DIESEL ENGINE DESIGN

By

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With 290 Illustrations
and 31 Engine Sections

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Chapter 3

ENGINE TYPES

DIESEL engines may be classified roughly into the following categories :

(1) Two-stroke or four-stroke working cycle.
(2) High, medium, or low-working speed.
(3) Single or double-acting.
(4) Vertical, horizontal, vee, opposed piston, etc.
(5) Duty – generating, marine, locomotive, road vehicle, etc.
(6) Supercharged or unsupercharged.

Considering these in more detail :

(1) Two-stroke or Four-stroke Working Cycle

Classification as to the mechanical cycle followed is, of course, general. At one time it was usual to assume all engines to be four-strokes " unless otherwise stated," but to-day it is necessary to be quite clear on the point, as there are as many of one type as the other. The fundamental difference between the two types is that in the case of the two-stroke, a separate pump is required to recharge the cylinder with air, whilst in the four-stroke engine the working cylinder itself performs that duty. This fundamental difference leads to a host of consequent problems, most of which are concerned with the twin subjects of higher rate of heat flow and the shorter time available for exhaust and air induction in the two-stroke. Owing to the fact that a firing impulse is received twice as frequently in a two-stroke, and that valves can be done away with, this type of engine is used universally for the higher powers, say above 3,000 B.H.P. per unit. Between 1,000 and 3,000 B.H.P., the supercharged four-stroke and the two-stroke are in equal competition, the two-stroke being possibly the favourite. Below 1,500 B.H.P. the four-stroke holds the field at present, chiefly owing to lower fuel and lubricating-oil consumption, but the new highly rated valve-exhausted two-strokes are rapidly gaining favour. It will be noted that in all cases of high-duty two-strokes, up to 1,000 B.H.P. at any rate, exhaust valves have been found to be essential, and it can no

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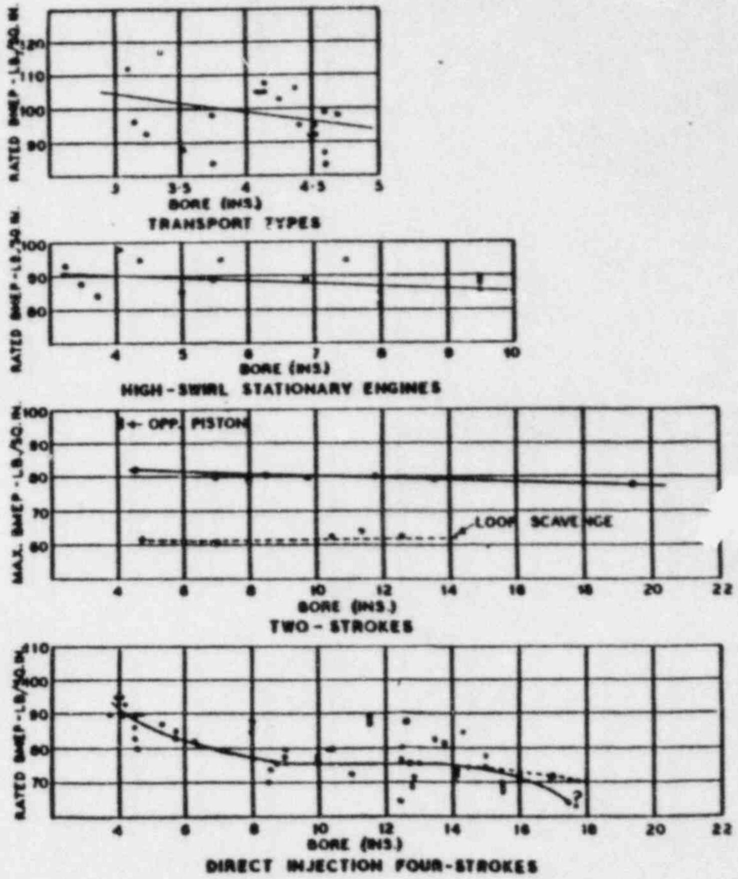

Fig. 4-1

pump power. A great deal will depend on scavenging efficiency. The old-fashioned "crankcase compression" engine rarely exceeded a B.M.E.P. of 60 lb./sq. in., whilst engines built under the Kadenacy patents have exceeded 120 lb./sq. in. on test. A plot of two-stroke B.M.E.P. is also shown in Fig. 4-1, but it should be noted that these are on the one-hour rating.

Piston Speeds

As the piston speed, $N_p = 2LN$, the horse-power expression may be rewritten :

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TABLE IX

Bore, in.	18	12	10	8	6	4
Speed, r.p.m.	300	428	500	600	[illegible]	1,250
Max. pressure, lb./sq. in.	700	740	790	850	900	1,050
Compression pressure, lb./sq. in.	420	430	440	460	490	510
Cylinder pressure at half stroke lb./sq. in.	75	75	80	90	90	100

The figures for compression pressure will, of course, depend to some extent on the pressure at the beginning of compression, but may be taken as a fair approximation. Similarly, the values for pressure at half stroke will depend on the value of " *n* " (and on the expansion ratio), but they have some interest, as it is near this point that the transverse stress due to the inertia of the connecting rod is a maximum.

The cylinder pressure ought to be regarded as a shock load when designing the running gear. Some reference has already been made to the rate of pressure rise in the cylinder, and from the point of view of fatigue stressing, the rates of rise per *second*, even in cases where the rise per degree is small, are quite serious. At 1,200 r.p.m., a rate of pressure rise of 40 lb./sq. in./degree is not far short of 300,000 lb./sq. in./sec. As will be pointed out later, under these conditions the *shape* of the part is almost as important as its strength if serious concentrations of stress are to be avoided.

(2) Inertia and Centrifugal Loads

Inertia forces increase with the *square* of the speed. Those due to the piston and connecting rod are in *opposition* to the gas loading at top dead centre, and their effect is to reduce the downward loading on the connecting rod and bearings. On T.D.C. of the exhaust stroke in four-strokes, the inertia forces introduce a *tension* load in the rod, and this means that the rod has to withstand some reversal of stress.

In the valve gear, almost the whole of the load is due to inertia, and the effect is cumulative. High accelerations require heavy valve springs to cope with them. This means that stronger gear is required to resist the stress due to spring forces, and this in turn increases the inertia loading. The exhaust-valve system of two-stroke engines is particularly troublesome in this respect.

Centrifugal forces due to rotating unbalanced masses impose loads on the bearings which travel round the bearing with the rotation of the crankshaft. They are often in opposition to the gas loading, but cause stresses in other parts of the engine, due to

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INTERNAL-COMBUSTION ENGINES

THEORY AND DESIGN

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period is the third stage or *gradual, or controlled, combustion*. The fourth stage, or *afterburning*, is burning of the fuel after injection terminates. This stage cannot be controlled and is very undesirable as its efficiency is comparatively low.

Figure 10-1 illustrates the process by a pressure-time diagram: the fuel is injected at point 1, but ignition does not start until point 2. Angle *a* represents the delay period also called *ignition lag*. For a certain engine the delay period depends upon many factors, as will be shown later. From point 2 to point 3, corresponding to a crank angle *b*, the flame spreads from the initial nucleus to the main body of the fuel charge. Similar to the conditions of spark-ignition engines, the flame velocity and

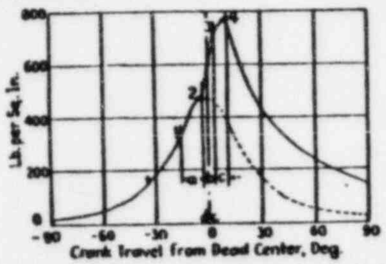

FIG. 10-1.—Combustion diagram of a high-speed oil engine.

the pressure rise depend upon turbulence; this second phase of the process is very important for a smooth operation of an engine. During the third stage, crank angle *c*, from point 3 to 4, the fuel burns as it leaves the fuel nozzle and the pressure may either increase, remain constant, or decrease, depending upon the rate at which it is delivered to the combustion chamber. The angle *c* is a function of the load which the engine is carrying. The fourth stage, afterburning, is not apparent on the indicator diagram. On a diagram taken from an engine, the dividing points 2, 3, and 4 are not pronounced, and the four stages merge one into another gradually.

10-3. Delay Period.—This period itself is made up of two parts: a heating period when the cold fuel droplets are heated, vaporized, and brought up to their ignition temperature and a period of true ignition delay that ends when the first particles actually ignite. However, in engines it is difficult to distinguish between these two periods, so the delay period is measured from the beginning of the injection to the moment of ignition.

The delay period presents a great interest, as it materially influences the operation of an engine: A shorter delay period gives a smoother operation; a longer delay period results in a rougher

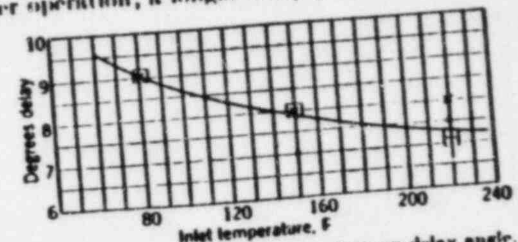

FIG. 10-2.—Effect of inlet temperature on delay angle.

and noisier running engine. The factors that influence the length of the delay period are:

(*a*) temperature of the air charge,
(*b*) pressure of the charge,
(*c*) atomization of the fuel,
(*d*) timing of injection,
(*e*) engine speed, and finally,
(*f*) ignition quality of the fuel, its cetane number.

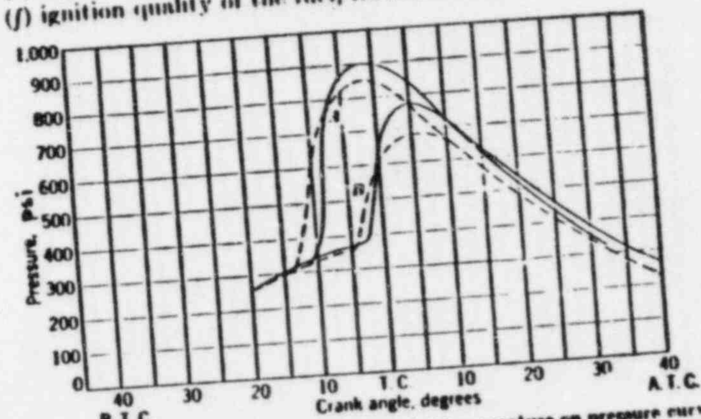

FIG. 10-3. Effect of injection timing and jacket temperature on pressure curves. Curves *A* and *B* injection starts 20 and 10° B.T.C., respectively; solid and broken-line curves jacket temperature 150 and 300F, respectively.

Temperature.—Figure 10-2 shows the typical effect of inlet temperature on the delay angle of an experimental engine.[1] The same condition is confirmed by another engine, Fig. 10-7.

[1] ROTHROCK, A. M., *SAE Journal*, vol. 34, June, 1934.

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Cooper-Bessemer compression-ignition oil engine equipped with a Buchi-Elliot turboblower. In this engine the mep when supercharged was limited not by available air but by the fuel-pump delivery.

Finally, Fig. 19-8 presents data about the increase of bmep in two compression-ignition aircraft oil engines as a function of the supercharging pressure.[1] Curve c gives the same data in respect to a large oil engine running at 300 rpm with an unchanged valve overlap of 30°.[2]

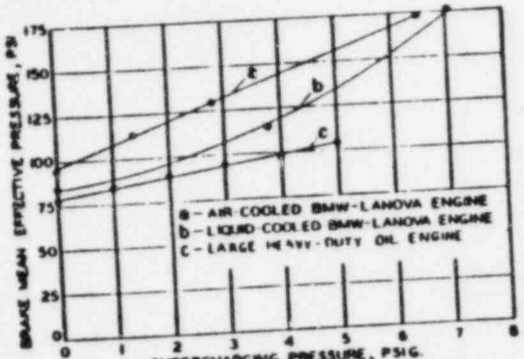

Fig. 19-8.—Effect of supercharging pressure on mean effective pressure.

Mechanical Efficiency.—The increase of friction losses with a supercharger driven by the engine itself is considerably smaller than the power gained through supercharging. As a result the mechanical efficiency, referred to the maximum load, increases with supercharging. Figure 19-9 shows the mechanical efficiencies of a 7-in. × 10-in. six-cylinder compression-ignition Cummins engine at different speeds and mean effective pressures when operating, supercharged with a Roots blower.[3] The mechanical efficiency of the same engine with natural aspiration at 1000 rpm when developing a bmep of 84 psi was found to be 73 per cent. The influence of supercharging upon the mechanical efficiency of two aircraft engines is brought out by Fig. 19-10.

[1] Maleev, V. L., *Mech. Eng.*, vol. 63, p. 446, 1941.

[2] *Diesel Power*, vol. 18, p. 877, October, 1940.

[3] Knudsen, H. L., Problems and Possibilities of Mechanical Supercharging of Diesel Engines, *Diesel Power*, vol. 19, p. 856, October, 1941.

Fuel Economy.—Owing to better combustion because of increased turbulence, better mixing of the fuel and air, and of an increased mechanical efficiency, the specific fuel consumption in most cases, though not in all, is lowered by supercharging.

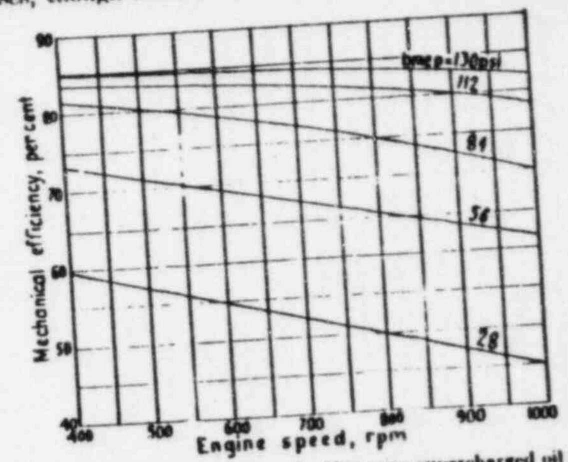

Fig. 19-9. Mechanical efficiencies of a Cummins supercharged oil engine.

Figure 19-11 gives an interesting comparison of the performance of a three-cylinder 9½-in. × 10½-in. Alco oil engine at 1000 rpm: curve c, unsupercharged; curve d supercharged with a Roots blower, power increase about 75 per cent, fuel consumption 5 per cent lower; curve e supercharged by a Buchi turboblower, power increase about 87 per cent, fuel consumption 13 per cent lower than in the unsupercharged engine.

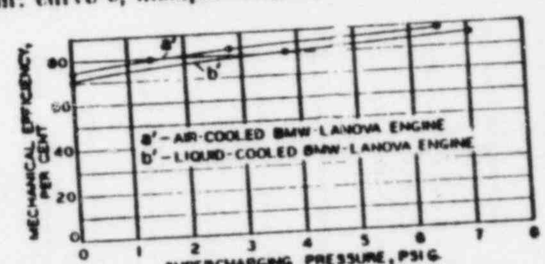

Fig. 19-10.—Effect of supercharging on mechanical efficiency of compression-ignition aircraft engines.

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Curves *a* and *b* show also how the performance of this engine was improved by putting in turbulence-creating pistons.

Figure 19-6, on the other hand, shows that the fuel consumption of that particular gasoline engine was improved very little and only at the maximum supercharged horsepower.

Fuel Knock.—In compression-ignition oil engines, increasing the inlet pressure decreases the ignition lag and consequently the rate of pressure rise in the cylinder, which results in an increasing smoothness of operation.[1] On the other hand, in a gasoline engine, if the engine is operated with a compression ratio that causes incipient detonation when supercharging and if a

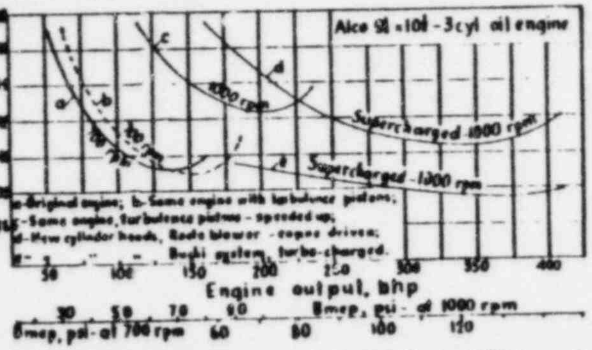

FIG. 19-11.—Performance development of Alco $9\frac{1}{2}$-in. × $10\frac{1}{2}$-in. engine.

fuel with the same antiknock characteristics is used, in order to prevent objectionable detonation, the compression ratio must be lowered so that the compression pressure of the supercharged engine will remain about the same as before supercharging. This will slightly lower the thermal efficiency, but the power output will be increased, as a greater amount of fuel will be burned. An engine operating with natural aspiration with a compression ratio 7:1, when supercharged should have a compression ratio about 6:1.

19-4. Limitations.—The permissible amount of supercharging depends upon the ability of the engine to withstand the increased pressure and heat stresses.

Pressures.—The increase of the mean effective pressure naturally increases the mean bearing pressures and mechanical

[1] *NACA Tech. Notes* 509, 1936, p. 8.

friction losses. On the other hand, the maximum pressures and temperatures will go up too. Thus a six-cylinder 12-in. × 15-in. by 650 rpm Enterprise oil engine unsupercharged develops continuously 670 bhp, or a bmep of 80 psi, with a maximum pressure of about 680 psi. When equipped with a Buchi turbine-driven blower, the engine can develop a maximum bmep of 162 psi and a continuous bmep of 125 psi, or a power increase of 56 per cent; the maximum pressure goes up to about 850 psi, the specific fuel consumption goes down from 0.40 lb/hp-hr to 0.375 lb/hp-hr, or an improvement of 6 per cent.

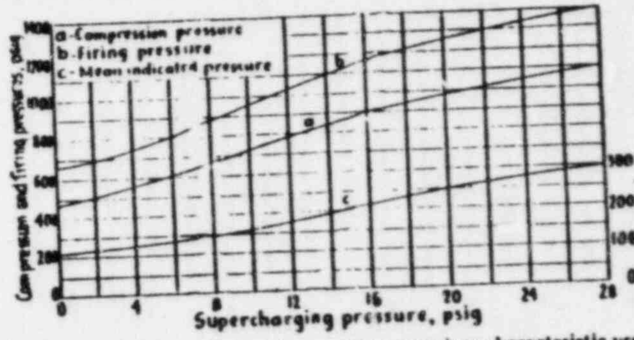

FIG. 19-12.—Effect of supercharging pressure on various characteristic pressures.

Figure 19-12 gives the relation between supercharging pressures, compression pressures, maximum combustion pressures, and obtained mean indicated pressures for sea-level operation. The curves represent computed values but are corrected on the basis of corresponding test data and give an idea of the limit of supercharging.

Temperatures.—The above-mentioned Enterprise engine has an exhaust temperature of 720 F unsupercharged and 960 F supercharged. The Cummins engine mentioned before has exhaust temperatures as shown on Fig. 19-13; Fig. 19-14 shows the temperatures for the Alco three-cylinder $9\frac{1}{2}$-in. × $10\frac{1}{2}$-in. engine at different loads.[1] Finally Fig. 19-15 gives a very complete picture of the performance of a six-cylinder $12\frac{1}{2}$-in. × 13-in. Alco engine, which supercharged is rated 1000 bhp at 740 rpm. The exhaust temperature goes up from 720 F, at 80 psi mep, to 1100 F, at 120 psi mep.

[1] *Diesel Power*, vol. 19, p. 864, October, 1941.

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CHAPTER 27

RUNNING GEAR

27-1. Trunk Pistons.—*The functions* of a trunk piston are:

1. To transmit the gas pressure to the crankshaft.
2. To take the side pressures due to angularity of the connecting rod.
3. To seal the inside of the cylinder from the crankcase.
4. To dissipate heat absorbed by the piston top during combustion and early part of expansion stroke.

Design Objects.—In designing a piston to meet these requirements the following objects must be sought:

1. Strength of the piston, particularly of its head.
2. Sufficient projected side area, and rigidity of the barrel.
3. Minimum work of friction.
4. Sealing of the working space against escape of gases.
5. Preventing the entrance of lubricating oil into the combustion space.
6. Good dissipation of the heat to the cylinder walls.
7. Minimum weight.

The design objects are listed not in the order of their importance but to conform with the order of functions as listed above. Good heat dissipation is one of the most important design requirements.

Materials used to make trunk pistons are in the order of their importance: cast iron, cast aluminum, forged aluminum, cast steel, and forged steel.

Cast iron is an excellent material; its main drawback is that it gives a slightly heavier piston than aluminum. However, with a proper design the difference is only about 10 to 20 per cent. Cast-iron pistons produce less cylinder-liner wear than aluminum ones,[1] especially if they are tin-plated.

[1] *Automotive Ind.*, Jan. 16, 1937, p. 81; *Z. Ver. deut. Ing.*, vol. 81, p. 610, 1937.

Cast aluminum alloy gives better heat dissipation and lighter weight but costs considerably more than cast iron. The strength is about the same as that of cast iron.

Forged-aluminum pistons are stronger and still lighter. They are used for aircraft engines and heavy-duty high-speed compression-ignition oil engines.

Alloy cast-steel pistons are used in some automotive engines and require liners of great surface hardness. The same is true of forged-steel pistons used in some aircraft engines.

Piston Head.—The thickness t_1 of the head or crown can be computed, considering it a flat round plate of uniform thickness fixed at the edges, from the formula

$$t_1 = 0.43D\sqrt{p/S} \tag{27-1}$$

where D is the cylinder bore, in.,
p is the maximum pressure during combustion, psi,
S is the allowable stress in bending, psi

A stress of 5500 psi can be allowed when using a good close-grained cast iron or an aluminum alloy with an ultimate tensile strength of 20,000 psi. If the material used has an ultimate strength of 30,000 psi, such as nickel cast iron, semisteel, or special aluminum alloy, normalized, then S can be taken as 8000 psi. For forged-steel heads S can be raised to 12,000 psi. The piston head has often four or six radial ribs a, Fig. 27-1, of a thickness t_2, one-third to one-half the thickness t_1 of the head, but it is safer not to consider these ribs when computing t_1.

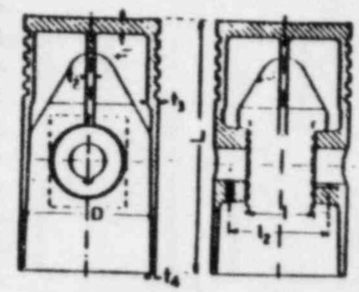

Fig. 27-1. Trunk piston of a gas engine.

TABLE 27-1.—THICKNESS OF PISTON HEAD

Type of engine	Piston material	Four-stroke	Two-stroke
Compression-ignition oil engines	Cast iron	0.11D-0.13D	0.16D-0.18D
Compression-ignition oil engines	Aluminum	0.13D-0.16D	0.17D-0.20D
Spark-ignition gas engines	Cast iron	0.12D-0.14D	0.20D-0.23D

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