Succeeding cycles can lead to dry residue or to concentrated solutions during operating conditions which provide moisture. In the case of the steam path, as the expansion crosses into the moisture region, the first droplets of water condensed from the steam will tend to contain most of the contaminants. Concentration-enhancement factors of 100 to 1,000 can be achieved. In modern reheat turbines the early-moisture region occurs in one of the later few stages of the low-pressure section, and at light loads can occur on the most highly stressed last-stage buckets.

Extreme care must be exercised in protecting turbines from chemical contamination during installation, operation, and maintenance. Any deviation from sound feedwater-treatment practice during condenser leaks should be done with the full realization that damage to the turbine may result.

9.5 POWER PLANT HEAT EXCHANGERS

by William J. Bow, assisted by Donald E. Bolt

NOTE: Standards for this industry retain USCS units except as indicated in the text.

SURFACE CONDENSERS

REFERENCE: Heat Exchange Institute Standards for Steam Surface Condensers.

The power plant surface condenser is attached to the low-pressure exhaust of a steam turbine (see Figs. 9.5.1 and 9.5.2). Its purposes are (1) to produce a vacuum or desired back pressure at the turbine exhaust for the improvement of plant heat rate, (2) to condense turbine exhaust steam for reuse in the closed cycle, (3) to deaerate the condensate, and (4) to accept heater drains, makeup water, steam drains, and start-up and emergency drains.

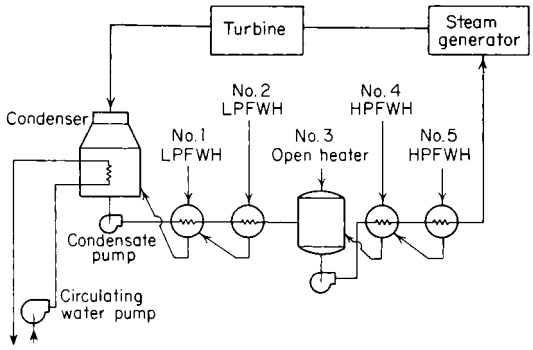


Fig. 9.5.1 Equipment arrangement, schematic.

An economical turbine back pressure is from 1.0 to 3.5 inHg abs. The factors involved in establishing this pressure are involved and will not be discussed here.

An equipment diagram of a closed power plant cycle is shown in Fig. 9.5.1.

For a condenser to deaerate the condensate, it must remove oxygen and other noncondensable gases to an acceptable level compatible with material selection and/or chemical treatment of the feedwater (condensate). Depending on materials and treatment, the dissolved O_2 level must normally be kept below 0.005 cm^3/L for turbine units operating with high-pressure and -temperature steam.

Deaeration in a condenser is accomplished by applying Henry's law, which states that the concentration of the dissolved gas in a solution is directly proportional to the partial pressure of that gas in the free space above the condensate level in the hot well, with the exception of those gases (e.g., $CO_2 + NH_3$) which unite chemically with the solvent. In a condenser droplets of condensate are continually scrubbed with steam, liberating the O_2 and permitting it to flow to the low-pressure air-removal section, where it is discharged to the atmosphere by the air-removal equipment.

To remove the last traces of O_2 from the condensate, an ammonia compound such as hydrazine is normally added. Free ammonia is liberated in the cycle and is either removed with the noncondensables as a gas or is condensed and retained in the condensate, depending on the detailed design of the condenser air-removal section. If the ammonia is concentrated as a liquid, it can be very corrosive to certain copper-base materials.

Most condenser manufacturers have **tube-bundle configurations** unique to their design philosophy. Basically, pressure losses from turbine exhaust to the air offtake are kept to a minimum and tubes are arranged to promote good heat-transfer rates. Small condensers are usually cylindrical, whereas large ones are rectangular for better utilization of space. Most turbines exhaust downward into the condenser, but condensers are also built to accommodate side as well as axial exhaust turbines.

Because of the inherent strength of cylindrical shapes as opposed to flat plates, condenser **water boxes** are generally made with curved surfaces. This has come about because of the increased pressure resulting from cooling towers, which in turn, are the result of environmental influences. With a cooling tower, pressures are in the 60 to 80 lb/in^2 range, whereas with water from lakes, rivers, etc., where a siphon system can be employed, water-box design pressures are in the 20 to 30 lb/in^2 range.

As a general rule, **tube selection** is based on economics; 18 BWG admiralty metal has been satisfactory for freshwater service and 90-10 copper-nickel material likewise for seawater. The current trend is to use 22 BWG titanium or one of the new specially formulated stainless-steel tube materials for this application. Material prices fluctuate greatly, and selection can be influenced by first cost. Lost revenue due to downtime caused by tube leaks or other causes, particularly in larger units, can usually justify the use of more exotic and expensive materials.

Low-pressure feedwater heaters are frequently located in the steam-inlet neck of a condenser. This is done to minimize pressure drop in the extraction steam piping and to utilize floor space surrounding the condenser better.

A sufficient number of **tube supports** must be provided within the condenser so that the tubes will not vibrate excessively, which will cause tubes to rub or crack circumferentially. During low-water-temperature operation, the steam entering the condenser will often reach sonic velocities, causing severe **flow-induced vibration** and ultimate tube failures if the tube support system is inadaquate.

Where once-through boiler or nuclear steam generators are used, it is imperative to dispose of large quantities of steam during starting and stopping of a turbine unit. The condenser, because of its large volume, has been used as a convenient dumping place for this steam. Means must be provided within the condenser to accommodate the high-energy steam without damage to condenser tubing, structural members, or the low-pressure end of the turbine.

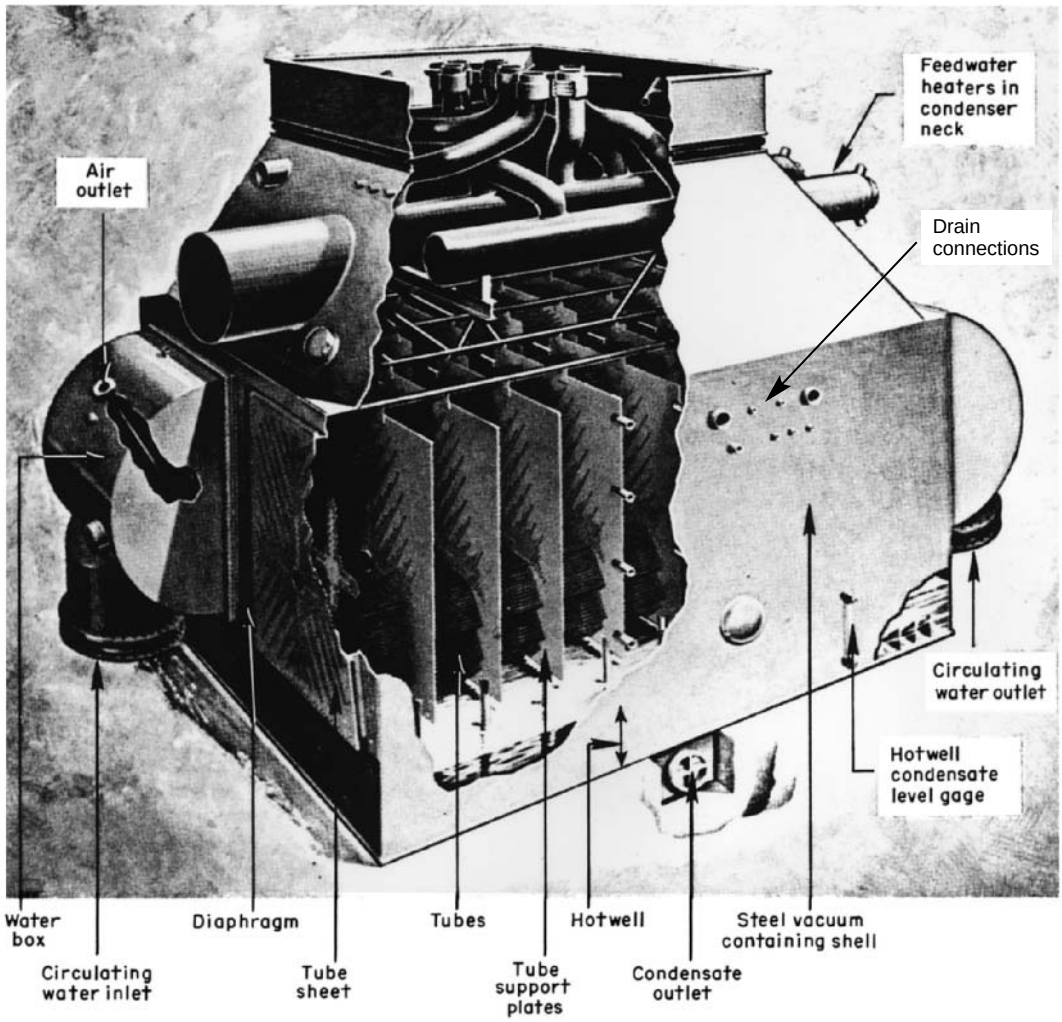


Fig. 9.5.2 One-pass rectangular surface condenser.

A major factor in condenser performance is **tube cleanliness.** Tubes can be plugged with leaves, marine life, and other debris deposited on the face of the tube sheets. By valving, various arrangements for back washing, or flushing, are effected to remove these materials.

The inside surfaces of the tubes can also become fouled by scaling, corrosion products, silt, products of biofouling such as slimes, or a combination of these. To clean the inside surfaces of the tubes, it is necessary to force brushes or plugs through them individually by the use of water or air jets. To do this, the affected portion of the condenser must be shut down. As an alternative, recirculating sponge rubber ball or brush systems are available that can be used during normal operation. To remove scale formations, sponge rubber balls with abrasive coatings are effective, but their continued use will shorten tube life.

Movement due to **temperature differences** between turbine and condenser is usually accommodated by a stainless-steel bellow or rubber-belt-type **expansion joint.** For small units the condenser may be supported on springs and rigidly connected to the turbine. To accommodate differential expansion between condenser shell and tubes, a flexible diaphragm or other expansion element may be installed.

The **tube-to-tube sheet joint** is usually rolled but has been welded in certain installations. Proper material selection must be made regardless of whether the joint is rolled or welded. Currently, considerable emphasis is placed on inward leakage of circulating water into the condenser steam space. Double tube sheets have been supplied for some large central-station condensers for nuclear units. They are standard in the Navy's nuclear-powered ships and submarines to minimize this source of contamination.

Condensers for some large turbine units having two or three low-pressure ends are designed for **multipressure application.** Usually there will be a gain, in the form of either heat-rate improvement or a reduction in capital investment, with a multipressure condenser. A rather complex analysis must be made for each application before a decision can be made.

The current version of ''Standards for Steam Surface Condensers'' is published by the Heat Exchange Institute in Cleveland, OH. These are available to the public and provide guidance in solving most structural and sizing problems related to surface condensers.

Performance Calculations and Sizing

Notation

S = condenser tube surface area, ft^2
C_c = cleanliness factor
C_1 = heat-transfer-rate constant
C_m = material and gage factor

C_t = temperature correction factor
c_p = specific heat, Btu/lb, °F
G = circulating-water quantity, gal/min
h = enthalpy, Btu/lb
h_r = heat rejected by steam, Btu/lb
L = length of water travel, ft
NPSH = net positive suction head, ft
OD = tube outside diameter, in
Q = heat transferred, Btu/h
R = temperature rise ($t_o - t_i$), °F
TDH = total dynamic head, ft
t = tube thickness, in
TTD = terminal temperature difference = $t_s - t_o$
t_i = inlet-water temperature, °F
t_o = outlet-water temperature, °F
t_s = saturation temperature in condenser, °F
U_o = overall heat-transfer rate, Btu/(ft² · h · °F)
V = water velocity, ft/s
W_s = steam to be condensed, lb/h
Δt_m = logarithmic mean temperature difference, °F

In **sizing** a condenser, the steam flow and heat rejected to the condenser are obtained from the turbine heat balance. Table 9.5.1 gives representative steam flows; heat rejected to the condenser is approximately 950 Btu/lb of steam for nonreheat turbines and 975 Btu/lb for reheat machines. Figure 9.5.3 shows approximate average water temperatures for the United States; local water temperature should be used, when known. The number of passes is usually dictated by the plant arrangement, with total length of water travel and tube diameter dictated by economic considerations. Normally, small-diameter tubes, single-pass condensers are used where water is plentiful, and larger-diameter tubes, two-pass condensers when water is scarce. The vacuum, or back pressure, is determined by economic evaluation, but Table 9.5.2 gives normal recommended values for average water temperatures. Table 9.5.3 is a pressure-temperature conversion table.

Table 9.5.1 Steam Flow to Condensers*

Turbine throttle conditions	Weight flow, lb/kWh
600 lb/in², 825°F	7.08
850 lb/in², 900°F	6.45
1,250 lb/in², 950°F	5.80
1,450 lb/in², 1,000°F	5.44
1,450 lb/in², 1,000°F, 1,000°F	4.70

* Approximate values for unit sizes to 100,000 kW and exhaust at 1.0 inHg abs.

NOTE: For exhaust pressures other than 1.0 inHg abs, multiply tabular values by 1.02 (1.04)[1.06] for 1.5 (2.0)[2.5] in.

Table 9.5.2 Normal Condenser Pressures and Circulating-Water Temperatures

Water inlet temp t_i, °F	Normal back pressure, inHg abs
55	1.0
70	1.5
80	2.0
85	2.5
90	3.0
95	3.5

A **cleanliness factor** is applied to the heat-transfer rate of new, clean tubes to allow for gradual decrease by fouling. A standarized cleanliness factor of 85 percent is frequently used, but this can often be misleading and even erroneous. The fouling is attributable to (1) sedimentation, (2) scaling, (3) steam-side deposits, (4) corrosion, and (5) biological growth. The fouling is correctly determined by use; in some cases the cleanliness factor may be 90 percent, whereas in other cases it may never rise above 75 percent.

Velocities normally used are: for clean water, 7.0 ft/s; for very clean water with cooling towers, 8.0 ft/s; and for seawater, with entrained sand, as low as 6.0 ft/s to minimize erosion. Prevalent velocities are 6.5 ft/s with aluminum-brass tubes, 7.0 ft/s with admiralty metal, and

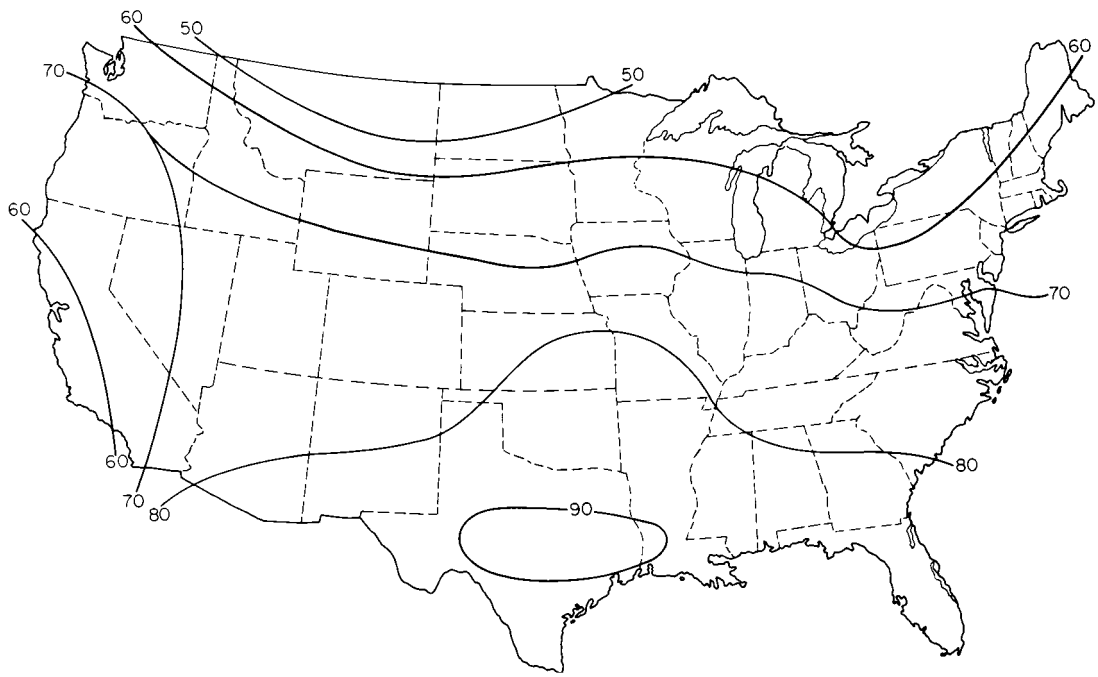


Fig. 9.5.3 Average inlet temperature of circulating water, °F, United States.

Table 9.5.3 Pressure-Temperature Conversion Table for Water*

Abs press, inHg	Sat temp, °F	Abs press, inHg	Sat temp, °F	Abs press, inHg	Sat temp, °F	Abs press, inHg	Sat temp, °F
0.20	34.57	1.35	88.36	2.50	108.71	8.00	152.24
0.25	40.23	1.40	89.51	2.55	109.38	9.00	157.09
0.30	44.96	1.45	90.64	2.60	110.06	10.00	161.49
0.35	49.06	1.50	91.72	2.65	110.72	11.00	165.54
0.40	52.64	1.55	92.77	2.70	111.37	12.00	169.28
0.45	55.89	1.60	93.81	2.75	112.01	13.00	172.78
0.50	58.80	1.65	94.80	2.80	112.63	14.00	176.05
0.55	61.48	1.70	95.78	2.85	113.25	15.00	179.14
0.60	63.96	1.75	96.73	2.90	113.86	16.00	182.05
0.65	66.26	1.80	97.65	2.95	114.45	17.00	184.82
0.70	68.41	1.85	98.56	3.00	115.06	18.00	187.45
0.75	70.43	1.90	99.43	3.20	117.35	19.00	189.96
0.80	72.32	1.95	100.30	3.40	119.51	20.00	192.37
0.85	74.13	2.00	101.14	3.60	121.57	21.00	194.68
0.90	75.84	2.05	101.96	3.80	123.53	22.00	196.90
0.95	77.48	2.10	102.77	4.00	125.43	23.00	199.03
1.00	79.03	2.15	103.56	4.20	127.21	24.00	201.09
1.05	80.53	2.20	104.33	4.40	128.94	25.00	203.08
1.10	81.96	2.25	105.09	4.60	130.61	26.00	205.00
1.15	83.33	2.30	105.85	4.80	132.20	27.00	206.87
1.20	84.64	2.35	106.58	5.00	133.76	28.00	208.67
1.25	85.93	2.40	107.30	6.00	140.78	29.00	210.43
1.30	87.17	2.45	108.01	7.00	146.86	29.921	212.00

* See also Sec. 4 for further data.

8 + ft/s with stainless steel or titanium. **Water-temperature rise** is about 10°F for single-pass condensers and 15°F for two-pass condensers, with a minimum 5° terminal temperature difference (TTD).

Approximate general rules for condensers serving turbines rated up to 100 MW. The **surface area,** ft², is equal to the steam flow, lb/h, divided by 10 for single-pass condensers and by 7.5 for two-pass condensers. **Circulating-water quantity,** gal/min, is equal to the area, ft², for a two-pass condenser and is twice the area for a single-pass condenser. Condenser **proportions** are given in Table 9.5.4. Empty **weight** of an installed condenser is 5 to 6 lb/ft² of surface.

Table 9.5.4 Typical Small Condenser Proportions

Surface area, ft²	Effective tube lengths, ft		
	¾-in OD	⅞-in OD	1-in OD
1,000–1,750	8, 10, 12, 14		
2,000–2,500	10, 12, 14, 16		
2,750–4,750	12, 14, 16, 18	12, 14, 16, 18	
5,000–7,000	14, 16, 18, 20	14, 16, 18, 20	14, 16, 18, 20
7,250–14,000	16, 18, 20, 22	16, 18, 20, 22	16, 18, 20, 22
15,000–19,000		18, 20, 22, 24	18, 20, 22, 24
20,000–27,500		20, 22, 24, 26	20, 22, 24, 26
30,000–47,500		22, 24, 26, 28	22, 24, 26, 28
50,000–75,000		24, 26, 28, 30	24, 26, 28, 30

Condenser Calculations

It is normally a tedious process to calculate the size and performance of a condenser directly by use of the logarithmic mean temperature difference. But using the following formula, the sizing can readily be determined by iteration, and performance can be found directly.

$$t_s = \frac{R}{\ln^{-1}[(S/G)(U_o/500)] - 1} + t_o \qquad (9.5.1)$$

Note that the specific heat c_p is normally neglected in condenser work, it being considered insignificant.

The basic diagram is shown in Fig. 9.5.4, and the applicable equations are

$$Q = U_o S\,\Delta t_m \qquad (9.5.2)$$

$$Q = 500 G c_p (t_o - t_i) \qquad (9.5.3)$$

$$R = W_s h_r/(500G) \qquad (9.5.4)$$

$$U_o = C_1\sqrt{V}C_t C_m C_c \qquad (9.5.5)$$

EXAMPLE. Calculate the surface and circulating-water requirements to condense 445,000 lb of steam per h to an absolute pressure of 2.00 inHg abs. The heat rejected is 980 Btu/lb of steam and is absorbed by circulating seawater at an average inlet temperature of 75°F. The condenser is to be two-pass with ⅞-in 18-gage 26-ft-active-length aluminum-brass tubes; cleanliness factor = 85 percent.

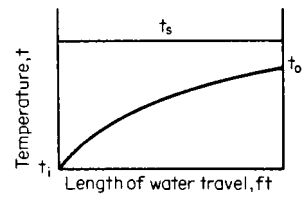


Fig. 9.5.4 Temperature vs. water travel, surface condenser.

SOLUTION. The heat transfer rate U_o is a function of the tube diameter, material, gage, water velocity, and cleanliness factors within the velocity limits of 3 and 10 ft/s. For ⅝- and ¾-in tubes $C_1 = 267$; for ⅞- and 1-in tubes $C_1 = 263$, and for 1⅛- and 1¼-in tubes $C_1 = 259$. The inlet-water-temperature correction factor is obtained from Table 9.5.7. The material and gage correction factor C_m is obtained from Table 9.5.6, and the cleanliness factor C_c is given in the conditions of the example.

Assume a surface of 59,000 ft² based on the typical steam loading for such a unit and a water velocity of 6.5 ft/s. For the selected surface and water velocity, by using the information in Table 9.5.5, the number of gallons per minute required by this selection is determined. It is now possible to determine the steam temperature that this selection would produce, by using Eq. (9.5.1).

The results show a steam temperature of 99°F, indicating that a smaller surface

is required to attain a 2-inHg abs condenser pressure. Try a smaller unit until a surface with no more than three significant figures will yield a steam temperature of no more than 101.14°F. The surface arrived at is 54,400 ft² with 43,900 gal/min of cooling water.

Table 9.5.5 Tube Characteristics

OD and ext. surface, ft²/ft	BWG no.	ID, in	Water, gal/min @ 1 ft/s velocity
	16	0.495	0.600
⅝-in OD	18	0.527	0.680
0.1636	20	0.555	0.754
	22	0.569	0.793
	16	0.620	0.941
¾-in OD	18	0.652	1.041
0.1963	20	0.680	1.132
	22	0.694	1.179
	16	0.745	1.359
⅞-in OD	18	0.777	1.478
0.2291	20	0.805	1.586
	22	0.819	1.642
	16	0.870	1.853
1-in OD	18	0.902	1.992
0.2618	20	0.930	2.117
	22	0.944	2.182

Table 9.5.6 Material and Gage Factor

	C_m						
	BWG no.						
Tube material	24	22	20	18	16	14	12
Admiralty metal	1.03	1.02	1.01	1.00	0.98	0.96	0.93
Arsenical copper	1.04	1.03	1.03	1.02	1.01	1.00	0.98
Copper iron 194	1.04	1.04	1.03	1.03	1.02	1.01	1.00
Aluminum brass	1.02	1.02	1.01	0.99	0.97	0.95	0.92
Aluminum bronze	1.02	1.01	1.00	0.98	0.96	0.93	0.89
90–10 Cu-Ni	0.99	0.98	0.96	0.93	0.89	0.85	0.80
70–30 Cu-Ni	0.97	0.95	0.92	0.88	0.83	0.78	0.71
Cold-rolled carbon steel	1.00	0.98	0.97	0.93	0.89	0.85	0.80
Stainless steel type 304/316	0.90	0.86	0.82	0.75	0.69	0.62	0.54
Titanium	0.94	0.91	0.88	0.82	0.77	0.71	0.63

Table 9.5.7 Inlet-Water-Temperature Correction Factors

Inlet temp t_i, °F	Correction factor C_t
40	0.740
50	0.830
60	0.920
70	1.000
80	1.045
90	1.075
100	1.100

If this condenser is single-shell with one main turbine exhaust inlet and has no auxiliary turbine exhausts entering it (such as from a turbine-driven boiler feed pump), then the air-removal equipment must have a minimum capacity of 10 scfm of air at 1-inHg abs suction pressure. (See Table 9.5.8.)

Performance curves (Fig. 9.5.5) are drawn for a condenser to show the back pressure for various condenser steam loads and inlet-water temperatures maintaining a minimum TTD of 5°. The zero-load back pressure and the cutoff pressure are shown in Fig. 9.5.6, where the cutoff

Table 9.5.8 Venting-Equipment Capacities*

One condenser shell			
Effective steam flow each main exhaust opening, lb/h	Total number of exhaust openings	1	2
Up to 25,000	scfm†	3.0	4.0
	Dry air, lb/h	13.5	18.0
	Water vapor, lb/h	29.7	39.6
	Total mixture, lb/h	43.2	57.6
25,001–50,000	scfm†	4.0	5.0
	Dry air, lb/h	18.0	22.5
	Water vapor, lb/h	39.6	49.5
	Total mixture, lb/h	57.6	72.0
50,001–100,000	scfm†	5.0	7.5
	Dry air, lb/h	22.5	33.8
	Water vapor, lb/h	49.5	74.4
	Total mixture, lb/h	72.0	108.2
100,001–250,000	scfm†	7.5	12.5
	Dry air, lb/h	33.8	56.2
	Water vapor, lb/h	74.4	123.6
	Total mixture, lb/h	108.2	179.8
250,001–500,000	scfm†	10.0	15.0
	Dry air, lb/h	45.0	67.5
	Water vapor, lb/h	99.0	148.5
	Total mixture, lb/h	144.0	216.0
500,001–1,000,000	scfm†	12.5	20.0
	Dry air, lb/h	56.2	90.0
	Water vapor, lb/h	123.6	198.0
	Total mixture, lb/h	179.8	288.0
1,000,001–2,000,000	scfm†	15.0	25.0
	Dry air, lb/h	67.5	112.5
	Water vapor, lb/h	148.5	247.5
	Total mixture, lb/h	216.0	360.0

* Heat Exchange Institute.
† 14.7 lb/in² abs at 70°F.
NOTE: These tables are based on air leakage only and the air-vapor mixture at 1 inHg abs and 71.5°F.

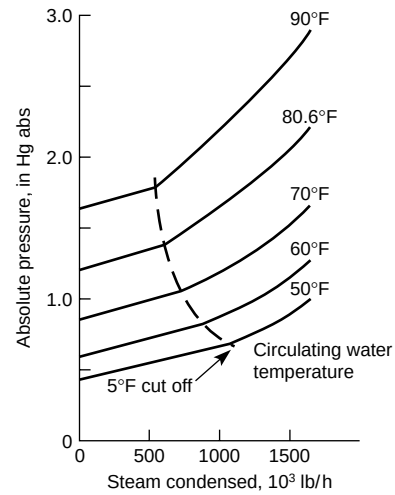


Fig. 9.5.5 Representative performance curves for a 157,500-ft² one-pass surface condenser, 1-in, 18-gage, 37.8-ft-active-length aluminum-brass tubes. Performance is based on 85 percent clean tubes, 221,400 gal/min cooling water at a velocity of 7 ft/s, and 968 Btu/lb rejected to the cooling water.

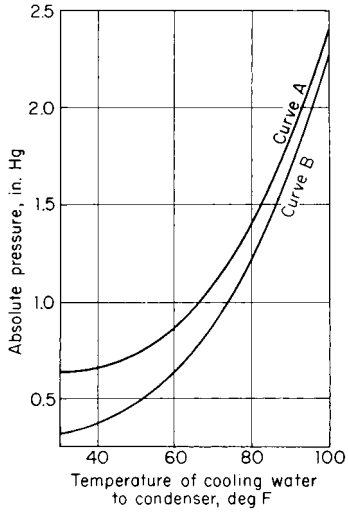


Fig. 9.5.6 Cutoff and zero load vacuums. Curve *A*, cutoff (except where the absolute pressure is limited to 5°F TTD); curve *B*, zero load.

limitation is set by the air-removal equipment. A combined turbine-condenser performance curve is sometimes drawn in which heat rejected vs. back pressure for a given turbine load is superimposed on the condenser performance.

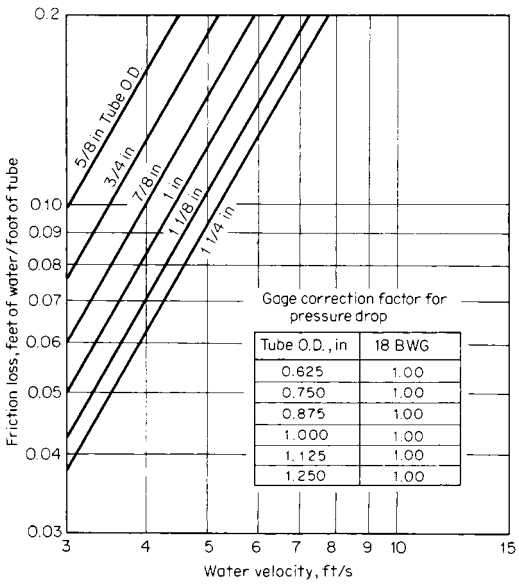


Fig. 9.5.7 Friction loss for water flowing in 18 BWG tubes, low velocity. *(Heat Exchange Institute.)*

Circulating-water pumps are usually half capacity, with 3 or 4 percent additional allowance on each pump for miscellaneous heat exchangers. The total dynamic head (TDH) is the sum of the condenser and water-box friction (Figs. 9.5.7 to 9.5.11), the pipe loss, and any unrecovered static head. Normal heads are about 20 ft; for cooling-tower installations, 60 ft. (See also Sec. 14 and Fig. 9.5.12.)

Condensate pumps are usually full capacity, determined by condenser flow plus any heater drains dumped into the condenser and 5 percent additional margin. Vertical-pit-type designs are commonly used because of low net-positive-suction-head (NPSH) requirements with water at the boiling point. TDH ranges from a normal value of 100 ft on small plants to 800 ft on larger plants. (See Sec. 14.)

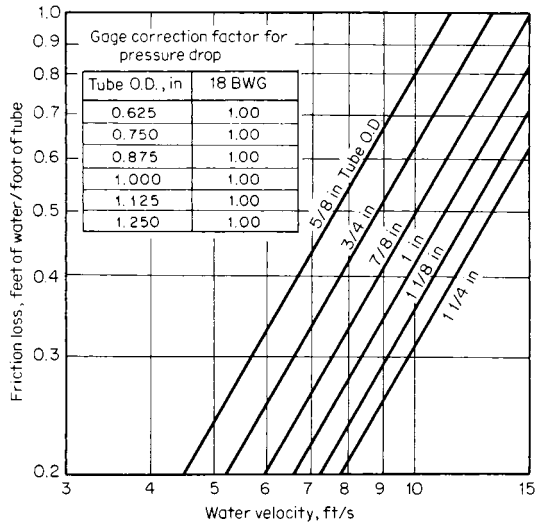


Fig. 9.5.8 Friction loss for water flowing in 18 BWG tubes, high velocity. *(Heat Exchange Institute.)*

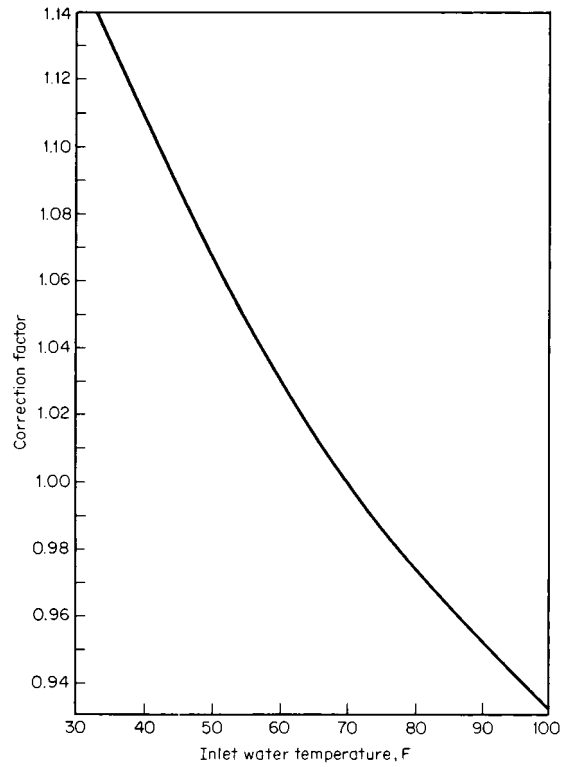


Fig. 9.5.9 Temperature correction for friction losses in tubes (see Figs. 9.5.7 and 9.5.8). *(Heat Exchange Institute.)*

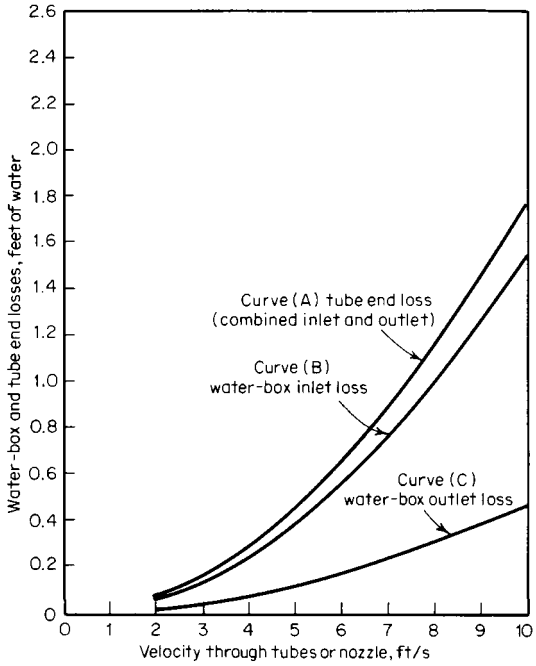


Fig. 9.5.10 Water-box and tube end losses, single-pass condensers. *(Heat Exchange Institute.)*

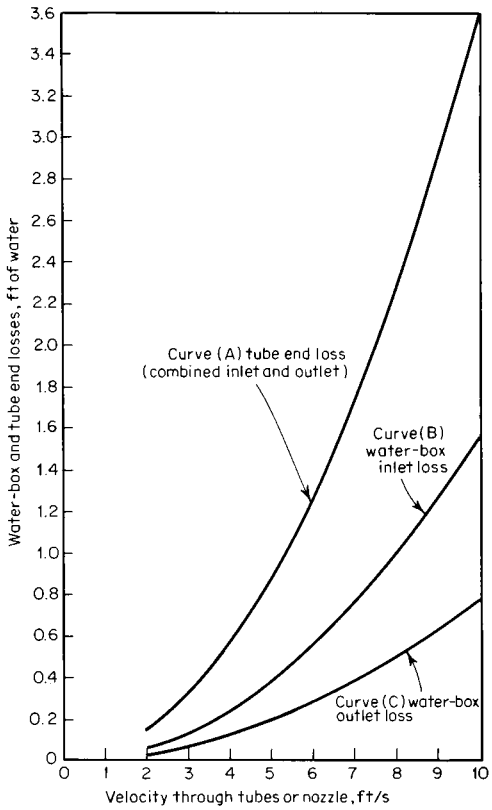


Fig. 9.5.11 Water-box and tube end losses, two-pass condensers. *(Heat Exchange Institute.)*

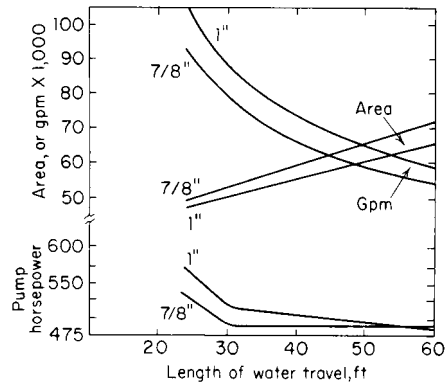


Fig. 9.5.12 Condenser configuration selection for a given heat load, W_s = 477,355 lb/h; h = 975 Btu/lb; t_i = 70.0°F; 1.5 inHg abs; 18-gage admiralty metal tubes; 7.0-ft/s water velocity; 85 percent cleanliness factor.

AIR-COOLED CONDENSERS

The air-cooled condenser is used where adaquate water is not available in sufficient quantities to permit the use of conventional cooling towers. The water lost by evaporation and cooling tower blow-down is approximately equal to the steam condensed from the turbine exhaust.

In an air-cooled condenser, the steam is condensed inside tubes while cooling air flows over fins on the external surfaces. These tubes are arranged in the area where packing or fill would be arranged in a conventional cooling tower, and propeller fans (see Sec. 14) supply air across the tube surfaces.

Aluminum is generally the material of choice for tubing because of its excellent heat transfer properties and availability. Items to be considered in design are the overall heat-transfer coefficient, steam-side pressure drop, uniformity of steam distribution, noncondensible gas concentration for air removal, and, of course, provision for freeze protection. Turbine design back pressures are normally in the range of 5 in Hga when served by air-cooled condensers.

DIRECT-CONTACT CONDENSERS

REFERENCES: Heat Exchange Institute Standards for Direct Contact and Low Level Condensers.

When (1) low investment is desired and (2) condensate recovery is not a factor, direct-contact condensers are effective. They are relatively simple to build and operate, are limited to sizes less than 250,000 lb of steam per h, and are built in three types: (1) barometric, (2) low-level, and (3) jet.

Figure 9.5.13 shows a self-supported counterflow **barometric condenser** with tail pipe, hot well, and air ejector. Steam and cooling water flow in opposite directions, with the coldest water for final condensation and cooling of noncondensables. The air pump must handle that part of the air disengaged from the cooling water as well as air leakage. The head required to pump water is pipe friction plus static head, minus 75 percent approx of the design vacuum. The barometric condenser is usually placed outdoors, and the water leg must be more than 34 ft high.

The **low-level condenser** substitutes a pump for the water leg of the barometric condenser to remove the liquid from the vacuum space. The **jet condenser** utilizes the aspirating effect of a jet for the entrainment of noncondensables and the consequent elimination of a separate air pump.

In the usual direct-contact condenser, where steam and raw circulating water are mixed, the recovery of pure condensate is precluded; greater feedwater makeup is necessary, and poorer vacuums are attained than with surface condensers. Direct-contact-condenser installations are not found in large plants, but there is some recent interest in their use with dry cooling towers.

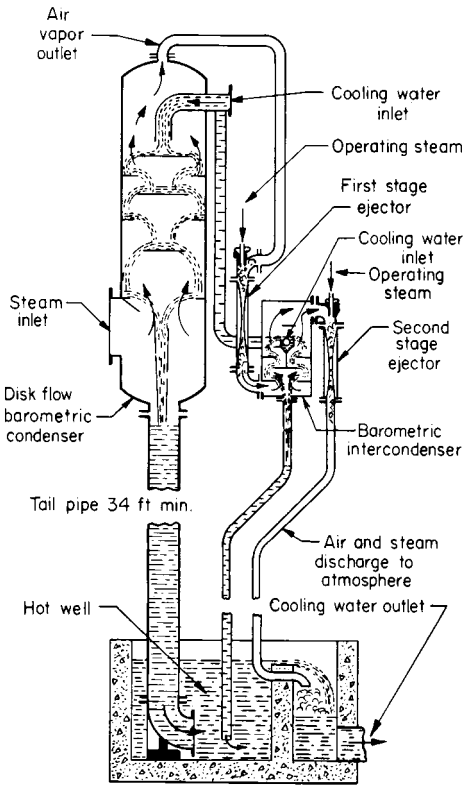


Fig. 9.5.13 Counterflow barometric condenser with two-stage air ejector. *(Ingersoll-Rand Co.)*

Table 9.5.9 gives values of expected air content of cooling waters; Fig. 9.5.14 gives typical performance curves; Fig. 9.5.15 defines flows W, temperatures t, and enthalpies h needed for the basic heat-balance equation $W_s(h_s - h_2) = W_w(t_2 - t_1)$. The latent heat $(h_s - h_2)$ is frequently taken as 950 Btu/lb of steam.

Shell materials are usually steel plate; with dirty or corrosive water, bronze, stainless steel, or linings of ceramic, plastic, or rubber may be used.

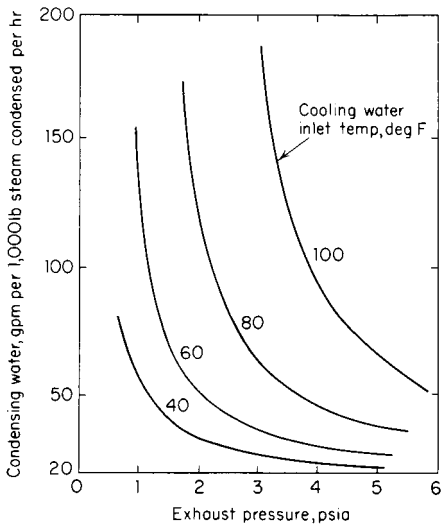


Fig. 9.5.14 Representative performance curves for a direct-contact condenser.

Table 9.5.9 Air in Cooling Water

Water temp, °F	Air, ft³/min per 1,000 gal/min
35	4.0
40	3.78
50	3.35
60	2.97
70	2.68
80	2.41
90	2.21
100	2.0

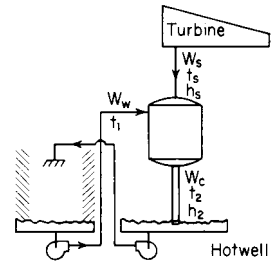


Fig. 9.5.15 Direct-contact condenser performance terms.

AIR EJECTORS

REFERENCE: Heat Exchange Institute, Standards for Steam Jet Ejectors.

The steam-jet ejector is used to remove noncondensable air and gases from condensers. It consists of a suction chamber, diffuser, and steam nozzle. The high-velocity jet of steam issuing from the nozzle entrains the gas, and the kinetic energy of the mixture is converted to pressure energy in the diffuser. Ratings are in lb/h at a given suction pressure. Ejectors are operated in series or are staged for absolute suction pressures of 1 ± inHg abs. They will handle wet or dry mixtures, they are easy to operate, installation costs are low, and there are no moving parts—with consequent long life, high sustained efficiency, and low maintenance.

Two-stage ejectors, with surface-type inter- and aftercondensers (Fig. 9.5.16), are common in steam-surface-condenser application. The main condensate serves as a coolant, returning heat regeneratively to the boiler feed system. Two-stage ejectors have a shutoff pressure of 0.5 inHg abs approx. Two elements are usually provided, one serving as a spare.

Computation of **ejector performance** requires application of Dalton's law of mixtures (see Sec. 4), where basically the total pressure is the sum of the condensable vapor (steam) pressure and the noncondensable gas (air) pressure. In power plant condensers, the mixture is saturated with steam so that the temperature of the mixture fixes the partial pressure of the vapor.

Air ejectors are usually rated with suction conditions of 1 inHg abs and 7.5°F subcooling. Since the saturation temperature of steam at 1 inHg abs is 79°F, the mixture entering the ejector is at 71.5°F. The partial pressure of the vapor at 71.5°F is 0.78 inHg abs, and the partial pressure of the air is $1 - 0.78 = 0.22$ inHg abs. Therefore, an ejector with a standard rating (see Table 9.5.8) of 12.5 ft³/min of free dry air at 30 inHg abs handles, by the gas laws, $12.5(30/0.22) = 1{,}704$ ft³/min at the ejector suction.

Motive steam to the ejector is usually supplied from turbine throttle conditions, e.g., 600 lb/in², 750°F; 1,500 lb/in², 1,000°F; 2,500 lb/in², 1,050°F, through a reducing valve to a pressure not higher than 600 lb/in².

The **inter- and aftercondenser,** one- or two-pass, is a small heat exchanger which condenses the motive steam and allows the noncondensable gases, such as O_2 and CO_2 to be expelled to the atmosphere. How-

ever, the highly soluble ammonia gas may be returned to the feedwater system. The water-box pressure of tube-side pressure must be designed for the condensate-pump shutoff head. The pressure drop through the inter- and aftercondensers varies from 1 to 10 ft of water, with 3 ft as a reasonable average.

Normal **materials** used are: nozzle, stainless steel type 303; steam chest, steel; inlet-air chamber, cast steel; diffuser, aluminum bronze; inter- and aftercondenser shell, steel; inter- and aftercondenser tubes, stainless steel type 304; inter- and after-condenser tube sheet, carbon steel.

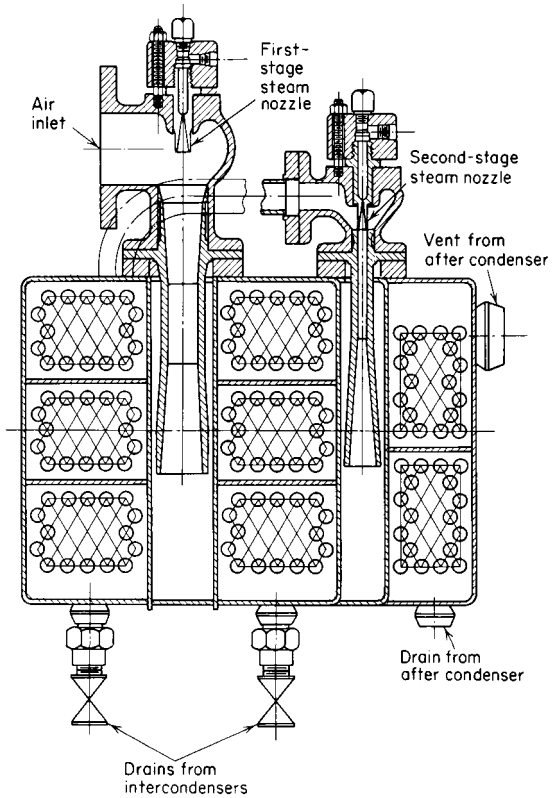


Fig. 9.5.16 Two-stage air ejector with surface-type inter- and aftercondensers. *(Westinghouse Electric Corp.)*

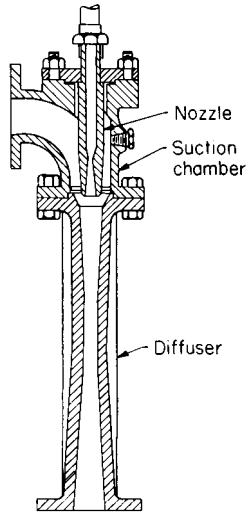


Fig. 9.5.17 Single-stage air ejector.

Hogging and Priming Ejectors Single-stage steam-jet ejectors (Fig. 9.5.17) are used for evacuating, or **hogging,** the air from the steam side of a surface condenser, in 15 or 20 min time, so that steam flow can be established and the condenser brought on the line. These ejectors are usually noncondensing and exhaust to the atmosphere. Ratings (Table 9.5.10) are typically in lb/h of free dry air at 70°F, suction conditions of 15 inHg abs, shutoff pressure about 2 inHg abs, motive steam pressures and temperatures as with multistage ejectors.

Priming ejectors are also single-stage noncondensing units and are used to withdraw air from the water side and to fill the condenser with circulating water during start-up periods. The hogging ejector may also serve this priming function by a suitable system of piping and valves.

Materials normally employed are: for the nozzle, stainless steel, 18-8 chrome nickel; steam chest, steel; air-inlet chamber and diffuser, cast steel.

Table 9.5.10 Hogger Capacities*

Total steam condensed, lb/h	scfm†—dry air at 10 inHg abs design suction pressure
Up to 100,000	50
100,001–250,000	100
250,001–500,000	200
500,001–1,000,000	350
1,000,001–2,000,000	700
2,000,001–3,000,000	1,050
3,000,001–4,000,000	1,400
4,000,001–5,000,000	1,750
5,000,001–6,000,000	2,100
6,000,001–7,000,000	2,450
7,000,001–8,000,000	2,800
8,000,001–9,000,000	3,150
9,000,001–10,000,000	3,500

* Heat Exchange Institute.
† 14.7 lb/in^2 abs at 70°F.
NOTE: In the range of 500,000 lb/h steam condensed and above, the table provides evacuation of the air in the condenser and low-pressure turbine from atmospheric pressure to 10 inHg abs in about 30 min if the volume of condenser and low-pressure turbine is assumed to be 26 ft^3/(1,000 lb/h) of steam condensed.

VACUUM PUMPS

REFERENCE: Woodman, Rotary or Steam-Jet Air Pumps, *Power,* Aug. 1948.

Mechanical vacuum pumps, used for hogging, holding, and priming, are of several types for power-plant service: (1) reciprocating—piston, diaphragm; (2) rotary—sliding-vane, oval-water- seal, eccentric-rotor. (See Sec. 14 for details.) Multistage pumps usually need intercooling. A silencer is generally provided on the air discharge. Normally, two half-capacity pumps are installed, both operating during hogging and one adequate for holding service. The relative capabilities of a vacuum pump and hogging and holding ejectors are illustrated in Fig. 9.5.18 (note the considerable hogging capacity of the vacuum pump on start-up).

Advantages of vacuum pumps compared to steam-jet air ejectors include (1) system independent of a steam supply; (2) start-up when steam is not available, as on a once-through boiler system; (3) system capable of completely automatic operation; (4) high-pressure steam piping eliminated; (5) recycling of noncondensables and ammonia eliminated; (6) quieter operation. Disadvantages include (1) damage by water entering the intake (except when pumps having a liquid rotating seal); (2) higher initial cost; (3) higher maintenance cost. Operating costs are approximately equal. Various combination arrangements are found in practice, e.g., an air-operated air-ejector first stage followed by

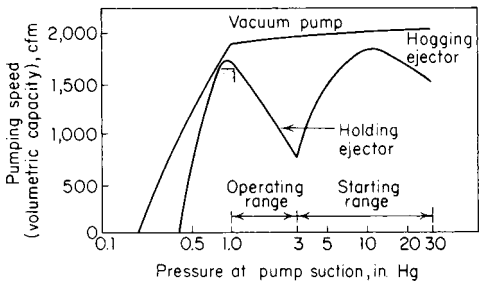


Fig. 9.5.18 Relative capabilities of air-removal pumps.

a vacuum-pump second stage, with consequent savings in motor-horsepower and space requirements.

COOLING TOWERS

REFERENCES: ''Evaluated Weather Data for Cooling Equipment Design,'' Fluor Products Co. Cooling Towers, *Power*, Mar. 1963. Baker and Shryock, A Comprehensive Approach to the Analysis of Cooling Tower Performance, *Trans. ASME*, 1961.

The prevalent type of cooling tower (see Figs. 9.5.19 and 9.5.22) dissipates heat by the evaporation of some of the water sprayed into the air circulated through the tower. It is used where water is in limited supply, where temperature pollution of natural water bodies is to be avoided, where water conservation is to be effected, or where otherwise polluted sources must be avoided. Figure 9.5.20 illustrates the functional cycle and the significant basic terms (see also Sec. 4).

Wet-bulb temperature, for design purposes, should not exceed the maximum expected value more than 5 percent of the time during summer (June to September). (See Fig. 9.5.21.)

Approach is the difference in temperature between the cold water leaving the tower and the ambient wet bulb.

Cooling range is the difference in temperature between the hot water entering and the cold water leaving the tower.

Drift is the water lost as mist or droplets entrained by the circulating air and discharged to the atmosphere. It is in addition to the evaporative loss and is minimized by good design.

Makeup is the water required to replace total losses by evaporation, drift, blowdown, and small leaks.

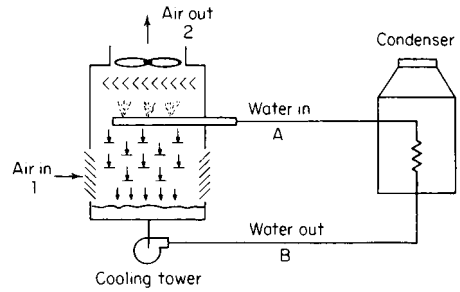


Fig. 9.5.19 Schematic for induced-draft, counterflow, cooling tower installation.

The early, simple atmospheric towers have, because of reliance on natural air circulation, high pumping heads, excessive spray losses, and makeup, been largely superseded by three important types: (1) forced-draft, (2) induced-draft, and (3) hyperbolic (Fig. 9.5.22).

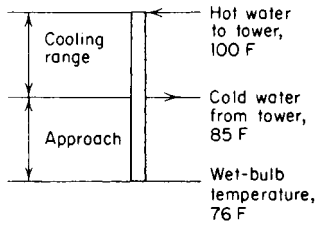


Fig. 9.5.20 Cooling tower terms.

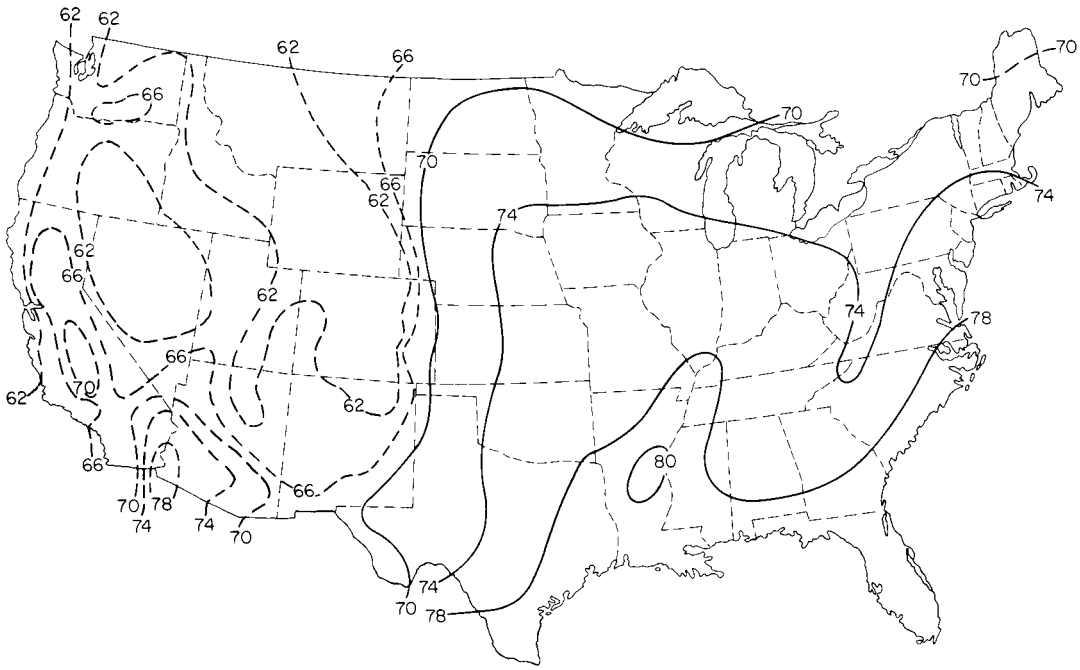


Fig. 9.5.21 Wet-bulb temperature, °F, isoclines at the 5 percent level, United States. *(Adapted from Fluor Products Co., by permission.)*

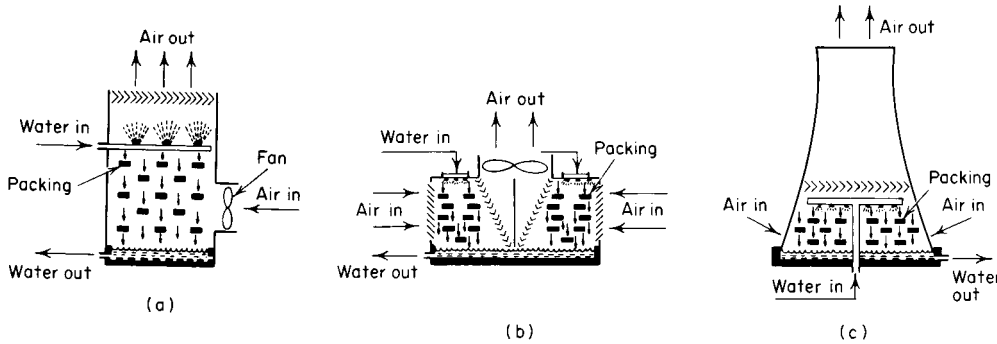


Fig. 9.5.22 Cooling tower types. (*a*) Forced draft; (*b*) crossflow-induced draft; (*c*) hyperbolic.

The **mechanical forced-draft tower** (Fig. 9.5.22*a*) gives (1) a controllable air supply from fans conveniently located for inspection and maintenance at ground level, and (2) reduced water- pumping head. Nonuniform distribution of air over the ground area of the tower cell, recirculation of vapor from the tower discharge to the tower inlet, with its deleterious effects and fan icing in cold weather, and limitations on the physical diameter of fans are all problems with forced-draft towers.

The **induced-draft tower** (Fig. 9.5.22*b*) is prevalent in U.S. practice. The fan is mounted on the top (discharge) of the cell, with consequent improved air distribution within the cell; drift eliminators reduce makeup requirements; spray nozzles, downspouts, splash plates, and splash bars ensure ample evaporative surface for the water, with maximum volumetric heat-transfer rates. In the **counterflow** design, air is introduced beneath the cell fill, but in the **crossflow** design (Fig. 9.5.22*b*), air is introduced at the sides of the fill.

The **hyperbolic tower** (Fig. 9.5.22*c*) utilizes the chimney effect (height, 300 ± ft) for natural circulation. It has been favored in large European installations where the prevalent lower dry-bulb ambient temperature gives a greater difference in density between entrance to and exit from the tower. The wide approach keeps the unit size within practical bounds, and the savings in fan power support the higher investment. Recent installations in the United States attest to its attractiveness.

Precautions are necessary to avoid freezing troubles in cold weather, fire hazards with intermittent operations, corrosion, scale, and microbiological growth problems. **Location** must avoid recirculation: for tower lengths to 250 ft, the long-axis orientation should be parallel to the prevailing wind; towers longer than 250 ft should be arranged broadside to the prevailing wind. The tower should be isolated as much as possible; adjacent heat sources compel the specification of higher design wet-bulb temperatures.

Performance Calculations

(See Fig. 9.5.19.)

Applicable equations are

$$W_{a1}h_{a1} + W_{v1}h_{v1} + W_{wA}h_{fA} = W_{a2}h_{a2} + W_{v2}h_{v2} + W_{wB}h_{fB} \quad (9.5.6)$$

$$W_{wB} = W_{wA} - (W_{v2} - W_{v1}) \qquad W_{a1} - W_{a2} \quad (9.5.7)$$

$$W_{wA}(h_{fA} - h_{fB}) = W_{a2}h_{a2} + W_{v2}h_{v2} - (W_{a1}h_{a1} + W_{v1}h_{v1}) - (W_{v2} - W_{v1})h_{fB} \quad (9.5.8)$$

$$h_{fA} - h_{fB} = t_{wA} - t_{wB} \quad (9.5.9)$$

and

$$W_a h_a + W_v h_v = \text{total heat, from psychrometric chart} \quad \text{(See Sec. 4.)} \quad (9.5.10)$$

$$W_{wA}(t_{wA} - t_{wB}) = \text{total heat at 2} - \text{total heat at 1} - (W_{v2} - W_{v1})h_{fB} \quad (9.5.11)$$

where W = flow, lb/h; a = air; w = water; v = vapor; h = enthalpy, Btu/lb; f = fluid; t = temperature, °F; 1, 2, A, B = locations.

EXAMPLE. Given: flow = 80,000; wet bulb = 76°F; dry bulb = 86°F; range (also condenser rise) = 100 − 85 = 15°F; approach = 9°F. Find: economical tower size, circulating-pump power, fan capacity and power, and makeup water.

SOLUTION. (1) *Economical tower size:* A study must be made to tie in with condenser. Heat to be dissipated is constant. Vary rise (range), approach, and then compare capital costs of cooling tower, circulating pump, and condenser with operating costs for fan and pump power. (2) *Circulating-water-pump power;* bhp = gal/min × TDH/3,960 × eff. TDH = condenser friction + pipe friction + static head. (3) *Fan capacity and power:* If air leaves tower saturated at 95°F, then from psychrometric charts (Sec. 4),

$$\text{Total heat} = 1 \text{ lb dry air } C_p(t_a - 0°) + W_v h_g \qquad \text{at } t_a \quad (9.5.12)$$

For air and vapor at 1, vapor = 120 grains/lb dry air. By Eqs. (9.5.10) and (9.5.12), total heat = 1 × 0.24(86 − 0) + (120/7,000)1,099.2 = 39.5 Btu/lb dry air. For air and vapor at 2, vapor = 256 grains/lb dry air. Total heat = 63.1 per lb dry air. Sp vol (air and vapor) = 14.80 ft³/lb dry air. By Eq. (9.5.11), $W_{wA}(100 - 85) = 63.1 - 39.5 - (256 - 120)\ {}^{63}/_{7,000} = 1.50$ lb water/lb dry air. 80,000 gal/min/(1.50 × 500) = 26.6 × 10⁶ lb dry air/h, or 26.6 × 10⁶ × 14.80/60 = 6.58 × 10⁶ ft³/min. With test data on static-pressure requirements for the tower and with the fan characteristics, the horsepower to drive the fan can be calculated. (4) *Makeup:* $W_{v2} - W_{v1}$ = (256 − 120) 26.6 × 10⁶/7,000 = 515,000 lb/h.

Normally, a cooling tower is purchased for only one guarantee point. It is well, however, to have **performance** curves (Fig. 9.5.23) showing operation for various wet-bulb temperatures and cooling ranges. The **investment** for a cooling tower is essentially a matter of water flow and is

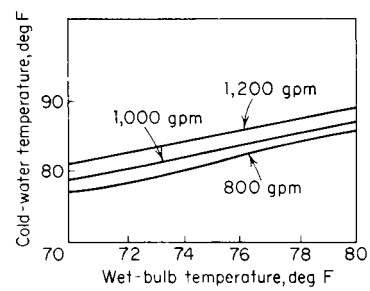


Fig. 9.5.23 Cooling tower performance. Design for 85°F cold water, 95°F hot water, 78°F wet-bulb temperature, 10° range; 1,000-gal/min cell.

influenced by approach, range, and wet bulb (Fig. 9.5.24). The **cost evaluation** should include consideration of tower frame and fill, fans, motors, basin and pump pit, pump head, fan horsepower, freight, labor, and erection. In the choice of a tower for a power plant, there should be coordinated study and evaluation of the turbine and condenser for best overall economy.

The **height** of a field-erected induced-draft tower, from basin curb to fan deck, ranges from 8 to 50 ft; widths vary from 6 to 60 ft; lengths from 8 to 500 ft; fan-stack height between 2 and 15 ft.

Materials used are: frame—redwood (treated or untreated), Douglas fir (treated), steel (galvanized), concrete; fill—red wood, plastics (polyethylene, polypropylene), cement asbestos board (extruded);

casing—cement asbestos board (corrugated, slat), redwood, fiber glass, aluminum, concrete; fan blades—aluminum, glass-reinforced polyester, stainless steel, monel; fan hubs—cast iron, galvanized steel, stainless steel; fan stack—redwood, steel, masonite, drift eliminators—redwood, cement, asbestos board; spray nozzles—ceramic, bakelite; louvers—redwood, cement asbestos board.

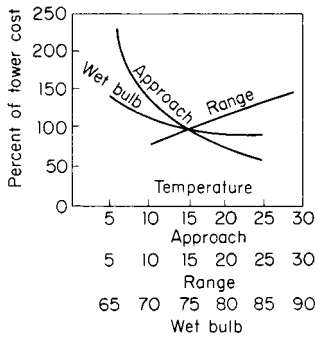


Fig. 9.5.24 Comparative cooling tower costs. *(Foster Wheeler Corp.)*

DRY COOLING TOWERS, WITH DIRECT-CONTACT CONDENSERS

REFERENCE: Ritchings and Lotz, Economics of Closed vs. Open Cooling Water Cycles, *Power Eng.*, May and June 1963.

The thermodynamic requirement for a heat sink can generally best be met with a natural body of water such as a river, lake, or ocean. When water supplies are limited, a **"wet" cooling tower** (water losses less than 2 or 3 percent) may be used for reclamation. A **"dry" cooling tower** will further reduce this loss. Figure 9.5.25 shows an application with a direct-contact condenser. The finned surfaces of the dry cooling tower are

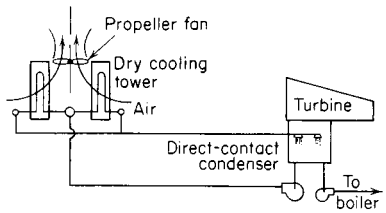


Fig. 9.5.25 Dry cooling tower system.

substituted for the tubes of a surface condenser. Heat-transfer rates (see Sec. 4) are appreciably lower, but the extended surface, relatively low in cost, may be installed in an induced-draft tower or in a hyperbolic tower, as in England for a 125,000-kW unit. Problems of recirculation, freezing, and air leakage must be solved. If a minor evaporative loss is permissible, peaking capacity is obtained by wetting the heat-exchange surface in the tower.

SPRAY PONDS

If land area is available, a spray pond may serve as an alternative reclamation system. Pipelines are typically arranged on 50-ft centers, with five nozzles (50 ± gal/min each) every 12 ft of pipeline. Cooling is limited by the relatively short period of contact between spray and air. Drift, especially with adverse wind conditions, can make losses higher than on towers. Generally, if cooling efficiencies above 50 percent are required, a tower is favored, where efficiency $= (t_h - t_c) \times 100/(t_h - t_{wb})$, where the temperature t subscripts are h = hot water, c = cold water, wb = wet bulb of the ambient air. (See Wright and Kirsopp, Pond Surface Cooling of a Chemical Plant Cooling Water, *Proc. Am. Power Conf.*, 1960.)

CLOSED FEEDWATER HEATERS

REFERENCES: Heat Exchange Institute Standards for Closed Feedwater Heaters. ASME Boiler and Pressure Vessel Code, Sec. VIII.

Feedwater heaters are used (1) to effect the thermodynamic gains of the regenerative steam cycle (see Secs. 4 and 9), and (2) to raise water temperatures to a sufficient value for the avoidance of thermal shock to the boiler metal. The **number and types of heaters** usually employed are: (1) plant sizes of up to 70 mW, two closed low-pressure heaters, one open heater, one closed high-pressure heater; (2) plant sizes of 75 to 300 mW, two or three closed low-pressure heaters, one open heater, two closed high-pressure heaters; (3) fossil-fueled plant sizes above 300 mW, three or four closed low-pressure heaters, one open heater, two or three high-pressure heaters. Nuclear units require very large feedwater flows, necessitating double or triple parallel strings of heaters. There are generally five or six low-pressure and one high-pressure heaters (no open heaters) in each string.

Minimum heater cost prevails with minimum restrictive specifications, e.g., horizontal, two-pass, high water velocity (10 ft/s at 60°F), no length limits. Overall heater length is limited by maximum available tube lengths of 100 ft for copper alloys, admiralty metal, and copper-nickel and 85 ft for Monel. With U-tube construction, this results in heater lengths of about 48 and 40 ft, respectively. A general rule, to ensure good steam distribution, is that the length in feet shall not exceed the shell diameter in inches plus 2; i.e., with a 30-in diam shell, the length should not exceed 32 ft. **Pressure drop** through the tubes must be economically evaluated as it varies approximately with the square of the water velocity.

If a **length restriction** is imposed, the designer may have to substitute a four-pass arrangement for the two-pass design, with consequent large-diameter shell and water chamber, heavier walls, more tube holes to be drilled, more tubes to be installed, and a cost increase. If a **pressure-drop restriction** is imposed, a lower water velocity results, with more tubes, larger shell and chamber diameter, and more surface because of the lower heat-transfer rate. Vertical heaters, with appropriate construction details, are also higher in cost.

The **condensing heater** (Fig. 9.5.26), like a surface condenser, performs at constant saturation temperature outside the tubes with no condensate subcooling. Feedwater is generally heated to within 5°F of the

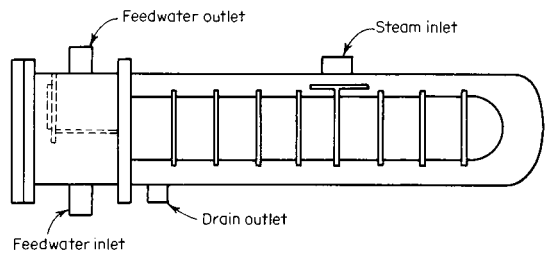


Fig. 9.5.26 Basic heater section for condensing heater.

saturation temperature. The addition of a **drain-cooling section** (Fig. 9.5.27), with a cover or enveloping baffle around the tubes of the inlet pass, can reduce the drip temperature to within 10 or 15°F of the incoming water. When the steam is at sufficient pressure (above 125 lb/in^2 abs) and contains enough superheat (more than 250°), a **desuperheating**

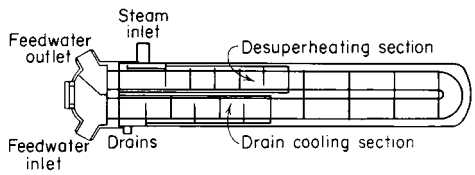


Fig. 9.5.27 Basic heater section with drain cooling and desuperheater sections added.

section (Fig. 9.5.27), using an enveloping baffle around the tubes at feedwater outlet, can raise the water temperature above the saturation temperature of the entering steam. Drain coolers and desuperheating sections improve overall plant heat rate by reduction of energy degradation. Subcooling of drains reduces flashing, erosion, vibration, and noise when drains are dropped to a lower-pressure heater. Special attention should be given to drains flashed to the main condenser, as this is a direct thermodynamic loss. As an alternate, these drains can be pumped forward into the feedwater line.

Heat-transfer rates for a condensing feedwater heater range, overall, between 500 and 900 Btu/(h · ft² · °F). (See Figs. 9.5.28 and 9.5.29.)

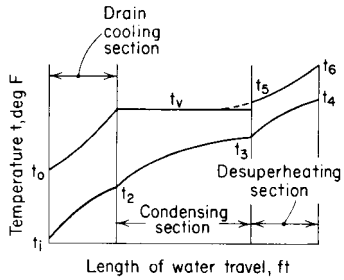


Fig. 9.5.28 Temperature vs. water travel, closed-feed heater.

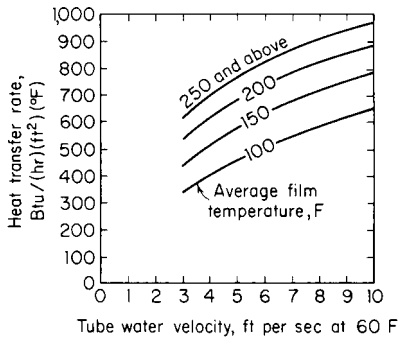


Fig. 9.5.29 Approximate condenser heat-transfer rates with 18 BWG admiralty metal tubes. Average film temperature $= t_s - 0.8\delta t_m$.

Heat-transfer rates for drain-cooling and desuperheating sections must be separately evaluated, with overall ranges from 300 to 500 on the former and 80 to 140 on the latter. (See Figs. 9.5.30 and 9.5.31.) Normally, the three sections are included in the same shell (Fig. 9.5.27); when the cooler becomes too large, as with the lowest-pressure heater, a separate shell-and-tube exchanger may be justified.

Low-pressure heaters (upstream of the boiler feed pump) are usually designed for tube-side pressure of less than 90 lb/in² gage. This is based on the shutoff head of the condensate pump plus 10 percent. For pressures up to 300 to 400 lb/in² gage, a bolted cover with flange and gasket

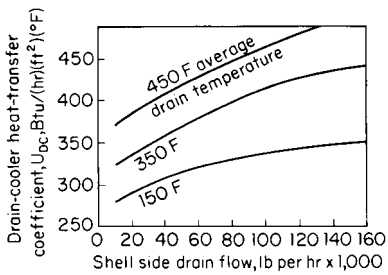


Fig. 9.5.30 Approximate drain-cooler heat-transfer rates.

(Fig. 9.5.32*a*) is used, while higher pressures usually justify a welded joint to eliminate the gasket (Fig. 9.5.32*b* and *c*). A manway is utilized for access to the tube sheet. Tubes are generally rolled into the tube sheet.

High-pressure heaters (downstream of the boiler feed pump) are designed for shutoff head of the pump plus 10 percent, resulting in about

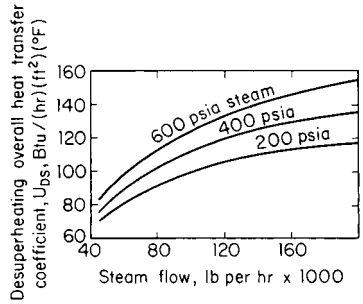


Fig. 9.5.31 Approximate desuperheater section heat-transfer rates.

1,500 lb/in² gage for a nuclear plant and up to 5,000 lb/in² gage for fossil-fuel plants. Hemispherical head construction (Fig. 9.5.32*c*) and welded tube-to-tube sheet joints are almost exclusively used. Since the cost of the tubes represents the largest portion of the heater cost, material selection is important (see Table 9.5.11). Depending on the desired water-chemistry requirements, low-pressure heaters usually use admiralty metal, 90-10 Cu-Ni, stainless or carbon steel. High-pressure heaters require the use of the higher-strength alloys of 70-30 Cu-Ni, monel, stainless, or carbon steel. The thinnest wall possible [$t = Pd/(2s + 0.8p)$] or the industry minimum standard of 18 BWG (20 BWG for stainless steel) is utilized.

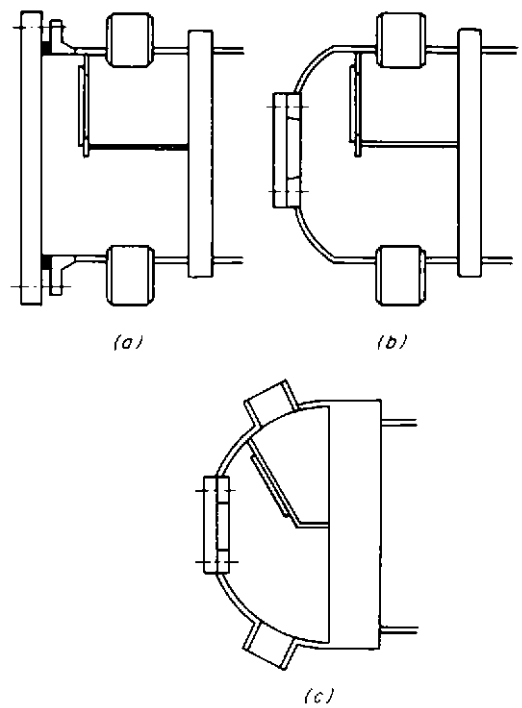


Fig. 9.5.32 Feedwater heater channels. (*a*) Bolted; (*b*) dished; (*c*) hemispherical.

Table 9.5.11 Material and Gage Correction Factor C_m
(For ⅝- to 1-in tubes)

BWG no.	Tube material: Ars-Cu	Admiralty	90-10 Cu-Ni	80-20 Cu-Ni	70-30 Cu-Ni	Monel	18-8 SS	Carbon steel
18	1.00	1.00	0.97	0.95	0.92	0.89	0.85	0.96
17	1.00	1.00	0.94	0.91	0.87	0.85	0.80	0.92
16	1.00	1.00	0.91	0.88	0.84	0.82	0.77	0.89
15	1.00	0.99	0.89	0.86	0.82	0.79	0.74	0.87
14	1.00	0.96	0.85	0.82	0.77	0.75	0.70	0.83
13	0.98	0.93	0.81	0.78	0.73	0.70	0.65	0.79

Performance Calculations

Notation

A = heat-transfer surface, ft²
C_m = material and gage correction factor, Table 9.5.11
D = tube ID, in, Table 9.5.12
d = OD of tube, in (⅝ and ¾ in are most common)
F_1 = friction factor, Table 9.5.13
F_2 = water-temperature correction factor, Table 9.5.14
h = enthalpy, Btu/lb
k = tube diameter and gage factor, Table 9.5.15
L = active length of tubes per pass, ft
N = number of passes
n = number of tubes
P = design pressure, lb/in²
p = pressure, lb/in²
Q = total heat transferred, Btu/h
S = allowable design stress at tube design temperature, lb/in², from ASME Code
t = wall thickness, in (see Table 9.5.16); temperature, °F
U_o = overall heat-transfer rate, Btu/(h · ft² · °F) (the material correction factor, Table 9.5.11, must be applied to the overall heat-transfer rate)
V = water velocity, ft/s
W = steam flow, lb/h
W_w = feedwater flow, lb/h
Δp = pressure drop, lb/in²
Δt_m = logarithmic mean temperature difference, °F (Fig. 9.5.28)

Table 9.5.12 Feedwater-Heater Tube Constants

Tube BWG	⅛-in OD D	⅛-in OD $D^{1.24}$	¾-in OD D	¾-in OD $D^{1.24}$	⅞-in OD D	⅞-in OD $D^{1.24}$
18	0.527	0.452	0.652	0.589	0.777	0.731
17	0.509	0.433	0.634	0.567	0.759	0.710
16	0.495	0.418	0.620	0.554	0.745	0.695
15	0.481	0.404	0.606	0.538	0.731	0.678
14	0.458	0.380	0.584	0.514	0.709	0.652
13	—	—	0.560	0.488	0.685	0.625

Table 9.5.13 Friction Loss, lb/in² per ft of Straight Travel

Water velocity (at 60°F), ft/s	F_1
3.0	0.018
4.0	0.031
5.0	0.047
6.0	0.065
7.0	0.085
8.0	0.108
9.0	0.134
10.0	0.162

Table 9.5.14 Water-Temperature Correction Factor*

Average water temp, °F	F_2
100	0.905
150	0.840
200	0.795
250	0.770
300	0.755
350	0.750
400	0.755
450	0.757
500	0.795
550	0.830
600	0.885

* Average water temperature $= t_s - \Delta t_m$.

Table 9.5.15 Tube Diameter and Gage Factor k

Diam, in	BWG no. 20	18	17	16	15
⅝	0.217	0.2407	0.2580	0.2728	0.289
¾	0.1732	0.1887	0.1995	0.2089	0.218
⅞	0.1445	0.1550	0.1624	0.1686	0.1748
1	0.1238	0.1314	0.1369	0.1413	0.1458

Table 9.5.16 Tube-Wall Thickness

Gage, BWG	Thickness, in
15	0.072
16	0.065
17	0.058
18	0.049
19	0.042
20	0.035
22	0.028

Heat-balance data will provide the major parameters of steam pressure, feedwater flows, and temperatures. The choice of tube diameter and material must be made first. Tube thickness is calculated by

$$t = Pd/(2S + 0.8P)$$

Test pressure is 1.5 times design pressure.

Condensing-heater-size equations are

$$Q = UA\,\Delta t_m = W_w(h_3 - h_2) = W_s(\Delta h_s)$$

(See Figs. 9.5.28 and 9.5.29.) The minimum terminal temperature difference is 2°F.

The active length of tubes is calculated from $L = 500AV/(NW_w k)$, where maximum velocity is 10 ft/s at 60°F; the number of tubes, from $n = 3.82A/(Ld)$; the pressure drop, from $\Delta p = (L + 5.5D)F_1F_2N/D^{1.24}$.

The shell design pressure is 20 percent greater than the maximum operating pressure, rounded to the higher 25 lb/in² as a minimum.

A **drain-cooling section,** when added, is treated as a separate heat exchanger where the quantities t_v, t_o, t_i, W_s, and W_w are known; t_2 is the only unknown and is to be calculated from a heat balance on the section or $Q = W_s(h_v - h_o) = W_w(h_2 - h_1)$. Compute Δt_m; find the area of the drain-cooler section A_{dc} from $Q = U_o A_{dc} \Delta t_m$, with overall U_o from Fig. 9.5.30.

A **desuperheating section** is similarly calculated with t_v, t_6, t_4, p_6, h_6, h_4, W_s known. Allow approximately 1 percent pressure drop through the section to get p_v, and 80 to 90 percent desuperheating (30°F superheat entering condensing section) to obtain t_5 and h_5. By a heat balance on section, calculate h_3 from $Q = W_s(h_6 - h_5) = W_w(h_4 - h_3)$. With equivalent t_3, calculate Δt_m and substitute in $Q = U_o A \Delta t_m$ for area of desuperheating section (overall U_o from Fig. 9.5.31).

Materials

Heaters are designed in accordance with the Heat Exchange Institute Standards for Closed Feedwater Heaters and the ASME Boiler and Pressure Vessel Code, Sec. VIII, using the following materials:

Tubes:

Material		Max temp, °F: Rolled joint	Max temp, °F: Welded joint
Admiralty metal	ASME SB-395	350	
90-10 Cu-Ni		400	450
80-20 Cu-Ni		400	475
70-30 Cu-Ni		400	525
70-30 Cu-Ni, stress-relieved		400	525
Monel, ASME SB-163		400	600
Carbon steel, ASME SA-556, grades A, B, and C		350	650
Stainless steel, ASME SA-249; TP 304 (welded)		350	650

Shell, nozzles: carbon-steel pipe and plant, ASME SA-515 grade 70; SA-106 grade B.

Channels, or heads, channel covers: carbon-steel plate, forgings, and pipe, ASME SA-515 grade 70; SA-105 grade II; SA-106 grade B.

Baffles and tube supports: steel plate, ASME SA-283 grade C.

Tube sheets: carbon-steel plate and forgings, ASME SA-515 grade 70; SA-105 grade II.

Carbon steel is used for temperatures up to 800°F. Chromium-molybdenum steel (ASME SA-387 grade C) is used for higher temperatures.

OPEN, DEAERATING, AND DIRECT-CONTACT HEATERS

REFERENCE: Heat Exchange Institute Standards for Deaerators and Deaerating Heaters.

Deaerators or deaerating heaters serve (1) to degasify feedwater and thus reduce equipment corrosion (see Sec. 6), (2) to heat feedwater regeneratively and improve thermodynamic efficiency (see Sec. 9), and (3) to provide storage, positive submergence, and surge protection on the boiler feed-pump suction (see Sec. 14).

Removal of oxygen and carbon dioxide from boiler feedwater and process water at elevated temperature is essential for adequate conditioning. In some power-plant applications, a well-designed surface condenser gives adequate deaeration and the accompanying exclusive use of closed feedwater heaters.

A modern **deaerator** will, by mechanical action, reduce O_2 content of effluent to less than 0.005 cm^3/L and CO_2 content to a negligible amount. Water must (1) be heated to and kept at saturation temperature, as the gas solubility is zero at the boiling point of the liquid, and (2) be mechanically agitated by spraying or cascading over trays for effective scrubbing, release, and removal of gases. Gases must be swept away by an adequate supply of steam. Since the water is heated to saturation conditions, the terminal temperature difference is zero with maximum improvement in associated turbine heat rate. Extremely low partial gas pressures, dictated by Henry's law, call for large volumes of scrubbing steam. **Vent condensers** of the shell-and-tube or direct-contact types, cooled by incoming feed, serve to recover heat and water before release of gases to the atmosphere.

The **tray-type deaerator** (Fig. 9.5.33) is prevalent. While it has some tendency to scale, it will operate at wide load conditions and is practically independent of water-inlet temperature. Trays can be loaded to some 10,000 $lb/(ft^2 \cdot h)$, and the deaerator seldom exceeds 8 ft in height.

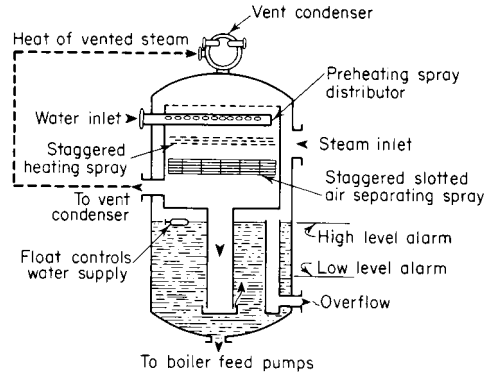


Fig. 9.5.33 Tray-type deaerating heater section.

The **spray type** uses a high-velocity steam jet to atomize and scrub the preheated water. Applications are (1) marine service, where it is unaffected by ship roll and pitch, and (2) industrial plants where operating pressures are stable. It requires a temperature gradient, e.g., 50°F minimum, to produce the fine sprays and vacuums with the cold water required.

Materials

Open-feed heaters are designed to the ASME Boiler and Pressure Vessel Code. Materials used are: shell—steel, ASME SA-285 grade C and SA-515 grade 70; trays—stainless steel, type 304; baffles in vent condenser—stainless steel, type 304; spray valves—stainless steel, type 304.

EVAPORATORS

For general treatment of evaporators used in the preparation of makeup water, see Sec. 6.

9.6 INTERNAL COMBUSTION ENGINES

by D. Assanis, C. Borgnakke, D. E. Cole, and D. J. Patterson

REFERENCES: Lichty ''Internal-Combustion Engines,'' McGraw-Hill. Obert, ''Internal-Combustion Engines and Air Pollution,'' Harper & Row. Taylor and Taylor, ''The Internal-Combustion Engine,'' International Textbook. Vincent, ''Supercharging the Internal-Combustion Engine,'' McGraw-Hill. Crouse, ''Automotive Engine Design,'' McGraw-Hill. Patterson and Henein, ''Emissions from Combustion Engines,'' Ann Arbor Science. Starkman, ''Combustion Generated Air Pollution,'' Plenum. *Trans. ASME Trans. SAE Proc. I Mech E,* Automobile Division, *Automotive Inds.* Ferguson, ''Internal Combustion Engines,'' Wiley. Heywood, ''Internal Combustion Engine Fundamentals,'' McGraw-Hill. Blair, ''The Basic Design of Two-Stroke Engines,'' SAE. Watson and Janota, ''Turbocharging the Internal Combustion Engine,'' Wiley. Benson and Whitehouse, ''Internal Combustion Engines,'' Pergamon.

GENERAL FEATURES

Cycles

(See also Sec. 4.1)

In piston-type **internal combustion engines,** the combustion process is assumed to occur at constant volume (Fig. 9.6.1), at constant pressure (Fig. 9.6.2), or by some combination of these (Fig. 9.6.3). The constant-volume process is characteristic of the **spark ignition** or **Otto cycle;** the

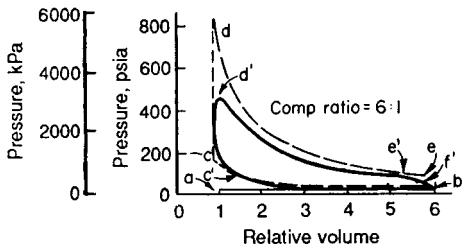


Fig. 9.6.1 Otto cycle (constant-volume combustion, no excess air). Process of the ideal cycle: *a-b,* suction stroke, admission of the charge; *b-c,* compression stroke; at *c,* ignition of the compressed charge; *d-e,* expansion; at *e,* exhaust valve opens; *b-a,* exhaust stroke.

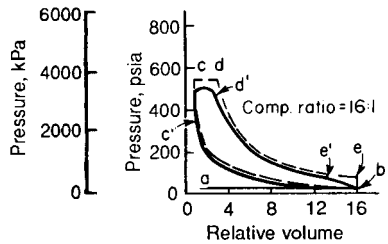


Fig. 9.6.2 Diesel cycle (constant-pressure combustion, 50 percent excess air). Processes of the ideal cycle: *a-b,* suction stroke, admission of air; *b-c,* compression of air; *c-d,* injection and combustion of the fuel; *d-e,* expansion; at *e,* exhaust valve opens; *b-a,* exhaust stroke.

constant pressure is found only in the slow-speed **compression ignition** or **diesel cycle;** with both processes, the cycle is sometimes called a **mixed, combination,** or **limited pressure cycle**. Actually, the indicator card obtained from a high-speed, mixed-cycle, compression ignition engine may be similar to that obtained from a spark ignition engine. The fundamental differences are the methods of mixing the air and fuel (before compression in the Otto cycle, and usually near the end of compression in the diesel cycle) and the methods of ignition.

The nominal **compression ratio** (usually specified) is the displacement plus clearance volume divided by the clearance volume. The actual compression ratio is appreciably less than the nominal value because of late intake valve or port closing.

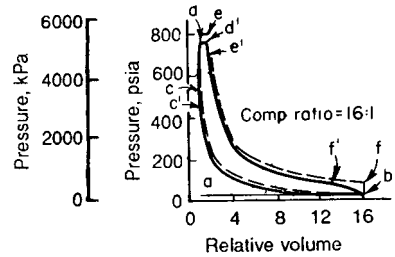


Fig. 9.6.3 Mixed cycle (constant-volume and constant-pressure combustion, 50 percent excess air). Processes of the ideal cycle: *a-b,* suction stroke, admission of air; *b-c,* compression of air; *c-d,* ignition and constant-volume combustion; *e-f,* expansion; at *f,* exhaust begins; *b-a,* exhaust stroke. Broken lines represent ideal cycle; solid lines show characteristics of actual cycle. Primes attached to letters show actual locations of events, though some lower-compression-ratio and multiple-chamber engines may require starting aids.

The **compression pressure** may be estimated from the relation $p = r_a^{1.33}$, where p_m is the intake manifold pressure and r_a is the actual compression ratio.

The actual compression pressure can be determined with a compression pressure gage that traps the gases by means of a check valve, thus indicating the maximum pressure under motoring conditions.

Spark ignition engines use volatile liquids or gases as fuel; have compression ratios between 6 : 1 and 12 : 1 (limited by combustion knock or the fuel/air mixture) and compression pressures from below 150 to above 300 lb/in^2 (1,030 to 2,060 kPa); use carburetors, gas mixing valves, or fuel injection systems; and operate on the Otto cycle. Gasoline is the fuel commonly used in airplane, automobile, small marine, small stationary, and tractor engines. Commercial gases such as blast furnace gas, coal gas, coke oven gas, carbureted water gas, producer gas, and natural gas are used in large stationary engines. The mixtures used generally are near chemically correct. Combustion pressures are usually 3.5 to 5 times the compression pressures. Load and speed are usually controlled by throttling the charge; piston speeds above 3,000 ft/min (15 m/s) are permissible.

Advantages are low first cost, low specific weight, low cranking effort required, wide variation obtainable in speed and load, high mechanical efficiency, and fairly low specific fuel consumption at high compression ratios and wide-open throttle.

Compression ignition engines use liquid fuels of low volatility varying from fuel oil and distillates to crude oil (see Sec. 7.1), have compression ratios between 11.5 : 1 and 22 : 1 and compression pressures 400 to 700 lb/in^2 (2,760 to 4,830 kPa), and operate on the diesel or mixed cycle. Generally no ignition devices are used, although some lower-compression-ratio and multiple-chamber engines may require starting aids. Load and speed are controlled by varying the fuel quantity injected.

The **dual-fuel engine** is a diesel engine with a compression ratio which is too low to result in ignition, at the desired time, of the gas/air mixture inducted into the cylinder. A pilot (small) injection of liquid fuel with good ignition quality is used to initiate the combustion process.

Advantages are low specific fuel consumption, high thermal efficiency at partial loads, possibly lower fuel cost, no preignition, low CO and hydrocarbon emission at low and moderate loads, suitability for

two-stroke operation, and excellent durability. The lower-compression engines are of simpler construction (usually valveless two-stroke cycle), are more lightweight, and have lower first cost, lower operating expense, and higher mechanical efficiency than the higher-compression engines.

The **four-stroke cycle** (four-cycle) engine requires four piston strokes or two crankshaft revolutions per cycle (Figs. 9.6.1 and 9.6.2). This engine cycle is used almost exclusively in automobile, tractor, and aircraft engines of all types and sizes and in engines of other classifications with the exception of most outboard engines. Small four-cycle engines are always single-acting (combustion on only one side of the piston). The pumping loss, indicated by the area between the exhaust and the induction curves (Fig. 9.6.12), is more than the negative loop of the indicator card and at wide-open throttle depends on valve openings and speed and amounts to 3 to 7 percent of the engine indicated power. Throttling increases the pumping loss and reduces the positive indicator card area.

Advantages compared with two-cycle crankcase compression spark ignition engines include the wider variation in speed and load, cooler pistons, common crankcase in multicylinder engines, good lubrication secured more easily, no fuel loss during exhaust, lower specific fuel consumption lower pumping losses, less exhaust dilution, lower hydrocarbon emissions, positive-displacement inlet and exhaust processes, and easier power regulation.

The **two-stroke cycle** (two-cycle) engine requires two piston strokes or only one revolution for each cycle. Exhaust ports in the cylinder wall are uncovered by the piston, or exhaust valves in the cylinder head are opened near the end (at 60 to 88 percent) of the expansion stroke, permitting the escape of exhaust gases and reducing the pressure in the cylinder (Fig. 9.6.4). The charge of air or combustible mixture flows into and is compressed in a separate crankcase compartment for each cylinder, by a compressor or a blower to slightly above atmospheric pressure. Intake ports are uncovered by the piston or intake valves and

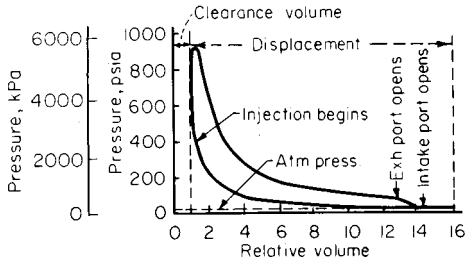


Fig. 9.6.4 Indicator card for two-cycle diesel engine (16 : 1 compression ratio).

are opened soon after the opening of the exhaust, and the compressed charge flows into the cylinder, expelling most of the exhaust products, some charge escaping with the exhaust. Blowers or compressors, driven either mechanically or by an exhaust turbine, are often used to supply sufficient air to scavenge the cylinder and clearance space and, in some cases, to supercharge the engine. Although small engines use only ports and crankcase compression (Fig. 9.6.5), large engines usually have separate compressors or blowers and either ports or both valves and ports. With crankcase compression, low scavenging of the cylinder occurs, and the volumetric efficiency is low (30 to 70 percent). The crankcase compression work amounts to 7 to 12 percent of the total work, but scavenging blowers with displacements 20 to 80 percent greater than the piston displacement may require as high as 30 percent of the indicated work in high-speed engines.

Advantages compared with four-cycle engines include the 50 to 80 percent greater power output per unit piston displacement and same speed (depending on scavenging) twice at many power impulses per cylinder per revolution, low cost for valveless designs employing crankcase compression, low NO_x emissions (spark ignited), and light weight.

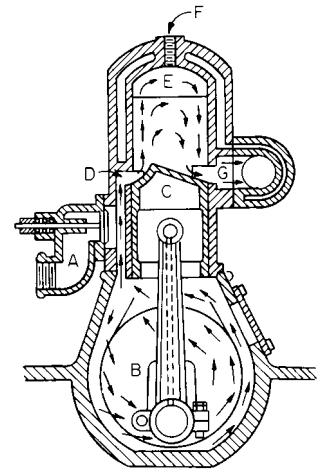


Fig. 9.6.5 Small two-cycle gasoline engine with crankcase compression.

ANALYSIS OF ENGINE PROCESS

Cycle Analysis The theoretical indicated thermal efficiency η_a of the air standard Otto cycle depends only on the volumetric compression ratio r_c (Sec. 4.1):

$$\eta = 1 - \left(\frac{1}{r_c}\right)^{0.4} \tag{9.6.1}$$

For the air standard diesel cycle (constant-pressure combustion),

$$\eta_a = 1 - \frac{(r_d^{1.4} - 1)}{1.4 r_c^{0.4}(r_d - 1)} \tag{9.6.2}$$

where r_d is the volumetric expansion ratio during constant-pressure combustion.

However, the diesel engine is more closely approximated by the limited pressure combustion (dual) cycle for which the thermal efficiency is given by

$$\eta_a = 1 - \frac{r_p r_d^{1.4} - 1}{r_c^{0.4}[r_p - 1 + 1.4 r_p(r_d - 1)]} \tag{9.6.3}$$

where r_p is the pressure ratio during the constant-volume heat addition.

The efficiencies indicated by the air standard analysis are much higher than can be attained, principally because of variable specific heats, dissociation, and heat and time losses. The fuel-air cycle analysis takes into consideration variable fuel/air mixture properties and dissociation of combustion products and results in lower, more realistic values for efficiencies (Figs. 9.6.6 and 9.6.7). Real cycle analyses include, in addition, combustion losses, heat losses, and leakages, resulting in thermal efficiencies approximately 80 percent of those predicted by the fuel-air cycle analyses.

The **mean effective pressure (mep)** is equal to net work divided by volumetric displacement. Horsepower follows from

$$\text{hp} = P_{\text{mep}}\, SA\, \text{rpm}/K \tag{9.6.4}$$

where P_{mep} = mean effective pressure, lb/in² (kPa); S = stroke, ft (m); A = total piston area, in² (m²); rpm = number of cycles completed per min; K = 33,000 (0.4566 for SI units).

Increasing the compression ratio and the air-fuel ratio increases the thermal efficiency of the Otto cycle (Fig. 9.6.6). Increasing the compression ratio and the air/fuel ratio also increases the thermal efficiency of the diesel cycle with constant-pressure combustion (Fig. 9.6.7). Mean effective pressures depend on charge input and thermal efficiency, both of which depend on compression ratio and air and fuel supply (Figs. 9.6.8 and 9.6.9).

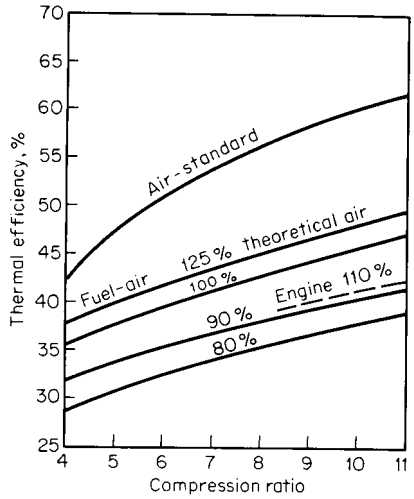


Fig. 9.6.6 Indicated thermal efficiencies of various Otto cycle analyses and engine test results.

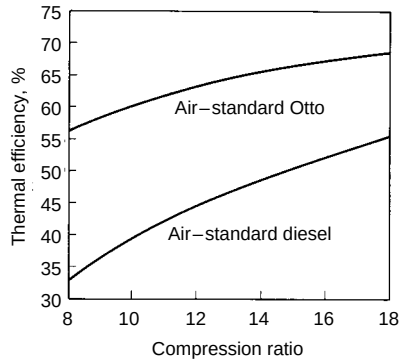


Fig. 9.6.7 Indicated thermal efficiency of the air standard diesel cycle with constant-pressure combustion.

Deviations from Ideal Processes

Changes in atmospheric pressure and temperature change power output although they do not appreciably affect thermal efficiency. The standard correction formulas adopted by SAE for brake power output are, for spark ignition engines,

$$\text{bhp}_s = \text{bhp}_o \left(\frac{P_s}{P_o}\right) \sqrt{\frac{T_o}{T_s}}$$

and for diesel engines,

$$\text{bhp}_s = \text{bhp}_o \left(\frac{P_s}{P_o}\right) \left(\frac{T_o}{T_s}\right)^{0.7}$$

where subscripts o and s indicate observed and standard conditions, respectively, P_o being the dry barometric pressure.

Pumping work (area EFGAE, Fig. 9.6.10) is required to overcome the resistance to flow of fresh charge into and exhaust products out of the cylinder. It varies from almost 0 for an engine running at slow speed with wide-open throttle to about 3 lb/in^2 (21 kPa) mep at high speed, and to 10 lb/in^2 with idle throttle position. The negative loop (area AXFGA, Fig. 9.6.10) in a normally aspirated engine represents about 60 percent of the pumping work.

Volumetric efficiency is the ratio of the volume of air (for a liquid-fuel) engine) and charge (for a gas engine) actually admitted, measured at atmospheric pressure and temperature, to the displacement volume. High manifold and cylinder temperatures and resistance to flow reduce volumetric efficiency. High-output aircraft engines have maximum volumetric efficiencies of 85 to 90 percent at rated speeds. Supercharged engines can have volumetric efficiencies well above 100 percent or can be designed to maintain high volumetric efficiencies with an increase in speed.

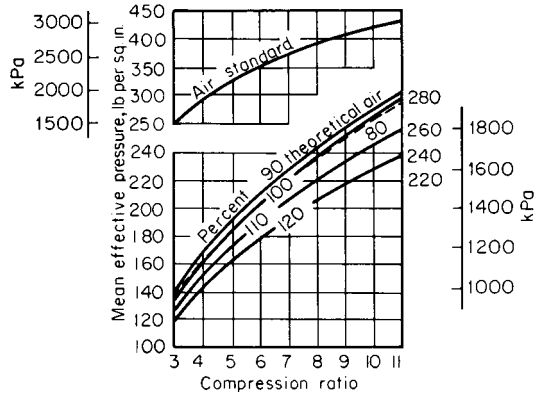


Fig. 9.6.8 MEP for Otto cycle engine [air and liquid iso-octane supplied at 60°F (16°C)].

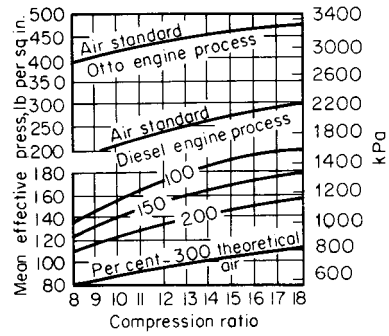


Fig. 9.6.9 MEP for diesel cycle engine [air and liquid dodecane supplied at 60°F (16°C)].

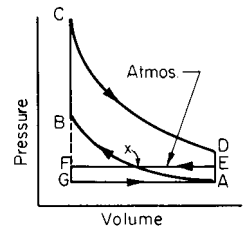


Fig. 9.6.10 Throttled Otto cycle.

Volumetric efficiencies of automobile engines (Fig. 9.6.11) diminish appreciably as piston speeds exceed 1,200 to 1,600 ft/min (6 to 8 m/s), decreasing to approximately 60 percent at top speeds of 2,500 to 3,200 ft/min (13 to 16 m/s). Volumetric efficiency varies appreciably with valve timing.

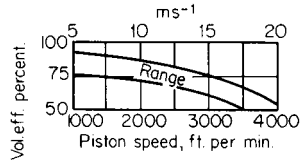


Fig. 9.6.11 Volumetric efficiencies of normally aspirated engines with poppet valves.

Table 9.6.1 Relative Ideal Volumetric Efficiencies with Liquid Fuels
(Air and liquid fuel at 60°F)

Fuel	Chemically correct air-fuel ratio	Heat required per lb of fuel for complete evaporation, Btu/lb (J/gm)	Percent evap with 200 Btu/lb fuel	Suction temperature, °F (°C)		Relative volumetric efficiency	
				Complete evaporation	200 Btu/lb fuel	Complete evaporation	200 Btu/lb fuel
Gasoline	15.11	330 (768)	81	106 (41)	81 (27)	100	100
Benzene	13.26	149 (347)	100 +	50 (10)	68 (20)	111	103
Ethyl alcohol	8.99	425 (989)	55	71 (22)	55 (13)	107	105
Methyl alcohol	6.46	519 (1,207)	49	68 (20)	45 (7)	108	107

Two-cycle crankcase compression engines have volumetric efficiencies of about 60 percent at low speeds and 50 percent or below at high speeds.

Fuels with high latent heats and low boiling temperatures reduce the charge temperature and result in high volumetric efficiencies (Table 9.6.1).

Air cleaners, governors, and carburetors increase the resistance to flow and decrease the volumetric efficiency.

The **compression stroke** begins with heat transfer from the cylinder walls to the charge but ends with heat transfer from the charge to the walls. The net result is a heat loss of about 0.5 to 1.5 percent of the heat value of the charge. Heat loss during compression lowers the compression curve and work, but lowers the expansion work a greater amount, thus slightly reducing the net work output.

The mean value of the exponent n in the approximate equation $pv^n =$ constant for the polytropic compression of a correct mixture of octane and air is about 1.33. For the diesel cycle in which the gas under compression is mainly air, the exponent is about 1.38 for a 15 : 1 compression ratio. Heat transfer into the charge increases the exponent, but leakage lowers its value.

The **combustion process** begins before the end of the compression stroke and ends after the beginning of the expansion stroke. Rassweiler, Withrow, and Cornelius (*Trans. SAE,* **34,** 1939) found a maximum variation of 30 percent in combustion time for a single-cylinder engine while variations of both maximum cylinder pressure and rate of pressure rise of about 30 percent were found by Patterson (SAE paper 660129) in a V-8 engine. These effects are thought to be due primarily to cyclic variations in swirl and turbulence at the spark plug, but can arise also from variations in air/fuel ratio and homogeneity of the mixture. Poor distribution in multicylinder engines also causes variations between cylinders. It has been shown that the loss due to the actual finite combustion time is 3 to 6 percent of the ideal efficiency with constant-volume combustion. Any energy released before or after top-center piston position has an availability corresponding to the expansion ratio for the piston position at which the energy is released. Heat loss during the combustion process amounts to 5 to 7 percent of the heat supply of the charge at rated outputs and speeds for automotive engines.

Flame Travel Flame begins at the spark plug in the spark ignition engine or at various points in the combustion chamber of the compression ignition engine and travels in all directions through the mixture. At the end of combustion, the first part of the charge to burn has reached a higher temperature than the last portion to burn, except in the constant-pressure process. Rassweiler and Withrow (*Trans. SAE,* **30,** 1935, p. 125) observed temperature differences of over 400°F (200°C) (nonknocking) and over 600°F (315°C) (knocking) in a nonturbulent combustion chamber.

The combustion of part of the charge compresses the remainder. Thus with 30 percent of the mass burned, the volume of the unburned portion will be about 35 percent of the total volume. This increases the temperature of the last fraction to burn in a constant-volume process by about 500°F (280°C) above its temperature at the beginning of combustion.

The mean adiabatic exponent for the **expansion process** is about 1.22, varying from about 1.20 at the beginning to 1.25 at the end of the process. Heat transfer and leakage increase the exponent, an average value of 1.33 being found by Rassweiler and Withrow (*Trans. SAE,* **33,** 1938, p. 185) in an L-head 2⅞- by 4¾-in (7.3- by 12.1-cm) cylinder. The heat transfer during the expansion stroke amounts to 8 to 12 percent of the heat value of the charge. Because of heat losses and real gas effects, the thermal efficiency of the engine is decreased by about 40 percent.

Blowdown begins at 80 to 90 percent of the expansion stroke. This reduces the work about 1 to 2 percent. High gas velocities [1,200 to 1,500 ft/s (360 to 460 m/s)] are attained, and heat transfer, including that through exhaust port walls, amounts to 10 to 20 percent of the heat value of the charge, but represents practically no loss of work or availability.

The **exhaust process** is the discharge of some of the gases from the cylinder by the piston. The exhaust pressure is usually above atmospheric (Fig. 9.6.12) but may run below atmospheric for part of the stroke. Heat loss during exhaust amounts to 3 to 5 percent of the heat

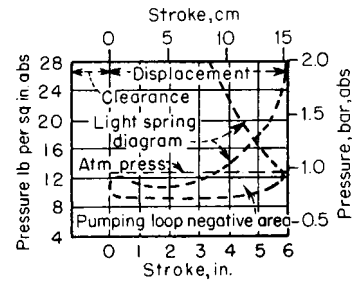


Fig. 9.6.12 Pumping loop of Otto cycle engine.

value of the charge but does not represent any loss of work or availability. At full load about 80 percent of the gases in the cylinder escape during blowdown, and about 15 percent are pushed out of the cylinder during exhaust, the remainder being left in the clearance space at the end of the exhaust stroke. At partial load, the blowdown pulse is smaller and the residual fraction higher.

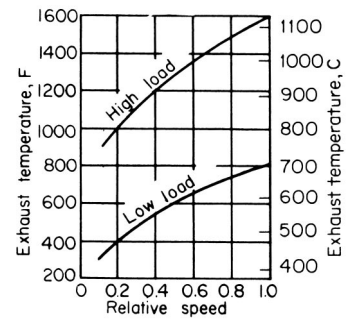


Fig. 9.6.13 Exhaust temperatures for Otto cycle engines.

Exhaust gas temperatures vary with speed and load (Figs. 9.6.13 and 9.6.14), high loads and speeds resulting in the highest temperatures. The first gases escaping during release are at the highest temperatures.

Exhaust Gas Analysis The composition of exhaust varies depending upon the equivalence ratio and the hydrogen/carbon ratio of the fuel. Figure 9.6.15 shows the calculated composition including water

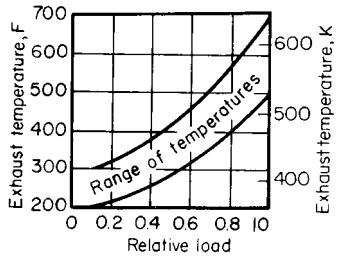


Fig. 9.6.14 Exhaust temperatures for diesel cycle engines.

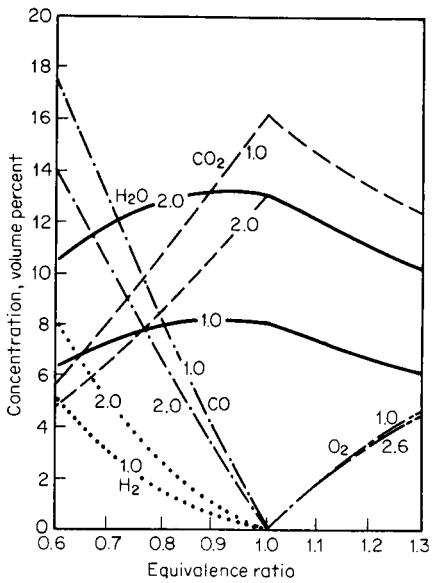


Fig. 9.6.15 Theoretical mole percent of principal combustion products of hydrocarbon fuels for fuel hydrogen/carbon ratios.

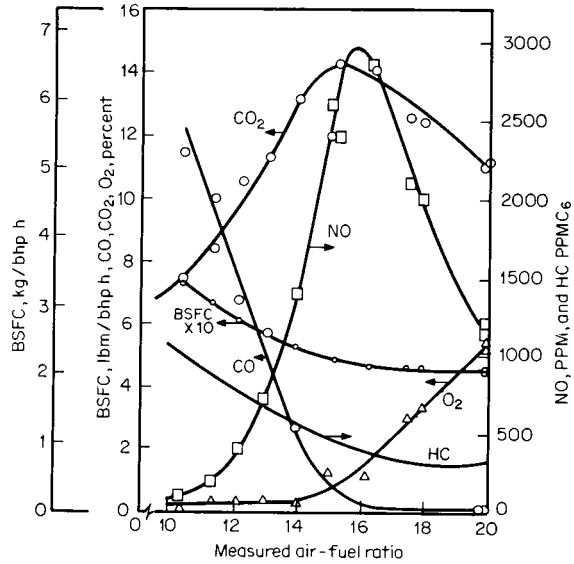


Fig. 9.6.16 Gasoline engine basic specific fuel consumption and emissions vs. air/fuel ratio.

(wet basis) for complete combustion. Charts of theoretical exhaust composition for various fuels have been prepared by D'Alleva (*General Motors Res. Rept.*, **372,** 1960) and Eltinge (*SAE Trans.*, 1968). In practice, nonequilibrium products, namely, unburned hydrocarbons and nitrogen oxide, appear in trace quantities. Figure 9.6.16 shows measured exhaust composition on a dry basis for a fuel with a hydrogen/carbon ratio of 1.85.

Equations for calculating the air/fuel ratio from exhaust CO, CO_2, HC, and O_2 have been developed by Spindt (*SAE Trans.*, 1966) and others. Such determinations and those for calculation of remaining but unmeasured exhaust species depend upon carbon, hydrogen, oxygen, and nitrogen balances employing measured values for some exhaust constituents (Obert, ''Internal Combustion Engines'').

Table 9.6.2 summarizes the principal exhaust gas constituents and combustion efficiency. HC and NO were not measured.

U.S. AUTOMOBILE ENGINES

The size (displacement) of U.S. automobile engines varies from 61 in^3 (1.0 L) to 488 in^3 (8 L). Rated horsepower and displacement have

Table 9.6.2 Summary of Exhaust Gas Constituents and Combustion Efficiency*

Air/Fuel	Percent by volume						$\frac{H_2O}{CO_2}$	$\frac{H_2O}{Fuel}$	Combustion† eff., percent
	CO_2	O_2	CO	H_2	N_2	H_2O			
11	8.76	0.15	9.14	4.66	77.08	13.78	1.57	0.972	66.7
12	10.18	0.44	6.65	3.39	79.13	13.93	1.37	1.043	73.8
13	11.60	0.59	4.31	2.20	81.09	14.16	1.22	1.122	81.5
14	13.02	0.63	2.09	1.07	82.99	14.46	1.11	1.205	89.6
15	13.23	1.35	0.99	0.50	83.72	14.09	1.06	1.247	93.8
16	12.62	2.49	0.68	0.35	83.65	13.30	1.05	1.256	94.8
17	12.00	3.55	0.48	0.25	83.51	12.54	1.05	1.261	95.5
18	11.45	4.49	0.30	0.16	83.39	11.88	1.04	1.267	96.2
19	10.90	5.36	0.20	0.10	83.23	11.25	1.03	1.269	96.5
20	10.40	6.15	0.11	0.06	83.07	10.68	1.03	1.272	96.9
21	9.92	6.86	0.08	0.04	82.90	10.16	1.03	1.271	96.9
22	9.44	7.55	0.06	0.03	82.71	9.65	1.02	1.268	96.8
23	9.00	8.18	0.05	0.03	82.53	9.19	1.02	1.266	96.7
24	8.60	8.74	0.06	0.03	82.37	8.78	1.02	1.264	96.6

* Gerrish and Voss, *NACA Rept.* 616, 1937.
† Low-heat-value basis.

trended downward while horsepower per unit of displacement has generally increased during the period from 1984 to 1994 (Fig. 9.6.17). Most U.S. automobiles are spark-ignited and are fueled with gasoline.

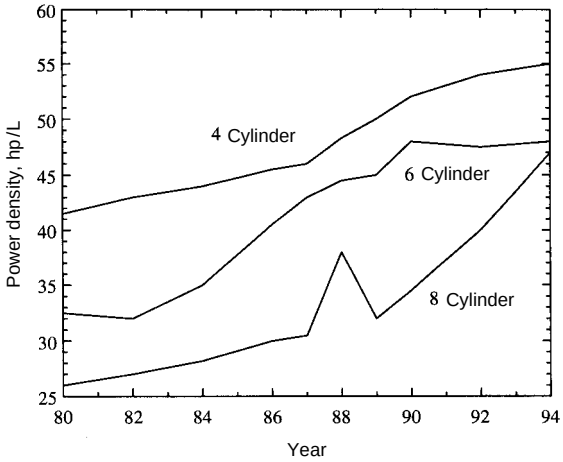


Fig. 9.6.17 Trend of average U.S. automobile power density.

Emission control has brought about (1) reduced compression ratios, (2) improved valves, (3) more precise and accurate fuel metering (carburetion and fuel injection), (4) controlled inlet air temperatures, and (5) electronically controlled fuel management (most with closed-loop feedback to maintain a chemically correct mixture), exhaust gas recirculation, and ignition timing. Current advertised horsepower is based on net rather than gross horsepower. Net horsepower is measured with a fully equipped engine, including all parts needed to perform its intended function unaided, such as fuel pump, oil pump, intake air system, and exhaust system.

Valve timing, particularly the closure of intake valves (approximately 50° crankshaft rotation after bottom dead center), is selected to give a rising volumetric efficiency and torque as the rotation speed is increased. At high speeds, restrictions in the intake system, developed primarily by the carburetor venturi and the intake valves, begin to throttle the engine. The 1994 base engines developed maximum torque at approximately 2,600 r/min. Despite reduced torque at higher speeds, the power peaks at 4,200 to 5,200 r/min. Higher powers were obtained with four-barrel (compound) carburetors, electronic fuel injection, and other modifications.

Engines have been produced with various numbers of cylinders over the years (from 2 to 16), but the most common engines in modern cars are in-line four-cylinder designs (Fig. 9.6.18) for smaller cars and V-6 and some V-8 designs for larger cars. All V-8 engines have an angle of 90° between the two banks of four cylinders, whereas the V-6 engines have an angle of either 60° or 90° between two banks of three cylinders. Smaller engines, in terms of both cylinder number and displacement, provide improved fuel economy, which is particularly important because of rigid federal corporate average fuel economy (CAFE) standards. Neither four- nor six-cylinder engines operate as smoothly as the V-8 because of less effective dynamic balancing and greater torque fluctuations. With the rapid growth in the use of transverse engines with front drive, and smaller and lighter vehicle structures with sloping aerodynamic front hoods, packaging constraints on automobile engines are severe.

All modern engines employ compact, rigid structures. Most cylinder blocks and heads are made of cast iron, although use of aluminum is increasing, particularly in cylinder heads. Four-cylinder engines use three or five crankshaft bearings; the V-6 uses four, and the V-8 uses five. Cylinder bore is usually greater than piston stroke.

Liquid cooling passages are provided around the bore of each cylinder as well as at the hot regions of the combustion chambers and exhaust ports in the head. Modern pistons are almost always made of aluminum. Crankshafts and connecting rods are usually made of cast iron or forged steel. Four-cylinder engines and some six-cylinder engines use a separate crank throw (eccentric portion of the crankshaft) for each connecting rod, whereas in V-8s, connecting rods for corresponding cylinders on each side of the V are side by side on the same crank throw.

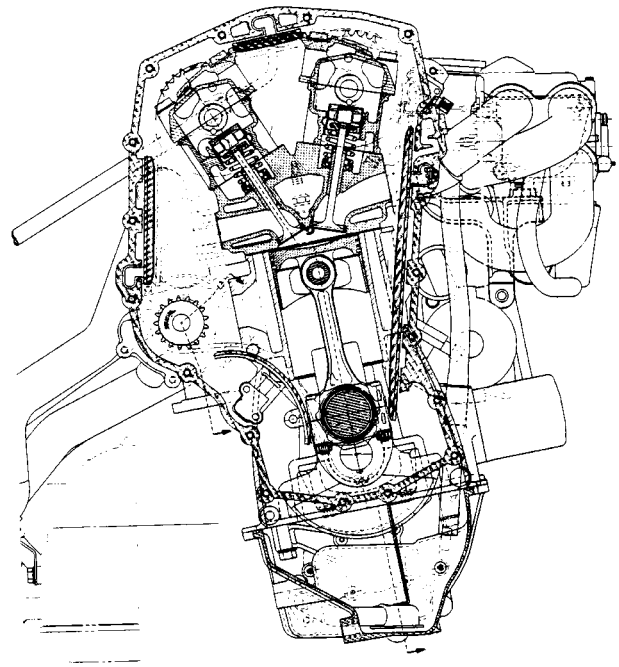


Fig. 9.6.18 Oldsmobile Division Quad 4 with four valves per cylinder (1987).

Detachable cylinder heads contain the inlet and exhaust valves (overhead valves). Most engines employ a single exhaust and inlet valve, although there is a growing trend toward designs with three or four valves per cylinder (Table 9.6.3). Many four-cylinder engines use a belt-driven overhead camshaft, although most V-6s and V-8s still have a camshaft in the valley between the cylinder banks. Exhaust gas is directed through short passages to the exhaust collector or manifold.

Fuel and air are delivered to the head by a compact inlet manifold. Fuel management is provided by either a carburetor or an electronic fuel injection system which may be either multipoint (one injector for each inlet port) or single-point (throttle body injection). In most cases, feedback from an exhaust gas composition sensor is used to provide precise control of air/fuel ratio and optimize performance of the emission control system. Electronic ignition virtually eliminates the need for periodic ignition system maintenance. Manufacturing precision has been improved considerably, leading to improved overall quality, reduced friction, and greater reliability.

Quiet operation is essential in automobile engines and requires close manufacturing control of clearances. Most engines have hydraulic valve lifters and means of rotating the exhaust valves, at least, to prolong life.

Table 9.6.3 Selected U.S. Automobile Engines

Make and model	Cylinders: Type and no.*	Bore, in (cm)	*B*/*S* ratio	Displacement, in³ (L)	No. of valves per cyl.	Comp. ratio	Published maximums: bhp	r/min	bhp/L	Car weight, lb (kg)
Chrysler Corp.										
Chrysler LeBaron	SOHC V-6	3.55 (9.02)	1.19	181 (3.0)	2	8.9	141	5,000	47	3,122 (1,416)
Chrysler New Yorker	SOHC V-6	3.78 (9.60)	1.19	215 (3.5)	4	10.5	214	5,800	61	3,457 (1,568)
Dodge Intrepid	OHV V-6	3.66 (9.30)	1.15	201 (3.3)	2	8.9	153	5,300	46	3,217 (1,459)
Plymouth Neon	SOHC I-4	3.45 (8.76)	1.06	122 (2.0)	4	9.8	132	6,000	66	2,320 (1,052)
Ford Motor Co.										
Ford Escort	SOHC I-4	3.23 (8.20)	0.93	114 (1.9)	2	9.0	88	4,400	46	2,304 (1,045)
Ford Mustang GT	OHV V-8	4.00 (10.2)	1.33	302 (5.0)	2	9.0	215	4,200	43	3,258 (1,478)
Lincoln Town Car	SOHC V-8	3.60 (9.14)	1.00	281 (4.6)	4	9.0	210	4,600	46	4,039 (1,832)
Ford Taurus†	DOHC V-6	3.50 (8.89)	1.11	182 (3.0)	4	10.0	200	5,750	67	3,104 (1,408)
Taurus SHO†	DOHC V-8	3.24 (8.24)	1.04	207 (3.4)	4	10.0	235	6,100	69	3,150 (1,429)
Contour†	DOHC I-4	3.30 (8.38)	0.94	121 (2.0)	4	9.6	125	5,500	63	2,500 (1,135)
General Motors										
Buick Century	OHV I-4	3.50 (8.89)	1.01	133 (2.2)	2	9.0	120	5,200	55	2,974 (1,349)
Cadillac Eldorado	DOHC V-8	3.66 (9.30)	1.11	279 (4.6)	4	10.3	270	5,600	59	3,774 (1,712)
Chevrolet Cavalier	OHV I-4	3.50 (8.89)	1.01	133 (2.2)	2	9.0	120	5,200	55	2,509 (1,138)
Chevrolet Geo†	SOHC I-4	2.91 (7.40)	0.98	79 (1.3)	2	9.5	70	5,500	54	1,949 (884)
Chevrolet Lumina†	DOHC V-6	3.62 (9.20)	1.10	207 (3.4)	4	9.7	215	5,200	63	3,309 (1,501)
Oldsmobile Cutlass	OHV V-6	3.50 (8.89)	1.06	191 (3.1)	2	9.5	160	5,200	52	3,086 (1,400)
Pontiac Formula	OHV V-8	4.00 (10.2)	1.15	350 (5.7)	2	10.5	275	5,000	48	3,425 (1,553)
Pontiac Grand Am	DOHC I-4	3.63 (9.22)	1.08	138 (2.3)	4	10.0	175	6,200	76	2,822 (1,280)
Saturn	SOHC I-4	3.23 (8.20)	0.91	116 (1.9)	2	9.3	85	5,000	45	2,314 (1,050)

* I = in-line, V = vee type.
† From 1996 Manufacturers Motor Vehicle Specifications.
SOURCE: *Wards Automotive*, 1994; *Auto Ind.*, July 1994.

Exhaust valve seats are hardened to prevent recession with unleaded fuel.

It is standard practice to provide pistons with three rings above the wrist pin—two narrow compression rings above one oil scraper ring, with several drain holes from the back of the ring groove to return oil to the crankcase. All compression rings are narrow, are made of cast iron from 0.063 to 0.0787 in (0.160 to 0.200 cm) wide, and have a wear-resistant coating, such as chromium (particularly for the top ring) and tin, or compounds (iron oxide, phosphates), which minimizes the running-in period without danger of scuffing or scoring the cylinders. Oil scraper rings are of steel or cast iron, generally are a little less than 3⁄16 in (0.476 cm) wide, and are provided with internal expander springs to increase pressure against the cylinder walls. Many oil scraper rings, especially those of steel, also use wear-resistant coatings.

Liquid coolant under a pressure of 15 lb/in² (103 kPa) is standard for U.S. cars. Coolant jacket temperatures are controlled by thermostats which open at approximately 195°F (90.6°C), permitting coolant from the cylinder heads to flow to the radiator. All engines either are designed with coolant all around each cylinder or are ''siamesed'' and with jackets the full length of the piston travel.

FOREIGN AUTOMOBILE ENGINES

Foreign automobile engines sold in the United States are increasingly similar to engines produced in the United States. (See Table 9.6.4.) These similarities include size and basic design features. They are mostly four-stroke, Otto cycle engines; the rest are diesels. All engines are liquid-cooled. Engines produced for areas of the world other than

Table 9.6.4 Selected Foreign Automobile Engines

Make and model	Cylinders: Type and no.*	Bore, in (cm)	*B*/*S* ratio	Displacement, in³ (L)	No. of valves per cyl.	Comp. ratio	Published maximums: bhp	r/min	bhp/L	Car weight, lb (kg)
Audi 90 and 100	DOHC V-6	3.25 (8.26)	0.96	169 (2.8)	4	10.3	172	5,500	61	3,400 (1,542)
BMW325i	DOHC I-6	3.31 (8.41)	1.12	152 (2.5)	4	10.5	189	5,900	76	3,087 (1,400)
Chrysler Colt	SOHC I-4	3.19 (8.10)	0.91	112 (1.8)	4	9.5	113	6,000	63	2,195 (996)
Ford Aspire	SOHC I-4	2.79 (7.09)	0.85	81 (1.3)	2	9.7	63	5,000	48	2,004 (909)
Honda Corp.:										
Accord	SOHC I-4	3.35 (8.50)	0.90	132 (2.2)	4	8.8	130	5,300	59	2,800 (1,270)
Civic	SOHC I-4	2.95 (7.49)	0.89	91 (1.5)	4	9.2	102	5,900	68	2,224 (1,009)
Mazda 626	DOHC I-4	3.30 (8.38)	0.92	122 (2.0)	4	9.0	118	5,500	59	2,606 (1,182)
Mercedes 220	DOHC I-4	3.54 (9.0)	1.04	134 (2.2)	4	9.8	110	5,500	50	3,150 (1,429)
Nissan Centra	DOHC I-4	2.99 (7.60)	0.86	97 (1.6)	4	9.5	110	6,000	69	2,346 (1,064)
Porsche 911	SOHC H-6	3.94 (10.0)	1.31	220 (3.6)	2	11.3	247	6,100	69	3,031 (1,375)
Saab 900	DOHC I-4	3.54 (9.0)	1.00	140 (2.3)	4	10.5	155	4,300	67	2,950 (1,338)
Subaru Legacy	SOHC H-4	3.82 (9.70)	1.29	135 (2.2)	2	9.5	130	5,600	59	2,825 (1,280)
Toyota:										
Camry	DOHC I-4	3.43 (8.71)	0.96	134 (2.2)	4	9.5	130	5,400	59	3,086 (1,400)
Tercel	SOHC I-4	2.87 (7.29)	0.84	89 (1.5)	3	9.3	82	5,200	55	1,975 (896)
VW Jetta	SOHC I-4	3.25 (8.26)	0.89	121 (2.0)	2	10.0	115	5,400	58	2,647 (1,200)
Volvo 850	DOHC I-5	3.32 (8.43)	0.92	149 (2.4)	4	10.5	168	6,200	70	3,280 (1,488)

* I = in-line, V = vee type.
SOURCE: *Ward's Automotive*, 1994; *Auto Ind.*, July 1994.

the United States exhibit a broader range of characteristics. Very small cars with engines of less than 40 in^3 (0.66 L) and 35 hp are found in some areas. Engines produced for the U.S. market are slightly larger, on average, in terms of piston displacement, than those used elsewhere. Some European engines, however, have very high specific output in terms of horsepower per unit displacement because of taxation based on engine displacement, higher vehicle speed limits, and less severe pollution requirements. Many countries tax larger engines at a higher rate than smaller engines.

Most engines are normally aspirated, although the use of turbocharging is increasing. Displacements range from three cylinders, 61 in^3 (1.0 L); four cylinders, 82 to 156 in^3 (1.35 to 2.6 L); five-cylinders, 109 to 183 in^3 (1.8 to 3.0 L); six cylinders, 152 to 258 in^3 (2.5 to 4.26 L); eight cylinders, 149 to 412 in^3 (2.46 to 6.8 L).

The maximum horsepower of foreign automobile engines ranges from approximately 50 to over 400 hp, equivalent to the range for U.S. engines. For a number of years, power has decreased because of emphasis on fuel economy and emission controls, but power levels have once more increased in response to consumer demand.

A number of foreign engines were considered ''high tech'' engines, with such features as three and four valves per cylinder, aluminum cylinder heads and blocks, and lightweight precision components. These features are now common in all cars produced worldwide. See Tables 9.6.3 and 9.6.4.

TRUCK AND BUS ENGINES

Truck and bus engines (Figs. 9.6.19 and 9.6.20) are similar to automobile engines but, in general, are larger, having 300- to 1,000-in^3 (4.6- to 16-L) displacement, are more rugged, and run at lower speeds. Both Otto and diesel cycles are used, with the diesel predominant in the larger sizes. The two-cycle blower scavenged diesel engine with or without a turbocharger is in common use (Fig. 9.6.20). Water cooling is universally used. The gasoline engines are naturally aspirated and have compression ratios slightly lower than automobile engines but are valved

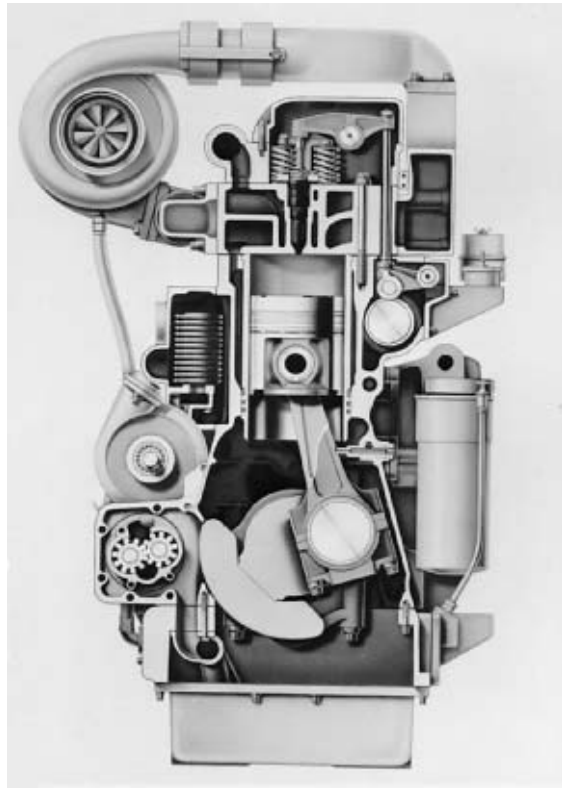


Fig. 9.6.19 Cummins model KT 450-hp turbocharged, four-stroke diesel engine.

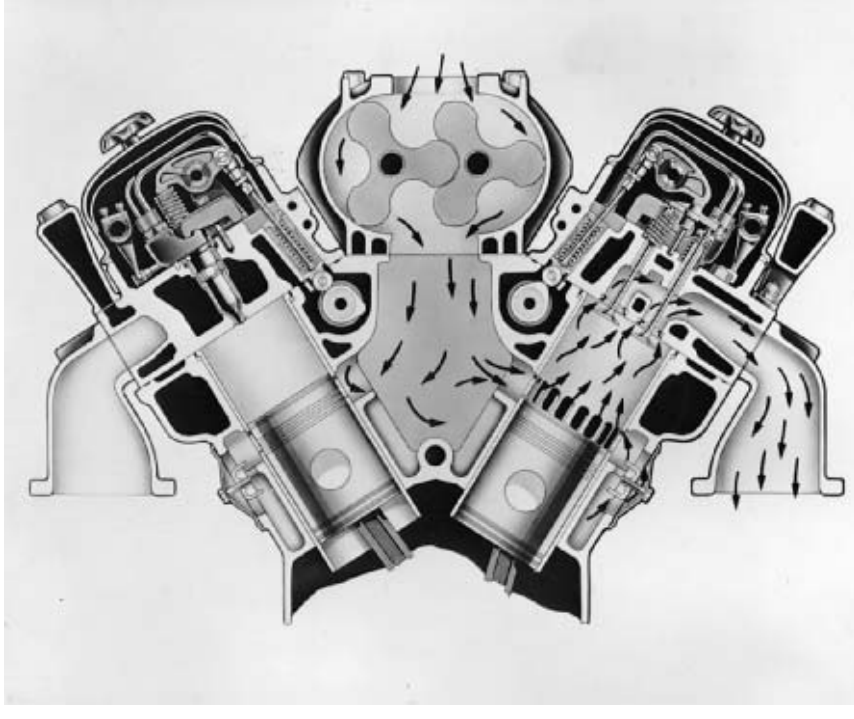


Fig. 9.6.20 Blower-scavenged Detroit diesel model 8V-71N two-stroke diesel engine.

Table 9.6.5 Selected Diesel Engines for Truck, Bus, Tractor, Marine, and Industrial Uses

Make and model	Designed for*	Cylinders No.	Bore, in (cm)	*B*/*S* ratio	Displacement, in³ (cm³)	Compression ratio	Max continuous bhp at r/min		Max torque, lb·ft (N·m), at r/min	
Allis-Chalmers 2800	Tr, Ind	6	3.88 (9.85)	0.91	301 (4,932)	16.25	68	2,200	225 (305)	1,400
Case 336 BD	Tr, Ind	4	4.63 (11.76)	0.93	336 (5,506)	16.5	72	2,200	281 (381)	1,400
Caterpillar D 348	T, Ind	12	5.41 (13.74)	0.83	1,786 (29,267)	16.5	725	1,800	2,840 (3,850)	1,500
Chrysler Nissan IN675	Ind	6	4.33 (10.99)	0.84	452 (7,407)	16	168	1,800	664 (900)	1,400
Cummins V-785-C	Ind	8	5.50 (14.0)	1.33	785 (12,864)	15.9	220	2,300	560 (759)	1,600
Deere 3164 D	Tr, Ind	3	4.02 (10.21)	0.93	164 (2,687)	16.3†	40	2,200	122 (165)	1,500
Detroit Diesel 8V-71T	T, Tr, Ind	8	4.25 (10.80)	0.85	568 (9,308)	17†	350	2,100	965 (1,312)	1,600
Deutz F6L-714	T, B, Tr, Ind	6	4.75 (12.07)	0.86	579 (9,488)	17.8	117	2,000	376 (510)	1,300
Ford 401-D	Tr	6	4.41 (11.20)	1.00	401 (6,571)	16.5	108	2,300	284 (385)	1,400
GMC DH478	T, B	6	5.13 (13.03)	1.33	478 (7,833)	17.5	155	2,800	337 (457)	2,000
Hercules D-3000	Tr, Ind	6	3.75 (9.53)	0.83	298 (4,883)	17.5	85	2,400	244 (331)	1,400
International DT-407	Tr, Ind	6	4.33 (10.9)	0.89	407 (6,670)	17†	130	2,400	393 (533)	1,800
Mack END 673E	T	6	4.88 (12.40)	0.81	672 (11,012)	16.11	168	2,100	540 (732)	1,400
Murphy MP-321	Ind	6	5.75 (14.61)	1.09	1,013 (16,600)	16	207	1,400	946 (1,283)	800
Oliver 1855	Tr	6	3.88 (9.85)	0.89	310 (5,080)	16	98	2,400	298 (404)	1,600
Perkins 6-354	Tr, Ind	6	3.88 (9.85)	0.78	354 (5,801)	16	88	2,250	288 (390)	1,400
MAN DO826	B	6	4.25 (10.8)	0.86	419 (6,871)	16	270	2,400	710 (960)	1,500

* B = bus; T = truck; Ind = industrial; Tr = tractor.
† Turbocharged.

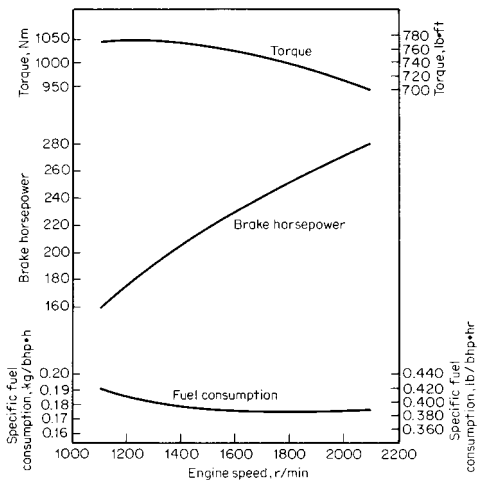


Fig. 9.6.21 Performance data for blower-scavenged Detroit diesel model 8V-71N two-stroke diesel engine. (See Fig. 9.6.20.)

similarly, have similar performance curves, and attain maximum brake horsepower (bhp) (55 to above 300) at a mean piston speed averaging about 2,600 ft/min (13 m/s). Maximum torque is obtained at about 1,500 r/min, appreciably lower than for automobile engines. Diesel engines, which are often supercharged and/or turbocharged, have piston speeds ranging from 1,500 to 2,100 ft/min (7.6 to 11 m/s) and operate at somewhat lower speeds than gasoline engines. Truck and bus gasoline engines can operate at speeds slightly higher than those at which maximum power is attained. Some diesel engines can be operated near the maximum power speed. Most are operated below (Fig. 9.6.21).

Truck and bus engines are usually built with four or six cylinders in line or with eight cylinders in V-type construction. The bore/stroke ratio ranges from about 0.75 to 1.4, the latter engines having the larger values (see Tables 9.6.5 and 9.6.6).

TRACTOR ENGINES

Both spark ignition (gasoline or liquefied petroleum gas) and compression ignition (diesel) engines are used in tractors. Almost all engines above 100 hp are diesels. Most engines are four-cycle, water-cooled, with valve-in-head arrangement. Integral bore, dry-sleeve, or wet-sleeve cylinder liner arrangements are utilized. One-, two-, three-, four-, five-, six-, and eight-cylinder engines are used. All engines except the

Table 9.6.6 Selected Gasoline Engines for Truck, Bus, Tractor, Marine, and Industrial Uses

Make and model	Designed for*	Cylinders No.	Bore, in (cm)	*B*/*S* ratio	Displacement, in³ (L)	Compression ratio	Rating† bhp at r/min		Max torque, lb·ft (N·m), at r/min	
Allis-Chalmers G-153	Ind	4	3.44 (8.74)	0.83	153 (2.5)	8	52	2,400	132 (179)	1,460
Jeep Cherokee	T, Tr, Gp, Ind	I-6	3.88 (9.53)	1.13	242 (4.0)	8.8	190	4,750	215 (292)	1,400
Case 201 G	Tr, Gp, Ind	4	4.0 (10.16)	—	251 (4.1)	7.5	87	2,400	221 (300)	1,400
Chevrolet 350	Gp	V-8	4.0 (10.16)	1.15	350 (5.7)	9.1	210	4,000	—	
Chevrolet 454	T	8	4.25 (10.79)	1.06	454 (7.4)	8.3	250	4,000	365 (495)	2,800
Chrysler H-225	M, Ind	6	3.41 (8.66)	0.83	225 (3.7)	8.4	132	4,000	203 (275)	2,000
Continental R 821-46	Tr, Ind	4	3.03 (7.70)	0.92	95.5 (1.6)	8.6	67.2	5,000	84 (114)	2,400
Diamond Reo 8-250	T, B	8	4.24 (10.79)	1.03	468 (7.7)	7.5	250	3,400	420 (569)	2,400
Dodge CT 900	T	8	4.19 (10.64)	1.12	413 (6.8)	7.5	238	3,600	355 (481)	2,000
Ford 158-G	Tr	3	4.20 (10.67)	1.10	158 (2.6)	7.8	45.7	2,100	120 (163)	1,350
Ford Aerostar	Gp	V-6	3.5 (8.89)	1.11	182 (3.0)	9.2	135	4,600	—	
Ford F150	T	I-6	4.0 (10.16)	1.00	300 (4.9)	8.8	145	3,400	—	
Hercules G-3400	Tr, Ind	6	4.0 (10.16)	0.89	339 (5.6)	7.5	143	2,800	311 (422)	1,400
International V5-401	T	8	4.13 (10.49)	1.10	400 (6.6)	7.7	226	3,600	355 (481)	2,000
Minneapolis-Moline HD 425	Tr	6	4.25 (10.79)	0.85	425 (7.0)	8.1	133	2,000	378 (512)	800
Oliver 1655	Tr	6	3.75 (9.53)	0.94	265 (4.4)	8.5	84	2,200	220 (298)	1,300
Volkswagen 124A	Ind	4	3.38 (8.59)	1.24	96.7 (1.6)	7.7	53	3,600	75.5 (102)	2,500

* B = bus; T = truck; Tr = tractor; M = marine; Gp = general purpose; Ind = industrial.
† hp without accessories.

six-cylinder are usually in-line in order to minimize the width of the hood in the interest of driver visibility. Tractor applications have a high load factor, are relatively low-speed, and deliver high low-speed torque (Fig. 9.6.22). Future off-road engines are expected to be emission-controlled.

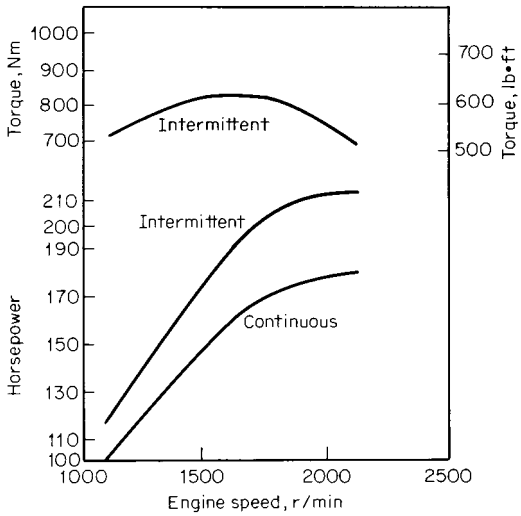


Fig. 9.6.22 Performance of John Deere model 6351 A, a diesel engine.

STATIONARY ENGINES

Stationary engines may be either Otto or diesel cycle, use either liquid or gaseous fuel, and use the two- or four-stroke cycle. The power output ranges up to about 48,000 bhp per engine. Piston rods, which permit the crankcase to be isolated from blowby gases and the bottom face of the piston to function as an air pump, are commonly used. Nearly all engines are now single-acting. In the larger sizes, cylinders are built up of several parts, cylinder liners are used, and long pistons for single-acting diesel engines usually have about five piston rings. The valves are usually massive and in the larger sizes may be cooled. The exhaust pipe is sometimes water-cooled, and the pistons for large-bore engines are always liquid-cooled with oil or water. In general, there is much less integral construction (Fig. 9.6.23) in the large stationary engine than in the automobile type.

An interesting medium-speed, four-stroke diesel engine, V-type design with rotating pistons, is illustrated in Fig. 9.6.23. The symmetric piston is cooled with lubricating oil, and besides the reciprocating motion, it performs a slow rotating motion around its longitudinal axis. At each stroke, a new oil-wetted portion of the piston is always in contact with the pressure side of the liner, thus minimizing wear, oil consumption, and local overheating of the liner resulting from blowby.

Stationary engines usually operate at constant speed and are governed by throttling the charge of the Otto engine and by varying the amount of fuel injected into the diesel engine. Compression ratios as low as 6 : 1 are used with spark ignition engines and as low as 12 : 1 with compression ignition engines. Stationary engines are usually built with 1 to 12 cylinders, with cylinders in a vertical line, or with a V-type arrangement having up to 16 or more cylinders.

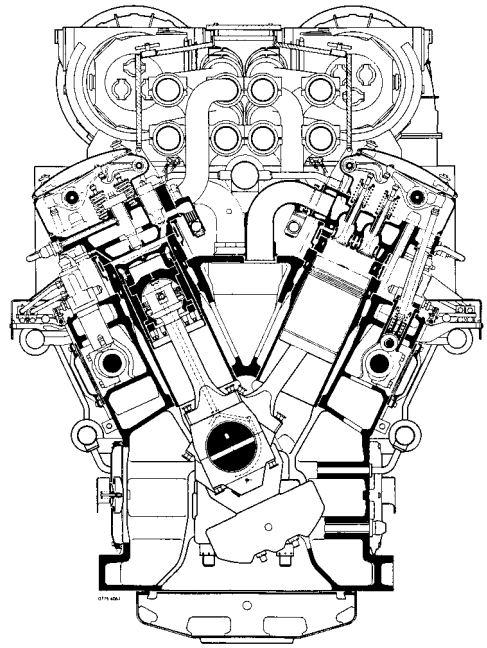


Fig. 9.6.23 Cross section of Sulzer V-type ZV-40 engine. Bore = 400 mm; stroke = 480 mm.

MARINE ENGINES

(See also Sec. 11.3)

Marine engines may be either Otto or diesel cycle and may be normally aspirated or supercharged. Outboard engines range in power from less than 1 to more than 200 bhp and are generally of the two-cycle gasoline type. Modified automotive gasoline engines are also used in larger boats. Diesel engines range in power from below 20 to 48,000 bhp and may be either two- or four-cycle. Both automotive and stationary types are used in marine work; consequently, wide variations in specific weights are found. (See Tables 9.6.5, 9.6.6, and 9.6.7).

The outboard two-cycle engines (see Table 9.6.7) have crankcase compression (Fig. 9.6.24) and may have two ports, three ports, or two

Table 9.6.7 Selected Outboard Engines

Make and model	Cylinders: Type and no.*	Bore, in (cm)	*B*/*S* ratio	Displacement, in^3 (cm^3)	Rating, bhp at r/min		Weight, lb (kg)
Evinrude Yachtwin 6	I-2	1.937 (4.92)	1.14	10 (164)	6	5,000	56 (25.5)
Evinrude 25	I-2	3.000 (7.62)	1.33	31.8 (521)	25	5,000	117 (53.6)
Evinrude 90	V-4	3.500 (8.89)	1.35	99.6 (1,632)	90	5,000	301 (136)
Yamaha Force 9.9	I-2	2.250 (5.71)	1.16	14.2 (232)	9.9	5,050	91 (41.5)
Yamaha Force C85	I-3	3.313 (8.41)	1.18	69.6 (1,140)	85	5,000	248 (113)
Johnson 15	I-2	2.375 (5.56)	1.35	15.6 (255)	15	5,500	77 (35)
Johnson 225	V-6	3.685 (8.89)	1.29	183.3 (3,000)	225	5,500	470 (212)
Mercury 8	I-2	2.13 (5.4)	1.20	12.8 (210)	8	5,000	69 (31)
Mercury 100	I-4	3.5 (8.9)	1.19	113 (1,848)	100	5,000	348 (158)
Mercury 200	V-6	3.5 (8.9)	1.32	153 (2,507)	200	5,500	422 (192)

* I = in-line, V = vee type.
SOURCE: Manufacturer's publications.

ports with a rotary or reed crankcase inlet valve for the highest outputs (Fig. 9.6.25). Otto four-cycle engines have either L-head or valve-in-head construction, and diesel four-cycle engines are invariably of valve-in-head construction.

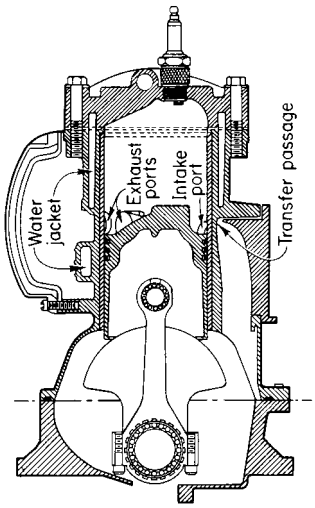


Fig. 9.6.24 Johnson two-cycle, 30-hp outboard motor.

Regular gasoline is required for Otto cycle engines, the fuel and lubricating oil being mixed for outboard two-cycle engines. A comparison of a number of fuel-consumption curves for various types of engine shows the large low-speed marine diesel engine as having the lowest specific fuel consumption (Fig. 9.6.26).

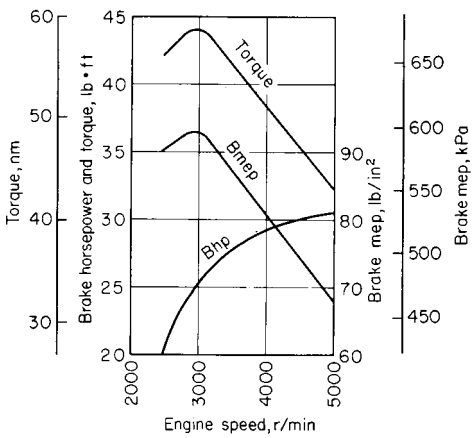


Fig. 9.6.25 Performance of Johnson outboard motor.

Low-speed engines are directly connected to the propellers, and high-speed engines are connected through reduction gearing. Outboard engines have maximum speeds of 4,500 to 6,000 r/min with the piston speeds ranging from 1,000 to about 2,500 ft/min (5.0 to 12.7 m/s).

Maximum engine speed is obtained at the intersection of the propeller/horsepower and the maximum horsepower curves (Fig. 9.6.27). This may occur well below the speed for maximum horsepower or beyond this speed. The effect of throttling to reduce speed is to increase the specific fuel consumption considerably.

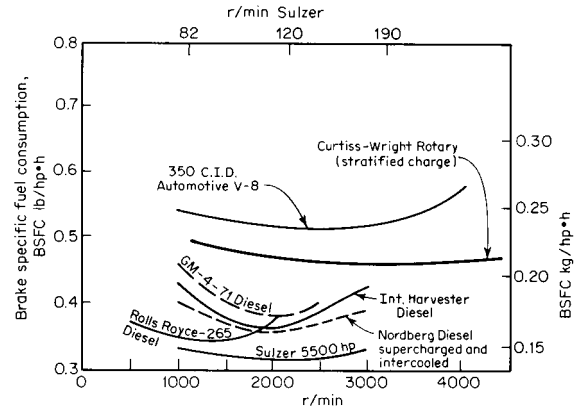


Fig. 9.6.26 Specific fuel consumption curves.

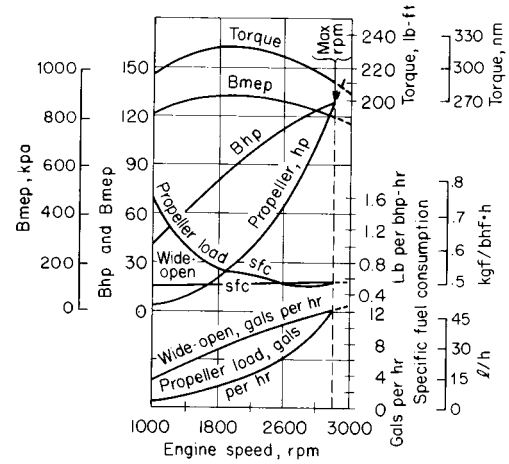


Fig. 9.6.27 Performance of six-cylinder $3\frac{7}{16} \times 4\frac{3}{4}$ in (8.73 × 12.07 cm) Chrysler marine engine with 7:1 compression ratio.

SMALL INDUSTRIAL, UTILITY, AND RECREATIONAL VEHICLE GASOLINE ENGINES

The Otto cycle engine with two or fewer cylinders is predominant in this class (Table 9.6.8). Fuel economy is generally not a major concern, whereas simplicity and performance are important. For reciprocating engines, the L-head design is predominant. Both high- and low-speed engines are used, depending on the application. Engines of this type generally employ air cooling of the cylinder and charge cooling of the interior. Water or liquid cooling is used on a few high-output small engines. Higher-mep air-cooled engines used forced fan cooling. Two-stroke engines are often used in applications where minimum weight and cost are important.

Carburetion, ignition systems, and other auxiliary equipment are usually less sophisticated than those on larger engines. A given engine is often used in a wide range of applications.

Engines used in motorcycles and in all-terrain vehicles are generally very sophisticated and constructed with great precision. Features such as overhead valves and overhead cams are common. The specific output of these engines is often high as measured by power per unit of piston displacement.

Table 9.6.8 Selected Small Gasoline Engines

Make and model	Cycles	Type	Cylinders: No.	Cylinders: Bore, in (cm)	Cylinders: *B*/*S* ratio	Displacement, in³ (cm³)	Compression ratio	Valve location*	Continuous rating, bhp at r/min	Max torque, lb·ft (N·m), at r/min	Weight, lb (kg)
Briggs & Stratton 100900	4	V	1	2.5 (6.50)	1.17	10.43 (171)	6.2	L	3.40 3,600	5.9 (8.0) 3,100	30.5 (13.9)
Briggs & Stratton 80400	4	H	1	2.38 (6.04)	1.36	7.75 (127)	6.2	L	2.55 3,600	4.6 (6.2) 3,100	25.25 (11.9)
Chrysler 500	2	H, V	1	2.00 (6.45)	1.26	5.00 (82)	6.5		3.33 7,000	3 (4.1) 5,000	8.75 (3.9)
Homelite XL-123	2	H	1	1.81 (4.60)	1.31	3.60 (59)	8.3				13.0 (5.9)
Kohler MV18	4	H, V	2	3.12 (7.92)	1.13	42.2 (692)	6	L	18 3,600	26 (35) 2,600	130 (59)
O & R 13A	2	V	1	1.25 (3.18)	1.15	1.34 (22)	9		1.0 6,300	0.88 (1.2) 5,200	3.75 (1.70)
Onan AI	4	V	1	2.75 (6.99)	1.10	14.90 (244)	6.25	L	3.86 3,600	8 (10.8) 2,100	150 (68.2)
Rockwell Zf-400	2	V	2	2.56 (6.50)	1.00	24.29 (398)	7.5		24.0 6,250	28 (38) 6,250	62 (28.2)
Tecumseh LAV 35-1A	4	V	1	2.50 (6.35)	1.36	9.06 (149)	—	L	2.27 2,400	5.25 (7.12) 3,100	21.5 (9.77)
Tecumseh HH160	4	H	1	3.5 (8.89)	1.22	27.66 (453)	8.7	I	16.0 3,600	25 (33.9) 2,400	84 (38.2)
WE400 Wisconsin TRA-10D	4	V	1	3.3 (8.4)	1.20	23.7 (388)	6.5	L	10.0 3,400	16.5 (22.4) 2,800	80 (36.4)
Fichtel & Sachs KM 914	4	H	1	—	—	18.5 (303)	8	RC	20 5,000	— —	56 (25.5)

* I = valve in head; L = L head; RC = rotary combustion.
SOURCE: *Auto. Ind.*, **48**, no. 7, Apr. 1, 1973, pp. 104–106.

Table 9.6.9 Selected Medium-Size Diesel Engines*

Make and model	Cylinder arrangement†	Cycles	Cylinders: No.	Cylinders: Bore, in (cm)	Cylinders: *B*/*S* ratio	Displacement, in³ (L)	Compression ratio	Max rating, bhp. at r/min	Weight, lb (kg)
Alco 251	V	4	12	9.0 (22.86)	0.86	8,016 (131.4)	11.5	3,150 1,000	35,581 (16,152)
DeLaval GVB-16	V	4	16	13.0 (33.0)	0.87	31,856 (522.0)	—	2,900 1,000	141,000 (64,005)
Electro-Motive 12-645	V	2	12	9.125 (22.0)	0.92	7,740 (128.4)	16	2,950 950	28,600 (11,350)
Fairbanks-Morse 12-38D 1/8	OP	2	12	8.13 (20.65)	0.81	12,443 (203.9)	16.1	3,840 900	45,000 (20,430)
White-Superior 40-VX-12	V	4	12	10.0 (25.4)	0.95	9,896 (162.2)	—	1,800 1,000	—

* ''Diesel and Gas Turbine Catalog,'' 1994. All engines are turbocharged.
† V = V type; OP = opposed.

LOCOMOTIVE ENGINES

Diesel engines are used for locomotive service because of high thermal efficiency (overall diesel brake thermal efficiency of about 38 to 40 percent with the net locomotive traction efficiency of about 32 percent for a diesel locomotive, compared with 6 percent for a steam locomotive), high availability for service, elimination of destructive pounding of rails, and elimination of shutdown fuel losses. The locomotive diesels are usually lighter, have lower displacement per cylinder, but higher outputs per cubic inch of displacement than the corresponding stationary diesel engine. These engines are built with 4 to 20 cylinders. Usually one engine is used per locomotive, and one or more locomotives are used per train. On an experimental basis, locomotive diesel engines have used coal-water slurries and natural gas as fuel. The convenience and low cost of conventional liquid diesel fuel will cause liquid fuels to predominate into the foreseeable future. (See Table. 9.6.9.)

AIRCRAFT ENGINES

Small aircraft piston engines are air-cooled and invariably four-stroke. All types are naturally aspirated or supercharged. These engines have the lowest specific weight of piston engines (Table 9.6.10). Aircraft engines always have valve-in-head construction; they also have low specific fuel consumption, and the guaranteed minimum may be as low as 0.40 lb (0.18 kg)/(bhp · h). Maximum cruising bmep ranges from 100 to 150 lb/in^2 (690 to 1,030 kPa), while takeoff bmep ranges from 120 to 200 lb/in^2 (830 to 1,380 kPa), depending principally on the compression ratio, fuel, and supercharge.

Aircraft engines are valved to obtain high mean effective pressure at rated speed. High-output engines usually operate with compression ratios and supercharging levels that prohibit full-throttle operation at or near sea level with the specified fuel. Such engines can be operated at full throttle at or above a critical altitude, beyond which the power decreases with an increase in altitude. Below the critical altitude, the power decreases with a decrease in altitude, at constant intake-manifold pressure, because of the increase in exhaust back pressure and ambient temperature. Turbocharged engines, however, maintain essentially constant-power with constant manifold pressure from sea level to a critical altitude, which can be as high as 20,000 ft (6,100 m).

The performance of an aircraft engine varies with speed, manifold pressure, and altitude. At any given speed and with a fixed supercharger or blower ratio (supercharger r/min to engine r/min) and wide-open throttle, the bhp of an engine varies almost linearly with the density of the atmosphere. This characteristic makes it possible to estimate the power curves at altitude if the sea-level output is known. The power at 20,000-ft (6,100-m) altitude is about 50 percent of the sea-level wide-open-throttle output for any given speed.

Low specific fuel consumption is obtained by operating with reduced r/min for the desired power with a lean mixture setting, the optimum output at this point being about one-half to two-thirds of the normal rated output of the engine.

Aircraft engine cylinders are limited to a maximum diameter of about 6 in (0.15 m) by piston cooling and knocking difficulties that arise with large cylinders, high compression ratios, and high intake manifold pressures. The bore/stroke ratio is commonly above 1.0. The larger piston engine sizes are being replaced by turbine-type engines.

Piston aircraft engines are mostly of the horizontally opposed design (desirable profile) with an even number of cylinders, usually four, six, or eight. They are equipped with a dual ignition system, resulting in more reliable combustion.

WANKEL (ROTARY) ENGINES

The Wankel (rotary) SI engine is used in the Mazda RX series. A triangular rotor rotates on an eccentric shaft inside an epitrochoidal housing. The rotor tips are in constant contact with the housing and form three working chambers. The operating cycle (Fig. 9.6.28) follows the four-stroke cycle principle, although the output shaft rotates at 3 times rotor speed, providing one power stroke per crankshaft revolution (as in the single-cylinder two-cycle). A fixed gear meshes with an internal gear in the rotor to maintain the proper relationship between the rotor and housing. Housing cooling is by either water or air and rotor cooling by oil or fuel/air charge. Auxiliary equipment (fuel system, ignition system, etc.) is similar to that used on piston engines. A major problem is the sealing grid (90° intersection of seals and single-line seal at apexes of rotor), and this tends to produce high hydrocarbon emissions.

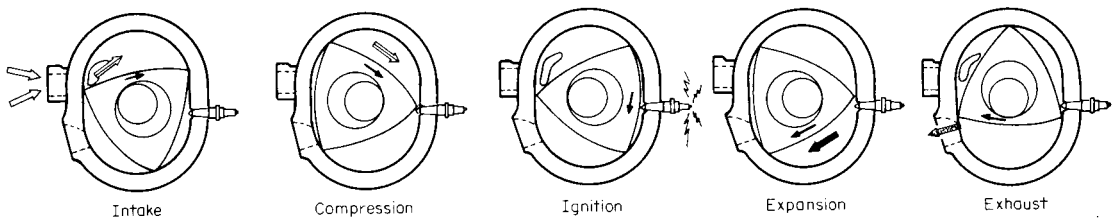


Fig. 9.6.28 Operating cycle of Wankel rotating combustion engine.

Table 9.6.10 Selected Piston-Type Aircraft Engines

Make and model	Cylinders: No.	Arrangements*	Bore, in (cm)	Stroke, in (cm)	Displacement, in^3 (cm^3)	Ratings (METO)†: Comp. ratio	bhp‡	r/min	At sea level (SL) or altitude, ft	Weight§: bmep at max bhp, lb/in^2 (kPa)	Engine, dry, lb (kg)	Fuel octane no. required
Avco Lycoming												
0-320-E	4	H	5.13 (13.03)	3.88 (9.86)	319.8 (5,240.6)	7.0	150	2,700	SL	138 (952)	244 (110.4)	80/87
I0-360-A	4	H	5.13 (13.03)	4.38 (9.86)	361.0 (5,915.7)	8.7	200	2,700	SL	163 (1,120)	293 (132.6)	100/130
TIGO-541-E	6	H	5.13 (13.03)	4.38 (9.86)	541.5 (8,873.6)	7.3	425	3,200	15,000	195 (1,340)	700 (316.8)	100/130
Teledyne Continental												
0-200	4	H	4.06 (10.31)	3.88 (9.86)	201 (3,293.8)	7.1	100	2,750	SL	143 (986)	190 (86.0)	80/87
I0-470	6	H	5.0 (12.7)	4.0 (10.16)	471 (7,718.3)	8.6	260	2,600	SL	166 (1,140)	465 (210.5)	100/130
GTSIO-520-F	6	H	5.25 (13.34)	4.0 (10.16)	520 (8,521.3)	7.5	435	3,400	19,000	195 (1,340)	647 (292.8)	100/130

* H = horizontal (production of radial and vertical cylinder arrangements has been discontinued).
† METO = maximum except during takeoff.
‡ Takeoff power is usually higher than the recommended maximum continuous power.
§ Weights are for dry engines without hub or starter.

Table 9.6.11 Data on Fuel Properties

Fuel	Formula (phase)	Molecular weight	Specific gravity (density* kg/m^3)	Heat of vaporization, kJ/kg†	Specific heat		Higher heating value, MJ/kg	Lower heating value, MJ/kg	LHV of stoich. mixture, MJ/kg	$(A/F)_s$	$(F/A)_s$	Fuel octane rating	
					Liquid, kJ/(kg·K)	Vapor c_p, kJ/(kg·K)						RON	MON
Practical fuels													
Gasoline	$C_nH_{1.87n}(l)$	~110	0.72–0.78	305	2.4	~1.7	47.3	44.0	2.83	14.6	0.0685	92–98	80–90
Light diesel	$C_nH_{1.8n}(l)$	~170	0.84–0.88	270	2.2	~1.7	44.8	42.5	2.74	14.5	0.0690	—	—
Heavy diesel	$C_nH_{1.7n}(l)$	~200	0.82–0.95	230	1.9	~1.7	43.8	41.4	2.76	14.4	0.0697	—	—
Natural gas‡	$C_nH_{3.8n}N_{0.1n}(g)$	~18	(~0.79†)	—	—	~2	50	45	2.9	14.5	0.069	—	—
Pure hydrocarbons													
Methane	$CH_4(g)$	16.04	(0.72*)	509	0.63	2.2	55.5	50.0	2.72	17.23	0.0580	120	120
Propane	$C_3H_8(g)$	44.10	0.51 (2.0*)	426	2.5	1.6	50.4	46.4	2.75	15.67	0.0638	112	97
Isooctane	$C_8H_{18}(l)$	114.23	0.692	308	2.1	1.63	47.8	44.3	2.75	15.13	0.0661	100	100
Cetane	$C_{16}H_{34}(l)$	226.44	0.773	358		1.6	47.3	44.0	2.78	14.82	0.0675	—	—
Benzene	$C_6H_6(l)$	78.11	0.879	433	1.72	1.1	41.9	40.2	2.82	13.27	0.0753		115
Toluene	$C_7H_8(l)$	92.14	0.867	412	1.68	1.1	42.5	40.6	2.79	13.50	0.0741	120	109
Alcohols													
Methanol	$CH_4O(l)$	32.04	0.792	1,103	2.6	1.72	22.7	20.0	2.68	6.47	0.155	106	92
Ethanol	$C_2H_6(l)$	46.07	0.785	840	2.5	1.93	29.7	26.9	2.69	9.00	0.111	107	89
Other fuels													
Carbon	$C(s)$	12.01	~2‡	—	—	—	33.8	33.8	2.70	11.51	0.0869		
Carbon monoxide	$CO(g)$	28.01	(1.25*)	—	—	1.05	10.1	10.1	2.91	2.467	0.405		
Hydrogen	$H_2(g)$	2.015	(0.090*)	—	—	1.44	142.0	120.0	3.40	34.3	0.0292		

(l) = liquid phase; (g) = gaseous phase; (s) = solid phase; RON = research octane number; MON = motor octane number.
* Density in kg/m^3 at 0°C and 1 atm.
† At 1 atm and 25°C for liquid fuels; at 1 atm and boiling temperature for gaseous fuels.
‡ Typical values.
SOURCES: E. M. Goodger, ''Hydrocarbon Fuels; Production, Properties and Performance of Liquids and Gases,'' Macmillan, London, 1975. E. F. Obert, ''Internal Combustion Engines and Air Pollution,'' Intext Educational Publishers, 1973. C. F. Taylor, ''The Internal Combustion Engine in Theory and Practice,'' vol. 1, MIT Press, 1966. J. W. Rose and J. R. Cooper (eds.), ''Technical Data on Fuel,'' 7th ed., British National Committee, World Energy Conference, London, 1977.

Advantages include small size, low weight, simple and fewer components (compared with piston-type four-cycle engines), superior breathing, no valves (rotor and its seals serve as valves), lower friction, low-octane fuel requirement, low NO emissions, and no reciprocating unbalance.

FUELS

Fuels for internal combustion engines are predominantly petroleum products. Natural and manufactured gases are also used in limited quantities. Data on properties of fuels which are commonly used for internal combustion engines are summarized in Table 9.6.11. Additional details on fuels are given in Sec. 7.1.

Gasoline

Gasoline consists of various amounts of many hydrocarbons, each having its vapor, pressure, and temperature characteristics. Also included are small amounts of additives such as knock suppressers, deposit modifiers, antioxidants, metal deactivators, antirust agents, anti-icing agents, detergents, upper cylinder lubricants, and dyes. Different grades of gasoline are marketed on the basis of octane number. The overall volatility of a gasoline sample is specified in U.S. industry by the ASTM Distillation Test (D86) and the ASTM Vapor Pressure Test (D323, Reid Method). The distillation test records the increasing temperature of the vapor in the neck of a flask vs. percent evaporated as 100 mL of fuel is gradually heated and distilled. The vapor pressure test gives the pressure in a split-chamber bomb which contains the fuel sample and room air in thermal equilibrium with a 100°F (311 K) temperature bath. The pressure, corrected to standard initial bomb air conditions, gives the **Reid vapor pressure (RVP)** of the gasoline. Typical results for summer- and winter-grade gasolines are given in Table 9.6.12.

In general, volatility is specified by giving the initial boiling point (IBP); the 10, 50, and 90 percent evaporation points; and the endpoint (EP) temperatures as well as the Reid vapor pressure. These data indicate the effects of fuel volatility on engine performance, e.g., ease of starting, warm-up, and fuel economy (Figs. 9.6.29 and 9.6.30.)

Reformulated Gasoline (RFG) The composition of gasoline is known to affect the exhaust and evaporative emissions. The C_4 and C_5 paraffins and olefins (photochemically active) are the main contributors to evaporative losses. Reformulated gasolines have much reduced amounts, and thus lower vapor pressures (Reid vapor pressure maximum of 48 kPa) and a maximum of 6 percent by volume olefins. Benzene, a toxic compound, is limited to a maximum of 0.85 percent by volume, while aromatics (photochemically active, deposits) as a class are limited to 25 percent by volume. Oxygenated fuel components (methanol, ethanol, etc.) are added at a level of 1.8 to 2.2 percent by weight. This reduces CO somewhat, especially from older vehicles without feedback fuel controls. For the federal RFG standards, sulfur is limited to 130 ppm; this lowers acid emissions and lengthens engine life. Further, the ASTM 90 percent distillation temperature is limited to 165°C. This minimizes deposits and leads to less enrichment required upon cold starting. Such a fuel may give fewer miles per gallon. The California Air Resources Board (CARB) applies even more stringent limits, with sulfur held to 30 ppm and the 90 percent distillation temperature limited to 143°C.

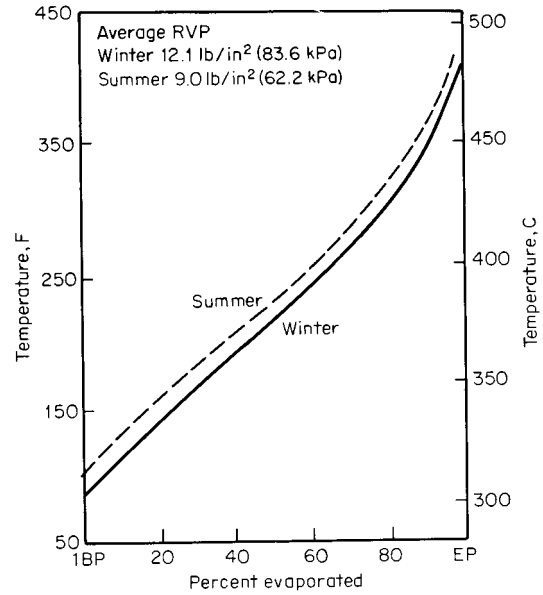


Fig. 9.6.29 Distillation curves for typical winter and summer gasolines. *(Ethyl Corp.)*

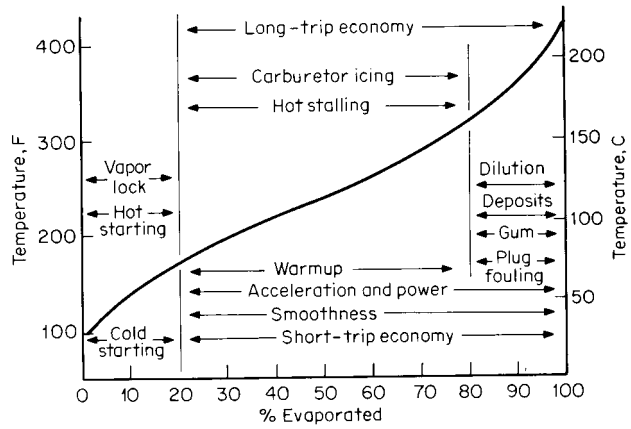


Fig. 9.6.30 Effects of volatility characteristics on engine performance.

Diesel Fuels Automotive and railroad diesel fuels are derived from petroleum refinery products which are referred to as **middle distillates;** the latter have a higher boiling range than gasoline. The properties of commercial distillate diesel fuels depend on the refinery practices employed and the nature of the crude oil from which they are derived, and thus vary substantially. Such fuels generally boil over a range between 163 and 371°C (325 and 700°F). Their makeup can represent various

Table 9.6.12 Average Volatility Characteristics of U.S. Motor Gasolines

	Summer		Winter	
	Premium	Regular	Premium	Regular
IBP temp, °F (°C)	93 (34)	94 (34)	82 (28)	82 (28)
10% temp, °F (°C)	126 (52)	125 (52)	107 (42)	107 (42)
50% temp, °F (°C)	216 (102)	207 (97)	207 (97)	198 (92)
90% temp, °F (°C)	321 (160)	338 (170)	316 (157)	333 (167)
Endpoint temp, °F (°C)	402 (205)	414 (212)	397 (203)	407 (209)
RVP, lb/in² (kPa)	8.9 (61.3)	8.8 (60.6)	12.3 (84.8)	12.1 (83.4)

Table 9.6.13 Detailed Requirements for Diesel Fuel Oils[a,h,i] (ASTM D975)

Grade of diesel fuel oil	Flash point, °C (°F) Min	Cloud point °C (°F) Max	Water and sediment, vol % Max	Carbon residue on, 10 percent residuum, % Max	Ash, wt % Max	Distillation temperatures, °C (°F), at 90% point Min	Distillation temperatures, °C (°F), at 90% point Max	Viscosity kinematic, cSt (mm^2/s), at 40°C Min	Viscosity kinematic, cSt (mm^2/s), at 40°C Max	Viscosity, Saybolt, SUS at 100°F Min	Viscosity, Saybolt, SUS at 100°F Max	Sulfur,[d] wt % Max	Copper strip corrosion Max	Cetane no. Min
No. 1-D—a volatile distillate fuel oil for engines in service requiring frequent speed and load changes	38 (100)	[b]	0.05	0.15	0.01	—	288 (550)	1.3	2.4	—	34.4	0.50	No. 3	40[f]
No. 2-D—a distillate fuel oil of lower volatility for engines in industrial and heavy mobile service	52 (125)	[b]	0.05	0.35	0.01	282[c] (540)	338 (640)	1.9	4.1	32.6	40.1	0.50	No. 3	40[f]
No. 4-D—a fuel oil for low- and medium-speed engines	55 (130)	[b]	0.05	—	0.10	—	—	5.5	24.0	45.0	125.0	2.0	—	30[f]

[a] To meet special operating conditions, modifications of individual limiting requirements may be agreed upon between purchaser, seller, and manufacturer.

[b] It is unrealistic to specify low-temperature properties that will ensure satisfactory operation on a broad basis. Satisfactory operation should be achieved in most cases if the cloud point (or wax appearance point) is specified at 6°C (10°F) above the tenth percentile minimum ambient temperature for the area in which the fuel will be used. The tenth percentile minimum ambient temperatures for the months of October, November, December, January, February, and March are given in ASTM D 975 in maps for the 48 contiguous states and Alaska. This guidance is of a general nature; some equipment designs, use of flow improver additives, fuel properties, and/or operations may allow higher or require lower cloud point fuels. Appropriate low-temperature operability properties should be agreed on between the fuel supplier and purchaser for the intended use and expected ambient temperatures.

[c] When cloud point less than − 12°C (10°F) is specified, the minimum viscosity shall be 1.7 cSt and the 90% point shall be waived.

[d] In countries outside the U.S., other sulfur limits may apply.

[e] Where cetane number by method D 613 is not available, method D 976 may be used as an approximation. Where there is disagreement, method D 613 shall be the preferred method.

[f] Low atmospheric temperatures as well as engine operation at high altitudes may require system use of fuels with higher cetane ratings.

[g] Millimeter squared per second (official SI unit) (Note: 1 cSt = 1 mm^2/s).

[h] The values in SI units are to be regarded as the standard. The values in U.S. Customary system units are for information only.

[i] Nothing in this specification shall preclude observance of federal, state, or local regulations which may be more restrictive.

combinations of volatility, ignition quality, viscosity, sulfur level, gravity, and other characteristics. Additives may be used to enhance specific properties. The SAE recommended practices are described in SAE J313 (March 1992). Permissible limits of significant diesel fuel properties are summarized in ASTM's classification chart D975, reproduced as Table 9.6.13.

Alternative Fuels Other than gasoline and diesel fuels, alternative fuels used in various parts of the world include propane, frequently referred to as LPG; hydrous ethyl alcohol (ethanol), particularly in Brazil; gasohol, a blend of 10 percent by volume of gasoline and 90 percent by volume of denatured anhydrous ethanol. Other possible fuels include compressed or liquefied natural gas (CNG or LNG)—primarily methane, other alcohols—particularly methanol, which is also used in racing, and biomass fuels. Details can be found in SAE J1297.

GAS EXCHANGE PROCESSES

Intake Manifolds The intake manifold distributes the air (diesel) or the air and fuel (Otto) to various cylinders of a multicylinder engine. Intake manifolds are **streamlined** (always with diesel), the **rake type,** or a combination of both. The sections may be either circular or rectangular. In some cases, the circular sections are combined with a flat **floor** or bottom to provide more surface on which the liquid fuel may spread and be vaporized. **Dams** are also used to prevent liquid fuel from flowing along the manifold floor. In carbureted engines, the updraft manifold should have a **riser velocity** of not less than 40 ft/s (12 m/s) to entrain or lift the liquid fuel particles from the carburetor to the **distributor** section of the manifold. This determines the minimum wide-open throttle speed for satisfactory operation. The downdraft manifold permits larger areas and lower velocities between the carburetor and manifold.

The overlapping of the intake valve periods produces complicated disturbances in the intake manifold and has resulted in the use of dual manifolds in some engines with six or more cylinders. V-8 automobile engines have complicated dual manifolds, integrally cast, with each manifold usually feeding two cylinders of each bank. Port fuel injection permits extensive manifold tuning because the wall wetting produced by the carburetor is eliminated. Rotary engines have a simple intake system requirement. A separate carburetor or fuel injector may be used to feed each chamber.

Preheating the intake air and controlling its temperature are desirable in that they reduce air density variation and thereby allow better control of the mixture ratio for economy and emission control. In carbureted engines, carburetor icing is thereby minimized and the need for an exhaust manifold crossover valve reduced. However, air preheating is undesirable in that it reduces volumetric efficiency, heats the carburetor, and evaporates fuel in the float chamber, and may increase intake system coking. While preheating the fuel is also undesirable in that the light fuel fractions will be lost by vaporization, many intake manifolds for liquid fuels have a **hot spot** which receives heat from the exhaust (usually thermostatically controlled) and on which the fuel particles impinge upon leaving the manifold riser. The dew points or condensation temperatures for fuel in an air/fuel mixture depend on the fuel, air/fuel ratio, and total pressure (Table 9.6.14), and they may be determined from equilibrium air distillation data.

Exhaust Manifolds In addition to adequate flow area, the most important consideration in exhaust manifold design is expansion. Small multicylinder engines employ a one-piece exhaust manifold. This is anchored at the center of the engine, and elongated holes are provided at other points for expansion. Large engines employ multipiece manifolds with expansion joints. They are usually water-cooled in marine applications to prevent strain and for safety reasons. In turbocharged engines, exhaust manifolds are usually insulated to conserve energy.

Manifold flow area should be large enough to avoid local back pressure buildup during exhaust and in connection with the total exhaust system should not raise back pressure excessively at maximum flow. A flow area of 0.7 times the intake manifold area may be used for four-cycle engines. For two-cycle engines, the exhaust flow area usually exceeds the intake flow area. Overlapping of the exhaust valve periods in multicylinder engines makes multiple manifolds desirable.

Mufflers (see Sec. 12.6) In order to be efficient as sound silencers, exhaust mufflers must decrease the exhaust gas velocity and either absorb the sound waves or cancel them by interference with other waves from the same source. Mufflers should have volumes 6 to 8 times the piston displacement and may contain baffles with or without holes. Mufflers that cancel sound waves by interference usually break the waves into two parts which follow different paths and meet again, out of phase, before leaving the muffler (Davis et al., NACA Rept. 1192).

Exhaust **muffle pits** are used with large engines, are usually made of concrete, have a volume about 20 times the piston displacement, and may be open or provided with baffling or partly filled with loose stone through which the gases must pass. A stack is usually provided for the escape of the gases, and a drain for the discharge of the condensed water from the exhaust products.

Exhaust **back pressure** should be kept to a minimum since an increase of 1 lb/in^2 (6.9 kPa) in back pressure decreases the maximum power output about 2 percent, about 1 percent being due to more exhaust work and the balance of the effect of increased clearance gas pressure on volumetric efficiency.

Supercharging

Supercharging increases the amount of charge per cycle (above that of the normally aspirated engine) by increasing its pressure and hence its density. Supercharging permits more fuel to be burned and is a practical means to increase engine power, but it increases mechanical and thermal stresses in the engine. For aircraft piston engines, it is a means to high-power output for takeoff and to offset the rare atmosphere at high altitude. For diesel engines supercharging results in smaller and lighter power plants. It is especially practical for diesel engines because supercharging does not add to the fuel quality required, and because diesel engines do not use combustion air as effectively as spark ignited engines.

Three alternative methods can be used to achieve supercharging. In **mechanical supercharging,** a positive-displacement type of Roots blower or compressor is geared directly to the engine shaft and thus is driven by engine power. This type is desirable for variable-speed engines (diesel railcar) where high torque is required at various speeds, since the pressure and capacity characteristics of this type do not decrease with speed. In **turbocharging,** exhaust energy drives a turbine, which, in turn, provides the necessary work to drive the turbocharger's compressor, mounted on the same shaft as the turbine (Fig. 9.6.31). The centrifugal compressor is particularly adaptable to aircraft engines because of high capacity, small size, and low weight. It is also used with truck diesel

Table 9.6.14 Condensation Temperatures of Liquid Fuels, °F (°C)

						Condensation temp.			
						Atmospheric pressure		Pressure, ⅔ atmospheric	
		ASTM distillation				Air/fuel ratio		Air/fuel ratio	
Fuel	Deg API	Initial point	50 percent	90 percent	Endpoint	12:1	15:1	12:1	15:1
Gasoline	60	100 (38)	249 (121)	354 (179)	400 (204)	118 (48)	114 (46)	111 (44)	107 (42)
Kerosene	43	350 (177)	450 (232)	510 (266)	550 (288)	204 (96)	200 (93)	201 (94)	197 (92)

engines to improve the volumetric efficiency and power at high engine speeds. Centrifugal blowers may have one or two stages with cooling between stages or after the single or last stage. The rotors run at maximum speeds of 15,000 to 70,000 r/min. Finally, in **pressure wave supercharging,** wave action and resonant tuning in the intake and exhaust systems are used to compress the charge and thereby boost volumetric efficiency.

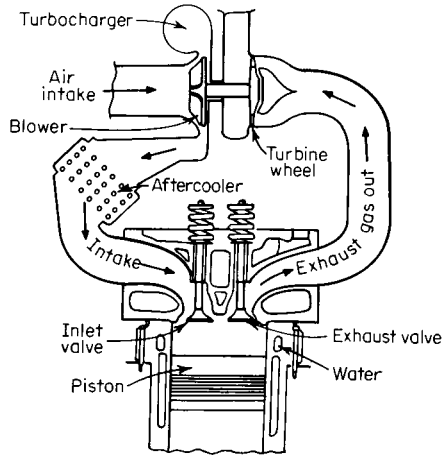


Fig. 9.6.31 Schematic of exhaust turbosupercharger installation on diesel engine cylinder.

Supercharging a spark ignition, premixed-charge engine which has the optimum compression ratio for the available fuel causes combustion knock and necessitates lowering the compression ratio, enriching the mixture, or increasing the octane number of the fuel. Supercharging a diesel engine increases the maximum pressure and necessitates lowering the compression ratio or changing the fuel injection timing, if the limiting pressure is already attained without supercharging. The use of variable engine compression ratio together with supercharging is very desirable but difficult and costly.

Turbocharging of diesel engines is usual for most applications, including truck, locomotive, and marine uses. The exhaust gas turbine provides more complete expansion of the combustion gases than is practical in the engine cylinder and contributes to improved thermal efficiency. Higher output also increases the engine mechanical efficiency (Buchi, *ASME Trans.,* **59,** pp. 85–96). Supercharging assists with NO_x control without a fuel economy penalty. The fuel economy and other performance characteristics of a supercharged Sulzer marine diesel engine of 30-in (760-mm) cylinder diameter are shown in Fig. 9.6.32.

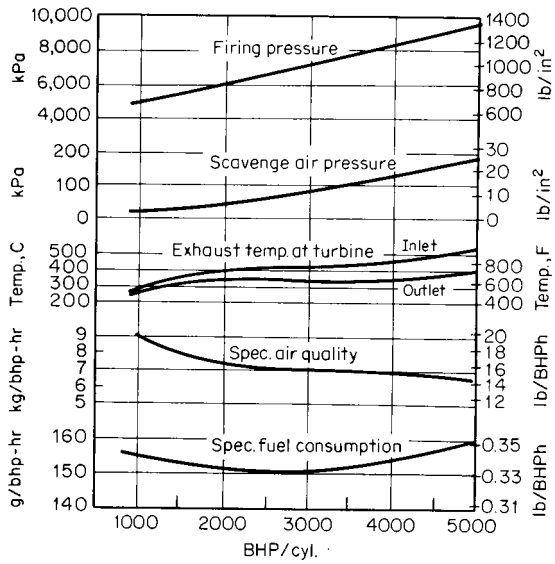


Fig. 9.6.32 Performance curves of a turbocharged diesel engine. *(Sulzer.)*

Scavenging Two-Stroke Cycle Engines

Crankcase compression for each cylinder or an engine driven blower may be used for scavenging and charging the cylinders of a two-stroke cycle engine (Fig. 9.6.33). With crankcase compression, used on small gasoline outboard marine engines, a third port uncovered by the piston near the top of the stroke, a rotary valve, or an automatic poppet valve may be used. The top of the piston is shaped to deflect the gases entering the cylinder and to prevent short-circuiting to the exhaust port. A ''dead'' spot of gases may remain in the lower center of the cylinder or in the upper corners.

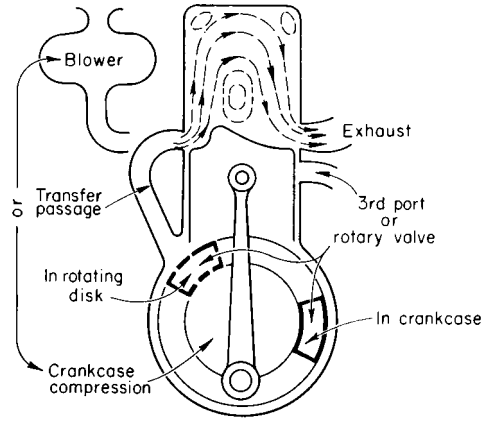


Fig. 9.6.33 Common scavenging systems.

The intake may be inclined and tangential, giving the entering charge an upward helical motion (Fig. 9.6.34). This motion keeps the entering charge near the cylinder walls, forcing the exhaust gases to the center of the cylinder and down to the exhaust port. Some engines employ loop scavenging (Fig. 9.6.35).

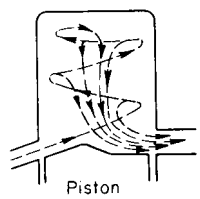


Fig. 9.6.34 Hesselman helical-loop system.

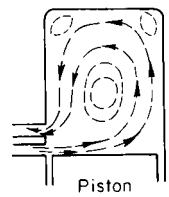


Fig. 9.6.35 M-A-N loop scavenging.

Some large diesel engines, such as the General Motors engine shown in Fig. 9.6.20, have poppet exhaust valves in the cylinder head, eliminating the hot exhaust ports from the cylinder walls. Intake ports can then completely surround the cylinder. The resulting flow pattern is referred to as **uniflow scavenging**. Typical blower pressures and related conditions are as shown in Table 9.6.15 for a General Motors two-stroke diesel engine.

Opposed-piston engines have intake and exhaust ports located in the cylinder walls at opposite ends of the cylinder. A blower forces the

Table 9.6.15 General Motors Two-Stroke Cycle Diesel Engine

	Air-box pressure at 500 to 2,300 r/min		Max bmep*		Max imep*	
Engine type	inHg	cmHg	lb/in^2	kPa	lb/in^2	kPa
Naturally aspirated	1–9	2.54–22.86	101	697	129	890
Turbocharged	1.5–40	3.81–101.62	136	938	163	1,125

* At 2,300 r/min.

entering charge through the intake ports which extend around the entire circumference of the cylinder.

The two-stroke cycle Sulzer diesel engine (Fig. 9.6.36) makes use of a crosshead and piston rod design. The air in the space between the piston and the diaphragm carrying the piston rod stuffing box is displaced by the descending piston and is forced into the cylinder through the intake ports as they are uncovered by the piston. This aids the scavenging of the cylinder. Supercharging is provided by exhaust gas turbine-driven blowers. Air from these blowers passes through intercoolers and is admitted through automatic one-way valves to the scavenging chambers. Electric motor-driven air blowers are sometimes used on large engines to assist scavenging at idle and low power.

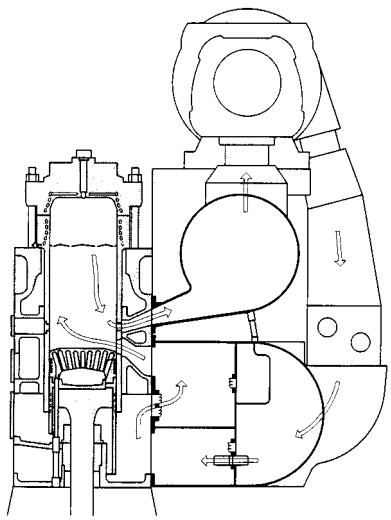


Fig. 9.6.36 A Sulzer series RL two-stroke crosshead diesel engine.

The crosshead design, with its stuffing box, also prevents combustion products from contaminating the crankcase oil system and is commonly used on large engines. This RL design (Fig. 9.6.36) is used for cylinder diameters of 560, 660, 760, and 1,900 mm. The largest cylinder, with a stroke of 1,900 mm, delivers a normal output of 4,000 bhp at 102 r/min corresponding to a bmep of 1,430 kPa (207 lb/in^2) and a piston speed of 6,480 m/s (1,300 ft/min). Figure 9.6.32 shows the performance curves of this type up to an overload of 5,000 bhp per cylinder. Smaller cylinders produce correspondingly lower outputs with roughly the same mep and piston speed.

FUEL-AIR MIXTURE PREPARATION

Mixture Distribution Perfect distribution of fuel-air mixture consists of equal distribution of both air and fuel to the various cylinders of a multicylinder engine for each cycle. The compression pressure of each cylinder is an indication of the air distribution if all cylinders have identical valve timing and compression ratios and no leakage. Cylinders receiving lean mixtures develop less power than those receiving correspondingly rich mixtures. Poor distribution can be masked by enriching the mixture, and high-power output attained at the cost of high fuel consumption rates and HC and CO emission increases. The CO, CO_2, and O_2 analysis of the exhaust gases from the individual cylinders and spark plug temperatures (thermocouple in the center electrode) may be used to determine the mixture distribution.

Distribution of load (and fuel) and variation in injection timing between cylinders of a diesel engine are similarly indicated by the exhaust gas composition and temperature of the individual cylinders, the mixtures in the various cylinders usually being lean under all conditions.

Air lines, extending from the engine to the outside air, should be designed for air velocities of 50 to 100 ft/s (15 to 30 m/s). Air cleaners remove dirt particles and reduce piston, ring, and cylinder wear. Air silencers, used at the entrance to the air lines, are combined with cleaners for automotive purposes.

Gasoline and **diesel fuel lines** should be designed for maximum velocities under 1 ft/s (0.3 m/s) for gravity feed. Gasoline lines should be in a cool location, because heating above 100°F (38°C) may vaporize the lighter fractions and cause **vapor lock**. Sediment and water traps as well as strainers or filters should be located in the fuel line ahead of the fuel injector or carburetor, fuel pump, or injection pump.

Gas lines should be designed for gas velocities of 30 to 60 ft/s (9 to 18 m/s). A pressure drop of 0.07 to 0.12 in (1.8 to 3 mm) of water should be allowed per 100 ft (30 m) of pipe. Close calculation of the **size of gas pipe** for industrial gas installations is undesirable, because of the reduction of the normal pipe cross section by tar and dust deposits. A large-diameter storage chamber should be located in the pipe near the engine. The gas line should be free of traps and as free as possible from changes in direction of flow, and it should be pitched slightly from both ends toward a low point in the center, where a sealed drain should be located. All water seals in the gas line should be as cool as possible, to minimize evaporation and the breaking of the seal.

Mixture Preparation in Spark Ignition Engines

A rich mixture is required for idling and small throttle openings because of dilution of the small incoming charge with the exhaust gases in the clearance space. A maximum-economy mixture is desired at intermediate loads, and a maximum-power mixture is usually desired (automotive practice) at wide-open throttle (Fig. 9.6.37). Maximum economy is obtained with lean mixtures. Maximum power is obtained with rich mixtures which permit optimum utilization of the oxygen with maximum energy liberation per unit of volume of the mixture. Mixture ratios

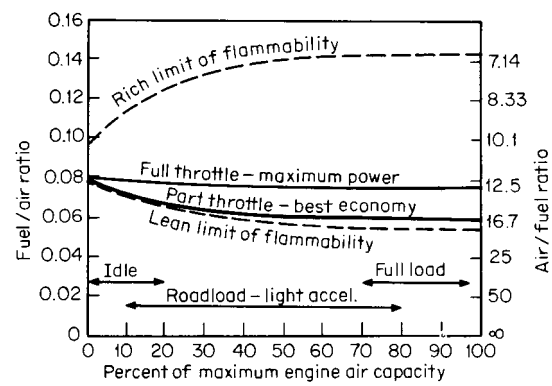


Fig. 9.6.37 Mixture ratios for spark ignition engines.

for minimum specific fuel consumption and for maximum power are also indicated on Fig. 9.6.37. The rich mixture which provides maximum power also results in large amounts of carbon monoxide and unburned fuel in the exhaust.

For many years, carburetion has been used to meter, atomize (if liquid), and mix the fuel with the air flowing to the engine. This method is still in use for mixing gaseous fuels with air, for small engines such as those used in household and off-road applications, for piston-type air-cooled engines, and for older passenger cars. However, except for parts of the world with less stringent emission standards, carburetion has been replaced by fuel injection for automotive applications. The two methods of fuel-air mixture preparation are described below.

Carburetion

The basic principle of carburetion is illustrated in Fig. 9.6.38. A float maintains a constant fuel level in a float chamber which is vented to the atmosphere or air entrance of the carburetor. The float chamber fuel level is slightly below the outlet of the fuel jet. The engine air flows through a venturi tube which has a throat diameter of 0.75 to 0.85 of the diameter of the carburetor bore. The reduction in pressure at the venturi

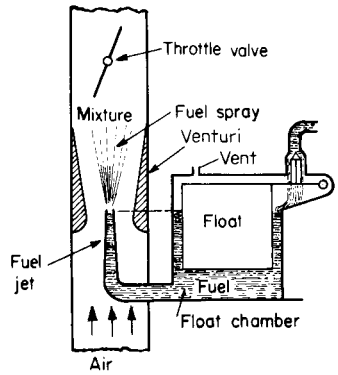


Fig. 9.6.38 Principle of carburetion.

throat causes fuel to flow from the float chamber through the main fuel jet into the airstream. Fuel atomization is accomplished by the velocity difference between the air and fuel. Vaporization of the fuel continues while flowing to the engine cylinder but is usually not completed in the intake manifold at wide-open throttle.

The metering characteristic of a carburetor is the weight ratio of fuel to air supplied over the operating range of airflow. These carburetor flow characteristics are commonly shown on a graph, similar to Fig. 9.6.37, which gives the fuel/air ratios at idle, along the ''road-load curve'' for an automobile engine, and at wide-open throttle. Although computations can be made to establish the fuel/air ratio under specified conditions, estimates are approximate because air is usually bled into the fuel channels to help control the fuel flow. Also, airflow through the venturi is pulsating in nature and makes both airflow and fuel flow somewhat unpredictable. Actual carburetor characteristics approach the desirable characteristics indicated in Fig. 9.6.37.

When the throttle is opened quickly, there is a momentary lag of the fuel flow in relation to the airflow due to the formation of a liquid film on the intake manifold walls. This necessitates the use of some device (accelerating pump or well) to enrich the mixture during sudden accelerations and thereby avoid engine backfire. At idle, the depression of the throat of the venturi is insufficient to overcome the viscosity of the fuel, thus requiring special idle circuits to overcome this problem and to enrich the otherwise very dilute, high-residual cylinder contents. Provisions are also made for mixture enrichment at high loads. In the 1980s, with the enforcement of more stringent emissions standards and the introduction of closed-loop control of the fuel/air ratio to ensure operation in a narrow window about the stoichiometric ratio, the electronic carburetor was introduced (Fig. 9.6.39). In this device, the flow area for fuel metering is computer-controlled by a pulse-width-modulated solenoid.

Carburetors for piston-type air-cooled aircraft engines provide fuel/air ratios that vary from 0.11 at idle to 0.085 for cruising rich (cruising lean being about 0.072). As the full-power condition is approached, the

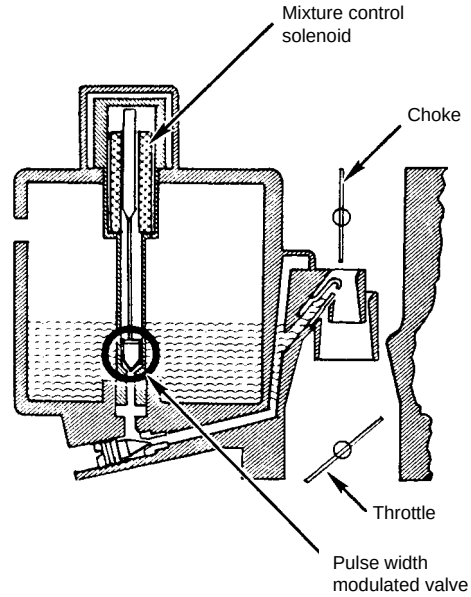


Fig. 9.6.39 Electronic carburetor. *(Amann, ''The Automotive Spark-Ignition Engine—An Historical Perspective,'' ASME-ICE vol. 8, 1989, pp. 33–45.)*

mixture richness is increased to aid cylinder cooling, becoming about 0.108 at maximum output. Aircraft carburetors must compensate for the effect of altitude on the air density and the resulting change in the mixture ratio. Compensation for the enrichment that occurs with increasing altitude is usually provided by reducing the area of the main fuel-metering orifice. This is accomplished by moving a tapered needle in the orifice, which may be done manually or automatically by means of an air-pressure-sensitive bellows. A carburetor providing a 0.067 fuel/air ratio at sea level would provide a fuel/air ratio of about 0.091 at an altitude of 20,000 ft (6,100 m).

Fuel Injection in Spark Ignition Engines A majority of new automobile and truck engines and some small marine and utility engines (future) employ fuel injection. Fuel is injected into the supercharger, intake manifold, intake valve ports, or combustion chambers of spark ignition engines. Fuel injection consists of an in-tank pump, common-rail recirculating supply, and injection nozzles. With single-point, central fuel injection above the throttle, as typified by the throttle body injector of Fig. 9.6.40, one or two injectors with injection pressure of about 70 kPa are used for multicylinder engines. This system provides an electronically controlled injector at the intake manifold entrance where the carburetor was formerly positioned. With multipoint fuel injection, an injector is positioned in each port, and pressures from 400 to 700 kPa are used to avoid vapor formation in the tip located in the hot spit-back region just upstream of the intake valve. In certain cases, an additional injector is used to supplement the fuel flow during starting.

The advantages of fuel injection over carburetion are the more uniform distribution of fuel between the engine cylinders, the reduction of combustion knock caused by nonuniform fuel distribution, the elimination of heat to the intake manifold to obtain the desirable fuel vaporization (thus obtaining higher-power outputs because of higher charge density), more rapid throttle response, and the use of less volatile fuel. Carburetor and induction system icing are eliminated. With an injection system, it is easier to cut off the fuel flow during deceleration (coasting) than is the case with the carburetor with its fuel-wetted intake manifold.

Note that throttle body injection encounters problems associated with the slower transport of fuel than air during transients, as with carburetors.

Fuel injection systems may have mechanical or electronic control of fuel flow, with the electronic type being predominant. In mechanical systems, such as the Bosch K-Jetronic, fuel is continuously injected into

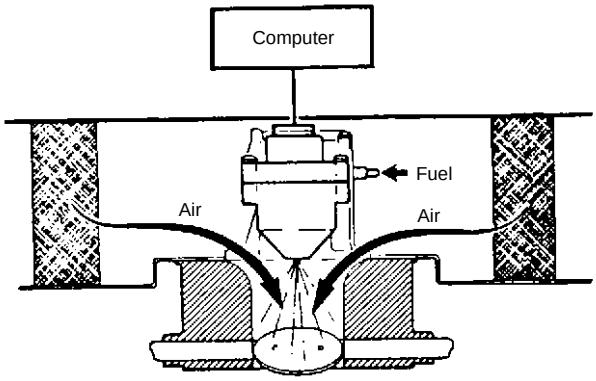


Fig. 9.6.40 Throttle body injector. *(Amann, ''The Automotive Spark-Ignition Engine—An Historical Perspective,'' ASME-ICE vol. 8, 1989, pp. 33–45.)*

the port where fuel quantity is mechanically determined by movement of an air valve. In electronic systems, an electromagnetic injector is pulsed on and off by the computer to supply the desired fuel quantity in proportion to the air. The **injector pulse** width may be trimmed by an exhaust sensor signal to achieve the desired mixture strength. The airflow must be sensed to establish the correct fuel/air ratio for various engine conditions. This can be done by sensing the airflow with a venturi or equivalent, a hot-film, hot-wire, or other air anemometer (such as in the L-Jetronic system), or by sensing the principal engine variables which control engine air consumption. The latter is referred to as a **speed-density** control system. A system of this type is shown schematically in Fig. 9.6.41. Engine speed and manifold pressure and air temperature are used to calculate the engine airflow from prior dynamometer data, a method which has proved less accurate in practice.

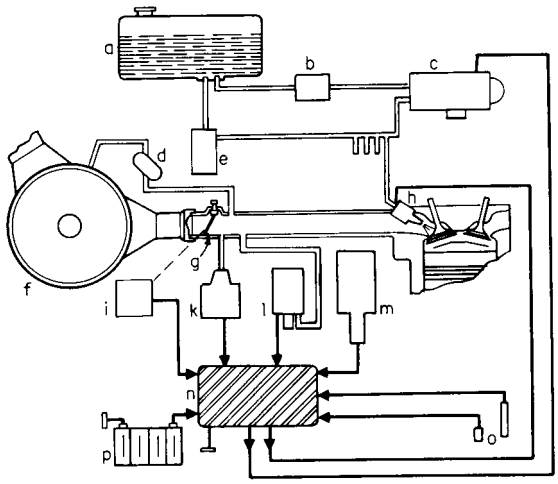


Fig. 9.6.41 Schematic of electronic gasoline injection system for an automobile. (*a*) Gasoline tank; (*b*) fuel filter; (*c*) fuel supply pump; (*d*) auxiliary idling air valve; (*e*) fuel pressure regulator; (*f*) air filter, (*g*) throttle plate; (*h*) fuel injection valve; (*i*) throttle position switch; (*k*) full load pressure switch; (*l*) manifold pressure sensor; (*m*) ignition distributor with trigger contacts; (*n*) electronic control unit; (*o*) temperature sensor; (*p*) battery. (*Robert Bosch.*)

For central fuel injection designs, the injector opening angle may be independent of engine cylinder events (asynchronous). For port injection designs, the opening is normally synchronized with cylinder events to open before the intake valve opens. This minimizes axial mixture stratification which occurs when the injector supplies fuel when the intake valve is open. Such axial stratification may be used advantageously to create a lean-burning engine (Toyota). Injection of fuel into the intake port usually occurs early in the intake stroke, to provide the maximum time for fuel evaporation and mixing to occur. Injection into the intake port provides good mixing as the fuel and air are drawn at high velocity through the intake valve passages, and it isolates the injector from combustion gases at high pressure and temperature. Direct injection into the cylinder occurs during intake or early compression, and it usually results in charge stratification.

Spray targeting in port fuel injection is especially important for fuel evaporation, wall wetting, and engine emissions. Typically when the spray is directed at the center of the intake valve, toward its valve stem base, hydrocarbon and CO emissions are minimized (Fig. 9.6.42). Injector design, spray angle, and droplet size distribution are important. Quader (1989) also showed that targeting the fuel spray at the intake valve centerline toward the valve head resulted in the lowest smoke and

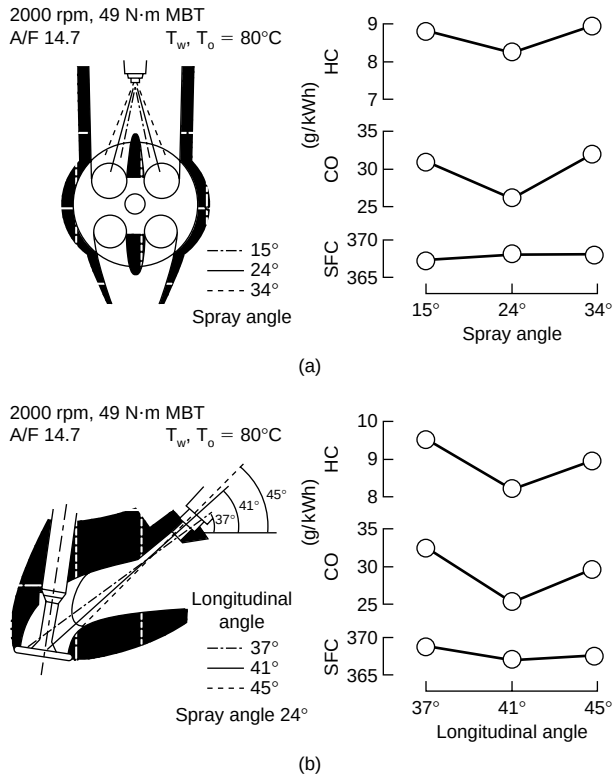


Fig. 9.6.42 Effects of spray angle and injection targeting on engine emissions. (*a*) Effect of dual stream separation angle; (*b*) effect of injection direction. *(Iwata et al., SAE paper 870125, 1987.)*

HC emissions under high-speed and heavy-load conditions. Some designs are less affected by deposits. Air-assisted fuel sprays provide smaller droplets and may be an advantage in both port and direct cylinder injection. Air-assisted direct cylinder injection has shown promise for improved economy and lower emissions in two-stroke, spark ignition, orbital combustion process (OCP) engines.

Fuel Injection in Diesel Engines The fuel injection system for diesel engines usually consists of a pump, fuel line, and nozzle. This system provides control of the beginning, rate and duration of injection, and

desired fuel quantity per injection, and atomizes and partially distributes the fuel in the combustion chamber. Methods of fuel injection include pump or unit injectors and common-rail systems. In the **pump injection system** (Fig. 9.6.43), a pump forces a desired quantity of fuel through a high-pressure fuel line and an atomizing nozzle into the combustion chamber. Unit injectors combine the plunger pump and spray nozzle, thus eliminating the high-pressure fuel line. Injection pressures usually range from 1,500 to 7,000 lb/in^2 (10,300 to 48,000 kPa) but may run above 30,000 lb/in^2 (206,000 kPa) at high injection rates. Fuel quantity

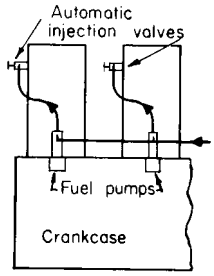


Fig. 9.6.43 Pump injection.

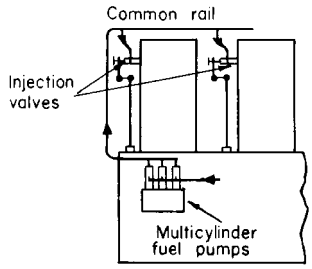


Fig. 9.6.44 Common rail.

control with this type of injection is usually affected by controlling a bypass valve in the pump or the pump discharge line or by varying the time of closure or opening of the fuel-pump inlet port. A separate fuel-pump is commonly required for each engine cylinder, although rotating plunger pumps serving from two to eight cylinders per plunger are in common use, especially on small engines. In the **common-rail system** (Fig. 9.6.44), a constant pressure is maintained in the fuel discharge line, and the discharge of fuel to the engine cylinder is controlled by the lift, timing, and duration of opening of the fuel valve. Considerable capacity of the discharge line is necessary in this case, a triplex pump being considered desirable to supply fuel at a constant rate.

Pumps Fuel pumps for pump injection systems require rugged construction, are usually actuated by a cam, and have constant mechanical stroke. The pump plunger and barrel (Fig. 9.6.45) are a very close lapped fit to minimize leakage. The inlet port is uncovered near the

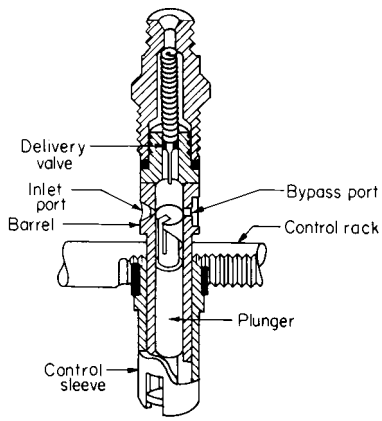


Fig. 9.6.45 Bosch plunger, barrel, and delivery valve assembly.

bottom of the plunger stroke, the low vapor pressure in the pump cylinder causing fuel flow into the barrel. On the upward plunger stroke, fuel flows past the discharge valve and out of the spring-loaded nozzle. Fuel spray from the nozzle continues until the pump bypass port is uncovered. The pressure then subsides, the nozzle closes, and the pump delivery valve closes to maintain a desired residual line pressure until the next injection. Fuel quantity control is obtained by turning the pump plunger in the barrel, which varies the time of uncovering the bypass port by the plunger helix. This in turn varies the duration of the effective plunger stroke and the amount of fuel injected. The high pressures which occur in these injection systems, together with the compressibility of the fuel oil, give rise to transient pressure wave phenomena in the connecting lines which change injection characteristics.

Fuel Lines The tubing connecting the injection pump with the spray nozzle must be thick-walled, of smooth and uniform bore, and made of ductile material to permit bending. Tubes range in size from ¼-in (0.64-cm) OD and 1/16-in (0.16-cm) ID to ½-in (1.3-cm) OD and 3/16-in (0.48-cm) ID. Fuel lines should be the same length for each cylinder, to obtain the same transient hydraulic characteristics. Pressure waves due to plunger motion and nozzle efflux characteristics are developed, travel back and forth, and create disturbances during the injection process (De Juhasz, *Trans. ASME, Oil and Gas Power,* **60,** p. 2; Bolt et al., *SAE Trans.,* 1971, paper 710569), making short lines desirable.

Fuel Injection Nozzles Injection nozzles are usually hydraulically operated, having a differential-diameter valve (Fig. 9.6.46), held on its seat by a spring, which opens when the fuel line pressure is increased by the injection pump. The valve closing pressure is always lower than the opening pressure, depending on the relative effective valve areas exposed to fuel pressure. The valves may be either the pintle or multiple-orifice type (Fig. 9.6.46). The former has a single orifice through which the end of the valve protrudes, and it is used mostly with precombustion

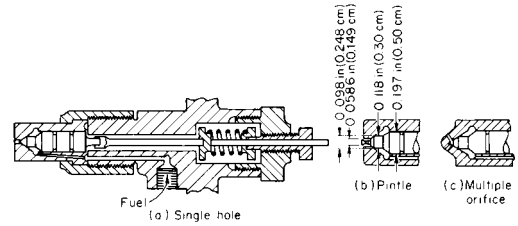


Fig. 9.6.46 Solid injection closed nozzle.

chambers. A pintle may be designed to restrict the fuel flow as it opens; this results in a throttling nozzle. The pintle nozzle provides a hollow conical spray with an included angle which varies from 4° to 60° with various pintle sizes. The multiple-orifice type of nozzle (Fig. 9.6.46*c*) is used principally with the open combustion chamber. The resulting fuel spray consists of a core and surrounding envelope of atomized fuel.

Diesel Fuel Injection System Characteristics At any given setting, most fuel injection systems deliver more fuel per cycle with a decrease in speed, except under the idling condition of very low speed and small fuel delivery. The rising fuel delivery curve with decrease in speed at full load contributes to the good torque rise and lugging ability of the high-speed diesel. At the idle condition, decrease in speed reduces fuel delivery and causes the engine to run still more slowly, etc. Thus the diesel engine is usually unstable and requires a governor for idling as well as at other loads.

COMBUSTION CHAMBERS

Spark Ignition Engines

Most combustion chambers for four-cycle engines, especially automotive, have an open design and employ overhead valves actuated by either pushrods or overhead cams. This design provides fast burning of the charge because of the concentrated combustion volume and minimizes surface area and thus heat losses and wall quenching. Figure 9.6.47 shows classical, two-valve chamber designs. In each case, the spark plug location and chamber shape yield a relation between flame front area and flame travel similar to that in Fig. 9.6.48. For a given engine, the maximum rate of pressure development is proportional to the peak flame-front area viewed at TDC. For the chambers of Fig. 9.6.47*a, b,* and *d,* virtually all the volume is in the head. This design allows large valves and minimizes piston inertia. The hemispherical or

domed head of Fig. 9.6.47*b* allows very large valves and with a domed piston provides a combustion chamber of normal compression ratio. Aircraft engines usually have this chamber shape. The chamber of Fig. 9.6.47*c*, used by Jaguar in its V-12 design, is in the piston bowl. The bathtub chamber of Fig. 9.6.47*d* is very compact for good fuel economy. To suppress knock, each chamber has the spark close to the exhaust valve and a well-cooled, quench region in the end gas far from the spark plug. While all these chambers have a squish area to generate turbulence, compression-induced turbulence is low and mixture motion depends on intake-induced swirl, which is generally quite considerable.

More recently, pent-roof heads with four valves per cylinder and a centrally located spark plug have become very popular. This arrangement gives minimum flame travel distance and hence fast burn, as well as large valve flow area and thus high volumetric efficiency. In such designs, turbulence has been generated through the complex interaction of the two inlet flows which produce a tumbling motion. Other fast-burn chambers have been designed by Nissan and Ricardo. In the Nissan NAPS-Z combustion system, there are twin spark plugs and an induction system producing a comparatively high level of axial swirl (Fig. 9.6.49). In Ricardo's high-ratio compact chamber (HRCC), the combustion chamber is centered on the exhaust valve. Chamber compactness, high turbulence, and lean burning allow this design to operate at compression ratios exceeding 11 without knock.

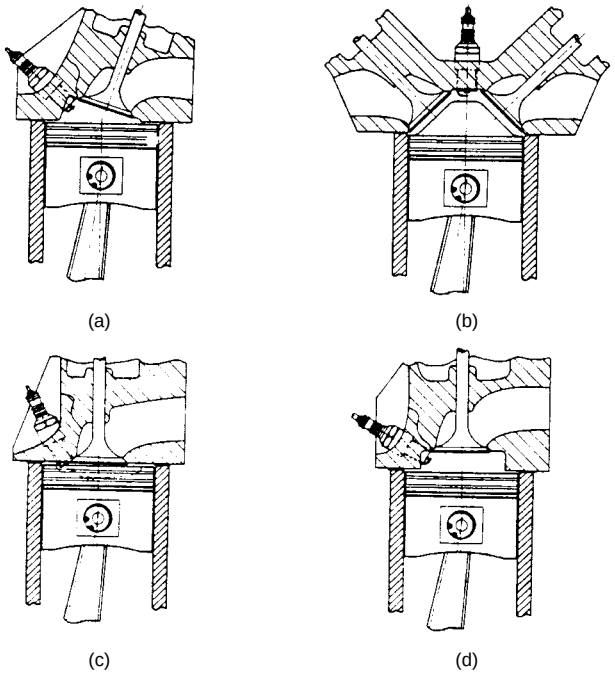


Fig. 9.6.47 Classical combustion chambers for spark-ignition engines; (*a*) Wedge; (*b*) hemispherical head; (*c*) bowl in piston; (*d*) bathtub head. *(Courtesy: R. Stone, SAE, 1995.)*

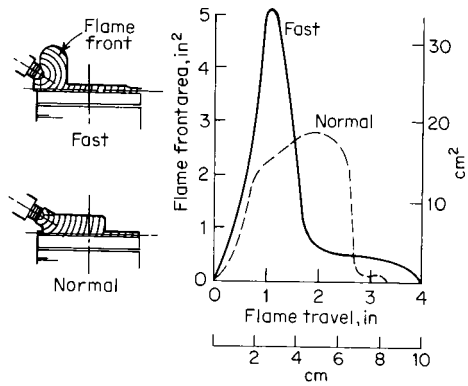


Fig. 9.6.48 Flame-front area curves for fast- and normal-burn combustion chambers.

Where cost or engine height is a principal concern, a design such as the L-head (Fig. 9.6.50), in which the valves are actuated from below directly by the cam, is commonly used. Spark plug location can compensate to some extent for the slow burn rate of the elongated chamber. The L-head has a relatively high surface area, and thus greater heat loss and wall quenching, and a small space for valves.

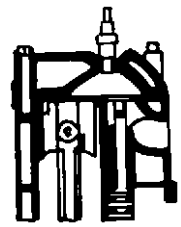


Fig. 9.6.50 L-head engine.

Combustion Roughness The high rate of pressure rise during combustion at relatively low engine speeds (1,000 r/min) and the maximum combustion pressure near peak indicated torque engine speeds subject the engine mechanism to severe stresses which may result in combustion roughness noise unless the pressure rate is controlled. Andon and Marks (*SAE Trans.* **72,** 1964,

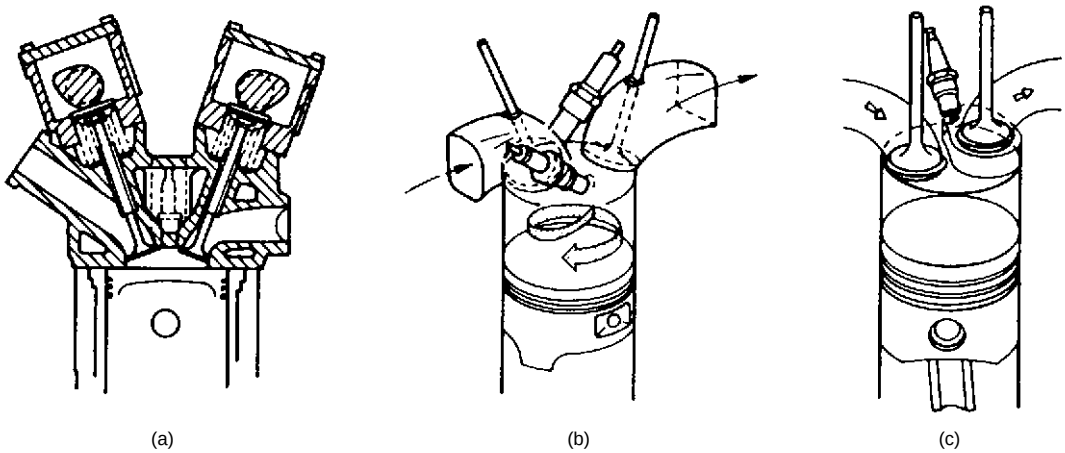


Fig. 9.6.49 Modern fast-burn combustion chambers. (*a*) Four-valve pent-roof; (*b*) NAPS-Z with twin spark plugs and high axial swirl; (*c*) high-ratio compact chamber (HRCC). *(Collins and Stokes, SAE, 1983.)*

p. 636) found that spark plug location, combustion chamber shape, and degree of turbulence are the principal factors that can be varied to control combustion roughness and the resulting noise. In a given engine, the maximum combustion rate of pressure rise can be reduced by retarding the spark timing or decreasing the volumetric efficiency.

A rate of pressure rise of 30 lb/in^2 (207 kPa) per degree of crank travel results in near-maximum efficiency, and higher rates of pressure rise usually result in combustion roughness noise. Increasing the rigidity of the crankshaft increases the natural frequency of vibration and reduces the deflection of the structure caused by combustion and inertial forces and thereby reduces roughness noise.

Die-cast aluminum transmits roughness noise more readily than cast iron. Structurally stiff compact engines such as the V-block are better than the in-line engines in this regard.

Compression Ignition Engines

Compression ignition engines have valve-in-head construction if operating with four cycles. Two types of combustion chamber arrangements are used: a single chamber, referred to as the *open* or *direct* injection type, and a two-chamber, referred to as a *prechamber* or *divided-chamber* type.

Open Combustion Chambers (Direct Injection) Fuel is injected under high pressure, usually through a multiple-orifice nozzle, directly into the clearance space or chamber between the piston and the cylinder head. The piston head is usually conformed (Figs. 9.6.51 and 9.6.52) to fit the fuel spray, and swirl moves the air into the fuel spray. Air swirl is accomplished by intake port design, by shrouding the intake valve, or, in the two-stroke cycle engine, by using tangential intake ports. High turbulence is accomplished by having the piston closely approach part

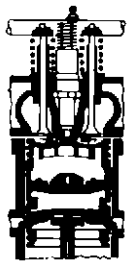


Fig. 9.6.51 Open combustion chamber with swirl produced by intake ports.

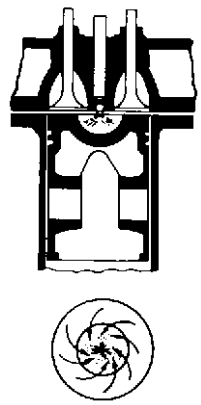


Fig. 9.6.52 Open combustion chamber with high turbulence and some swirl.

of the cylinder head. This forces the gases out of the small clearances and agitates the mixture.

Precombustion chambers are divided into two parts, the major volume being between the piston and the cylinder head and connected by a small passageway to the minor volume located in the cylinder head (Fig. 9.6.53). Fuel is injected into only the smaller chamber, and except under light loads, partial combustion occurs and discharges the burning mixture into the large chamber in which combustion is completed. This

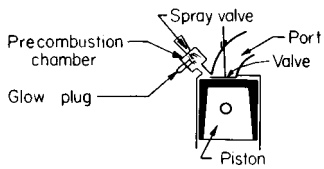


Fig. 9.6.53 Precombustion chamber.

type of combustion chamber produces smooth combustion but has fairly high fluid friction and heat transfer losses.

Turbulence or **divided combustion chambers** are modifications of the precombustion chamber, having the major chamber in the cylinder head and usually only a small clearance space between piston and cylinder head (Fig. 9.6.54). The passage between the two chambers is considerably larger than that in the precombustion chamber. Close piston clearance produces high turbulence in the prechamber and promotes rapid combustion. Part of the prechamber containing the transfer passage commonly is thermally insulated. The chamber (Fig. 9.6.53) is typical of those used in automotive diesel engines for passenger automobile use. An electrically heated hot-wire glow plug is used to assist cold starting, together with compression ratios up to 24 : 1.

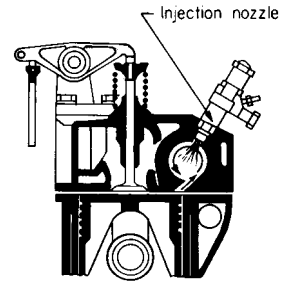


Fig. 9.6.54 Ricardo divided combustion chamber.

A **stratified-charge spark-ignited engine** has potential for burning lean mixtures for improved fuel economy and reduced emissions. In theory, an overall lean mixture is formed in the combustion chamber, while controlled to be slightly richer than stoichiometric in the vicinity of the spark plug at the time of ignition. The richer portion is thus easily ignited, and this in turn ignites the remaining lean mixture. Some of the benefits claimed are (1) lower flame temperature, (2) improved cycle efficiency, (3) less heat loss, (4) less dissociation, and (5) less pumping loss. In addition, less NO_x would be expected because of the lower combustion temperatures; likewise, less unburned hydrocarbon and CO emissions would result because of the overall excess of oxygen.

For effective stratification, very careful development has proved to be necessary to obtain complete combustion of fuel under the wide range of speed and load condition required of an automotive type of engine. Three rather distinct means for accomplishing the stratified charge condition have been explored.

1. A single combustion chamber with a well-controlled rotating air motion is used. This arrangement is illustrated (Fig. 9.6.55) by the Texaco combustion process (TCP), patented in 1949. Ford Motor Company's programmed combustion (PROCO) is a similar, single-chamber concept.

2. There is a prechamber or two-chamber system. This is illustrated by Fig. 9.6.56, which shows the general arrangement of the Honda compound-vortex, controlled-combustion (CVCC) system.

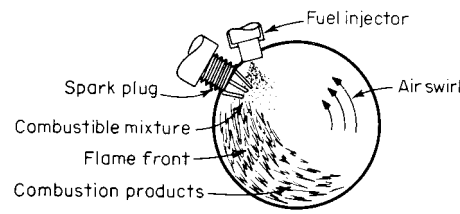


Fig. 9.6.55 Texaco combustion process (TCP).

3. There is axially stratified charge. By turning on the port fuel injector toward the end of the filling process, a rich mixture is produced near the spark plug, while the overall mixture may be lean. This method is known as the *Toyota lean-burn system.*

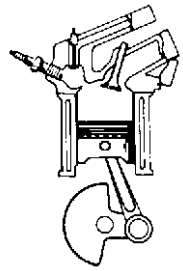


Fig. 9.6.56 Honda CVCC combustion process.

SPARK IGNITION COMBUSTION

Ignition System The ignition system usually consists of a 12-V storage battery, induction coil, and high-tension distributor, or a high-tension magneto with a distributor. Recent automotive practice employs electronic means to interrupt primary current in place of a contact breaker, and the trend is toward individual coils for each plug. For aircraft, the low-tension distribution system to transformer spark plugs, or other apparatus for obtaining a high-tension spark at the plug, appears to eliminate some of the troubles of the conventional ignition system in aircraft engines at high altitude. For details on ignition systems, see Sec. 15.1.

The usual firing order for in-line engines is 1, 3, 4, 2, or 1, 2, 4, 3 for four cylinders; 1, 5, 3, 6, 2, 4 (U.S. automobile engines) or 1, 4, 2, 6, 3, 5 for six cylinders in line; and 1, 6, 5, 4, 3, 2 for V-6 engines. A common procedure for V-8 engines is to number the cylinders from front to rear, with the odd numbers in the left bank, as viewed from the driver's seat. For this arrangement a typical firing order is 1, 8, 4, 3, 6, 5, 7, 2. Radial engines have firing orders that skip every other cylinder. For a nine-cylinder, single-row engine the order is 1, 3, 5, 7, 9, 2, 4, 6, 8.

Spark plug gaps vary from 0.020 to 0.080 in (0.5 to 2.0 mm). The smaller gaps require 4,000 to 8,000 V while the larger gaps require 10,000 to 34,000 V. In wedge-shaped, two-valve combustion chambers, spark plugs were typically located near the exhaust valve (hottest spot) so that the flame progressed toward the cooler part of the combustion chamber; this arrangement permitted higher compression ratios without combustion knock. In modern, four-valve, pent-roof combustion chambers, a central spark plug location results in minimum combustion time and good tolerance for dilution. Dual ignition (two spark plugs located opposite each other) also reduces combustion time and results in a slight gain in efficiency (usually less than 1 percent) compared with single ignition with optimum spark advance. Rotary (Wankel) engines employ one or two spark plugs per rotor. Single plugs are centrally located either above or below the trochoid waist. In modern, direct injection, stratified-charge, lean-burn engines, optimum placement of the spark plug is crucial to provide a combustible mixture near its vicinity.

Flame Speed Flame speeds in spark ignition engines are low immediately after ignition, attain maximum values when about one-half the combustion chamber has been inflamed, and decrease toward the end of the piston. Mean flame speeds are a maximum for mixtures usually 10 to 20 percent richer than the chemically correct air/fuel ratio, and they vary with fuel, engine speed, and turbulence.

At 900-ft/min (4.5-m/s) mean piston speed, mean flame speeds with maximum-power gasoline-air mixtures and optimum spark advance range from about 50 to 100 ft/s (15 to 30 m/s). Mean flame speeds under the same mixture, spark, and speed conditions in the CFR Waukesha engine are estimated at 55 (70) [80] ft/s (17, 21, 24 m/s) for a 4.6 : 1 (6.5 : 1) [8 : 1] compression ratio, while with ethyl alcohol–air mixtures the corresponding mean flame speeds are 50 (65) [75] ft/s (15, 20, 23 m/s).

The mean flame speed in an automotive engine under the foregoing conditions is estimated from the optimum spark advance (approximately five-ninths of combustion time) to be 90 ft/s (27 m/s) for a 6.5 : 1 compression ratio. In the same engine, mean flame speeds with optimum conditions vary from about 60 ft/s (18 m/s) at 500 ft/min (2.5 m/s) to about 170 ft/s (0.9 m/s) at 3,000 ft/min (15 m/s) mean piston speed. The higher flame speeds in the automotive engine at comparable piston speeds are due to greater turbulence than that obtained with the usual flat valve-in-head (CFR) design.

Spark Advance Since combustion requires finite time, it is necessary to ignite the mixture in the cylinder before the piston reaches the end of its compression stroke. This should distribute the combustion process before and after top center in order to obtain maximum torque and power. Spark advance is measured by the number of degrees the crankshaft rotates between the time of the spark and the end of the compression stroke. **Optimum spark advance** is the minimum advance which develops the **maximum brake torque (MBT)**. Usually this causes (1) the maximum pressure to peak at about 16° after top center and (2) 50 percent of the charge to be burned by 10° after top center (Heywood, 1988). The torque lost by a spark timing earlier or later than optimum is at first only slight but increases rapidly as the timing further deviates from optimum (see Fig. 9.6.57).

Optimum spark advance depends principally on air-fuel mixture, amount of residual gas, emission control requirements, combustion chamber design, turbulence, engine speed, number of spark plugs, and

Table 9.6.16 Ignition Temperatures of Air-Gas Mixtures at Atmospheric Pressure

		With ignition lag*		Instantaneous† ignition	
Gas	Chemical symbol	% gas in mixture	Avg temp, °F (K)	% gas in mixture	°F (K)
Hydrogen	H_2	8–24	1,130 (893)	10	1,377 (1,020)
Carbon monoxide	CO	13–47	1,204 (924)	50	1,708 (1,204)
Methane	CH_4	5–38	1,202‡ (923)	25	> 1,832 (1,273)
Ethane	C_2H_6	§	968‡ (793)		
Propane	C_3H_8	§	914‡ (763)		
Acetylene	C_2H_2	4–22	804 (702)		
Ethylene	C_2H_4	6–19	1,004 (813)	10	1,832 (1,273)
Benzene	C_6H_6			50	1,943 (1,335)
Ether	$C_4H_{10}O$			50	1,892 (1,306)
Gasoline:					
92 octane					734 (663¶)
100 octane					804 (1,702¶)

* Dixon and Coward, *Trans. Chem. Soc.*, **95,** 1909, p. 514.
† David, *Trans. Chem. Soc.*, **111**, 1917, p. 1003.
‡ Minimum value, since spread in values was appreciable.
§ Composition not reported.
¶ Jones, *U.S. Bur. Mines Bull.*, 1946.

Table 9.6.17 Ignition Temperatures at Various Air Pressure, °F (K)

Gas	Air pressure, atm								
	1	3	5	7	10	15	20	25	30
Hydrogen*	1,134 (885)	1,132 (884)	1,112 (873)	1,100 (866)					
Methane*	1,343 (1,001)	1,269 (960)	1,215 (930)	1,175 (908)					
Gasoline†			590 (583)		480 (522)	420 (489)			
Kerosine†			670 (628)		490 (528)	430 (494)	400 (478)	385 (469)	
Gas oil†			580 (578)		500 (533)	450 (505)	435 (497)	415 (486)	
Machine oil†			710 (650)		610 (594)	550 (561)	520 (544)		
Benzene†			685 (636)		585 (580)	545 (558)	530 (550)	515 (541)	500 (533)
Cylinder oil†			825 (714)		710 (650)	645 (614)	620 (600)	595 (586)	572 (573)
Benzol†					1,095 (864)	970 (794)	930 (772)	905 (758)	880 (744)

* Bone and Townsend, ''Flame and Combustion in Gases, p. 69.
† Tausz and Schulte, *Zeit. V.D.I.*, 1924, p. 574.

spark plug location. The optimum spark advance is also affected by knock considerations, as described later. Maximum power air/fuel ratios require the minimum spark advance. Low-speed engines require 10° to 20° spark advance, and high-speed automotive engines require 30° to 40° spark advance. Racing engines require still more. Full load requires less spark advance than part throttle. Spark advance in contemporary automotive engines is based on computer-stored values and is a function of speed, load, and other engine operating variables.

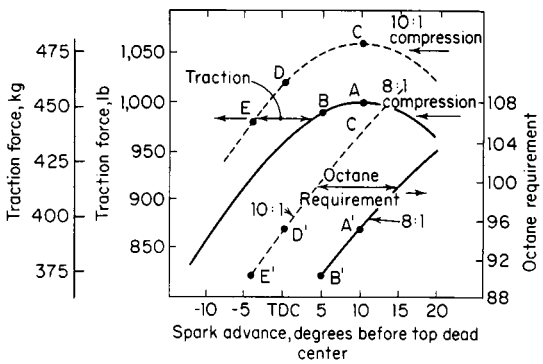


Fig. 9.6.57 Effect of spark advance on traction force and octane requirement.

Autoignition The autoignition temperature of an air-fuel mixture is the lowest temperature at which chemical reaction proceeds at a rate sufficient to result eventually (long time lag) in inflammation. This temperature depends principally on the air-fuel mixture, fuel properties, mixture pressure, and test apparatus. Subjecting a mixture to a temperature higher than the autoignition temperature results in inflammation after a shorter time lag, there being a temperature for each mixture which results in practically instantaneous ignition (Table 9.6.16). An increase in pressure decreases the ignition temperature of an air-fuel mixture (Table 9.6.17). See CRC Report on Project CF-37-53.

Compression ignition engines, such as diesel engines, rely on autoignition for initiation of combustion. However, in spark ignition engines, autoignition of the gas ahead of the flame front is undesirable as it leads to combustion knock. This abnormal combustion phenomenon is described in greater detail in the following section.

COMBUSTION KNOCK

Combustion knock in the internal combustion engine is produced by the spontaneous combustion or autoignition of an appreciable portion of the charge (Rassweiler and Withrow, *Trans, SAE,* **31,** 1936, p. 297), which results in an extremely rapid local pressure rise and sharp metallic knock. In the spark ignition engine, this occurs when the last fraction of the charge to burn is compressed appreciably above its autoignition temperature by the fraction that has burned. However, if the lag in time (ignition lag) between the attainment of this temperature and the spontaneous local appearance of the flame is increased by the use of a knock-suppressor blending agent or additive, then the flame from the spark plug will be permitted to travel through the unburned fraction before it can knock. Thus knocking tendency depends on the autoignition temperature, ignition lag, and flame speed of the air-fuel mixture.

The tendency of a spark ignition engine to knock increases almost directly with spark advance (Fig. 9.6.57). Prior to the introduction of exhaust emission control requirements, it was general practice in automobile engine design to use a spark advance a few degrees less than optimum at wide-open throttle. This materially reduces the tendency to knock with only a negligible loss of torque. At part-throttle operation, where combustion knock may not be a problem, the spark is advanced closer to optimum. However, in many cases a compromise must be made between spark advance for good torque and spark retard for emission control.

In the compression ignition engine, combustion knock occurs when the ignition lag is comparatively long, thus permitting the evaporation of a considerable portion of the injected fuel, which suddenly inflames and produces the characteristic knock at the initiation of combustion. Thus knock suppression requires long ignition lags (high octane, low cetane) in spark ignition engines and short ignition lags (high cetane, low octane) in compression ignition engines.

Knock Rating of Fuels

For detailed information regarding methods, apparatus, reference fuels, and suppliers of apparatus and reference fuels, see ASTM Manual of Engine Test Methods for Rating Fuels, current edition.

Octane Number

The octane number (ON) of a fuel is the percentage by volume of iso-octane (2,2,4 trimethylpentane) in a mixture of iso-octane and normal heptane which matches the unknown fuel in knock tendency when compared by a specific procedure in the ASTM CFR knock-testing engine. A fuel of high octane will have a low cetane number and vice versa (Fig. 9.6.58).

The **performance number** scale for aviation gasolines (Fig. 9.6.58) is designed to relate fuel rating to average knock-limited performance (imep). A gasoline of the 100/130 grade indicates that in the aviation test (lean mixture, normally aspirated) the fuel is equivalent to iso-octane in imep while in the supercharge test (rich mixture) it permits an imep of 130 percent of that of iso-octane.

There are four methods—**research, motor, aviation** (lean mixture), and **supercharge** (rich mixture) (ASTM D 908, 357, 614, and 909, respectively)—in general use in the rating of gasolines according to knocking tendency. These methods vary principally in the engine operating conditions and the means for indicating knock intensity (see ASTM manual).

Engine conditions, as indicated by the air or mixture temperature and by the coolant temperature, are less severe in the research method than in the other methods. Thus, some fuels rate lower by the motor method than by the research method. The sensitivity of a fuel is defined quanti-

tatively as the difference between the research and motor octane numbers. Although most gasolines rate lower by the motor method than by the research method, straight-run gasolines have about the same rating by both methods. With present fuels and engines, the average of the research and motor method numbers provides the best correlation with road ratings.

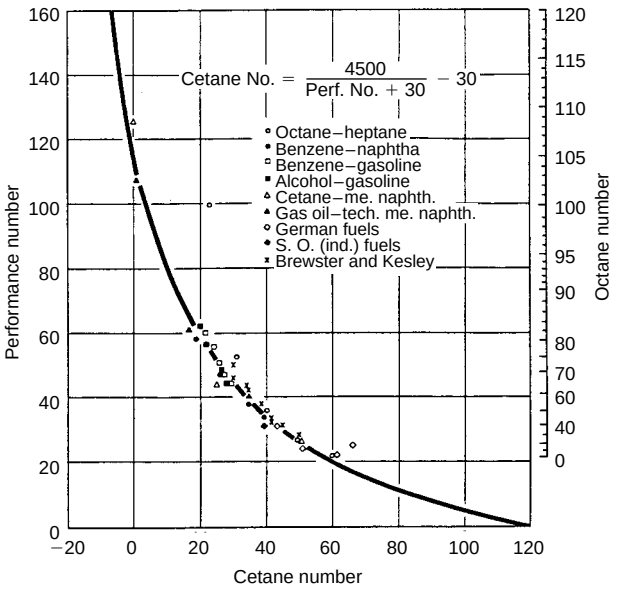


Fig. 9.6.58 Relationship between antiknock scales (octane number and performance number) and cetane number. *(Brewster and Kesley, SAE, 1963.)*

The use of the aviation and supercharge methods for aviation gasolines results in two performance numbers (such as 100/130). This grading indicates that the fuel is less sensitive than iso-octane to the difference in engine severity in the two test methods.

Road ratings of gasoline are determined by means of borderline knock curves for the test fuel and various blends of the reference fuels at various spark advances. The car is accelerated with wide-open throttle, and the speed at which steady knock ceases is recorded as the knock die-out speed (CRC procedure E-2). A plot of these data on a spark-advance vs. knock die-out speed chart (Fig. 9.6.59) makes possible the determination of the octane number rating of the test fuel over the desired speed range.

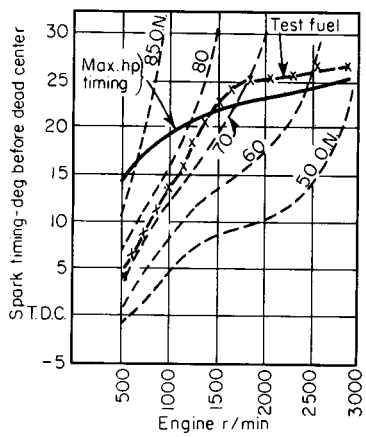


Fig. 9.6.59 Road octane rating.

Mechanical octane number is the resulting decrease in fuel octane requirements brought about by mechanical changes in engine design or control. Efficiency design variables include ignition control, combustion chamber design, cooling, valve timing, fuel system, and engine-transmission combinations.

Knock Suppression The antiknock characteristics of a fuel can be improved by blending in either another fuel of better antiknock characteristics or a knock-suppressor fuel additive (Fig. 9.6.60). Tetraethyllead (TEL) and tetramethyllead (TML) are the most effective knock suppressors (Table 9.6.18) and have been used extensively in small

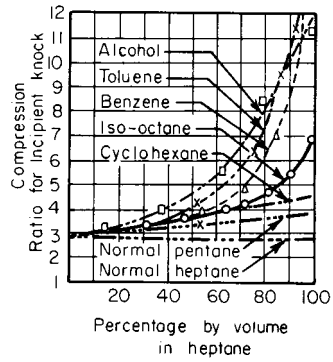


Fig. 9.6.60 Effect of fuel composition on knock-limited compression ratio.

amounts (up to 3 mL/gal of automotive fuels) in many gasolines. Their addition has been eliminated because of adverse effects on health and exhaust emissions. Other compounds can raise antiknock quality, but the much larger amount needed typically changes volatility and stoichiometric point significantly and may affect engine life, emissions, and lubrication.

Cetane Number The cetane number (CN) of a diesel fuel is determined by matching its ignition quality with that of a blend of two reference fuels, normal cetane (CN = 100) and heptamethylnonane (CN = 15). The comparison is made in an ASTM CFR diesel engine (Waukesha Motor Co.) using a specified procedure (ASTM D 613 or CRC-F-S). Cetane number for the reference blend is calculated by

$$\text{CN} = (\text{vol \% } n\text{-cetane}) + 0.15(\text{vol \% heptamethylnonane})$$

The **ignition quality** of a fuel for a compression ignition engine is indicated by the time delay between beginning of injection and the start of rapid pressure rise caused by combustion. The compression ratio in the test engine is adjusted to result in a 13° crank-angle delay for the unknown fuel when injection begins at 13° before top center. The mixture of reference fuels which has the same lag under the same conditions indicates the cetane number of the fuel.

The factors that tend to aggravate knocking in a spark ignition gaso-

Table 9.6.18 Relative Effect of Antiknock Compounds

Compound	Chemical symbol	Weight for a given effect, g
Tetraethyllead (TEL)	$Pb(C_2H_6)_4$	0.0295
Aniline	$C_6H_5NH_2$	1.0000
Ethyl iodide	C_2H_5I	1.55
Ethyl alcohol	C_2H_5OH	4.75
Xylene	$C_6H_4(CH_3)_2$	8.0
Toluene	$C_6H_5CH_3$	8.8
Benzene	C_6H_6	9.8

SOURCE: ''International Critical Tables,'' vol. 2, pp. 162–163.

Table 9.6.19 Antiknock Indices (ASTM D4814)

Antiknock index (RON + MON)/2	Application
Unleaded fuel* (for vehicles that can or must use unleaded fuel)	
87	Designed to meet antiknock requirements of most 1971 and later model vehicles
89	Satisfies vehicles with somewhat higher antiknock requirements
91 +	Satisfies vehicles with high antiknock requirements
Leaded fuel* (for vehicles that can or must use leaded fuel)	
88	For most vehicles that were designed to operate on leaded fuel

* Unleaded fuel having an antiknock index of at least 87 should also have a minimum motor octane number of 82 in order to adequately protect those vehicles that are sensitive to motor octane quality.

SOURCE: Abstracted from ASTM D4814, with permission.

line engine tend to suppress it in a diesel engine. Thus, octane has a low, while heptane has a high, cetane number. High-CN fuels burn more smoothly and start cold engines more readily than low-CN fuels.

Cetane number has been correlated with the mid-boiling point and the API gravity of the fuel, the relation for computing the approximate cetane number being (*Jour. Inst. Petroleum,* **30,** 1944, p. 193)

$$CN = 175.4(\log \text{mid-boiling pt, °F}) + 1.98(\text{API gr}) - 496$$

The deviation from engine test CN for 579 fuels was less than ± 5 CN for 94 percent of the fuels (see also Sec. 7.1).

Ignition accelerators increase the cetane number of diesel fuels, but the susceptibility of the fuel to the accelerator decreases with an increase in the amount of the accelerator (Fig. 9.6.61). Commercial amyl

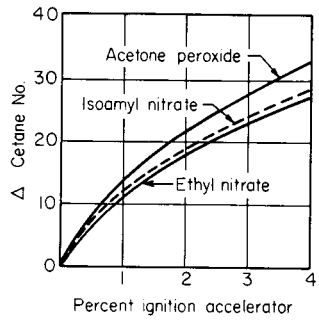


Fig. 9.6.61 Improvement in CN with ignition accelerators. *(Bogen and Wilson.)*

nitrate is slightly less effective than the pure isoamyl nitrate (Bogen and Wilson, *Petroleum Refiner,* July 1944).

Permissible Compression Ratios Data obtained on single-cylinder test engines and multicylinder automotive engines show wide variations in the compression ratios producing incipient knocking of fuels of various octane numbers. High compression ratios in engines using fuels of a given octane number do not necessarily indicate excellent design and high performance, since valve timing, manifolding, and porting may act as throttling devices, or spark timing may be retarded. Average compression ratios vary considerably depending on the design and fuel used. An increase in cylinder size usually necessitates a lower compression ratio (Fig. 9.6.62), which may be due in some cases to the use of lower grades of fuels with the larger spark ignition engines. The larger diesel engines require less compression to obtain the desired temperatures since the heat transfer effect is less than in the smaller engine.

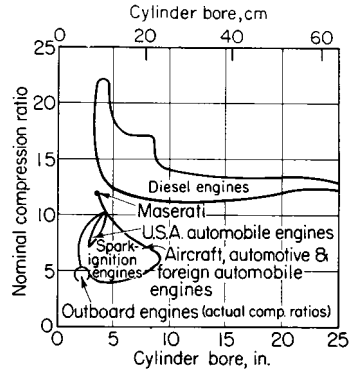


Fig. 9.6.62 Range of compression ratios as a function of cylinder bore.

OUTPUT CONTROL

Speed and **load regulation** are usually obtained in four-stroke cycle gasoline engines by throttling the charge or by quantitative governing. The air/fuel ratio may vary some with load, depending on the engine requirements. Intake throttling reduces the pressure in the cylinder at the end of the intake stroke and thereby reduces the mass of charge trapped per cycle. While throttling is the simplest means for regulation and provides a stable idling condition, it increases the pumping work and consequently decreases the efficiency of the engine. Presently, there is a trend toward using variable valve timings and variable valve lift as a means of controlling load without increasing pumping losses. In direct injection, stratified-charge engines, load control may be accomplished through control of the amount of fuel injected per cylinder.

Power control of carbureted, spark-ignited, two-stroke cycle engines is difficult because intake throttling reduces or prevents cylinder scavenging. Qualitative governing is used in some cases with constant-speed gas engines. The fuel is throttled while the air is unrestricted. Qualitative governing is limited by the lean limit of flammability. In addition, lean mixtures burn more slowly and may lower the thermal efficiency of the engine. Advancing the ignition timing, as the mixture is made leaner, reduces this loss of efficiency.

Governing by variation of ignition timing, as used on some small two-stroke cycle gasoline engines, is undesirable since this is regulation by variation of thermal efficiency and may result in overheating the engine. Variation of ignition timing is required for each change in speed and load to obtain optimum results.

Power control or regulation of diesel engines is obtained by varying the amount of fuel injected per cycle, which provides increasing richness of the fuel-air mixture as the engine load is increased. Maximum specific fuel consumption usually occurs at about 85 percent of the maximum load.

Diesel engines require a speed governor to provide a stable idle. Mechanical, hydraulic, and electronic speed-control governors are used on diesel engines. These control the fuel pump injection quantity and provide speed regulations at all operating conditions.

COOLING SYSTEMS

Engine cylinders must be cooled to maintain a lubricant film on the cylinder walls and other sliding surfaces. The cylinder heads, pistons, and exhaust valves are cooled to prevent combustion knock or destruction of these parts due to overheating. The lubricant must be cooled to maintain the desirable viscosity under operating conditions. Liquid or air cooling systems are used. Pistons, exhaust valves, and lubricants in comparatively small or low-duty engines are sufficiently cooled by contact with other engine parts or the lubricant between them and do not require separate cooling systems.

Liquid Cooling The heat removed by the cooling liquid from the

cylinders and cylinder heads ranges from 15 to 20 percent of the energy input for large diesel engines, 20 to 35 percent for automotive engines, and may run as high as 40 percent for automotive engines at one-third load. An increase in speed decreases the heat loss to the coolant. Heat loss can range from 40 to 50 percent of the brake output for large diesel engines and 100 to 150 percent of the brake output for automotive engines.

Pistons in small engines are cooled by heat transfer to the cylinder walls and lubricant. In some cases, an appreciable quantity of oil is directed against the piston head to maintain desirable head temperatures.

Some large engines have even used liquid-cooled pistons. Liquid is usually fed to the piston by swing joints or telescopic connections. The coolant jacket surrounding the high-temperature areas of the engine is maintained at a pressure considerably above atmospheric—10 to 15 lb/in^2 (70 to 100 kPa). This permits operation at higher coolant temperatures and/or decreases the tendency for local boiling in the region of ''hot spots,'' which could lead to component failures. The average percentage distribution of the coolant to the various jackets is approximately 60 percent to the cylinder jacket and 40 to the cylinder-head jacket with uncooled piston; 50 to the cylinder jacket, 25 to the cylinder-head jacket, and 25 to the piston and piston rod with liquid-cooled piston.

The cooling system may be noncirculating, in which case cold water is supplied, flows through the water jacket, and is wasted. With a temperature rise in large engines of 90°F (32°C), from 2 to 4 gal/(bhp · h) [7.6 to 15 L/(bhp · h)] of coolant must be supplied. In some engines, coolant outlet temperatures as low as 120°F (49°C) must be maintained, which, with high inlet temperatures, may increase the required quantity of coolant to as much as 10 gal/(bhp · h) (38 L)/(bhp · h). Coolant is circulated at a rate of 25 to 50 gal/(bhp · h) [95 to 190 L/(bhp · h)] with a temperature rise of 10 to 20°F (5.6 to 11°C) while flowing through the water jacket.

Natural circulation (**thermosiphon**), with low velocities, requiring large connections, may be used if the coolant forms a complete circuit. In this case, hot coolant rises in the engine jacket, flows to the radiator, is cooled, descends, and flows back to the engine jacket.

Small stationary engines are often **hopper-cooled**. This is an **evaporative cooling system** requiring about 4 to 6 lb/(bhp · h) [1.8 to 2.7 kg/(bhp · h)] of makeup water. The ASTM CFR knock-testing engines are cooled by evaporation, the vapor being condensed by a **reflex condenser** (copper coil with cold water flowing through) and returned to the cylinder jacket.

In high-output engines, the coolant is directed at the hottest spot, usually the exhaust valve seat, with either an external or internal inlet manifold from a common cylinder block jacket; otherwise vapor bubbles will form, cling to the surface, and cause overheating. In vertical engines, the coolant usually flows upward, around the cylinder barrel into the cylinder head jacket, and to the outlet. Higher output and reduced knocking tendency have resulted from directing the entering coolant at the hot spots in the cylinder head and then letting the coolant flow downward around the cylinders.

The **size of piping** required for the inlet to the jacket may be calculated for 3-ft/s (1-m/s) coolant velocity, and for the discharge 2 ft/s (0.6 m/s) for large engines. In engines with recirculation, these values are increased up to 10 to 15 ft/s (3 or 4.5 m/s), and the size of the inlet line to the circulating pump is usually larger than the outlet from the coolant jacket. It is desirable to have an open or visible outlet from each separately cooled part, showing the flow of coolant. To ensure totally filled systems which are as air-free as possible, overflow bottles may be used. Water is an outstanding liquid coolant. However, where freeze protection is necessary or where higher jacket temperatures are desired, other cooling media are generally used. For example, light-duty engines typically use a mixture of water and ethylene glycol. (See Sec. 6.8.)

Air Cooling Many small industrial engines and most motorcycle and aircraft engines are air-cooled. Even here, however, higher specific output engines are increasingly liquid-cooled. Air cooling eliminates the necessity for water or other liquid cooling media, coolant jackets, pumps, radiators, and coolant connections, but necessitates single-cylinder construction, finning, baffles, and, in some cases, blowers. Air-cooled engines also tend to be more noisy since the combustion chamber walls radiate sound and the engine structure is less rigid. The permissible compression ratio and output of air-cooled aircraft engines depend on the efficiency of the cooling of the cylinder head, exhaust valves, and seats. Long fins [1 to 2 in (25 to 50 mm) depending on cylinder size], closely spaced [0.10 to 0.20 in (2.5. to 5 mm)], with baffles directing the air at high velocity at the hot spots, have made high-output aircraft engines possible at the expense of considerable air resistance or drag. Properly designed cooling ducts may make use of the heat added to the cooling air to obtain a jet effect, thereby reducing drag.

Oil Cooling Shearing of various oil films by moving parts and oil contact with hot parts of an engine cause a rise in oil temperature until energy input to the oil equals the energy transferred by the oil to the cooler parts that it contacts. Oil coolers are required for high-duty engines to maintain oil temperatures of 200°F (94°C) or under. Temperatures of 250°F (120°C) are not uncommon in automobile engines on hot days: 300°F (150°C) is considered too high, particularly for oils which decompose rapidly under conditions causing the high oil temperatures. The desired oil temperature is maintained by circulating the oil through a radiator or cooler to an oil sump and then through the engine. Transmission oil cooling is also accomplished by placing plate coolers in the end tanks of radiators or with separate radiators.

LUBRICATION

(See also Secs. 6.10 and 8.4)

General A suitable liquid lubricant is vital to the satisfactory operation of an internal combustion engine. It prevents excessive wear and deposit accumulation and removes heat from the areas of relatively high temperature within the engine.

Most engine oils are composed of base oils and additives. The base oils are usually mineral oils, but some are synthetic oils. Chemical additives are incorporated in engine oil formulations to impart desirable performance characteristics which are not provided by base oils alone.

Physical Properties Both the physical and chemical properties of an engine oil affect its performance. The principal physical property involved is viscosity. It must be low enough at low temperatures to permit cranking and starting and high enough at high temperatures to provide an adequate oil film between rubbing surfaces. Sufficiently high viscosity is also required to help prevent excessive oil consumption at high temperatures. Another physical property affecting consumption is volatility. Consumption can be undesirably high with an oil of excessive volatility. Both viscosity and volatility are influenced by base oil selection. Viscosity can also be modified by means of additives.

Chemical additives perform a number of other functions in an engine oil. For example, oxidation inhibitors reduce oxidative and thermal degradation, which can lead to varnish and sludge deposition and to excessive thickening. Detergent-dispersant additives suspend insoluble products formed during engine operation, so they can be removed from the engine when the used oil is drained. Antiwear agents help protect rubbing surfaces, especially those subject to boundary lubrication. Antifoam agents prevent foam and air entrainment and their adverse effects on oil pressure and heat transfer. Rust inhibitors eliminate the formation of corrosion products on ferrous surfaces and reduce corrosive wear. Friction-reducing additives lower boundary friction.

SAE Classifications Engine oils are classified by the Society of Automotive Engineers in two general groups, viscosity (SAE J300 standard) and performance (SAE J183 standard). Viscosity is measured at both 0°F (−18°C) and 210°F (98.9°C). SAE 5W, 10W, 15W, 20W, and 25W viscosity numbers are related to 0°F (−18°C) measurements; SAE 20, 30, 40, 50, and 60 to 210°F (98.9°C) measurements. Multigrade oils, such as SAE 5W-20, SAE 10W-30, and SAE20W-40, are those which have viscosity characteristics meeting requirements at both temperatures. Multigrade oils can be used throughout the year, reduce

friction and wear, and provide better lubrication and fuel economy during a cold start; e.g., the viscosity of an SAE 20W-40 oil will be less than that of an SAE 40 oil at ambient conditions.

The SAE performance classification includes engine oils ranging in performance quality from that of a straight mineral oil plus a small amount of pour-point and/or foam depressants, to those required in severe passenger car service and heavy-duty truck and off-highway operation. Letter designations such as SG apply to active test techniques for oils for gasoline engines existing since 1988; designations CD and CE apply to oils for naturally aspirated and turbocharged diesel engine service; and designation CD-II applies to oils for two-stroke cycle diesel engines. Performance of these oils is evaluated in both single- and multicylinder engine tests. Factors measured include rust, wear, deposit formation, oil consumption, oil thickening, and fuel economy.

REFERENCES: Most of the foregoing information on engine lubrication is covered in greater detail, and references are provided in the ''SAE Handbook.'' See also the list of references in Sec. 6.11.

AIR POLLUTION

(See also Sec. 18.1)

General Relationship

Air pollution from automobile engines (smog) was first detected about 1942 in Los Angeles, CA (see Haagen-Smit, *Sci. Amer.*, **210,** no. 1, 1964, p. 25). Smog arises from sunlight-induced photochemical reactions between nitrogen dioxide and the several hundred hydrocarbons in the atmosphere (see Caplan, SAE Pt. 12, p. 20). Undesirable products of the reactions include ozone, aldehydes, and peroxyacylnitrates (PAN). These are highly oxidizing in nature and cause eye and throat irritation. Visibility-decreasing nitrogen dioxide and aerosols are also formed.

According to an EPA study, transportation engine emissions comprised 42 percent by weight of all U.S. artificial air pollutants in 1980. At that time, virtually all the transportation CO, about one-half of the hydrocarbons and oxygenates, and about one-third of the nitrogen oxides came from gasoline engines. Diesel engines account for particulates. Table 9.6.20 illustrates the situation extant in 1980 and addresses the five categories of air pollutants listed which were attributed to all transportation sources and highway vehicles. Even more current statistics will show trends only, inasmuch as there is a sizable time lag between the collection and publication of data. Suffice it to say that improvements have been noted in many parts of the country and have resulted from improved engine design, universal use of catalytic converters in new automobiles, improved fuel technology, and so forth.

Locally, artificial pollutants can exceed those from natural sources and can become dominant. However, emitted mass does not reflect pollution severity. For example, 1 unit mass of oxides of sulfur may equal 30 units of carbon monoxide or more in terms of health effects.

Emissions from internal combustion engines include those from blowby, evaporation, and exhaust. These can vary considerably in amount and composition depending on engine type, design, and condition; fuel-system type; fuel volatility; and engine operating point. For an automobile without emission control it is estimated that of the organic emissions, 20 to 25 percent arise from blowby, 60 percent from the exhaust, and the balance from evaporative losses primarily from the fuel tank and to a lesser extent from the carburetor (if used). All other nonhydrocarbon emissions emanate from the exhaust.

At least 200 hydrocarbon (HC) and other organic compounds have been identified in exhaust. Some of those compounds, such as the olefins, are referred to as **reactive** since they react rapidly and to a great extent in the atmosphere to form smog products. Others such as the paraffins are virtually unreactive. Methane is deemed unreactive. Of several reactivity scales, the Carter scale (Table 9.6.21) has become most widely used. It ranks nonmethane organic gases (NMOGs) on a scale from 0 to 11 g ozone/g NMOG. The reactivity of a mixture is computed as the sum of the grams of ozone from individual species, with lower levels typical of the larger molecules. A value of 3.42 g NMOG/mi is representative of non-catalyst-equipped cars.

Table 9.6.21 Carter Scale of Reactivity

Nonmethane organic gas	g ozone/g NMOG (approx. range)
Methane	0
C_2-C_5 paraffins	0–1.5
Acetylene	0
Benzene	0.42
C_6 + C_8 paraffins	2–0.2
Toluene (and higher aromatics)	2–10
Alcohols	0.5–2.5
I-alkenes	9–1
Diolefins	11–9
Aldehydes	3–7
Internally double-bonded olefins	10–3

SOURCE: Carter and Atkinson, *Emu. Sc. Tech.* **23,** 1989, p. 864.

Atmospheric oxides of nitrogen (commonly termed NO_x) are a mixture of nitrogen oxide (NO) and nitrogen dioxide (NO_2). Oxides of sulfur (SO_z) arising from fuel sulfur are a mixture of SO_2, SO_3, and SO_4. Sulfuric acid may be a product in some oxidizing catalytic exhaust converters.

Emission quantities are most often reported as mole percent for the principal exhaust constituents (CO, H_2O, CO_2, H_2, O_2, N_2) and as moles per million moles (ppm) for trace constituents (HC, NO). Parts per million may be assumed to be equal to volume percent and ppm by volume. Hydrocarbons are reported commonly as either ppm equivalent hexane (C_6H_{14}) or as ppm carbon (C_1). One ppm C_6 is equivalent to 6 ppm C. Other reporting parameters are (1) specific emission rate, g pollutant/kWh or g pollutant/vehicle mi (km), and (2) emission index, g pollutant/1,000 g fuel.

Emission Sources

Homogenous Combustion and Spark Ignition Engines ORGANIC GAS EMISSIONS. Emissions of unburned hydrocarbon and other organic gases can be generated from several sources, as shown in Fig. 9.6.63. The largest source is thought to be flow during compression of un-

Table 9.6.20 Estimated Total Annual U.S. Emissions from Artificial Sources (1980)*

	Carbon monoxide	Hydrocarbons	Sulfur oxides	Nitrogen oxides	Particulates
Total, Tg/yr	85.4	21.8	23.7	20.7	7.8
All transportation, %	81	36	3.8	44	18
Highway vehicles, %	72	29	1.7	32	14

* Current data from the EPA and other sources will provide an updated version of these values. Rapid changes result from greater use of catalytic converters (universal in all spark ignition automotive engines), weather conditions, seasonal variations, etc.

SOURCE: EPA Rept. 450/4-82-001, 1982.

burned fuel-air mixture into the crevice volume between the piston and liner above the top ring; the crevice mixture returns to the combustion during expansion at a time when it is too late to burn. A similar mechanism is caused by the absorption of fuel vapor into oil layers and combustion chamber deposits during compression, and subsequent desorption during expansion and exhaust.

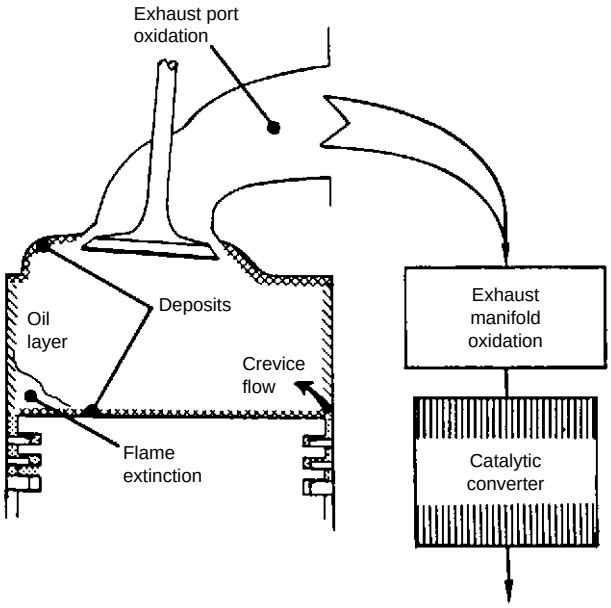


Fig. 9.6.63 Emission sources of unburned hydrocarbon and other organic gases. *(Amann, ''The Automotive Spark-Ignition Engine—An Historical Perspective,'' ASME-ICE vol. 8, 1989, pp. 33–45.)*

Emissions also are generated from flame extinction or wall quenching due to the inability of the flame to propagate totally to a surface or into a crevice. Daniel (''Sixth Symposium on Combustion,'' Reinhold, 1957, p. 886) described the result of wall quenching to be a visibly dark layer of unburned and partially burned fuel-air mixture left adjacent to all engine combustion chamber surfaces. Subsequent studies showed that the layer adjacent to single walls is largely burned prior to being exhausted. Thus the organic emission from the engine is mainly due to crevice quenching, of which the top piston land clearance is the major source.

Under engine operating conditions where cylinder exhaust residual is high, incomplete flame propagation may leave a portion of the combustion chamber contents unburned. This can produce organic emissions of several thousand ppm C_6. When residual exceeds 15 percent by weight, some incomplete combustion may occur. In some stratified-charge, direct-injection designs, bulk quenching as the flame comes in contact with cooler or leaner gases may also be present.

Wankel engine organic emissions arise from the above sources, but significant additional quantities may arise from mixture leakage through the leading rotor apex seal directly into the exhaust.

In two-stroke engines, where intake pressure exceeds exhaust during the scavenging or overlap period, raw-mixture loss to the exhaust creates very high organic emissions. Emissions from such two-cycle engines may be 10 times higher than those from four-cycle engines. Note also that most two-stroke designs utilize once-through lubrication; i.e., the oil is mixed with the fuel, typically in a fuel/oil ratio of 50 : 1. Hence, increased deposits and oil films generate organic emissions by physical and chemical storage and release of fuel components.

Any unburned hydrocarbons and other organic gases escaping the cylinder during exhaust can still oxidize in the exhaust temperature and manifold if the exhaust temperature is high enough for the available residence time and if there is sufficient oxygen present. In addition, where combustion is close to stoichiometric, oxidation of unburned gases is accomplished in catalysts (see later).

CARBON MONOXIDE EMISSIONS. The principal source of exhaust carbon monoxide (CO) is rich-mixture combustion. Figure 9.6.16 shows exhaust composition as mixture strength is varied. Cycle-to-cycle and cylinder-to-cylinder maldistribution of the fuel-air mixture, especially during transients, increases CO. Additional CO may arise from incomplete combustion of organics in an exhaust treatment device.

OXIDES OF NITROGEN EMISSIONS. The principal oxide of nitrogen produced inside the engine cylinder is nitrogen oxide (NO). However, since the residence time of gases within the reaction zone is very short, NO forms primarily in postflame combustion gases which do not attain chemical equilibrium. The formation depends primarily on peak combustion gas temperature and to a lesser extent on oxygen content. Once formed, NO tends to be frozen and persists in high, nonequilibrium concentrations during expansion and exhaust. Nitrogen oxide variation with mixture strength is shown in Fig. 9.6.16. Other important variables that affect NO emissions include the amount of exhaust gas recirculation (EGR) and spark timing. When mixtures are more than 30 percent lean or where air is injected into the exhaust, about 5 percent of NO may be oxidized to NO_2 prior to leaving the exhaust pipe. Typically NO and NO_2 emissions are measured together with a chemiluminescent analyzer, and their sum is reported as NO_x emissions.

ALDEHYDE EMISSIONS. Aldehydes are incomplete combustion products of the oxidation of organics. They arise from the same wall quenching and incomplete combustion mechanisms as the unburned organic emissions themselves, but in addition may be formed in low-temperature exhaust system oxidation reactions. Typically gasoline engine exhaust contains 100 ppm C aldehyde, less for rich and more for lean mixtures. Methanol fuel tends to produce high emission levels of formaldehyde.

PARTICULATE EMISSIONS. Solid-particle emissions from gasoline engines consist primarily of carbon, metals, and metal oxides. Metals arise from fuel and lubricant additives such as lead, phosphorus, zinc, and barium, as well as engine wear and corrosion. Sulfuric acid particles may be emitted as a result of reactions involving SO_2 over platinum metal catalysts. SO_2 is formed during combustion from fuel sulfur. Liquid aerosols of heavy hydrocarbons are emitted not only during starting, but also during running. Of these, polynuclear aromatics (PNAs) are health concerns.

Heterogeneous Combustion and Diesel Engines The hydrocarbon, carbon monoxide, and nitrogen oxide emission formation in heterogeneous combustion processes in diesel engines derives from the same basic mechanisms as in the spark ignition engine. However, the relative importance of each is different. In the diesel engine, the emission formation in combustion can be considered to arise from a broad distribution of individual fuel-air elements, some of which burn rich and others lean. Patterson and Henein (Emissions from Combustion Engines and Their Control, *Ann Arbor Sci.*, 1972, p. 260) have suggested the various emission sources in the combustion of a fuel spray injected into swirling air, as shown in Fig. 9.6.64. In addition, fuel impingement on chamber surfaces contributes to unburned hydrocarbons and carbon smoke particles. Hydrocarbon emissions from diesels contain many

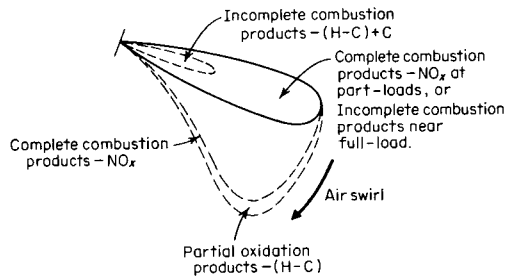


Fig. 9.6.64 Emission formation in a fuel spray injected in swirling air.

heavy fuel types. Normally, heated sampling lines are required to avoid HC condensation during sample analysis.

The amount of carbon, carbon monoxide, hydrocarbon, and nitrogen oxide emissions from diesel engines varies considerably with engine and fuel system design as well as with the operating point. Moreover, because of the high air mass flow rate at partial load, emissions from diesel and gasoline engines cannot be compared directly except on a specific emission basis. Diesel emissions as mole percent or ppm may be corrected for dilution to an equivalence ratio of unity for comparison with spark ignition engine data. On a corrected basis, diesel engines emit in the range of 50 to 500 ppm C_6 and NO_x up to 3,000 ppm. Diesel engines which run lean tend to show higher NO_2/NO ratios (between 10 and 30 percent), especially at light loads when cooler regions quench any NO_2 formed in the reaction zone (Hiliard and Wheeler, SAE 790691, *SAE Trans.*, **88,** 1979). Carbon monoxide emission from the diesel engine is very low, typically under 1,000 ppm (0.1 percent).

Diesel engines emit considerable amounts of carbon smoke and odor constituents. Most carbon arises from pyrolysis in rich combustion regions. Spindt, Barnes, and Somers (SAE paper 710605, 1971) have identified low concentrations of many potential oxidation products and certain fuel fractions as being the odor-producing compounds. Human-panel techniques are commonly used to measure odor.

Strategies for Controlling Spark Ignition Engine Emissions: Effects of Engine Design Variables

Combustion Chamber Shape The shape of the combustion chamber affects organic emissions through a change in surface area and thus heat loss. Surface area relative to the entire clearance volume is an indicator of the volume fraction of organic emissions. This may be stated as

$$\text{Organic, ppm} \propto \frac{A_s}{V_c}$$

where A_s is the chamber surface area and V_c the clearance volume at TDC. According to Sheffler (SAE, Pt. 12, p. 60), the following change the surface/volume ratio, thereby changing hydrocarbon emissions: chamber shape, bore/stroke ratio, compression ratio, and cylinder displacement. In automotive reciprocating engines, surface/volume ratios range from 5 to 8 in^2/in^3 (2 to 3 cm^2/cm^3). Wankel rotary engines have surface/volume ratios about 50 percent higher. In well-designed engines, combustion chamber shape has little, if any, effect on CO. A fast-burning chamber increases NO because of an increase in peak temperature.

Ignition System Under most running conditions, improvements in spark energy, rise time, and spark duration produce little improvement in emissions. However, a high-energy, long-duration spark in conjunction with extended electrodes and wider spark gap has been found to minimize the increase in organic emissions due to misfire, as mixtures are leaned well beyond what is chemically correct. (See Quader, *SAE Trans.*, **85,** 1976.)

Exhaust back pressure and valve overlap affect organics and NO through changes in the exhaust residual. Increasing overlap or back pressure lowers NO_x and may lower organic emissions. At very light loads, excessive residual leads to incomplete combustion, thereby increasing organic emissions.

Induction Systems and Fuel-Air Mixture Preparation Optimum mixture preparation and distribution are a key to minimum emissions. With carburetors, high-velocity manifolds, increased manifold heating, and staged throttles have shown emission reductions (see Bartholomew, SAE Pt. 12, 1971). Port fuel injection should be well-targeted on the intake valve to minimize hydrocarbon and soot emission. (See Kosowski, SAE paper 850294, 1985.)

Effects of Operating Variables

Mixture ratio and ignition timing are the most important operating variables affecting emissions. Figure 9.6.16 shows the effect of the mixture ratio on major emission amounts. Retarding ignition timing from the best-efficiency setting reduces HC and NO_x emissions significantly. Excessive retard increases HC and CO emissions. Increasing engine speed reduces HC emissions as ppm. NO_x emissions increase as load increases. Increasing coolant temperature normally reduces HC emissions and increases NO_x emissions.

Limiting fuel enrichment during warm-up is critical to low emissions, especially without exhaust treatment devices. With exhaust treatment, emissions may be reduced to virtually zero once the system is warmed up. Thus the only significant emissions from such systems are those generated during starting and warm-up.

Exhaust Treatment Devices

Catalytic or noncatalytic exhaust treatment devices may be used to clean up remaining exhaust emissions. Thermal reactors are noncatalytic devices which rely on homogeneous bulk gas reactions to oxidize CO and HC. Commonly they appear to be enlarged exhaust manifolds, but they may have internal baffling. In thermal reactors, NO_x is unaffected. Reactions are enhanced by increased exhaust temperature (reduced compression ratio, retarded timing) or by increasing exhaust combustibles (rich mixtures). Typically, temperatures of 1,500°F (800°C) or more are required for peak efficiency. Usually the engine is run rich to give 1 percent CO, and air is injected into the exhaust. A typical thermal reactor residence time is 100 ms. Required reactor volume is commonly 1.5 times engine displacement. Maximum allowable reactor temperature is governed by materials and is typically 1,750°F (950°C). Thermal reactors are seldom used because the required engine calibration produces low fuel economy.

Catalytic systems are capable of reducing NO_x as well as oxidizing CO and HC. However, a reducing environment for NO_x treatment is required which necessitates a richer than chemically correct engine mixture ratio. A two-bed converter may be used in which air is injected into the second stage to oxidize CO and HC. It is highly efficient but results in lower fuel economy. Single-stage, three-way catalysts (TWCs) are widely used, but they require extremely precise fuel control to be effective. As illustrated in Fig. 9.6.65, only in the vicinity of stoichiometric ratio is the efficiency high for all three pollutants. Such TWC systems employ either a zirconia or a titanium oxide exhaust oxygen sensor (EGO, HEGO, or UEGO) and a feedback fuel system to maintain the required mixture strength near stoichiometric. Although a

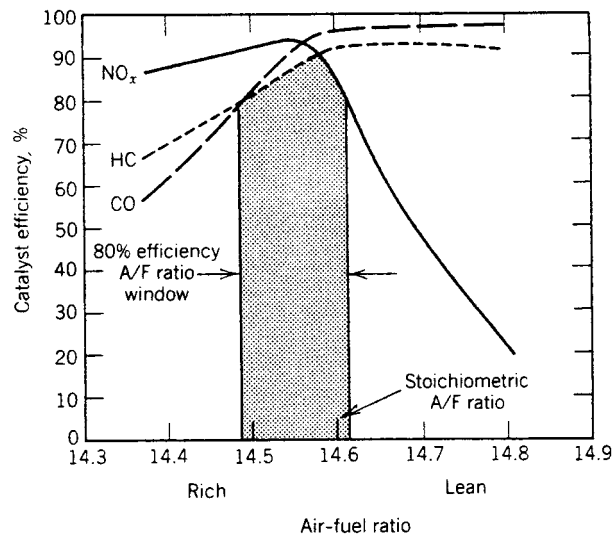


Fig. 9.6.65 Conversion efficiency of three-way catalyst (TWC) as a function of air/fuel ratio. *(Kummer, Prog. Energy Combust. Sci.,* **6,** *1981, pp. 177–199. Reprinted with permission.)*

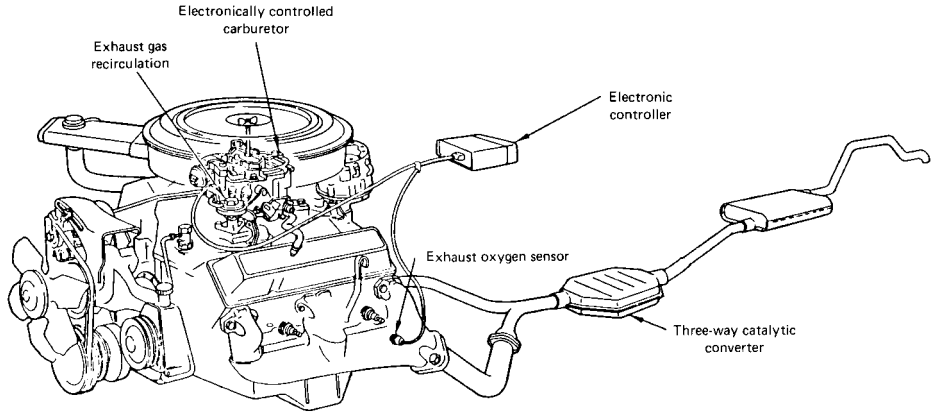


Fig. 9.6.66 Closed-loop three-way catalyst (TWC) system.

carburetor is shown in Fig. 9.6.66, the feedback concept is similar for both central port and multiport fuel injection. Table 9.6.22, which lists average hot stabilized conversion efficiencies during an FTP cycle, demonstrates that many modern vehicles achieve average conversion efficiencies of over 90 percent for all three pollutants once the catalyst has reached normal operating temperatures. Efficiencies are considerably lower on the high-speed acceleration cycle.

Catalyst support beds may be of the pellet or honeycomb (monolith) type. Materials for reducing catalysts include rhodium, Monel, and ruthenium. Materials for oxidizing catalysts include platinum and palladium. The amount of active catalyst material required may be as low as 0.05 troy oz (1.4 g) per vehicle system. In 1996, catalyst volume is about 50 to 100 percent of the engine displacement. The weight of the catalyst bed typically is about 2.2 to 5 lb (1 to 2.3 kg) excluding the metal container. Many aged catalysts can oxidize at 85 percent or higher efficiency at temperatures as low as 800 to 1,000°F (430 to 540°C). Above 1,500°F (820°C), they deteriorate rapidly. Compared with thermal reactors, catalytic converters warm up slowly because of the larger mass of material. Catalytic converters may be located farther from the engine than thermal reactors, since converters operate efficiently at lower temperatures. However, this slows warm-up. Monolithic converters, which can normally be operated hotter, can be smaller and can be placed closer to the engine. Catalysts with metal substrates (monoliths) warm up even faster and are more resistant to high-temperature deterioration. A small cold-start catalyst located near the engine, especially one with supplementary heating, is very effective. (See Laing, *Auto. Eng.,* Apr. 1994, p. 51.)

Strategies for Controlling Diesel Emissions: Engine Design and Operating Variables

Horrocks (*Proc. Inst. Mech. Engr.,* **208,** 1994, pp. 289–297) has summarized the strategies for controlling particulate and HC emissions as follows: (1) optimization of fuel injection and air motion; (2) good atomization of the fuel spray, particularly at idle, at partial load, and during transients; (3) precise control of fuel injection timing, particularly during transients; (4) minimization of parasitic losses in the combustion chamber; (5) low sac volume or valve cover orifice nozzles for direct injection; (6) minimization of contribution of lubricating oil; (7) central injector and combustion chamber for high-speed, direct injection engines; (8) modulated EGR; (9) electronic controls; (10) rapid warm-up of engine.

Control of NO_x emissions from diesels presents additional challenges since suppression of NO formation tends to cause an increase in particulates. Nevertheless, the following strategies are frequently used (Horrocks, op. cit.): (1) injection retard; (2) injection rate shaping; (3) intercooling of turbocharged engines; (4) centralized combustion for high-speed, direct injection; (5) modulated EGR; (6) electronic controls.

Exhaust Treatment Devices

Since diesel engines burn lean overall, the exhaust will always contain excess oxygen. Hence the reduction of NO_x with conventional three-way catalysts is not feasible. Currently, NO_x is removed from the diesel exhaust by either selective catalytic reduction (e.g., Feeley et al., SAE-

Table 9.6.22 Three-Way Catalyst Conversion Efficiency, Percent

			Hot, stabilized FTP			High-speed cycle		
Vehicle	Year	Disp (L)	HC	CO	NO_x	HC	CO	NO_x
Escort	1993	1.9	—	—	—	89.3	41.2	57.8
Taurus	1993	3.0	94.4	90.2	69.9	90.8	56.5	81.5
Mustang	1993	5.0	96.3	97.4	71.9	96.6	89.4	73.9
Cutlass Supreme	1994	3.1	98.3	91.5	84.7	96.0	70.8	84.1
Grand Prix	1994	3.4	97.5	95.1	88.3	88.0	60.2	76.1
Olds 98	1994	3.8	96.7	83.9	94.0	96.9	76.4	92.1
Seville	1994	4.6	97.3	87.6	76.6	93.1	60.1	74.7
Custom Cruiser	1992	5.7	97.1	87.4	90.2	92.9	47.0	62.6
Saturn	1994	1.9	—	—	—	91.6	43.2	91.1
Metro	1993	1.0	—	—	—	85.3	23.0	70.1
Grand Am	1993	2.3	98.3	92.1	93.9	93.0	63.9	98.0
Civic	1992	1.5	98.5	92.8	95.2	86.7	57.6	97.2
Mirage	1993	1.8	98.8	96.4	98.9	94.4	83.6	99.0
Camry	1993	3.0	95.7	92.8	98.4	95.1	79.5	96.3
Corolla	1993	1.8	91.8	86.3	94.9	85.3	66.9	92.3
420 SEL	1993	4.2	99.3	94.4	97.7	99.3	90.9	96.5

SOURCE: German, SAE paper 950812.

Table 9.6.23 Federal Gasoline and Diesel Engine Vehicle Emission Standards (EPA 6684200)*†

Year(s)	Test procedure	Nonmethane organic gases, g/mi	Nonmethane hydrocarbons, g/mi	Carbon monoxide, g/mi	Spark-ignited oxides of nitrogen, g/mi	Diesel oxides of nitrogen, g/mi	Particulates, g/mi	Cold carbon monoxide, g/mi	Evaporative losses, g/test
Automobiles: 50,000-mi schedule (100,000-mi); phased in; light-duty trucks up to 3,750-lb loaded vehicle weight (LVW) (*Fed Reg,* v. 56, no. 108, June 5, 1991)									
1995–2002	FTP tier I	0.257 (0.319)	0.25 (0.31)	3.4 (4.2)	0.4 (0.6)	1.0 (1.25)	0.08 (0.1)	10	2
2003 and later	FTP-tier II	(0.0128)	(0.0125)	(1.7)	(0.2)	(0.4)	(0.1)	10	Not specified
Light-duty trucks between 3,751- and 5,750-lb LVW, phased in (*Fed Reg,* **56,** no. 108, June 5, 1991)									
1995–2002	FTP-tier I		0.32 (0.40)	4.4 (5.5)	0.7 (0.976)	— (0.97)	0.08 (0.1)		2
Light-duty trucks over 5,750-lb LVW, phased in (*Fed Reg,* **56,** no. 108, June 5, 1991)									
1995–2002	FTP-tier I		0.39 (0.56)	5.0 (7.3)	1.1 (1.53)	— (1.53)	— (0.12)		2

		Total hydrocarbons	Carbon monoxide	Oxides of nitrogen	Hydrocarbons + oxides of nitrogen	Particulates	Accel/Lug Pea %
Heavy-duty diesel, emission in g/(bhp·h)							
1994	EPA Trans.	1.3	15.5	5.0		0.10 (0.05 buses 1996)	20/15/50
1998	EPA Trans.	1.3	15.5	4.0		0.10 (0.05 buses)	20/15/50
Off-highway diesels 37 kW and greater, emission in g/kWh (*Fed. Reg.* **59,** no. 116, June 17, 1994); phase-in 1996 to 2000							
1996–2000							
I. 130 kW or more		1.3	11.4	9.2		0.54	20/15/50
II. 75–130 kW; 8 mode				9.2			20/15/50
III. 37–75 kW				9.2			20/15/50
Motorcycles (50 cm^3 and over), emission in g/km (*Fed. Reg.* **42,** no. 3, June 28, 1979)							
1980 and later	FTP	5	12				0.6
Small utility engines, emisssion in g/kWh (*Fed. Reg.*, **60,** no. 127, July 3, 1995)							
1997 and later							
(SAE cycle J-1088)							
I. Nonhandheld under 225 cm^3			469		16.1		
II. Nonhandheld 225 cm^3 and over			469		13.4		
III. Handheld under 20 cm^3		295	805	5.36			
IV. Handheld 20–50 cm^3		241	805	5.36			
V. Handheld 50 cm^3 and over		1,661	603	5.36			
SI outboard and personal watercraft. (*Fed. Reg.* **59,** No. 216, Nov. 9, 1994)							
Standard calls for average 75% reduction in HC emission from 1990 baseline levels Baseline HC, g/kWh = 151 + 557/Power$^{0.9}$ Final = 0.25 (baseline HC)							
1998–2006	ICOMIA	Phase in	400	6			

* Numbers in parentheses refer to 100,000 mi standard.
† Refer to the standard, which is periodically updated, for further details of terms, etc.

paper 950747) or catalyzed thermal decomposition of NO into O_2 and N_2. In addition, lean NO_x catalysts are under development.

Diesel particulate traps have been developed which employ ceramic or metal filters (see Wade et al., SAE 830083, 1983). Thermal and catalytic regeneration can burn out the material stored. Particulate standards of 0.2 g/mi may necessitate such traps. Both fuel sulfur and aromatic content contribute strongly to particulates. Catalysts have been developed for diesels which are very effective in oxidizing the organic portion of the particulate (see Ball and Stack, SAE paper 902110, 1990).

Emission Regulations

Light-Duty Passenger Cars and Trucks Emissions are measured on a light-duty driving cycle termed the **Federal Test Procedure (FTP)**. This involves various accelerations, decelerations, and cruise models on a chassis dynamometer. The vehicle is started after a 12-h soak in a 60 to 86°F (16 to 30°C) ambient temperature. Top cycle speed is 56.7 mi/h (91.3 km/h). Test procedures, fuel, and instrumentation are detailed in the Code of Federal Regulations (70, parts 85 and 86, 1986) for cars, trucks, and diesel engines.

In the FTP, the exhaust is introduced into a constant-volume sampler (CVS) in which ambient air is mixed with raw exhaust. Emissions are calculated in grams per mile from average concentrations and total sampler volume flow. In theory, the constant-volume sampler yields true mass emission results. Three sample bags have been used to segregate emissions during the key cycle portions of cold start, hot restart, and cruise. The emission concentrations from each bag are weighed, and the sums are used for certification purposes. Particulate emissions from diesels are measured by weighing filters. Summaries of light-duty, passenger car, and truck standards can be found in Table 9.6.23 (Federal) and Table 9.6.24 (California).

In order to meet the **ultra-low emission vehicle (ULEV)** standards in the cold phase of the FTP-75 test cycle, the time period where the standard three-way catalyst has low efficiency (light-off time) needs to be reduced drastically. Exhaust gas ignition (run rich, pump air into the exhaust, and use ignition device such as glow plug to ignite mixture in exhaust), electric-heated catalysts, burner-heated catalysts (Achleitner et al., SAE paper 950479), stainless-steel or double-wall exhaust pipes for insulation, and insulated catalysts are some of the enabling technologies currently being explored. In addition, more sophisticated transient fuel strategies and closed-loop control are employed.

Table 9.6.24 California Air Resources Board Exhaust Emission (CARB) Standards for Passenger Cars (Phased in), 50,000-mi (100,000-mi) Standards

CARB	Effective date	Nonmethane organic gases, g/mi	Carbon monoxide, g/mi	Oxides nitrogen, g/mi
1993 base	1993	0.25 (0.31)	3.4 (4.2)	0.4
TLEV	1995	0.125 (0.156)	3.4 (4.2)	0.4 (0.6)
LEV	1995	0.075 (0.090)	3.4 (4.2)	0.2 (0.3)
ULEV	1995	0.040 (0.055)	1.7 (2.1)	0.2 (0.3)
ZEV	1998	0.000 (0.000)	0.0 (0.0)	0.0 (0.0)

TLEV = transitional low-emission vehicle; LEV = low-emission vehicle; ULEV = ultra-low-emission vehicle; ZEV = zero-emission vehicle.

Heavy-Duty Engines Emissions from heavy-duty diesel engines are determined in transient engine dynamometer tests. This involves starts, stops, and speed/load changes. Gaseous and particulate emissions are determined in g/(bhp · h). These standards are shown in Table 9.6.23.

Additional Emission Control The California Air Resources Board (CARB) and the EPA have set standards for a variety of off-highway and marine engines. Some of these are given in Table 9.6.24. Future standards may include a wider group of off-highway engines, locomotives, and light aircraft.

9.7 GAS TURBINES

by John H. Lewis and Albert H. Reinhardt

REFERENCES: Kerrebrock, ''Aircraft Engines and Gas Turbines,'' MIT Press. Koff, Spanning the Globe with Jet Propulsion, 1991, AIAA paper 2987. ''Turbine Engine Directory,'' Flight Int., June 8–14, 1994. Hill and Peterson, ''Mechanics and Thermodynamics of Propulsion,'' Addison-Wesley. Bejan, ''Advanced Engineering Thermodynamics,'' Wiley. Horlock, ''Combined Cycle Powerplants,'' Pergamon. ''Gas Turbine World Handbook,'' published annually, Pequot Publishing, Inc.

INTRODUCTION

The **gas turbine** is a turbine-type engine which operates with gas (usually air) as distinguished from steam or water as the working substance. In its most basic configuration, the gas-turbine engine consists of a compressor to draw in and compress the gas, a combustor or heat source to add energy to the compressed gas, and a turbine to extract power from the heated gas flow. Practical applications of gas turbines first occurred from 1939 to 1941. In 1939, the firm of Brown Boveri of Switzerland used a gas turbine to generate electricity. Also in 1939, the first flight of an aircraft powered by a gas turbine developed by Hans von Ohain took place in Germany. Another aircraft gas turbine was developed by Frank Whittle, who powered an aircraft in 1941 in England. From these early applications, the gas turbine has been developed to the point where today it is the most important aircraft power plant in use. In addition to its aviation applications, the gas turbine is a significant power plant for electric power generation, pipeline pumping power, and marine propulsion. It is a key component of combined-cycle power plants and cogeneration plants (see Fig. 9.7.18 and Sec. 9.4).

Depending upon its development roots and applications, the gas turbine is referred to by many names. In aviation, it is called a jet engine, a turbojet, a jet turbine, a turboprop, a turbofan, and a fanjet. In land and marine applications it is called a gas turbine, a turboshaft, a combustion turbine, and an industrial turbine. In the industrial world, engines generally fall into two categories. **Heavy-duty** or **frame machines** refer to gas turbines designed and built strictly for ground applications. **Aeroderivative machines,** as the name implies, are gas turbines originally designed for aircraft use and adapted for industrial applications. A typical aeroderivative is shown in Fig. 9.7.1. Future trends to marry heavy industrial and aerospace technologies will blur this distinction between the two types.

Gas turbines offer significant advantages compared with other types of engines because they are compact, lightweight, reliable, and efficient. They are capable of rapid start-up, follow transient loading well, and can be operated remotely or left unattended. They have a long service life, long service intervals, and low maintenance costs. Cooling fluids are usually not required. Their compact size high power-to-weight ratio, and high reliability make gas turbines the ideal power plant for aircraft and for portable power plants. In industrial applications, the engine's small footprint allows relatively small foundations and enclosure buildings.

FUELS

Gas turbines use a wide variety of **gaseous and liquid fuels**. For ground and marine applications, natural gas and light distillate oils are most commonly used, but gasified coal and heavy residual oils can also be

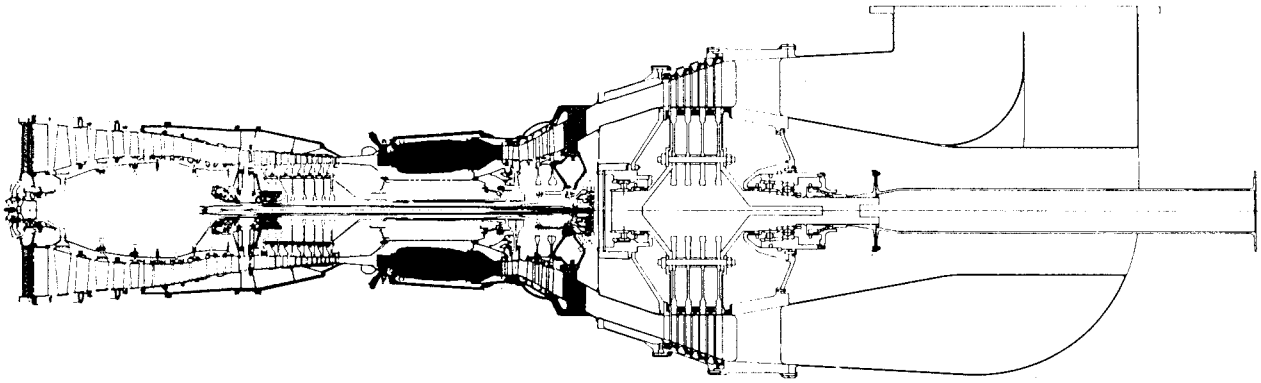


Fig. 9.7.1 Cross section of Pratt and Whitney FT8 gas turbine derived from the JT8D commercial aircraft engine.

used. When fuels that contain significant amounts of sulfur, salt, vanadium, and other metals are used, the hot-section components such as the combustor and turbine must be designed to withstand the corrosive effects of these impurities. For aircraft applications, specific kerosine-type fuels have been developed that control important qualities of the fuel, to obtain maximum performance.

THERMODYNAMIC CYCLE BASIS

The gas turbine operates on the principle of the **Brayton** (or **Joule**) **cycle**. The working substance, usually ambient air, is compressed adiabatically to high pressure where heat is added at constant pressure to elevate the temperature, at which point it is expanded, also adiabatically, back to the original pressure. Because of the expansion from a higher temperature, the work extracted in the expansion process exceeds the work required for compression by an amount that equals the net output work of the cycle. The cycle is completed at the original pressure with heat rejection in the exhaust. See Fig. 9.7.2. At the end of the expansion process, the working substance temperature is higher than its initial temperature by an amount that corresponds to the waste energy to be dissipated at constant pressure to the surroundings. Unique to the Brayton cycle are the heat addition and heat rejection processes at (ideally) constant pressure. The constant-pressure process lends itself to continuous flow of the working substance. When coupled with compressors and turbines which operate on the rotating turbomachinery principle

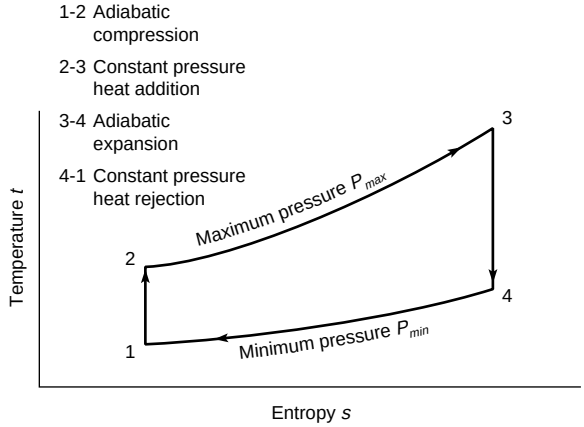


Fig. 9.7.2 Brayton cycle temperature vs. entropy diagram. Ideal cycle.

(see Sec. 14), the entire gas turbine acts as a **continuous flow engine**. The continuously flowing working substance (air) results in very high power density. The gas turbine's intrinsic compactness favors its use over other engine types in many power generation applications.

Temperature-entropy diagrams describe the Brayton cycle. Figure 9.7.2 shows the ideal case where all component efficiencies are 100 percent. Figure 9.7.3 shows the real cycle with nonideal components, including friction losses in the ducting between the compressor and turbine.

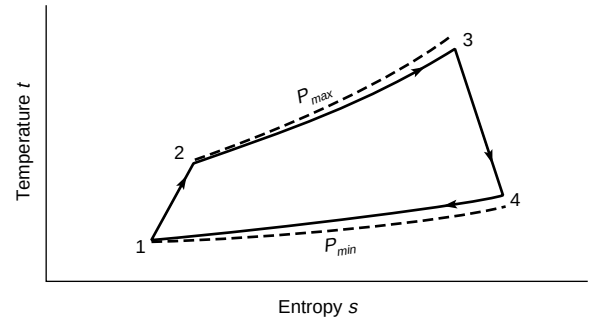


Fig. 9.7.3 Brayton cycle temperature vs. entropy diagram. Real cycle reflects component inefficiencies and pressure losses.

The **ideal Brayton cycle efficiency** η is related to the cycle pressure ratio by the following expression, where $P_{\max}/P_{\min}$ = pressure ratio, R = gas constant, and c_p (the latter being specific heat at constant pressure properties of the working substance):

$$\eta \text{ Brayton, ideal} = 1 - 1/(P_{\max}/P_{\min})^{R/C_p} \qquad (9.7.1)$$

Thus with ideal components, the gas-turbine efficiency will be increasingly high as its characteristics maximum/minimum pressure ratio continues to increase approaching 100 percent when the pressure ratio becomes infinitely large. With real components, however, when the pressure ratio increases, the component inefficiencies become larger contributors to exhaust waste heat. The net result is that efficiency reaches a peak at some level and then falls off. The point where this peaking occurs is directly related to the quantity of cycle heat addition, which, in turn, is related to the maximum cycle temperature. As illustrated in Fig. 9.7.4, the peak shifts to a higher efficiency level at high pressure ratio as the turbine temperature increases. At the same time, the specific power output has a similar trend of increasing up to a peak and then falling off as the pressure ratio increases, while the amount of

specific output power increases continuously with heat addition as determined by the maximum cycle temperature. These trends are illustrated in Fig. 9.7.5.

Among manufacturers and users, the **maximum cycle temperature** is variously referred to as turbine inlet temperature, combustor exit temperature, turbine rotor inlet temperature, or firing temperature. Precise definition depends on the design details for a specific engine.

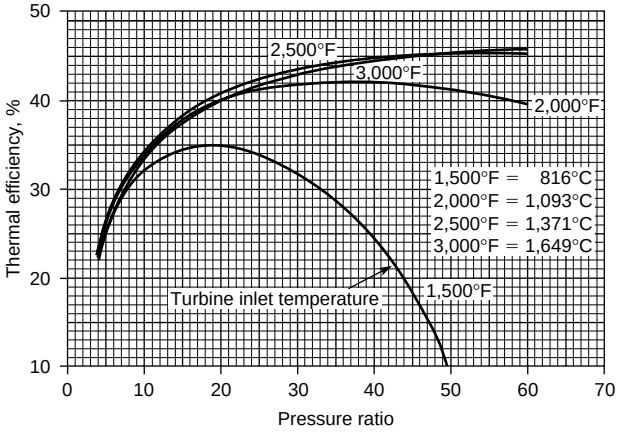


Fig. 9.7.4 Representative Brayton cycle variation of thermal efficiency with pressure ratio at different turbine inlet temperatures.

Specific power refers to output per unit mass flow of the working substance, expressed in hp/(lb · s) or its electrical equivalent kW/(kg · s). As shown in Fig. 9.7.5, a given cycle fixes specific power. When specific power is constant, absolute power then varies directly with the mass flow rate of the working substance. The mass flow quantity, in turn, sets the overall size of the machine

$$\text{Power} = \text{specific power} \times \text{mass flow rate}$$
$$= \text{hp/(lb} \cdot \text{s)} \times \text{(lb/s)}$$
$$= \text{kW/(kg} \cdot \text{s)} \times \text{(kg/s)}$$

Generally, the gas turbine's working substance is atmospheric air which is continuously drawn into the front of the compressor through inlet ducting and is expelled back to atmosphere through exhaust ducting. In this case, the minimum cycle pressure is identical to the surrounding

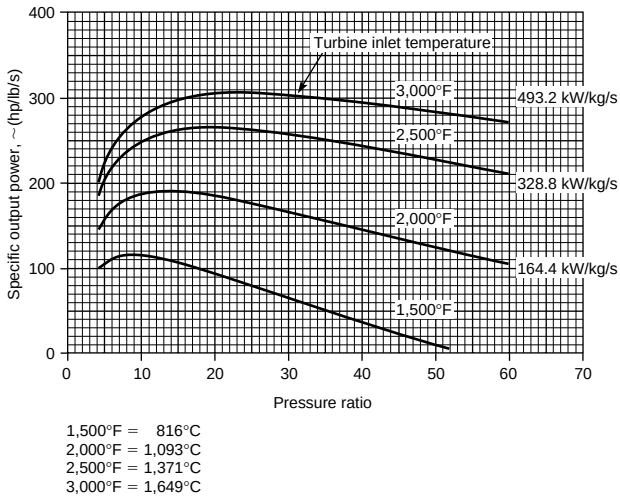


Fig. 9.7.5 Representative Brayton cycle variation of specific output power with different pressure ratios and turbine inlet temperatures.

ambient pressure while the initial working substance temperature is the ambient temperature of the surrounding atmosphere. The schematic in Fig. 9.7.6 shows the typical component arrangement of a continuous flow simple-cycle gas turbine. In this typical arrangement, the heat addition section consists of an internal combustor where fuel is added directly into the through-flowing air and is continuously burned. The rotating compressor is attached to the turbine by a shaft which transmits the power needed for compression. The excess power available from expansion through the turbine at high temperature is fed by a rotating power output shaft to an external load.

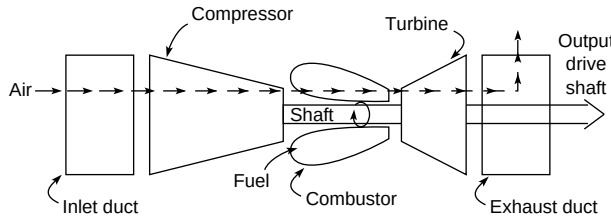


Fig. 9.7.6 Simple Brayton cycle gas-turbine component arrangement.

BRAYTON CYCLE VARIATIONS

Regenerative Cycle (Fig. 9.7.7) In this cycle, air exiting the compressor is directed through a heat exchanger, or **regenerator** before being introduced to the combustion section. Then the heated air exhausting from the turbine passes through the other side of the regenerator. By this

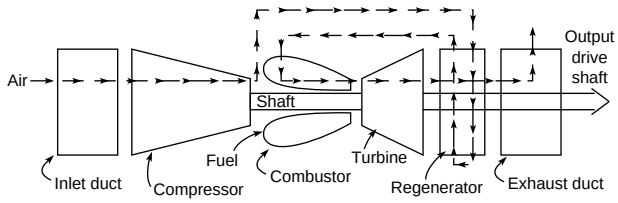


Fig. 9.7.7 Regenerative Brayton cycle gas-turbine component arrangement.

means, some of the exhaust heat is recovered and used as an additional source of energy to heat the high-pressure air, instead of being dissipated as waste heat in the low-pressure exhaust. The effect of regeneration is to increase cycle efficiency, but it is only effective at low levels of pressure ratio. This is because as the pressure ratio increases, the compressor exit air temperature increases and the turbine exhaust temperature is lowered, thereby reducing the regenerator temperature difference necessary for heat transfer. At some point, depending on the maximum cycle temperature, the compressor exit and exhaust temperatures become equal and there can be no further regenerative heat recovery. Figure 9.7.8 compares regenerative cycle trends with those of the simple cycle.

Intercooled Cycle (Fig. 9.7.9) In this cycle, the compression process is split between two compressors. An **intercooling heat exchanger** is placed between the front and rear compressors to reduce the temperature of the working substance. This cycle takes advantage of the thermodynamic principle that the work of compression is directly proportional to the level of the entering temperature. With a lowered inlet temperature, the work input needed to drive the rear compressor is reduced for the same pressure ratio. The reduced compressor work is directly available as increased net cycle output. This is true for both ideal and real cycles. In the ideal cycle, because the intercooler is a source of added heat rejection, the intercooling cycle efficiency is always lower than that of a simple cycle having the same pressure ratio. However, in real cycles, the intercooling effect can help offset the effect of real component efficiencies. In some circumstances, intercooling improves both specific power output and efficiency over simple cycle levels. Theoretically, the maximum power output occurs when the total

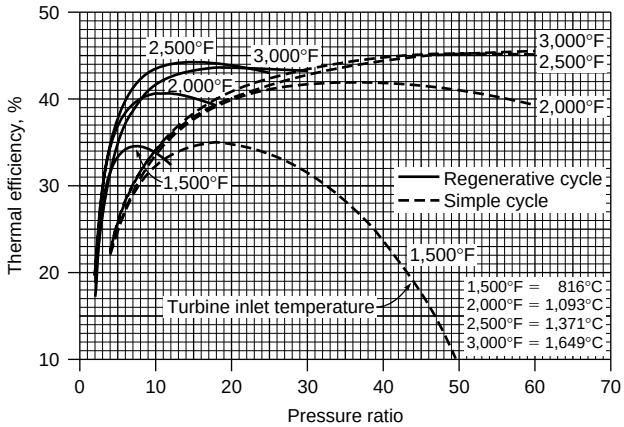


Fig. 9.7.8 Regenerative Brayton cycle variation of thermal efficiency with pressure ratio at different turbine inlet temperatures.

pressure ratio is split evenly between front and rear compressors. However, maximum cycle efficiency occurs at a split which depends on the turbine inlet temperature as well as the efficiency levels of compressors and turbines. Typically, peak efficiency occurs when intercooling takes

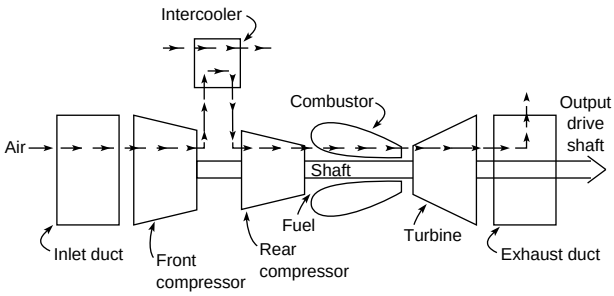


Fig. 9.7.9 Intercooled Brayton cycle gas-turbine component arrangement.

place well in front of the midpoint of the compression process. Figures 9.7.10 and 9.7.11 compare intercooled thermal efficiency and specific power output trends for pressure-ratio splits optimized for power output and for pressure-ratio splits optimized for efficiency.

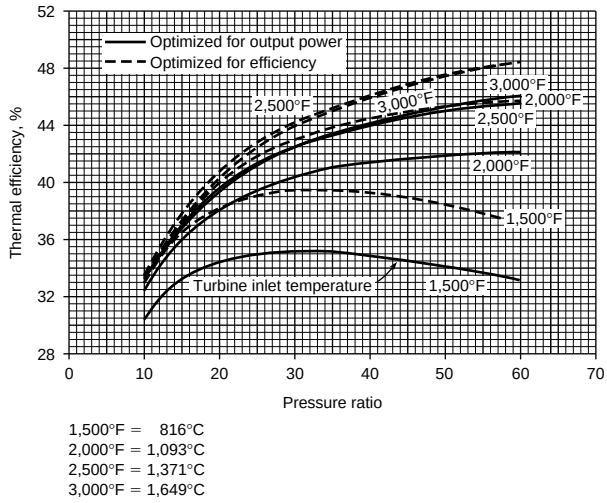


Fig. 9.7.10 Intercooled Brayton cycle variation of thermal efficiency with pressure ratios at different turbine inlet temperatures, optimized for output power and for efficiency.

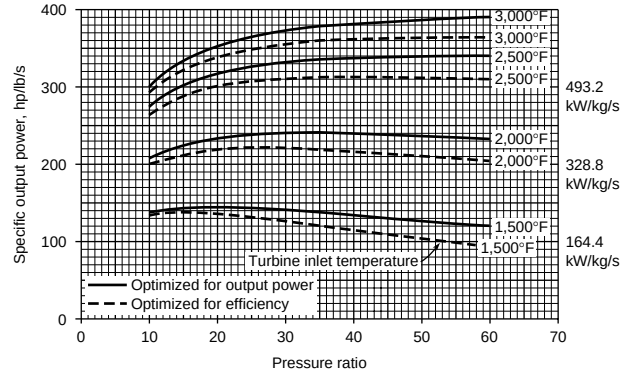


Fig. 9.7.11 Intercooled Brayton cycle variation of specific output power with pressure ratio at different turbine inlet temperatures, optimized for output power and for efficiency.

Reheat Cycle (Fig. 9.7.12) In this cycle, the turbine expansion process is interrupted at some intermediate point before it reaches minimum (or ambient) pressure in the exhaust. At this point, additional heat is added to the working substance by combustion at constant pressure in a second combustor. This takes advantage of the basic thermodynamic principle that the work of any expansion process increases in direct

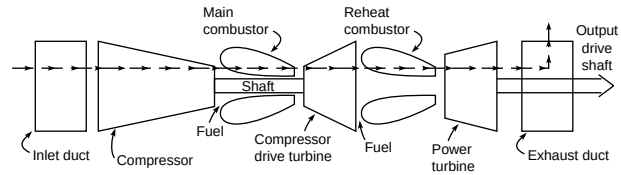


Fig. 9.7.12 Reheat Brayton cycle gas-turbine component arrangement.

proportion to the temperature entering the turbine. (This is the converse of the principle that applies to compression, as cited above for the intercooled process.) The elevated temperature entering the second turbine increases its work output; this additional work appears directly as greater power on the output shaft. Since the extra work is more than offset by the added heat energy of the second combustor, the result is also reduced cycle efficiency.

Associated with the **reheat cycle** is an increase in exhaust temperature which ordinarily represents higher exhaust waste heat. However, in the case of a **combined cycle,** the higher exhaust temperature is a source of waste heat recovery that can be used to generate steam. The higher exhaust temperature can improve cycle efficiency of the steam system to the extent that the **overall system efficiency** can be improved. Reheat thus lends itself to use with exhaust waste heat recovery systems.

Intercooled and Regenerative Cycle This cycle (Fig. 9.7.13) has both an intercooler in the middle of the compression process and a regenerator to capture exhaust waste heat. The intercooling results in a

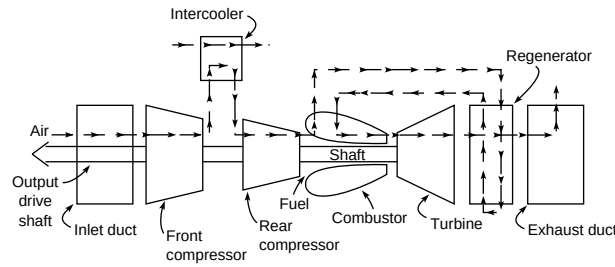


Fig. 9.7.13 Intercooled regenerative Brayton cycle gas-turbine component arrangement.

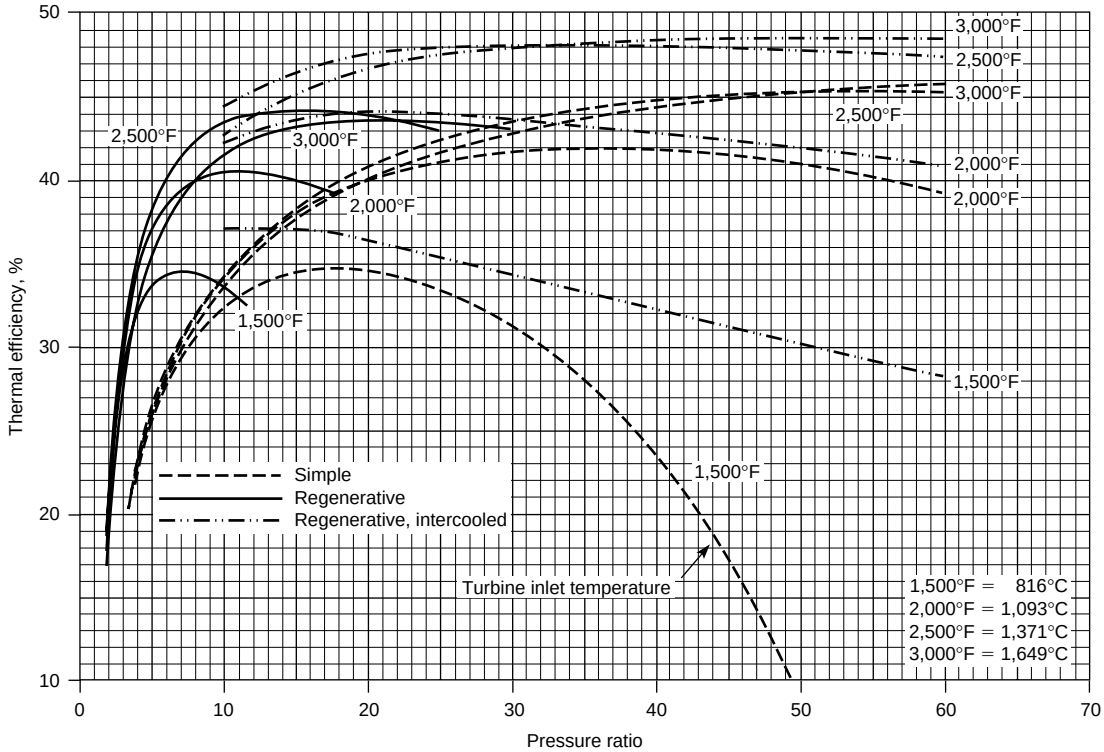


Fig. 9.7.14 Variation of Brayton cycle thermal efficiency with pressure ratio and at different turbine inlet temperatures, for regenerative cycle and for simple cycle.

lower temperature leaving the rear compressor. This circumvents the temperature difference limitation reached in regeneration. That is, at increasingly high pressure-ratio levels, the turbine exhaust temperature remains sufficiently higher than the compressor exit temperature to sustain heat exchange and thus waste heat recovery. Overall, the cycle efficiency of the intercooled regenerative cycle is superior to that of the simple Brayton cycle and all its common variants over a wide range of cycle pressure-ratio levels. Figure 9.7.14 shows variation of efficiency for an intercooled regenerative cycle compared to simple and pure regenerative cycles.

In actual practice, the choice of a simple cycle or one of its variants depends on overall economics applied to a specific use. The extra cycle benefits must trade favorably against the additional cost and complexity of the added heat exchangers or combustors and the penalties of the added ducting in between.

CONFIGURATION VARIATIONS

Since the gas turbine is made up of rotating compressors and turbines connected by driveshafts, there are various ways to arrange these elements. The simplest, Fig. 9.7.15, is the single shaft or ''spool'' arrangement. In this case, the compressor, turbine, and output driveshaft all turn at the same rotational speed.

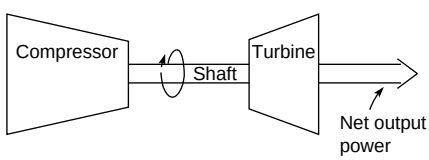


Fig. 9.7.15 Single-spool gas-turbine component arrangement.

Free Turbine (Fig. 9.7.16) To provide more flexibility by allowing the load to turn independently at a speed different from that of the compressor and its drive turbine, a **free turbine** configuration connects the compressor and its turbine on one driveshaft, while hot air continues to expand through a separate turbine which delivers net output power. The free turbine finds use either in electric generator power

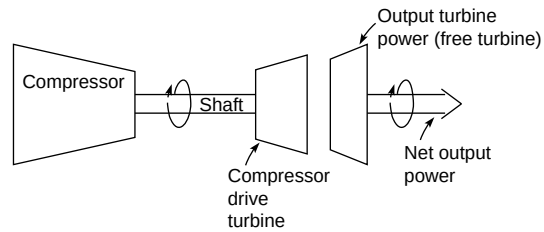


Fig. 9.7.16 Free turbine gas-turbine component arrangement.

applications, where the generator must operate at a fixed speed (either 3,600 r/min or 3,000 r/min for 60- or 50-Hz installations, respectively), or in pumping applications, where speed can vary. In either case, the compressor or partial load can operate at its most favorable speed independent of the driveshaft load.

Dual Spool (Fig. 9.7.17) At very high compressor pressure ratios, the part load operation of a compressor on a single spool tends to become unstable. When a compressor designed for high pressure ratio is forced to operate at low pressure ratio (as load is reduced), the front and rear portions of the single-spool compressor become aerodynamically unbalanced. This is alleviated by dividing the compression process into two or more spools on separate shafts driven by separate turbines. With this arrangement, the front compressor can slow down more rapidly

than the one behind, thus maintaining a more stable overall aerodynamic balance. Conventionally, to avoid unnecessary turning and ducting of the air, the spools are arranged in a straight-through alignment. This is made possible by **concentric shafting**. The shaft connecting the second compressor and its drive turbine (commonly known as the *high-pressure spool*) is made hollow to accommodate the driveshaft which connects the first or front compressor with its drive turbine at the rear (commonly known as the *low-pressure spool*).

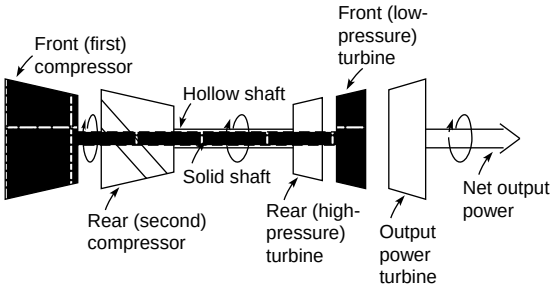


Fig. 9.7.17 Dual-spool gas-turbine component arrangement.

Closed vs. Open System In circumstances where heat addition by internal combustion is not practical, the open cycle using atmospheric air as the working substance may be replaced by another working substance continuously flowing in a **closed circuit**. An example of such a power plant would be the helium-cooled nuclear reactor. As the name implies, helium under high pressure replaces air. With the working substance completely contained, the minimum cycle pressure can be made much higher than ambient. Even such a low-molecular-weight gas as helium can be compressed into a low volume and thus keep the overall power plant relatively compact. Helium lends itself to the nuclear reactor application by virtue of its being almost completely chemically inert, which prevents radioactive transport.

External Combustion The gas turbine can maintain its continuous flow feature without internal combustion; such is the case with the direct use of coal as a fuel. Early attempts to use coal directly in internally fired gas turbines always resulted in severe and rapid damage to the turbine materials. Associated with the coal burning are not only the erosive particulates but also highly corrosive impurities such as sulfur. As an alternative, gas-turbine power plants using indirect firing of coal are being considered. Coal, on a Btu/lb basis, is much cheaper than any other available fossil fuel. Conversely, indirect heat addition through a heat exchanger not only adds expense to the power plant but also wastes energy because of heat exchanger effectiveness limitations. The overall economics of a specific application will dictate power plant choice.

WASTE HEAT RECOVERY SYSTEMS

A characteristic of gas-turbine engines is the incentive to operate at as high a ''firing'' (turbine inlet) temperature as the prevailing technology will allow. This incentive comes from the direct benefit to both specific output power and cycle efficiency, as illustrated in Figs. 9.7.4 and 9.7.5. Associated with the high maximum temperature is a high exhaust temperature which, if not utilized, represents waste heat dissipated to the atmosphere. Schemes to capture this high-temperature waste heat are prevalent in industrial applications of the gas turbine.

Combined Cycle This term commonly refers to the scheme that places a boiler and superheater directly behind the gas turbine to generate steam and produce additional power in a steam turbine (Rankine cycle). A simplified version of a combined cycle is shown in Fig. 9.7.18. Figure 9.7.19 illustrates the combined-cycle process on a temperature vs. entropy diagram. The steam portion is sometimes called the **bottoming** cycle. Early combined cycles had **fired** as well as **unfired** versions. The supplementary firing was required because of the high sensitivity of the Rankine cycle to maximum superheat temperature, as

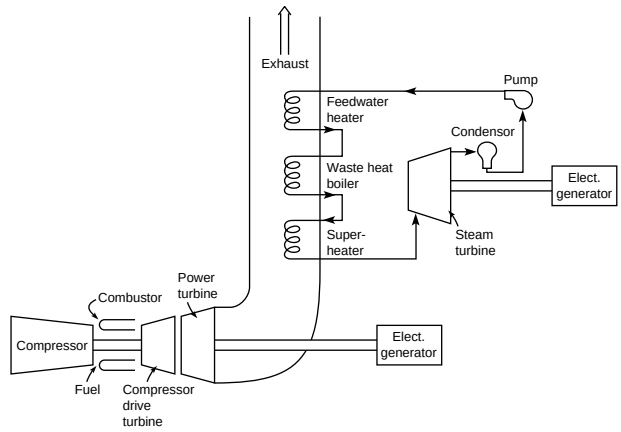


Fig. 9.7.18 Schematic of combined-cycle heat recovery system.

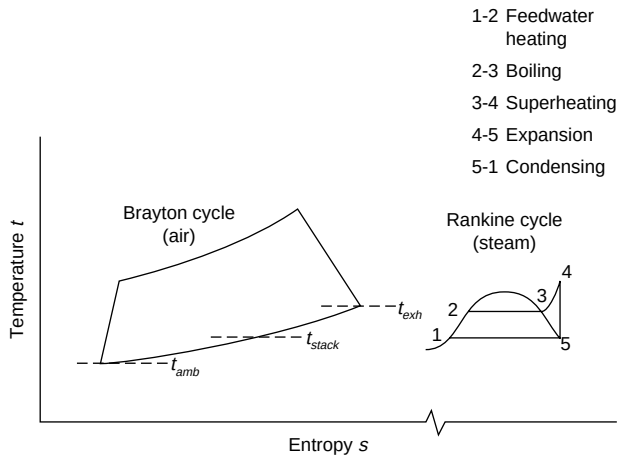


Fig. 9.7.19 Combined-cycle temperature vs. entropy diagram.

shown in Fig. 9.7.20. When the gas-turbine exhaust temperature is too low to sustain a good steam cycle efficiency, the extra combustion in the gas-turbine exhaust could result in an overall system efficiency improvement. Modern gas turbines, with their increasingly high turbine inlet and exhaust temperatures, can approach or exceed 60 percent in **overall combined-cycle plant efficiency** without supplementary firing.

If η_{gt} is the gas-turbine (Brayton) cycle efficiency and η_{st} is the steam

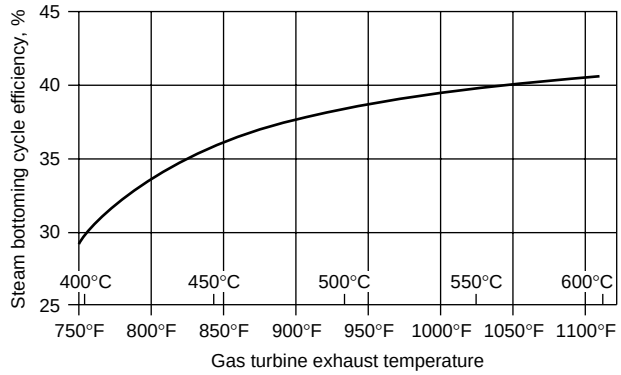


Fig. 9.7.20 Variation of typical steam system heat recovery with turbine exhaust temperature.

bottoming cycle (Rankine) efficiency, then the overall combined-cycle plant efficiency η_{cc} is:

$$\eta_{cc} = \eta_{gt} + (1 - \eta_{gt})\ \eta_{st} \left(\frac{t_{exh} - t_{stack}}{t_{exh} - t_{amb}} \right) \qquad (9.7.2)$$

where t_{stack} is the minimum practical exhaust temperature to be released into the atmosphere after heat recovery, t_{exh} is the temperature of the gas turbine exhaust gases leaving the turbine, and t_{amb} is the outside ambient temperature. (Typically, stack exhaust temperatures are limited to 200 to 300°F (93 to 150°C) to avoid condensation of the potentially corrosive combustion products.)

Steam Injection (Fig. 9.7.21) As a variation of the conventional combined cycle described above, some gas-turbine installations generate steam by exhaust waste heat recovery and, instead of using that steam in a separate steam turbine, inject the steam directly back into the turbine section of the gas turbine. In this system, the turbine acts not just on air (plus combustion products) as the working substance, but instead on a mixture of air products and steam. Such a scheme eliminates the need for additional equipment and is thus less expensive than the normal combined cycle. This system has a drawback in that direct steam injection consumes water at the plant site and thus raises environmental concerns.

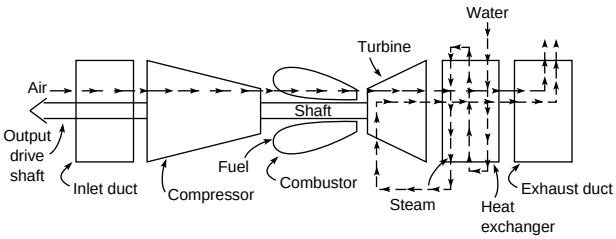


Fig. 9.7.21 Arrangement of components for gas turbine with steam injection.

Cogeneration There are many power plants used to provide the total energy needs of industries which use process heat. In these instances the exhaust heat of the gas turbine provides a source of high energy that can be used directly in the plant process rather than for producing more power. This is particularly true in the chemical industry, where many operational **total energy plants** can be found. In addition, some of or all the exhaust waste heat can be used for **district heating** where, especially in colder climates, steam or warm water generated from the gas-turbine exhaust waste heat can be circulated around the community to heat homes and buildings. Use of gas-turbine exhaust for heating is especially suited to provide the total energy needs of moderate-size to large institutions such as schools, hospitals, or commercial complexes like shopping malls.

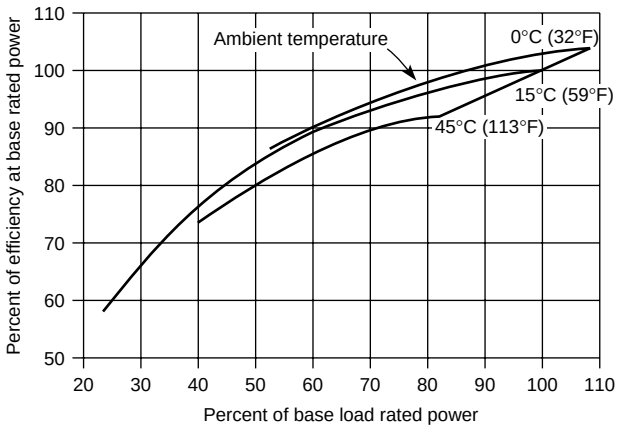


Fig. 9.7.22 Typical partial-load performance characteristics for fixed output speed.

OPERATING CHARACTERISTICS

Partial Load Gas turbines respond rapidly to throttle variations. They are capable of accelerating from idle to full power in minutes or, in the case of the lightweight flight engine designs, seconds. They can be started in a matter of minutes and, via automatic controls, can operate for long periods of time unattended. Partial-load fuel efficiency remains high. Typical partial-load performance characteristics are shown in Fig. 9.7.22 for fixed output rotating speed (r/min) and in Fig. 9.7.23 for a variable-speed (free-turbine) drive.

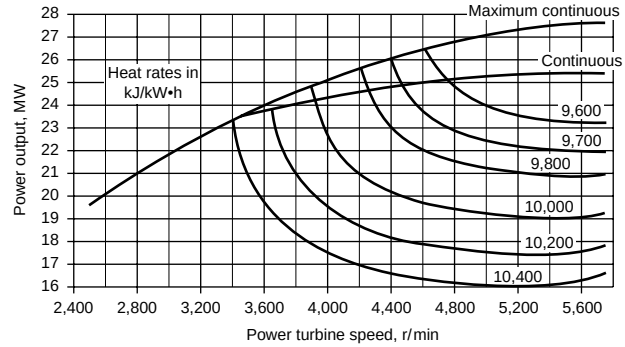


Fig. 9.7.23 Typical partial-load performance characteristics for variable output speed in a free-turbine drive.

Ambient Conditions Gas-turbine operation is sensitive to ambient conditions. Being directly dependent on mass flow for a fixed cycle, power output varies directly with air density and thus ambient pressure. Variations in ambient temperature affect the thermodynamic cycle and engine operating characteristics. Normally, any particular gas-turbine model will have an associated **power rating curve** which varies as shown in Fig. 9.7.24.

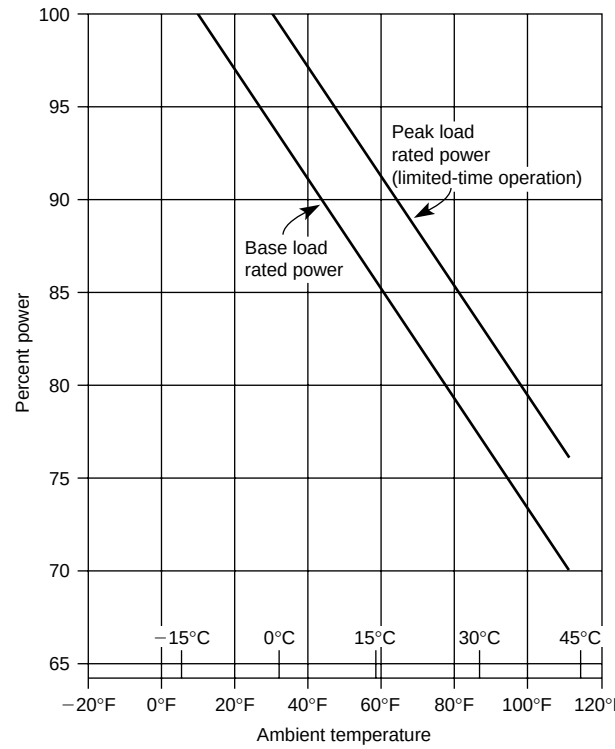


Fig. 9.7.24 Typical variation of output power with ambient temperature.

GAS-TURBINE COMPONENTS

The primary components of a gas turbine consist of a **compressor,** a **combustor** and a **turbine**. Other elements of a gas turbine, depending on the installation, are the electronic control, inlet to the compressor, exhaust nozzle, power turbine, regenerators, and intercoolers. The component that poses the greatest aerodynamic and thermodynamic design challenge, as well as having the greatest influence on engine performance, is the compressor.

Compressors Compressors can be centrifugal flow, axial flow, or a combination centrifugal and axial flow. In a **centrifugal flow compressor,** the airflow enters at the hub and is then turned from axial to radial by the compressor rotor. Early gas turbines and some modern small engines employ centrifugal compressors. In an **axial flow compressor,** air flows in an axial direction through a series of rotating blades and stationary vanes which are concentric with the axis of rotation. All large gas turbines employ multistage axial flow compressors because of their higher efficiency and capacity. Figure 9.7.25 illustrates the rotor, stators, and assembly of a multistage axial flow compressor. To achieve high pressure ratios and maintain stability at all operating conditions, **dual-spool** or three-spool arrangements are employed, which permit each spool to operate at optimum speed. Interstage bleeds, discharge bleeds, and variable-angle vanes are also employed to avoid surge. Compression ratios in any one spool can exceed 20 : 1 with spools of 10 or more stages, and polytropic efficiency can be as high as 92 percent.

Combustors A number of arrangements are used for the **combustion chamber** including side-by-side can annular combustors or a single annular combustion chamber (Fig. 9.7.26). Fuel is introduced through a manifold and an arrangement of multiple fuel nozzles. The **combustor** must maintain high combustion efficiency, and low levels of pollutant emissions and smoke, have a minimum pressure loss, maintain stable combustion over a wide range of operating conditions, and produce a uniform temperature in the hot gases distributed to the turbine. The combustor must be designed to utilize some of the airflow to atomize the fuel to achieve efficient combustion, generate turbulent mixing needed to produce a uniform exit temperature, and cool the metal parts of the combustor for durability at high operating temperature. Burner efficiency usually exceeds 99 percent except at low power in the idle region.

Turbines The **turbine** consists of one or more stages of blades and interstage guide vanes located immediately to the rear of the combustor section. Figure 9.7.27 illustrates the rotors or wheels, sets of stationary

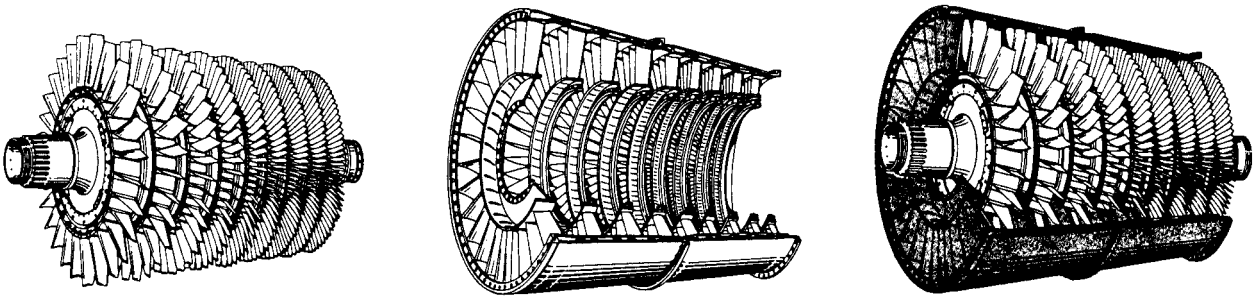
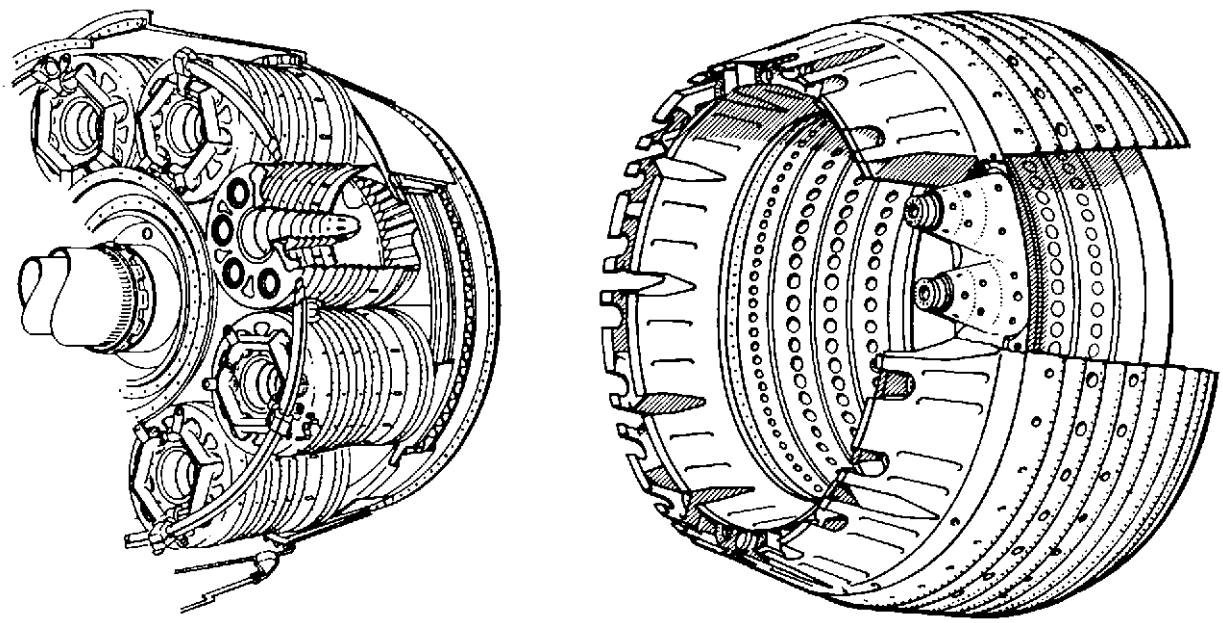


Fig. 9.7.25 Components and assembly of axial flow compressor.



Can annular combustion chamber

Annular combustion chamber

Fig. 9.7.26 Combustion chamber (combustor) configurations.

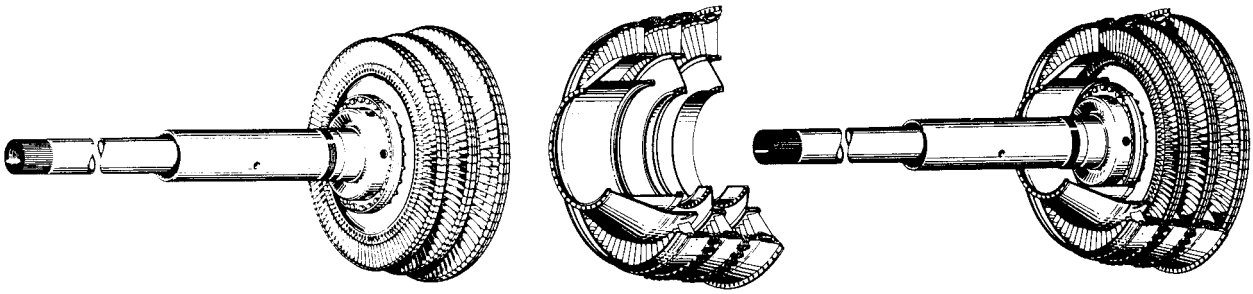


Fig. 9.7.27 Turbine elements and assembly.

vanes or nozzles, and their assembly. The turbine extracts kinetic energy from the heated gas stream and converts it to shaft power to drive the compressors and accessories. In the case of a turboprop, turbofan, or turboshaft, power is also extracted to drive a propeller, a fan, or an output shaft (see Sec. 11.5). The arrangement of the turbine is similar to that of the compressor with two major differences. First, since the expansion process is aerodynamically easier than the compression process, turbines have fewer stages than compressors. Second, the gases are much hotter and, therefore, require materials and cooling schemes to maintain the life of the turbine parts. High-temperature alloys, internal cooling passages (for air usually supplied from the compressor), and ceramic thermal barrier coatings are used in the turbine. The normal range of turbine efficiency is 90 to 95 percent.

APPLICATIONS

A worldwide directory of currently available **gas-turbine aircraft engines** published in *Flight International* (June 8–14, 1994) lists 37 manufacturers and 166 models. A much larger number of companies are manufacturers or suppliers of parts or components to the original equipment manufacturers. More than 700 model designations are now or have been in service.

For **industrial gas turbines** above 15,000 hp (11,185 kW), there are 28 gas-turbine models designed, manufactured, and currently being marketed by nine original equipment manufacturers (OEMs). In addition, there are many other companies that, under license from these OEMs, either manufacture and package or only package these gas-turbine engines. For industrial gas turbines under 15,000 hp (11,185 kW), there are 73 gas-turbine models designed and made by 17 companies; again, there are many other licensee associates. Aeroderivative gas turbines account for almost two-thirds of the gas turbines installed worldwide to date. There are 4 manufacturers of 10 aeroderivative engine models over 15,000 hp (11,185 kW) currently available in the marketplace.

Military Aviation **Military aircraft,** including fixed-wing combat and transport as well as helicopters, are almost exclusively gas-turbine-powered. Fighter aircraft are powered with afterburning turbofan engines. Modern heavy-lift cargo and troop transports are powered by high-bypass-ratio turbofans often derived from commercial engines. Many older cargo, troop, tanker, and utility aircraft still in service are powered by turboprop engines. Most helicopters are powered by turboshaft engines. New developments in military engine technology aim at increasing overall performance by increasing the power of the core engine components through use of higher-temperature materials and turbine technology. The military strives to achieve a balanced tradeoff of performance, weight, and stealth against reliability, maintainability, and cost.

Commercial Aviation **Commercial aircraft** are primarily powered by turbofan and turboprop engines. Design of large commercial wide-body transports tends toward long-range twin-engine turbofan power, but three-and four-engine aircraft are still being manufactured. With twin-engine aircraft approaching the size and range of four-engine aircraft, thrust requirements are ever increasing. Turbofan engines in the 60,000- to 90,000-lb (267- to 400-kN) thrust range have been certified, and versions to 100,000 lb (444.8 kN) are planned. Medium-size commercial transports are also turbofan-powered, with engines in the 20,000- to 40,000-lb (89- to 178-kN) thrust range. Small regional or commuter aircraft are primarily powered by turboprop engines from 500 to 3,000 hp (372 to 2,237 kW). Commercial engine technology is aimed at higher bypass ratios for increased thrust and reduced emissions and noise; higher component efficiency for better fuel consumption; and increased reliability. Future developments include advanced ducted propulsors (ADPs) which couple a gear-driven prop fan to the gas turbine, geared fan engines, and engines for the second generation of supersonic aircraft.

Electric Power Of the three generic industrial gas-turbine application markets, i.e., electric, mechanical, and marine (ship and boat), electric power generation comprises roughly 90 percent of the total power added to generating capacity. Gas turbines, including the combined-cycle steam turbine, account for about 40 percent of the world's new installations for electric power generation of all types. Provided appropriate fuels are economically available, this level is expected to increase to over 50 percent. Historically, gas turbines were used primarily for peak duty. Because of marked increases in ease of operation, reliability, thermal efficiency (approaching 60+ percent) as well as the availability of relatively inexpensive natural gas, gas turbines are largely utilized in the combined-cycle configuration for base-load duty. As their fossil fuel steam boiler counterparts historically did in growing to over 1 GW in size, gas turbines are likewise pursuing economy-of-scale cost ($/kW) advantages by growing to very large sizes. In the 1970s, a 75-MW combined-cycle system was a fairly large unit; by the early 1990s, systems of over 400 MW are common.

Marine **Aeroderivative gas turbines**, in sizes up to 35,000 hp (26,100 kW), provide power for the bulk of the surface warships (e.g., cruisers, destroyers, frigates, corvettes) recently added to the world's fleets. Two new market opportunities for marine aeroderivative gas turbines are emerging. The first is the growing market for commercial lightweight, very fast (40+ kn), novel-type (catamaran, surface effect ship, slim monohull, etc.) ferries and cargo ships. The second will be created when the rigid air quality standards that exist for aircraft and land-based installations in developed countries are extended to the marine industry. The world's cargo and cruise ships are primarily diesel-powered due to their ability to utilize poor-quality, less expensive fuels. However, resulting effluent levels are high, especially regarding sulfur dioxide and nitrogen oxides (NO_x). Although sulfur in the fuel can be eliminated at the refinery, NO_x is primarily engine-related. From a diesel NO_x emissions are 600 to 1,700 ppmv (parts per million of exhaust volume), while a gas turbine emits 200 to 220 ppmv. Even lower emissions, down to 42 ppmv, utilizing distillate fuel can be achieved with gas turbines equipped with lean-burn, dry-low NO_x burners.

Mechanical Drive The increased popularity of low-polluting natural gas as a preferred fuel stimulates the market for gas turbines in mechanical drive applications. Mechanical drive gas turbines, up to 40,000 hp

(29,800 kW), are largely aeroderivative and are found in such applications as powering natural gas pipeline compressors, providing gas-lift compressor power for enhanced oil well recovery in offshore platforms and land-based installations, and powering compressors for natural gas gathering, liquefied natural gas production, and various processing plants.

Small Engines Gas turbines in smaller sizes [less than 1,000 hp (746 kW)] are used in several fields of application. Such engines usually feature centrifugal compressors and are all manufactured by specialized companies. In the ground transportation field, automobile and truck manufacturers have experimented since 1950 with gas turbines, primarily of the low-pressure-ratio (less than 10 : 1) regenerative type. While several prototypes have been tried over the years, displacement of the widespread, well entrenched automobile piston engine has not taken place, with one exception: The U.S. Army has found the gas turbine to be well suited as a tank engine. As with the automobile engine, gas turbines have been experimentally tried in railroad trains, but have yet to make inroads to replace the diesel engine.

Small gas turbines are used as a compact, simple, and reliable power source on board aircraft for start-ups and auxiliary power. They are known as auxiliary power units or APUs. Similar engines find use as quick-start standby units to drive generators for emergency power. Finally, very small jet engines provide propulsion for remotely controlled aircraft with surveillance or cruise missile military missions.

9.8 NUCLEAR POWER

by Louis H. Roddis, Jr., Daniel J. Garner, John E. Gray, Edwin E. Kintner, Nunzio J. Palladino, supplemented by George Sege and Paul E. Norian of the NRC

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NOTE: The material related to nuclear power plant safety and licensing was revised and amended by personnel of the Nuclear Regulatory Commission (NRC) on official time. That material, therefore, is in the public domain and not covered by copyright.

FISSION AND FUSION ENERGY

Nuclear power is the energy derived from the fission (or splitting) of the nuclei of heavy elements, such as uranium or thorium, or from the fusion (or combining) of the nuclei of light elements, such as deuterium or tritium. Particles set in motion by these processes yield heat energy virtually instantaneously. The amount of energy released per atom exceeds by a factor of several million the amount of energy obtainable per atom in a chemical reaction, such as the burning of fossil fuels. While control of the fusion process is still surrounded by tremendous technical problems, considerable success has been achieved in utilizing heat energy produced by fission for power generation, propulsion, industrial production, and scientific experiments.

NUCLEAR PHYSICS

A nuclear reaction occurs when the particles making up the nucleus of an atom are rearranged—as distinct from a chemical reaction, which is caused by changes in the electron structure surrounding the nucleus.

Fundamental (elementary) particles are, theoretically, the irreducible constituents of the material world. More than 20 particles were regarded as elementary in 1972, the exact number being dependent on how they are classified. New experimental techniques lead to discovery of more particles; advances in theory indicate that some are compounds of particles. For example, atoms and atomic nuclei were at one time believed to be noncomposite. Except for the electron and proton, all the fundamental particles are unstable.

An **electron** carries one unit of negatively charged electricity [defined as 1.6×10^{-19} coulomb (C)] and has a mass at rest equal to 1/1,836 times the mass of the hydrogen atom. Electrons emitted by radioactive atoms are called **beta** (β) **rays** and have energies up to several MeV (million electron volts).

A **proton** carries one unit of positively charged electricity equal in magnitude to that of the electron and has a mass, at rest, of 1.0072764 atomic mass units [u when 1 u (1.66057×10^{-27} kg) is defined as $\frac{1}{12}$ the mass of a neutral atom of the most abundant isotope of carbon]. It is identical to the nucleus of the hydrogen atom. Protons are not emitted spontaneously from atomic nuclei.

A **neutron** has a mass approximately equal to that of the proton but lacks an electric charge. It cannot be detected by subjecting it to electric or magnetic fields, nor can its presence be shown by its passage through cloud chambers. The presence of neutrons can be shown only by their interaction with other particles.

An **atom,** which is the basic unit of any chemical element, consists of a central nucleus surrounded by planetary electrons. The number of its electrons determines its chemical characteristics and in electrically neutral atoms equals the positive charge on its nucleus. The nuclei of all atoms except the hydrogen atom are composed of protons and neutrons. The nucleus of the hydrogen atom consists of a single proton, which gives it an atomic number of 1. This number is designated by the letter *Z*. The number of neutrons in the nucleus is represented by the letter *N*. From this it follows that the mass number *A*, which represents the atomic weight, is equal to $N + Z$. In 1977, 105 elements were known. All with atomic numbers above 92 (uranium) must be produced artificially and have a short half-life.

Isotopes are elements which have identical chemical characteristics but different atomic weights, i.e., certain atoms of the same element have different numbers of neutrons in their nuclei. There are considerably more than a thousand isotopes of the known elements. Only 320 of these exist in nature, of which approximately 40 are unstable and radioactive.

For example, hydrogen has three isotopes. One is ordinary hydrogen with a proton nucleus. Another is deuterium, whose nucleus consists of a proton and a neutron. The third is tritium, which consists of one proton and two neutrons. At the heavy end of the natural periodic table is uranium, with 92 protons. When it has 146 neutrons, it is $_{92}U^{238}$, which comprises 99.28 percent of natural uranium. The remainder of natural uranium consists of 0.006 percent U^{234} and 0.71 percent U^{235}. The latter isotope, with 92 protons and 143 neutrons, is the only one of the three which is readily fissionable and is the one utilized in most of the nuclear power reactors operating in the United States. It is separated from the natural element in the enrichment desired by a gaseous diffusion process in government facilities.

Energy and mass are used interchangeably in nuclear physics, and the total energy of a nucleus can be determined by measurement of its exact

mass m, which is related to its energy E by the Einstein equation $E = mc^2$, where c is the velocity of light, 3×10^{10} cm/s. This law of equivalence of mass and energy unifies two long-accepted laws: the conservation of energy and the conservation of matter. Singly, the two older laws must be regarded as high-order approximations, adequate in all engineering fields except atomic energy. The exact mass m of a nucleus differs by only a few hundredths of a percent from an integral number A of proton masses, but this small deviation is of significance in nuclear theory, since it measures the difference in energy between the nucleus and its separate components, i.e., its binding energy. The energies of nuclides are recorded in the tables in terms of their masses.

Physical atomic weights do not quite equal chemical atomic weights because the oxygen used in the chemical determinations is a mixture of isotopes, whereas in mass spectrographic measurements the single isotope C^{12} is used for reference. The relationship is: Physical atomic weight = 1.000280 × chemical atomic weight.

Planck's Constant Since radiation has the properties of both waves and particles, it is theorized that in the emission or absorption of energy by atoms or molecules, the process takes place by steps, each step being the emission or absorption of an **individual quantity of electromagnetic energy,** which is considered the elementary quantum of action. It is known as **Planck's constant** and has the value $h = 6.624 \times 10^{-27}$ erg·s. It is used in virtually all quantum relationships, including Schrödinger's equation and Heisenberg's uncertainty principle, and in the formula for the energy levels of atomic hydrogen. The standard notion is $\hbar = h/2\pi$, where $\hbar$ signifies the angular momentum of an orbiting electron.

Relativistic Mass When moving particles approach the speed of light, the approximation of kinetic energy, $\Sigma\frac{1}{2}m_0v^2$, fails; it then becomes necessary to use the equation $E = mc^2$ and to use the relativistic mass, a function that increases with velocity according to the law $m - m_0/\sqrt{1 - (v/c)^2}$, where m_0 is the rest mass (i.e., the mass of newtonian mechanics) and v is the coordinate velocity of the particle.

Binding energy is the work required to disintegrate an atom completely into Z protons and $A - Z$ neutrons. It is a measure of the total kinetic and potential energy of the nucleons in the nucleus and may be calculated from the equations

$$B = C^2\Delta \qquad \text{and} \qquad \Delta = Zm_{\mathrm{H}} + (A - Z)\, m_n - m$$

where m is the mass of the nuclide, m_{H} the mass of the hydrogen atom, and m_n the mass of the neutron, all on the physical atomic-mass scale (u). The quantity of Δ is known as the **mass defect**. The energy involved is usually stated in MeV.

Fusion Figure 9.8.1 shows that if two deuterium (H^2) nuclei react to produce one He^3 nucleus plus one neutron, the total binding energy of the four particles (two neutrons plus two protons) involved in the reaction is increased from approximately 4.4 to 7.6 MeV, releasing 3.2 MeV for the reaction. Such a process is called **fusion**. The deuterium-deuterium fusion reaction may also produce an H^3 plus an H^1 nucleus, with a release of 4.0 MeV.

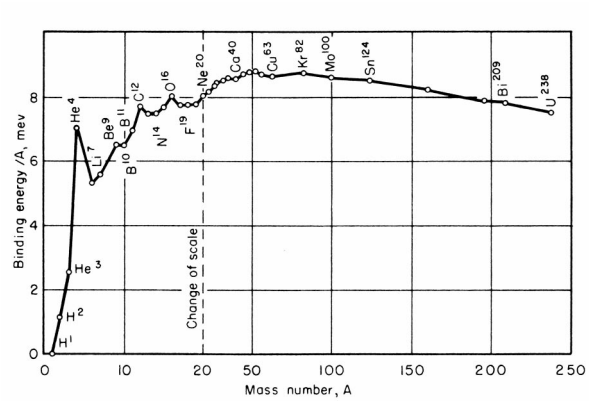


Fig. 9.8.1 Binding energy per nuclear particle for stable nuclei.

For fusion to occur, the two nuclei must approach each other with exceptionally high kinetic energy in order to overcome their electrostatic repulsion; and since kinetic energy is proportional to absolute temperature, effective fusion can occur only at extremely high temperatures. For fusion to continue as a chain reaction, it is further necessary that nuclear collisions be sufficiently frequent to maintain the high temperature in spite of heat radiation. The condition necessary for frequent collisions is high pressure.

A light-element chain reaction is the principal source of heat in the sun, hydrogen being converted to helium as the net result of a multistage process. The temperature and pressure at which this reaction is sustained at the center of the sun are 35 million °F and 1.5×10^{11} lb/in^2 respectively.

A deuterium-deuterium reaction can be sustained at temperatures and pressures which are considerably lower than those in the solar reactions but still fantastically high by terrestrial standards. Achievement of a controlled light-element chain reaction for power generation by fusion depends on producing high particle velocity and high density locally by electromagnetic means, without development of correspondingly high temperatures and pressures at the container walls.

Fission is the division of an atomic nucleus into parts of comparable mass, either naturally (spontaneously) or under bombardment (induced) with neutrons, α particles, γ rays, deuterons, or protons. In elements with $Z > 90$, fission can be produced by neutrons of low or moderate energy.

Fission is important because in the process neutrons are emitted that may, under certain conditions, be utilized to produce further fissions, thus leading to a self-sustaining, or ''chain,'' reaction. It is thus possible to ''burn'' uranium to obtain an energy release per atom approximately 10^8 greater than from a chemical fuel. The energy release in the fission of U^{235} is

	MeV
Kinetic energy of fission products	168
Kinetic energy of fission neutrons	5
Energy of γ rays	10
Energy of β rays	5
Energy of neutrons	11
	199

The total of 199 MeV is equivalent to 3.2×10^{-11} W·s, which indicates that it requires the fissioning of 1.12×10^{17} atoms to release a kilowatthour of energy. This number of atoms forms only a speck of matter six thousands of an inch in diameter.

Radioactivity occurs when an unstable nucleus undergoes atomic disintegration by emitting α, β, or $\beta+$ particles, γ or X-ray electromagnetic radiation.

The **alpha** (α) **particle** is identical with the nucleus of a helium atom. The emission of an α particle creates a new nucleus, with Z reduced by 2 and A decreased by 4 mass units.

Beta (β) **decay** involves emission of an electron from the nucleus of an atom, thereby increasing Z by unity.

The **gamma** (γ) **ray** is electromagnetic radiation originating in the nucleus—as differentiated from X-rays, which usually are less energetic and arise from energy adjustments as electrons move between orbital shells outside the nucleus. Gamma rays are emitted instantaneously when a neutron is captured in a nucleus and are frequently emitted following ejection of a particle from a radioactive nucleus.

The **positron** ($\beta+$) is a particle whose mass is the same as that of an electron and whose charge is equal in magnitude but opposite in sign. Positron emission decreases Z by unity.

Electron capture (EC) results when an unstable atom decays by capturing an orbital electron in the nucleus, resulting also in a decrease of Z by unity. This capture produces a vacancy in the orbital shell which is filled by an electron moving from an outer shell, giving the rise to X-ray characteristics.

Half-life is the time required for half the original nuclei in a sample of an isotope to decay; it is given as $0.693/\lambda$ s. Half-lives of the radioactive nuclides vary from 10^{-7} s to 10^{10} years.

UTILIZATION OF FISSION ENERGY

Nonpower Types

Nuclear reactors having a purpose other than the generation of usable power may be classified into two general types: research and test reactors and weapons material production reactors.

The latter category of **production reactors** includes both light- and heavy-water and gas-cooled reactors of high thermal power (comparable in thermal output to large power reactors). Their fuel cycle is adjusted to produce plutonium and tritium of weapons quality for nuclear weapons programs. Most production reactors operate with outlet temperatures near or below the boiling point of water, so the thermal heat is wasted to the environment. A few **''dual-purpose''** production reactors operate at temperatures which allow the coupling of useful electric power output facilities.

Teaching reactors have the characteristics of very little excess reactivity, very low power capability (a few to a few thousand watts of heat energy), and relatively easy access to the core. Teaching reactors are used mainly for the purpose of instruction in the behavior of a nuclear reactor. With a teaching reactor novice reactor operators may experience the response of reactor instrumentation as reactivity is inserted and criticality is approached, ''just critical'' behavior, and power increases or decreases.

Teaching reactors are generally owned and operated by universities. In addition to being used for operator training, they are used in a variety of nuclear-engineering instructional experiments. Typical experiments include measurements of neutron-flux distribution; measurement of neutronic characteristics such as generation time, leakage fraction, thermal-utilization factor, resonance escape probability, and other characteristics affecting criticality and kinetics; and measurement of parameters affecting control such as control-rod worth and reactivity feedback coefficients. Although the neutron flux available from a teaching reactor is relatively small, it is nevertheless significant and can be used for limited production of radioactive isotopes.

Research reactors generally have a power capability of 10 thermal MW approximately. They are for the purpose of providing an intense radiation source, both neutrons and gamma rays. A research reactor is usually designed to have a high leakage flux in order to facilitate its use as a radiation source.

The neutrons from a research reactor have a variety of uses including neutron activation analysis, neutron radiography, neutron diffractometry, and production of radioactive isotopes. The gamma rays are applied in materials testing and in studies using the interaction of gamma rays with the nuclei of various elements.

In **neutron activation analysis,** an unknown sample is irradiated in a calibrated neutron flux, and the constituents of the sample become radioactive through neutron absorption. The radiation from the sample is then analyzed. Since the radiation from a specific radioactive isotope is unique to that isotope, a qualitative and quantitative determination of the constituents of the sample is possible. Extremely small quantities of radioactivity can be detected; therefore, neutron activation analysis is a very sensitive analytical tool.

Neutron radiography is a technique similar to X-radiography. The perturbation of a collimated beam of neutrons that has passed through the radiographed item is determined by film exposure or other means. Unlike x-rays that serve to outline dense, highly absorptive materials, neutron radiography serves to outline light materials that preferentially scatter neutrons from the incident beam.

Neutron diffractometry is a technique used to determine the molecular and crystalline structure of materials. The analysis is accomplished through a precise determination of the diffraction pattern from a collimated monoenergetic beam of neutrons that has interacted with the sample.

Certain radioactive isotopes are useful and effective in the **treatment of some diseases**. Isotopes that do not occur naturally can be produced in large quantities by exposure of the precursor element to the neutron fluxes available from research or test reactors.

Test reactors generally have a power capability ranging from 25 to 100 thermal MW. They are used largely for the purpose of providing an experimental radiation environment similar to that which exists in the core of a power reactor. They are also used for large scale radioactive isotope production.

Power Cycles

Although the energy of fission appears as kinetic energy (85 percent) and photon energy of γ rays, there is no practical method of converting this energy directly into useful work on a large scale. Therefore, a nuclear reactor must be treated as a heat source which differs from a chemical heat source in that no oxygen is required and the heat does not have to be removed from gaseous combustion products which possess poor heat-transfer properties.

In the large power capacities associated with nuclear power plants, the **Rankine vapor cycle** is the one in use. The Rankine cycle appears to be best suited for nuclear power because (1) the maximum practicable operating temperature for a reactor corresponds to or is lower than the temperature used in the conventional Rankine vapor cycle; (2) the problem of containment of radioactivity is better solved by this cycle than by other cycles; (3) reactors are most economical in large sizes, as are Rankine engines.

Three variations of the Rankine cycle (Fig. 9.8.2) are being used. The **direct cycle** is the least complicated and thermodynamically the most desirable. Its disadvantages are the facts that the boiling process does not permit the high power densities which are attainable with liquid cooling and that radioactive steam is carried into the turbine and condenser. The **indirect cycle** gives greater power density and eliminates radioactivity in the turbine, although it too produces steam at lower

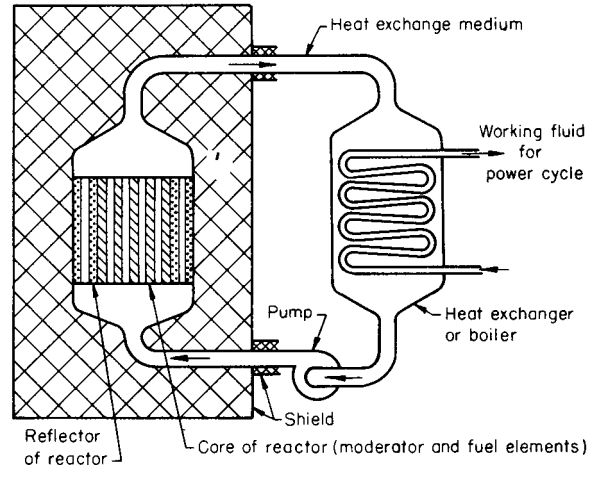


Fig. 9.8.2 Plant cycles—coupling with a working fluid. *(Amorosi, Selection of Reactors, ''Nuclear Engineering Handbook,'' McGraw-Hill, 1957.)*

pressures and temperatures than those considered most efficient for modern turbines. Both concepts are highly developed and commercially available. Plants utilizing sodium or other liquid metal as the heat-exchange medium use an intermediate-link system to isolate the water system from radioactive sodium, but the ability to operate at high temperature with liquid-metal coolants permits the generation of high-temperature steam to utilize the present state of steam turbine technology.

The only other cycle which has been given consideration is the **Brayton cycle**. This cycle is being examined with renewed interest because of the rapid advances in gas-cooled reactor technology and the availability of large gas turbines suitable for reliable utility industry service. The use of an open cycle using air has been discarded because of the spread of activation products from the turbine exhaust and the corrosion problems in the reactor core and fuel elements. The use of a closed Brayton cycle with the latest high-temperature gas-cooled reactors (HTGR) has potential advantages in that the gas temperatures can match the latest gas-turbine designs and a very compact plant can be built.

The ability of a nuclear reactor to produce high temperatures not limited by chemical reaction equilibrium has generated interest in advanced concepts such as magnetohydrodynamics and thermionic generators. These systems are limited by the availability of suitable materials of construction and appear destined to remain in the research phase until suitable materials are developed.

Power Reactor Types

During the early years in the development of power reactors, a large number of possible reactor designs were investigated. In the present state of the industry, the designs which have gained the greatest acceptance all are solid fuel in the form of uranium dioxide. The moderator may be either graphite or water and the coolant may be water, gas, or liquid sodium. Experimental power reactors using a solution of uranium compounds in water or molten salt have been built but have not assumed industrial importance because of many unsolved corrosion problems. Light-water (moderated and cooled) reactors (LWRs), comprised of the boiling water reactor (BWR) and pressurized water reactor (PWR), represent about 99 percent of the operating reactors in the United States. A high-temperature gas-cooled reactor (HTGR) is in operation in a small plant. The liquid-metal-cooled fast breeder reactor (LMFBR) concept is employed in two operating test reactors in the United States, but has been successfully implemented in full-scale operating plants in France and Russia. For the efficient utilization of the energy represented by the world's uranium reserves, a successful breeder reactor is desirable since the present generation of reactors are all consumers of only the naturally occurring fissionable isotope of uranium, U^{235}.

For utilization of natural uranium as a fuel, D_2O is substituted for H_2O as the coolant moderator. The reactor must be refueled more frequently than enriched reactors to maintain an adequate operating reactivity margin. However, D_2O reactors such as the Canadian designs (CANDU) utilize on-line refueling to minimize shutdown time. The success of this type of reactor depends on the availability and cost of D_2O.

Boiling water reactors have a simpler design and can utilize relatively thin-walled vessels and pipes because they operate at moderate pressures compared with pressurized-water reactors. Fuel cladding temperatures are only slightly higher than steam temperatures, and there is an inherent safety factor because the steam-void volume increases on a transient power increase. Some of the problems caused by radioactivity carryover, such as maintenance on the turbine and condenser and the prevention of radioactive leakage, are offset by the cost of the boiler in pressurized water reactors. However, boiling water reactor vessels, despite their lower design pressure, are larger and heavier than pressurized water vessels of similar rating and extensive water purification equipment is required to remove corrosion products and other impurities from the feedwater before introduction into the reactor.

Limitations on the **power density** imposed by exit voids (i.e., the vapor volume of the exiting steam-water mixture from the reactor core) are a disadvantage in that low-power density contributes to high fuel inventory charges. Load changes are accomplished by steam bypass control or rods since adjustment of the turbine throttle and consequent reduction in steam flow will cause an increase in pressure in the reactor, which in turn will collapse the steam bubbles and increase reactivity. The control system functions to maintain constant reactor pressure, and reactivity control is achieved in part by varying the recirculation rate in the reactor. The net decomposition of the water into O and H is much greater than in a pressurized water reactor.

Pressurized Water Reactors Properties of water and steam which led to their predominance as general-purpose heat-transfer media have also caused their widespread application as a reactor coolant. A major disadvantage of water results from its relatively high vapor pressure. However, this can be partially overcome by allowing boiling in the reactor. Thermal efficiencies up to 36 percent are possible.

The use of H_2O as a coolant and moderator is based on well-developed technology which indicates that the ultimate size (or capacity) will be dictated primarily by heat transfer requirements. The average heat flux q/A is around 200,000 Btu/(h · ft²) (630,000 W/m²), with a maximum of 600,000 Btu/(h · ft²) approximately (1,890,000 W/m²).

The **light-water** (H_2O) **reactors** require slightly enriched fuel. However, the cost of enrichment is economically justified because the increased power density reduces inventory charges. To provide sufficient neutron moderation, the H_2O/U volume ratio is kept slightly above 2 : 1.

The use of oxide fuel minimizes corrosion. Design problems are reasonably well understood, although costly structural provisions must be made to load and unload fuel because of the high pressures. Fuel enrichment usually runs between 1.5 and 4.5 percent, depending on the alloy used for corrosion resistance.

Control rods, burnable neutron absorbers, and soluble boron in the coolant are used to control excess reactivity, achieve high fuel burnup, and maintain power distribution within the design limits.

Gas-cooled reactors offer low fuel and operating costs because of their ability to utilize natural uranium as fuel. This advantage is offset by higher capital costs caused by the large system size, in turn resulting from the lower power density usually used. More advanced designs such as the high-temperature gas-cooled reactor and its variants can use low enriched uranium and are able to use higher temperatures and the required structural material for such temperatures. These newer gas-cooled designs are much more compact than the natural-uranium reactors which they are beginning to supplant.

Carbon dioxide is the coolant usually used in natural uranium reactors, as it is nontoxic, only mildly radioactive, nonflammable, noncontaminating in case of leakage, and relatively inexpensive. Helium gives indication of being an ideal coolant, but its limited availability and high cost make it attractive in a power reactor only where the inventory can be kept small and its superior performance at high temperatures can be utilized. Helium, being inert, does not chemically react with graphite and the reactor structure at high temperatures as does carbon dioxide.

The latest gas-cooled reactor designs are prestressed-concrete vessels which contain the core, gas circulators, and steam generators. The inclusion of as much equipment and piping as possible inside the vessel structure greatly simplifies the control of gas leakage.

Fast breeder reactors operate at extremely high power densities (as they are unmoderated) and are liquid-metal-cooled. Their attractiveness stems from an ability to ''breed,'' i.e., produce more fuel than they consume. Aside from savings in fuel cost, this characteristic is believed necessary to conserve the world's supply of energy.

Two conflicting definitions of a breeder reactor are in common use: (1) a nuclear reactor that produces more fissionable material than it consumes, regardless of type of fuel used; (2) a nuclear reactor that produces the same species of material as it consumes, regardless of the net gain or loss. The first definition has wider acceptance in the United States.

The selection of a coolant for a breeder reactor is restricted by the requirement that it not act as a moderator. This requirement rules out the use of water, although the attainment of some breeding in a water-cooled reactor has been investigated. The choice of coolant is limited to helium or liquid metal.

Choice of a liquid-metal coolant is based mainly on nuclear properties and on the engineering difficulties associated with melting temperatures and corrosion effects. Those of prime interest are sodium, sodium-potassium alloy, bismuth, lead, and lead-bismuth alloy. Separated Li^7 would be very attractive were it not for its high cost. Sodium has been selected as the preferred coolant because of its availability, good nuclear properties, and the short half-life of its induced radioactivity. However, in the presence of oxygen, it burns and can react violently with water.

Advantages of operating in the fast neutron spectrum include (1) structural material for the reactor core can be selected without consideration of neutron absorption; (2) very little excess reactivity is needed to compensate for fission product buildup; and (3) reprocessed fuel in which fission product removal may not be complete can be used.

The disadvantages are the increased damage to structural materials by the high fast flux and the control problems resulting from the short neutron lifetime.

Fuel and Waste Cycle

The steps and processes in a typical fuel cycle for light-water reactors are described below. Figure 9.8.3 illustrates the relationship of these steps and processes.

Natural Uranium Uranium occurs naturally in ores in very low concentrations, less than ½ percent by weight. In certain very unusual cases, the uranium content in the ore, called ore grade, may be as high as 10 percent, but these occurrences are extremely rare.

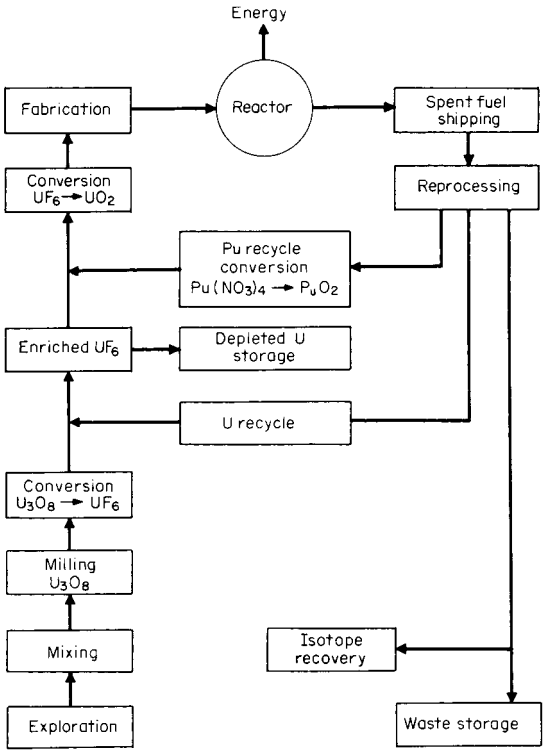


Fig. 9.8.3 Nuclear fuel cycle.

Uranium ores are mined by conventional techniques and then crushed and ground in a mill, which has special equipment for the recovery and purification of the ore into uranium concentrates. The product is in the form of a powder called yellow cake, which has the chemical form of either ammonium or sodium diuranate, both of which are yellow, hence the name.

Conversion of U_3O_8 to UF_6 The process of enriching the U^{235} isotope (from 0.711 percent to the 2 or 3 percent required by water reactors) requires that the uranium be in gaseous form. The gas UF_6 is used; so the natural U_3O_8 must be converted to natural UF_6.

Enrichment alters the relative amounts of U^{235} and U^{238} in uranium, raising the U^{235} concentration from 0.711 percent to the percentage required for use in a reactor. By so doing, the input (feed) stream of natural UF_6 is broken into two output (product plus tails) streams, one enriched and one depleted.

The process of enrichment makes use of the mass difference between U^{235} and U^{238} isotopes. In a gaseous form, the average energies of all UF_6 molecules are equal whether the U atom is U^{235} or U^{238}. Since the energies are equal, the average velocity of the lighter $U^{235}F_6$ molecule is greater than that of the $U^{238}F_6$. Because of this greater velocity and, hence, a greater momentum, the $U^{235}F_6$ molecules can diffuse through a barrier more easily. Gas diffusion plants work on this principle.

In the **gas centrifuge** process, gaseous UF_6 is fed into centrifuges which rotate at very high speed. The heavier U^{238}-bearing molecules drift to the outside wall, while those containing the U^{235} atoms accumulate close to the axis of the machine. The separation process is enhanced by longitudinal gas circulation along the wall and near the axis, in currents counter to each other. This also allows end scoops to collect the enriched and depleted streams. Centrifuges are connected in cascade to obtain specific enrichment assay levels and the cascades are operated in parallel to obtain adequate amounts of enriched product. A plant will contain several thousand stages. Electric power demands for the centrifuge enrichment process are typically 5 to 10 percent those of the diffusion process for similar enrichment and amount. The term **separative work unit (SWU)** is used to express the quantity. Centrifuge plants are in use in European countries, but the United States continues to use the more energy-intensive diffusion process, after abandoning a major centrifuge plant half-built.

There are a number of other possible separation processes for uranium (or for that matter, other isotopes of any material). These include various thermal separation processes and plasma processes, activated either by electronic or laser energy, which work on atomic or molecular ions with separation by magnetic field. The lighter isotope will have a slightly smaller radius of curvature and can be caught in a pocket separate from the heavier. Several of these processes have been proven at the experimental level, and some have been used to produce significant amounts of material in the past. The United States, when it abandoned the centrifuge plant, authorized further work on the **atomic vapor laser isotope separation (AVLIS)** process to carry it to full-scale development. This process utilizes vaporized uranium metal, the U^{235} and U^{238} atoms of which are selectively energized to different ionized states by laser energy, and then separated in a magnetic field, all at high temperature. This is potentially even more energy-efficient than the centrifuge process.

Fabrication The gaseous enriched UF_6 that comes out of the enrichment plant is delivered in standard cylindrical gas bottles. The UF_6 is then converted to UO_2, the fuel form is which it is used. The UO_2 is in the form of a powder which is then milled to provide suitable sintering properties. The powder is formed into pellets by cold pressing, sintered, and ground to final dimensions. The pellets are loaded into a zirconium alloy tube (cladding) which is backfilled with helium and sealed at each end with a welded-end plug. The fuel rods are assembled into complete fuel assemblies. The number of fuel rods in an assembly varies from 50 to 200 or more, depending on the reactor type and specific design.

Spent Fuel Storage and Transportation Several options are available for managing spent fuel after its discharge from the reactor. It may be stored at the reactor site for a minimum time of 3 months or longer and then shipped in a shielded cask to a reprocessing plant for recovery of residual fuel valves. This procedure is used in most power reactors, except in the United States. Alternatively, it may be stored at reactor water pools, or in an away-from-reactor storage facility for an extended period of time, and then shipped to an interim dry storage site, possibly after fuel mechanical compaction, or sent to a geological repository for long-term disposal. This latter procedure is mandated by current U.S. law.

Reprocessing and Reconversion Spent nuclear fuel has substantial residual value in its uranium and plutonium, and possibly in other fission products or transuranic elements. The plutonium and other transuranics are the longer-lived radioactive components. The spent fuel is mechanically chopped and dissolved in acid, and the solution processed through subsequent steps to purify the plutonium, uranium, and any other desired product. The remaining fission wastes remain in the high-level waste streams. The plutonium product is usually in the form of a nitrate, and the uranium, either directly or in a subsequent process as UF_6.

Uranium and Plutonium Recycle As described above, the components of value in the spent fuel are the remaining uranium, now of lower enrichment, and the plutonium generated through neutron capture in U^{238}. The uranium can either be reenriched or blended with new enriched uranium and used to make new fuel elements. The plutonium can also be used either alone or mixed with uranium in making new fuel elements. A breeder reactor, by definition, will reuse the plutonium as fuel. Water reactors are nearly all designed to reuse fuel, but this is

being done only outside the United States. By law the United States has mandated the use of a once-through cycle, without reprocessing the spent fuel. The economics of the two approaches depend on the relative cost of newly mined uranium.

Waste Disposal Disposal of **low-level radioactive wastes,** from power plants or other radioactive material uses, such as medical is usually done by dry burial in a dedicated site.

High-level waste in the form of solid spent fuel containing transuranics as well as high-level fission products, or reprocessing plant high-level waste converted to a solid form as a glass or ceramic, can be disposed of in a mined geologic repository to ensure long-term isolation of the waste from the environment. Current U.S. policy calls for acceptance of commercial high-level waste by the Department of Energy for disposal in mined repositories beginning in 1998, in accord with the Nuclear Waste Policy Act of 1982. At this time (1996), it seems likely that this goal will remain unmet; rather, it is expected that such repositories will not be ready to accommodate these wastes until sometime in the early twenty-first century. The electric power producing utilities pay a fee on the basis of kilowatthours generated for this service. Isolation will be by the use of both engineered (waste package and container) and geologic (geologic media, such as salt, basalt, or granite) barriers. Other countries, after separating the uranium and plutonium, generally plan to store the waste for extended periods of time before final disposal. Key technical areas in implementation include the development of highly stable materials for encapsulating the waste (e.g., glass, ceramic, corrosion-resistant metals), methodologies for predicting repository performance over extended time periods, and repository closure techniques. Demonstration of the ability to safely dispose of waste has become critical to public acceptance of nuclear power in many countries.

PROPERTIES OF MATERIALS

(See also Table 9.8.1.)

Materials used in nuclear reactors can be divided into six basic categories: fuel, radiation, control, coolant, structural, and shielding. While the nature of the materials may be different, they must be physically and chemically stable and compatible with their environment for a sufficient period to enable the reactor to operate predictably, stably, and economically under the particular set of parameters imposed by the design. (See also Sec. 6.)

In addition to the environmental conditions to which materials may ordinarily be exposed, reactor materials must also sustain the extraordinary bombardment of X-rays (photons) and other atomic particles which are products of nuclear fission. In their passage through matter, these photons and atomic particles may inflict severe damage, since the energy of these particles is spent mainly in ejecting or pulling electrons from their orbits, i.e., producing **ionization**. Gamma photons and neutrons are highly penetrating, whereas fission fragments have a path under 0.001 in the solids and liquids (but cause intense ionization over that path).

The **behavior of materials** following breakage of a bond by ionization may differ markedly. The fragments of simple, highly stable molecules such as water (H_2O) or carbon dioxide (CO_2) will most probably recombine. However, it is highly probable that the parts of complex **organic compounds** such as lubricants and electrical insulation will suffer permanent fragmentation and/or recombination in a different manner to produce new molecules.

Metals, on the other hand, are essentially unaffected by ionization since the probability of displacement of excited atoms in a metal lattice is small and the atoms will merely revert to their original status on ridding themselves of their excess energy. The atoms of **nonconducting crystals** and glasses are similarly immobile, and most of their properties are insensitive to the ionization. However, since these crystals are nonconducting, some of the electrons displaced by ionization become trapped in lattice vacancies, producing changes of optical properties (such as darkening of glass by gamma irradiation) and change of electrical resistivity.

On direct collision, **fast neutrons** may displace atoms from a chemical compound without affording opportunity for immediate recombination. Similarly, **energetic neutrons** may displace atoms from their normal lattice positions in a metal or other crystalline material and push them into abnormal interstitial positions in the lattice, generally producing an effect on metals similar to work hardening. The hardness and tensile strength of the metal will increase, but the metal becomes more brittle. **Thermal neutrons** affect the properties of metals only in those cases where the probability of absorption is high enough to cause significant transmutation to other elements.

Most severe damage occurs when the atoms of a fuel material fission and produce two or more new atoms from each of the original fissioned atoms. Volumetric changes occur by the substitution of these additional atoms in the lattice for the original atom, and the lattice is further damaged by the intrusion of the high-velocity fission fragments.

Fuel Materials

Fuel materials are generally divided into two categories: fissionable and fertile. The **fissionable materials** are those isotopes of uranium and plutonium which fission upon interaction with thermal neutrons. These are the isotopes U^{233} U^{235}, Pu^{239}, and Pu^{241}, which have odd atomic weights. The **fertile materials** are those isotopes of uranium and plutonium which have even atomic weights. These absorb neutrons under proper conditions and undergo a series of nuclear reactions, with the eventual formation of fissionable isotopes. Thus U^{238} forms Pu^{239}, and Pu^{240} forms Pu^{241}. In addition, thorium^{232} forms the fissionable isotope U^{233}.

Both fissionable and fertile materials fall short of possessing the **properties considered ideal** for a nuclear fuel, namely, corrosion resistance, good thermal conductivity, strength, and ductility. To overcome these problems as well as to prevent fission products from entering the coolant, various techniques are used, including plating, cladding, and alloying.

Materials for this use must be corrosion-resistant, be compatible with the fuel materials, be physically stable under irradiation, have good heat transfer characteristics, have good nuclear properties, and be reasonably fabricable. From a consideration of cross section alone, only aluminum, beryllium, magnesium, and zirconium are attractive for use in thermal reactors. Magnesium usually is ruled out on the basis of strength and corrosion resistance but has been widely used in some gas-cooled systems. In fast reactors, a number of other materials become attractive, the most important being stainless steel, which also is used under special conditions in thermal reactors. Consideration also is being given to ceramic materials, BeO, Be_2C, SiC, and $MoSi_2$, which are very attractive in all phases except resistance to thermal shock.

Moderating material must be capable of reducing neutron energy very rapidly and should have good strength at high temperatures, corrosion resistance, thermal and radiation stability, and reasonable cost. Such properties are not available in a single material. Graphite is the most widely used solid moderator. Attention also is being given to beryllium and its compounds, deuterium, oxygen, and hydrogen.

The nuclear and physical properties considered desirable for **fuel diluents and cladding** are, in general, desirable for structural materials. For the latter, however, more emphasis is placed on strength and corrosion resistance than on nuclear properties. Stainless steel, aluminum, zirconium, molybdenum, titanium, niobium, and their alloys are used. As operating temperatures continue to rise, ceramics and cermets will have to be developed for this purpose.

Materials used for **control purposes** must have a high cross section for absorption of neutrons, adequate strength, low mass to permit movement, good corrosion resistance, chemical and dimensional stability under heat and irradiation, and low cost. Boron, cadmium, and hafnium are given the most consideration in various metallic and ceramic forms and as compounds dissolved in the coolant.

Coolant Properties

The choice of a coolant also is a compromise, since no known material possesses all the **desirable characteristics**. These are good heat transfer coefficient and good heat capacity on a volume basis, low absorption

Table 9.8.1 General Properties of Reactor Materials before Irradiation*†

Materials	Density, g/cm^3	Melting point, °C	Specific heat cal/(mol·°C)	Thermal conductivity† cal/(s·cm·°C)	Coefficient of thermal expansion, 10^{-6} per °C	Ultimate tensile strength† 10^3 lb/in^2	Modulus of elasticity† 10^4 lb/in^2	Capture cross sections, 0.025 eV (σ_e barns)
Fuels and fertile materials								
Uranium, normal	19.3	1,133 ± 1	6.649	100°C, 0.063	25–125°C, 45.8	56	24	7.68
Thorium, normal	11.71	1,690 ± 10	100°C, 6.59	100°C, 0.090	30–100°C, 11.5	30.6	10.6	7.56
Plutonium^{239}	19.60	623 ± 7		100°C, 0.061		125	13–16	1.026
Uranium oxide (UO_2)	10.96	2,500–2,600	15.38	100°C, 0.022	22–363°C, 9.3–10.8		21	24.3
Al-16 weight % U alloy		~650		200°C, 0.42	20–100°C, 20	17.7	10.6	
Zr-4 weight % U alloy	6.72	1,840 ± 25		100°C, 0.033	190–300°C, 6.66	63	14.4	
Diluents and cladding materials								
Zirconium	6.50	1,845 ± 25	6.31	50°C, 0.05	5.82	30–38	13.8	0.18
Zircaloy 2	6.55	1,820 ± 25		< 0.04	20–100°C, 5.2 ± 0.6	65	14	
Aluminum (28)	2.694	600.2	100 C, 6.07	0.53	20–100°C, 23.8	13	10.3	0.26
Magnesium	1.74	650	25 C, 6.08	18°C, 0.376	49°C, 26	Annealed, 27	Annealed, 6.5	0.069
$MoSi_2$	6.24	1,870		370°C, 0.088	0–1,500°C, 5.1	27°C, 22		
Solid Moderators								
Graphite	2.27	Sublimes at 3,650 ± 25°C	2.066	100°C, 0.37–0.48	20–250°C, 1.9–4.0	0.500–2.4	0.5–1.2	0.0032
Be_2C	6.24	1,870		30°C, 0.02	25–200°C, 7.7		400°C, 23	
Beryllium	1.847	1,315	100°C, 13.69	100°C, 0.34	25–100°C, 11.54	29	44	0.01
Sintered BeO†	2.2–2.8	2,550 ± 25	100°C, 7.7	200°C, 0.19	25–100°C, 5.5	400°C, 15	400°C, 39	0.00072
SiC	3.21	Decomposes at at 2,500°C	327°C, 10.06	400°C, 0.060	0–1,700°C, 4.4	1,700°C, 50		
Structural materials								
Stainless steel (347)	8.027	1,427	100°C, 0.12‡	100°C, 0.037	20–100°C, 16.5	85	29	3
Inconel X	8.51	1,395–1,425	25–100°C, 0.109‡	0–100°C, 0.036	20–100°C, 11.5	Annealed, 100–120		4.1
Molybdenum	10.2	2,622 ± 10	100°C, 6.24	0°C, 0.32	5.1	67	48	2.7
Titanium (commercial purity)	4.507	1,690	0–500°C, 6.61	25°C, 0.41	25°C, 8.5	85–100	15.5	5.8
Niobium	8.57	2,415	0°C, 6.01			1,200°C, 14.7		1.1
Control materials								
Cadmium	8.694	321	25–32°C, 6.19	0.22	20–100°C, 31.8	10.3	7.1–10	2.450
Hafnium	13.36	2,130 ± 15	25–2,227°C, 6.16		0–100°C, 5.9	67.5	14	105

* All properties change with irradiation at room temperature except when specified.
† Thermal conductivity, strength, and the modulus of elasticity vary directly with the density; however, the coefficient of thermal expansion appears unaffected.
‡ Cal/(g · °C).
SOURCES: H. A. Seller, Properties of Reactor Materials, ''Nuclear Engineering Handbook,'' McGraw-Hill, 1958; ''Reactor Physics Constants,'' ANL 5800, 1963.

Table 9.8.2 Properties of Nuclear Coolants*†

	Aqueous and organic liquids				
	Water	Heavy water	Diphenyl	Dowtherm-A	Santowax R
Density, lb/ft³	42.4 (600°F)	46.2 (600°F)	46.4 (675°F)	46.6 (675°F)	52.29 (675°F)
Viscosity, lb/(h · ft)	0.20 (600°F)	0.220 (600°F)	0.353 (675°F)	0.7986 (675°F)	0.630 (675°F)
Melting point, °F	32	39	156	53.2	293
Boiling point, °F	212	214	49	496	687–784
Heat capacity, Btu/(lb · °F)	1.54 (600°F)	1.68 (600°F)	0.637 (675°F)	0.675 (675°F)	0.539 (675°F)
Thermal conductivity,	0.294 (600°F)	0.294 (600°F)	0.0763 (675°F)	0.098 (675°F)	0.064 (675°F)
Capture cross section σ_a, 0.025 eV (barns)	0.66	0.0011	3.3	3.3	4.6

	Gases						
	Hydrogen	Helium	CO_2	Air	Nitrogen	Steam	Neon
Density, lb/ft³	0.00185	0.00375	0.041	0.0275	0.026	0.0167	0.0190
Viscosity, lb/(h · ft)	0.042	0.094	0.081	0.089	0.086	0.069	0.1445
Heat capacity, Btu/(lb · °F)	3.625	1.245	0.283	0.263	0.268	0.518	0.246
Thermal conductivity, Btu/(h · ft · °F)	0.224	0.159	0.0328	0.0337	0.0327	0.0341	0.0536
Prandtl no.	0.660	0.735	0.6988	0.6898	0.7097	1.048	0.6632

	Liquid metals							
	Gallium	Lithium	Potassium	Rubidium	Sodium	Tin	Na (56%) K (44%)	Pb (44.5%) Bi (55.5%)
Density, lb/ft³	359	29.9	44.6	84.4	51.2	421	48.8	625.5
Viscosity, lb/(h · ft)	1.836	1.188	0.3816	0.415	0.5040	2.736	0.435	2.88
Melting point, °F	85.86	354	147	102	208	449	66.2	257
Boiling point, °F	3,601	2,403	1,400	1,270	1,621	4,118	1,518	3,038
Heat capacity, Btu/(lb · ft · °F)	0.082	1.0	0.180	0.0877	0.3005	0.0639	0.2484	0.035
Thermal conductivity, Btu/(h · ft · °F)	18.0	18.0	21.15	13.2	37.7	19.0	16.35	8.05
Capture cross section σ_a, 0.025 eV (barns)	2.77	71.0†	1.97	0.70	0.505	0.60		0.094

* At 1,000°F unless otherwise specified. Adapted from L. Green, Reactor Coolant Properties, *Nucleonics*, **19,** no. 11, which contains a detailed bibliography.
† Thermodynamic properties of a variety of other specific materials are listed also in Secs. 4.1, 4.2, and 6.1.

cross section, low vapor pressure at the operating temperature, minimum of chemical reaction with the material contacted by the coolant, good resistance to forming γ emitters with long half-lives, stability under irradiation and high temperatures, and workable melting points.

Coolants are divided into three broad **classes:** aqueous and organic liquids, liquid metals, and gases. **Water** has found the widest application, largely because it is readily available and economical, has fairly good heat transfer properties, and offers no corrosion problems which cannot be handled. Its most serious drawback is its high vapor pressure. **Heavy water** has superior qualifications except for cost, which is high. Diphenyl and other **organic liquids** have demonstrated an ability to stand up under irradiation but are handicapped by a tendency to deposit decomposition products on the fuel element plates. See Table 9.8.2.

Liquid metals are attractive because they provide high heat-transfer rates and do not require pressurizing to operate at high temperatures. Sodium is most attractive because of its high boiling point and high heat conductivity, but it reacts violently with water and has poor lubricating qualities. Lithium 7 better meets qualifications but is harder to obtain and is costly.

Gas coolants offset their advantage of being inert at elevated temperatures with poor heat transfer qualities. They must be operated at high pressures to be useful in power reactors. Sodium hydroxide and yellow phosphorus are among other materials considered as coolants, but their acceptability has not been demonstrated.

FISSION REACTOR DESIGN

Neutron Balance The main objective of reactor design is to achieve neutron balance during reactor operation. At steady state, the number of neutrons produced during fission is equal to the neutrons lost by absorption and leakage processes. The net rate of change of neutron density n (neutrons per cubic centimeter) is given by the neutron conservation equation:

$$\frac{\partial n}{\partial t} = \text{production} - \text{absorption} - \text{leakage}$$

If $\partial n/\partial t$ is positive, the reactor is said to be supercritical; if it is zero, the reactor is at steady state and is said to be critical; if it is negative, the reactor is subcritical.

Cross Section The microscopic cross section σ for a particular nuclear process is the effective target area of the nucleus with which a neutron must interact to produce the given reaction. The unit of σ is barn (1 barn $= 10^{-24}$ cm²). Neutron cross sections (absorption, scattering, and fission) constitute the basic nuclear data for reactor design. Absorption cross sections for thermal neutrons range from 0.00046 barn for deuterium to 3.3×10^6 barns for xenon^{135}. Moderating atoms for the slowing down of fast neutrons have low atomic mass, small absorption, and large scattering cross sections. Examples are hydrogen, deuterium beryllium, and carbon. Table 9.8.3 gives thermal neutron cross sections in barns for selected elements (based upon the naturally occurring combinations of isotopes).

Multiplication Factor The number of neutrons produced in any one generation in a reactor for each neutron produced in a previous generation is called the multiplication factor k. If $k = 1$, the reactor is critical; if $k < 1$, it is subcritical; and if $k > 1$, it is supercritical. For an infinite thermal reactor, multiplication factor $k_\infty = \eta f p \varepsilon$. η is the number of fission neutrons produced per neutron absorption in fuel, f is the thermal utilization factor (neutron absorption rate in fuel/total absorption rate), ε is the fast-fission factor, and p is the resonance capture escape probability. The leakage of neutrons from the reactor core is accounted for by multiplying k_∞ by the factor $1/(1 + M^2B^2)$, where M^2 is the migration area (sum of the squares of the diffusion length L and the slowing-down

Table 9.8.3 Absorption (σ_α) and Scattering (σ_s) Cross Section for Thermal Neutrons, in barns

Element	σ_α	σ_s	Element	σ_α	σ_s	Element	σ_α	σ_s
Aluminum	0.23	1.4	Helium	0.007	0.8	Potassium	1.97	1.5
Beryllium	0.01	7	Hydrogen	0.33	38	Plutonium^{239}	1.026	9.6
Bismuth	0.032	9	Iron	2.53	11	Sodium	0.505	4.0
Boron	755	4	Lead	0.17	11	Thorium	7.56	12.6
Cadmium	2,450	7	Magnesium	0.063	3.6	Tin	0.6	4
Carbon	0.0032	4.8	Molybdenum	2.7	7	Titanium	5.6	4
Chromium	2.9	3.0	Nickel	4.6	17.5	Uranium	7.68	8.3
Copper	3.7	7.2	Niobium	1.1	5	Zirconium	0.18	8
Hafnium	105	8	Oxygen	<0.0002	4.2			

NOTE: Individual isotopes have markedly different cross section. For example, U^{235} has total $\sigma_a = 682$ barns and fission cross section $\sigma_f = 580$ barns.

length τ) and B^2 is the geometric buckling ($B^2 = 0$ for an infinite reactor).

Breeding Ratio If the ratio of a number of fissile nuclides produced by the capture of neutrons by the fertile nuclides to the number of fissile nuclides consumed is greater than unity, it is called the **breeding ratio;** if it is less than unity, it is called the conversion ratio. For breeding, the value of η (defined above) must be greater than 2, as one neutron is required to maintain the chain reaction by fission, another neutron maintains breeding by its capture by the fertile nuclide, and the remaining to compensate for the loss of neutrons due to absorption and leakage. Neutron economy favors the breeding of Pu^{239} fissile nuclides from fast neutron capture by U^{238} fertile nuclides. This breeding process is the basis of LMFBR (liquid-metal fast breeder reactor). On the other hand, thermal breeders are based upon the breeding of uranium 233 fissile nuclides from thorium 232 fertile nuclides by the capture of a thermal neutron.

Diffusion Theory Multienergy group diffusion theory is presently the basis of reactor design. For a critical reactor the neutron balance equation reduces to

$$D_g \nabla^2 \phi_g - \sum R_g \phi_g + S_g = 0$$

where ϕ_g and S_g are the gth energy group flux and the slowing-down neutron source; D_g and R_g are the diffusion coefficient and removal cross section. The latter are called group constants, which are obtained by the analysis of the $\Sigma R_g \phi_g$ basic fuel lattice. The solution of the above equations provides the reactor multiplication factor, reactor critical size, neutron fluxes, power distributions, isotopics of elements, and fuel-cycle length.

Control Extra fuel is required for depletion to compensate fuel lost in producing energy, to account for the loss of reactivity due to the buildup of fission products, and to account for the change in reactivity due to changes in fuel and moderator temperatures. The control of excess reactivity is undertaken by the control rods of strongly absorbing materials having large absorption cross sections (such as boron, hafnium, or silver-indium-cadmium) and the burnable poison rods containing boron or gadolinium. In PWR, soluble boron is also employed. Control rods are inserted to shut down the reactor and are withdrawn to make the reactor critical.

A fraction of prompt neutrons (0.0065) are delayed neutrons. The presence of the delayed neutrons makes control of the reactor easier.

Thermal

Sources of Heat The energy produced by fissioning appears in several forms, each of which eventually degrades to heat. To provide for removal of this heat, the core, or active portion of the reactor, usually consists of uranium bearing fuel elements with passages around these elements for flow of coolant. The rate of coolant flow and the temperature of the coolant must be such as to permit removal of all the heat generated in the elements without exceeding allowable materials properties.

Hot Spot and Hot Channel Factors The problem of heat removal is complicated by the fact that heat is not generated uniformly throughout the core. The geometric configuration of most power reactors is approximately represented by a finite cylinder of height H and radius R. Diffusion theory predicts the following thermal neutron flux pattern for an unreflected, unrodded, and homogenized cylindrical reactor as a function of axial Z and radial r variables:

$$\phi(z, r) = \phi_0 \left(\cos \frac{\pi Z}{H} \right) J_0 \frac{2.405r}{R}$$

where ϕ_0 is peak flux and J_0 is the Bessel function.

In addition, the power generation is influenced by mechanical factors such as variables in fuel element dimensions, fuel concentrations, and flow. These nuclear and mechanical factors are generally called hot spot factors. Thus the peak power density of heat flux in the core is greater than the average by an amount indicated by the hot spot factor. In addition, hot channel factors are identified which indicate how much greater the temperature rise of coolant is in the hot channel than in the average channel. As an example of how such factors are used, consider a reactor in which the axial heat generation pattern is a cosine and the total flow passes uniformly once through all channels of the core; the maximum fuel element surface temperature T_{sm} depends on the hot channel and hot spot factors $F_{\Delta T}$ and F_θ and on the three temperatures—the coolant temperature at the core inlet T_1, the coolant temperature rise in the core ΔT, and the average film temperature drop in the core θ_a. (F_θ includes the axial peak-to-average of $\pi/2$.)

$$T_{sm} = T_1 + \tfrac{1}{2} F_{\Delta T} \Delta T + \tfrac{1}{2} \sqrt{F_{\Delta T}^2 \Delta T^2 + 4 F_\theta^2 \theta_a^2}$$

Design Criteria The maximum fuel element surface temperature T_{sm} is of interest because it may be limited by corrosion effects or by a desire to avoid film boiling, a condition which lowers the heat transfer from the fuel to the coolant. However, other limiting conditions also exist. In addition the internal fuel element temperature during all anticipated reactor transients is generally limited to a maximum value (generally below the melting point of the fuel) or to a maximum average value.

Coolant temperature rise generally is limited indirectly by fuel temperature limits. Occasionally, however, thermal stresses within the fuel element may impose a direct limit. In pressurized water reactors, the coolant temperature is limited not to exceed the saturation temperature of the fluid; thus it is more significant to speak of the maximum enthalpy rise, which is related directly to the steam quality of the fluid leaving the hot channel.

Density changes within the coolant may be limited to provide control stability in reactors that use boiling heat transfer.

Coolant velocity usually is limited to a maximum value by erosion or by vibration resonances, and sometimes is required to exceed a minimum value because of crud deposition. The burnout condition is determined in a given fuel element geometry for a particular combination of fuel element heat flux, primary coolant flow, and primary coolant enthalpy. Steady-state, local, or bulk boiling may have to be limited, even when burnout is not a danger, because of pitting of fuel element surfaces or crud deposition on the fuel surfaces. The burnout problem is most acute during reactor transients; hence steady state burnout generally is not limiting.

Mechanical

The mechanical design of nuclear reactors is influenced by three factors not encountered with other apparatus: the need to control criticality, the

effects of irradiation, and the high power density produced in the core.

Control of Criticality The following requirements are introduced by the need to control criticality:

1. Not only must components perform reliably, but also if they fail to function as designed, the reactor will shut down. Small parts located elsewhere in the core should be so designed that if they fail they do not interfere with the action of control rods in shutting down the reactor.
2. Emergency shutdown mechanisms must be fast-acting devices to reduce the time delay in initiating shutdown.
3. During normal operation, components of the core must be rigid enough so that deflections due to mechanical or thermal stresses do not introduce unpredicted criticality variations.

Effects of Irradiation Neutron and gamma irradiations cause the following effects, which must be considered in the design of reactor systems and reactor components:

1. Radiation effects on properties of materials—particularly on the brittle fracture characteristics of the reactor vessel, on the relaxation characteristics of bolts, and on crevice corrosion.
2. Heat generation brought about by attenuation of radiation within components materials—probably insofar as such heat influences heat removal requirements, thermal stresses, and thermal distortions of parts.
3. Induced radioactivity in corrosion and wear products that could be carried by the coolant and deposited in such regions as pumps and valves.

High Power Density A nuclear reactor requires a large amount of heat transfer surface for removal of the high power per unit volume which can be produced. Thus the reactor consists of many small, closely packed fuel elements. Mechanical tolerances are often appreciable fractions of the dimensions and spacing of these elements and therefore influence thermal performance. The influence of such tolerances on performance must be evaluated. Parts also must be rigid and vibration-free in the face of the high fluid flow rates required for power removal. The high power density may also lead to large thermal stresses in fuel elements.

Furthermore, moving parts may have to operate without benefit of organic lubricants because such lubricants tend to sludge under irradiation, and if deposited on heat surfaces could interfere with removal of heat from the core.

Residual radioactivity also influences programs and facilities for refueling of the core and for periodic maintenance of components in the reactor system. In addition, personnel exposure by the above radiation requires adequate shielding.

Shielding To protect reactor operating personnel against damaging biological effects of neutrons and gamma rays, shielding is required around a nuclear reactor. Neutron and gamma ray fluxes in the range of 10^{13} to 10^{14} must be attenuated to 10^3 particles/(cm^2·s) to meet the tolerance radiation level.

To attenuate gamma rays, which interact primarily with the orbital electrons of atoms, a material with a high atomic number containing a high density of these electrons is required. Examples are lead, tungsten, depleted uranium, or concrete containing high-Z elements in the form of scrap or heavy ore.

To attenuate neutrons, they must be slowed down and then absorbed. Hydrogenous materials such as water, concrete, or polyethylene are excellent moderators. The slowed neutrons must be absorbed without producing high-energy-capture gamma rays by using boron 10. Therefore, some gamma shielding outside the neutron shield is generally required.

Electrical

Neutron level instruments, located adjacent to but outside the core or core vessel, are used for the basic control of nuclear reactors. Thermocouples, flowmeters, pressure taps, and fission chambers are used inside the core for power calibration of nuclear instruments and for determining the distribution of power and/or temperature within the core; in turn, the power distribution is used to compute heat fluxes and fuel temperatures for comparison with limiting values.

Neutron level instruments utilize the fact that neutrons can produce ionizing particles to indicate their presence. The two most common processes used for this purpose are the (n,α) reaction with boron and the fission reaction. The resulting ion pair is used to produce ionization of a gas in a tube containing a central anode and outer electrode. The electric discharge produced by the ionization is used for measurement purposes. The detailed design features vary widely with the range of the instrument and the type of reactor in which it is to be used.

Different neutron level instruments are employed during the reactor operation at various power levels, because of the wide range of sensitivity required to monitor neutron fluxes from start-up to full-power condition. In power reactors at least three instrument ranges are used with the overlap between the adjacent ranges. These are (1) source range, (2) intermediate range, and (3) power range. During the reactor start-up (source range) the BF_3 proportional counters are used to measure neutron flux in the range of 10^{-1} to 5×10^4 neutrons/(cm^2·s). The compensated ionization chambers are employed to monitor neutron flux in the intermediate power range of 2.5×10^2 to 2.5×10^{10} neutrons/(cm^2·s). In the power range, where neutron flux ranges from 2.5×10^7 to 2.5×10^{10} neutrons/(cm^2·s), uncompensated ionization chambers are employed.

Compensated ionization chambers are designed to count neutrons against a background of gamma radiation. Uncompensated chambers are useful for providing axial power distribution.

All the above instruments are located outside the core and monitor neutron flux, hence reactor power. The latter is also obtained from the measure of ΔT (change in temperature) across the reactor core from the temperature measuring instruments located in the primary loop of the reactor. However, these instruments do not provide local flux peaking and local anomalous coolant temperature situations. In-core instrumentation must be used for this purpose.

In-core instrumentation is employed to provide information on neutron flux and fuel-assembly-coolant outlet temperature at a few selected locations in the reactor core. Thermocouples measure the coolant outlet temperature and thus provide the rise in local coolant temperature, a measure of the increase in local reactor power. Miniature, fixed, and movable fission chambers containing U_3O_8 which is 90 to 95 percent enriched U^{235} constitute neutron flux detectors. The experimental data obtained during the reactor operation from the in-core instrumentation provide the check against the reactor design parameters such as the assembly distributions and hot channel factors.

In addition to thermocouples and fission chambers, other in-core instruments such as pressure, displacement, and strain transducers are also desirable. None of these instruments except thermocouples have operated satisfactorily over a long period. The installation of in-core instrumentation is vital to the safe and economic operation of the reactor.

NUCLEAR POWER PLANT ECONOMICS

Decisions by electric utilities as to what type of generating plants to order to meet future base load requirements are based largely, though by no means wholly, on a comparison of the costs for which the alternative types can produce electricity over their lifetimes. Other factors, which have become relatively more important in recent years, include adequacy and reliability of fuel supply, including dependence on foreign sources; environmental and safety considerations; availability of suitable sites; public acceptance as influenced particularly by environmental opposition; the costs, uncertainties, and delays involved in government regulation; lead times required from initial decision to get plants on line; and plant reliability and availability.

As a practical matter in the United States, utilities seeking to add to their base load capacity have the choice only of plants utilizing nuclear fuel or one of the fossil fuels; coal, gas, or oil. Most usable hydrosites have already been developed.

The discussion of nuclear generating costs here is directed to light water reactors of either the pressurized water reactor (PWR) or boiling water reactor (BWR) types. Although other types are in use (e.g., high-temperature gas-cooled) or under development (e.g., fast breeder), the

overwhelming preponderance of reactors in commercial use are, and will for some time continue to be, light-water types.

It is customary to compare generating costs for nuclear and fossil fueled plants in three broad categories: (1) fixed costs on capital investment, (2) fuel costs, and (3) operating, maintenance, and insurance costs.

Fixed Costs on Capital Investment

In standard accounting practice, the capital investment in a nuclear power plant is considered to be the total cost of building the plant and placing it in commercial operation. Accounting classification systems employed by both NRC and FPC provide separately for ''direct'' and ''indirect'' costs. Direct costs are associated on an item-by-item basis with land and land rights, the equipment and structures which comprise the complete power plant, and coolant and moderator materials. Equipment and structures are customarily subdivided into reactor plant and equipment, turbine plant and equipment, other electrical equipment, equipment and facilities of a general nature, and transmission plant. Indirect costs consist mainly of expenses which apply to all portions of the physical plant, including engineering and design costs, construction facilities, taxes, interest during construction, and the owner's administrative and overhead costs.

The total of these items tends to be higher for nuclear than for fossil fueled plants largely because of (1) the need for containment shells, shielding, instrumentation, and other measures to contain radioactivity and ensure safety; (2) the greater complexity, hence greater costs, of reactor equipment compared to conventional boiler equipment; and (3) the lower turbine steam temperatures and pressures, and the lower thermal efficiencies of nuclear plants compared to conventional plants, requiring the use of larger, hence costlier, turbines. The discrepancy is wider in warm climates, which permit fossil fuel plants to utilize outdoor (unenclosed) construction.

Mitigating factors which prevent the construction cost discrepancy between nuclear and coal plants, in particular, from being still larger are (1) the lower cost of fuel handling equipment in nuclear plants and (2) the need in coal plants but not in nuclear plants, for smoke control and ash removal equipment.

Unit construction costs (expressed in dollars per kilowatt) for both nuclear and fossil fueled plants are lower at any given time as plants increase in size because of various economies of scale. The decreases are sharper in the case of nuclear plants, largely because of the prominence of certain minimum and relatively fixed costs to ensure safety. The existence of these inescapable costs places small nuclear plants at a particularly great economic disadvantage.

Adding a nuclear unit to a generating station which already has one or more nuclear units is less costly than constructing a first nuclear unit of otherwise identical characteristics. The principal savings in multiple-unit plants result from use of a developed site; reduced engineering and design efforts; and use of joint facilities and equipment such as cooling water intake and discharge facilities, control rooms, warehouses, shops, offices, roads, fuel handling and storage facilities, and temporary construction facilities.

The unit capital costs of nuclear plants often can be reduced substantially over their lifetimes by using improved fuel cores to achieve outputs higher than the original rating, which is usually conservative. To take advantage of this opportunity, allowance for higher output must be made at the outset in the capacity of turbine generators. Conversely, should it be necessary to derate plants to take account of safety uncertainties such as the effectiveness of emergency core-cooling systems, the effect would be to increase unit capital costs.

Total construction costs of nuclear plants have risen rapidly since about 1963, when it was believed that a large light-water plant could be built for $125 per kilowatt or less. Plants now are being estimated to cost in excess of $2,000 per kilowatt. The principal factors making for such increases in cost have been inflation, stretch-outs of plant schedules (adding to interest during construction), lower labor productivity, increases in the scope of projects to add safety and environmental features, and more stringent quality control.

Some of these factors may continue to push unit construction costs upward in the future. Counterbalancing factors tending toward a moderation in unit costs include continued increases in plant size, the likelihood of greater standardization of systems and components, a larger sales volume over which to spread manufacturing overhead costs, and improved field assembly and construction methods. One should take note of the fact that the historical tendency in the industry has been to underestimate construction costs by a wide margin.

The fixed cost component of generating costs (mills per kilowatthour) can be obtained from construction costs (dollars per kilowatt) by applying an annual fixed charge rate and an annual plant capacity factor.

Annual fixed charges cover all costs that are in direct proportion to the initial capital investment and are independent of the extent to which the facilities represented by this investment are used. A simple procedure for evaluation of a plant, which approximates the levelized annual charges obtained through utility accounting procedures, is to use constant annual fixed charges, i.e., to apply a constant fixed charge rate each year to the initial investment.

The fixed charge rate for a utility will vary because of a number of factors including amount of state and local taxes, plant life and accounting method for depreciation purposes, and debt-equity composition of the utility's financial structure. A representative range of fixed charge rates for depreciating capital of an investor-owned utility illustrate the components and relative importance of these items, it being understood that the actual components and range will vary, depending on the time of implementation and the then-current financial climate.

Component	*Fixed charge range, %*
Return on investment	7.0–9.0
Depreciation	1.0–1.3
Interim replacements of short-lived assets	0.3–0.5
Property insurance	0.3–0.4
Federal income taxes	2.0–2.8
State and local taxes	2.9–3.5
	13.5–17.5

A lower fixed charge rate would be applied to the nondepreciating capital (land and land rights and working capital) because depreciation, interim replacements, and property insurance are eliminated and associated adjustments are made in taxes. Not included in annual fixed charges are relatively constant fixed costs, such as nuclear liability insurance or plant staffing expense, which are only indirectly related to plant investment.

The annual capacity factor is calculated by dividing the kilowatthours generated by the plant in a year by the kilowatthours that would have been generated had the plant operated at full capacity throughout all the hours in the year (8.760 except in leap years).

The conversion from capital to generating costs may be illustrated as follows: Assume a plant of 1,000,000-kW capacity costing $2 billion, with annual fixed charges of 14 percent and operating at 80 percent capacity factor. The construction-cost contribution to generating costs would then be 40 mills/kWh, calculated as follows:

$$\frac{\$2{,}000{,}000{,}000 \times 14\%}{1{,}000{,}000\ \text{kW} \times 8{,}760\ \text{h} \times 80\%} = 40\ \text{mills/kWh}$$

Fuel Costs

The economic advantage of nuclear power plants lies essentially in fuel costs which are substantially lower than those of fossil fueled plants.

The fuel cost advantage is implicit in the compactness of nuclear fuel: 685 lb of uranium, if fully consumed, contains the energy equivalent of 1.7 million tons of coal, 7.2 million bbl of oil, or 32 billion ft^3 of natural gas. It has not yet proved possible in any water cooled reactor to consume more than a small fraction of the uranium used in nuclear fuel; yet this fraction is high enough to yield the cost advantage noted above.

Determining nuclear fuel costs is a far more complicated matter than determining fossil fuel costs because of the need to take account of a complex sequence of events which begins long before the uranium fuel

is inserted in the reactor and ends long after the spent fuel has been removed from the reactor. The entire sequence, which is called the nuclear fuel cycle, includes mining and milling of uranium, refining of uranium and conversion to uranium hexafluoride, enrichment of the uranium in the isotope uranium 235, conversion of enriched uranium to fuel material, fabrication of reactor fuel elements, use of fuel elements in nuclear power plants, recovery and marketing of by-product plutonium (not in the United States), chemical reprocessing of spent fuel to obtain reusable fuel material (not in the United States), and disposal of radioactive wastes.

Some principal items of nuclear fuel cost and possibilities of change include the following:

Fabrication The cost of fuel element fabrication is taken to include all the steps necessary to change a starting material (UF_6 in the case of enriched uranium elements) into a usable element. These steps involve chemical conversion to a powder or metal, metallurgical and mechanical processing to form and clad the elements, inspection, testing, and scrap recovery. Fabrication costs are expected to continue declining as greater quantities are handled, processing becomes more efficient, and fuel design is improved.

Recycling of Bred Plutonium Uranium-fueled light-water reactors in normal operation produce a certain amount of fissionable plutonium 239 through capture of neutrons in the nonfissionable isotope uranium 238. The separation is done by well-established chemical processes, the end products being uranium dioxide and plutonium dioxide. The plan to recycle spent nuclear fuel resulted in several reprocessing plants being built in the United States, but only one was ever operated. Widespread public concern over proliferation of nuclear weapons and the possible use of plutonium by terrorists and others resulted in a ban on reprocessing in the late 1970s. The ban was lifted in 1980, but since that time there has been no economic incentive to embark on an extended program of industrial reprocessing of nuclear fuel. This situation is entirely different elsewhere; reprocessing is currently a going operation in France, Russia, Great Britain, and elsewhere. The confluence of many factors will foster increased reprocessing of spent nuclear fuels, among which will be the matter of depletion of fossil fuels, accumulated pollution products from the combustion thereof, growth of power requirements worldwide, and so on. Certainly, at such future time as breeder reactors may become viable, they will provide an ideal place to utilize plutonium derived from reprocessed spent nuclear fuel produced in light-water reactors.

Shipping Charge The costs of transporting irradiated fuel elements from a reactor site to a chemical processing plant include charges for freight, shipping casks, handling, and insurance. The shape of the fuel element (having an effect on the size of cask required), the irradiation level, the cooling time required, the shipping distance, the route and type of transport, and the regulatory requirements are all factors which affect the cost.

Enrichment Enrichment in the fissionable isotope uranium 235 is accomplished at present (in the United States) only in three DOE-owned plants. This is the only part of the nuclear power industry not yet in private hands.

Burnup The amount of heat that can be obtained from a given weight of nuclear fuel before it is discharged from the reactor is known as the fuel's burnup. Higher burnups are expected as a result of technological improvements. This would have the consequences of reducing the amount of fuel that must be fabricated, reprocessed, and shipped per unit of electrical output. Cost reductions should be achieved as a result, but they may be limited by the fact that fuel elements capable of achieving the higher burnups may be more expensive to fabricate and process and may require higher enrichment.

Chemical Reprocessing and Waste Disposal The cost of chemical reprocessing, less any income from the plutonium or uranium recovered, is usually a net credit for foreign reactors. For U.S. and foreign reactors, the cost of waste disposal is an appreciable cost factor. Under U.S. law, that cost is assessed against the user utility as the power is generated, even though the disposal will take place decades later.

Decommissioning The cost of decommissioning the nuclear power plant when it is eventually closed down is treated differently in different countries. In the United States, these costs are required to be guaranteed by various accounting practices, which in effect accumulate a reserve for costs to be accrued in the future. Reactors decommissioned and dismantled are utilized as case studies to accumulate data on the costs incurred thereby. Other reactors that have been shut down have been defueled, and either stored under guard, or entombed in concrete, with ultimate decommissioning postponed to the future.

Operating and Maintenance Costs

The main cost items included in this category are station labor, training of replacement personnel, expendable materials, and supplies other than fuel, maintenance operations, and insurance.

The novelty and complexity of nuclear plants, a conservative approach to health and safety problems, and the need for specialized knowledge are all factors requiring a more skilled staff in nuclear plants and a longer period of training. Problems of coal and ash handling and maintenance of tubes and related boiler components require a larger working force in coal plants. While insurance costs have been higher for nuclear plants, it is expected that a continuance of safe operating experience will eventually equalize this item.

Numbers of operating personnel per generating unit are less for a plant having two or more nuclear units than for a single-unit plant. Increasing the size of a nuclear unit does not materially affect the number of people required to operate it. For these two reasons, one can expect declining personnel requirements per kilowatt of capacity at such time as the nuclear power industry expands present plant capacity and/or adds new plants to the overall inventory.

Total Generating Costs

Although many uncertainties becloud cost estimates for both nuclear and fossil fuel plants, in the 1970s, comparisons tended to favor nuclear power in many parts of the United States on either economic or noneconomic bases, or both. The continued and expanded use of fossil fuels in power plants is driven by several considerations, not the least of which is the current (1995) apparant plentiful supply at reasonable prices. In looking to the future, however, there are certain problems which are recognized and which must be addressed in the matter of fossil fuels. Domestic supplies of low-sulfur oil are diminishing, and dependence on imports makes for uncertain supply and is always subject to sharp price increases. Domestic coal is plentiful, but most of it, especially in the eastern United States, has a relatively high sulfur content which requires extensive and expensive steps to be taken to desulfurize it or to treat the effluent gases to meet local air pollution standards. Vast quantities of natural gas have come on the market in the last decade, but it is recognized that this source, too, is finite. Oil and natural gas have required deeper and deeper wells, in locations which add to the difficulties of extraction (i.e., offshore platforms in deep waters) and shipment.

The large-scale plans for expansion of the nuclear power plant base in the United States were dramatically scaled back in the 1980s, for a number of reasons. Nationwide, actual and forecasted growth of demand was markedly lower; the regulatory climate and length of time required from inception to the plant's going on-line were such that construction financing presented onerous economic problems which largely offset the erstwhile advantages to be gained from the use of nuclear fuel; and there was denial of retraction of plant certification by local authorities.

In the past several years, efforts have been made to reduce the time required from planning to final operation of nuclear plants. To that end, the industry has designed a number of standard size and type nuclear reactors which, once approved by the NRC, will not have to undergo the long-time, one-of-a-kind review for each proposed installation. Although no plants are on order in the United States, the major U.S. fabricators and suppliers of nuclear reactors are engaged, either solely or in partnership, with the growing nuclear energy programs in many other countries. It is expected that the store of expertise gained thereby will be available for application in the United States at such future time

when, as expected, nuclear power plants once more will serve to meet growing demands. At this time (1995), approximately 20 percent of the electric power generated in the United States originates in nuclear reactors. Outside the United States, there is great activity in the emplacement and operation of an expanded nuclear power base in the electric power generating industries.

Effect on Fuel Resources

The emergence of nuclear power as a viable energy source comes at a time when additional environmentally acceptable sources of energy are sorely needed, both in the United States and in other countries, to meet continued rapid increases in demand. As indicated above, domestic production of both natural gas and oil appears to be approaching its peak and may decline in the future. Dependence on foreign sources entails risks with regard to dependability of supply and the U.S. balance of payments. New technology is needed in order to be able to mine and use this country's extensive coal reserves in an environmentally and economically acceptable manner.

Uranium supplies are limited and, if utilized only in light-water receptors, would rapidly diminish. Breeder reactors, now under active development outside the United States, can utilize most (60 percent or more) of our uranium (and/or thorium) resources rather than just a minute percentage.

A practically unlimited source of energy is available from the deuterium in the world's oceans should development work now underway in several countries lead to production of useful power from controlled fusion reactions. The technical problems are formidable, however, and the harnessing of fusion as an energy source in large power generating plants is not expected until the latter half of the twenty-first century.

NUCLEAR POWER PLANT SAFETY

Revised and amended by George Sege

Through the end of 1993, over 1800 reactor-years of operation had been accumulated by commercial nuclear power plants in the United States, and about 6,900 worldwide, not including military or shipboard reactors (International Atomic Energy Agency, ''Nuclear Power Reactors of the World,'' 1994). This operational history has been accompanied by an impressive safety record with respect to health and safety impacts on the public, yet accidents have occurred. The most disastrous occurred in 1986 at the Chernobyl nuclear power plant complex located in the then U.S.S.R., resulting in loss of life and long-term radioactive contamination over a large area in the vicinity of the plant as well as transport of short-lived radioactive contamination dictated by the prevailing local weather patterns. Ultimate disposition of the destroyed reactor, now encased in an added containment structure (''sarcophagus''), remains a concern. In the United States, two of the worst commercial accidents occurred at the Brown's Ferry facility in 1975 and the Three Mile Island (TMI) facility in 1979. In the Brown's Ferry accident, a fire burned through hundreds of electrical power and control cables, threatening the plant's capability to be shut down and maintained in a safe condition. There was ultimately no damage to the reactor and no off-site radioactive releases occurred. At Three Mile Island significant reactor damage occurred through a sequence of events that began with a stuck-open primary system relief valve. Radioactivity was released to the environment, mostly in the form of noble gas fission products. However, estimated exposures to members of the public were found by a Federal interagency working group to be exceedingly small, even for the maximally exposed individual. The TMI reactor vessel did not fail despite relocation of approximately 19 tons of molten core materials. Although these accidents had minimal public impact, safety systems and plant operators were severely challenged during the events.

Commercial reactors have inherent safety features that prevent them from exploding like bombs. In a fission bomb, the fissioning of the fuel is caused by neutrons whose energy is about the same as the level at which they are produced by fission of the material in the previous generations. As a result, the average neutron lifetime is very small, and this allows the supercritical chain reaction to increase the power level to incredibly high levels before the device explosively destructs itself. In a power reactor, light-water or gas-cooled, enrichments of the fuel is very small (a few percent). As a result, essentially all fissioning of the fuel is caused by neutrons that have been reduced greatly in energy by the moderator (water or carbon). This results in a long average neutron lifetime compared to that in a bomb. If the chain reaction is allowed to increase in an uncontrolled fashion, this long lifetime will prevent the power of the reactor from rising to very high levels before the inherent design characteristics of negative temperature and void coefficients cause the reaction to shut down. While the Chernobyl reactor had design deficiencies in this regard, the violence of the accident was very far removed from what would result from detonation of a nuclear bomb.

There are two fundamental inherent safety features that contribute to the controllability of commercial power reactors. The first of these is the **Doppler effect**. As the power level of a reactor increases rapidly, the temperature of the fuel increases very rapidly. This rise in temperature causes the low-energy resonant peaks in the microscopic cross section to broaden, effectively creating a larger probability that neutrons will suffer nonfission captures by the U^{238} in the fuel. The second inherent safety feature is caused by the physical densities of materials, particularly the moderator, that decrease as the temperature increases. This results in higher probabilities that neutrons will escape from the reactor rather than cause fission in the fuel. Both of these inherent safety features feed back to the chain reaction in a negative way such that an uncontrolled increase in power level tends to shut the reactor down. This is called a **negative temperature coefficient**. Steam voids in a water moderator have a similar effect.

In spite of the inherent safety features, a reactor still provides a potential hazard to plant workers and members of the public in the vicinity of the plant. This is because of the generation of highly radioactive fission products within the fuel. The essential safety challenge is to assure that these fission products are kept confined at all times.

Nuclear power plant safety is based on a defense-in-depth philosophy. The defense-in-depth relies not only upon the inherent safety features, but also on high quality standards that provide a high degree of assurance that accidents are not likely to occur, and that if they do occur, the radioactivity will be contained locally.

High quality begins with the selection of highly qualified personnel to design, construct, and operate a plant. Each applicant for a license to build a nuclear plant must institute a quality assurance program to be applied to design, fabrication, construction, and testing of the structures, systems, and components of the facility. An applicant for a license to operate a plant must also provide for appropriate managerial and administrative controls to be used to assure safe operation. The function of the quality assurance program is to provide a high degree of confidence that the structures, systems, and components in the plant will perform as intended during the service life of the plant.

Nevertheless, accidents can occur, and other safety features are designed to keep fission products confined in the event of a broad range of accidents that could conceivably occur. Two categories of anticipated accidents are undercooling events and positive reactivity addition events. An undercooling event can be caused by such things as ruptures in the reactor cooling system, loss of flow through the reactor core, or loss of feedwater flow. Positive reactivity addition events are the result of such things as inadvertent control-rod withdrawals or dissolved boron dilution, main steam line breaks (pressurized water reactor), or overfeeding of a steam generator.

There are various types of engineered safety features found in today's commercial reactors. They can be grouped into four major areas: (1) reactor protection system, (2) emergency core-cooling systems, (3) containment, and (4) emergency power supplies. The following discussion of these features is generally applicable to light-water-cooled reactors, pressurized water and boiling water reactors, but the same general features are applicable to other types.

The purpose of the **reactor protection system** is to rapidly insert the control rods into the reactor upon receipt of a signal indicative of a malfunction that potentially threatens the integrity of the core. For ex-

ample, a low reactor coolant system pressure is indicative of a loss-of-coolant accident. High temperatures in the reactor coolant system or the core indicate the possibility of a loss of coolant or loss of flow through the core. The reactor protection system processes the various input signals and compares them to predetermined set points. If a set point is reached or exceeded, the system is designed to automatically insert the control rods into the reactor. The control rods absorb neutrons and quickly reduce the fission rate in the core. This rapidly drops the power level in the reactor and aids in prevention of damage to the fuel.

In the event of a loss-of-coolant accident, it is not sufficient to shut the reactor down in order to prevent damage to the fuel. Decay heat produced from the radioactive decay of fission products (initially some 4 percent of the previous operating power, but decreasing rapidly with time after stopping the chain reaction) would cause damage to the fuel if the coolant inventory losses were not made up by an injection system. **Emergency core-cooling systems** are designed to permit injection of coolant at high reactor coolant system pressures that exist during small ruptures as well as at low reactor coolant system pressures resulting from large breaks. Redundancy is provided to assure that equipment failures do not result in the loss of capability to keep the core covered with water. Large sources of water are stored to meet the long-term cooling requirements to remove the decay heat. In the event that a pipe rupture results in the depletion of the stored injection water, provisions are made to allow recirculation of water from the emergency sumps in the containment building back through the injection pumps. In time the rate of heat generation drops to the point where forced-circulation cooling is no longer required. After the Three Mile Island core damage event the system was cooling by natural convection to the surroundings at atmospheric pressure after about 3 years.

Containment features provide an important function essential to the defense-in-depth philosophy. There are actually three levels in the containment scheme. The first level is the fuel cladding, which not only mechanically holds the fuel pellets in the core, but also provides a boundary to prevent migration of fission products to the coolant. The second level is the reactor cooling system pressure boundary. This consists of the reactor vessel system piping, and other components designed to withstand pressures in the order of hundreds of atmospheres without failure. Parts of these systems that are outside containment in normal operation, as in a boiling water reactor, are isolated at the containment building boundary in case of an accident. Thus even if the fuel cladding should fail, the pressure boundary is likely to keep the fission products contained. In the event that the accident would cause both of these boundaries to fail, a third level is provided in the form of a containment building. The typical containment building consists of a steel shell surrounded by a concrete structure subjected to precompression by steel cables. Thick steel only, reinforced concrete, and combinations have all been used in containment schemes, most of which are designed to withstand 5- to 10-atm pressure, based on containing the total energy available during an accident. The destroyed Chernobyl reactor did not have a containment building; it had only confinement provisions for small local pipe breaks.

Emergency power supplies also exhibit redundancy and defense-in-depth features. The power requirements for auxiliary systems within a power plant are normally supplied from the reactor output itself through the main electrical generator and auxiliary station transformer. In the event of an accident, the reactor would not be available to produce useful power; thus the preferred power source for supplying emergency electrical loads is from the transmission lines from other power plants. At least two off-site power supplies are required. In case of loss of these sources, redundant on-site sources are also required, usually powered by quick-start diesel generators. Battery systems are also needed to provide certain direct-current loads and some alternating-current loads through inverters. Control, computer, and instrument supplies are typically so served.

Protection is provided not only for events originating in the plant but also for external events. Plants are designed to withstand safely the worst anticipated earthquakes, floods, and winds. Containment and other safety features of nuclear power plants generally offer significant protection against consequences of accidents exceeding the severity of design-basis accidents. In recent years, experimental work and probabilistic studies have been carried out to determine the extent of such margins and discover residual vulnerabilities that would then become the basis for possible further safety improvements, generally on a plant-specific basis.

As the operating years of plants have accumulated, there has been increasing attention to detecting and counteracting age-related degradation of plant materials and equipment. A notable example is the varying degrees of radiation-induced embrittlement of reactor vessel materials, which if unchecked should increase vulnerability to overcooling accidents (pressurized thermal shock).

Another development in recent years has been the increasing use of probabilistic safety assessments as a partial guide to plant safety actions as well as safety regulation. In such assessments, **fault trees** are used to trace malfunctions through all antecedent events to the initiating causes and **event trees** to trace undesired events through all their consequences to ultimate outcomes. Known or estimated probabilities are assigned at each step in the chain of causation, to derive probabilities of sequences of safety (or risk) interest.

The various inherent and engineered safety features, along with the high quality standards which are required in the design, construction, and operation of a nuclear power plant, make accidental releases of radioactivity to the environment extremely unlikely. Nonetheless, additional defense-in-depth is provided in the form of trained operators and procedural adherence. As a final feature of the defense-in-depth philosophy, emergency planning is required that establishes the protective measures that would be taken to protect workers and members of the public against possible radioactive releases. Such emergency measures are required to be practiced at least annually in the United States.

Further assurance of nuclear plant safety is provided by the regulatory requirements summarized in the next section.

NUCLEAR POWER PLANT LICENSING

Revised and amended by Paul E. Norian

NOTE. The following describes the process in the United States. Nearly all countries have similar processes, but the degree of public participation and appeal opportunities may differ in significant degree, generally resulting in less complex and time-consuming processes in other countries, especially those with nationally owned industries and utilities.

Licensing of commercial power reactors is conducted in accordance with the Atomic Energy Act of 1954, as amended. The applicable regulations appear in Title 10 of the Code of Federal Regulations (10 CFR). Licensing is currently conducted by the NRC in a two-step process involving the issuance of a construction permit (CP) and then an operating license (OL) when the plant is completed. The process begins with the filing by a utility of an application for the construction permit and is followed by safety, environmental, security, and antitrust reviews by the NRC staff. These reviews are conducted in parallel. Following the staff reviews, and report to the NRC, another review is conducted by the Advisory Committee on Reactor Safeguards (ACRS), a body independent of the NRC, which gives a formal opinion letter to the NRC. Finally a mandatory public hearing is held by a three-member Atomic Safety and Licensing Board (ASLB), which reports to the NRC. If all issues are resolved satisfactorily, the staff is allowed by the NRC to issue the construction permit. The applicant may then begin construction, provided other local, state, and federal requirements are met for a variety of nonnuclear matters such as water discharge, building construction, worker safety, etc. The total number of permits required may exceed one hundred.

Several years before the plant construction is completed the applicant files an application to the NRC for an operating license. The process that follows is similar to that for the construction permit except that the public hearing is not mandatory during this step. However, a hearing may be requested by affected members of the public or by the commission (NRC). In practice this usually occurs on request from some member of the public.

Major features of the licensing process, are summarized below.

Preliminary Safety Analysis Report (PSAR) The PSAR is the pri-

mary component of the application for a CP. It generally comprises 10 or more volumes of information concerning site characteristics, plant design features, operating features, and documentation of compliance with NRC regulations.

Final Safety Analysis Report (FSAR) The FSAR accompanies the application for an operating license. This document provides more detailed information than the PSAR about changes in design, requests for information from the NRC staff during the CP phase, changes in regulations since the issuance of the CP, and conformance with the additional regulations that are only applicable at the OL stage.

NRC Staff and ACRS Safety Reviews The applicant's safety analysis reports provide the basis for the safety reviews. The staff reviews the reports against standard NRC acceptance criteria derived from the regulations and documents the results of these reviews in **Safety Evaluation Reports (SERs)** at both steps in the licensing process. Clarifications are usually required from the applicant both in writing and in meetings. Following completion of the SERs, the ACRS independently completes its reviews. These ACRS reviews typically include many meetings with the NRC staff and applicant. At the conclusion of the ACRS review, a letter is sent to the chairman of the NRC providing the ACRS recommendation as to whether the CP or OL should be issued. Supplements to the SERs are issued by the staff that incorporate appropriate changes as a result of the ACRS recommendations.

Environmental Review (ER) This review is held at both the CP and OL steps pursuant to the National Environmental Policy Act of 1969. The staff review is based on a Safety Evaluation Report (SER) prepared by the applicant. Upon completion of the staff's analysis, a draft Environmental Statement is prepared and distributed for comment to the various federal, state, and local agencies and to members of the public. Comments are taken into account in the preparation of a final Environmental Statement.

Antitrust Reviews This review is conducted by the NRC and the attorney general (Department of Justice) to ensure that operation of the facility will not be in violation of the antitrust laws.

Public Hearings Separate hearings may be held on safety and environmental matters, or the issues may be consolidated into a single hearing. If an antitrust hearing is required, it is always held separately. Hearings are normally held close to the reactor site to allow local public participation.

License Issuance and Commission Reviews Since 1981 it has been NRC practice for the staff to initially issue an operating license restricting operations of the plant to power levels not exceeding 5 percent of the full power rating. This low power license is issued after the completion of the safety, environmental, and antitrust reviews, and after completion of the public hearings, and resolution of any appeals. The exception to this is that public hearings on off-site emergency planning need not be completed prior to the issuance of a low-power license. The low-power license permits plant heat-up and reactor system testing, which typically takes 2 or 3 months. After the major remaining safety issues are resolved and the staff has confidence that the applicant can operate the plant safely, the staff brings the matter before the five-member commission (NRC) for review. Following this review and a favorable finding by the commission, the staff is allowed to issue a license amendment authorizing full-power operation of the plant. Federal courts may review the commission decision on appeal from interested parties, with or without a stay of execution of the license as the court may decide. Following the term of the full-power license, the license can be renewed for a period up to 20 years in accordance with the requirements established by the NRC.

The NRC has issued a safety goal policy statement that established goals that broadly define an acceptable level of radiological risk that might be imposed on the public as a result of nuclear power plant operation. To implement the goals, numerical objectives on core damage frequency and containment performance have been established. The safety goals are among the considerations underlying NRC's generic regulatory actions, but do not supplant NRC regulations and are not applied to individual plants.

Continuing Review A licensed reactor is subject to continual oversight by the NRC throughout its operating life to assure that regulations and license conditions are observed. The NRC maintains an active program for evaluating and resolving generic safety issues, i.e., issues relevant to all plants or to a class of plants. Plant or procedural modifications are required if needed for adequate safety or justified on a safety-cost tradeoff basis. Licensees are required to monitor the performance or condition of certain structures, systems, and components, and the effectiveness of maintenance, to provide reasonable assurance that capability to perform their intended safety functions will be retained.

Reactor licensees are also required to perform a probabilistic risk assessment, called an **individual plant examination,** for their facilities. These risk assessment analyses are a result of the NRC policy regarding assessment of beyond-design-basis accidents (severe accidents) and are intended to identify any plant vulnerabilities needing attention.

On-site resident inspectors carry out a continuous inspection program and are augmented by specialized inspectors based in the NRC regional offices or headquarters. Significant events are subjected to thorough investigations to determine causes and to assure corrective actions. Should a safety hazard ever be presented at a facility, the NRC is empowered to take any necessary enforcement action, including the shutdown of the plant.

Standardized Design Standard plant designs can lead to greater assurance of safety, reduced costs through quality production, and less licensing review time by NRC. Under NRC's procedures, a standard design is reviewed using bounding site conditions for the range of sites at which the plant might be located. For safety considerations which do not involve site-specific issues, the licensing review is therefore conducted only once, even though many applicants may reference the design. The NRC is in the process of certifying several standard nuclear power plant designs, and it strongly encourages future applicants for licenses to reference them in their applications. The operation of multiple plants having essentially the same design features will provide a substantial safety benefit because experience gained at one facility may be applied to other like facilities. This should lead to more effective maintenance of key safety components and improved reliability of plant operations. Standardization would also stimulate programs of quality control and lead to better and faster training of operators and workers. It appears likely that when U.S. utilities do begin ordering nuclear plants again, the standard plant concept will be used, as has successfully been done in other countries.

Operator Licenses In addition to issuing operating licenses to the owners of nuclear power plants, the NRC also licenses persons who actually manipulate the controls of a reactor, at two responsibility levels, reactor operator and senior reactor operator. The licensing process involves training, experience requirements, and examinations, both written and practical, to verify an acceptable degree of knowledge on the part of the candidate before the license is granted or renewed. Because of the high cost of using an operating reactor for training and demonstration of practical facility in operation, many utilities use elaborate replica simulators of the control-room panels and systems of the power plant for operator training.

Decommissioning Five years before an operating license is to expire, licensees are required to inform the NRC for approval of their program for funding the management of spent fuel until disposal in a permanent repository. No later than 1 year before the license is to expire, the licensee must file an application to terminate its operating license, together with a detailed plan for decommissioning. With license renewal, decommissioning follows the period of extended operation.

Radioactive Waste The nuclear waste generated at nuclear power plants is generally classified as low-level waste, mixed waste, and spent fuel, although these plants are not significant generators of mixed waste. The low-level and mixed wastes can be shipped off-site to regional storage facilitates, if available. If not available, the low-level and mixed waste can be safely stored on-site until a facility is available. There is currently no permanent repository for storage of spent fuel, although the Department of Energy is working to establish one. Spent fuel is stored in spent fuel storage pools at each facility, and when they are filled, additional storage can be provided on-site by dry storage in independent spent fuel storage installations (ISFSIs). The spent fuel has normally

been allowed to cool for at least 5 years before it is placed in an ISFSI; these facilities are certified for 20 years, can be recertified, if needed, and have a design life of 50 years.

OTHER POWER APPLICATIONS

Nuclear Ships The earliest and most successful of the efforts to develop nuclear power for ship propulsion were those by the U.S. AEC and U.S. Navy. These initial efforts toward this end predated and contributed to the development of large electric power generators. The U.S. Navy launched the *USS Nautilus* in 1954 and followed it with about 125 other operational nuclear submarines and 13 operational surface vessels. Additional submarines and surface vessels have been committed by the U.S. Navy. Nuclear submarines and ships are also being operated by the British, Russian, and French and Chinese navies.

Although a ship can be propelled by diesel engines, steam engines, gas turbines, and other devices, the steam turbine has emerged as the principal modern marine propulsion unit. When a nuclear reactor of a type developed for generating stations is used to produce steam, the combustion process is eliminated. Such a propulsion plant is especially advantageous for a submarine, since it is then completely independent of the earth's atmosphere. The U.S. Navy's nuclear ships and submarines all employ the pressurized water type of nuclear reactor.

On commercial ships, the advantages of nuclear power are related to higher-speed operation over long run routes. Experience indicates that an increase in speed usually produces substantial increases in cargoes, thereby increasing a ship's load factor. In conventional ships, the power required for higher speed is increased by the third power of the speed ratio. This means larger engines and greater fuel storage capacity. On the other hand, a nuclear ship can operate continuously at the maximum level of its hull form without penalty in payload. Such advantages are especially desirable for icebreakers, large bulk cargo vessels, and high-speed long-distance passenger service. Efforts in the 1960s and 1970s to build commercial nuclear-powered ships resulted in the N.S. *Savannah* in the United States, the N.S. *Otto Hahn* in West Germany, the N.S. *Mutsu* in Japan, and the icebreaker *Lenin* in the Soviet Union. All proved to be failures. The first two were technical successes, but economic failures. The *Savannah* is a museum, and the *Otto Hahn* had her power plant removed and replaced by diesel propulsion units. The *Mutsu* had a radiation shield leak, never operated, and was eventually broken up. The *Lenin* had her nuclear power plant replaced by a more modern one, and with three larger nuclear-powered icebreakers and a large supply ship, also nuclear-powered, operates in the Russian Arctic. Economics of a state-owned ship are difficult to assess, but these are the only unarmed nuclear-powered surface ships in operation.

Other Vehicles The use of nuclear power in airplanes, automobiles, and railroad trains has been deterred by the requirements for heavy shielding to protect against radioactivity and collisions.

Remote Power Sources Small nuclear power plants have been built and operated by the U.S. Corps of Engineers in such remote regions as the Greenland ice cap and the Antarctic ice sheet. A barge-mounted plant was also built and operated in the Panama Canal Zone, but was scrapped.

Isotopic Power Generators Isotopic power generators are direct conversion devices that convert the energy of decaying radioisotopes to electricity, using direct thermoelectric or thermionic devices, or thermal conversion systems. (See Sec. 9.1.) The first isotopic power generator, completed in 1959, produced 2.5 W of power and used polonium 210 as fuel. The first commercial isotopic power generator, fueled by strontium 90, became available in 1966, and produced 25 W of power. There are now eight radioisotopes available in this area (see Table 9.8.4). Plutonium 238 was used in the long-range space probes Voyager 1 and 2.

Space Power Sources Most satellites and space vehicles use solar and battery power sources. However for very high powers, into the multikilowatt region, and for very deep space probes where solar energy is not available, nuclear sources (either radioisotopes or reactor heat systems) are required. Several dozen such systems have been deployed since 1970. Most are isotopic, but the United States orbited one reactor system in 1965. The Soviet Union has put several reactor-powered systems into orbit. On at least two occasions such U.S.S.R. systems have decayed back into the earth's atmosphere, and in one instance radioactive parts landed in Canada. In 1984, the United States reactivated development programs for high-energy reactor-powered systems for space use.

NUCLEAR FUSION

Development of nuclear fusion reactors for the production of useful energy will require significant advances from the present state of the art in both plasma physics and engineering. A large fusion effort, especially in physics research, has been carried on since 1956 in many industrial nations, notably the Soviet Union, Japan, the United States, and several in western Europe; considerable progress has been made.

There are a number of possible **fuel cycles** involving the fusion of light ions into heavier atoms, the most promising of which are shown in Table 9.8.5. The most accessible of these is the deuterium-tritium (DT) reaction, which involves fusion of two hydrogen isotopes. Even this lowest-temperature reaction of the possible fusion cycles has a thresh-

Table 9.8.5 Possible Fusion Fuel Cycles

Reactor equation	Approx threshold plasma temp*	Approx avg energy gain per fusion
$D + T \rightarrow {}^4He\ (3.5\ MeV) + n(14.1\ MeV)$	10 keV	1,800
$D + D \rightarrow {}^3He\ (0.82\ MeV) + n(2.45\ MeV)$ $D + D \rightarrow T\ (1.01\ MeV) + p(3.02\ MeV)$	50 keV	70
$D + {}^3He \rightarrow {}^4He\ (3.6\ MeV) + p(14.7\ MeV)$	100 keV	180

* 1 keV = 11,000,000 K approximately.

Table 9.8.4 Radioisotopic Materials

Radioisotope	Half-life, yr	Typical power density,* W/cm^3	Radiation present†	Shielding required
$Strontium^{90}$	28	1.4	Beta and bremsstrahlung	Heavy
$Cesium^{137}$	30	0.21	Beta and gamma	Heavy
$Cerium^{144}$	0.78	24.5	Beta and gamma	Heavy
$Promethium^{147}$	2.7	1.8	Beta	Minor
$Polonium^{210}$	0.38	1,210	Alpha	Minor
$Plutonium^{238}$	89	3.5	Alpha	Minor
$Curium^{242}$	0.45	1,150	Alpha	Minor
$Curium^{244}$	18	26.4	Alpha and neutron	Moderate

* Reduced below theoretical maxima in order to account for expected isotopic impurities.
† Of dominant importance in heat generation and for consideration of radiation shielding.
SOURCE: AEC TID-20079.

old temperature of about 50 million K, and a practical reactor would require operating temperatures of about 100 million K. At these temperatures the ionized gas or plasma cannot be contained in vessels of conventional materials.

Fortunately, ionized plasmas can be contained by magnetic or inertial forces on earth. (The level of gravitation forces which produce fusion in the sun and stars cannot be achieved on earth.) Research seeking how this can be done at high enough combinations of temperatures, pressures, and densities to obtain practical net energy output is the main effort of fusion programs worldwide. This research is concentrated in three major approaches:

1. **Toroidal sytems,** especially of the tokamak variety, in which very high currents are circulated through an endless plasma confined by toroidal (doughnut-shaped) magnetic systems of various physical cross sections and magnetic shaped fields.
2. **Magnetic mirror systems,** in which the ends of a solenoidal (linear) magnetic field are ''plugged'' by a combination of electrostatic and magnetic forces to make the plasma reflect back and forth.
3. **Inertial systems,** in which small pellets of fusionable material are rapidly compressed by beams of laser-produced light, electrons, heavy ions, or other compressive forces. This is the principle employed in nuclear fusion weapons.

For practical energy production, a DT plasma should have a temperature of about 100 million K. This is usually expressed as equivalent ion temperature in electron volts (1 eV = 11,000 K approx.). Coincidentally with this temperature the product of density (in ions per cubic centimeter) and confinement time (in seconds) should reach 10^{14}. These conditions have been achieved experimentally.

Three large tokamak installations (TFTR in the United States, JET in Europe, and JT-60 in Japan) have begun experimental operations. Each of these machines is expected to create plasma conditions approximating those required in practical power producing reactors, although they are not in themselves net power producing devices. Likewise, a large magnetic mirror experiment is under construction at Lawrence Livermore National Laboratory in the United States to determine if similar plasma conditions can be achieved in a mirror configuration. Also at Livermore a very large laser device is being used to attempt to establish similar conditions by inertial means, using very tiny pellets filled with DT mixtures. All these devices are experimental.

After practical reactor conditions have been achieved in a confined plasma, much engineering development associated with removing the energy and converting it into useful form will be required. Developmental work on fusion engineering continues, but formidable obstacles lie ahead, especially in the requirements for materials capable of surviving the environment of burning plasmas for long periods of time.

Although scientific feasibility seems ensured and is expected to be demonstrated in the late 1990s, the very considerable engineering problems yet unresolved put the operation of a demonstration fusion power reactor well into the twenty-first century, with any significant effect on world energy supplies occurring at some future time. Since uranium resources appear adequate for many decades, the essentially unlimited fuel supplies for fusion (from the oceans) may not be needed until then.

9.9 HYDRAULIC TURBINES

by Robert D. Steele

REFERENCES: ''Water Power Engineering,'' McGraw-Hill. Creager and Justin, ''Hydroelectric Handbook,'' Wiley. Daugherty and Ingersoll, ''Fluid Mechanics,'' McGraw-Hill. Hammitt, ''Cavitation and Multiphase Flow Phenomena,'' McGraw-Hill. Kovalev, ''Hydroturbines, Design and Construction,'' Russian translated by U.S. Department of the Interior and National Science Foundation. Mosonyi, ''Water Power Development,'' Hungarian Academy of Science.

GENERAL

Notation

B = width of distributor or height of wicket gates, in (mm)
D = diameter of runner, in (mm)
D_l = diameter of runner, at centerline of guide case, in (mm)
D_{th} = throat diameter of runner, in (mm)
D_r = relative diameter of runner, in (mm) (= entrance diameter for low-speed Francis turbines; throat diameter for high-speed Francis turbines; bore of throat ring at centerline of blade for propeller turbines)
D_d = diameter top of draft tube, in (mm)
D_p = pitch diameter of impulse-turbine runner, in (mm)
d = jet diameter of impulse turbine, in (mm)
e = overall efficiency of turbine = $e_h e_m$
e_h = hydraulic efficiency (including draft tube)
e_m = mechanical efficiency
g = acceleration of gravity, ft/s^2 (m/s^2)
H = net effective head, ft (m)
b = head change due to load change, ft (m)
n = r/min
n_l = r/min at 1 ft (m) head = $n/\sqrt{H}$
n_s = specific speed = $n\sqrt{P}/H^{5/4}$
P = horsepower = 0.746 kW
P_1 = horsepower (kW) at 1 ft (m) head = $P/H^{3/2}$
Q = discharge, ft^3/s (m^3/s)
Q_1 = discharge at 1 ft (m) head = $Q/\sqrt{H}$, ft^3/s (m^3/s)
t = thickness, in (mm), or time, s
u = circumferential velocity of a point on runner, ft/s (m/s)
V = absolute velocity of water, ft/s (m/s)
V_u = tangential components of absolute velocity = $V\cos\alpha$
V_r = component of V in radial plane
v = velocity of water relative to runner, ft/s (m/s)
w = weight of water lb/ft^3 (kg/m^3)
z = number of runner buckets (blades for propeller turbines)
α = angle between V and u (measured between positive directions)
β = angle between v and u (measured between positive directions)
θ = angle of water deflection relative to bucket
ϕ = peripheral coefficient = $[\pi Dn/(720\sqrt{2gH})][\pi Dn/(60{,}000\sqrt{2gH})]$
σ = cavitation coefficient

Fundamental Formulas The actual power of a hydraulic turbine is the theoretical power P_t multiplied by the turbine efficiency e.

Power in U.S. Customary System (USCS) Units

$$P = eP_t = \frac{eHQw}{550}$$

In the power equation above, **power** is calculated in horsepower. In the following text, when power P is used in stated equations, it is in horsepower.

Power in Metric Units

$$P = eP_t = \frac{eHQw \times .746}{76.04} = \frac{eHQw}{101.93}$$

The metric equation stated above for power calculates power in kilowatts. In the following text, where power P is used in metric equations (shown in parentheses), the power stated in the equations is in kilowatts. [The customary unit of horsepower in the United States should not be

confused with the term *metric horsepower* (see the conversion and equivalency tables of this handbook).]

The laws of proportionality for homologous turbines are:

For constant runner diam	For constant head	For variable diam and head
$P \propto H^{3/2}$	$P \propto D^2$	$P \propto D^2H^{3/2}$
$n \propto H^{1/2}$	$n \propto 1/D$	$n \propto H^{1/2}/D$
$Q \propto H^{1/2}$	$Q \propto D^2$	$Q \propto D^2H^{1/2}$

Nomenclature The nomenclature used throughout this section is based on NEMA's standards publication HT1-1957, ''Hydraulic Turbines, Governors and Accessory Equipment,'' which also contains definitions.

Types of Turbines Three characteristic types of hydraulic turbines are now in general use: the **Francis reaction** type (Figs. 9.9.1 and 9.9.2);

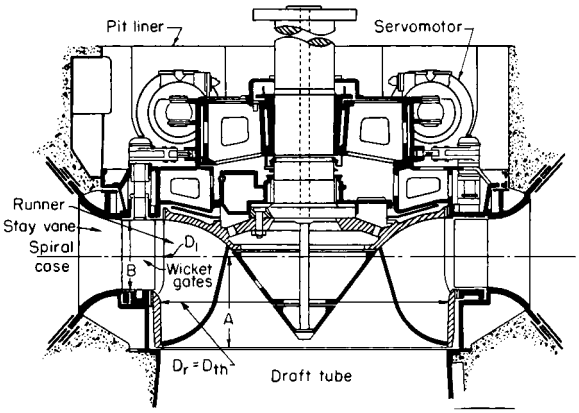


Fig. 9.9.1 Medium-head Francis turbine.

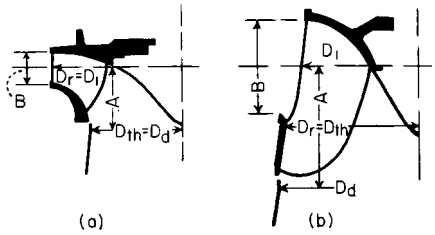


Fig. 9.9.2 Typical profile of Francis runners. (*a*) Low specific speed; (*b*) high specific speed.

the **propeller reaction** type (Fig. 9.9.3) and the **impulse** type (Fig. 9.9.12). The propeller type may be further divided into fixed and adjustable blade types. All three types have in common a stationary guide case (or nozzle in the case of the impulse type) in which the static head is transformed partly, or wholly, into velocity, and a revolving part, the runner.

In the guide case of the impulse turbine, the static head is completely transformed into velocity, so that air surrounds both the jet issuing from the nozzle and the runner.

In the guide case of the reaction types, the static head is only partly transformed into velocity, leaving an overpressure between the guide case and the runner. This overpressure causes an acceleration of the relative velocity of the water passing through the runner, the discharge area of which is smaller than the entrance area. Except when operating vented at low loads, the water passages are completely filled with water from the intake to the end of the draft tube.

Selection of Turbine Type and Casing Impulse turbines receive

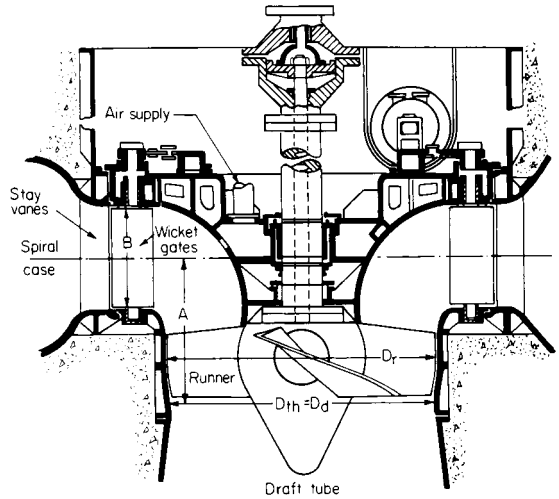


Fig. 9.9.3 Adjustable-blade Kaplan turbine.

their water supply directly from a pipeline. Francis and propeller reaction types are set in a case of concrete or metal. While the limits of head to which the impulse and reaction types are adopted may be fairly well defined in practice, as roughly outlined in Table 9.9.1, there is no definite line where the application of one type ends and the other begins. The choice between impulse and reaction types depends upon the size of the unit as well as upon the head and other considerations.

Specific Speed The common basis of comparison between turbine runners of different types and between runners of the same type but different design and characteristics is termed *specific speed* n_s. This is the relationship between the speed of a runner at the point of highest efficiency and the maximum power output at this speed regardless of size. However, since both power and speed vary with head, specific speed is defined as the relationship between the speed n_1 and power P_1 at 1 ft (m) head. Subscript 1 denotes that the value is reduced by the similarity law to a 1 ft (m) head basis. Since $n \propto 1/D$ and $P \propto D^2$, the product $n_1\sqrt{P_1}$ remains a constant regardless of the size of the runner and is designated the specific speed of the runner. The term *specific speed* for this relationship stems from the fact that $n_1\sqrt{P_1}$ also is the value of the speed in r/min at best efficiency which the runner would have if operated under 1 ft (m) head, the runner being of such size as to develop 1 hp (kW) ($P_1 = 1$).

Since $n \propto \sqrt{H}$ and $P \propto H^{3/2}$, the specific speed of any runner operating under head H will be $n_s = n\sqrt{P}/H^{5/4}$, where n is the best efficiency speed and P the maximum power output at this speed, all at head H.

Selecting the Speed Hydraulic turbines are usually connected to ac generators. The turbine speed must agree with one of the synchronous speeds required for the system frequency. The prevailing frequency for most systems in the United States is 60 Hz. Synchronous speeds are determined by the formula $n = 120 \times$ frequency/number of poles in generator. The number of poles must be even.

The speed should be as high as practicable, as the higher the speed the less expensive will be the turbine and generator and the more efficient will be the generator.

A convenient way of determining the highest practicable speed is by the relation of the specific speed to the head. For Francis turbines this may be taken as $n_s = 900/\sqrt{H}$ $(1{,}900/\sqrt{H})$; for propeller-type turbines as $n_s = 1{,}000/\sqrt{H}$ $(2{,}100/\sqrt{H})$. The adoption of these values of specific speed is predicated on the use of a reasonable setting of the unit with reference to tailwater level, i.e., selection of proper cavitation coefficient.

Number of Units From the standpoint of reducing the number of auxiliaries and the amount of associated equipment and also reducing initial and maintenance costs for the entire plant, the number of units should be kept to a minimum. Also the larger the unit, the higher the

Table 9.9.1 General Arrangements of Turbine Installations and Usual Head Limits Employed

Type	Setting	Construction	No. of runners	Usual head limits for direct-connected units	
				ft	m
Reaction turbines, 5 to 1,000 ft (1.5 to 300 m) head	Axial flow	Vertical or horizontal or slanted	1	5–60	1.5–20
	Encased	Concrete vertical	1	15–130	5–40
		Cast or welded plate steel. Stainless steel. Vertical or horizontal	1 or 2	30–1,600	10–500
Impulse turbines, 500 to 6,000 ft (150 to 1,800 m) head	Encased	Horizontal (1–2 nozzles)	2	500–6,000	150–1,800
		Vertical (1–6 nozzles)	1	500–6,000	150–1,800

efficiency and generally the lower the cost per unit power output. However, other considerations, such as flexibility of operation, higher efficiency operation during low-load demands, and minimum loss of capacity during shutdown for repair or maintenance, might dictate the use of multiple units where one unit would be feasible in terms of physical size. For some projects, the physical size of the unit has been limited to the maximum size runner that could be shipped in one piece; this is largely due to the extra manufacturing costs involved in furnishing split runners. However, since split runners present no serious mechanical difficulties, the tendency in recent years has been to disregard this limitation.

REACTION TURBINES

Francis Figure 9.9.1 shows a Francis-type reaction turbine for medium head. The runner consists of a relatively large number of shrouded buckets. Movable wicket gates with axes parallel to the turbine shaft control the flow. This type of turbine is normally used for heads ranging from 75 to 1,600 ft (25 to 500 m). Specific speeds vary from 15 to 100 (50 to 400).

Propeller Turbines, Fixed and Adjustable Blade The propeller turbine has a runner which is normally provided with from 3 to 10 unshrouded blades, either fixed or adjustable. This type of turbine usually is used for heads from 5 ft (1.5 m) up to 120 ft (40 m), although in a few cases they have been used for heads up to 200 ft (60 m). The higher the head, the greater the number of blades used. Specific speeds vary from 80 to 250 (300 to 1,000). Propellers have very steep efficiency vs. power curves (4 and 5, Fig. 9.9.5). Adjustable-blade propeller runners are used to produce a flat efficiency curve over a wide range of power (6, Fig. 9.9.5) and to produce considerably more power beyond the maximum efficiency point than can be obtained with a fixed-blade runner of equal diameter. For fixed-blade runners, the blade angle is usually set between 16 and 28°, where maximum efficiency occurs. For adjustable blade runners, the blade angle may vary from −10° min to +45° max. Normally, automatic oil-pressure-operated blades are used in a **Kaplan turbine** (Fig. 9.9.3). The blades are adjusted by means of an oil-operated piston located either within the main shaft or in the runner hub. The oil is admitted to and discharged from the piston by means of a distributor head either on top of the generator shaft or surrounding the main shaft. The oil pressure is supplied from the governor oil pressure system. The controls are so arranged that the blade tilt varies automatically with the wicket gate opening to produce a maximum efficiency envelope curve (Fig. 9.9.9). A large operating-cylinder capacity is required. The capacity S, ft · lb (J) may be approximated from the formula $S = 20Pn_s^{1/4}/\sqrt{H}$ ($14Pn_s^{1/4}/\sqrt{H}$). If antifriction bearings are used in the hub instead of bronze bearings for the blade trunnions, this formula becomes $S = 10Pn_s^{1/4}/\sqrt{H}$ ($7Pn_s^{1/4}/\sqrt{H}$). However, since the hydraulic unbalance on the blades varies with speed, gate opening, and the location of the blade pivot axis, the oil-operated cylinder size should be calculated from model test data.

Axial-Flow Turbines Axial-flow turbines use the propeller type runner with either fixed or adjustable blades. Their characteristic feature is the straight-through, or nearly straight-through, water passageway from intake to discharge. The shaft is, therefore, either horizontal, vertical, or inclined (Fig. 9.9.4). The spiral or semispiral case and elbow draft tube, which require substantial widths and depth of excavation, are eliminated. Therefore, with a reduction in height and area of the powerhouse and the turbine's suitability for location directly within the dam, an overall construction-cost savings may be obtained for the power plant part of the project compared with conventional vertical units. This may make it possible to build or redevelop power plants for low heads or in small sizes that have previously been considered uneconomical.

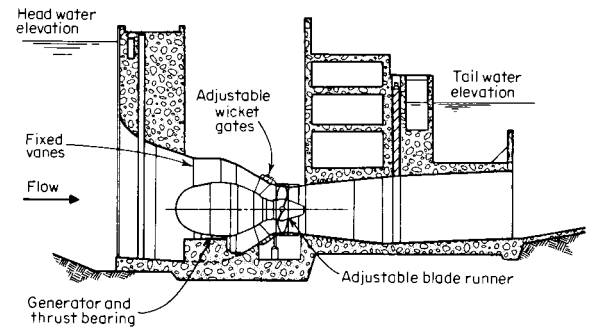


Fig. 9.9.4 Axial-flow bulb turbine.

In recent years, axial-flow turbines have been installed for use with tidal power since they can be arranged to operate with water flowing in either direction and, when required, to pump with water flowing in either direction.

There are four general types of axial-flow turbines. The first type has the generator rotor mounted around the periphery of the turbine runner. This type of turbine and generator arrangement has been called a **rim** turbine. The second type is the **bulb** turbine, where the generator is directly connected to the turbine at a submerged elevation. The bulb turbine has the generator enclosed in a streamlined, watertight housing located in the water passageway on either the upstream or downstream side of the runner (Fig. 9.9.4). The third type of axial-flow turbine is the **pit** turbine. In the pit turbine arrangement, a high-speed generator is driven from the high-speed output shaft of a speed increaser, and the low-speed input torque to the speed increaser is supplied from the low-speed shaft of the turbine. The generator of an axial-flow pit turbine can be mounted in line with the turbine, or in smaller units the generator may be mounted inclined or vertical.

The fourth type of axial-flow turbine is the **tube** type, which has the generator located outside the water passages. The construction characteristic of this turbine is the slight bend of the water passageway that permits the extension of the turbine shaft to the outside of the water passageway. The tube turbine is typically arranged so that the generator is on the downstream side of the turbine, but it could be constructed to have the generator on the upstream side. The shaft from the turbine to

the generator of a tube turbine can be inclined to reduce excavation, thereby raising the generator higher above tailwater elevations. In some cases, a gear-type speed increaser is used to reduce the combined equipment cost and generator size and weight.

Axial-flow turbines are suitable for heads up to at least 90 ft (30 m) with basically the same limitations that apply to the conventional Kaplan or other propeller-type turbines. Maximum unit capacity is limited, however, by maximum practical speed increaser torques and maximum practical horizontal generator capacities. This appears to be approximately 100 MW at present. Either fixed-position or movable wicket gates can be used. Power and efficiency performance is comparable to conventional vertical shaft propeller turbines.

Design The water entering the runner is given a whirl or tangential component by the guide vanes. This whirl is taken out by the runner so that, at the design point (point of best efficiency), the water leaves the runner without appreciable whirl. According to Euler's theorem, the power of a turbine in steady motion equals the angular velocity multiplied by the change in angular momentum experienced by the mass of water flowing in a unit time in its passage through the turbine. This is expressed in the following equations:

$$P = 62.4Q(u_1V_{u1} - u_2V_{u2})/(550g) = QHe_n/8.81 \quad (9.9.1)$$

and

$$u_1V_{u1} - u_2V_{u2} = e_hgH$$

$$P = 1{,}000Q(u_1v_{u1} - u_2v_{u2})/(101.93g) = QHe_hw/101.93 \quad (9.9.2)$$

(See notation at beginning of this section.) Subscript 1 refers to the inlet edge of the bucket and 2 to the discharge edge.

In practice, e_h may be taken as 0.92 to 0.94, and the term u_2V_{u2} may be omitted as it becomes zero for discharge in a radial plane, $\alpha_2 = 90°$.

The right hand member of Eq. (9.9.2) may be regarded as a fixed value for a given head. Consequently, in order to have a high-speed-type runner (u_1, large) the whirl V_{u1} placed in the water by the guide case must be small. This means that for the best load, low-speed low-capacity (small n_s) runners will have relatively small gate openings and short wicket gates, whereas high-speed high-capacity (large n_s) runners will have relatively large gate openings and long wicket gates.

Equations (9.9.1) and (9.9.2) are used in designing the runner and guide case. The P and Q used in Eq. (9.9.1) are for the point of best efficiency and not for the rated values. For specific speeds up to about 60 (250) the power at best efficiency may be taken as 85 percent of the rated power. Above that specific speed the point of best efficiency occurs at higher loads as indicated in Fig. 9.9.5. Adjustable-blade propeller turbine-runner blades are laid out for an intermediate load of 75 to 80 percent of the rated load. The actual design of a runner should be worked out by an experienced designer on the basis of coordinated test data.

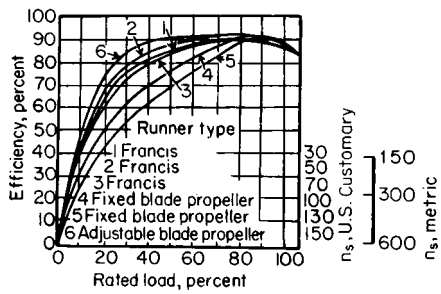


Fig. 9.9.5 Variation of efficiency vs. load for reaction turbines.

The general proportions of a normal line of turbine runners may be determined as follows for any combination of net head H and rated power P. (Rated power is usually considered to be 95 percent of full gate power.)

First select the specific speed appropriate to the head (see ''Specific Speed'' above). Next calculate n from the formula $n_s = n\sqrt{P}/H^{5/4}$. If $n/H^{5/4}$ is not a synchronous speed, select the nearest synchronous speed and calculate the corresponding n_s. This value of n_s may be used in Fig. 9.9.6 to determine the peripheral coefficient ϕ for D_r and D_1. Then the diameter may be determined from the formula

$$\phi = \pi Dn/720\sqrt{2gH}\ [\ = \pi Dn/(60{,}000\sqrt{2gH})]$$

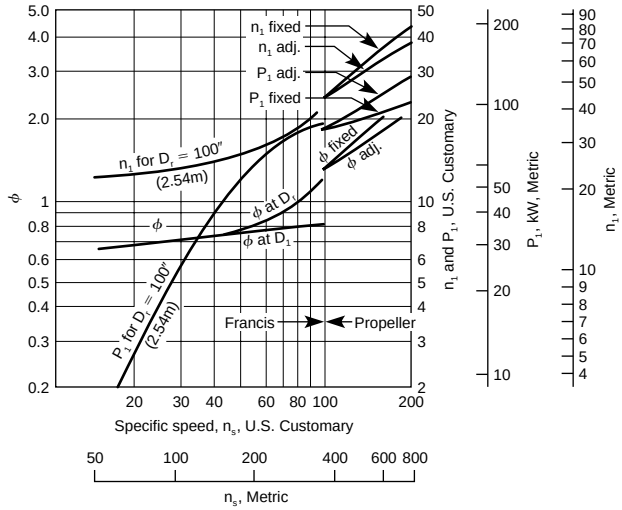


Fig. 9.9.6 Turbine characteristics and specific speed.

Other proportions of the runner may be determined from Fig. 9.9.7, which shows the ratio to D_r of width of distributor B; diameter at top of draft tube D_d; and centerline of distributor to top of draft tube A. See Fig. 9.9.1 for D_r, B, and A.

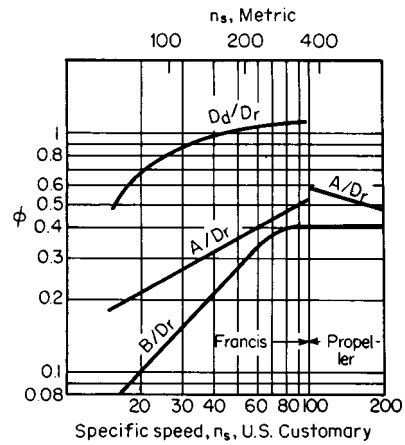


Fig. 9.9.7 Turbine characteristics and specific speed.

The values of unit power P_1, and unit speed n_1, for $D_r = 100$ in (2,540 mm), which correspond to the specific speed of the runner may be determined from Fig. 9.9.6. The following formulas may then be used: $P = P_1H^{3/2}(D_r/100)^2\ [\ = P_1H^{3/2}(D_r/(2{,}540)^2]$ and $n = n_1\sqrt{H}\ (100/D_r)\ [n_1\sqrt{H}\ (2{,}540/D_r)]$. These formulas may be used to verify the diameter calculated from the peripheral coefficient, and will prove useful in selecting runner sizes for units which must meet power requirements other than rated power at design head. In such cases the values of n_1 and P_1 from Fig. 9.9.6 should be used in conjunction with Fig. 9.9.8.

For preliminary powerhouse layouts these ratios of diameter to D_r may be used for all specific speeds: gate circle, 1.20; stay ring, 1.65; pit liner, 1.50; unit spacing, 3.5 to 4.5.

The clearance space between the discharge tips of the wicket gates and the entrance edge of the runner buckets is treated as a free vortex space in which the tangential component of the absolute velocity of

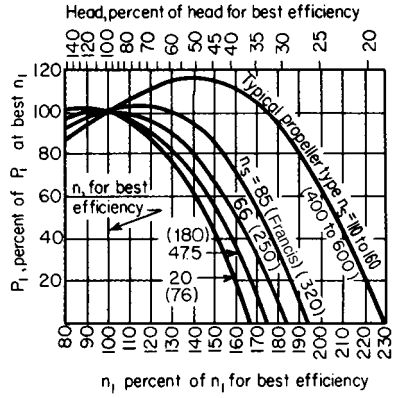


Fig. 9.9.8 Variation of unit power with unit speed and specific speed.

discharge from the gates (V_u) increases in inverse proportion to the radius while the component in a radial plane (V_r) increases in inverse proportion to the area.

To obtain the best efficiency, the water must enter the runner without shock and leave it with as little velocity as possible; i.e., the entrance angles of the buckets must be approximately in line with the relative direction of the entering flow and the direction of the absolute discharge velocity should be approximately in a radial plane (axial in the case of propeller runners). This condition prevails only at the most efficient load. Above that load the water enters the draft tube with reverse whirl and below that load with a forward whirl.

The shape of the buckets should be kept as simple as possible, consistent with meeting the proper angles. The curvature of the bucket along a flow line is made greatest near the entrance and reduces as the discharge edge is approached, somewhat the shape of a portion of a parabola.

Turbine Characteristics Speed, power, and efficiency characteristics vary widely with specific speed as shown in Figs. 9.9.5 and 9.9.6. The variation of unit power P_1 with the unit speed (as a percent of normal basis) is shown in Fig. 9.9.8 for typical specific speeds ranging from 20 to 160 (70 to 600). For low specific speed Francis-type turbines, P_1 decreases as n_1 increases, as when operating at heads below normal. This is due to the centrifugal effect, which tends to decrease the flow as n_1 increases. This characteristic of low specific speed Francis runners makes them less suitable for operation under extreme variations in head. With the higher specific speeds and particularly with the propeller type, P_1 increases as n_1 increases above normal. This makes the higher specific speeds particularly suitable for operation at widely varying heads.

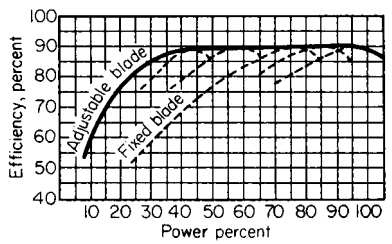


Fig. 9.9.9 Variation of efficiency vs. power for fixed- and adjustable-blade propeller turbines.

Figure 9.9.9 shows the advantage of blade adjustment for propeller-type turbines. By angular adjustment of the blades as the load changes, the peaked power-efficiency curve of the fixed blade propeller is transformed into a flat power-efficiency curve, and the maximum power output is considerably increased.

Runaway Speed If a turbine runner is allowed to revolve freely without load and with the wicket gates wide open, it will overspeed to a value called the *runaway speed.* The runaway speed, at normal head, varies with the specific speed and for Francis turbines ranges from 170 percent (normal speed = 100 percent) at low specific speed [n_s = 20 (76)] to some 195 percent at high specific speed [n_s = 100 (383)]. For propeller turbines, the runaway speed varies with blade angle—the steeper the blade angle, the lower the runaway speed. For fixed-blade propellers with the blades set from 16 to 28°, where maximum efficiency is usually obtained, the runaway speed ranges approx from 255 to 180 percent, respectively. For adjustable blade turbines, where the minimum blade angle is sometimes as low as 6 to 16° in order to obtain high efficiency at part load, the maximum possible runaway speed will be about 290 to 270 percent, respectively. However, with adjustable-blade propeller turbines, there is from the standpoint of efficiency an optimum relationship between runner-blade angle and wicket gate opening, usually controlled by a cam in the operating mechanism; the higher the gate opening, the steeper the blade angle. Thus the combination of wide-open gate and minimum design blade angle can only occur in the so-called ''off-cam'' position, which is an extremely rare possibility. In some units, this maximum possible off-cam runaway speed is reduced by limiting the minimum design blade angle to 16°. Another method is the use of a runaway speed limiter. There are several designs in use. One of the most frequently used is a valve designed to open by centrifugal force at speeds of 120 percent (approximately). The open valve bypasses the runner blade servomotor piston, thus equalizing the oil pressure on both sides of this piston. If the runner blades are designed to open because of hydraulic unbalance at overspeeds, they will then go to a higher blade angle under off-cam overspeed conditions. The off-cam runaway speed thus can be limited to some 185 percent if the runner is designed for a maximum blade angle of 28 to 40°.

For all turbines, if the maximum head is higher than the normal head, the runaway speed will be increased in proportion to the square root of the head. Therefore, runaway speeds should be based on the maximum operating head rather than on the normal head. Any runaway speed above 180 percent can increase appreciably the cost of the generator.

Turbine Thrust The water passing through the turbine creates a thrust load, that is carried by a thrust bearing, typically supplied with the generator. For vertical units, the thrust T consists of the hydraulic thrust T_h plus the weight of the turbine runner and shaft T_w, lb (kg). Units have been constructed with the thrust bearing supporting the total weight of rotating components of the turbine and generator, plus the hydraulic thrust, located on the turbine head cover.

For horizontal or inclined units, the thrust T consists of the hydraulic thrust T_h and a component of the weight T_w of the rotating components, if the unit is inclined. The thrust bearing can be located in the generator, in a speed increaser, in the hydraulic turbine, or in a separate thrust bearing housing.

The hydraulic thrust may be obtained by using a **thrust coefficient K_t** multiplied by the weight, lb (kg), of a circular column of water whose diameter is the diameter of the runner D_r, in (mm), and whose height is equal to the maximum head H_m, ft (m), under which the turbine must operate, or

$$T_h = \frac{K_t D_r^2 H_m}{2.94} \quad \text{lb} \qquad \left(T_h = \frac{K_t D_r^2 H_m}{1{,}273} \quad \text{kg} \right)$$

The thrust coefficient for Francis turbines varies with the specific speed and may be taken as approximately $K_t = n_s/250$ ($K_t = n_s/954$). In using this value of K_t, it is assumed that the runner seals are properly arranged and that the crown of the runner inside the seals is properly drained; otherwise, the thrust may be much higher.

For propeller turbines, with either fixed or adjustable blades, the thrust coefficient K_t may be taken as 0.90 for all specific speeds.

Reaction Turbine Elements

Runner and Wearing Rings The number of buckets z for Francis runners ranges from about 21 for low specific speed to 12 for high specific speed. The approximate number may be taken as $z = 55/n_s^{1/3}$ $(86/n_s^{1/3})$.

Materials that can readily be repaired by welding with either carbon-steel or stainless-steel electrodes are selected for the construction of runners. Most runners are made of cast carbon steel, cast stainless steel, formed carbon steel plate, formed stainless-steel plates, or a combination of carbon steel and stainless steel. The majority of new equipment is supplied manufactured of stainless steel, using a combination of martensitic and austenitic stainless-steel casting, forgings, and plates. The use of high-strength stainless steel for runners is increasing for high heads and for conditions where cavitation pitting may be a design consideration.

The number of blades for propeller-type turbines ranges from 10 for low specific speeds to 3 for high specific speeds. Propeller runner blades are usually made of cast carbon steel, formed carbon steel, formed stainless steel, or cast stainless steel. If they are made of cast steel or formed carbon plate steel, the surfaces over which pitting may be expected are overlay-welded with stainless steel before finishing to final contour. In place of overlay welding, solid stainless-steel inserts are sometimes welded into propeller blades, and stainless trim is added to the periphery of the blades.

The functions of the runner seals for Francis turbines are to prevent excessive leakage loss and thus improve the efficiency and control the hydraulic thrust.

Rolled, forged, and cast steels make excellent wearing rings for low- and medium-head units. To prevent seizure in case of contact the rotating ring material should differ from that of the stationary rings. Stainless steel, bronze, or steel with bronze inserts make excellent wearing rings. For extremely high heads, stainless steel should be used to prevent undue wear and erosion.

Seal clearances are made as small as practicable to reduce leakage, particularly on high-head units. Larger clearances are required with water-lubricated main bearings, subject to considerable wear before being readjusted.

Main Shaft and Bearings The main shaft, for a typical vertical hydraulic turbine unit, must be rigid and should be made of a medium-grade carbon-steel forging. The size of the shaft is determined by limiting the torsional stress from 3,000 to 6,000 lb/in^2 (20 to 41 N/mm^2).

The shafting system of a horizontal or an inclined hydraulic turbine unit is subject to the torsional stress produced by the torque transmitted and the bending stress produced by the weight of the runner, the shaft, and the components that are attached to the shafting system. The design of a shaft for a horizontal unit must be subjected to review by experienced mechanical engineering designers to ensure acceptable stress levels, to account for points of stress concentration, and to determine the overall safety margin of a design. The allowable stress levels in a horizontal shaft must be evaluated after the fatigue properties of the shafting material and the points of stress concentration are taken into consideration. Should the bearing be designed to be water-lubricated, a stainless-steel sleeve is installed on the shaft to provide a bearing surface for the guide bearing. A stainless-steel sleeve is typically placed on the shaft in the area of a packing box or stuffing box.

Vertical hydraulic turbines are usually provided with one main bearing located in the head cover, as near the runner as practicable. The main turbine bearing is generally a babbitted oil bearing, supplied in halves, or as independently adjustable segmented babbitted shoes. The oil is supplied to the bearing by an independent low-pressure oiling system. Self-lubricated babbitted shell or shoe bearings, containing an oil reservoir, and pumping grooves in the babbitt are sometimes used.

Water-lubricated bearings of wood, lignum vitae, rubber, or special composition are sometimes preferred, particularly for small and medium-size propeller-type turbines where the bearing is located at the bottom of the head cover cone, where the packing box and the oil-lubricated bearing would be inaccessible, and where it would be difficult to avoid water contamination. With the water-lubricated bearing, the packing box is placed above the bearing.

Spiral Case The spiral case must be proportioned so as to cause relatively low friction losses, as well as to prevent eddying which would travel into the runner and affect its efficiency. The cases generally are made of (1) metal: cast steel, cast iron, or steel plate, and (2) concrete. Metal cases are made as complete spirals, customarily with uniform or slightly increasing velocity from the throat to the small end. When a valve is used at the case entrance, the net area at the centerline of the valve should be at least equal to the case entrance area.

Concrete cases may be complete spirals or semispirals, rectangular or oval in cross section. There should be no piers close to the turbine.

Rated load velocities for general practice for cases of various types are shown in Fig. 9.9.10. Higher velocities are sometimes used with the larger units to reduce size and cost.

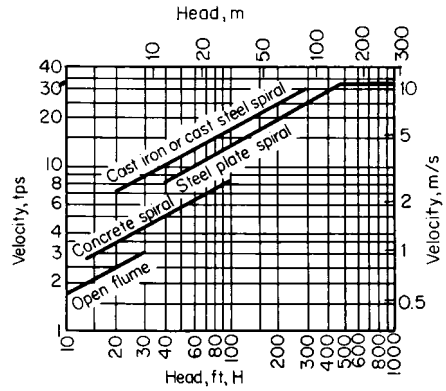


Fig. 9.9.10 Case velocities.

Stay Ring That part of the guide apparatus between the spiral case and the wicket gates, containing stationary stay vanes, is called the stay ring. The water is accelerated within this space as it approaches the gates. The number of stay vanes employed is usually made equal to or one-half the number of gates. They are placed at the angle which will cause the least obstruction to the flow.

The stay ring is cast integral with cast cases and is made separately of welded or cast steel for concrete or steel plate spiral cases. It should be a continuous ring to facilitate erection, and very rigid, because it serves as a foundation for the rest of the turbine and generator.

Wicket Gates and Operating Mechanism Wicket gates control the power and speed of the turbine. The number of gates m ranges from 12 for small units to 28 for large units. The overall dimensions of the turbine decrease as the number of gates increases.

To prevent interference between the gates and the runner buckets, which may cause noise and vibration, the discharge tips of the fully open gates should be kept well away from the inlet edges of the runner buckets of Francis turbines, the radial clearance being large enough to prevent the gate tips from overhanging the curved part of the discharge ring. Gate tip diameter = 1.1 (maximum inlet diameter D_r).

The height and angular movement of the gates increase with the specific speed. The angular movement varies from 15° for low specific speed to 80° for high specific speed.

Most wicket gates are made of cast steel; a few, for higher heads, are made of stainless steel; some, for lower heads, are weldments built from rolled materials and castings.

Each gate connection to the operating ring should be provided with a breaking element to protect the gate and other mechanism in case of an obstruction. Because of the inherent instability of a free gate and its tendency to flutter, it is advisable to install gate-restraining devices to hold the gate if a breaking element fails. This device also avoids potential resonant flutter with the natural frequencies in the system. Each gate

should also be provided with stops to prevent it from striking the runner or reversing after the breaking element fails.

The capacity G, ft·lb (J), of the oil pressure cylinder or cylinders (sometimes known as governor capacity) may be computed from the formula

$$G = 18Pn_s^{1/4}/\sqrt{H} \qquad (13Pn_s^{1/4}/\sqrt{H})$$

Some units now provide individual servomotors on each wicket gate. This synchronized system allows the hydraulic control system to dampen a tendency to gate flutter.

One or more vacuum breakers or air valves are installed in the head cover to admit air to the runner or draft tube, to improve efficiency at low gate openings or to alleviate draft tube vortex cavitation. An air valve is also necessary on propeller-type units to break the vacuum under the head cover and to help prevent the runner from ''screwing up'' in the water when the gates are suddenly closed. The air valve is piped to the outside of the powerhouse above the floodwater level.

Draft Tubes The draft tube may serve the double purpose of (1) allowing the turbine to be set above tailwater level, without loss of head, to facilitate inspection and maintenance, and (2) regaining, by diffuser action, the major portion of the kinetic energy delivered to it from the runner.

At rated load the velocity at the upstream end of the tube for modern units ranges from 24 to 40 ft/s (7 to 12 m/s), representing from 9 to 25 ft (3 to 8 m) head. As the specific speed is increased and the head reduced, it becomes increasingly more important to have an efficient draft tube. Good practice limits the velocity at the discharge end of the tube to 5 to 10 ft/s (2 to 3 m/s), representing less than 1 ft (0.3 m) velocity head loss.

The elbow type of tube is now used with most vertical turbine installations. With this type the vertical portion begins with a conical section which gradually flattens in the elbow section and then discharges horizontally through substantially rectangular sections to the tailrace. Most of the regain of energy takes place in the vertical portion, very little in the elbow section which is shaped to deliver the water to the horizontal portion so that the regain may be efficiently completed. Figure 9.9.11 shows proportions of a good elbow tube. One or two vertical piers are placed in the horizontal portion of the tube for structural reasons.

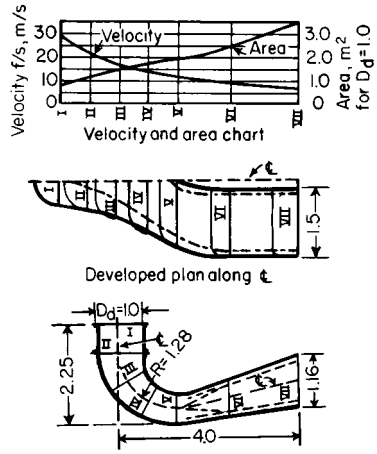


Fig. 9.9.11 Elbow draft tube layout.

Most tubes are made of concrete with a steel plate lining extending from the upper end to a point where the velocity has been sufficiently reduced [say 20 ft/s (7 m/s)] to prevent erosion of the concrete. Pier noses are also lined where necessary to prevent erosion and for structural reasons.

IMPULSE TURBINES

Impulse turbines are utilized when the head is too high for the practical use of Francis turbines, which is normally a head exceeding 1,600 ft (500 m). Impulse turbines are also sometimes used for heads below 1,600 ft (500 m) where excessive erosion due to foreign materials in the water presents a problem. The main disadvantage of impulse turbines, especially for low heads, is their low specific speed. In the past, this was overcome on conventional horizontal shaft units by the use of two runners or two jets per runner. In recent years, the vertical shaft multijet impulse turbine (Fig. 9.9.12) has become popular.

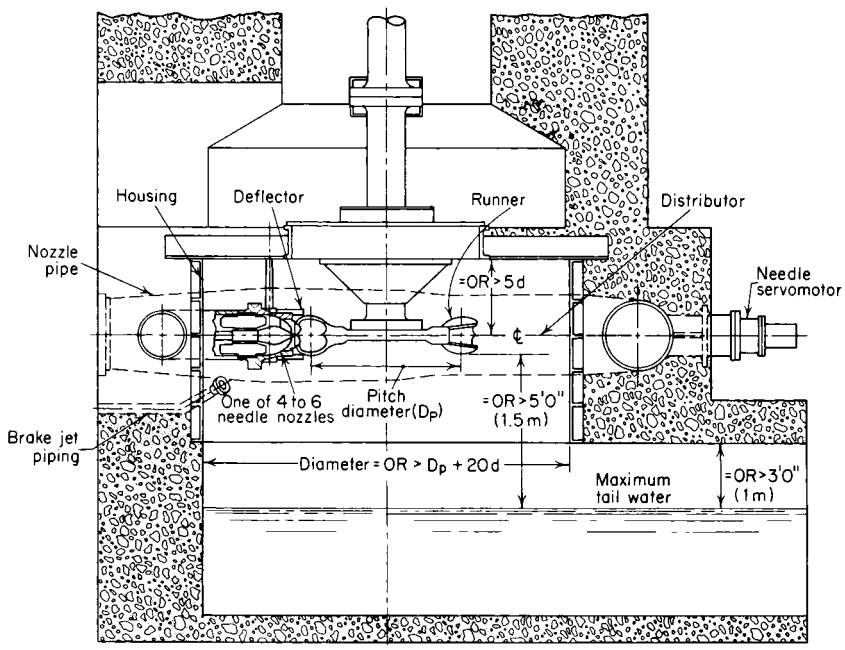


Fig. 9.9.12 Vertical-shaft multijet impulse turbine installation.

The efficiency obtainable from a horizontal shaft impulse turbine is about 90 percent. Field tests on multijet vertical units have shown efficiencies as high as 91.5 percent. The use of multiple jets on the vertical units reduces the percentage loss due to windage of the runner. The unit can be operated with a reduced number of jets at part load, thus increasing part load efficiency. Six jets are about the maximum number that can be used on one runner without jet interference.

Selection of Speed The first consideration is the selection of suitable n_s. As with reaction turbines, there is a relation which determines approximately the limiting value of specific speed n_s (per jet) for any head. For heads around 1,000 ft (300 m), $n_s = 5.0$ to 5.5 (19 to 21) gives high efficiency. For heads around 2,000 ft (600 m), maximum efficiency is attained near $n_s = 4.0$ to 5.0 (15 to 19). Care has to be exercised in selecting an n_s (per jet), especially for the higher heads, so that the proper number of buckets can be used on the wheel disk. The higher the n_s, the fewer the number of buckets required (Fig. 9.9.13) but the smaller the pitch circle. The latter can create stress problems in the attachment of the buckets to the disk. If the resulting speed n is quite low for the power to be developed, the n_s of the unit and consequently the speed n can be increased by increasing the number of runners or the number of jets per runner. The n_s of the unit will then be n_s (per jet) times the square root of the number of jets.

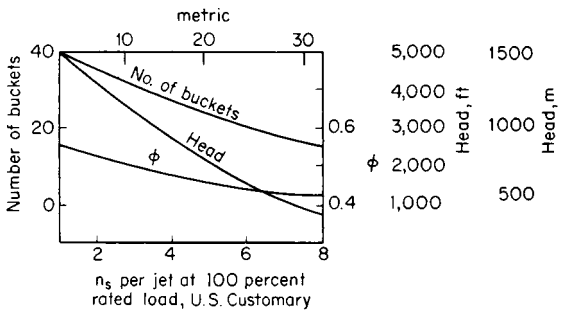


Fig. 9.9.13 Impulse turbine characteristics and proportions.

Basic Dimensions The pitch diameter D_p of the runner is twice the distance from the center of the runner to the axial centerline of the jet and is determined by the value of the coefficient ϕ selected, $D_p = 1{,}840\phi\sqrt{H}/n$ $(84{,}578\phi\sqrt{H}/n)$. The variation of ϕ with n_s is established by experience and model tests (Fig. 9.9.13).

The quantity of water Q discharged per jet is $550P/(wHe)$ $[101.93P/(wHe)]$, where P is the horsepower (kilowatts) developed by each jet.

The diameter of the jet is $4.85\sqrt{Q}/H^{1/4}$ in $(0.55\sqrt{Q}/H^{1/4}$ m), using a velocity coefficient of 0.97 for the jet.

The **velocity** in the inlet should not exceed $0.12\sqrt{2gH}$ (approximately), and it is considered good practice not to exceed an absolute value of 40 ft/s (12 m/s).

The **valve** employed in the inlet pipe should be of a type which, when open, permits a smooth, uninterrupted flow, free of eddies or obstructions. Modern preference is for rotary sphere valves for this purpose.

The **housing** serves primarily to carry off the discharged water to the tail pit below. On horizontal shaft units, at the place where the runner receives the discharge from the jet, the housing should be about 10 to 12 times the jet diameter. At the place where the runner has been cleared of the discharge, the housing should be as narrow as possible to decrease windage. Suitable baffling should be used to carry the discharge away from the buckets. The housing should be vented adequately near the center of the runner.

For vertical-shaft units, the housing should be of ample size to prevent discharge water from interfering with the buckets. The housing should also be vented adequately near the center of the runner.

The **setting** of the horizontal-shaft unit should be such that the lower edges of the buckets are at least 3 ft (1 m) above maximum tailwater elevation. For vertical shaft units, this distance should be at least 5 ft (1.5 m). Impulse turbine discharge entrains large amounts of air, and unless this is completely replaced, a vacuum will be produced in the housing which will draw the tailwater up and drown out the runner. Thus the roof of the discharge tunnel or passageway, for both horizontal- and vertical-shaft units, should be at least 3 ft (1 m) above the maximum operating tailwater elevation to permit the free circulation of air to the runner (Fig. 9.9.12).

In cases where extremely high tailwater occurs for a short period, compressed air can be used to depress the tailwater.

Experience has shown that the ft^3/s (m^3/s) of air (at the discharge pressure of the compressor) required to maintain the depressed tailwater is about 15 percent of the ft/3/s (m^3/s) of water (Q) discharged by the turbine.

Runner The force F exerted by a stream upon a stationary bucket (Fig. 9.9.14) is

$$F = MV(1 - \cos\theta) = wQV(1 - \cos\theta)/g$$

where M = mass per s and θ = angle in deg through which the water is turned relative to the bucket (see section *Y-Y*, Fig. 9.9.14). If the bucket is moving in the direction of the stream with a velocity u, $F = wQ(V - u)(1 - \cos\theta)/g$. The work done by the stream $= Fu = wQn(V - u)(1 - \cos\theta)/g$. When θ is greater than 90°, the cosine becomes negative. Thus the closer the angle θ is to 180°, the greater the amount of work done with the same Q and V. However, in order to prevent the water from striking the succeeding bucket, θ should never be 180° (see section *Y-Y*, Fig. 9.9.14). Also, since the water discharged from the bucket must have some velocity, the value of u is made somewhat less than $V/2$.

The angle β_1 at the centerline of the bucket should be calculated from the velocity triangle (Fig. 9.9.14). The value of β_1 is greatest as the entrance lip E enters the jet. The angle of the underside of the bucket should be greater than β_1 to avoid pitting.

In terms of the jet diameter which will give maximum guaranteed or rated capacity, the approximate proportions of the buckets are $B = 3d$, $L = 2.6d$, $D = 0.85d$ (Fig. 9.9.14). The contour of the entrance lip E is important, since the manner in which this lip comes in contact with the jet affects the path of the water passing to the preceding bucket. The size of the bucket with relation to the jet diameter also determines the percentage of full load at which maximum efficiency occurs. The larger the bucket, the higher this percentage of load.

The number of buckets on a runner should be such that no water can pass through the buckets without being deflected by the buckets. Figure 9.9.13 indicated approximately the most efficient number.

The **needle nozzle** should be placed as close to the buckets as possible, as the jet tends to lose its compactness of form shortly after emerging from the nozzle. The needle and the seat of the nozzle should be designed for easy replacement and should be of a material highly resistant to erosion. The needle should terminate in a cone of 30 to 45°. The nozzle tip should have a subtended angle of 45 to 60° and a diameter at discharge about 20 percent greater than the calculated diameter of the jet. The maximum diameter of the needle should be about 15 percent greater than the discharge diameter of the nozzle tip. The diameter of the upstream portion of the nozzle should be such that the velocity does not exceed $0.10\sqrt{2gH}$. (See Lowy, Efficiency Analysis of Pelton Wheels, *Trans. ASME*, Aug. 1944, for additional information on the design of impulse turbines.)

Regulation The inertia of the water flowing through the long penstocks usually employed with impulse turbines prohibits rapid reduction in velocity because of the pressure rise which would occur. Therefore, to minimize the speed rise following a sudden load rejection, it is necessary to reduce the hydraulic power delivered to the runner without changing the flow in the penstock too rapidly. This is usually accomplished by placing a governor controlled jet deflector between the needle nozzle and the runner. The governor moves this deflector rapidly into the jet, cutting off the load. It is not unusual for the deflector to cut off the entire jet in 1½ s. Since the deflector acts on the jet after it leaves the nozzle, there is no change of flow in the penstock; hence, there is no pressure rise. The governor then moves the needle at a permissible rate (in terms of pressure rise), with simultaneous automatic withdrawal of the deflector. The jet is finally reduced the necessary amount to correspond to the reduced load. The needle must also move slowly in the

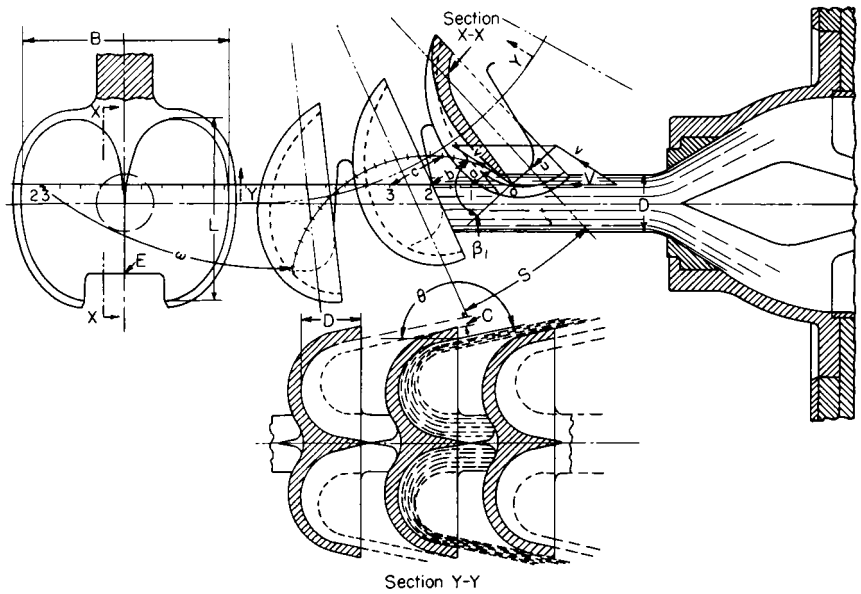


Fig. 9.9.14 Impulse turbine bucket design.

opening direction for oncoming loads to avoid penstock collapse because of large pressure drops.

Runaway Speed The runaway speed for impulse turbines ranges from 160 to 190 percent of normal speed, depending upon the specific speed of the runner; the higher the specific speed, the higher the runaway speed.

REVERSIBLE PUMP/TURBINES

In this type of project, surplus low-value energy from either hydro or thermal plants is used to pump water during off-peak periods to an elevated reservoir, where it becomes available for generating high-value peaking energy. While separate pumps and hydraulic turbines of conventional design can and have been used for this purpose, the development of single reversible pump/turbines has made many pumped storage projects economically feasible.

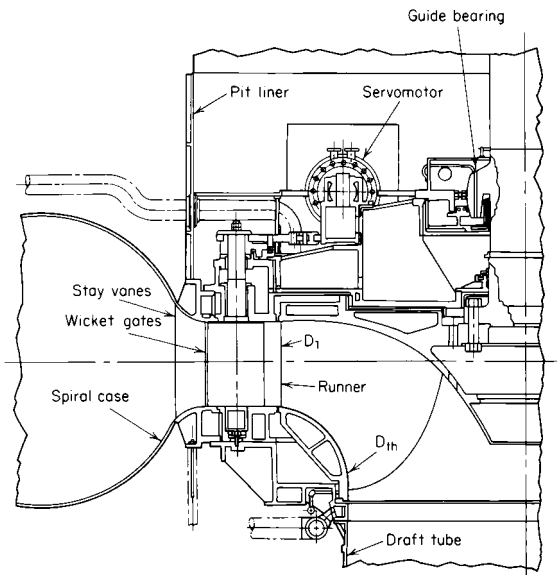


Fig. 9.9.15 Reversible pump turbine.

Figure 9.9.15 is a cross section of a reversible pump/turbine, showing that, with the exception of the runner, it is essentially a conventional turbine. The wicket gates must be designed for flow in both directions. A few units have been built without movable wicket gates. The runner is

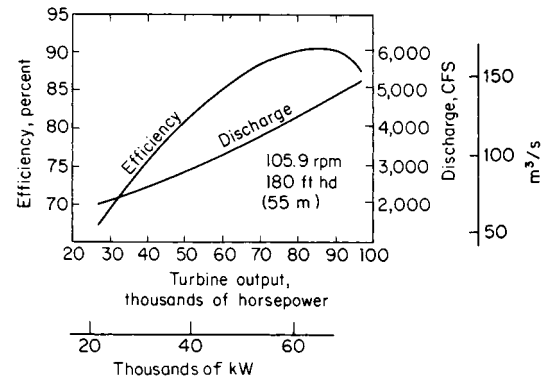


Fig. 9.9.16 Generating.

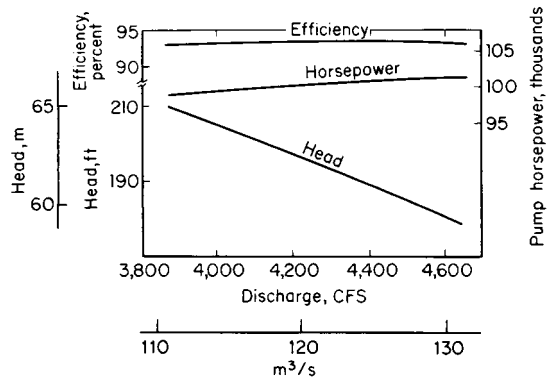


Fig. 9.9.17 Pumping.

Figs. 9.9.16 and 9.9.17 Typical performance curves for a pump turbine.

Table 9.9.2 Reversible Pump/Turbine Installations

	Generating				Pumping				
	Power		Head		Capacity		Head		
Plant	1,000 hp	MW	ft	m	ft³/s	m³/s	ft	m	Year in operation
Pedreira ''A''	5.2	3.9	49	15	690	19.5	49	15	1939
Hiwassee	62	46	190	58	3,900	111	205	62.5	1956
Taum Sauk	220	165	790	241	2,650	75.0	765	233	1963
Cruachan	100	74.6	1,130	343	1,010	28.6	1,175	358	1966
Cabin Creek	166	125	1,190	363	1,375	39.0	1,060	323	1967
Villarino	135	100	1,250	382	1,020	29	1,325	404	1969
Kisenyami	240	180	755	220	3,880	110	645	197	1969
Drakensburg	360	269	1,380	422	1,685	42	1,443	440	1977
Dinorwic	405	302	1,682	513	1,647	46.7	1,771	540	1975
Bajina Basta	422	385	1,968	600	1,520	43	2,132	650	1976
Raccoon Mt.	465	350	940	287	3,850	110	1,000	305	1974
Bath Co.	510	380	1,080	329	4,100	116	1,100	335	1983
Palmiet	335	250	988	301	2,012	57	1,000	305	1988
La Muela	284	212	1,725	525	1,165	33	1,637	499	1990
Bad Creek	482	360	1,200	367	3,531	100	1,240	378	1991
Mingtan	368	275	1,322	403	2,896	82	1,345	410	1992

essentially a pump runner modified for optimum performance while generating power. (See Sec. 14 for the design of centrifugal pumps and their characteristics.) Conventional turbine runners, because of their short blades, are not suited for the pumping cavitation requirements.

The reversible pump/turbines have certain fundamental performance characteristics which are inherent in the design. The relationship between pumping and generating performance for a given specific speed is more or less fixed and can be modified only to a minor degree by alterations in the design. Figure 9.9.16 shows the expected generating performance, and Fig. 9.9.17 shows the expected pumping performance based on model tests of a reversible pump/turbine with a specific speed $n_s = 47.3$ (180) generating and $n_s = 2{,}600$ (50) pumping (see ''Centrifugal and Axial Pumps'' for definition of pump specific speed).

The best efficiency, with reversible pump/turbines, occurs at a lower speed when generating than when pumping. This can be compensated for by using a generator/motor capable of operating at two speeds with a constant frequency. Several units of this type have been built, but since two-speed generator/motors cost considerably more than those built for single speed, a careful study should be made of the advantages to be obtained in operating at two speeds.

Single reversible pump/turbines can be built for any heads up to 2,000 ft (600 m). Beyond this, either multiple-stage reversible units or separate pumps and turbines should be used. Some of the more notable reversible pump/turbine installations are shown in Table 9.9.2.

Runaway Speed The runaway speed for pump/turbines is considerably lower than that for conventional turbines. It ranges from 150 percent of normal for low specific speed runners to 175 percent for high specific speed runners.

MODEL TESTS

Model tests serve several purposes. They are primarily used to check turbine runner, wicket gate, draft tube, casing, and (sometimes) inlet works designs for optimum performance. Correctly interpreted, they may also be used as a reliable indication of the performance of the units in the field. In many cases, purchasers specify the performance of a homologous model test, which is used as an acceptance test of the unit in lieu of field tests. In such cases, the field conditions, particularly the casing and draft tube, must be reproduced faithfully. Model tests should be run in accordance with the International Code for Model Acceptance Tests of Hydraulic Turbines, Publication 193; International Code for Model Acceptance Tests of Storage Pumps, Publication 497 of the International Electrotechnical Commission (IEC); and International Standard Publication, IEC 995, that supplements IEC 193 and 497.

Figure 9.9.18 shows typical model test results of a reaction-type turbine, in which the power P at 1 ft (0.3 m) head and the efficiency are each plotted against the speed n at 1 ft (0.3 m) head, for each of several gate openings.

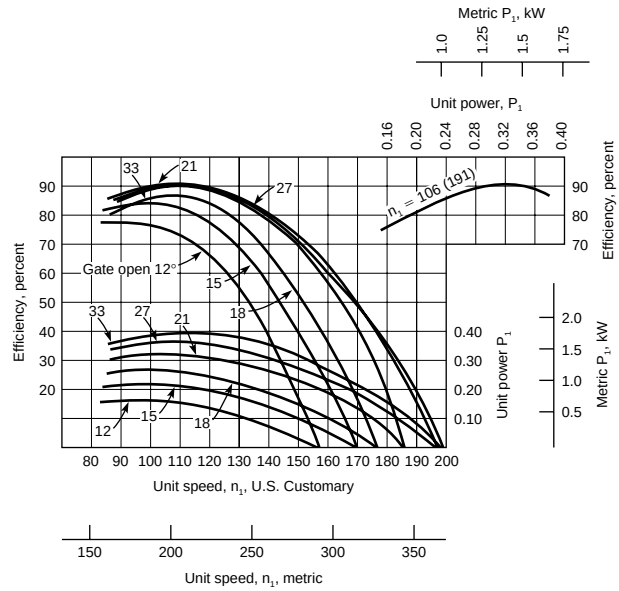


Fig. 9.9.18 Model test curves for a Francis runner ($D_r = D_{th} = 16.2$ in (0.41 m) and n_s = about 65 (250).

Laws of Proportionality for Homologous Turbines The laws of proportionality shown previously are used to calculate from model tests the power, speed, and discharges of a homologous turbine of different diameter under a different head. The field unit will have a somewhat higher efficiency and power owing to proportionally smaller frictional and bearing losses. Expected field efficiency is customarily computed from the model efficiency by the Moody formula

$$c' = 1 - \frac{2}{3}\left[(1 - e)\left(\frac{D}{D'}\right)^{1/5}\right]$$

in which the prime letters refer to the field installation. The step-up efficiency is usually computed for the point of best efficiency only, and the corresponding differential is applied as a constant value from, say, half load to full load.

CAVITATION

Cavitation occurs when the pressure at any point in the flowing water drops below the vapor pressure of the water.

The relationship which produces cavitation is between vapor pressure, barometric pressure, setting of the runner with respect to tailwater, and net effective head on the turbine and is expressed by the **Thoma cavitation coefficient** $\sigma = (H_b - H_v - H_s)/H$, where H_b = barometric head, ft (m) of water; H_v = vapor pressure of water, abs; H_s = elevation, ft (m) of the runner above tailwater, measured at the centerline of the distributor of a Francis runner and at the centerline of the blades of a propeller runner (if the runner is submerged, H_s becomes negative); and H = total or net effective head, ft (m), on the turbine. The setting of the runner depends upon the value of σ, which varies with specific speed n_s of the runner and the individual characteristics of a particular runner design. In practice, the model of the proposed runner is first tested with relatively high back pressure (H, small or negative). Then the back pressure is reduced in increments until the breaking point, as indicated by a drop in power, efficiency, and discharge, is reached. This breaking point is designated as the critical σ and will vary with gate opening and speed and, on propeller turbines, with blade angle. Consequently, σ must be determined for a range of limiting conditions.

In the absence of cavitation tests, the value of σ should not be lower than $\sigma = n_s^{3/2}/2{,}000$ ($n_s^{3/2}/15{,}000$) for Francis and propeller runners and $\sigma = n_s^2/25{,}000$ ($n_s^2/350{,}000$) for adjustable-blade propeller runners.

The value of σ at which a plant operates, depending largely upon the setting of the runner with respect to tailwater, is called the **plant** σ. To avoid excessive cavitation, the plant σ should exceed the critical σ. The greater this margin, the less possibility of cavitation during operation. For a general discussion of cavitation phenomena, see Knapp, Recent Investigations of the Mechanics of Cavitation and Cavitation Damage, *Trans. ASME,* Oct. 1955. Laboratory tests and experience have shown that materials having a high resistance to cavitation erosion (pitting) and suitable for use in hydraulic turbines are the stainless steels and aluminum bronzes, especially when used as welding overlays. (See Rheingans, ''Resistance of Various Materials to Cavitation Damage,'' ASME Report of 1956 Cavitation Symposium.)

SPEED REGULATION

(See also Sec. 16.)

Regulation is accomplished by changing the flow of water to the turbine. The flow is controlled by the wicket gates of reaction turbines and by the needle valve or jet deflector of impulse turbines. The governor moves the gates or needle in response to speed changes resulting from load or head changes.

A schematic diagram of a classic governor is shown in Fig. 9.9.19. The parts consist of a speed-responsive device, a power element that changes the gates or needle position, and a follow-up or compensating device that prevents hunting.

The **speed-responsive device** originally was a pair of spring-loaded flyballs mounted directly on the turbine shaft, or driven from the shaft by belt or gears, or driven by an electric motor that receives its power either from the bus line or from an independent generator driven from the main turbine generator shaft.

The **power element** consists of oil-operated power cylinders or servomotors which operate the turbine gates or needle. Oil pumps and a pressure tank or accumulator maintain a supply of oil. A valve operated by the speed sensor controls the flow of oil to the servomotors or acts as a pilot valve controlling a larger relay valve, which in turn controls the oil to the servomotors. The pump capacity is usually 3 servomotor volumes per minute. The capacity of the pressure tank is generally made 20 times the servomotor volume, allowing for 8 volumes of oil and 12 volumes of air. The velocity of oil in the pipelines is kept below 15 ft/s (5 m/s). When using individual servomotors on each gate, these values are usually increased 50 percent to balance the usual damping effect of the common massive two-servomotor system.

The **follow-up** or **compensating device** connects the power piston of the servomotor to the control valve, usually through a dashpot, and causes the motion of gates or needle to stop when they have moved sufficiently to compensate for the load change.

The time for a full stroke or traverse of the governor is controlled by the rate of flow of oil to the servomotors; most governors have provisions for varying this time. The gate opening changes at a uniform rate

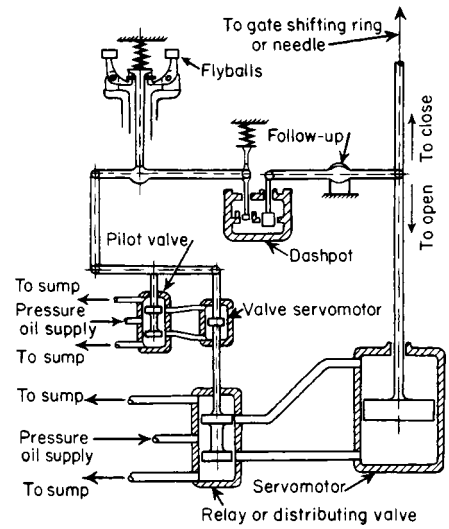


Fig. 9.9.19 Schematic diagram of governor.

over the major portion of the stroke and at a somewhat slower rate at the ends of the stroke. The governor *dead time,* or the elapsed time from the initial speed change to the first movement of the gates, is usually less than 0.2 s.

Electric Governor In recent years, the functional requirements placed upon the hydraulic-turbine governor have increased to the point where electrical control of the hydraulic turbine is attractive in view of the simplicity with which electrical signals can be manipulated. The basic elements of an electric-hydraulic governing system are (1) permanent magnet generator (PMG) or equivalent for measurement of turbine speed and the means for transmittal of such speed signals to the electrical portion of the governor; (2) an electric circuit sensitive to speed variations about some adjustable reference point; (3) amplifying circuits to convert speed reference changes, speed error signals, and auxiliary signals into a useful electric current; (4) an electrohydraulic transducer to transform the electric current into a hydraulic output signal; (5) hydraulic amplifying equipment to deliver suitable power and the desired signal to the gate servomotors as a function of the output of the electrohydraulic transducer; (6) power supplies for the electric and hydraulic portions of the control. For further particulars of the electric governor, see Leum, Electric Governors for Hydro Turbines, ASME Paper 62-WA165.

Electronic Governors With the advent of minicomputers, microprocessors, and programmable controllers, many of the traditional mechanical functions in the hydraulic turbine governor have been replaced electronically. While the basic elements of the governor remain the same, the use of electronics in the feedback sensing and control circuits has greatly reduced the size and mechanical complexity of these units. The basic elements of the modern electronic governoring system are: (1) Inductive pickups or permanent magnetic generators or equivalent to measure the turbine speed and to transmit the information to the processor. (2) Electronic circuitry to establish the speed-control function of the governor (integrated circuits or digital programming). Basically the circuitry consists of a proportional, an integrator, and a derivative function. (3) A servo valve or equivalent to translate the electronic signal from the processor to a hydraulic signal. (4) Where required, a means to

hydraulically amplify the output from the servo valve so that a suitable power level can be delivered to the gate and blade servomotors. (5) Auxiliary equipment as required in the form of power supplies, etc. to support the system.

Speed Regulation Requirements Usually, a sufficient measure of the regulation provided is the maximum speed rise resulting from sudden rejection of full load, as from the breaker tripping. A maximum speed rise of 30 percent of normal speed for this condition is a common limitation.

Speed Rise Following Load Reduction For sudden load reductions, the approximate speed rise is

$$n_t/n = [1 + 1{,}620{,}000T_tP_t(1 + h/H)^{3/2}/WR^2n^2]^{1/2} \quad \text{(USCS)}$$
$$= [1 + 365{,}000T_tP_t(1 + h/H)^{3/2}/GD^2n^2]^{1/2} \quad \text{(SI)}$$

where n_t is the speed, rpm, at the end of time T_t; n is the speed, rpm, before the load decrease; T_t is the time interval, seconds, for the governor to adjust the flow to the new load; P_t is the reduction in load, hp (kW); h is the head rise caused by the retardation of the flow, ft (m); H is the net effective head before the load change, ft (m); WR^2 is the product of the revolving parts weight, lb, and the square of their radius of gyration, ft; GD^2 is the product of the revolving parts weight, kg, and the square of their diameter of gyration, m. For approximate values of h, see below. Very rapid gate closure produces a reduction of pressure in the draft tube and the possibility of breaking the water column, with subsequent violent resurge which may damage the turbines.

Speed Drop Following Load Increase For sudden load increases, the approximate speed drop is

$$n_t/n = [1 - 1{,}620{,}000T_tP_t/WR^2n^2(1 - h/H)^{3/2}]^{1/2} \quad \text{(USCS)}$$
$$= [1 - 365{,}000T_tP_t/GD^2n^2(1 - h/H)^{3/2}]^{1/2} \quad \text{(SI)}$$

where P_t is the actual load increase and h is the head drop caused by the increase of the flow. If the speed drop is to be determined for a given increase in gate opening, the governor time T_t for making this increase and the normal change in load for the change in gate opening, under constant head H, can be used in the following formula:

$$n_t/n = [1 - 1{,}620{,}000T_tP_t(1 - h/H)^{3/2}/WR^2n^2]^{1/2} \quad \text{(USCS)}$$
$$= [1 - 365{,}000T_tP_t(1 - h/H)^{3/2}/GD^2n^2]^{1/2} \quad \text{(SI)}$$

The actual change in load, however, will be $P_t(1 - h/H)^{3/2}$.

For derivation of the above speed variation formulas and for a more accurate determination, see Strowger and Kerr, Speed Changes of Hydraulic Turbines for Sudden Changes of Load, *Trans. ASME,* 1926; and Rich, ''Hydraulic Transients,'' McGraw-Hill.

Water Hammer in Penstocks If a gate movement is considered as a series of instantaneous movements with a very small interval between each movement, the pressure variation in the penstock following the gate movement will be the effect of a series of pressure waves, each caused by one of the instantaneous small gate movements. For a steel penstock, the velocity of the pressure wave $a = 4{,}660/\sqrt{1 + d/100t}$ $(1{,}420/\sqrt{1 + d/100t})$, where d is the penstock diameter, in (m), and t is the penstock wall thickness, in (m). The pressure change at any point along the penstock at any time after the start of the gate movement may be calculated by summing up the effect of the individual pressure waves. See ''Symposium on Water Hammer,'' ASME, 1933; and Parmakian, ''Water Hammer Analysis,'' Dover.

Approximate formulas (De Sparre) for the increase in pressure h, ft (m), following gate closure, are given below. They are quite accurate for pressure rises not exceeding 50 percent of the initial pressure, which includes most practical cases.

$h = aV/g$	for $K < 1$, $N < 1$
$h = aV/\{g[N + K(N - 1)]\}$	for $K < 1$, $N > 1$
$h = aV/[g(2N - K)]$	for $K > 1$, $N > 1$

where $K = aV/(2gH)$; $N = aT/(2L)$. V and H are the penstock velocity, ft/s (m/s), and head, ft (m), prior to closure; L is the penstock length, ft (m), and T is the time of gate closure. For full load rejection, T may be taken as 85 percent of the total gate traversing time to allow for nonuniform gate motion.

For pressure drop following a complete gate opening, the following formula (S. Logan Kerr) may be used with T not less than $2L/a$:

$$h = \frac{aV}{g}\left(\frac{-K + \sqrt{K^2 + N^2}}{N^2}\right) = \text{pressure drop, ft (m)}$$

Pressure variations exceeding 40 percent rise and about 25 percent drop should be avoided. When the control, directly by the governor, causes undesirable pressure variations, a surge tank, a pressure regulator, or a jet deflector may be used. A **surge tank** is a standpipe with an atmospheric tank, attached to the penstock as close as possible to the casing inlet. The tank provides a reservoir and expansion chamber for the water demand or the water rejection following sudden gate movements, so that sudden accelerations or decelerations of the flow in the penstock are avoided.

Pressure regulators may be of either the water-wasting or water-saving type. The **water-wasting type** is a synchronous bypass, generally attached to the turbine casing. It is operated directly from the governor, or the gate mechanism of the turbine, and wastes such an amount as to keep the total water discharge equal at all times to the full-load discharge of the turbine. The bypass is a needle nozzle or a mushroom-shaped disk valve or a cone valve which opens and is partly balanced hydraulically by a piston under pipeline pressure. The **water-saving type** permits the regulator to open upon rapid closure of the turbine gates, and then close slowly, so that the total water discharge is gradually reduced and finally limited to that through the turbine, adjusted for the new load.

AUXILIARIES

Valves and head gates are provided for shutting off the water to each turbine for safety, for ease of maintenance, and to reduce water leakage losses. Motor-operated steel head gates are generally used for low- and medium-head plants with concrete scroll cases, although a few still use stop logs. The butterfly type of valve placed close to the turbine casing is suitable for medium- and high-head units with metal casings of circular inlet diameter. Butterfly valves of 8-ft (2.5 m) diameter for 1,000-ft (305 m) head and 27-ft (8.2 m) diameter for 100-ft (30 m) head have been built. The new flow through valve or biplane valve is becoming more popular for this application. In recent years, rotary sphere valves have replaced gate valves for high heads and where the loss through the butterfly valve is excessive because of the obstruction to flow by the valve disk.

COMPUTER-AIDED DESIGN

The continued development, refinement, and use of proven high-technology computer-aided design and analysis tools help create new generations of designs. Design objectives today are to produce equipment which has characteristics of reliability, easy servicing, operational flexibility, and high performance. The success of this approach is made possible by integrating design history and field experience into hydraulic, mechanical, and operational computer design and analysis systems.

TURBINE TESTS

Field testing of hydraulic turbines to determine the absolute efficiency and output involves careful and accurate measurement of the power available in the water supplied to the turbine [water hp (kW)] and the turbine output [developed hp (kW)] e = developed hp (kW)/water hp (kW) = $550P/(wQH)$ $[101.93P/(QHw)]$. The tests should be conducted in accordance with the International Code for Field Acceptance Tests of Hydraulic Turbines and/or Storage Pumps, IEC Publications 41 and 198.

Because of the difficulties and costs involved in making accurate measurements of horsepower, net head, and discharge in the field, there has been a trend in recent years to dispense with the field test, especially where a laboratory test on a homologous model turbine is available. Instead, an index test is made on the unit in the field, which measures the turbine output and relative discharge under various conditions. Index tests should be conducted in accordance with the International Code for Field Acceptance Tests of Hydraulic Turbines, IEC Publication 41.