angle for the valve opening will be one-half the valve range of operation, the latter being expressed in degrees of crankshaft rotation.

One of the means used to increase the maximum power of an engine is to increase its volumetric efficiency. The volumetric efficiency is usually defined as the ratio, in percent, of the volume of air at standard atmospheric conditions entering the cylinder of an engine during its intake stroke to the volume displaced by the piston. The volumetric efficiency and the power of an engine can be increased, within limits, by increasing the valve lift.

Combustion Chamber Design

The size and shape of the combustion chamber has a decided effect upon the proper mixing of the fuel and air, and also upon the fuel detonation. In general, the combustion chamber is designed to create turbulence of the mixture for better combustion and greater flame propagation velocity, thereby decreasing the combustion time and improving antiknock characteristics. A short and compact combustion chamber generally requires less time for combustion and improves the antiknock characteristics.

Effect of Compression Ratio

The thermal efficiency of an engine does not approach the theoretical efficiency of an Otto cycle engine as defined by

$$\text{Eff.} = 1 - \frac{1}{r^{n-1}} \tag{2}$$

There are many reasons why the actual efficiency is substantially below the theoretical efficiency. Some of the discrepancy can be accounted for by the following factors:

1. The expansion and compression of gases is not adiabatic.
2. There are friction losses in the engine.
3. Work is required to draw in and exhaust gases.
4. For a free-breathing engine (i.e., one not supercharged) the volumetric efficiency will be less than 100 percent.
5. Complete combustion does not occur.

Nevertheless, there is a definite relationship between the compression ratio r, and the specific fuel consumption of an engine. For one LP gas engine the effect of the compression ratio upon efficiency is shown clearly in figure 5-17. Large changes in the compression ratio of diesel engines are not practo be equivalent to the system with two concentrated masses, m_{c1} and m_{c2}, located at distances a and b apart from the center of gravity G as shown in figure 5-19(b). For the system (b) to be dynamically equivalent to the system (a) in figure 5-19, the following conditions must be satisfied:

1. Total masses for the two systems are equal.
2. Locations of the centers of gravity are the same.
3. Moments of inertia for both systems about the center of gravity are equal.

From the conditions 1 and 2 the following expressions are derived, noting that $a + b = l$.

$$m_c = m_{c1} + m_{c2}$$

$$m_c b = m_{c1} l \qquad \text{or} \qquad m_{c1} = \frac{b}{l} m_c$$

$$m_c a = m_{c2} l \qquad \text{or} \qquad m_{c2} = \frac{a}{l} m_c$$

Thus the mass m_c of the connecting rod can be divided into two masses, m_{c1} and m_{c2}, located at distances a and b, respectively, from the center of gravity. For condition 3 the moment of inertia I for the equivalent system is expressed in the form

$$I_b = m_{c1}a^2 + m_{c2}b^2$$

If I_a is the moment of inertia of the original system [fig. 5-19(a)], the inequality $I_a < I_b$ holds in general. To compensate, a ring of radius R and mass Δm will be placed about the center of gravity as shown in figure 5-19(c). Then the moment of inertia ΔI of the ring about G is expressed as

$$\Delta I = \Delta m R^2$$

Hence ΔI can be determined to satisfy

$$I_a = I_b - \Delta I$$

Addition of ΔI increases the total mass of the system by Δm and results in condition 1 not being satisfied. However, this problem can be practically solved by taking the ring radius R to be large enough so that Δm can be made small enough to be negligible. Then, considering the inertia force of the connecting rod, only the inertia forces of the reciprocating mass m_{c1} and the rotating mass m_{c2} are taken into account, neglecting Δm. For the inertia couples about the center of gravity, the inertia couple due to ΔI must be considered in addition to the couples due to m_{c1} and m_{c2}.

chamber in the head or the top of the piston located between the auxiliary chamber and the injection nozzle.

The starting characteristics of a diesel engine using an auxiliary chamber are good since the fuel is first injected through the main chamber where the air is hottest. The precision and care of the injection nozzle must be better than for the other types of combustion chambers, because the fuel must be directed accurately into a small opening that is several centimeters away from the nozzle.

The duration of combustion is longer for this type of chamber, resulting in lower peak pressures at the expense of slightly higher fuel consumption.

The maximum pressure occurring in the combustion chamber of the four designs previously mentioned is graphically illustrated in figure 5-29. The peak pressure is the highest for the open-chamber design.

Figure 5-30 illustrates some of the care required in the design of diesel combustion chambers. Note that a 20 percent variation in specific fuel consumption results from changing the direction of the injection charge into the swirl chamber. It is obvious that the performance of a diesel engine would be significantly affected by the adjustment and condition of the injection nozzle.

Fuel-Injection Systems

Although fuel injection is sometimes employed on spark ignition engines, its primary use is, and has been, on diesel engines. Its application to spark ignition engines has been limited because of the comparative simplicity, dependability, and low cost of the carburetor. The following discussion is limited to injection systems as applied to compression ignition engines used on tractors.

All fuel pumps that produce the high pressures necessary for fuel injection are of the piston type. There are many variations of fuel-injection systems used on diesel engines. Only those that are used on high-speed tractor engines will be discussed. The three systems most commonly used on tractors are:

1. Individual or *in-line* injection pumps of the timed, metered type (figs. 5-31 and 5-32)
2. The *distributor* system in which one injection pump serves all the injection nozzles by delivering a metered fuel charge at the correct instant through a distributor (fig. 5-33)
3. The *unit injector* system in which the fuel-injection pump is combined with the injection nozzle in one assembly on the cylinder head (fig. 5-36)

The pumping principle of the in-line type is shown in figure 5-32. This pump is of the constant-stroke, lapped-plunger type and is operated by a

Pressure ratios required to obtain desired power output must be assumed or known for a given type of engine, and temperature rise across the compressor may be calculated from assumed values of compressor efficiency.

Once approximate airflow values are obtained, the next step in the matching process is to select a compressor characteristic on the basis of a number of important considerations. One must have a good idea of what maximum pressure ratio can be used at sea level at the full-load, full-speed rating point.

The complete field of required airflow over the entire speed and load range of the engine may be established and then superimposed on the compressor performance characteristic. The following factors have the greatest influence:

1. Desired torque rise, if any
2. Speed range of the engine
3. Altitude operating requirements
4. Aftercooling the compressed air

Selection of a compressor characteristic that will give the maximum efficiency over the engine operating range is the primary consideration. However, some compromises may be made on the basis of size and cost of the superchargers that might be applicable.

Referring to figure 5-38, a vehicle engine application will usually have a rather broad speed range over which it will be required to operate at various loads. This load variation requires the turbocharger to operate at various speeds, producing the operating line referred to as 100 percent, N_s. Lower values of engine speed move the air requirement line to the left, producing the other speed lines shown as 80 percent and 60 percent. Reducing the engine speed at full load will follow the dotted line from *A* in a manner determined by the amount of torque rise to be produced by the engine. Higher torque rise requirements result in broadening the engine air requirement field shown by the 5 percent and 15 percent torque rise lines. The low-idle point is shown at *B*, and high idle or governed speed at no-load is represented by point *C*.

As the turbocharged engine is taken to higher altitudes without derating, the rotational speed of the supercharger increases, as indicated by points *D* and *E*. Pressure ratio and airflow delivered to the engine both increase. Since there is no mechanical connection between the tubocharger rotor and the engine, it can "float" at various speed values depending on the altitude of operation and the energy in the engine exhaust gas. Thus the turbocharger can partially compensate for losses in ambient air density as the engine is taken to high altitude.

Another important consideration that affects the compressor match is the use of aftercooling of the compressed air. Aftercooling presents a means