REPORT No. 276

COMBUSTION TIME IN THE ENGINE CYLINDER AND ITS EFFECT ON ENGINE PERFORMANCE

By CHARLES F. MARVIN, Jr.
Bureau of Standards

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SUMMARY

As part of a general program to study combustion in the engine cylinder and to correlate the phenomena of combustion with the observed performance of actual engines, this paper, which was outlined by Mr. S. W. Sparrow, and the work undertaken at the request of the National Advisory Committee for Aeronautics, presents a sketchy outline of what may happen in the engine cylinder during the burning of a charge. It also suggests the type of information needed to supply the details of the picture and points out how combustion time and rate affect the performance of the engine.

A theoretical concept of a flame front which is assumed to advance radially from the point of ignition is presented, and calculations based on the area and velocity of this flame and the density of the unburned gases are made to determine the mass rate of combustion. From this rate the mass which has been burned and the pressure at any instant during combustion are computed.

This process is then reversed in an effort to determine actual rates of combustion and flame velocities from the pressures as recorded on indicator diagrams.

The effects of different rates of combustion on engine performance are then discussed and the importance of proper spark advance is emphasized.

INTRODUCTION

When the intake valve of a gasoline engine closes, it traps a definite weight of charge in the engine cylinder. If leakage by the piston and valves is neglected, this weight remains constant until the exhaust valve opens. During the short interval between these two events, the trapped gases undergo extremely rapid chemical and physical changes which enable them to do work on the piston. The amount of this work can be determined with fair accuracy and many tests show that it varies considerably for different engine designs and operating conditions.

Determinations of just what happens in the cylinder during the burning of a charge to cause the observed variations in engine performance are rendered very difficult and uncertain because of the complexity and speed of the chemical reactions and the presence of a number of operating variables, such as fuel characteristics, pressure, temperature, turbulence, and piston movement, all of which influence combustion and complicate its study.

Experiments with simple explosive mixtures under controlled conditions in bombs of various sorts have yielded important information regarding the phenomena of combustion, but there is need for supplementary data which will correlate the fundamentals of combustion with the observed performance of actual engines.

FLAME PROPAGATION IN THE ENGINE CYLINDER

As a first step in the visualization of what might happen in an engine cylinder when a charge is burned, a series of sectional views through the cylinder at the spark plug may be taken during combustion as shown in Figure 1. To make the picture as simple as possible it is assumed that the charge is homogeneous, that there is no turbulence or piston movement, and that the combustion chamber is a thin disk.

The small dotted portion of A (fig. 1) represents the original volume of that portion of the charge which burns during the first short interval of time "t" after the occurrence of the spark. As this increment of charge burns, its temperature rises and it therefore expands, pushing the flame front ahead of it, compressing the unburned charge, and causing a uniform increase in pressure throughout the cylinder. At the end of the time "t" the volumes of the burned and unburned gas, separated by the flame front, are shown at B.

For the sake of simplicity, it is assumed that during the second time interval "t" the flame proceeds into the unburned mixture at the same constant rate as during the first time interval. The amount of gas to be burned during the second time interval may be represented, before the flame front enters it, by the dotted section of C. After the second increment has burned and expanded, the position of the flame front and the relative volumes of burned and unburned gas are shown at D. Similar calculations give the position of the flame front after each successive equal interval of time.

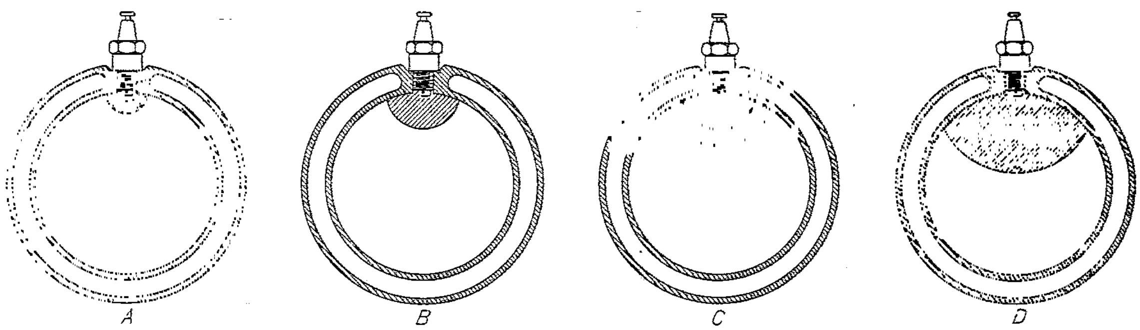

FIG. 1

RATE OF BURNING—MASS BURNED—PRESSURE RISE

The mass rate R_m at which the charge burns at any instant during combustion will be equal to the area, A, of the flame front multiplied by the velocity, V, with which the flame front is advancing into the unburned charge multiplied by the density, D, of the unburned charge or—

$$R_m = AVD \tag{1}$$

It is apparent that in Figure 1 the area of the flame front increases from zero to a maximum and then decreases to zero again as the flame advances across the cylinder from the spark plug to the opposite wall. It is also evident that the shape of the combustion chamber and the location of the ignition point may have a considerable effect on the area of the flame front and consequently on the mass rate of burning and the pressure rise. Consider, for example, the extreme cases of a spherical bomb ignited at the center, and a long thin tube of the same volume ignited at one end. In the spherical bomb, the flame can spread radially in all directions without interference, its area increasing as the square of the radial spread, until it reaches the walls of the bomb and the charge is entirely burned. In the tube, however, the flame can travel in only one direction and its area remains at a constant small value throughout the combustion. A much more rapid pressure rise and a shorter combustion time would be expected in the case of the bomb. In a somewhat similar manner, the number of ignition points would also affect the area of the flame front and hence the rate of burning.

Of course, the flame may not advance as a spherical surface in the actual engine cylinder. Its shape may be distorted from true sphericity by turbulence or stratification of the charge, by reflected pressure or sound waves, or by the movement of the piston. Under certain conditions or at some stage in the combustion, the whole of the unburned portion of the charge may be ignited simultaneously by compression or by incandescent particles. These possibilities invite further investigation.

The velocity at which the flame front advances into the unburned charge is known to be different for different fuels and mixture ratios. It also depends in some way on the pressure, temperature, and density of the burning gases. The presence of catalyzers or inert gases in the charge would also be expected to affect the velocity.

During combustion the unburned charge is undergoing continual compression by the increasing amount of burned charge so that its density increases steadily.

By making certain simple assumptions in regard to A, V, and D, it is possible to compute the mass rate of burning, R_m, and the total amount which has been burned at any instant during combustion for the simple case shown in Figure 1.[1] These quantities are plotted in Figure 2. Since the pressure rise is almost proportional to the mass burned, the solid curve in Figure 2 also indicates the type of pressure rise to be expected.

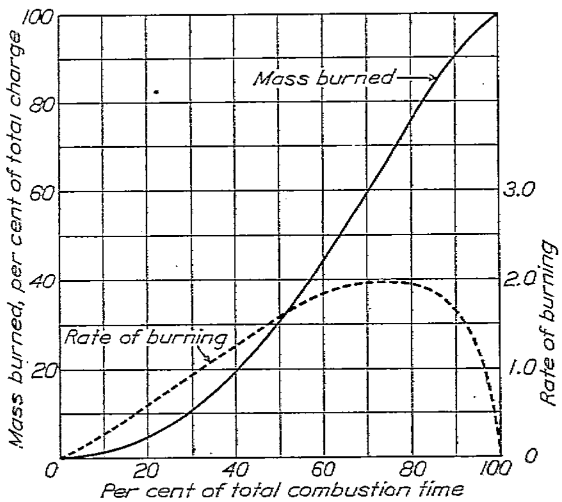

FIG. 2

ANALYSIS OF INDICATOR DIAGRAM

The fact that calculations of pressure rise can be made from a concept of an advancing flame front suggests that this process might be reversed and that the pressure rise as recorded on an indicator diagram might be made to yield information in regard to flame propagation in the actual engine cylinder.

A portion of a typical indicator diagram is shown plotted on logarithmic coordinates in Figure 3. The solid line shows actual pressures as measured by the indicator and the dotted line represents the pressures which would have been expected had the charge been instantly burned at upper dead center as in the theoretical Otto cycle. Theoretically and actually, compression is represented on the logarithmic plot by a straight line to the point where the spark occurs. Here the actual diagram begins to rise more sharply due to the burning of the charge. By the time the piston has reached upper dead center the actual pressure is at "c" instead of at "b" where it would have been had there been no burning. If it is assumed that the pressure rise above the straight compression line is proportional to the mass of charge burned, then the fraction of the total charge which is actually burned at upper center would be equal to $\frac{c-b}{d-b}$. In a similar manner the fraction burned at any other instant during combustion can be computed from a similar ratio of pressures. At "e" combustion is complete and expansion of the burned gases is represented by a straight line.

Seven indicator diagrams,[2] taken under different operating conditions as listed in Table I, were analyzed by this method and the results are plotted in Figures 4–7. The engine

[1] Details of these calculations are given in Appendix I.

[2] These diagrams were made in the "altitude chamber" at the Bureau of Standards with a balanced diaphragm pressure indicator described in N. A. C. A. Report No. 107.

on which these indicator cards were taken has a cylindrical combustion chamber similar to that pictured in Figure 1 except that in the engine two plugs are used, one on each side of the cylinder. As would be expected, the general shape of the curves, Figures 4–7, is similar to that of the theoretical curve shown as a solid line in Figure 2, but because of the various operating conditions there are variations in the rate of burning and the total time required for complete combustion.

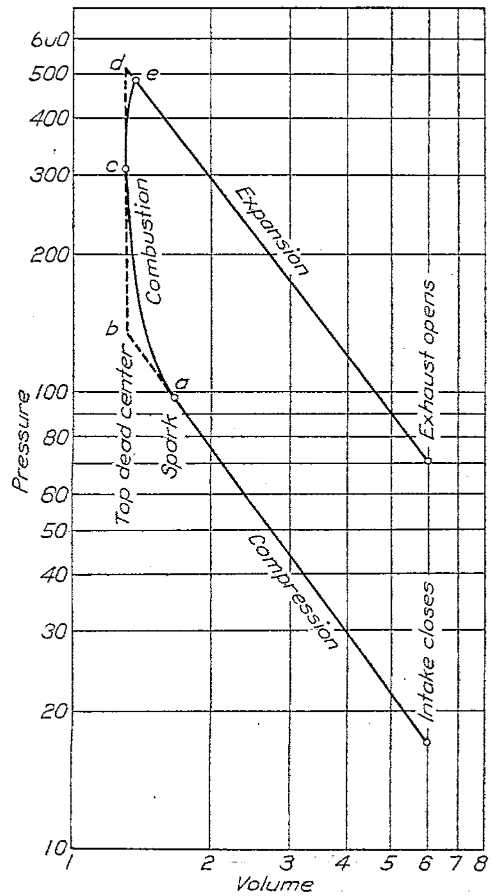

FIG. 3

As shown in Figure 4, combustion is more rapid when the charge is fired by two plugs than when one plug only is used. This is the natural result of having two flame fronts originating at opposite sides of the cylinder.

Figure 5 compares a lean mixture with a mixture producing maximum power. In this case the more rapid burning of the maximum power mixture must be due to a higher velocity of flame propagation rather than to a difference in the flame area, since two plugs were fired in both cases.

The curves in Figure 6 were made at different "altitudes," the density of the charge being the chief operating variable. The velocity of flame propagation is evidently higher for the denser charges. The very slow burning as represented by the curve for 25,000 feet altitude is partly accounted for by the fact that the mixture ratio in this case happened to be very lean.

The load, speed, charge density, and mixture ratio were all different for the two runs shown in Figure 7, and it would be difficult to isolate the effects of these various factors on the curves.

Returning to Figure 4, an attempt was made to estimate approximately, by means of Formula (1), the actual flame velocities at different instants during combustion.

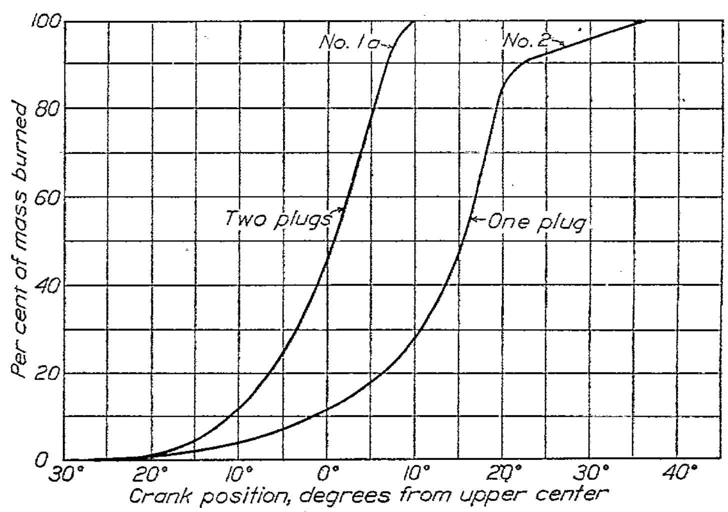

FIG. 4

Prior to ignition, conditions in the engine cylinder were almost identical for the two runs represented by the two curves in Figure 4. The weights, pressures, temperatures, and compositions of the two charges were practically the same and the engine was operated at the same

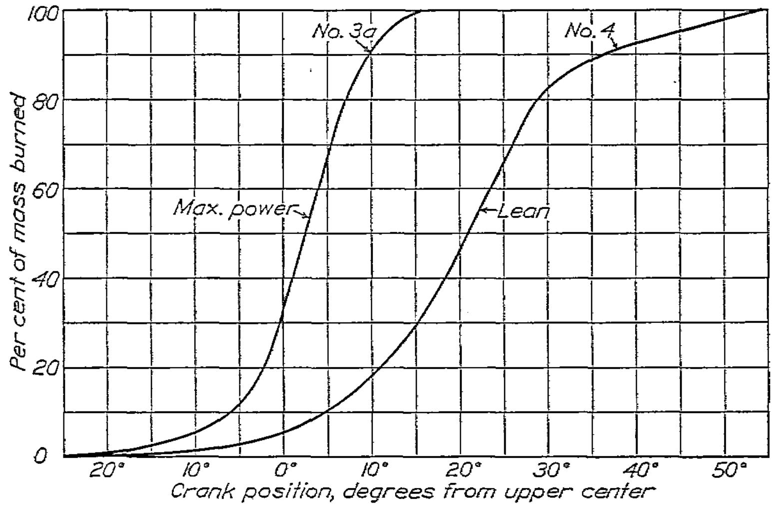

FIG. 5

speed and with the same spark advance in both runs. The flame velocities in the two cases should therefore be equal at the start of combustion and would be expected to vary only with temperature, pressure, and charge density as the burning progressed. For an appreciable time near top dead center the volume change is negligible and pressure, temperature, and charge density are proportional. During this period it would be expected that flame velocities in both

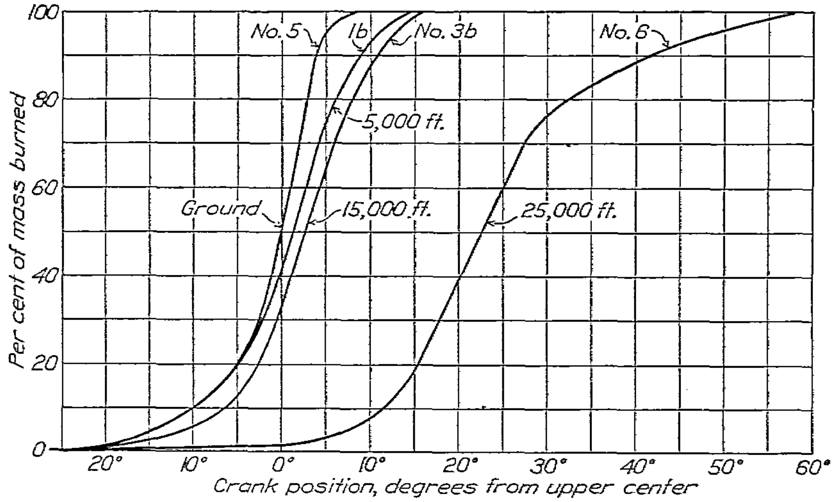

FIG. 6

runs would vary in the same manner with pressure. If it is assumed that the flame advances radially from the point of ignition, flame velocities can be estimated for the two cases by means of Formula 1.[3] The calculated velocities are plotted in Figure 8, and contrary to expectations, do not vary in the same manner with pressure for the two cases. The most plausible explanation for the fact that all of the points plotted in Figure 8 do not fall on the same smooth curve would seem to be that the areas of the flame fronts in the actual engine are not such as would be

[3] The method of calculating velocity is given in detail in Appendix II.

expected from the theoretical picture in Figure 1, and that the estimated velocities therefore reflect the errors involved in assuming this radial advance.

COMBUSTION TIME

It is obvious that the time required to completely burn the charge must affect the power and specific fuel consumption of an internal-combustion engine. Data have been presented

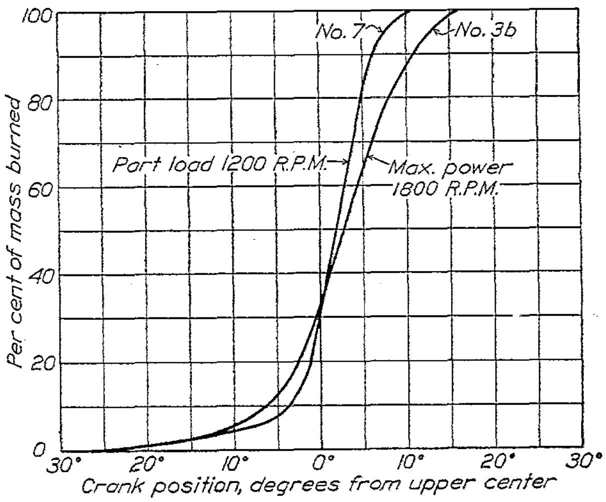

FIG. 7

from time to time showing this effect to be surprisingly small. For the most part these presentations have dealt with what happened and have assumed that the reader knew why it happened. While, in general, this assumption may be justified, it is believed that a somewhat detailed discussion of the reasons underlying this effect may be of value.

In the theoretical Otto cycle, it is assumed that the charge is completely burned at upper

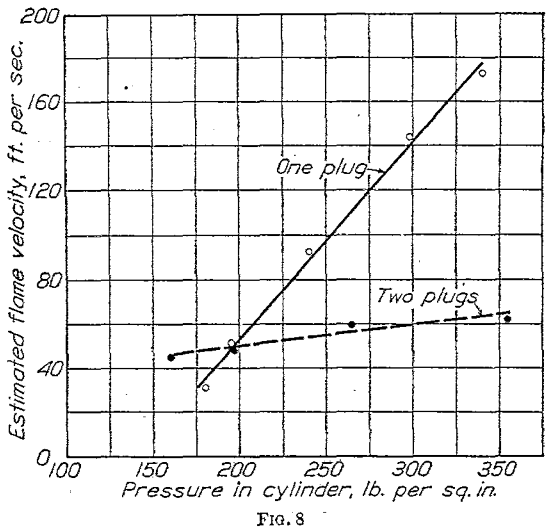

FIG. 8

dead center. The efficiency under such conditions depends only on the expansion ratio which is the ratio between the volumes above the piston at the end and at the beginning of the power stroke. The efficiency is given by the expression

$$E = 1 - \frac{1}{r^{n-1}}$$

where E is what is commonly known as the air cycle efficiency, r is the expansion ratio and n is the ratio between the specific heats at constant pressure and constant volume, which for air is approximately 1.4.

If the ignition of the charge is delayed, but the charge is still assumed to burn completely in a single instant, the efficiency obtainable will be less than was the case when the charge was burned at top center, as the volume above the piston will be greater at the time of combustion than when the charge is burned at top center, and hence the expansion ratio will be less.

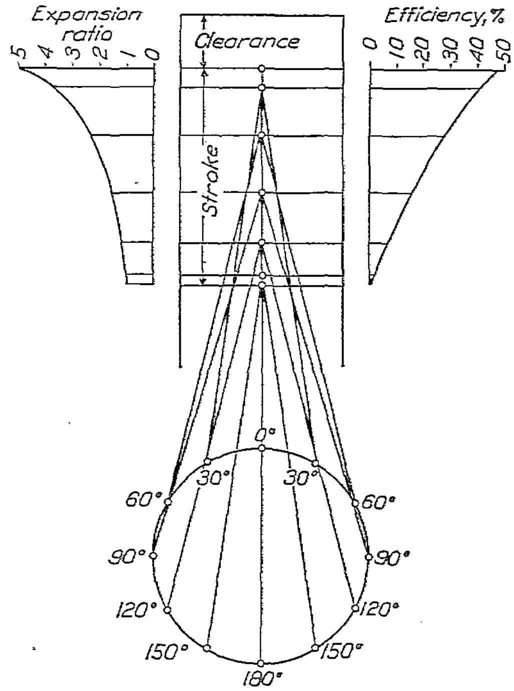

Fig. 9

If now the charge is burned—again instantly—with the piston in the position last considered but before center instead of after center, additional work will be required to compress the burned charge to dead center. However, in the absence of heat loss, this energy will be returned on the power stroke and the effect on engine performance will be the same as in the previous case where the charge was burned after center.

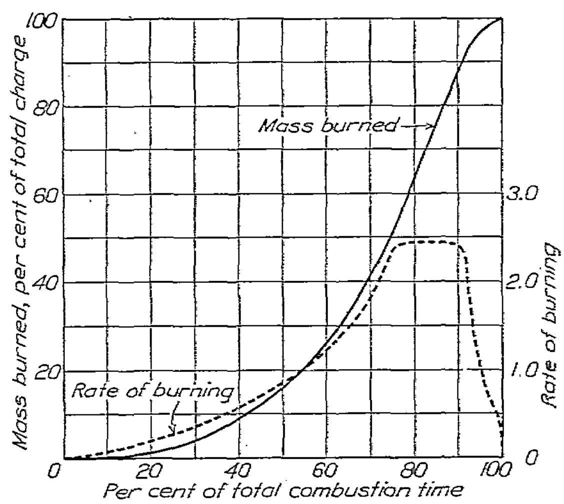

Fig. 10

In the actual engine the charge is, of course, not burned instantly but combustion continues over an appreciable part of the cycle. However, the efficiency obtainable from any small portion of the charge may be calculated as outlined above from the position of the piston at the time that portion is burned.

Figure 9 shows the positions of the top of the piston corresponding to several crank positions for a typical engine. If the expansion ratio based on upper dead center is 5 to 1, then the expansion ratios for portions of the charge burned at other positions of the piston are shown on the left in Figure 9. The air cycle efficiencies corresponding to these expansion ratios are plotted on the right in the same figure.

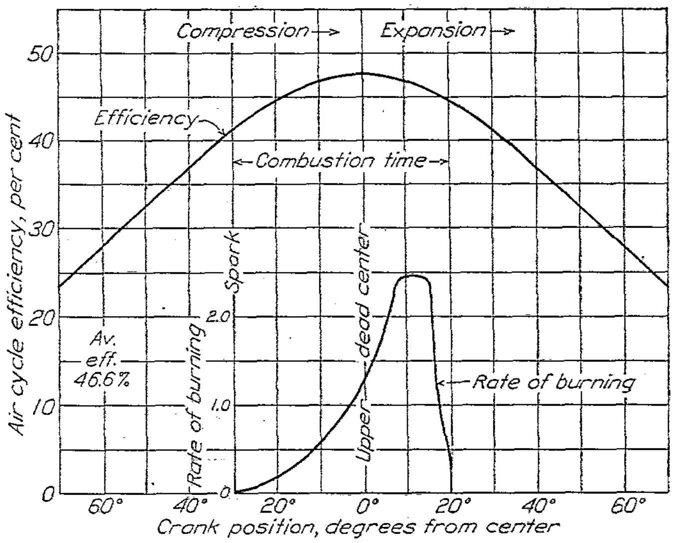

FIG. 11

It is now necessary to determine how much of the charge is burned at each position of the crank, or, in other words, the rate of burning at various stages of the combustion. This information can be obtained from the curves in Figures 4–7. Curve No. 1a in Figure 4 is typical of normal combustion, and to make the case perfectly general this curve has been replotted as the

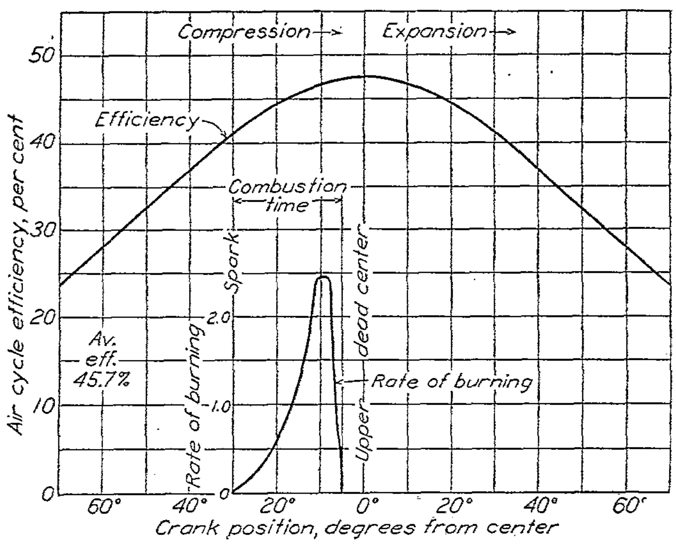

FIG. 12

solid line in Figure 10. The slope of this line is plotted as a dotted line in the same figure and represents the mass rate of burning at each instant during combustion. From this curve the amount of charge burned during any small increment of crank travel may be determined.

From the information in Figures 9 and 10 the average air cycle efficiency for the whole combustion can be computed, as illustrated in Figure 11. The upper curve in the figure gives the air cycle efficiencies corresponding to different crank positions as obtained from Figure 9.

The lower curve is the rate of burning taken from Figure 10, it being assumed that the charge is completely burned during 50° of crank travel and that the spark advance is 30°. The average efficiency for this particular operating condition is obtained by multiplying the amounts of charge burned during successive small increments of crank travel by the corresponding efficiencies and dividing the sum of the products so obtained by the total amount of charge burned, the result in the case being 46.6 per cent.

Assume now that the total time required to burn the charge is only one-half as great as before; that is, combustion is completed in 25° of crank travel instead of 50°. With a 30° spark advance the efficiency and rate curves would be as shown in Figure 12, and the average efficiency for this case is 45.7 per cent.

If combustion required 100° of crank travel and a spark advance of 30° was still retained, the curves would appear as in Figure 13 and the average efficiency would be only 35.7 per cent. So far as this analysis is concerned the reduction or increase in the combustion time as shown in Figures 12 and 13 may result from either a change in actual time of burning due to a change in fuel, mixture ratio, or operating conditions, or merely from a change in engine speed.

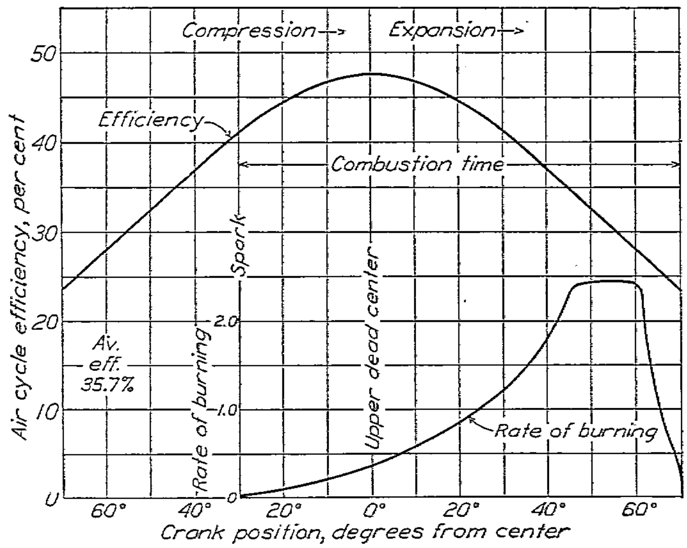

FIG. 13

SPARK ADVANCE

It is quite evident that 30° is too much spark advance for the rapid combustion shown in Figure 12 and not enough for the slow combustion shown in Figure 13. These two figures demonstrate clearly the need for a variable spark advance which can be adjusted to obtain the greatest power and economy possible with each different rate of combustion. As the spark advance does not affect the amount of charge that enters the cylinder, the spark advance that gives the highest average air cycle efficiency will give both maximum power and minimum specific fuel consumption.

Average efficiencies for values of spark advance other than 30° can be determined by shifting the rate curves along the horizontal axes of the charts to each desired spark advance and repeating the calculations described above. For example, Figure 14 shows the rate curve for the 50° combustion time set for values of spark advance of 60°, 40°, 20°, and 0°. The average efficiency for each setting is plotted as a circle against the corresponding spark advance. The peak of the curve drawn through these points gives the optimum spark advance and the maximum efficiency obtainable with this combustion time. This average efficiency curve is reproduced in Figure 15 where it may be compared with similar curves computed for combustion times of 25° and 100° of crank travel and for the theoretical case of instantaneous combustion.

The optimum spark advance and the maximum efficiency corresponding to each of the several combustion times have been taken from the curves in Figure 15 and plotted as circles against combustion time in Figure 16. It will be observed that the spark advance required for

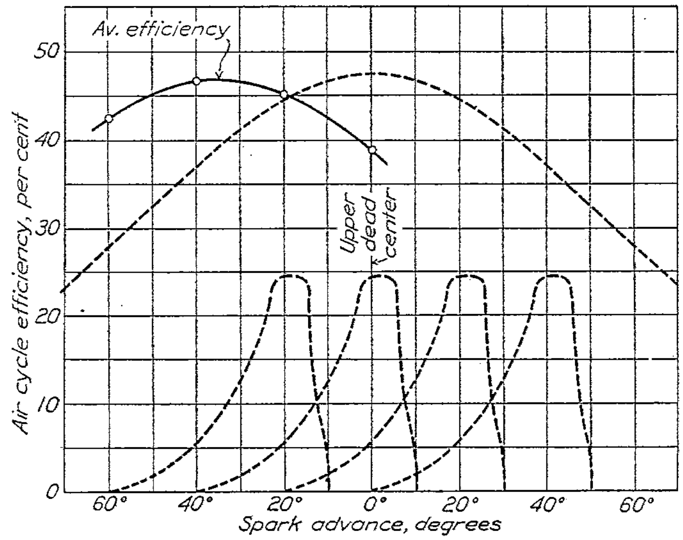

FIG. 14

highest efficiency is radically different for the different combustion times and agrees very well with Upton's rule (Reference 1), based on experimental data, viz: "The optimum spark advance is such that the half pressure rise occurs at the dead center, and that this stage of the pressure rise occurs practically at 75 per cent of the explosion time after ignition." This rule is represented by the dotted line in Figure 16. This figure also shows that the efficiencies obtainable with widely different combustion times are not greatly different, provided the optimum spark advance is used in each case. This also is in agreement with experimental results.

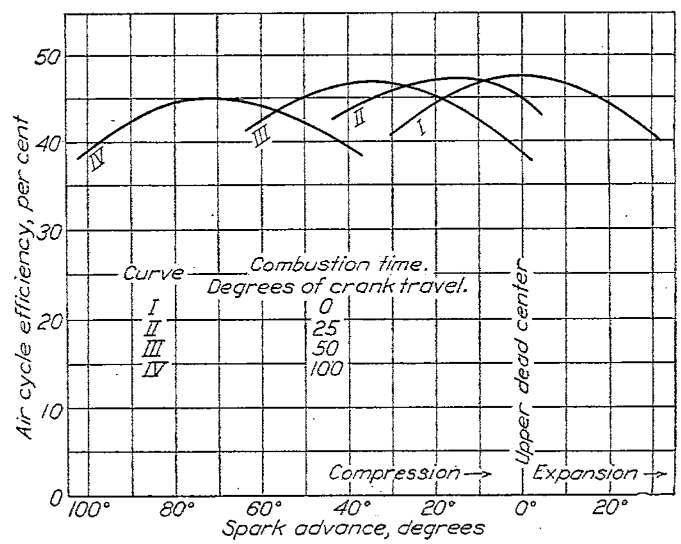

FIG. 15

RATE OF BURNING

Throughout the foregoing discussion of the effect of combustion time and spark advance on engine performance the charge was assumed to burn at a rate represented by the dotted curve in Figure 10. For actual engines the shape of this rate curve will vary somewhat, depending on engine design and operating conditions. However, it may be demonstrated by substituting a different rate curve in Figures 11–14, that considerable difference in the shape of the rate curve will have very little effect on engine performance provided the proper spark advance is maintained. Thus the rate curve shown in Figure 2 is quite different in shape from the one shown in Figure 10, but the maximum efficiencies computed on the basis of these two curves differ by less than 1 per cent.

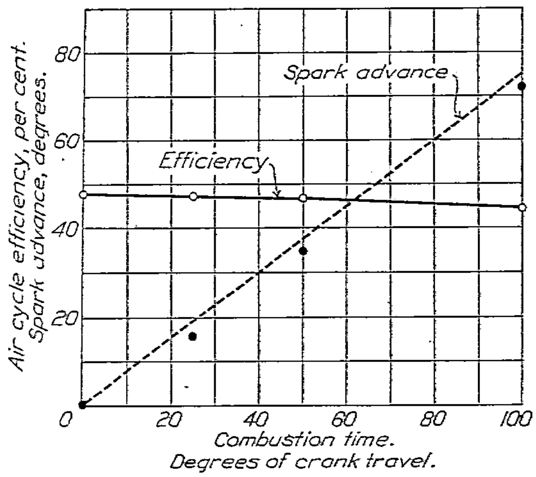

FIG. 16

CONCLUSIONS

From a theoretical standpoint, from the analysis of indicator cards, and from observation of actual engines, it is evident that the time required for complete combustion of a charge varies considerably with different engine designs and operating conditions. It is also known that the rate of burning at any particular instant after ignition may differ even in cases where the total combustion time is the same. Both the rate of combustion and the time required for complete burning of the charge have an effect on the power and economy of the engine. The greatest power and economy are, of course, attained in the theoretical case of instantaneous combustion at upper dead center. The slower combustions obtainable in practice may result in considerably reduced efficiencies if improperly timed with respect to piston position. However, the maximum theoretical performance may be approached very closely with a wide range of combustion times and rates, provided the proper spark advance is maintained for each particular condition.

REFERENCE

1. The Society of Automotive Engineers, Transactions, Vol. 18, Part II, p. 117.

BUREAU OF STANDARDS,
Washington, D. C., *April 2, 1927*.

TABLE I.—*Operating conditions during test of 300-horsepower, 8-cylinder aviation engine in the altitude chamber at the Bureau of Standards*

Curve numbers	1 (a & b)	2	3 (a & b)	4	5	6	7
"Altitude"—ft	5, 000	5, 000	15, 000	15, 000	Ground.	25, 000	15, 000
Speed—R. P. M	1, 800	1, 800	1, 800	1, 800	1, 800	1, 800	1, 200
Load	Maximum.	Maximum.	Maximum.	Maximum.	Maximum.	Maximum.	Part.
Spark advance—deg	26. 5	26. 5	26. 5	26. 5	26. 5	26. 5	26. 5
Number of plugs firing	2	1	2	2	2	2	2
Air density at carburetor, lb./cu. ft	0. 068	0. 066	0. 047	0. 048	0. 081	0. 034	0. 048
Mixture ratio	12. 25:1	12. 8:1	12. 9:1	14. 7:1	12. 6:1	15. 4:1	8. 2:1
Weight of charge per hour—lb	2, 293	2, 288	1, 550	1, 553	2, 757	1, 094	624

APPENDIX I

METHOD OF CALCULATING RATE OF BURNING FOR RADIALLY ADVANCING FLAME FRONT SHOWN IN FIGURE 1

A circle 10 inches in diameter was drawn and was divided into sections one-tenth of an inch wide by circular arcs of increasing radius drawn about an "ignition point" on the circumference of the 10-inch circle. The area of each of these sections was determined by measuring the average length of the section and multiplying by one-tenth inch. A curve was then plotted showing the area (or volume) behind the flame for each different flame position or radius from the point of ignition.

It was assumed that the cylinder was filled with a homogeneous charge at 600° C. absolute and that during the first interval of time "t" after the occurance of the spark the first one-tenth inch section of the charge was burned. For purposes of calculation this portion of the charge was assumed to burn at constant volume, its temperature being raised 2,700° in the process, the final temperature being 600 + 2,700 = 3,300°. The ratio of pressures before and after burning will be equal to the ratio of temperatures or

$$\frac{p_u}{p_b}=\frac{600}{3{,}300}$$

where—

p_u = original pressure in the cylinder before ignition
p_b = pressure of burned gases before expansion

The burned gases are then assumed to expand, compressing the unburned gases ahead of them, according to the equation $pv^{1.3}$ = constant, until the pressure throughout the cylinder is uniform. (The exponent 1.3 was selected because it is the approximate mean between an exponent for compression of 1.39 and an exponent for expansion of 1.22 as determined from indicator cards.) The following equations will then apply:

$$p_b v_b^{1.3} = P V_b^{1.3} \text{ expansion of burned gases.} \tag{1}$$

$$p_u v_u^{1.3} = P V_u^{1.3} \text{ compression of unburned gases.} \tag{2}$$

where—

v_b = original volume of burned gases before expansion.
v_u = original volume of unburned gases before compression = $V - v_b$.
V = total volume of cylinder.
V_b = final volume of burned gases after expansion.
V_u = final volume of unburned gases after compression = $V - V_b$.
P = final uniform pressure throughout the cylinder.

The simultaneous equations (1) and (2) may be reduced to the following general form:

$$\log\left(\frac{V}{V_b}-1\right)=\log\left(\frac{V}{v_b}-1\right)+\frac{\log\frac{p_u}{p_b}}{1.3}$$

This equation may be solved for V_b the volume of the burned gases after expansion, and the position of the flame front after the first interval of time "t" may thus be determined.

In the next equal interval of time the flame will proceed into the unburned mixture another one-tenth of an inch, constant volume burning being assumed. The increment of volume corresponding to this second one-tenth of an inch of movement can be found from the curve of volumes and flame positions. Since this new increment of volume contains compressed gas, its density must be known before its mass can be calculated. The total mass of unburned gas before the second increment is burned is $V-v_b$, mass being proportional to volume in the charge at original conditions. The volume of this total mass of unburned charge after compression by the first burn is $V-V_b$, and its density will therefore be $\frac{V-v_b}{V-V_b}$. Multiplying this density by the second increment of volume will give the mass burned during the second time interval. Adding this to the mass burned during the first interval will give the total mass burned after two intervals of time. In the calculations it is now assumed that this total mass is burned as a single unit. It thus becomes the new v_b in the equation and its volume after expansion is determined as in the case of the original increment. The computations are repeated for equal time intervals until the flame has traveled entirely across the cylinder. Such computations supply the data for plotting both the total mass burned and the mass rate of burning.

APPENDIX II

METHOD OF ESTIMATING FLAME VELOCITIES AT ANY INSTANT DURING COMBUSTION

$$R_m=AVD$$

R_m = mass rate of burning in pounds per second.
A = area of flame front in square feet.
V = velocity of flame front with respect to unburned gas in feet per second.
D = density of unburned gas in pounds per cubic foot.

Take, for example, curve No. 2 (fig. 4). Data taken during the engine test make it possible to transpose the values of per cent of mass burned to pounds burned and to express crank position as seconds of time after ignition. If curve No. 2 is replotted on these coordinates, its slope at any point will be the mass rate of burning (R_m) in pounds per second at that point.

The density of the unburned charge at any instant will be equal to the weight of the unburned charge divided by its volume. Consider, for example, the point at which 80 per cent of the charge is burned, the crank position at this point being given in Figure 4. The weight of the unburned portion is 20 per cent of the total weight of charge in the cylinder. If no burning had taken place this portion would occupy 20 per cent of the total volume above the piston, and its pressure would be the engine compression pressure for the given piston position as determined from the straight compression line on the indicator card. The burning of 80 per cent of the charge has, however, compressed the unburned 20 per cent to a smaller volume and a higher pressure according to the equation $pv^{1.4}$ constant.* This higher pressure is found on the indicator card and the final volume can therefore be calculated.

With the relative volumes of burned and unburned gas known, the position of the flame front in the cylinder can be determined and its area computed from the cylinder dimensions. In making these calculations the flame is assumed to advance across the cylinder in the manner illustrated in Figure 1. In the case of curve No. 1a, this advance takes place simultaneously from both ends of the diameter.

With values of R_m, D, and A thus determined the equation $R_m=AVD$ is solved for V.

* An average exponent of 1.39 for compression has been determined from indicator cards.