016 \times -3916.1 \\ &+ 0.17087 \times 32 \times 846.8 \\ &+ 3.86193 \times 28.016 \times 917.5 = -7725\end{aligned}$$

Similarly, at 2200 K

$$\sum_P (nMh_f) = 1221 \text{ Btu/(lb·mol fuel)}$$

By interpolation, the adiabatic-flame temperature = 2186 K = 3935°R = 3475°F.

The combustion temperatures as calculated in the previous two examples are higher than actual because *dissociation* of some of the products takes place at high temperatures. For example, some CO_2 dissociates to CO and $\frac{1}{2}O_2$, which is an endothermic reaction that lowers the temperature. *Chemical equilibrium* occurs at a certain temperature when the reaction rate is the same in both directions, i.e., when

$$CO + \frac{1}{2}O_2 \rightleftarrows CO_2 \tag{4-23}$$

and

$$H_2 + \frac{1}{2}O_2 \rightleftarrows H_2O \tag{4-24}$$

The effect of dissociation on the theoretically obtained temperatures above is greater the higher the temperatures but is lower for lean mixtures because the excess oxygen tends to drive the reactions toward completion and the higher the pressure. Thus the effect of dissociation is minimal in steam-generator furnace combustion, which occurs with lean mixtures around 3000°F.

Finally, it should be recalled that all combustion calculations can be done if there is heat exchange to or from the environment (as in Example 4-4), work done by or on the system, or even changes in kinetic or potential energy between products and reactants by including the proper terms into Eq. (4-17).

PROBLEMS

4-1 A sample of coal has the following molal analysis C 67.35%, H_2 26.26%, O_2 2.28%, N_2 0.57%, S 1.37%, H_2O 2.17%. Write the complete combustion equation in stoichiometric air and calculate the coal ultimate analysis, mass percent.

4-2 Write the complete combustion equation for the anthracite coal in Table 4-2, assuming stoichiometric air and find the dew point, degrees centigrade, of the combustion products if the total pressure is 1 bar.

4-3 H_2 burns in pure oxygen in a chemically correct (stoichiometric) mixture. Write the combustion equation and calculate (*a*) the mass of products per unit mass of H_2, and (*b*) the lower heating value of H_2 if its higher heating value is 61,100 Btu/lb_m.

4-4 Calculate the higher and lower heating values, in Btus per pound mass, using the Dulong-type formula, of the anthracite coal in Table 4-2, if the total pressure is 1 atm.

4-5 A gaseous fuel that is derived from coal (Sec. 4-11) has the following ultimate volumetric analysis H_2 47.9%, methane (CH_4) 33.9%, ethylene (C_2H_4) 5.2%, CO 6.1%, CO_2 2.6%, N_2 3.7%, and O_2 0.6%. It burns in 110 percent of theoretical air. Calculate (*a*) the volume flow rate of air required per unit volume flow rate of the gas when both are measured at the same pressure and temperature, and (*b*) the dew point of the combustion products, in degrees fahrenheit, if the total pressure is 2 atm.

4-6 10,000 U.S. gal of a fuel oil are burned per hour in 20 percent excess air. The fuel oil has the following ultimate analysis by mass: C 87%, S 0.9%, H_2 12%, ash 0.1%. Write the combustion equation and find the volume flow rate of air required, in cubic feet per minute, if the fuel has a density of 7.73 lb_m/U.S. gal and the air is at 1 atm and 60°F.

4-7 A southern California natural gas has the following ultimate analysis by mass: H_2 23.3%, C 74.72%, N_2 0.76%, and O_2 1.22%. The flue gases have the following volumetric analysis: H_2O 15.583%, CO_2 8.387%, O_2 3.225%, N_2 72.805%. Calculate (*a*) the percent theoretical air used in combustion and (*b*) the dew point, in degrees centigrade, if the flue gases are at 2 bars.

4-8 A fuel oil composed only of carbon, hydrogen, and sulfur is used in a steam generator. The volumetric flue gas analysis on a dry basis is: CO_2 11.7%, CO 0.440%, O_2 4.002%, SO_2 0.176, and N_2 83.682. Find (*a*) the fuel mass composition, (*b*) the air-fuel ratio by mass, (*c*) the excess air used, in percent, and (*d*) the dew point, in degrees centigrade, of the flue gases if their pressure is 2 bar.

4-9 A fuel oil burned in a steam generator has a composition which may be represented by $C_{14}H_{30}$. A dry-basis flue-gas analysis shows the following volumetric composition: CO_2 11.226%, O_2 4.145%, CO 0.863%, N_2 83.766%. Write the complete combustion equation for 1 mol of fuel and calculate (*a*) the air-to-fuel ratio by mass, (*b*) the excess air, in percent, and (*c*) the mass of water vapor in the flue gases per unit mass of fuel.

4-10 A crushed bituminous coal to be used in a fluidized-bed combustion chamber varies in size between 1/4 and 3/4 in and has a density of 80 lb_m/ft^3. The coefficient of drag when fluidized is 0.60. Calculate (*a*) the minimum gas velocity that fluidizes all the coal if the gas is at 1600°F and 9-atm pressure, and (*b*) the pressure drop in the bed, psi. Assume that the coal in the collapsed state has a height of 2 ft and a porosity of 0.25 and that the gas density can be approximated by that of pure air.

4-11 10,000 tons of coal are burned in a powerplant per day. The coal has an as-received ultimate analysis of C 75%, H_2 5%, O_2 6.7%, H_2O 2.5%, S 2.3%, N_2 1.5%, ash 7.0%. It burns in excess air in a fluidized-bed combustor. Calculate (*a*) the mass of calcium carbonate to be added, in tons per day, and (*b*) the mass of calcium sulfate to be disposed of, in tons per day. (The molecular mass of calcium = 40.)

4-12 Write the chemical formula and sketch the molecule for the following hydrocarbons: (*a*) ethane, (*b*) ethene or ethylene, (*c*) decane, (*d*) iso-decane (2,2,3,3 tetramethyl hexane), (*e*) pentatriacontane (do not sketch), (*f*) isobutene (2-methyl propene), (*g*) 1,5-heptadiene (the numbers indicate the positions of the carbon atoms that precede double bonds), (*h*) cyclohexane, (*i*) naphthalene, (*j*) 1-methyl naphthalene (a methyl radical CH_3 attached to a carbon atom instead of a hydrogen atom), (*k*) tetracontane (do not sketch), and (*l*) dotriacontahectane (do not sketch).

4-13 A coal-oil mixture (COM) is composed of the bituminous medium volatility coal listed in Table 4-2 and a distillate oil no. 2 that has an ultimate analysis on a mass basis of C 87.2%, H_2 12.5%, S 0.3%, N_2

0.02%, and negligible O_2 and ash, and has a higher heating value of 19,430 Btu/lb_m. The mixture is 50-50 on a heating-value basis. It burns in 15 percent excess air. Write the complete combustion equation and calculate the air-to-fuel ratio by mass.

4-14 Purified medium-Btu gas at 77°F is burned in 110 percent theoretical air. It leaves the combustion chamber at 1700°F. 5 percent of the heat released is lost to the environment. Calculate the useful heat added to the chamber in Btus per pound mass and Btu per standard cubic foot (at 14.696 psia and 60°F) of the gas.

4-15 A purified low-Btu gas at 77°F burns in stoichiometric air and gives off 52.49 Btu/lb_m in the combustion chamber. Calculate the exit temperature, in degrees Fahrenheit.

4-16 Gaseous propane at 77°F is mixed with air at 300°F and ignited. What percentage theoretical air must be used if the temperature of the products is 1700°F? Assume an adiabatic process.

4-17 100,000 standard cubic feet per minute of a purified low-Btu gas enters a gas turbine combustion chamber at 440°F, where it burns adiabatically in 150 percent theoretical air. It then drives a gas turbine and leaves it at 1160°F. Calculate the power input to the turbine in megawatts.

4-18 Ethane at 25°C burns completely in stoichiometric air in a vertical cylinder so that a piston is pushed upward. The cylinder diameter is 0.15 m and the piston mass is 20 kg. The air pressure above the piston is 1 bar. The initial cylinder volume is 0.1 m^3. Calculate the final cylinder volume, in cubic meters, if the products of combustion at the end of expansion were at 1600 K. (The Universal gas constant in SI units is 8314.3 J/kg·mol·K.)

4-19 300 lb_m/h of liquid octane is burned in an internal combustion engine with 125 percent theoretical air. The initial temperature is 77°F and the products leave the cylinder at 1160°F. 85% C burns to CO_2; the rest burns to CO. (*a*) Write the complete combustion equation for 1 mol of C_8H_{18}, (*b*) find the heat released in the process, Btu/mol C_8H_{18}, and (*c*) find the horsepower developed if the engine efficiency is 15 percent.

4-20 A chemically correct mixture of gaseous methane and air at 77°F is admitted into a nozzle where it burns completely. Calculate the nozzle exit velocity in feet per second if the exit temperature is 2000 K.

4-21 Calculate the higher and lower heating values in Btus per pound mass at 25°C of fuel oil no. 2 (Table 4-5), assuming that O_2 and N_2 have mass percentages of 0.1 each.

4-22 Gaseous propane is mixed with air at 77°F and burned. The products reached a temperature of 1520°F. There was a loss of 10 percent of the reactants energy by heat transfer. Calculate the percentage theoretical air used.

4-23 Calculate the adiabatic flame temperature, in degrees fahrenheit, for the subbituminous coal in Table 4-2 if it burns from 77°F with 100 percent air. Ignore the effect of sulfur on the energy balance.