FACTORS CONTROLLING DIESEL ENGINE PERFORMANCE

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Received July 26, 1937

INTRODUCTION

The performance of any internal-combustion engine, irrespective of its operating cycle, is primarily a function of the thermal energy liberated per cycle, the part of the cycle in which it is liberated, the compression ratio, and the mechanical limitations imposed by the engine structure. For the purpose of this paper, engine performance is determined by two quantities: first, the highest indicated mean effective pressure (i. m. e. p.) attainable, and second, the minimum fuel consumption expressed as pounds per indicated horsepower-hour. In order to secure optimum values for these quantities it is necessary that the combustion be completed as early in the power stroke as possible, with the provision that neither the explosion pressure nor the rate of pressure rise shall exceed certain limiting values.

The combustion in a compression-ignition or Diesel engine differs from that in a spark-ignition engine in that a large portion of the fuel must be mixed with air during the combustion period. The Diesel engine must be a compromise between extensive mixing before combustion, with its attendant early efficient burning but excessive rate of pressure rise, and limited mixing before combustion, giving a permissible rate of rise but a lower thermal efficiency. Provision must therefore be made for supplementary mixing of the major portion of the fuel charge with the available air after ignition by a proper coördination of the injection system, the combustion chamber, and directed air flow. It is the purpose of this paper to outline the influence of these factors on combustion in Diesel engines as revealed by a number of investigations conducted for the National Advisory Committee for Aeronautics (N. A. C. A.) on injection systems, combustion chamber design, and engine operating conditions, and to show how these factors control engine performance.

OPERATION OF TYPICAL INJECTION SYSTEMS

The fuel for most modern high-speed Diesel engines is injected, as a liquid under high pressure, through one or more small orifices directly into the combustion space of each cylinder by means of a displacement pump connected to an injection valve by a tube of appropriate dimensions. Each of the individual parts of such a system should be designed and coördinated to fulfill the following conditions: small variation in timing and fuel quantity from cycle to cycle; proper spray penetration and distribution for the particular combustion space; a sharp but controllable spray cut-off for various load conditions; and perhaps some control of injection rate and start of injection if the engine is intended to operate efficiently over a great speed range.

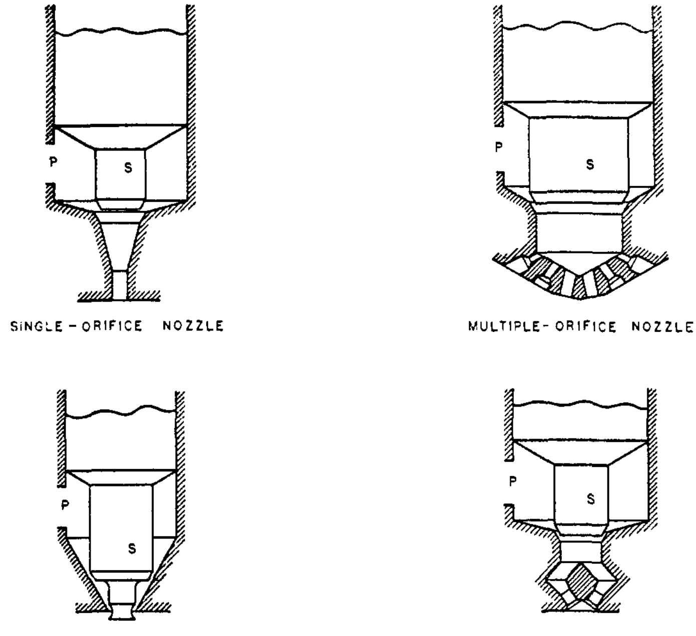

FIG. 1. Typical closed injection nozzles

The plain, pintle, multiple-orifice, and two-impinging-jets nozzles are representative of the many different nozzle styles used in commercial engines. Such nozzles are classified as open or closed, depending upon whether or not the cylinder gas has access after spray cut-off to an appreciable passage length above the flow-restricting orifice section of the nozzle. The nozzles sketched in figure 1 are all of the closed type with the valve stem S shown in the raised or injection position. If the oil pressure in the passage P falls sufficiently, a suitable spring above the valve stem forces the latter down until its seating surface is in contact with a corresponding surface on the nozzle, thus providing a gas-tight seal between the cylinder interior and the fuel passage above the nozzle.

The purpose of the nozzle is twofold,—to distribute the fuel throughout the combustion chamber and to assist in breaking up the fuel jet into the numerous droplets necessary for the intimate mixing of the fuel and air. Disturbances of the fuel jet as it issues from the nozzle produce surface irregularities that are drawn out into fine ligaments by the relative motion of the fuel and air. The liquid in each ligament, after detachment from the main jet, quickly contracts into a spherical droplet by the action of surface tension. This process is illustrated in figure 2 for a low jet velocity

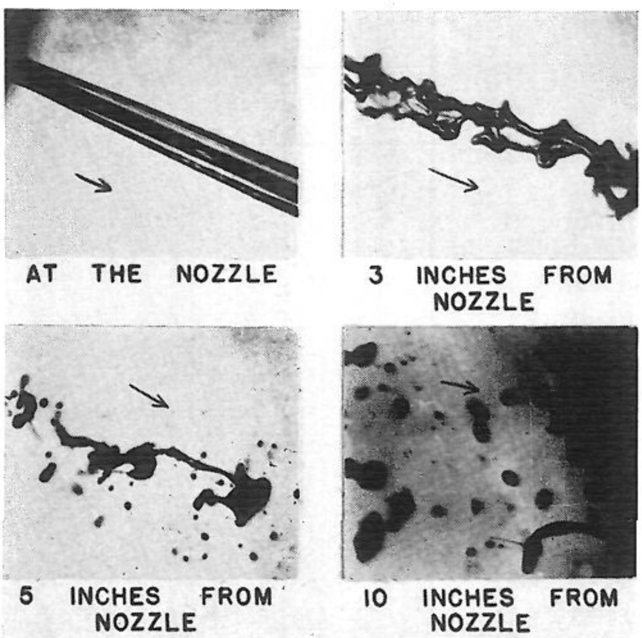

Fig. 2

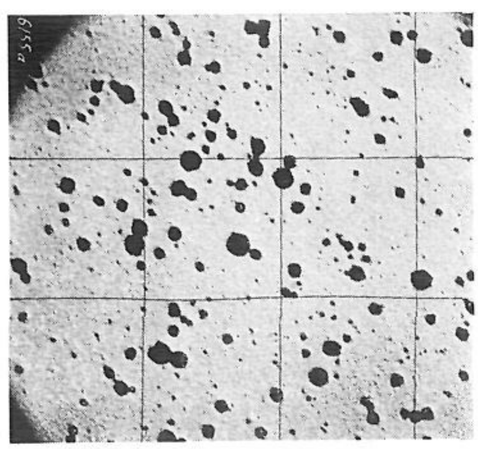

Fig. 3

Fig. 2. Disintegration of low-velocity jet in atmospheric air. Injection pressure = 100 lb. per square inch.

Fig. 3. Fuel drop impressions in lampblack surface

(8). Figure 3 is a photomicrograph of the impressions made by such drops after settling upon a lampblack surface (5). A detailed analysis of the size distribution of these impressions corresponding to various injection pressures (or jet velocities) furnished the data for the curves in figure 4. The percentage of fuel in each group was based upon the volume corresponding to those drop impressions having diameters within 0.00025 in. of the nominal or "mean" group diameter and the total volume for all such groups. It may be seen that for a given nozzle the average drop size decreases as the injection pressure increases, although the change is not very great at the higher jet velocities. Other conclusions drawn from these tests were as follows: Decreasing the orifice diameter results in more uniform atomization of the spray and a smaller mean drop size. The density of the air into which the fuel is injected has little effect on the drop size distribution. Visual inspection of sprays cannot be used to estimate the relative drop size distribution.

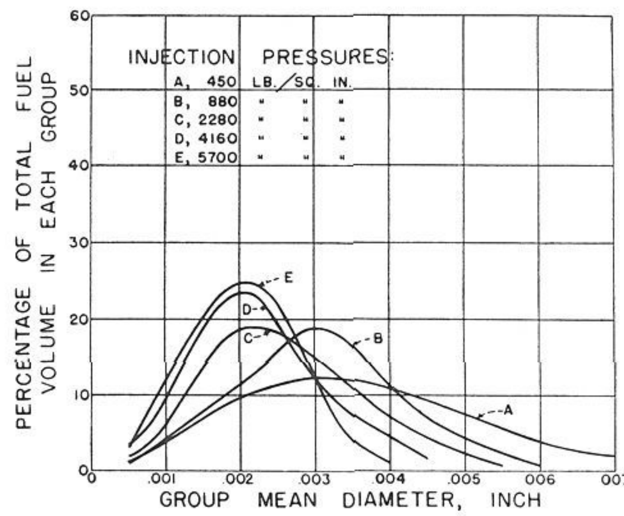

FIG. 4. Effect of jet velocity on atomization

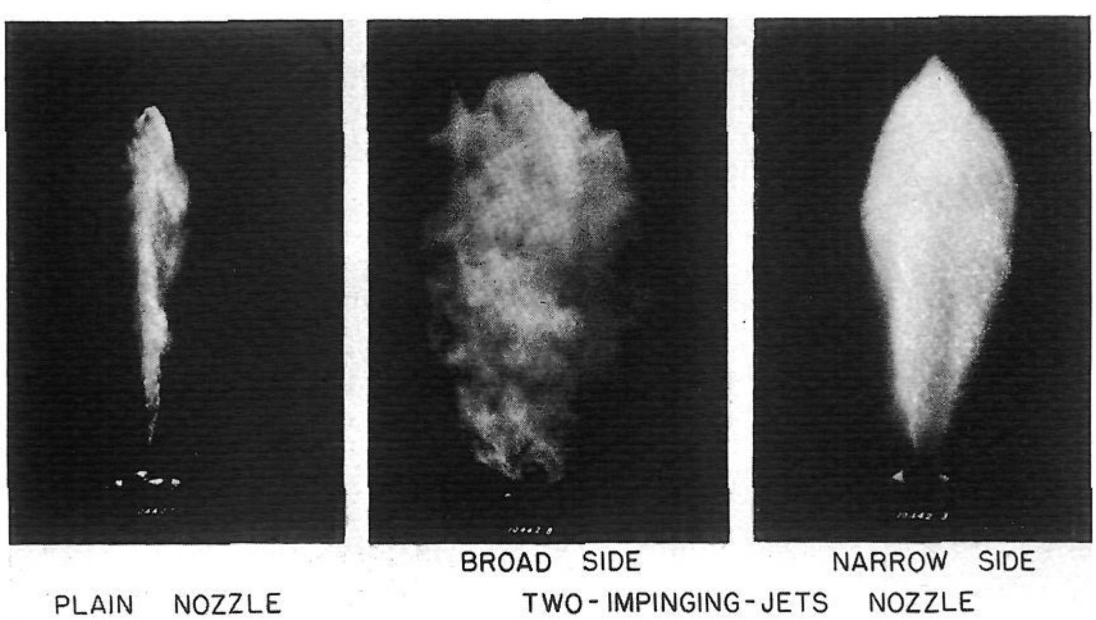

FIG. 5. Spark photographs of sprays injected into atmospheric air

Spark photographs (6) of the symmetrical spray from a plain nozzle and of two views of the spray from a two-impinging-jets nozzle, each of which was injected into atmospheric air at a high jet velocity, are reproduced in figure 5. The multiple-orifice spray is simply a composite of plain sprays, whereas the pintle nozzle gives a spray of greater lateral spread. At gas densities greater than atmospheric the spray penetration decreases, as is illustrated in figure 6 for two plain nozzles and an impinging-jets nozzle. The penetration into any gas is determined by the gas density rather than the gas pressure (4). For some applications the lower penetration of the impinging-jets nozzle may be advantageous, because of the greater lateral spread of the spray and the decreased impingement of the spray on the walls of the combustion chamber. Four such nozzles of the open type are used per cylinder in the Junkers aircraft Diesel. The length–diameter ratio of round-hole orifices giving the best spray penetration lies between 4 and 6, whereas the greatest lateral spread, or spray angle, occurs with ratios between 2 and 3 (3). It is for this reason that the smaller orifices are shown countersunk in the multiple-orifice nozzle in figure 1.

Further information as to the relative spatial distribution of fuel in sprays from various nozzle types has been obtained (7) by determining the

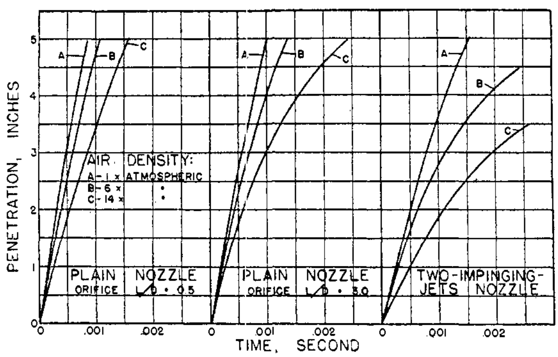

FIG. 6. Effect of air density on penetration of spray tip. Injection pressure = 4000 lb. per square inch.

quantity of fuel caught by suitable receiving apparatus set up in front of the nozzle for a definite number of injections into air of the desired density. Typical curves for the distribution obtained with three of the nozzle types shown in figure 1 are given in figure 7 for an air density fourteen times that of atmospheric air. The impinging-jets nozzle gave the best distribution within the spray, the pintle nozzle was intermediate, and the plain nozzle gave the poorest distribution. For plain round-hole orifice nozzles some improvement in lateral distribution within the spray resulted from an increase in injection pressure, the improvement being a function of orifice size. Low viscosity fuels also gave better distribution. The curves illustrate the difficulty of finding a satisfactory combination of orifice and combustion chamber design and the necessity of employing air movement within the combustion chamber to assist in the mixing of fuel and air. At present, the pintle nozzle is widely used in commercial

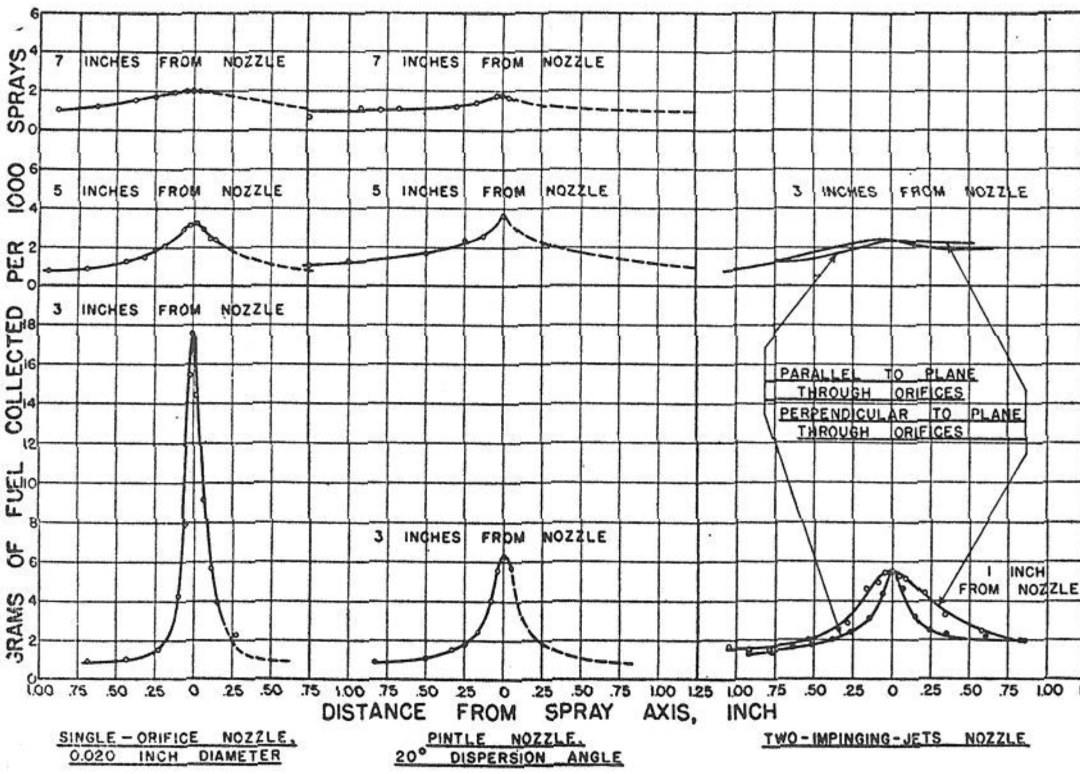

FIG. 7. Distribution of fuel within sprays

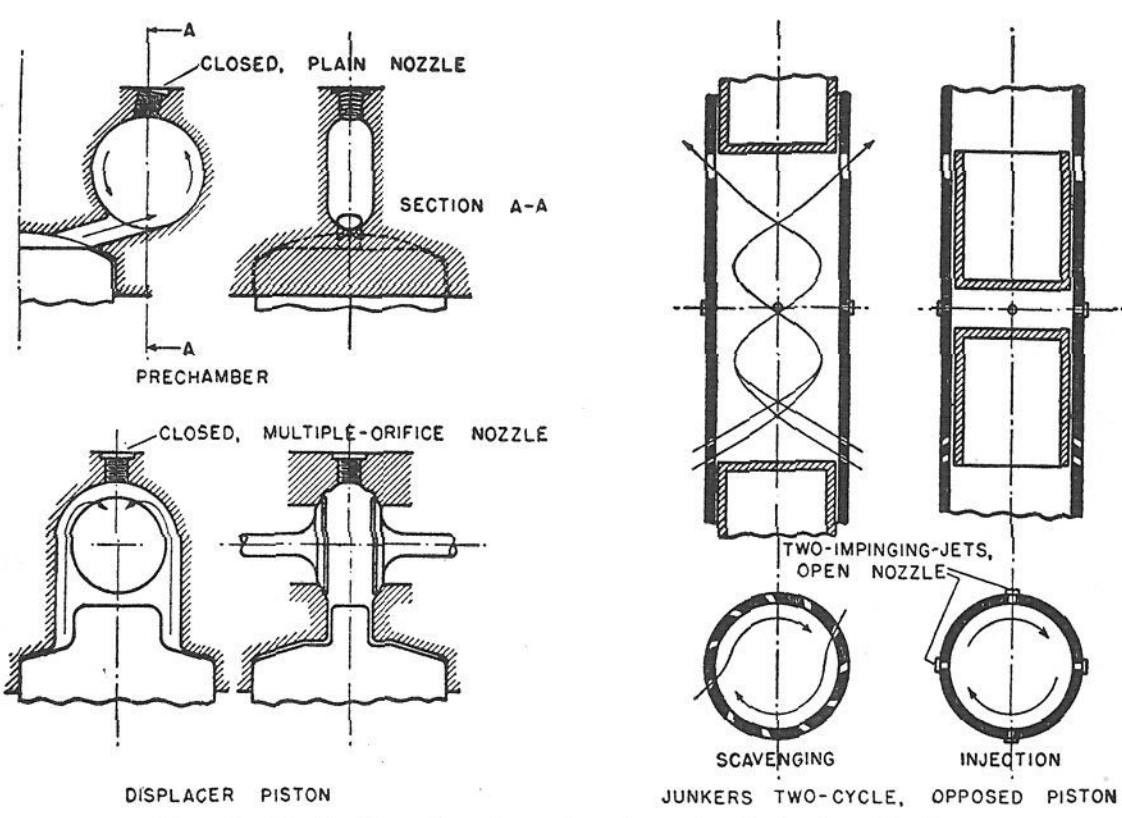

FIG. 8. Typical combustion chambers for inducing air flow

applications employing some form of precombustion chamber, but the multiorifice nozzle has been found most effective in tests at this laboratory with integral types of combustion chambers.

COMBUSTION AND COMBUSTION CHAMBER DESIGN

If it were not for the fact that available injection systems are incapable of distributing the fuel in a completely satisfactory manner, there would be little excuse for the varied combustion chamber designs that have been used in commercial Diesel engines. All such designs have as their basic purpose one or both of two aims: (*1*) to furnish supplementary fuel–air mixing to correct partially for the deficiency in the spray distribution and (*2*) to diminish the combustion shock or Diesel knock. As fuels of sufficiently high ignition quality are available, the problem of securing adequate mixing is distinctly of greatest importance, particularly with high-speed engines. On this basis the combustion chamber should possess two characteristics: first, it should have a shape conducive to the best distribution of the fuel with respect to the air charge and second, it should contribute as far as possible, by induced air flow and turbulence, to the secondary mixing of unburned or incompletely burned fuel with unused oxygen. Obviously an optimum coördination of injection system and combustion chamber shape minimizes the required auxiliary mixing by air flow and turbulence. For this reason high dispersion or multiple-orifice nozzles generally give better results.

Three typical systems for obtaining air flow in the combustion space are illustrated diagrammatically in figure 8, each of these being at present an optimum design for its type. The air flow is induced by the restricted connecting passage between two portions of the combustion space in the prechamber type (9), and by the restriction between the combustion chamber and displacement volume during the latter part of the compression stroke in the displacer-piston type (10, 11, 13). In the Junkers two-cycle design the flow is generated by directed intake ports or baffles. It is particularly important to note that each design is basically a disk chamber in which is induced a directed flow of the air charge. The displacer-piston type is really an adaptation of the so-called quiescent-chamber engine without any projection on the piston as developed by the National Advisory Committee for Aeronautics over a period of years (**2**).

The fuel must not only be atomized and distributed throughout the air in the combustion chamber, but it must also be vaporized through absorption of heat from the air so as to form a combustible mixture. The rapidity with which the injected fuel absorbs heat from the surrounding air is indicated by figure 9 (**23**). The upper record shows the decrease in total gas pressure attending the heat transferred to Diesel fuel upon its injection into heated and compressed nitrogen. This decrease began with the very first portion of the fuel injected. The lower record corresponds to the third injection into the same air charge after reëstablishment of thermal equilibrium following the first two injections. The decrease in the initial rate of pressure drop (or heat transfer) evident in the lower record must be attributed to one or both of the following: the greater heat capacity of the air–vapor mixture existing after the second injection, or an inhibition of the vaporization of a portion of the fuel, which presumably occurred in the absence of vapor.

An indication of the rapidity of the vaporization process in an engine is furnished by the high-speed motion pictures reproduced in figure 10 (16). The photographs in each series were obtained by means of suitably timed sparks and show a silhouette of a quiescent combustion chamber provided

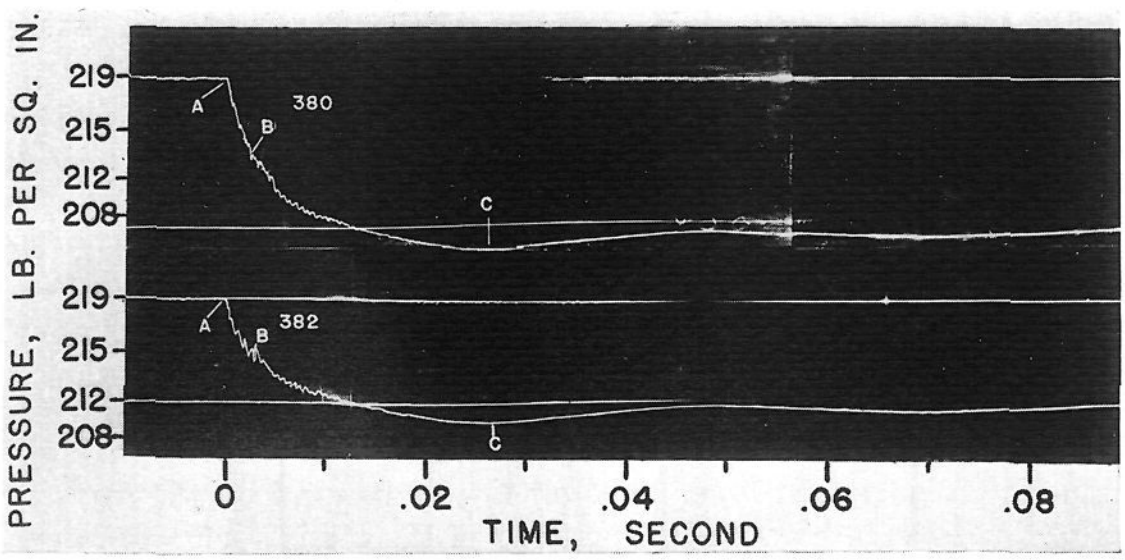

FIG. 9. Effect of fuel vapor concentration on rate of heat transfer

with front and rear windows 2.5 in. in diameter. In taking the photographs, the engine was motored over and a single charge of fuel injected into the combustion chamber. Combustion was prevented by the extremely low temperature at which the engine was operating. Each series of photographs corresponds to a fuel having a dew point markedly different from that of the others, the injection advance angle being maintained constant. The chamber contents became more transparent as the spray core disappeared through its vaporization and its movement beyond the field of view. On the expansion stroke the contents of the chamber suddenly became opaque, as a result of the condensation of the vapors. The order of condensation in terms of crank angle was the same as that for the volatility or dew points of the fuels. The fact that the fuels were vaporizing was confirmed by the results of further tests in which the injection advance angle, the fuel quantity, the engine speed, and the engine temperature were varied independently.

The extent to which the fuel is mixed with the air in the combustion chamber can be studied by investigating the efficiency of the combustion and the efficiency with which the heat energy extracted from the fuel is transformed into useful work. By combustion efficiency is meant the fraction of available heat energy that is extracted from the fuel during the latter part of the compression stroke and during the power stroke. The efficiency with which this extracted heat energy is transformed into indicated work is termed the cycle efficiency. Although these two efficiencies are independent of each other, they are generally studied together. The

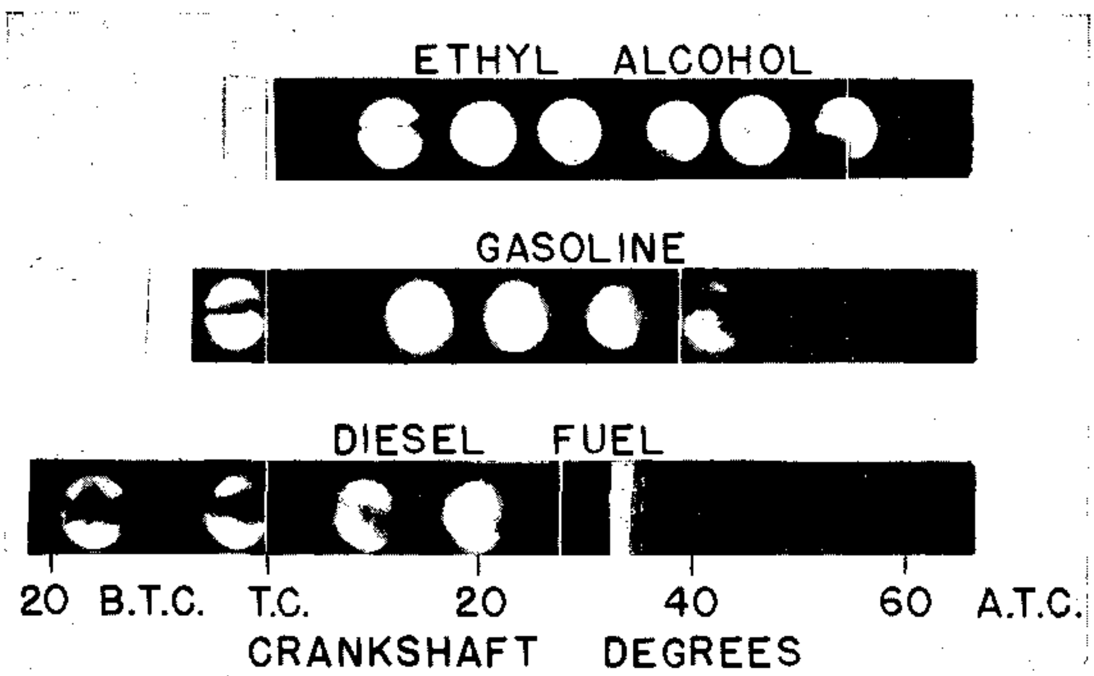

FIG. 10. Vaporization and condensation of fuel. Engine R. P. M. = 1500; I. A. A. = 20°

product of these two efficiencies is the indicated thermal efficiency of the engine and is the factor of primary interest.

The extent to which the fuel is burned is indicated to a certain degree by high-speed motion pictures of the combustion process and to a greater degree by an analysis of the indicator cards taken during engine operation. High-speed motion pictures were taken on the same single-cylinder quiescent-chamber test engine, operating under its own power for a single cycle, as was used for the vaporization work. In these tests the effects of high wall temperatures and of exhaust gas dilution were not present; otherwise, the test conditions meet very closely those experienced in actual engines. The quiescent chamber provided a minimum of air movement so that the distribution and dispersion of the fuel by the injection nozzle could be

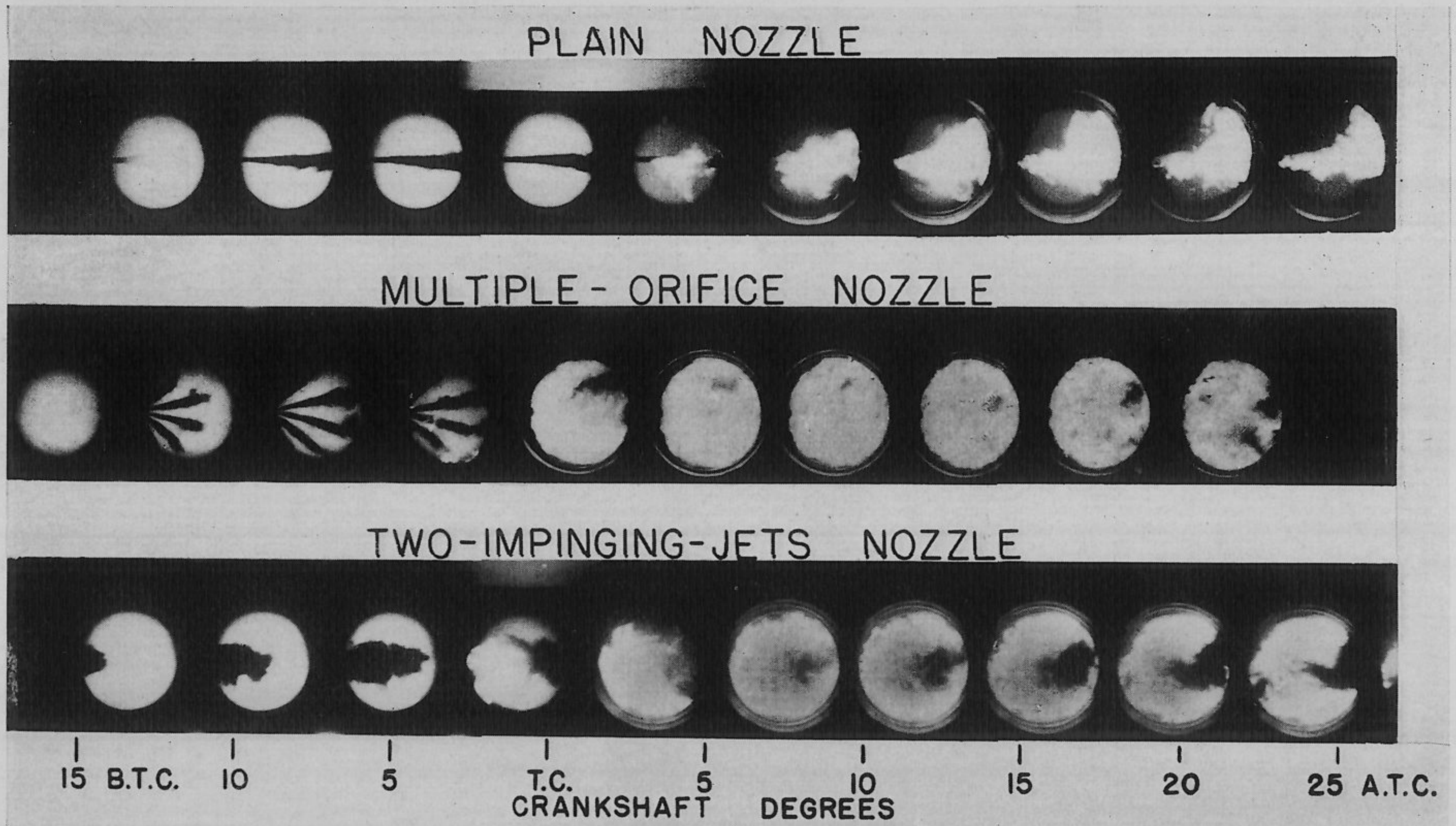

FIG. 11. Variation of fuel distribution and flame formation with nozzle design. Engine R. P. M. = 1500; I. A. A. = 15°

studied. Photographs of the injection and combustion of the fuel taken under this condition at a rate of about 2000 frames a second are presented in figure 11 (19). The figure emphasizes the inadequacy of the fuel dispersion and distribution from a single round-hole orifice, and the improvement that is obtained from the multiorifice nozzle. With the single round-hole orifice the area reached by the flame indicates the area of the fuel dispersion. Although there is continual vaporizing of the fuel sprays, the spray core and at least part of the envelope are visible in every case up to the start of combustion. Much of the combustion chamber is never reached by the burning gases. With the six-orifice nozzle the flame not only reaches all the visible portion of the combustion chamber, but persists for an appreciable time interval. The fact that ignition often occurred between the sprays is of interest. The slit nozzle, although giving fairly uniform distribution throughout the spray, did not disperse the fuel throughout the chamber, so that the mixing was inferior to that given with the six-orifice nozzle, even though the latter contained the six dense spray cores. The dense core is apparently necessary to obtain sufficient spray penetration in a combustion chamber which does not have high-velocity air flow.

The effect of air flow on spray development and on combustion is shown in the three series of schlieren, high-speed, motion-picture photographs, together with the respective combustion chamber arrangements, as in figure 12 (20). Except for the definite deflection of the opaque fuel in the two lower series, these pictures do not convey the impression of air motion nearly so well as the projected motion pictures. Other than for the schlieren feature, the upper series of photographs is similar to that in figure 11. As the displacer on the piston, as shown opposite the second series, entered the combustion chamber a violent swirl of the air as a whole was produced in the chamber. Even though the spray core itself was not destroyed by the air swirl, a greater combustion chamber area was reached by the flame than was the case without the air movement. With the multiorifice nozzle and the two-passage displacer the pictures indicate some air movement, but a comparison with similar pictures taken without air flow shows that there was no apparent improvement in the mixing of the fuel and air. Nevertheless, both the single and double passage displacers showed a decided improvement in performance for a combustion chamber of similar design on a single-cylinder test engine. Apparently with a nozzle that distributes the fuel reasonably well, the chief function of the air flow is not to disrupt completely the spray cores, but to improve the mixing between the individual sprays. In fact, it seems that the sprays contribute to the destruction of the air movement, but in so doing the desired mixing of the air and fuel is partially achieved.

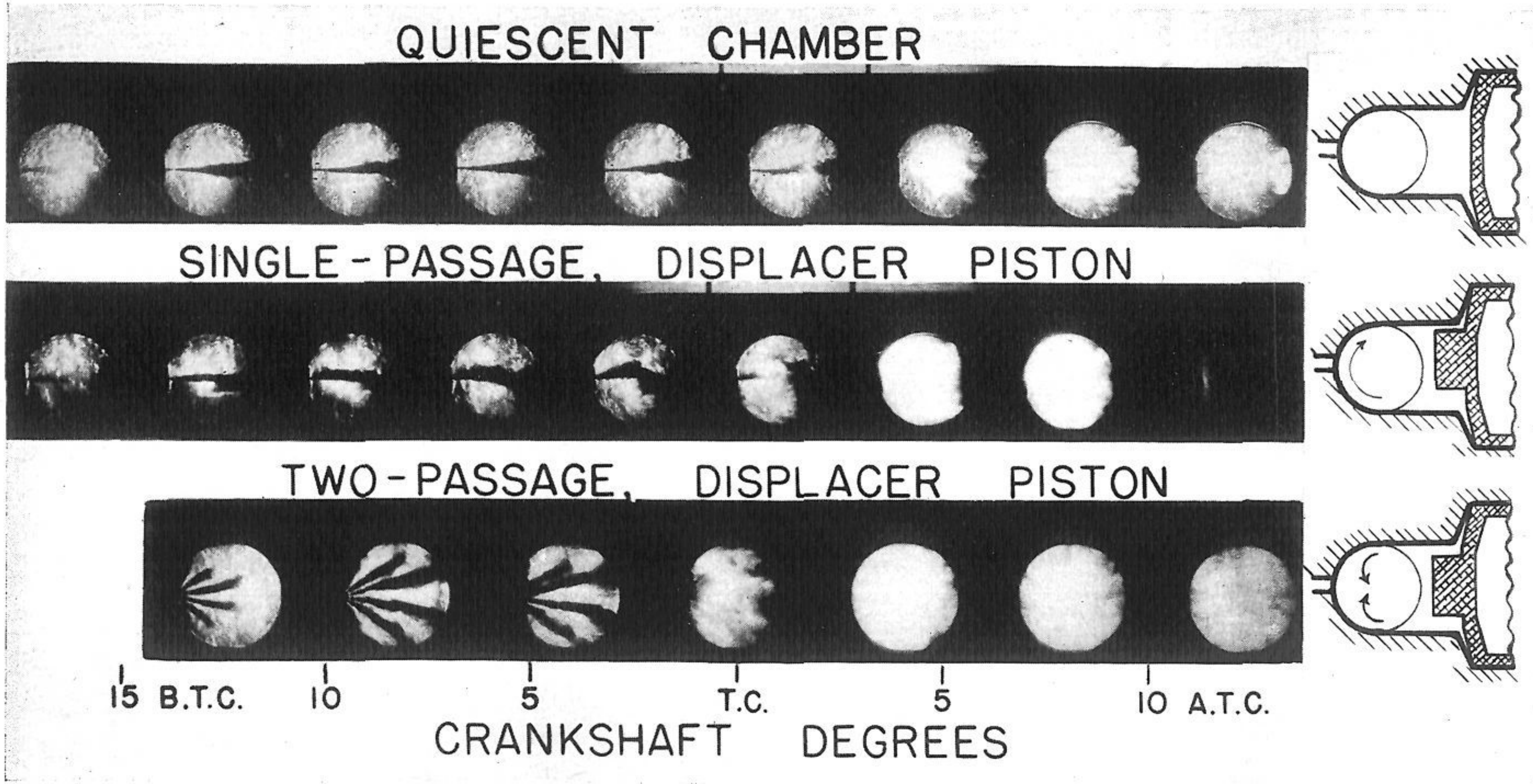

FIG. 12. Effect of air flow on fuel sprays. Engine R. P. M. = 1500; I. A. A. = 15° to 20°

By an adaptation of a method developed by Schweitzer (21) for the analysis of engine indicator cards not only the combustion efficiency but also the cycle efficiency can be obtained to a fair degree of accuracy. The curves in figure 13 show the effect of fuel-injection advance angle (I. A. A.) on the indicator card and on the rate of combustion in the quiescent combustion chamber (14). The corresponding combustion and cycle efficiencies were computed from the indicator cards shown and the known fuel quantity injected. These efficiencies are given in table 1.

The data show that the ignition lag remained practically constant for the different conditions. Therefore it seems reasonable to believe that a

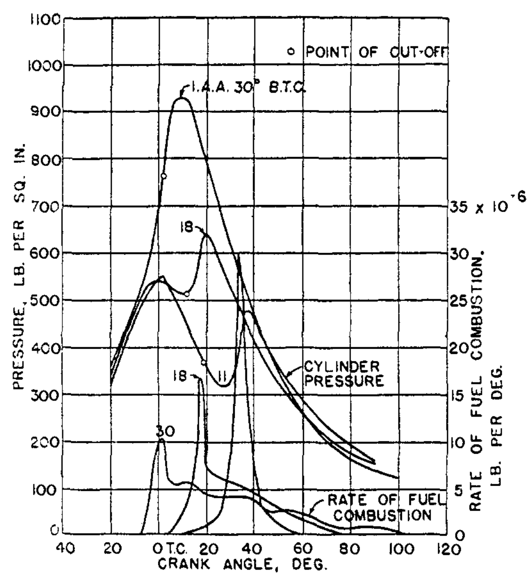

FIG. 13. Effect of injection advance angle on combustion rates

factor equally important with the ignition lag is the time interval between spray cut-off and the start of rapid combustion. Spray photographic tests have shown that only after spray cut-off does the spray core disintegrate. A comparison of figure 13 and table 1 shows that as the maximum rate of combustion was increased, the combustion efficiency also increased, but that the cycle efficiency decreased because of the lateness of the burning. As a result the thermal efficiency was highest for the earliest I. A. A. Had the tests been run on different fuels instead of different injection advance angles, then the fuel giving the highest combustion efficiency would also have given the highest thermal efficiency. This same effect has been obtained by using the same fuel and injection advance angle, but operating the engine at different jacket temperatures (2). In these tests an increase in the jacket temperature decreased the ignition lag, but also decreased the combustion efficiency because of the shorter lag. These data have led to the conclusion that the Diesel engine should be operated on the lowest cetane number fuel consistent with easy starting and permissible rates of pressure rise.

Figure 13 also shows the long after-burning period that exists in most high-speed Diesel engines. Until means are derived for burning this fuel earlier in the expansion stroke, the cycle efficiencies inherent in the high expansion ratio of the Diesel engine will not be realized. In fact, the elimination of this after-burning period will probably do more to advance the Diesel engine than any other single factor. In other tests conducted by the National Advisory Committee for Aeronautics (1, 15, 17) it has been shown that if the lag between cut-off of injection and start of combustion pressure rise is too great, the combustion efficiency and the combustion rates decrease. It appears, therefore, that there is a certain value

TABLE 1

Fuel-injection advance angle and efficiency

I. A. A.	FUEL-AIR RATIO	COMBUSTION EFFICIENCY	CYCLE EFFICIENCY	THERMAL EFFICIENCY
30	0.039	0.59	0.57	0.33
18	0.046	0.64	0.46	0.30
11	0.051	0.69	0.36	0.25

of this lag which gives a maximum combustion efficiency and rate-of-pressure rise. For this reason, if the after-burning period can be eliminated, not only the cycle efficiency but perhaps the combustion efficiency as well will be increased.

The after-burning is probably caused by inadequate mixing of the fuel and air. This mixing is controlled by the form of the fuel spray, the degree of atomization, the rate of fuel vaporization, the rate of vapor diffusion, and the degree and form of air movement within the combustion chamber. It is questionable whether any improvement in atomization over that obtained at present will further improve the distribution of the fuel. Tests were conducted at the laboratories of the National Advisory Committee for Aeronautics, in which a high-velocity air jet was directed through the injection nozzle with the fuel spray. Although atomization and burning tests showed the fuel to be much better atomized, the system did not give any improvement in engine performance. The rate of fuel vaporization is apparently comparatively high (16), and the test results indicate that, with the fuels used on commercial engines, the vaporization is sufficiently fast in every case to provide good mixing of the fuel and air. The diffusion of the fuel vapors appears to be too slow to provide adequate mixing even though nozzles with good distribution characteristics are employed (18, 19). It can be concluded that the improvements in mixing must be obtained through close coördination of the spray form and the air flow in the combustion chamber.

It is evident from the solid curves in figure 14 that the indicated mean effective pressure and the specific fuel consumption improve with N. A. C. A chamber design in the following order: quiescent chamber, prechamber, and quiescent chamber in combination with a displacer piston (2, 9, 11). The dashed curves show that the response to boosting[1] measured in terms of the increase in indicated mean effective pressure is about the same at

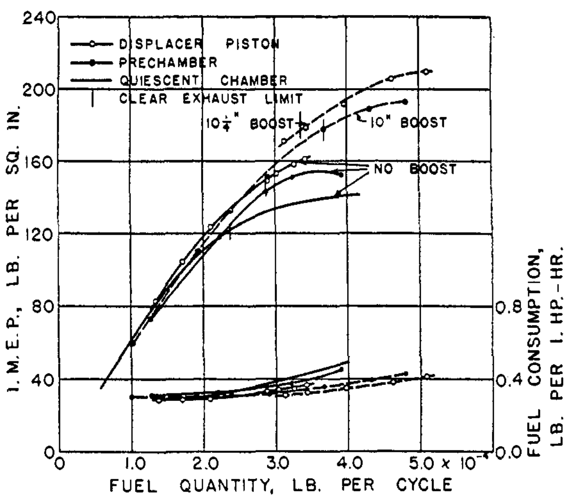

FIG. 14. Performance with typical combustion chambers

1500 R.P.M. for the prechamber and displacer engines. The clear exhaust limit for the quiescent chamber occurs at a considerably lower fuel quantity than for the unboosted prechamber and displacer engines. Considered on the basis of the limiting indicated mean effective pressure with clear exhaust, the displacer piston possesses an advantage, either boosted or unboosted.

The displacer-piston tests have been extended to 2500 R.P.M. and a boost of 20 in. of mercury, the resulting fuel consumption and maximum indicated mean effective pressure for a clear exhaust being 0.34 lb. per indicated horsepower-hour and 235 lb. per square inch, respectively, at a

[1] The boost data presented in this paper have not been corrected for supercharger power.

maximum cylinder pressure of 1200 lb. per square inch (12). These high performance figures at 2500 R.P.M. result from a combination of one or more of the following factors: greater permissible maximum cylinder pressure, improved cylinder scavenging with inlet and exhaust valve overlap, and better response to boosting.

The improvement in thermal efficiency with boosting, as shown by the decrease in specific fuel consumption at the higher loads, is another notable feature shown by figure 14. This same tendency is again evident in figure 15 (22). In this figure the ratio of the absolute explosion pressure 0.004 sec. after ignition to the initial pressure is plotted against the ignition lags obtained when the fuel charge is injected into a constant-volume bomb containing air at several respective temperatures and densities.

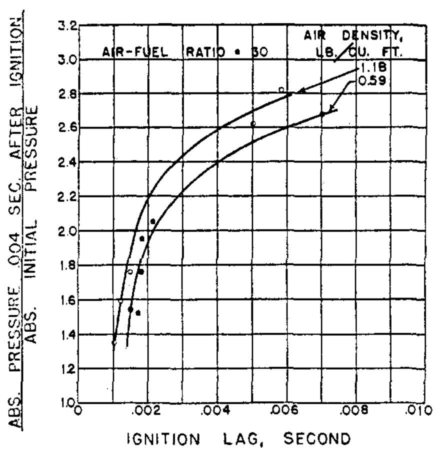

FIG. 15. Improvement in combustion with increasing air density

The lower density approximates those existing in a Diesel engine at normal injection advance angles, whereas the higher density corresponds to that in an engine at top center operated with a boost of about 10 in. of mercury. The improvement in combustion efficiency with increasing density is appreciable, particularly for the shorter ignition lags such as are permissible in engines.

CONCLUSION

From the data obtained on the high-speed Diesel engine at the laboratories of the National Advisory Committee for Aeronautics and at various other research institutions it is possible to draw certain definite conclusions and to outline the direction in which experimentation should go. One important conclusion is that the flat-disk combustion chamber offers the greatest promise. In the researches reported here, it was found that the flat-disk chamber gave the best performance with either the precombustion or the direct-injection chamber. In the precombustion chamber it was found that the best performance was obtained with most of the volume in the precombustion chamber of the flat-disk form. The outstanding performance obtained in Germany on the Junkers engine is also obtained with the flat-disk chamber. The chief difference between the three engines is in the nozzles used for the injection of fuel, the displacer-piston engine using a multiorifice nozzle, the precombustion chamber engine a single round-hole orifice nozzle, and the Junkers engine four impinging-jets nozzles. In each case the choice of nozzles has been the outcome of considerable experimentation and represents the best combination of combustion chamber design, air swirl, and nozzle design obtained so far. All three combustion chambers use a high-velocity air swirl, although the method of producing the swirl differs. The precombustion chamber engine has the disadvantage of an unscavenged space, which definitely limits its performance to a value below that of the other two types. More research is necessary on the effects of the design of the nozzle for the injection of fuel to determine, if possible, an optimum form of nozzle.

More data are necessary on the fuel distribution within the combustion chamber. There is little quantitative information on the distribution throughout the combustion chamber from the start of injection to the completion of the first 30 degrees of the expansion stroke. Finding the air–fuel ratio in different parts of the chamber during this time interval presents a difficult problem, but efforts to find it should be made. Tests should be conducted to determine whether the after-burning period is caused by inadequate mixing or by certain chemical phenomena. If it is chemical, it is possible that the use of certain fuel dopes will help to speed up the late burning.

Although it has not been brought out in this paper, there is a need of data on the effect of the rate of injection on the combustion efficiency. It is probable that to obtain widely different rates with adequate nozzle distribution, it will be necessary to use a unit injector. The unit injector, in which the piston of the injection pump is mounted as closely as possible to the injection nozzle, shortens the comparatively long time interval for pressure changes to be transmitted throughout the injection system.

The question of fuel rating, which is proving so bothersome in the gasoline engine field, appears to be fairly straightforward for Diesel engines. There are at least two methods of Diesel fuel rating, either of which is satisfactory, namely, the ignition lag as measured in the engine or the ignition lag as measured in a bomb at temperatures and densities equivalent to those in the engine.

As the speed of the Diesel engine is increased there is going to be a need for even closer coördination of the injection system and combustion chamber design. In addition, the injection system will need further development in order to inject the fuel in the shortened permissible time. Here again, it is probable that the unit injector will be at an advantage over the systems in general use at present, in which there is an injection tube of appreciable length between the injection pump and the injection valve.

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