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INCREASING THE CYLINDER PERFORMANCE

OF THE GAS ENGINE

Thesis for Degree of M. E.

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1919

1. During the past few years the demand for gasoline has increased far more rapidly than the supply of raw material from which it is produced. Consequently, there has been a general change in refinery practice with the production of so-called heavy gasolines, which are actually of relatively low volatility. Although this change in quality is not necessarily deterioration, it has brought to the attention of designers and users of internal combustion engines, problems that did not exist in the past. The main problem has been that of carburetion. Other troubles, such as the rapid formation of carbon and the dilution of the lubricating oil by the less volatile parts of the fuel are the result of poor carburetion or rather incomplete vaporization of the fuel.

The fuels marketed at present as gasoline are produced in various ways and are as follows:

1. Straight refinery gasoline
2. Cracked gasoline
3. Casing head gasoline blended with kerosene
4. Casing head gasoline blended with heavy straight refinery gasoline
5. Casing head gasoline blended with naptha

2. These fuels can be used for starting gas engines under practically all conditions as about twenty per cent of the fuel will vaporize at temperatures below 100 or 200° F. If means were provided for completely vaporizing the fuel soon after the engine gets started, they should give good service. The difficulty lies in the fact that in order to vaporize the least volatile part of the fuel and have a "dry" mixture enter the the engine, temperatures ranging from 320 to 550° F. are necessary.

This temperature is commonly called the "end point" of the fuel. In order to properly vaporize these fuels, the mixture of gasoline vapor and air should enter the cylinder at a temperature near if not at that of the end point of the fuel being used. Few carburetors or inlet manifolds give any such temperature. This is evidenced by the fact that passenger cars require changing of the lubricating oils in the crank case at least once a month, and trucks about every other week, because of the excessive dilution of the oil and consequent lubrication. The remedy, of course, is to heat the mixture more highly. In the case of automobiles, the economy is increased somewhat by by doing this but there is a falling off in maximum or full load power of the engine. It should be remembered that this discussion relates to full load conditions in all cases.
H. L. Horning of the Waukesha Motor Co. stated that "It is conservative to estimate a reduction in horsepower of 10 to 20 per cent because of such heating, but economy of fuel is increased considerably" in the March 22, 1917 issue of"The Automobile", now called "Automotive Industries." It is evident that heating the mixture entering a gas engine cylinder will reduce the volumetric efficiency. There are various proposed definitions for volumetric efficiency. In this paper, it will be considered as the ratio of volume of air and fuel drawn into the cylinder at atmospheric pressure and 62$^{\circ}$F. to the "suction volume." In the case of present engines, suction volume is represented by the piston displacement. Later on it will become clear why this term is used in place of the term "piston displacement."

3. In the average case, it is found that the mixture

in the cylinder of present engines at the beginning of the compression stroke is at a temperature of $390^{\circ}$ C abs. This figure is the result of investigations by three competent authorities quoted by O. A. Malychevitch in the September 19 and 26, 1918 issues of "Automotive Industries." This is equivalent to a temperature of $702^{\circ}$ F. absolute. This temperature is the temperature resulting when the new cool charge is heated by coming in contact with the cylinder walls and by mixing with the hot gases left in the clearance space at the end of the previous exhaust stroke. If it be considered that the charge entering the cylinder when these temperatures are obtained is at a temperature of $100^{\circ}$ F.; and that, in order to vaporize present fuels, a temperature of $400^{\circ}$ F. is needed, less mixture will be drawn in. The ratio of the amounts of mixture (by weight) will be equal to the ratio between the absolute temperatures of the cylinder contents at the beginning of the compression stroke. This ratio is found as follows.

4. With the charge entering the cylinder at a temperature of $100^{\circ}$ F. with grades of fuel as used in the past, the temperature of the cylinder contents becomes $702^{\circ}$ F. abs at the beginning of the compression stroke. The temperature of the charge has been increased $702 - (460 + 100) - 142^{\circ}$ F.

5. The mixture entering the cylinder at $400^{\circ}$ F. will have the temperature raised the same amount, thus its temperature at the beginning of the compression stroke will be $460 + 400 + 142 = 1002^{\circ}$ F. abs. The ratio between the absolute temperatures of the charges is $\frac{702}{1002}$. Therefore $\frac{702}{1002} \times 100 = .69\frac{1}{2}$ per cent as much mixture enters the cylinder when the charge is heated. This charge will contain only $69\frac{1}{2}$ per cent as much fuel,

and, therefore, the full load power of the engine will be reduced by about 30 per cent simply by pre-heating the charge as shown.

6. Therefore, the proper use of present fuels means that an engine of a given size will give considerably less horsepower. Nor is there any relief in sight as regards fuels. There are no great undeveloped fields that can be drawn on, and even if there were, it is a question as to whether or not it would be advisable to change the fuel back to that which was used in the past. National economy of resources seems to demand that the present grades be used. In fact, the U. S. Bureau of Mines is recommending a so-called "straight run" fuel which shall include the present grades of gasoline and a good per cent of the present kerosene and possibly some distillate. Present indications are that this will gradually come about in the relatively near future. Temperatures of the mixture will then have to be such that the temperature of the gases at the beginning of the compression stroke will be held at 400° F. at least, if not 500° F. in order to insure good operation. Evidences of what this will mean have already come to light in the case of a great many tractors that could not be made to pull their rated load on kerosene, the owner being forced to use gasoline in order to do so. It has been found in many cases that an engine delivered approximately three-fourths as much power on kerosene as it did on gasoline. In other words, a given size cylinder produces less horsepower. The weight in an ordinary commercial engine cannot be reduced proportionately, since many of the dimensions of these engines are determined by considerations having to do with foundry practice rather than strength. Then too, gasoline is used in order to warm up an

engine before attempting to use kerosene, and the engine should be designed to take care of the higher explosions incident to operating with the cool mixture. It is probable, also, that higher explosions would result in the engine after being warmed up in case it missed a few times. The cylinder would then be filled completely with a combustible charge instead of one diluted with exhaust gases so a higher explosion pressure would have to be cared for in the design. Suppose that a given engine for automotive use weighs sixteen pounds for each horsepower developed when operating with a relatively cool mixture. Certain parts, such as the connecting rod, crank shaft and possibly the flywheel might be made lighter. Perhaps one pound could be removed in this way figuring on the output being the same as at present. The engine would then weigh fifteen pounds per horsepower developed as rated on the cool mixture, but when operating on the heated mixture the actual weight per horsepower developed would be close to twenty pounds per horsepower developed if the loss be considered as 25 per cent. Therefore, it is probable that the "weight of engine per horsepower developed" has increased and that it will increase further unless means other than the makeshift of increasing the speed is provided to counteract the tendency. In other words, some means should be provided to cause the output from a given cylinder to be increased and, if possible, to bring about conditions whereby the parts are subjected to the maximum strains at all times instead of only at the time the engine is being warmed up or when it misses once in a while, as in the case of a hit-and-miss stationary engine.

7. By increasing the compression pressure to the limit for the fuel, the power output will be increased somewhat

but the other conditions remain. The volumetric efficiency is increased by the use of higher compression pressures, but in any case, the charge consists of a goodly percentage of burned gases from the previous cycle mixed with the new charge, so that the explosion pressures while running are lower than when starting or after one or two "misses." It is found in the average case that the volumetric efficiency is seldom over 75 per cent and that 65 per cent to 70 per cent is more often obtained.

8. By using water injection, the compression can be carried above that permissible without. There are cases where it has been carried as high as 150 pounds per square inch with gasoline as the fuel. This is at least 60 pounds per square inch above that at which pre-ignition is a danger without the use of water, even in a refined design. The use of water would improve the volumetric efficiency a great deal, but it should not be resorted to. Few users are expert enough nor are they willing to care for the necessary adjustments in most cases. On automobiles, trucks, tractors and airplanes, it is practically impossible to use water as it adds considerably to the weight of the vehicle even if it were always easily procurable in sufficient quantities.

9. However, the author believes that the real remedy lies in increasing the volumetric efficiency, but by means other than increasing the compression pressure which entails the chance of trouble from pre-ignition in case of the slightest trouble in the cooling system, or by the use of water. In 1913, the Automobile Club of America made some rather extensive tests on a Moline Knight four-cylinder automobile engine. The cylinder dimensions of this engine were 4" bore and

6" stroke. Among the tests were some made to ascertain the volumetric efficiency. The air entering the carburetor was measured by means of a venturi meter from which the air flowed to an air box surrounding the carburetor. Various readings were taken as indicated in the following table. Tests were also made with the carburetor on but with the engine driven by a dynamometer, and also without the carburetor, the engine being driven in the same manner. Since we are concerned with the actual performance, i.e. the operation with the carburetor on and the engine running under its own power, these latter two tests are not shown.

Volumetric efficiency of Moline Knight 4 x 6 automobile engine. Data taken from Table 4 of the report of the Automobile Club of America test made in December 1913.

| Test | Duration | R.P.M. | B.H.P. | Vol. eff. |
|---|---|---|---|---|
| $A_3$ | 6 min. | 1715 | 52.5 | 70.5 |
| $B_3$ | 5 | 1523 | 51.1 | 71.4 |
| $C_3$ | 5 | 1318 | 45.1 | 75.9 |
| $D_3$ | 5 | 1108 | 38.2 | 78.0 |
| $E_3$ | 5 | 902 | 30.8 | 79.0 |
| $F_3$ | 4 | 701 | 25.1 | 78.6 |
| $G_3$ | 5 | 491 | 16.8 | 72.9 |
| $H_3$ | 5 | 295 | 9.5 | 51.0 |

10. The volumetric efficiency of poppett valve engines of the automobile type is generally much lower than that of the Knight engine because the valve openings are smaller and less direct than those in the latter type. Where particular care has been taken in laying out the cylinder and valve

arrangements, higher volumetric efficiencies do obtain as in the case of modern airplane motors. Since noise from valve parts is not objectionable in these engines, quick opening and closing cams may be used, thus making the valves more efficient. Then too, the cooling facilities are much better, in that the water circulation is much better. The cylinder is generally of steel, consequently it may be relatively thin, so that the inner surface is not as hot as the inner surface does not have as great an effect in heating the incoming charge from the carburetor. However, the volumetric efficiency even under these conditions cannot be conceivably greater than 85 per cent.

11. Since the tests on the Moline Knight were made over five years ago when a good quality of fuel was easily procurable, it was unnecessary to pre-heat the charge to insure vaporization of the fuel. It is probable that, if tested today, the figures would be somewhat different because pre-heating is absolutely necessary unless some method of direct injection of the fuel is used. It is interesting at this point to consider the probable increase in power of the Moline Knight if the volumetric efficiency were 100 per cent. From test $B_3$, it is seen that the volumetric efficiency is 71.4 per cent at 1523 R.P.M. The probable increase in power is then $\frac{100 - 71.4}{71.4} \times 100 = 40\%$.

12. Taking 85 per cent as the volumetric efficiency of the best aviation engine, the probable increase in power because of increasing the volumetric efficiency to 100 per cent is $\frac{100 - 85}{85} \times 100 = 17.67\%$.

13. From the last two paragraphs, it is apparent that there is a great deal of lee-way in which to set to work to fit engines with new accessories so that more mixture is forced

into the engine, or to redesign engines so that they may draw in a greater charge during their intake stroke. As it stands at present, approximately one-fourth of the intake stroke of the piston is used in expanding the gases left within the clearance space at the end of the exhaust stroke and is therefore ineffective in drawing in a new charge. Air scavenging, precompression, and the author's design (which incorporates an auxiliary piston in the cylinder) are ways in which this condition may be partially if not entirely corrected.

14. Air scavenging might be worked out as follows: A third valve could be arranged in the cylinder in such a position that air could be pumped through it into the cylinder, blowing the products of combustion from the previous cycle out through the exhaust valve, thus leaving the clearance space filled with air instead of with a spent or burned gases. A rich mixture could then be drawn into the cylinder from the carburetor to mix with this air and to form a correctly proportioned mixture for the next explosion. A possible valve timing that suggests itself is-

| | | |
|---|---|---|
| Exhaust opens | $45^{\circ}$ to $55^{\circ}$ | before outer center |
| Air valve " | $45^{\circ}$ to $30^{\circ}$ | before inner center |
| Exhaust closes | $15^{\circ}$ to $25^{\circ}$ | after inner center |
| Air valve " | $15^{\circ}$ to $25^{\circ}$ | after inner center |
| Inlet opens | $18^{\circ}$ to $30^{\circ}$ | after innar center |
| Inlet closes | $30^{\circ}$ to $45^{\circ}$ | after outer center |

The auxiliary air pump or blower would need to have a capacity of at least double the clearance volume in order to insure through scavenging. This means that its capacity would be 40 per cent to 60 per cent as great as the piston displacement of the

engine itself. On the face of it, it appears that the increase in weight of the complete power plant would counterbalance the increase in power, to say nothing of the extra complication and attention necessary, due to the pump and extra valve parts. This type, therefore, will probably not come into any great use.

15. A pump or blower with a capacity greater than the piston displacement of the engine might be arranged to pre-compress the charge. Thus more mixture would be forced into the cylinder. Because of the possibility of loss of fuel from condensation, a pump, if used, would probably be arranged to force the air through the carburetor. The carburetor bowl would have to be sealed and connected to the inlet pipe else there would be difficulty in procuring fuel. On the other hand, a blower would assist in breaking up the fuel and could be placed between the carburetor and engine to good advantage on this account. The pump would add greatly to the weight of the power plant and would not be used on that account. It is conceivable that the blower might be a drag on the engine rather than an aid to it at low speeds. It is also a question whether or not the blower could be made so efficient that there would be any net gain.

16. The author has worked out a design which he believes will solve the problem with a minimum of complication. In this paper, he will show his design of a 5" bore by 7" stroke six cylinder aviation engine using sliding sleeve valves to control the gases in a manner somewhat similar to the Knight engine which most engineers are now familiar with. As already stated, an auxiliary piston is also used. This piston is

operated by an eccentric or crank which rotates at one-half the speed of the crank shaft. The stroke of this piston is so proportioned that the clearance between it and the main piston is very small at the end of the latter's exhaust stroke; and so that the two are the proper distance apart at the end of the compression stroke to give a clearance volume that insures the desired compression pressure. Figure I shows the arrangement of the piston and sleeves at the left, the auxiliary piston taking the place of the cylinder head. At the right, the distance between the two pistons is shown throughout the cycle of operation. It is seen that the auxiliary or head piston is in its outer position at the time of explosion and that it is in its inner dead center position when the main piston has returned on its exhaust stroke. A variable clearance throughout the cycle is the result. The head piston assists in clearing the cylinder of the burned gases during the exhaust stroke and assists in drawing in the new charge during the suction stroke of the main piston. Since the clearance volume is very small at the end of the exhaust stroke, a very small quanitty of hot exhaust gases will be left in the cylinder with which the new charge will be diluted. Therefore, the new charge will not have its temperature raised appreciably by mingling with these spent gases; the charge in the cylinder at the end of the suction stroke will be cooler so that a greater weight of mixture will be handled by the engine. Since the cylinder contents at the time of ignition will always be made up of a much greater proportion of new unburned gases than spent gases, ignition will always result even when the engine is operated with the throttle nearly closed.

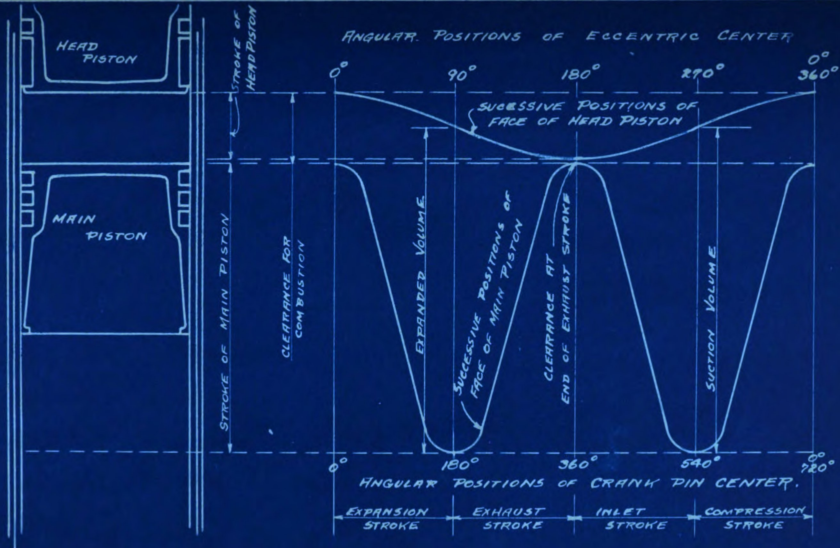


FIG. 1. DISPLACEMENT CURVES SHOWING RELATIVE POSITIONS OF THE FACES OF THE MAIN AND HEAD PISTONS IN JACKLIN ENGINE.

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17. It should be clear from Figure I why the author uses the term "suction volume" in his definition of volumetric efficiency. The suction volume in this case is somewhat greater than the piston displacement of an ordinary engine of the same bore and stroke because of the action of the head piston.

18. In order to compare this new design with existing engines of about the same size, data has been selected from descriptions of three of the best German designs used in their airplanes between 1914 and 1918. From this data, a card is constructed and constants derived which may be used in constructing a card for the new type. The analysis of the three types is presented first.

## ANALYSIS OF THREE GERMAN AVIATION ENGINES.

19. The three engines to be examined are the Benz six cylinder 160 H.P., and the Benz six cylinder 230 H.P., and the Mercedes six cylinder 260 H.P. From the general data and test results of these engines as published in "Automotive Industries" for January 24, 1918; February 14 and 28 and March 7, 1918; and September 6 and 13 and October 4, 1917 respectively, the following items are selected.

| | Benz | Benz | Mercedes | "Av.Engine" |
|---|---|---|---|---|
| 1. Rating-H.P. | 160 | 230 | 260 | 216.7 |
| 2. Bore-inches | 5.12 | 5.71 | 6.3 | |
| 3. Stroke-inches | 7.09 | 7.48 | 7.09 | |
| 4. Brake mean effective pressure | 103.4 | 113 | 107.5 | 107.96 |
| 5. Fuel consumption lbs. per H.P.-HR | 92.5 | 135 | 140.63 | 122.71 |
| 6. Speed-R.P.M. | 1400 | 1400 | 1400 | 1400 |

| | | | | |
|---|---|---|---|---|
| 7. Compression ratio | 4.5 | 5.00 | 4.94 | 4.813 |
| 8. Piston displacement for one cylinder- cu. inches | 146.05 | 191.4 | 220.82 | 186.1 |
| 9. Mechanical efficiency | .875 | .85 | .85 | .858 |

20. Since these engines operate with a free exhaust and are of a rather refined design, the following assumptions are believed reasonable.

1. Pressure within cylinder at the end of the exhaust stroke - 15.5 lbs. per square inch absolute.

2. Temperature of the gases remaining in the cylinder at the end of the exhaust stroke - 740° F. absolute.

3. Temperature of the air entering the carburetor - 517° F. absolute.

4. Ratio by weight of air to gas 14 to 1.

5. Atmospheric pressure - 14.7 lbs. per square inch absolute.

21. With an air-gas ratio of 14 to 1, the average engine as shown in paragraph 19, would require $\frac{122.71}{60} \times 14 =$ 28.62 lbs of air per minute.

22. This air would occupy $\frac{28.62}{0.076} = 376.7$ cu. ft. at 14.7 lbs per square inch and 62° F., the factor (0.076) being the weight of one cu. ft. of air under these conditions.

23. The volume of the fuel vapor used in one minute under these conditions is $\frac{122.71}{60} \times 4.2 = 8.58$ cu. ft., the factor (4.2) being the value given by A. M. Levin on page 125 of his book on the Modern Gas Engine (Wiley and Son, 1912) for the volume of one pound of gasoline vapor.

24. The total volume of air-gas mixture drawn into the cylinder of the "average engine" per minute is

$$376.7 + 8.58 = 385.28 \text{ cu. ft.}$$

25. The piston displacement of the "average engine" per minute at 1400 R.P.M. is $\frac{3 \times 1400 \times 186.1}{1728} = 452$ cu. ft.

26. The volumetric efficiency of this engine is then $E_v = \frac{385.28}{452} = .853$

27. In order to construct a complete indicator card from which constants may be derived for this "average engine", it is necessary to find the pressure and temperature of the cylinder contents at the beginning of the compression stroke.

The method used by J. R. Du Priest on pages 385 to 387 inclusive of the Trans. A.S.M.E., Vol. 39, to find the pressure works out as follows for this case.

28. The clearance volume of the "average engine" is $\frac{186.1}{4.813 - 1} = 48.85$ cu. in. This is shown on Fig. 2. This volume is filled with exhaust gases at a pressure of 15.5 lbs. per square inch and a temperature of 740° F. absolute.

29. These gases, when expanded isothermically, occupy $\frac{15.5}{14.7} \times 48.85 = 51.5$ cu. in. at 14.7 lbs. per square inch.

30. The volume of air-gas mixture drawn in is (from paragraphs 19 and 26) $.853 \times 186.1 = 158.8$ cu. in.

31. The total volume of exhaust gases and air-gas mixture at 14.7 lbs. per square inch to be expanded to occupy the whole cylinder volume is $51.5 + 158.8 = 210.3$ cu. in.

32. The volume which these gases occupy at some lower pressure is the sum of the piston displacement and the clearance volume, and is $186.1 + 48.85 = 234.95$ cu. in.

33. Then $P_a$, the pressure at the beginning of the compression stroke is $P_a = 14.7 \left(\frac{210.3}{234.95}\right)^{0.85} = 13.48$ lbs. per square inch absolute. The exponent (0.85) was arrived at as a result of experiment by Mr. Du Priest as shown on pages 389-390

Trans. A.S.M.E., Vol. 39.

34. In finding the temperature of the cylinder contents before compression starts, it is assumed that the specific heat of the exhaust gases is the same as that of the new aid-gas mixture in order to simplify the computations and because the error introduced is negligable in the final results. The temperature of the air as it reaches the carburetor is 57° F. (517° F. absolute). There is a drop of approximately 40° when this air-gas mixture is expanded to the pressure within the cylinder and the fuel has been vaporized. Upon entering the cylinder, the air-gas mixture encounters the hot cylinder walls with a resulting rise in temperature of 55°. Thus the temperature of the air-gas mixture before it has mixed with the hot exhaust gases is 517 - 40 + 55 = 532° F. abs.

35. The weight of the air-gas mixture is

$$M_n = \frac{158.8}{1728} \times 0.076 = .00698 \text{ lbs.}$$

36. The weight of the exhaust gases is

$$M_r = \frac{48.85}{1728} \times \frac{522}{740} \times \frac{15.5}{14.7} \times 0.076 = .0016 \text{ lbs.}$$

37. Then 0.00698 lbs. of air-gas mixture at 532° F. are raised to a temperature, $T_a$, by mixing with 0.0016 lbs. of exhaust gases at 740° F. abs. With equal specific heats, the equation becomes $(T_a - 532)\ 0.00698 = (740 - T_a)\ 0.0016$ from which $T_a = 572°$ F. abs.

38. The above results for $P_a$ and $T_a$ are checked as follows. The air-gas mixture occupies 158.8 cu. in. at 14.7 lbs. and 522° F. At 13.48 lbs. and 572° F, it will occupy $\frac{572}{522} \times \frac{14.7}{13.48} \times 158.8 = 189.9$ cu. in. The exhaust gases remaining in the cylinder occupying 48.85 cu. in. at 15.5 lbs. and 740° F.

At 13.48 lbs. and 572° F. they occupy

$\frac{572}{740} \times \frac{15.5}{13.48} \times 48.85 = 44.5$ cu. in. The total volume occupied by the air-gas and exhaust gas mixture is 189.9 + 44.5 = 234.4 cu. in. which corresponds very closely to the volume they are supposed to occupy - 234.95 cu. in.

39. Having obtained the values of $P_a$ and $T_a$, it is now possible to construct the compression line as shown in Fig. II, as well as to compute the temperatures as shown by the dashed line. The volumes at the various points are:

$V_a$ = 234.95 cu. in. $\qquad$ $V_e$ = 110.90

$V_b$ = 203.94 $\qquad$ $V_f$ = 79.88

$V_c$ = 172.92 $\qquad$ $V_g$ = 48.85

$V_d$ = 141.92

The corresponding pressures in lbs. per square inch absolute, using N = 1.33 in the equation $PV^n$ = constant, are

$P_a$ = 13.48 $\qquad$ $P_e$ = 36.8

$P_b$ = 16.20 $\qquad$ $P_f$ = 55.6

$P_c$ = 20.20 $\qquad$ $P_g$ = 108.2

$P_d$ = 26.50

The corresponding temperatures in degrees Fahrenheit absolute, using the equation $PV^{n-1}$ = constant, are

$T_a$ = 572 $\qquad$ $T_e$ = 742

$T_b$ = 607 $\qquad$ $T_f$ = 824

$T_c$ = 640 $\qquad$ $T_g$ = 968

$T_d$ = 675

40. The explosion pressure $P_x$ Fig. 2 is derived by means of P. M. Heldt's formula as stated in the Horseless Age, page 1 of the engineering section, December 15, 1916. It is as follows: $\text{M.E.P.} - \text{m.e.p.} = \frac{P_a}{0.3}(a-1)\left(\frac{1}{r-1}\right)\left[1 - \left(\frac{1}{r}\right)^{0.33}\right]$

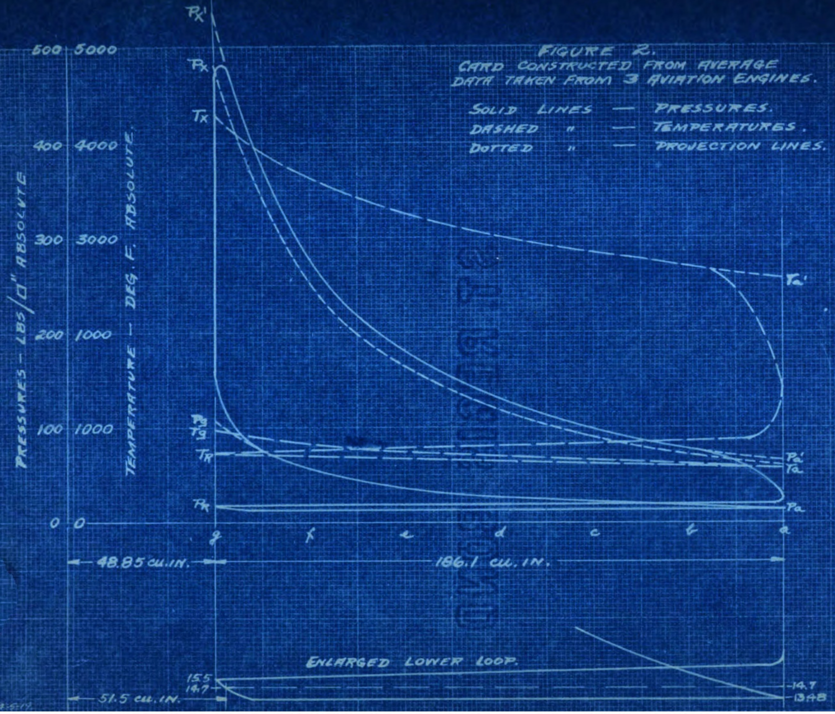
FIGURE 2.
CARD CONSTRUCTED FROM AVERAGE
DATA TAKEN FROM 3 AVIATION ENGINES.
SOLID LINES — PRESSURES.
DASHED " — TEMPERATURES.
DOTTED " — PROJECTION LINES.
PRESSURES - LBS/□" ABSOLUTE
TEMPERATURE - DEG. F. ABSOLUTE.
500 5000
400 4000
300 3000
200 1000
100 1000
0 0
Px'
Px
Tx
Pg
Tg
TR
PR
Ta'
Pa'
Ta
Pa
g f e d c b a
48.85 CU. IN.
186.1 CU. IN.
ENLARGED LOWER LOOP.
15.5
14.7
14.7
13.48
51.5 CU. IN.

in which

M.E.P. = mean pressure during expansion

m.e.p. = " " " compression

$P_g$ = compression pressure in lbs. per square inch = 108.2

a = ratio in which $P_g$ is multiplies on explosion

r = volumetric compression ratio = 4.813

It is apparent that $P_g$ (a - 1) is the rise in pressure due to the explosion. It remains to derive a value for the mean effective pressure (M.E.P. - m.e.p.) for this "average" engine before the equation can be solved for the value of $P_g$ (a - 1).

41. The brake mean effective pressure is 107.96 lbs. per square inch and the mechanical efficiency is 0.858. From these values M.E.P. - m.e.p. $= \frac{107.96}{.858} = 125.8$ lbs. per sq. in.

42. Then $P_g\ (a - 1) = \dfrac{125.8 \times 0.3}{\left(\frac{1}{4.813-1}\right)\left[1-\left(\frac{1}{4.813}\right)^{0.33}\right]}$

The denominator of the second number becomes $\left(\frac{1}{3.813}\right)\left(1 - \frac{1}{1.678}\right) = \frac{1}{9.42}$

Then $P_g\ (a - 1) = 125.8 \times 0.3 \times 9.42 = 374$ lbs. and the explosion pressure $P_x = 108.2 \times 374 = 482$ lbs. per square in. abs.

43. The temperature at explosion is $T_x = \frac{P_x}{P_g} \times T_g = \frac{482 \times 968}{108.2} = 4300^{\circ}$ F. abs.

44. The pressure during the expansion stroke, using n = 1.33 in equation $PV^n$ = constant, were then calculated and plotted as shown by the dotted line in Fig. 2. Upon checking the area, and the m.e.p., it was found necessary to plot this line from some higher point as $P_x^1$. In this case, it was found that $P_x^1$ should be 10 per cent greater than $P_x$. Allowance is made for the increase in pressure due to ignition occurring early, for a rounding of the curve at its highest point and for the drop

in pressure when the exhaust opens.

$P_x{}^1 = 1.10\ P_x = 530$ lbs. per square in. abs. The curve resulting is shown in the full line Fig. 2. The difference in area of the large and small loops is 7.565 sqare inches. The card is 6" long and the pressure scale is 1" = 100 lbs., so the m.e.p. figures out to 126.08 lbs. which is very close to the value desired. THE pressures at the various points are-

| | |
|---|---|
| $P_x{}^1 = 530$ | $P_c{}^1 = 99.0$ |
| $P_f{}^1 = 278$ | $P_b{}^1 = 79.6$ |
| $P_e{}^1 = 182$ | $P_a{}^1 = 66.0$ |
| $P_d{}^1 = 130$ | |

and the temperatures are

| | |
|---|---|
| $T_x = 4300$ | $T_c{}^1 = 2838$ |
| $T_f{}^1 = 3630$ | $T_b{}^1 = 2700$ |
| $T_e{}^1 = 3242$ | $T_a{}^1 = 2578$ |
| $T_d{}^1 = 3030$ | |

44. From paragraphs 35 and 36, it is apparent that the total weight of the cylinder contents at the time of explosion is $M = .00698 + .0016 = .00858$ lbs. From paragraphs 39 and 43, it is seen that the temperature rise is $T_x - T_g = 4300 - 968 = 3332$. The heating value of the fuel in each charge is $Q = \frac{122.71}{60} \times \frac{19000}{4200} = 9.2$ B.T.U.

122.71 = weight of gasoline consumed per hour

19000 = B.T.U. per lb. of fuel

4200 = number of inlet strokes per minute.

The above values may be substituted in the equation $Q = Mc_v\ (T_x - T_g)$, and a value deduced for $c_v$. Then

$$c_v = \frac{9.2}{.00858 \times 3332} = .322$$

46. The writer is well aware that a card of this type is not usually obtained from an engine operating at 1400 R.P.M. It is probable that the explosion line would be inclined toward the right instead of being vertical, and that the pressure at the time the exhaust valve opens would be higher than that shown. However, this more or less ideal card has been made to fit test results by using a correction factor, so that a value for the specific heat may be derived. It should be noted that the figure for the specific heat is approximately double that accepted for air. This agrees with observed results, where about one-half the expected rise in pressure has been obtained if the specific heat for air has been used in computing the expected pressure. By applying the factors as derived for this average engine to another engine operating under similar conditions, the performance of the latter may be quite accurately prophesied. In working out the data for this engine the following data will be used.

1. Pressure within cylinder at the end of the exhaust stroke = 15.5 lbs. per square inch absolute

2. Temperature of above gases = 740° F. abs.

3. Temperature of air entering carburetor = 517° F. abs.

4. Atmospheric pressure 14.7 lbs.

5. Ratio of air to gas by weight = 14 to 1.

6. J. R. Du Priest's exponent = 0.85

7. Weight of air at 62° and 14.7 lbs. per square inch = 0.076 lbs. per cu. ft.

8. Factor with which to multiply the explosion pressure to find point from which to plot the expansion line = $\frac{Px^1}{Px}$ = 1.10

9. Specific heat of cylinder contents at time of explosion = $c_v$ = .322

10. Pressure in cylinder at beginning of compression stroke = 13.5 lbs. per square inch absolute

11. Drop in temperature of air-gas mixture due to drop in pressure and vaporizing of fuel = $40^{\circ}$ F.

12. Increase in temperature of air-gas mixture due to coming in contact with cylinder walls = $50^{\circ}$ F. This is slightly less than that used in the analysis of the "average engine." It is deemed proper because the surface with which the charge comes in contact is actually less per unit of weight in this new engine than in the "average engine" and because there is more air-gas mixture entering the cylinder to be heated by such contact. Using as accurate comparative figures as possible, it was found that this rise in temperature would be $46^{\circ}$ F. The writer has compromised on $50^{\circ}$ F. as an acceptable figure.

## The Jacklin Engine

47. The arrangement of the two pistons and sleeves in the cylinder of this engine is shown in Fig. 4 (large print in pocket on back cover), while the displacement curves through $720^{\circ}$ of crank shaft rotation for the various parts are shown in Fig. 3. The operation of the sleeves and auxiliary or head piston in timing the valves and in varying the clearance volume is clearly shown. By observing the intersections of the curved lines with the vertical lines representing the degrees of rotation of the crank, the position of the parts at any instant may be accurately found. All necessary design data as to centers, etc., is noted on this drawing so that no trouble should

be experienced in constructing a lay-out.

48. It is seen that the clearance volume is very small at the end of the exhaust stroke, as represented by $V_r$. The distance between the faces of the two pistons at this time is 1/16", which is ample to care for all differences of expansion of the various parts due to heating. The volume between the pistons at this time is $V_r = \frac{1"}{16}$ x area of 5" circle + the volume X which is a shallow ring $= \frac{1}{16} \times 19.655 + .43 = 1.86$ cu. in.

49. The volume of these gases expanded isothermically to 14.7 lbs. per square inch is $\frac{15.5}{14.7} \times 1.86 = 1.96$ cu. in.

50. Since such a small quantity of hot gases would have a negligable effect in heating the new charge, these gases should also be considered as cooled to the same temperature as the cylinder contents at the end of the suction stroke, which is $T_7 = 517 - 40 + 50 = 527^{\circ}$ F. abs. so that the exhaust gases occupy $\frac{527}{740} \times 1.96 = 1.396$ cu. in. at 14.7 lbs. per square inch and $527^{\circ}$ F. abs.

51. The volume between the pistons at the end of the suction stroke is $V_7 = 7\ 7/8$ x area of 5" circle - .43 cu. in. $= 7.875 \times 19.635 + .43 = 155.26$ cu. in.

52. With (Y) the volume of new air-gas mixture drawn into the cylinder as outlined in paragraphs 27 to 33 inclusive, the following equation is true $13.5 = 14.7 \left(\frac{Y + 1.396}{155.26}\right)^{0.85}$ from which Y = 139.6 cu. in. air-gas mixture at atmospheric pressure and temperature.

53. On the basis of "suction displacement" the volumetric efficiency of the Jacklin engine is $E_V = \frac{139.6}{155.26 - 1.86} = .91$

54. A comparison of the volumetric efficiency

of the Jacklin engine and a conventional engine of the same bore and stroke (in the case of the Jacklin, the bore and stroke of the main piston) is interesting at this point. With a 5 x 7 cylinder, the piston displacement is 137.45 cu. in. Then, the volumetric efficiency of the Jacklin engine on this basis is $E_v^1 = \frac{139.6}{137.45} = 1.016$. Therefore, this engine will handle $100(\frac{1.016 - 853}{.853}) = 19.1$ per cent more aig-gas mixture than a conventional engine or, in this case, the "average engine", since .853 is the volumetric efficiency of the "average engine." Since, in general, the power developed depends on the amount of fuel consumed in the engine, it seems to be reasonable to expect a large increase in power when this design is used.

55. It remains to figure out an indicator card to complete the analysis and comparison. The volume in cubic inches between the pistons at the various points in the stroke of the main piston are (referring to Fig. 3).

$$V_7 = 155.26$$

$$V_6 = 7.1875 \times 19.635 + .43 = 141.43$$

$$V_5 = 6.3125 \times 19.635 + .43 = 124.33$$

$$V_4 = 5.3750 \times 19.635 + .43 = 105.93$$

$$V_3 = 4.4380 \times 19.635 + .43 = 87.53$$

$$V_2 = 3.5000 \times 19.635 + .43 = 69.13$$

$$V_1 = 2.5313 \times 19.635 + .43 = 50.13$$

$$V_0 = 1.5625 \times 19.635 + .43 = 30.77$$

The corresponding pressures in lbs. per square inch. abs. at these points (using $n = 1.33$ in the equation $PV^n = \text{constant}$) will be

| | | |
|---|---|---|
| $P_7 = 13.5$ | $P_4 = 22.8$ | $P_1 = 61.1$ |
| $P_6 = 15.5$ | $P_3 = 29.25$ | $P_0 = 117.8$ |
| $P_5 = 18.8$ | $P_2 = 39.2$ | |

The temperature at the end of the compression stroke will be

$T_0 = T_7 \times (\frac{V_7}{V_0})^{0.33} = 527 \times (\frac{155.26}{30.77})^{0.33} = 901^{\circ}$ F. abs.

This temperature is lower than the temperature $T_g$, paragraph 39, so that there is no likelihood of pre-ignition occurring in spite of the fact that the compression pressure has been increased about 10 lbs. per square inch.

56. At 1525 R.P.M. this engine would use $\frac{4575 \times 139.6}{1728} = 370$ cu. ft. of air-gas mixture at $62^{\circ}$ F. and 14.7 lbs. per square in. absolute. The factor 4575 is the number of suction strokes in a six cylinder four-stroke engine at 1525 R.P.M. Then the weight of the air-gas mixture (assuming the fuel vapor weighs the same as the air which is reasonably accurate as the vapor occupies approximately but 2 per cent of the volume) is $370 \times 0.076 = 28.15$ lbs.

57. When a 14 to 1 ratio of air to gas is used, this mixture will contain $\frac{28.15}{14 - 1} = 1.875$ lbs. of fuel, the heating value of which is $1.875 \times 19000 = 35{,}610$ B.T.U.

58. The total weight of the cylinder contents is 28.15 lbs. plus the weight of the spent gases left in the cylinder at the end of the exhaust stroke. These weigh $\frac{4575 \times 1.396}{1728} \times 0.076 = .281$ lbs. so the total weight of the cylinder contents becomes $M = 28.15 + .281 = 28.43$ lbs.

59. All values except the explosion temperature $(T_0^1)$ in the equation $Q = Mc_v\ (T_0^1 - T_0)$ are now available, so that $35{,}610 = 28.43 \times .322\ (T_0^1 - 901)$ from which

$T_0^1 = 4916^{\circ}$ F. abs.

60. Then the explosion pressure $P_0^1 = \frac{T_0^1}{T_0} \times P_0 = \frac{4916}{901} \times 117.8 = 641$ lbs. per square inch absolute.

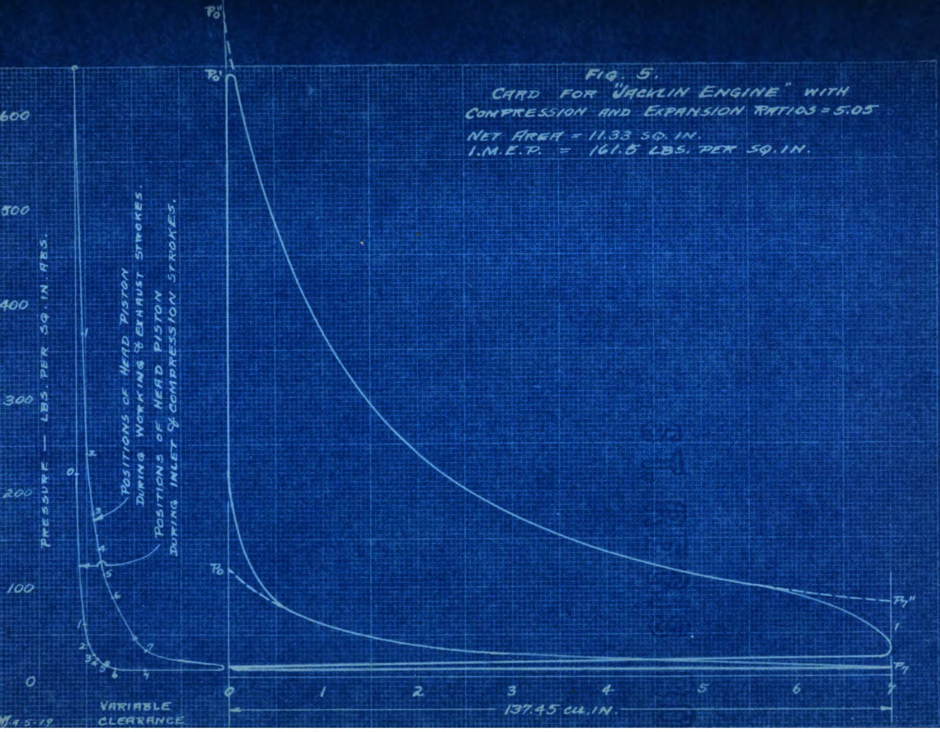


FIG. 5.
CARD FOR "JACKLIN ENGINE" WITH COMPRESSION AND EXPANSION RATIOS = 5.05
NET AREA = 11.33 SQ. IN.
I.M.E.P. = 161.5 LBS. PER SQ. IN.

61. $P_o''$, the point from which the expansion line should be drawn is $P_o'' = 1.10 \times P_1' = 1.10 \times 641 = 704$ lbs.

62. The pressures in lbs. per square inch abs. during the expansion stroke at the various points will be

| | |
|---|---|
| $P_o'' = 704$ | $P_4'' = 136.2$ |
| $P_1'' = 367$ | $P_5'' = 112.7$ |
| $P_2'' = 239.3$ | $P_6'' = 93.1$ |
| $P_3'' = 175.3$ | $P_7'' = 88.8$ |

63. Fig. 5 shows the compression and expansion curves as plotted from paragraphs 55 and 62, with allowances made for the rise in pressure at the time ignition occurs and for the drop in pressure after the exhaust valve opens. The difference in area of the upper and lower loops is 11.33 square inches. The mean effective pressure is then 161.5 lbs. per square inch.

64. Since the head piston moves outward and inward $\frac{1}{2}''$ during the compression and expansion strokes, respectively, the effective compression and expansion stroke is $7 - \frac{1}{2} = 6\frac{1}{2}$ inches.

65. Assuming the mechanical efficiency of this engine at 0.80 instead of .853, which is considered a liberal allowance for the increased power necessary to operate the valves, the brake mean effective pressure becomes B.M.E.P. = .80 x 161.5 = 129.2 lbs. per square inch.

66. The brake horsepower to be expected from this engine is then $B.H.P. = \frac{129.2 \times 6.5 \times 19.635 \times 1525 \times 3}{12 \times 33000} = 190.7.$

67. Using the B.M.E.P. of the "average engine," the brake horsepower of a conventional six-cylinder 5 x 7 engine at

1525 R.P.M. is $B.H.P. = \frac{107.96 \times 7 \times 19.635 \times 1525 \times 3}{12 \times 33000} = 171$

68. The increase in horsepower by the use of the Jacklin design is 190.7 - 171 = 19.7 H.P. The per cent increase is $100\left(\frac{19.7}{190.7}\right) = 10.32\%$.

69. A final comparison is not possible unless a complete design is worked out so that the weight may be estimated. Figs. 6, 7 and 8 show three views of a 5 x 7, six-cylinder engine according to this design. In the following paragraphs, the various features of the design are described, followed by a table showing the specifications of this engine arranged parallel to those of the three engines referred to in paragraph 19.

70. The piston is an aluminum casting of a very heavy section as is apparent in Figs. 4, 6 and 7. The center of the head is relatively thin and becomes thicker toward the edge, from which point the metal is tapered until at the lower edge of the skirt it is very thin. This type was decided upon because of the advantages it presents. The heat from the piston head is conducted down along the skirt and is communicated to the surrounding sleeve. Also no ribs are necessary and a cleaner casting should result. The rings are placed a considerable distance from the head in order that they may not expand into the parts in the inner sleeve. They are of the three piece type. The bearing for the piston pin is the aluminum casting itself. This is a good bearing material, and in addition the weight of bushings is saved. The head piston is also an aluminum casting. The piston pin bearings are exactly the same as those for the main piston. Since there is a relatively long connecting rod (over 7 inches), while the stroke is only $1\frac{1}{2}$ inches, it is

permissible to make this piston very short as has been done. The wide junk ring is pinned so that its joint comes between the two ports in the inner sleeve. The triple ring just above the junk ring should prevent all possibilities of leak at that point. As is apparent in Figs. 6 and 7, the head piston is water cooled, it being cast with a double head. This is deemed necessary because the spark plugs are carried in the head piston. The water connections and spark plug connections are very simple and should not give any difficulty because of the shortness of the stroke. The weight of the main piston with rings and pin is 4.5 lbs; while that of the head piston is 4.5 lbs.

71. The piston pins are 1 3/8 inches in diameter and are held in the upper end of the connecting rod. They are taper reamed so that they should be very stiff and amply strong. The connecting rods for the main engine piston and the head piston are exactly alike except as to length. That for the engine piston is 12 inches between centers, while the one for the head piston is 7 9/32 inches. Both are machined all over from the rough. They are 1 3/8 inches outside diameter and 1 1/8 inches inside diameter. This gives plenty of strength to resist the estimated high explosion pressure of 640 lbs. per square inch, and in addition should not give out due to whipping because of the large diameter. The bearings in the piston are 1 3/8 inches diameter by a total of 3 inches length. The bearing at the main ends is 2 1/2 inches diameter by 2 3/4 inches length. The weight of the piston pin is 1.2 lbs. while that of the long connecting rod is 5.3 lbs., and the short rod weighs 4.6 lbs.

72. The crank shaft is of the six throw type with

seven main journals; the front 3 inches long, all intermediate bearings 1 3/4 inches long and the rear 1 3/4 inches. The crank pins are 2 3/4 inches long as already explained. Both main journals and crank pins are 2 1/2 inches in diameter. The first two crank cheeks next to the propeller are one onch think. All others are 15/16. The width of the cheek in all cases is 3 inches. The main journals and crank pins are drilled out 1 1/2 inches. The weight of the crank shaft complete is 87 pounds. Owing to the small throw of the eccentric shaft it is impossible to drill out either the main journal or the crank pin as is done with the crank shaft. However a 1/2 inch oil hole is drilled from end to end. Cheeks of the eccentric shaft are made use of as eccentrics for operating the sleeve valves. The cheeks are lightened by drilling as shown in Fig. 5. In addition to these holes two 5/8 inch holes are drilled into the crank pins from an angle as shown in Fig. 7. Altogether this lightening is considerable. The main journals and crank pin journals of this shaft are exactly the same size as those for the crank shaft. The weight of the eccentric shaft is 61.3 journals.

73. In order to make the sleeves as light as possible it was decided to use steel tubing. The inner sleeve of course must be the heavier and is made 3/32 inch thick, while the outer sleeve is only 1/16 inch thick. The upper end to which the pin is attached is only a segment of a circle and is made 1/4 inch thick. The pin is to be welded to the sleeve. A slot into which the guide pins fit serve to prevent the sleeves from rotating, which tendency they are liable to have owing to the shortness of the rods--only slightly over 4 inches for a stroke of 3 inches. It is intended to finish the pin by drilling and

hollow milling. This construction is relatively light and should be as cheap as any other which might be adopted. The inner sleeve weighs 6.7 lbs. while the outer one weighs only 4.3 lbs. As already stated, the rods for operating these sleeves are very short. They are bronze or brass castings and are not fitted with other bearing material or with bushings. That for the outer sleeve weighs 1.94 lbs., while that for the inner sleeve which is slightly longer weighs 2.03 lbs.

74. The cylinders and upper half of the crank case are cast en-bloc of aluminum with large openings on both sides of the water space and also on both sides of the upper case. Since the cores can be located very accurately a thickness of 5/32 inch for the cylinder barrels as finished is deemed sufficient. The crank case casting is well ribbed and should be very rigid when assembled. The explosion pressure is communicated through the two pistons to the two connecting rods, and from there to the two shafts. Long through bolts are provided between the bearing caps for the main journals of the crank shaft and eccentric shaft, so that none of the direct stress from this source is communicated to the cylinder. These through bolts pass through the water jacket. Water leaks are provided against by the use of gaskets and half nuts at the upper and lower ends of the jackets. The weight of cylinders and upper crank case complete with all bolts, covers, etc., is 129.5 lbs.

75. After comparative layouts, it was decided to use the straight spur gear drive for the eccentric shaft as about 15 H.P. must be transmitted to operate the head pistons. Then, too, the two magnetos and the water pump can be mounted

on the gear housing with a minimum required weight for making connections.

76. All pumps are driven by an eccentric at the rear end of the crank shaft. Two oil cylinders are provided and one air cylinder. Oil is drawn from the bottom of the crank case and pumped to the tank by one pump. The other pump serves to take this cool oil and pump it under pressure to the main journals whence it is distributed to the crank pins, the cylinders being lubricated by the oil thrown from the rotating parts. Excess oil goes through the vertical pipe in the gear housing to above the top gear where a pressure regulator and pulsation damper are provided. The overflow is directed onto the gear train, from whence it flows to the lower crank case and is returned to the tank. The air pump supplies air to maintain pressure on the gasoline tank which may be mounted anywhere in the fusilage.

# SPECIFICATIONS

| | Jacklin 190 H.P. | Benz 160 H.P. | Benz 230 H.P. | Mercedes 260 H.P. |
|---|---|---|---|---|
| Cycle of Operation | Modified 4-stroke | 4-stroke | ditto | ditto |
| No. of Cylinders | 6 | 6 | 6 | 6 |
| Bore | 5 | 5.12 | 5.71 | 6.3 |
| Stroke | 7 | 7.09 | 7.48 | 7.09 |
| Stroke-Bore-Ratio | 1.400 | 1.385 | 1.31-1 | 1.125-1 |
| Area of Piston sq. in. | 19.635 | 20.60 | 25.59 | 31.164 |
| Total Piston area sq. in. | 117.81 | 123.60 | 153.57 | 186.984 |
| Piston displacement of one cylinder | 137.45 | 146.05 | 191.386 | 220.82 |
| Total Piston displ. | 824.67 | 876.30 | 1148.32 | 1324.92 |
| Vol of clearance space | 30.77 | 41.65 | 48.66 | 56.12 |
| Comp_ression ratio Total vol to cl.vol | 5.05-1 | 4.50-1 | 5-1 | 494-1 |
| Normal B.H.P. | 190 | 160 | 230 | 252 |
| " speed R.P.M. | 1525 | 1400 | 1400 | 1400 |
| Piston speed ft./min. | 1780 | 1655 | 1744 | 1655 |
| Brake mean effective pressure | 129.2 (est) | 103.4 (est) | 113 | 107.5 |
| Mech. efficiency | 80% | 87.50% | 85% | 85% |
| Indicated mean pressure | 161.5 | 117.7 | 133 | 126.5 |
| Cu. in. of Piston displ. per B.H.P. | 4.34 | 5.48 | 4.99 | 5.25 |
| Sq. in. of Piston area per B.H.P. | .620 | .773 | .667 | .74 |
| H.P./cu.ft. of Piston displ. | 374 | 315.5 | 346.3 | 329.14 |

| | | | | |
|---|---|---|---|---|
| H.P./sq.ft. of Piston area | 279.5 | 186.4 | 215.9 | 194.6 |
| Direction of rotation of crank & propellor | Clock | anti-clock | anti-clock | anti-clock |
| No. of Carburetors | 2 | 2 | 2 | 1 |
| Type | Zenith | Benz 2 jet | Benz 2 jet | Mercedes 2 jet |
| Mixture control | auto | auto | auto | auto |
| Ignition | | | | |
| No. of units | 2 | 2 | 2 | 2 |
| Mag. or Batt. | mag. | mag. | mag. | mag. |
| Spark plugs per cyl. | 2 | 2 | 2 | 2 |
| " " position | head center | sides | sides | sides |
| Cylinder numbering from propeller | 1-2-3 4-5-6 | 1-2-3 4-5-6 | 1-2-3 4-5-6 | 1-2-3 4-5-6 |
| Firing order from propeller | 1-5-3 6-2-4 | 1-5-3 6-2-4 | 1-5-3 6-2-4 | 1-5-3 6-2-4 |
| Timing | $30^{o}$ | $32^{o}$ early | $30^{o}$ early | $31^{o}$ early |
| Mag. speed | 1 1/2 eng. | | 2/3 eng. | |
| Type of Cylinder | I-head | | | Individual |
| Material | al | C.1 | C.1 | Steel |
| How made | cast en-bloc | cast & welded | cast & welded | welded |
| Type of valves | Slide | poppett | poppett | poppett |
| Location | opposite sides | head | head | head |
| Valve Timing | | | | |
| Inlet opens | $0^{o}$ | $0^{o}$ | $10^{o}$ | $1^{o}$ |
| Inlet closes | $45^{o}$ | $60^{o}$ | $55^{o}$ | $49.3^{o}$ |

| | | | | |
|---|---|---|---|---|
| Exhaust opens | $57^\circ$ | $60^\circ$ | $60^\circ$ | $50.6^\circ$ |
| Exhaust closes | $7^\circ$ | $18^\circ$ | $20^\circ$ | $17.6^\circ$ |

<u>Valve Detail</u>

Inlet Valves

| | | | | |
|---|---|---|---|---|
| Port dia | | 2.420 | 2.98 | 2.193 |
| Lift | | 1433 | .465 | .402 |
| Width of opening | 5.75 | | | |
| Max. height | .875 | | | |
| Mean height of opening | .465 | | | |
| Mean area | 2.68 | | | |
| Max. area | 5.04 | 3.29 | 5.96 | 5.44 |

Exhaust Valves

| | | | | |
|---|---|---|---|---|
| Port dia | | 2.420 | 2.06 | 2.193 |
| Lift | | .433 | .443 | .402 |
| Width of opening | 5.75 | | | |
| Max. height " | .6875 | | | |
| Mean height " | .406 | | | |
| Mean area | 2.34 | | | |
| Max. area | 3.96 | 3.29 | 5.68 | 5.4 |

Gas Velocity - ft./sec.

| | | | | |
|---|---|---|---|---|
| Inlet valve | 142 | 172.8 | 124 | 158 |
| Exhaust valve | 150 | 172.8 | 130 | 145 |

Piston Pin Bearings

| | | | | |
|---|---|---|---|---|
| Dia | 1.375 | 1.18 | 1.52 | 1.667 |
| Length | 3 | 2.91 | 3.57 | 3.78 |
| Location | piston | in rod | in rod | in rod |

| | | | | |
|---|---|---|---|---|
| Material | Bosses Al. | Phosphor Bronz | Phosphor Bronz | Phosphor Bronz |
| Fastening | in rod | in rod | in rod | in rod |
| Area in sq.in. | 4.125 | 3.435 | 5.38 | 6.3 |
| Ratio of piston area to projected area | 4.76 to 1 | 6 to 1 | 4.75 | 5.04 |

Piston Type

| | | | | |
|---|---|---|---|---|
| Material | Al | C.1 | C.1 | C.1 |
| Total | | 5.00 | 7.62 | 11.00 |
| Distance from center of pin to top of piston | 2.500" | | | |
| Length over all | 5.00" | | | |

| | | | | |
|---|---|---|---|---|
| Piston Rings - no | 2 | 3 | 3 | 4 |
| Material | C.1 | C.1 | C.1 | C.1 |
| Type | 3 piece | plain | plain | plain |
| Width | 5-16" | .3175" | .3175" | .199" |

Crank Shaft

| | | | | |
|---|---|---|---|---|
| Type | hollow | hollow | | |
| No. of bearings | 7 | 7 | | |
| Dia of main bearings | 2.500 | 2.165 | 2.46 | 2.519 |
| Dia of crank pin | 2.500 | 2.165 | 2.38 | 2.519 |
| Length of front | 3.000 | 2.756 | 3.14 | 4.094 |
| " of intermediate | 1.750 | 2.2047 | 2.142 | 2.519 |
| Length of rear | 1.750 | 1.968 | 2.18 | 2.519 |
| No. of bolts ea. big. | | | | |
| Front | 4 | 4 | | |
| Intermediate | 2 | 2 | | |

| | | | | |
|---|---|---|---|---|
| Dia hole in main | 1.500 | 1.1024 | 1.072 | 1.39 |
| Dia hole in crank pins | 1.500 | 1.1024 | 1.072 | 1.39 |
| Cheek thickness next to propeller | 1" | | | $1\frac{1}{4}$ |
| Cheek thickness of balance of crank | 15/16" | 0.945 | | 1 1/18 |
| Width of crank webs | 3" | 2.598 | | |
| Weight of crank | 87 | 72.70 | 109.25 | 139.5 |
| Cylinder centers | 6.500 | 6.69 | | |
| Dia of drilled oil holes in shaft | (4)-$\frac{1}{4}$" | .1575 | | |
| **Connecting Rod - No** | 6 | 6 | 6 | 6 |
| Type | Tubular | Tubular | Tubular | H section |
| Length c to c | 12" & 7 9/32 | 12.37 | | 12.95 |
| No. of bolts- lower end | 4 | 4 | | |
| Dia | .375 | .3937 | | |
| Weight of upper end | 1.37 | 1.8 | 2# 5oz. | 2# 2oz. |
| Weight of lower end | 3.93 | 3.2 | 4# 12oz. | 4# 14oz. |
| Total | 5.3 & 4.6 | 5.00 | 7# 1oz. | 7# |
| Dia | 1.375 | | 1.429 | |
| Bored out | 1.125 | | 1.190 | |
| **Connecting Rod Bearing** | | | | |
| Dia | 2.500 | 2.165 | 2.38 | 2.519 |
| Length (actual) | 2.750 | | | 3.175 |
| Effective length | 2.500 | 2.244 | 2.8 | 2.925 |
| Area | 6.25 | 4.86 | 6.65 | 7.36 |
| Ratio piston area to projected area | 3.14 | 4.18 | 3.82 | 4.3 |

General Analysis of Weights and Measurements.

| | Jacklin 190 H.P. | Benz 160 H.P. | Mercedes 160 H.P. | Benz 230 H.P. | Mercedes 260 H.P. |
|---|---|---|---|---|---|
| Cylinders<br>Base chamber (top) | 129.50 | 168.00<br>100.00 | 115.5<br>118.59 | 265.50<br>103.25 | 205.70 |
| " " (bottom) | 19.75 | 65.75 | 73.00 | 110.25 | |
| Crank shaft complete | 87.00 | 72.70 | 70.00 | 109.25 | 139.50 |
| Propeller hub " | 12.00 | 12.00 | 13.00 | 19.32 | |
| " bearing | 3.75 | 3.81 | 2.56 | | |
| Camshaft complete | 61.30 | 10.50 | 10.50 | 16.50 | |
| Pistons complete | 54.60 | 30.00 | 41.10 | 45.72 | 64.50 |
| Connecting rods | 59.40 | 30.00 | 30.00 | 42.78 | 42.00 |
| Valves complete | 66.00 | 11.91 | 13.68 | 27.36 | 19.00 |
| " operating gear | 23.82 | 14.20 | 5.94 | 38.4 | 15.00 |
| Inlet manifolds | 1.50 | 8.125 | 11.75 | 5.25 | |
| Carburetors complete | 8.00 | 7.25 | 12.73 | 13.08 | |
| Magnetos complete | 27.00 | 29.00 | 30.00 | 21.50 | |
| Ignition wiring & tubes | 4.75 | 4.35 | 4.00 | 4.24 | |
| Water pumps complete | 5.83 | 4.81 | 6.87 | 8.75 | |
| " pipes | 17.00 | 3.00 | 19.00 | | |
| Oil pumps complete | 5.00 | 4.55 | 12.00 | 2.89 | |
| Air " " | 1.50 | 1.125 | 1.50 | | |
| Fuel pumps | | | | 6.75 | |
| Exhaust Manifold | 10.00 | | | 15.00 | 26.00 |
| Miscellaneous | 15.00 | | | 7.53 | |
| Total | 612.7 | 591.5 | 618. | 848.32 | 936.00 |

| | Jacklin 190 H.P. | Benz 160 H.P. | Mercedes 160 H.P. | Benz 230 H.P. | Mercedes 260 H.P. |
|---|---|---|---|---|---|
| Weight per B.H.P. | 3.23 | 3.70 | 3.87 | 3.68 | 3.71 |
| Weight of fuel per hour | 112.5 | 92.5 | 92.5 | 135 | 140.63 |
| Weight of oil per hour | 6.00 | 4.5 | 4.5 | 5.06 | 5.31 |
| Total weight of fuel and oil per hour | 118.5 | 97 | 97 | 140.06 | 145.94 |
| Gross weight of engine in running order less fuel and oil. Cooling system at .65 lbs. per. B.H.P. | 726.5 | 695.5 | 722 | 996 | 1098 |
| Weight per B.H.P. ditto | 3.822 | 4.35 | 4.51 | 4.43 | 4.36 |
| Gross weight of engine in running order with fuel and oil for six hours. Tankage reckoned at 10% of weight of fuel and oil | 1508.6 | 1277.5 | 1304 | 1920.89 | 2061 |
| Weight per B.H.P with fuel and oil for six hours. | 7.93 | 7.98 | 8.15 | 8.35 | 8.18 |
| Overall length | 64 | 61.7 | | 76.8 | 77.5" |
| " width | 18 | 25.0 | | 23.8 | 26.6 |
| " height | 46 | 41.2 | | 53.5 | 46.0" |
| Total cubic space cu. in. | 53,000 | 63,510 | | 97,800 | 94,900 |
| Cubic space necessary per B.H.P. - cu. in. | 279 | 397 | | 425 | 365 |

77. The figures from the table showing the comparioen of weights, etc., show that this engine would have an advantage over existing types as regards weight, and that it has a greater advantage as regards the space necessary for a given H.P. The writer purposely selected the most unfavorable conditions for making the comparison by using the aviation type, as there is no question but that these engines have received more real thought and development than any other gas engines. The weight of this engine could be further reduced by lightening the vital parts, such as the crank shaft, eccentric shaft and connecting rods, and by hand finishing the cylinder and crank case castings to thinner sections than shown, to between 550 and 575 pounds. This would mean one horse-power to each 2.9 to 3.2 pounds of engine at 1525 R.P.M.

78. Conclusions:-

a. Even when the vital parts are not designed with the lowest possible factor of safety, there is an indicated reduction in the "weight of engine per horsepower developed."

b. With the parts trimmed down and with a good deal of hand finishing, there is an indicated reduction of about one-half pound in the weight of the engine per horsepower developed.

c. A muffler can be used with this engine with less effect on the power developed than in present engines, since a very small quantity of exhaust gases are left in the cylinder. This might be a great advantage for certain purposes in aviation, and is absolutely necessary in motor cars. The effect of back pressure on the distribution and induction of the new charge is entirely removed, so that all cylinders can easily procure a proper charge.

d. No springs, valve rockers, etc., are present to give trouble. The valves are opened and closed positively at the proper time.

e. When used in a motor car, this engine should operate on low throttle better than present types, since practically all the charge in the cylinder is inflammable and ignition would always occur.

f. The shape of the combustion chamber is more favorable to the obtaining of higher thermal efficiency than that used in most poppett valve engines.

g. When lower compression pressures are used as required when the engine is to be operated on present grades of gasoline or kerosene, the gain in power would be greater than that shown in this case. The stroke of the head piston would be greater than when higher compression ratios are used. Therefore the head piston would have a greater effect in increasing the amount of fuel handled. The volumetric efficiency of this engine would be no lower for a low compression engine for a higher compression type as in the case already investigated. Since few automobile engines have a volumetric efficiency of over 75 per cent, the use of this type engine may result in a 33 per cent or even greater increase in power.

h. If more power were not desired, a greater drop in pressure during the inlet stroke might be used. This would assist in vaporizing the fuel.

i. A considerable decrease in the space required per horsepower developed is indicated. The fusilage of an

airplane or the engine compartment (hood) of a motor car may be made smaller if this engine is used. This would result in a reduction of the weight of the vehicle and should be credited to the engine.

j. All in all, this investigation shows that this type has attractive possibilities. It has prompted the writer to construct an engine of this type with a few variations from the construction shown to increase the thermal efficiency and to adapt it to motor car use. This engine is expected to be in operation in the near future.

[redacted]

Pocket has:
(Fig. 3-
(Fig. 6-

112
497.
THS
Fig. 3-4