COMBUSTION-CHAMBER DESIGN FOR OIL-ENGINES

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THE PRINCIPLES OF THE COMBUSTION-CHAMBER DESIGN FOR OIL-ENGINES WITH MECHANICAL INJECTION OF FUEL

by

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INTRODUCTION

The success of every oil-engine to a very large extent depends on the design of its combustion chamber.

A badly designed chamber causes bad combustion of fuel oil, accompanied by heavy carbonization of cylinders, contamination of lubricating oil, increased cylinder wear, sometimes cracking of cylinder covers and pistons, and also increased wear on all working parts of the engine primarily due to contamination of oil.

Yet, notwithstanding the importance of this problem, its fundamental principles are still very little known.

No theoretical investigation of it is possible, and, as the process of combustion in the cylinder is, unfortunately, invisible, designing a combustion chamber for a new engine remains perforce a matter for conjecture.

However, a careful analysis of experiences with engines of many types and a study of the history of the development of combustion chamber design should help us to understand better the fundamental requirements of a rational combustion chamber.

A critical survey of all these problems, the study of their history and various practical and theoretical considerations, the knowledge of which is essential when designing a combustion chamber, form the subject matter of this book.

CHAPTER I

HISTORICAL NOTES ON THE DEVELOPMENT OF COMBUSTION-CHAMBER DESIGN

In the early days of diesel engines, no consideration at all was given to the shape of the combustion chamber, which was simply left to take care of itself, and whatever shape was given it was primarily due to constructional requirements of the engine design.

Although the first diesel engine was intended by Dr. Rudolf Diesel to work with mechanical injection of fuel, he could never get satisfactory results from this method of injection, and eventually the engine was re-designed to run with air injection.

The fundamental difference between the two types of injection is, that with air injection the fuel is first atomized and then the combustion of it is further assisted by compressed air supplied to the fuel injector under high pressure, whereas with mechanical injection the fuel is atomized and burned without compressed air.

The most important effect of injecting air was the creation of the turbulence in the combustion chamber, which, as we know it now, is of paramount importance for good combustion.

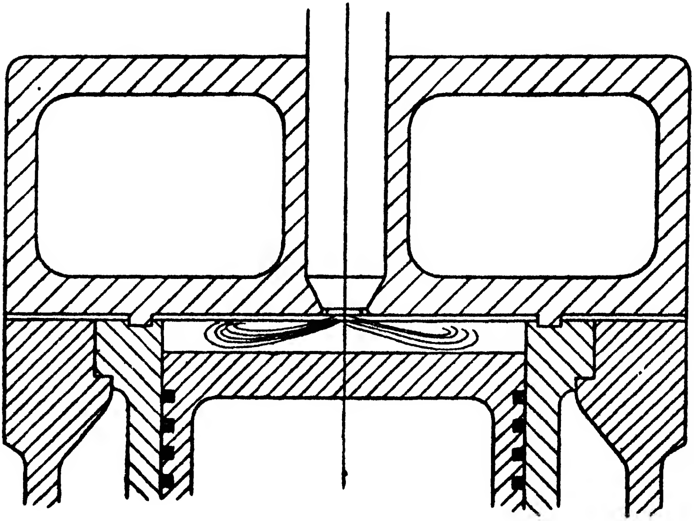


Fig. 1

Well-designed diesel engines with air injection can work with absolutely invisible exhaust, even if their combustion spaces are not ideal from the point of view of present-day knowledge.

The typical combustion chamber of an early diesel is shown in Fig. 1.

In diesel engines, as is, of course, well known, the ignition of fuel is caused by the heat of compression of working air in the cylinder.

Many attempts have been made by designers to develop a satisfactory oil engine with mechanical injection, but, due to lack of technical knowledge of both mechanical atomization of fuel and requirements of fuel combustion in a confined space, all these attempts proved abortive for a long time.

The first engines with mechanical injection in which heavy oil could be burned with satisfactory practical results, although at a low efficiency, were the so-called hot bulb engines.

In this type of oil engines, although the fuel oil was injected by mechanical means, without any high pressure air to assist atomization, the cylinder head was so constructed that part of it could be pre-heated by external blow-lamps and this portion of the cylinder head was further maintained in a hot state by the heat of combustion.

In these engines the compression pressure was usually fairly low, and the ignition of fuel was caused by local overheating of a mixture of fuel vapours and air, partly due to heat of compression and partly due to hot cylinder head.

A very typical example of the combustion-space in these engines is shown in Fig. 2 as it was built by a firm of Mietz and Weiss, about twenty-five to thirty years ago.

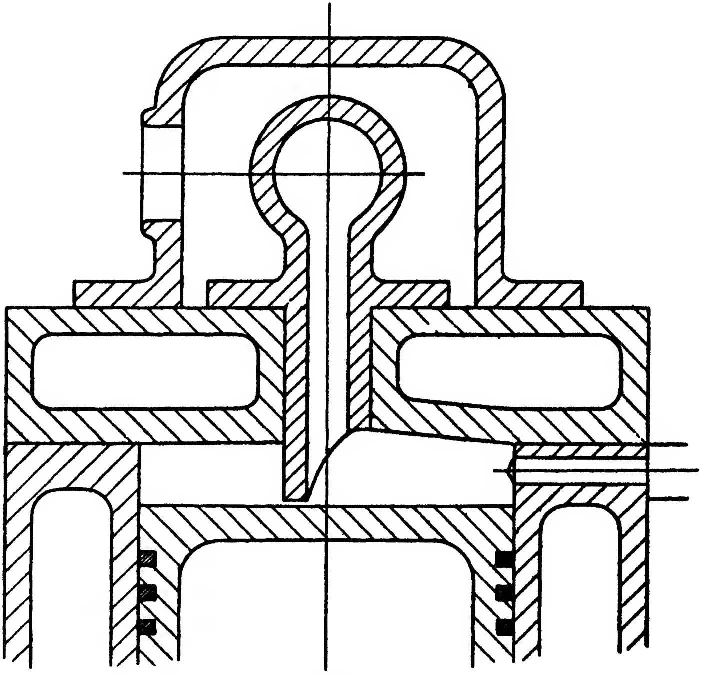


FIG. 2

The combustion chamber is divided into two parts—one of a larger volume inside the cylinder, and the smaller one in the hot bulb.

The hot bulb is spherical in shape, and is connected to the major combustion space by a fairly narrow throat.

The fuel is " squirted " on to the tongue, which is formed by the continuation of the portion of the hot bulb throat protruding into the major combustion space.

This type of combustion chamber is really a development of the very old idea of the ignition tube, which was originally used to ignite the charge in the early gas engines.

The theory of this combustion chamber was not perfectly understood at first, and the bulb was originally supposed only to act as an ignition tube.

It is, however, well known now that the effect of this type of hot bulb was much more important than was then imagined.

The fuel charge which is squirted on the protruding tongue of the hot bulb throat is partly vaporized, and these vapours, together with some of the remaining drops of the unburned fuel, are being blown into the hot bulb by the inrush of air from the major combustion chamber during the compression stroke, where the remaining drops of liquid fuel are being vaporized, and then the charge present in the hot bulb ignites and burns almost instantaneously, at constant volume, with a considerable rise in pressure. The amount of air in the hot bulb is, however, only sufficient to burn a very small portion of the fuel charge, and thus when this partly burned mixture is expelled from the hot bulb by the explosion, the combustion of the remaining fuel continues in the major chamber.

The explosion in the hot bulb produces a considerable turbulence in the major chamber, which facilitates the combustion of the remaining portion of fuel.

In the first engines built on this principle more than thirty years ago, the compression pressure was very low, usually three atmospheres or about, and fuel consumption was sometimes more than a pound per 1 B.H.P. per hour.

As more experience was, however, being gained with hot bulb engines, the compression pressure was gradually increased, and consequently better combustion efficiency and lower fuel consumption were obtained.

This type of combustion chamber still survives, although in somewhat modified form and under different new names, and with much higher compression ratio.

Another notable example of the hot bulb engine type is shown in Fig. 3.

In the latter type of cylinder head the fuel is sprayed directly on the hot part of the head.

This type of combustion chamber is still used in some two-cycle oil engines with crankcase type scavenging.

In course of time, however, all the original low compression hot bulb engines gradually developed into what is at present known as the semi-diesel engines. ([illegible])

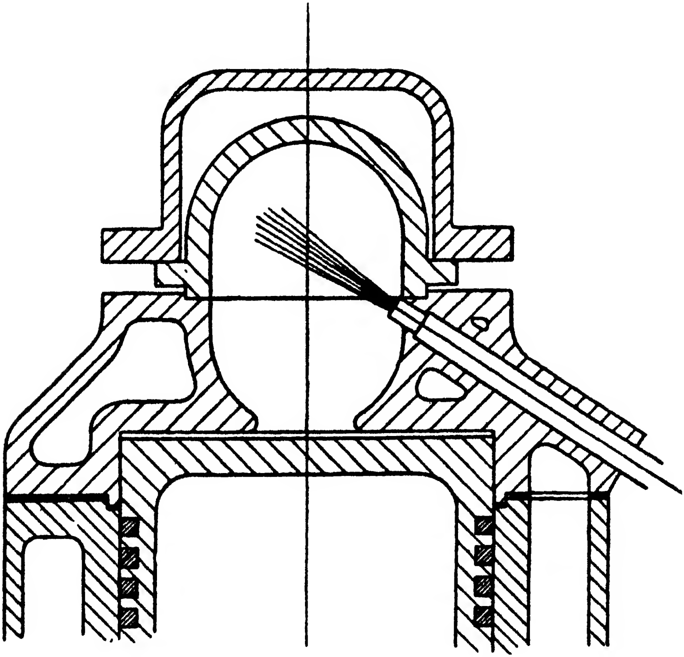


FIG. 3

The principal difference between the early low compression hot bulb engine and the later semi-diesel engine is in the compression ratio.

While in the hot bulb type engines the compression pressure varied between three and ten atmospheres, and consequently the ignition of fuel was caused mainly by the heat of the hot bulb, the compression pressure in semi-diesel engines was raised to sixteen and even to twenty atmospheres, and the compression temperature of air in the presence of hot bulb was sufficient to cause both the vaporization and the ignition of fuel. The hot bulb thus only assisted the ignition, and was not the main cause of it. Its chief purpose was really to help the atomization and vaporization of fuel, for in those days various complex problems of the modern mechanical injection were not well enough understood.

The next historically very important step was the appearance on the market of the first pre-combustion chamber type engine invented by a Dutch carpenter Brons. This head was first patented in 1904 and is shown in Fig. 4.

The fuel was supplied by gravity during the suction stroke into a small tumbler-shaped chamber, the volume of which was not more than from one-fifth to one-eighth of the total compression space volume. The quantity of fuel was regulated by a mechanically operated valve and an adjustable needle, through which the fuel passed.

During compression stroke, the fuel in the tumbler was heated, evaporated and ignited, so that the explosion of the air and fuel mixture in the tumbler expelled the unburned fuel from the tumbler through small holes in its bottom corner, thus further atomizing it

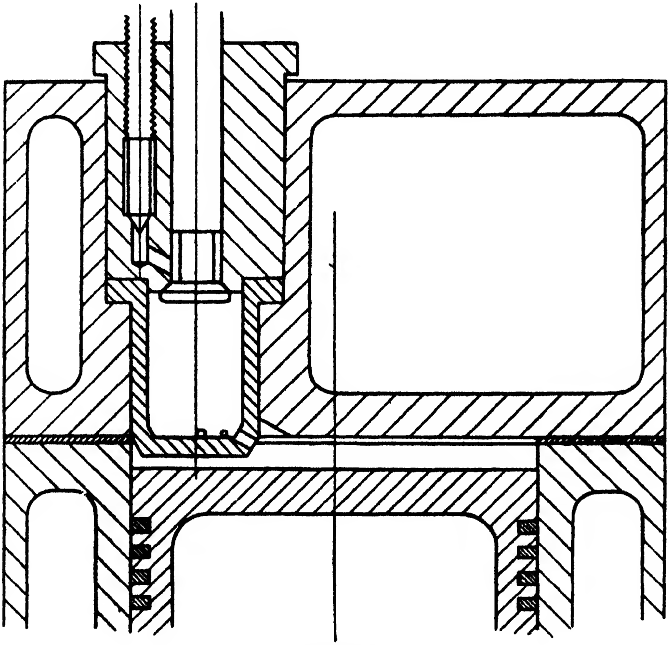


FIG. 4

and creating a turbulence in the major portion of the combustion chamber which assisted the combustion of the remaining unburned fuel.

The Brons engine was very popular about twenty-five years ago, and many different new types of pre-combustion chambers have been further evolved from it, which are still extensively used now, though under different names.

One of the disadvantages of the original Brons combustion chamber was the high compression pressure required to ensure easy starting. This difficulty has now been overcome by fitting an electrically heated plug into the pre-combustion chamber.

From this brief historical survey it can be observed, that in the earliest oil engines with mechanical injection of fuel their designers relied more on vaporization and ignition of oil by means of pre-heated surfaces, and mixing it with combustible air through turbulence created by the explosion in the hot bulb or the pre-combustion chamber.

The first attempts to burn oil fuel satisfactorily injecting it directly into the main combustion chamber by purely mechanical means, without any assistance of compressed air, since the earliest attempts by Dr. R. Diesel, were made by Messrs. Vickers, Ltd., in an engine which they have developed for the British Admiralty, for use in the submarines.

The combustion chamber used was somewhat similar to the one originally fitted in

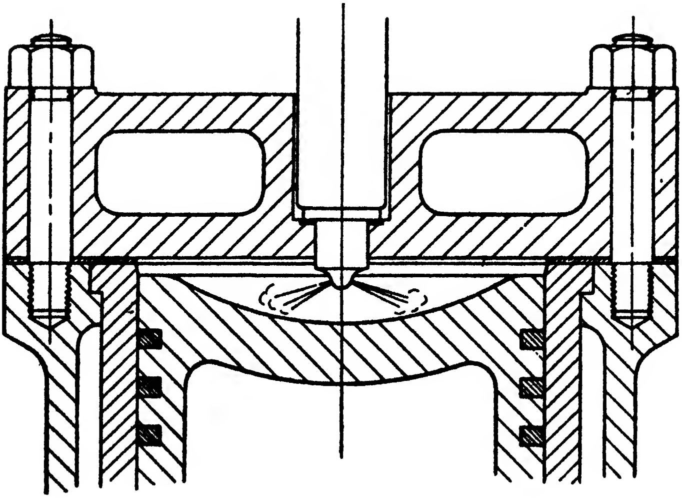


FIG. 5

diesel engines with air injection, but with slightly more dished piston top.

In this instance it was intended to obtain good combustion by perfecting the method of fuel injection.

As it is well known, a fairly satisfactory result was obtained.

This engine, the combustion chamber of which is shown in Fig. 5, marked the turning point in the development of the solid injection oil engine, and from that time more attention was being paid to the quality of fuel injection and atomization.

It would be true to say that in the first hot bulb and semi-diesel engines the fuel injection was very unsatisfactory. The fuel was simply squirted on to the hot surface of the hot bulb, or in case of Brons engine simply dropped into the tumbler by gravity. No attention whatever was paid to the rate of injection, or to the injection pressure. Any old pump and pump drive was used, and correct atomization of fuel was not even attempted.

When the first results with direct mechanical injection of fuel without any hot bulbs in Vickers engine became known, the importance of a technically correct fuel injection process was first realized.

In the meantime, the builders of the semi-diesel engines were also making considerable progress, by gradually eliminating the hot bulb and improving the quality of the fuel injection, until the semi-diesel engine became what is known now as a compression ignition engine with mechanical injection of fuel.

In very much the same way as was with

sails, which were still being fitted to steamers for nearly fifty years after the introduction of steam, hot bulb, though much reduced in size, was used for some years after a satisfactory injection system had been evolved, and is even now still used in one or two high compression engines.

Almost simultaneously with the beginning of the development of correct injection systems, the designers began to pay more attention to the shape of the combustion chamber and the movements of air in it both during the injection and combustion periods.

The air movements in the combustion space during injection period, as it is now definitely ascertained, are of the greatest importance for efficient combustion.

The first attempt of creating an organized rotational type of turbulence can be traced back to the patent No. 28,753 of 1909, by L. Laurin, Fig. 6.

Whether this construction was used in its original form, the writer is not aware of, but there is no doubt that this was the prototype of a great multitude of spherical type combustion chambers with rotational turbulence created by tangentially arranged throat, which were evolved during the last twenty years, and are still very popular even now, being made under different names and patents.

Producing turbulence in a spherical combustion chamber of this type is fairly easy, as it only requires the throat connecting it to the cylinder to be arranged tangentially to the surface of the sphere. On the diameter of the throat depends the circumferential

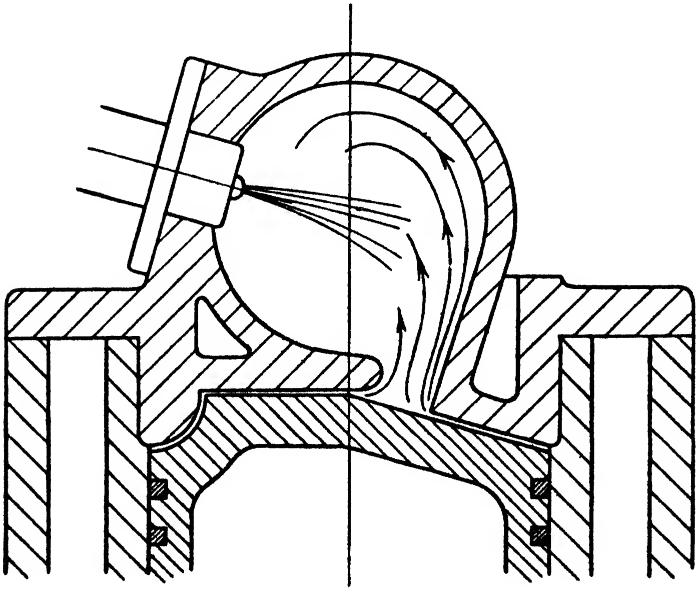


Fig. 6

velocity of the swirl, which can thus be controlled.

The same type rotational turbulence can be produced in the open type combustion space somewhat similar in shape to the one used originally in the first diesel engines with air injection. Such turbulence can be produced by a special design of inlet valves, known as " masked valves," shown in Fig. 7.

The original inventor of this construction was the well-known Swedish designer Hesselman, who for the first time built an engine with such turbulence in 1924.

Although the spherical or partly spherical combustion chamber with tangential throat was invented earlier, it was only after Hesselman demonstrated in his engine the beneficial effect of the rotational turbulence on efficiency of combustion that the spherical combustion chamber with rotational swirl became really popular.

Hesselman's combustion chamber with organized rotational turbulence produced by masked valves is shown in Fig. 26.

This development represents a notable step in the evolution of a really efficient combustion chamber. As we shall see later, all further developments in combustion chambers represent various compromises and combinations of the basic designs of the different

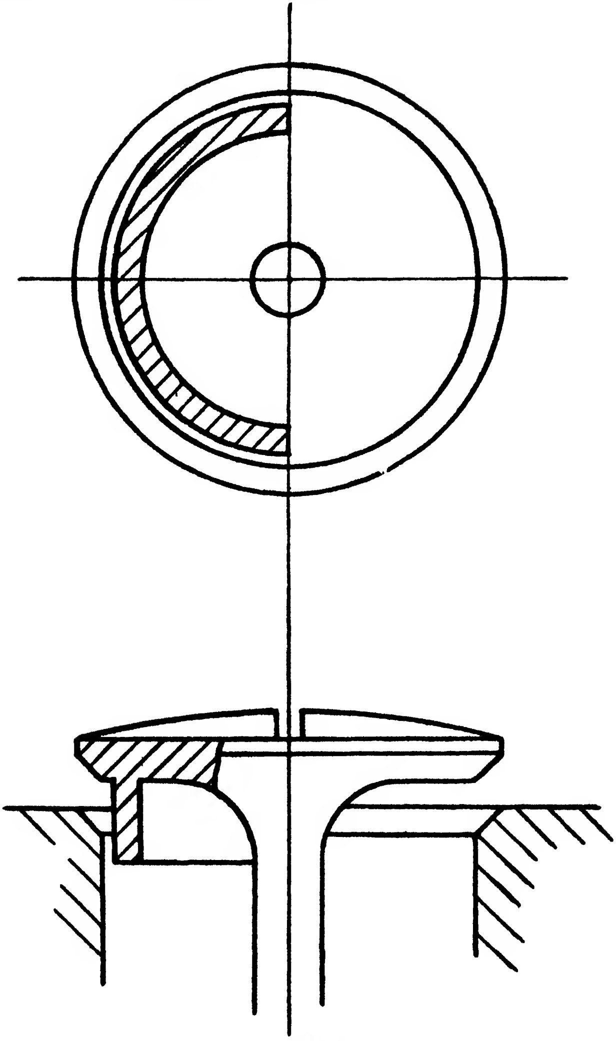


FIG. 7

combustion chambers mentioned in this brief historical survey.

The combustion chambers as shown in Figs. 1, 5, 8 and 10, with direct injection of fuel and low velocity turbulence can be classed as " open type " combustion chambers. The combustion chambers shown in Figs. 2, 4, 20 and 21 are typical representatives of the pre-combustion chamber design, and Figs. 6, 23, 24 and 25 show the " high velocity turbulence " type of combustion chambers.

CHAPTER II

THE BASIC REQUIREMENTS OF THE DESIGN OF A RATIONAL COMBUSTION CHAMBER

THE main purpose of every combustion chamber is to ensure an efficient combustion of fuel.

In the original air injection diesel engines, the fuel was atomized by means of high pressure air and thoroughly mixed with air in the combustion chamber during the combustion period also due to turbulence created by the high pressure injection air, so that in these engines combustion did not present any particularly great difficulties.

In diesel engines with mechanical injection of fuel, the fuel is injected in the shape of a pulverized conical spray, which does not create any turbulence, and does not mix readily with compressed air.

Consequently, some relative motion of air is required to facilitate the mixing of pulverized fuel and air available for combustion.

This motion of air in the combustion

chamber in relation to the atomized fuel jet is usually called the "turbulence."

It is created naturally during the compression stroke, and sometimes even during suction stroke, practically in every known type of combustion chamber.

In the combustion chamber as shown in Fig. 1, the air is moving during compression stroke in the same direction as the piston and parallel to cylinder axis.

If, however, the piston top is cup-shaped in the centre, and its sides come close up to the cylinder head, then at the end of compression there will be quite a considerable stream of air created, directed from all sides radially towards the axis of the cylinder, due to displacement of air around the cup-shaped cavity in the centre of the piston top, as shown in Fig. 28.

The conically shaped jet of atomized fuel has the expansion angle of approximately 20°, although this angle varies considerably, and depends, amongst other things, on the length of the hole in the nozzle, compression pressure, etc.

To assist the burning of atomized fuel, the relative position of the fuel jet in the combustion chamber should be such that the direction of the stream of air during the injection period *should be across the fuel jet.*

It is obvious from elementary practical consideration that this is the only rational relative position of the fuel jet and air stream ;

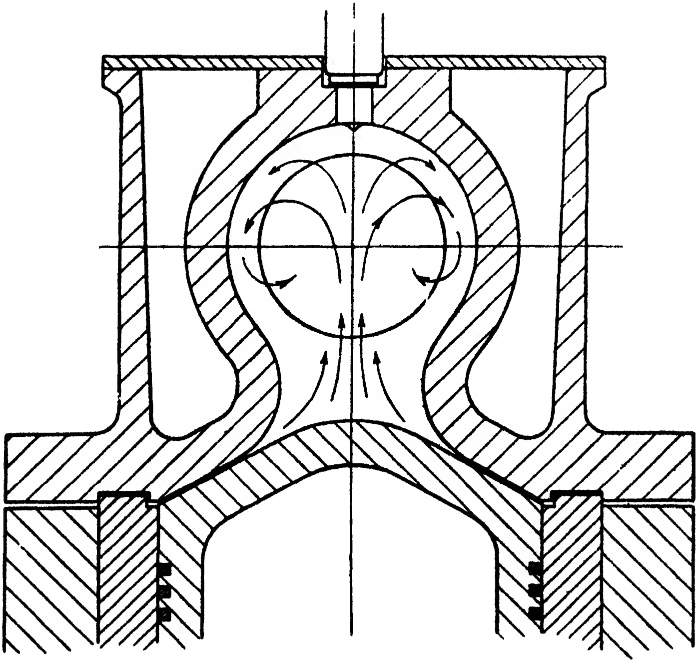


FIG. 8a

for should the latter be in the same or opposite direction to the fuel jet, as soon as the combustion begins, products of combustion instead of fresh air will be carried through and mixed with the yet unburned fuel in the jet.

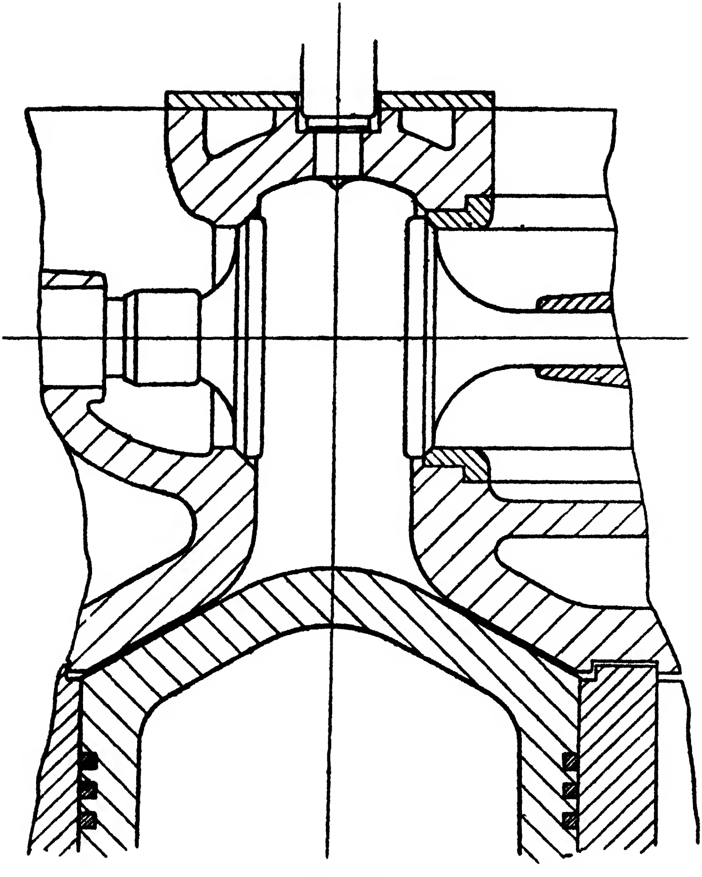


FIG 8b

This rule, of course, does not apply to various pre-combustion chamber type heads, where various other important requirements peculiar to each particular design of chamber are sometimes of greater importance.

In view of the importance of the relative position and directions of the fuel jet and the air turbulence, the problems of the formation of air streams in combustion chambers during the end of compression stroke and during the combustion period should be well studied.

Below two examples are given of the turbulence produced during compression in slightly different combustion chambers.

The first, Fig. 8a, represents a very typical combustion chamber of a horizontal type diesel engine.

The air comes into the chamber through the central throat in a solid stream, and meeting the opposite wall of the chamber, divides into two streams, each curling itself up in opposite directions.

Two to four fuel jets at about 70° or 80° included angle in one plane usually give the best results with this combustion chamber, and generally speaking this should be considered a very satisfactory combustion chamber design.

Sometimes the throat is made somewhat wider, without departing from the principle of the construction.

Fig. 9 shows a similar construction, but with somewhat different arrangement of the throat, which is made tangential to the one side of the circular chamber.

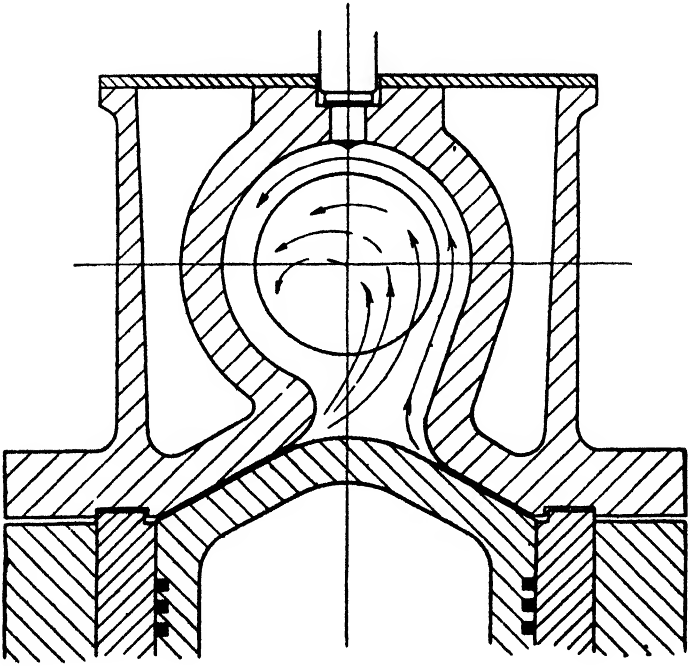


FIG. 9

In this case the turbulence in the combustion chamber will be of a very different type—one continuous vortex. A single jet nozzle would probably give the best result with this class turbulence.

In both cases the fuel jet is given a good opportunity to mix with the air.

It is also of the greatest importance, that all the air available for combustion purposes in the cylinder should be concentrated in the combustion chamber.

For this reason, all clearances and air pockets outside the combustion chamber proper should be reduced to the minimum or even eliminated. Unless this requirement is taken into consideration when designing the combustion chamber, difficulties may be experienced with obtaining a high mean effective pressure in the cylinder, although the fuel consumption may be quite satisfactory at low mean pressures.

The whole problem of efficient combustion of fuel thus boils down to a few very simple rules.

First, get all the air available in the cylinder concentrated in one space, as compact as possible, and of a shape suitable for distribution of jets radiating from one point.

Second, get all fuel jets, and as few in number as possible, evenly distributed over the combustion space, radiating from one point in the combustion space so as to be in a favourable relative position to the flow of air during compression stroke.

Third, take care that the combustion space is so shaped that a definite stream of air, preferably of moderate velocity, is produced in it during the end of compression stroke.

Fourth, the combustion chamber should never be so narrow or flat as to cause the jet to graze the sides.

If, notwithstanding all the precaution taken in the design of combustion chamber, you do not get a satisfactory result from it, you may be sure that it is the fuel injection arrangement which is at fault.

Very often, particularly with the open type combustion chamber, the designers find that better results are obtained with a large number of jets—often as many as ten or twelve.

It must be remembered, that even with a perfectly flat type of combustion chamber, such as shown in Fig. 1, and using very wide angle between jets, no useful purpose is served if more than five jets are used, for otherwise the expanding cones of the jets will overlap.

Therefore, if it is found that more than five jets are required, the design of fuel injection must be wrong.

The most frequent fault met with in the design of the fuel injection equipment is a too long injection period in terms of the crank angle.

This bears a very important relationship to the turbulence in the combustion chamber, for in most of combustion space designs the turbulence in them is reversed as soon as the piston gets over the top dead centre. If a very long injection period is used, terminating only after the top dead centre, as soon as the turbulence in the combustion chamber is reversed, *products of combustion* and *not* fresh air will be passing through the fuel jet, thus retarding the combustion.

As this is a very important point, it will be dealt with in a separate chapter.

CHAPTER III

HEAT LOSSES DURING COMBUSTION, TEMPERATURE OF COMBUSTION AND ITS INFLUENCE ON THERMAL EFFICIENCY

Most of the heat losses during combustion from burning fuel to combustion space walls are due to two causes :

(1) Direct radiation.
(2) By direct contact with hot gases.

In the direct radiation losses temperature plays a very important part, for the amount of heat radiated is proportional to T^4 of combustion. Notwithstanding this, the time element is also very important, and it is well known that if the combustion of fuel is very slow and inefficient, the heat losses to cylinder walls may be excessive, although the temperature of combustion may be lower than with more efficient combustion process.

For this reason the high combustion temperature, which is a sign of efficient combustion, is very desirable.

Heat losses by direct contact between hot gases and walls depend mainly on the velocity of hot gases in relation to cooled walls. It can be definitely stated that the interchange of heat between hot gas and metal wall is practically proportional to this relative velocity.

Consequently, in all combustion chambers having high velocity turbulence, the heat losses from hot gases to combustion chamber walls are very heavy.

The same, of course, applies to the compression of air ; in all combustion chambers with high velocity turbulence more heat is lost from air to cylinder walls during compression, and consequently a higher compression ratio or electric heating plugs are required to ensure easy starting.

As far as heat losses from gas to cylinder walls are concerned, whether they are big or small, they do not affect much the overall thermal efficiency of the process in the cylinder, except maybe in some extreme cases.

The same, of course, applies to savings in heat losses, which have not been found worth the trouble.

As a matter of fact, a temperature above 2100° C. abs. at the end of combustion seems to have a retarding effect on further combustion process.

However, although excessive heat losses from gases to combustion chamber walls during combustion are not detrimental to thermal efficiency of the engine, they produce considerable heat stresses in cylinder head walls, which may eventually result in cracks.

The heat losses during compression stroke are very undesirable, for the starting of engine from cold becomes more difficult, and the running of engine may become rough.

From this point of view the combustion chambers of the " open " type are much more advantageous.

To overcome the difficulties of high heat stresses in combustion chambers with high turbulence velocity, the designers had to resort to using metals for cylinder head castings having better heat conductivity and lower modulus of elasticity.

The importance of this will be seen from the approximate formula for the heat stresses in the watercooled cylinder walls, viz.,

$$(1)\ S = C \times E \times c \times f \times q \times G \times \text{I.H.P.} \times s/D^2\ (1 + h)g \text{ kg. per cm.}^2$$

where : E = modulus of elasticity, in kg/cm.^{2}
f = proportion of heat of combustion absorbed by walls, usually from ·25 to ·32
q = fuel consumption in kg. per I.H.P./
c = coefficient of linear expansion of metal of the walls

G = calorific value of fuel, in Cal/kg.
s = thickness of wall, in cm.
D = cylinder diameter in cm.
h = stroke to bore ratio
g = coefficient of heat transmission, in calories per 1 cm.^{2} × 1 cm. thick per 1°C. temperature difference
C = coefficient, depending on the units chosen
S = heat stresses in wall, kg/cm.^{2}

Whatever the actual numerical value of this stress is, from the eq. (1) we can make the following conclusions :—

(1) The greater the modulus of elasticity of metal, the higher the stress.

(2) The same applies to the coefficient of linear expansion.

(3) The thicker the walls the higher the heat stresses.

(4) The greater the coefficient of heat transmission of the metal of the wall, the smaller the stress, which is due to smaller temperature drop through the wall.

Consequently for engines with high power output and heavy heat losses to the walls (with high velocity turbulence) a metal with a small modulus of elasticity and of high conductivity is desirable, to reduce heat stresses. Some aluminium alloys would be found suitable for this purpose.

The variation of temperature in the combustion chamber walls is very much greater near the inner surface, and it changes comparatively very little on the cooled side.

In the early oil engines with mechanical injection of fuel it was considered desirable to have part of the cylinder head hot, to assist combustion.

As the knowledge of both the combustion and the fuel injection advanced, it was found that hot surfaces in combustion chambers are totally unnecessary. As a matter of fact, a very hot surface in the path of oil jet may even be harmful, for the oil striking the hot surface "cracks," *i.e.*, decomposes, and thus produces "dusty" exhaust.

In the old semi-diesel engines, to avoid fuel "cracking," it was found necessary to inject into the hot bulb a considerable quantity of water, to keep its temperature within the desirable limits. This quantity of cooling water, which was lost with the exhaust, was sometimes as much as ten times the quantity of fuel. The effect of water injection on the cleanness of the exhaust and engine efficiency was, however, very marked.

CHAPTER IV

THE IMPORTANCE OF THE CORRECT SHAPE OF COMBUSTION CHAMBER DESIGN

WHEN designing a combustion chamber, the shape of every portion of it should be very carefully considered, not only as it appears when the piston is in its top dead centre, but also with the piston in several intermediate positions from the beginning of the injection to the end of combustion period.

The approximate velocities and expected directions of the air streams in the combustion chamber should be plotted and the influences of these streams on the shape of the fuel jets should be carefully studied.

If the air stream is across the fuel jets, it is certainly reasonable to expect that fuel jets will be either deflected or at least bent to conform to the direction of the air flow.

It must also be remembered that as soon as piston gets over its dead centre, the direction of its movement is reversed, which is bound to affect the turbulence in the combustion chamber in one way or another.

It is also very important to consider carefully the possible influence of the shape of combustion chamber on the heat losses by modifying the air velocities in various passages and also the ratio of the combustion chamber surface exposed to the heat of combustion to its volume.

How much it can be affected by the shape of combustion chamber can be seen from the following example.

We will consider now two very simple types of combustion chambers for the same engine, one perfectly flat and cylindrical, the other—spherical.

Cylinder dia. = D″
Stroke = S″
Stroke to bore ratio = S/D = L
Compression ratio = 14 to 1.
Cylinder volume V = $\cdot785 \times D^2 \times S =$

(2) $= \cdot785\, D^3 \times L$

Combustion chamber volume—

(3) $v = \cdot785\, D^3 \times L/13 = \cdot06\, D^3 \times L$ cu. inches.

The surface of a flat cylindrical combustion space is $S_1 = \cdot785 \times 2 \times D^2 + \pi \times D^2 \times L/13 =$

(4) $= 1\cdot57 \times D^2 + \cdot242 \times D^2 \times L$ sq. inches.

Its surface-to-volume ratio will be

$$r_1 = (1\cdot5 \times D^2 + \cdot242 \times D^2 \times L) \,/\, \cdot06 \times D^3 \times L$$

(5) $= (26\cdot2 + 4\cdot04 \times L)/D \times L$

If we assume $L = 1\cdot5$, which is a very popular ratio, then

(6) $r_1 = 21\cdot7/D$.

From this we can see that the surface-to-volume ratio of a flat cylindrical type combustion chamber for the same stroke-to-bore ratio is inversely proportional to cylinder diameter.

For a spherical combustion chamber, we will consider that all the combustion is taking place inside the spherical chamber, and that all other surfaces are not exposed to direct flame of combustion.

The volume of the spherical chamber will have to be the same as for the flat type chamber.

Volume of spherical chamber

(7) $v = \cdot06\ D^3 \times L = \pi d^3/6$ cu. inches, where d = spherical chamber diameter.

From equation (7), we can find the value of *d* in relation to *D* and *L*.

(8) $d = \cdot485 \times D \sqrt[3]{L}$ inches

Surface of the chamber—

(9) $S_2 = \pi \times d^2 = {\cdot}74\, D^2 \times L^{2/3}$ sq. inches.

ratio $s/v = r_2 = {\cdot}74\, D^2 \times L^{2/3} / {\cdot}06\, D^3\, L =$

(10) $= 12{\cdot}3/D \times \sqrt[3]{L}$

If we assume again $L = 1{\cdot}5$, then

(11) $r_2 = 10{\cdot}7/D$.

We can see from comparison between eq. (6) and (11) that the surface exposed to direct flame of combustion in case of the spherical chamber is only about one-half of that in a flat combustion chamber, for the same stroke-to-bore ratio.

For this reason, smaller total heat losses to combustion chamber walls may be expected in a spherical chamber during combustion.

This condition, however, may be completely changed if a high velocity turbulence is provided in the spherical chamber.

From the point of view of combustion, the spherical chamber appears to be more suitable, for it provides a more concentrated volume of air for combustion, and permits a better distribution of fuel jets.

A perfectly flat type cylindrical combustion chamber is obviously not rational, for it does not provide for the expansion of the fuel jets, and due to its flatness it is almost impossible in it to avoid the jets grazing the piston top. The modified form of it, as for example is used in Hesselman's engine shown in Fig. 26, is certainly much superior, and is well adapted for the type of rotational turbulence used in it.

The open type combustion chamber as shown in Fig. 28 should also be very satisfactory, and with efficient injection system could probably be used even without the air swirl.

A combustion chamber approaching spherical in shape but with a wide throat connecting it to the working cylinder should also be considered an " open " type chamber, but it would be wrong to suggest that a perfectly spherical shape is the best for it from point of view of efficiency.

Considering the shape of fuel jets and the natural air turbulence in it, the best shape for a chamber of this type is that of a double-spherical lens with different curvature for each side. A combustion chamber of this type is shown in Fig. 10.

The top half of it is almost half a sphere, but the bottom half of it is described by a radius from the centre situated in the injector nozzle.

The air turbulence is indicated by curved lines with arrows.

Its influence on the shape of the jets is indicated in the shaded area representing the jets. The throat of this combustion chamber is fairly wide, so that little heat is lost by air during combustion and the starting of engine is very easy even if the engine is turned by hand.

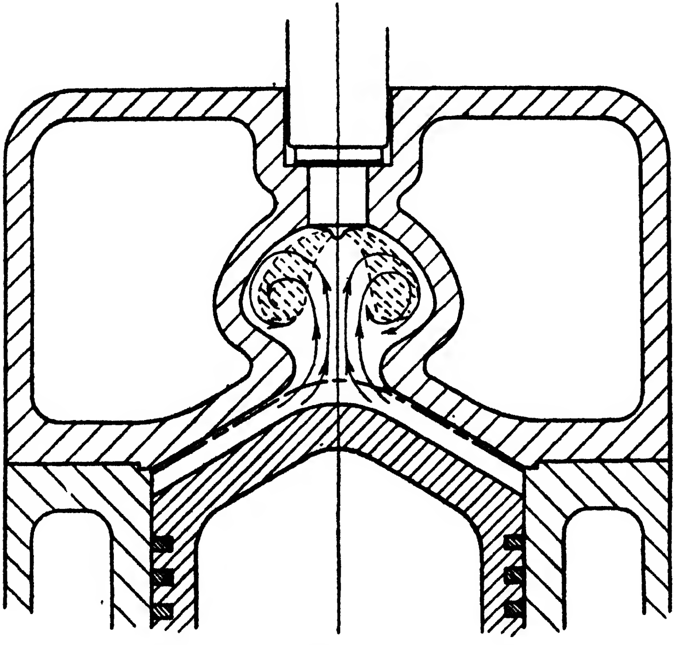


FIG. 10

From actual experience this combustion

chamber was found to give very efficient combustion, extraordinarily smooth running, invisible exhaust and is capable of developing a very high mean effective pressure.

The reason for the difference in curvature in the top and bottom halves of this cover is entirely due to considerations of the shape of fuel jets and air turbulence. Actual experience has shown that this is very important, and that conical shape for its top half or purely spherical shape does not give equally satisfactory results with moderate velocity turbulence as used in this combustion space.

The velocity of air stream for this type of combustion space at the end of compression stroke can easily be calculated from the following considerations.

When piston approaches its top dead centre, the cylinder volume decreases, but combustion chamber volume remains constant. Assuming that pressure and temperature of compressed air both in the cylinder and in the combustion space is always the same, and knowing the variation in cylinder volume between two positions of the piston, it is possible to calculate the quantity of air which passes through the throat, and its velocity.

Fig. 11 gives a curve showing the variation in this velocity for a combustion chamber of the type illustrated in Fig. 10.

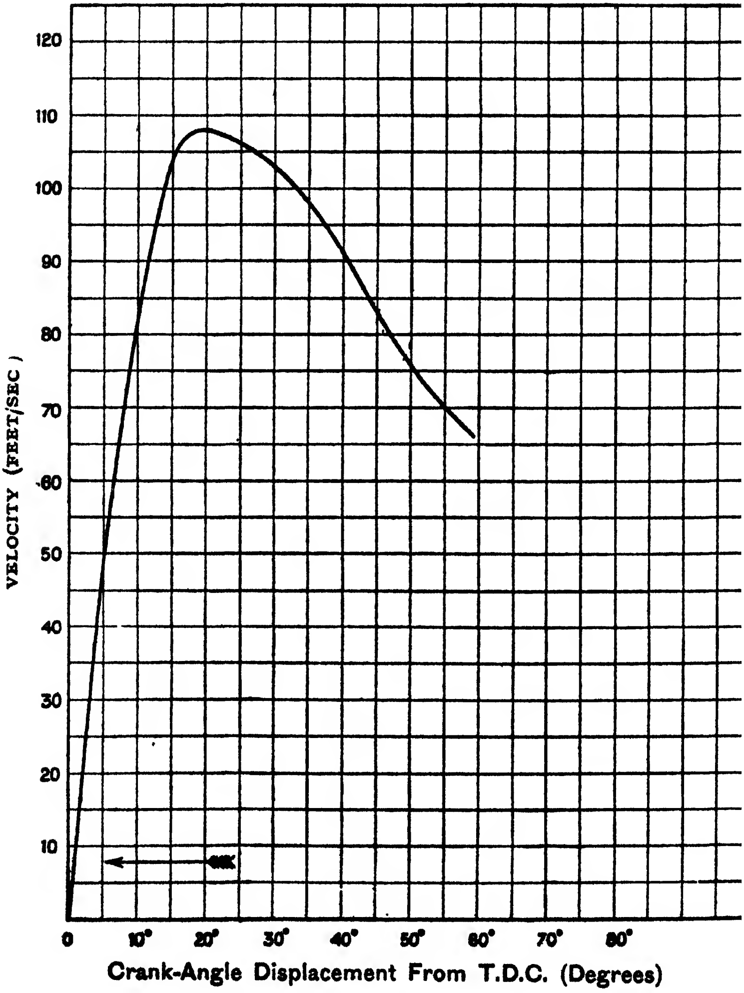

120
110
100
90
80
70
60
50
40
30
20
10
VELOCITY (FEET/SEC)
0
10°
20°
30°
40°
50°
60°
70°
80°
Crank-Angle Displacement From T.D.C. (Degrees)

FIG. 11

The engine is $3\frac{1}{2}''$ bore by $5\frac{1}{2}''$ stroke, capable of running at 2,500 r.p.m., and the velocity of air is calculated for the throat diameter of ·75″ at 1,000 r.p.m. The actual diameter of throat used is considerably larger, and consequently the velocity is less in the inverse proportion to the area of the throat.

Air turbulence velocity can also be easily calculated on the same principle for any open type combustion chamber, and the velocity of the rotational turbulence with masked valves can be calculated from the velocity of air through the inlet valve and its position in the cylinder cover.

CHAPTER V

EFFICIENCY OF FUEL INJECTION AND ITS INFLUENCE ON COMBUSTION

FUEL injection efficiency is of the greatest importance in all combustion chambers of the open type or having low velocity turbulence.

It is interesting to note, that all the first oil engines with mechanical injection of fuel have been either of the so-called hot bulb type, or of the pre-combustion chamber type.

In hot bulb engines the injected fuel was not atomized at all. It was simply squirted through a small hole on an incandescent surface of the hot bulb, and the hot bulb was relied upon to partly atomize and partly vaporize the fuel jet.

From point of view of the modern knowledge of fuel injection, the efficiency of the early type of injection systems was *nil.*

It could not be used at present in any open type combustion chamber with any hope of even moderate success. Such important factors as injection pressure, rate of fuel, delivery, constant spray velocity, length of injection period, etc., were entirely unknown and never considered. With the hot bulb, it was reasonably satisfactory for the requirements of its time.

Even modern fuel injection equipment presents such technically complicated problems that many manufacturers are still using pre-combustion chambers which do not require a high injection efficiency.

That injection efficiency is really of small importance with all pre-combustion chambers is well illustrated in the forerunner of all modern pre-combustion type engines, the Brons engine, shown in Fig. 4.

In this engine the fuel is not even injected at all, but drops by gravity into a cup during suction stroke, and is vaporized during compression.

It is, however, obvious from its mode of operation that only distilled oils can be used with all, both hot bulb and pre-combustion, chamber type engines.

Apart from exacting technical requirements of the modern fuel injection equipment, there are some very important points which must be considered in conjunction with the cylinder cover design and the type of turbulence used in it.

The most important factor is the length of the fuel injection period in terms of crank angle.

If the injection period is too long, part of the fuel will be injected after the top dead centre, which should be considered highly unsatisfactory.

It is very important, in all open type combustion spaces, that all the fuel at full load should be injected before the t.d.c. and the injection must be completed at least 5° before it.

If this requirement is not satisfied, the indicator diagram will have a very thin top and show a lot of afterburning, and the engine thermal efficiency will be low.

This is to a great extent due to the reversal of the direction of turbulence, which usually occurs after the piston passes the top dead centre.

As the injection of fuel usually begins at about from 20° to 15° before t.d.c. it follows that the length of the injection period should not be more than from 10° to 15° crank angle.

This condition is not very easy to satisfy, particularly in four cycle engines, as it will be found that large pump plungers and very fast cams are necessary.

With pre-combustion chamber engines or in combustion chambers with high velocity turbulence the length of injection period is not of such great importance, as the preliminary combustion in the pre-combustion chamber is more like explosion, and occurs after most of the fuel has already been injected.

Apart from the length of the injection period, the position of the injector in the chamber and the directions of the fuel jets are also of considerable importance.

With the combustion chamber illustrated in Fig. 10, the best position for the injector is in the centre of the combustion chamber, with the jets at about 70° included total angle.

Two jets were found quite satisfactory even in engines of fairly large size, although with larger size engines four jets will probably be necessary.

In combustion chambers of the "flat" type, such as illustrated in Figs. 1 and 5, usually five jets are found satisfactory.

If five jets do not give good results, the technical side of the fuel injection equipment should be investigated, provided the combustion chamber is designed according to principles recommended in Chapter II, for it is most unlikely that more than five jets should be necessary with both correctly designed combustion chamber and efficient injection system.

There is one more very important factor to consider with regard to fuel injection, which is the penetrating power of the fuel jet.

Many experiments have been carried out in the past for determining the factors which are mainly responsible for the greater or smaller penetration, but the results of all these tests appear to be contradictory.

It is, of course, quite natural to expect that a fuel jet issuing from an orifice of larger diameter will have a greater penetrating power than a jet from a smaller diameter orifice ; and this obvious fact was confirmed by experimental data. With regard to many other factors, the results appear to be confusing.

For instance, a higher injection pressure does not always improve the penetration, for sometimes the fuel gets better atomized and thus the jet more quickly loses its penetrating power. Higher compression pressure in the cylinder certainly also reduces the penetrating power of the fuel spray.

The turbulence also affects penetration, for it can deflect or curve the jet. The quantity of fuel injected per charge also affects the penetration very considerably, and so does the chattering of the needle valve in the injector. The increased length of the holes in the nozzle reduces the expansion angle of the jet and increases its penetrating power.

How important some of these factors are can be seen from the fact that in the combustion chamber illustrated in Fig. 10, two holes of ·025″ diameter could be used without the jets reaching the opposite surface of the combustion chamber, in an engine of only 3½″ cylinder diameter, where the distance from the nozzle to the opposite wall is a little over an inch, whereas in many other combustion chambers even at a distance of six or even eight inches the jet shows signs of impinging on the opposite wall.

It is, however, quite impossible to give any definite and rigid rules, and the desirable degree of penetration can always be attained by experimental means.

The efficiency of the fuel injection equipment as well as of the turbulence in the combustion chamber can be easily ascertained from the thermal analysis of the indicator diagrams, taken at the normal working mean effective pressure.

As an example, two indicator diagrams are shown, the one, Fig. 11a, with poor combustion efficiency, and Fig. 11b with good combustion efficiency both for the same indicated mean effective pressure of 100 lb. per sq. inch.

The efficiency of combustion in Fig. 11b was improved by shortening the injection period, making the fuel delivery rate during injection constant and by eliminating the chattering of the needle valve in the injector.

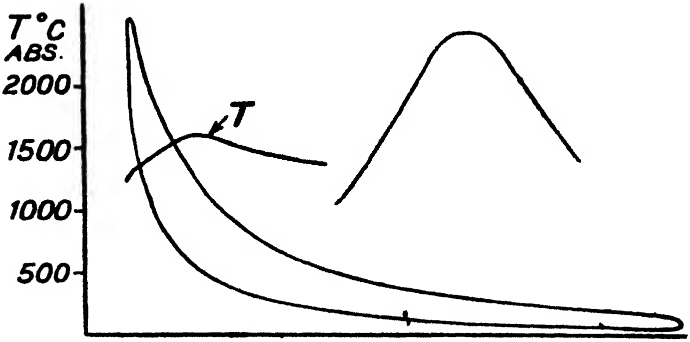


FIG. 11a

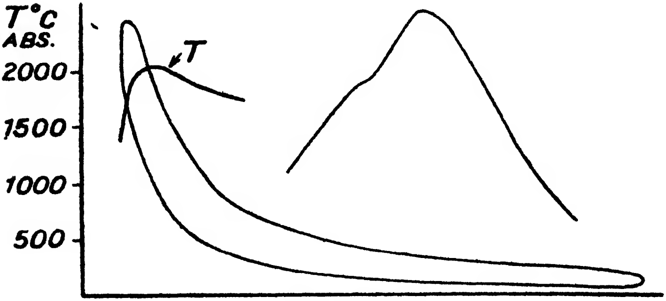


FIG. 11b

The improvement is, first of all, noticeable in thickening of the indicator diagram top, and again by the difference in the rate of combustion and the maximum combustion temperatures.

In Fig. 11a the maximum combustion temperature is about 1650° C. abs., and in Fig. 11b, the maximum temperature is about 2050° C. abs. The temperature curve in Fig. 11a shows a considerable amount of afterburning, noticeable from the slow drop in temperature after combustion, but the temperature drop in Fig. 11b is very much quicker.

The improvement in fuel consumption was about 20 per cent.

Both of these diagrams are for an engine with combustion space similar to the one shown in Fig. 8. The best fuel consumption was below ·30 lb. per I.H.P./hour.

The temperature curves can be very easily worked out from the pressure diagrams, if the compression ratio is known, and also the point of the beginning of compression.

Then, knowing the air volume and pressure in the beginning of compression, and assuming its temperature to be about 60° or 70° C., we can easily calculate from the pressure-volume curves on the indicator diagram the temperature for each point of the whole process, by using the equation :

$$(12)\quad pv = RT,$$

where p and v are pressure and volume of air, T – its absolute temperature, and R – the gas constant.

This constant must be first found for the scale of the diagram from the initial point of the beginning of compression, where p, v, and T are known, and then T can be found for all other points knowing p, v, and R. All volumes must be, of course, taken including the volume of compression chamber, and pressures—from absolute zero.

The thermal analysis is extremely useful in determining the influences of various changes in fuel injection and turbulence.

It is quite impossible to give any general recommendations with regard to fuel injection for either pre-combustion chamber type or for combustion chamber with high velocity turbulence, for in both of these types the combustion is much more dependent on the air turbulence than on the injection efficiency.

Generally speaking, however, with both latter types of combustion chambers the exhaust is never so clean, and the fuel consumption never so low as can be obtained in open type combustion chambers with low velocity turbulence.

Both the pre-combustion chamber type and the high velocity turbulence type combustion chambers must therefore be considered as being only temporary dodges, still used at present because the complete technical requirements of existing injection systems are not yet perfectly understood, and the pre-combustion chamber or high velocity turbulence offers an easy solution to the difficulties.

There is no doubt, however, that they will all disappear in course of time, and, indeed, they are already fast disappearing now.

They have, however, served a very useful purpose in the initial stages of the development of small high speed oil engines, particularly for road transport, for in this stage the problems of efficient fuel injection have been little known to most oil engine manufacturers.

CHAPTER VI

INVESTIGATION OF THE COMBUSTION CHAMBER EFFICIENCY

MOST of the oil engine manufacturers even now make the great mistake of judging the efficiency of the combustion chambers of their engines by the results obtained from brake tests.

It is easy to prove, however, that the brake test results have little connection with the combustion chamber efficiencies in different engines, and can only be used for comparison of combustion efficiency in the same engine and under the same conditions of running.

This is because the mechanical losses in various engines differ very considerably, and these mechanical losses have a far greater influence on the engine output on the brake test than any possible variations in combustion efficiency, provided the fuel injection equipment and combustion chamber design are not hopelessly wrong.

The mechanical losses in every engine depend on a great number of factors, amongst which the following are the most important ones :

(1) Stroke to bore ratio.
(2) Length of all bearings.
(3) Ratio of the conrod length to the crank.
(4) Tightness of bearings.
(5) Length of pistons.
(6) Weight of reciprocating parts, such as conrods, pistons, etc.
(7) Balancing of the cranks.
(8) Engine speed.
(9) Indicated mean pressure used.
(10) Size of valves in four cycle engines.
(11) Number of piston rings used.
(12) Type of drive used for camshaft, and many other minor factors.

It is, of course, an impossibility to give an accurate estimate of the magnitude of each influence, but in several diagrams following, curves are given showing the influence of the speed and indicated mean pressure on mechanical losses and mechanical efficiency.

These curves do not apply to any particular engine, but represent rather an average case computed from a number of tests of high speed oil engines of the automotive type.

As an example, a most popular size of engine is chosen, having $4\frac{1}{4}''$ bore × $6''$ stroke, running normally at speeds from 800 r.p.m. to

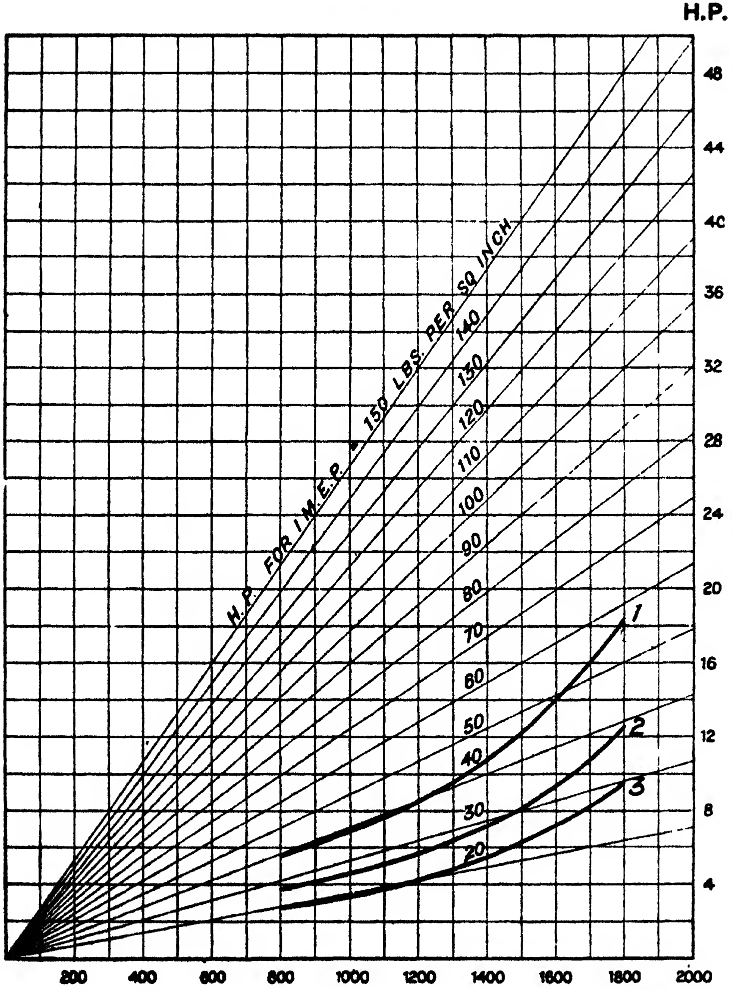

H.P.
48
44
40
36
32
28
24
20
16
12
8
4
H.P. FOR I.M.E.P. = 150 LBS. PER SQ. INCH
140
130
120
110
100
90
80
70
60
50
40
30
20
1
2
3
200
400
600
800
1000
1200
1400
1600
1800
2000
R.P.M.

Fig. 12

1800 r.p.m., and indicated mean effective pressures from 80 lb. per sq. inch to 160 lb. per sq. inch.

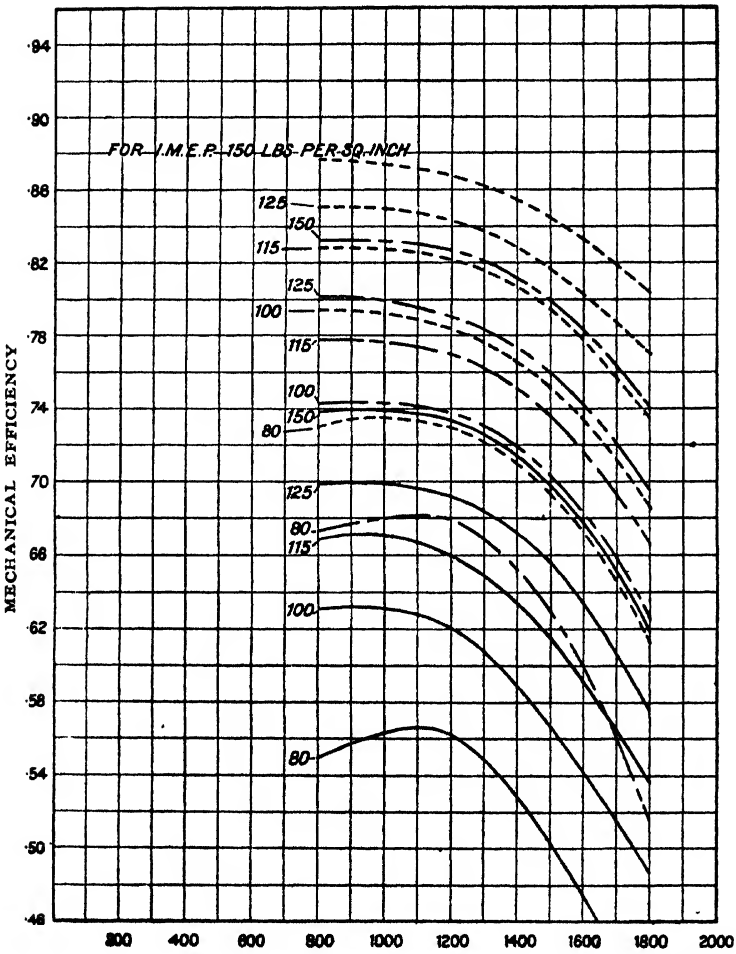
In Fig. 12, straight lines show the horse-power of the engine at different speeds, each line corresponding to a certain mean effective pressure.

Obviously, if the power is considered as brake horse-power, the mean pressure must be considered as brake mean effective pressure, or, if the power is considered as indicated, then the lines will also correspond to indicated mean effective pressures.

The three curves in the lower part of the diagram show mechanical losses for three engines with three different mechanical efficiencies, which three engines we shall further consider as a *standard* for comparison.

Curve 1, showing high mechanical losses, is for an engine with low mechanical efficiency, *curve* 2 probably represents a good average engine, and *curve* 3 is for an engine with exceptionally good mechanical efficiency.

It is a well-known fact that in the average case the mechanical losses remain approximately constant, and are independent of the mean effective pressures. This will enable us to construct curves for mechanical efficiencies at different mean pressures for all three engines.



MECHANICAL EFFICIENCY
·94
·90
·86
·82
·78
74
70
66
·62
·58
·54
·50
·46
FOR I.M.E.P. 150 LBS PER SQ.INCH
125
150
115
125
100
115
100
150
80
125
80
115
100
80
200
400
600
800
1000
1200
1400
1600
1800
2000

FIG. 13

R.P.M.

It is important to note that mechanical losses first increase approximately in proportion to speed, and then gradually get heavier as the speed increases.

This is mainly due to greater pressure losses in the cylinder and also due to inertia of reciprocating parts, which increase frictional losses in the bearings.

Curves in dotted lines are for engine with small mechanical losses, curves in chain-dotted lines are for medium losses, and solid line curves are for engine with heavy mechanical losses.

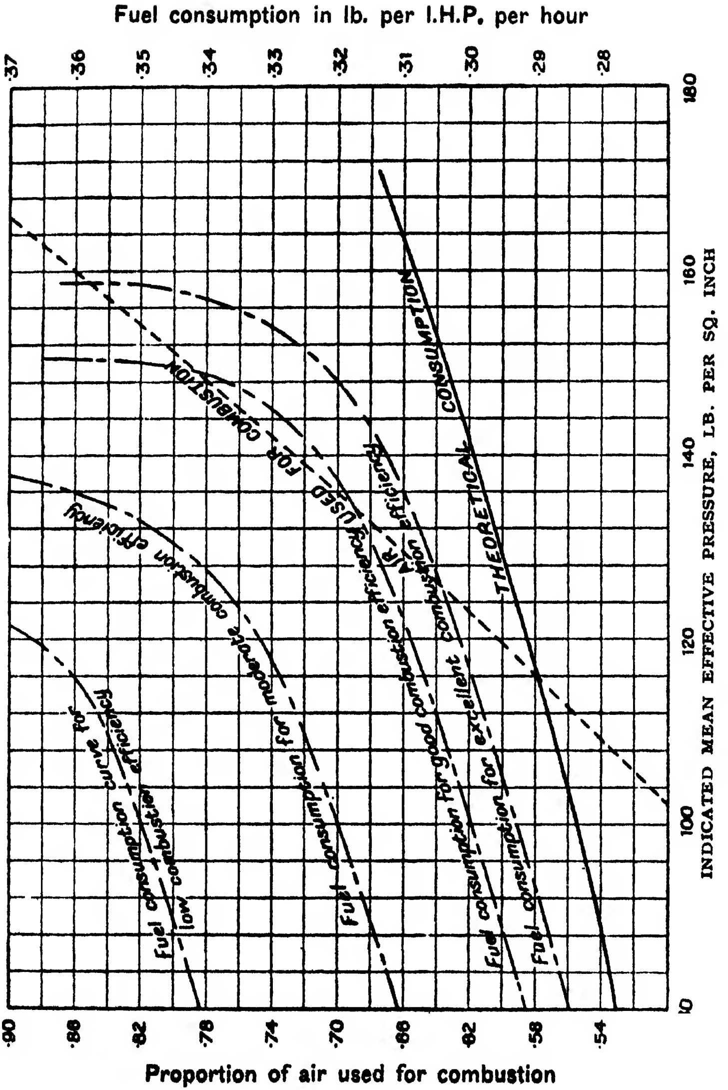
Figures for each curve denote the indicated mean effective pressure for which each curve is computed.

In Fig. 13 curves are shown for mechanical efficiencies of all three engines at different speeds, each curve corresponding to a different indicated mean effective pressure.

It is obvious, of course, that although mechanical losses are independent of the mean effective pressure, the engine mechanical efficiency depends on it more than on anything else, for the mechanical efficiency is :—

$$(13)\quad \text{Mechanical Efficiency} = \frac{\text{Indicated H.P.} - \text{mechanical losses in H.P.}}{\text{Indicated H.P.}}$$

FIG. 14


Fuel consumption in lb. per I.H.P. per hour
·37
·36
·35
·34
·33
·32
·31
·30
·29
·28
180
160
140
120
100
0
INDICATED MEAN EFFECTIVE PRESSURE, LB. PER SQ. INCH
·90
·86
·82
·78
·74
·70
·66
·62
·58
·54
Proportion of air used for combustion
Fuel consumption curve for low combustion efficiency
Fuel consumption for moderate combustion efficiency
Fuel consumption for good combustion efficiency
Fuel consumption for excellent combustion efficiency
THEORETICAL CONSUMPTION
AIR USED FOR COMBUSTION

These curves for mechanical efficiencies of three standard engines will be useful to us when determining their fuel consumptions.

In Fig. 14 curves are given for fuel consumptions in engines with different combustion efficiencies, and also a curve for theoretical fuel consumption, at different indicated mean pressures, for compression ratio 14 : 1.

A curve is also given for the theoretical percentage of air used in combustion for different indicated mean effective pressures.

These curves, of course, have been computed for the average type of fuel oil.

Fig. 15 shows how the indicated mean effective pressures and mechanical efficiencies vary in all three standard engines at the same constant brake mean effective pressure of 100 lb. per sq. inch at different speeds.

From these curves and the four actual consumption curves in Fig. 14 it is possible to construct fuel consumption curves for brake horse-power for all three engines, Fig. 16, corresponding to four indicated horse-power fuel consumption curves in Fig. 14, all curves for the same brake mean effective pressure of 100 lb. per square inch.

Curves in dotted lines are for the engine with small mechanical losses, chain-dotted curves are for the medium mechanical losses,

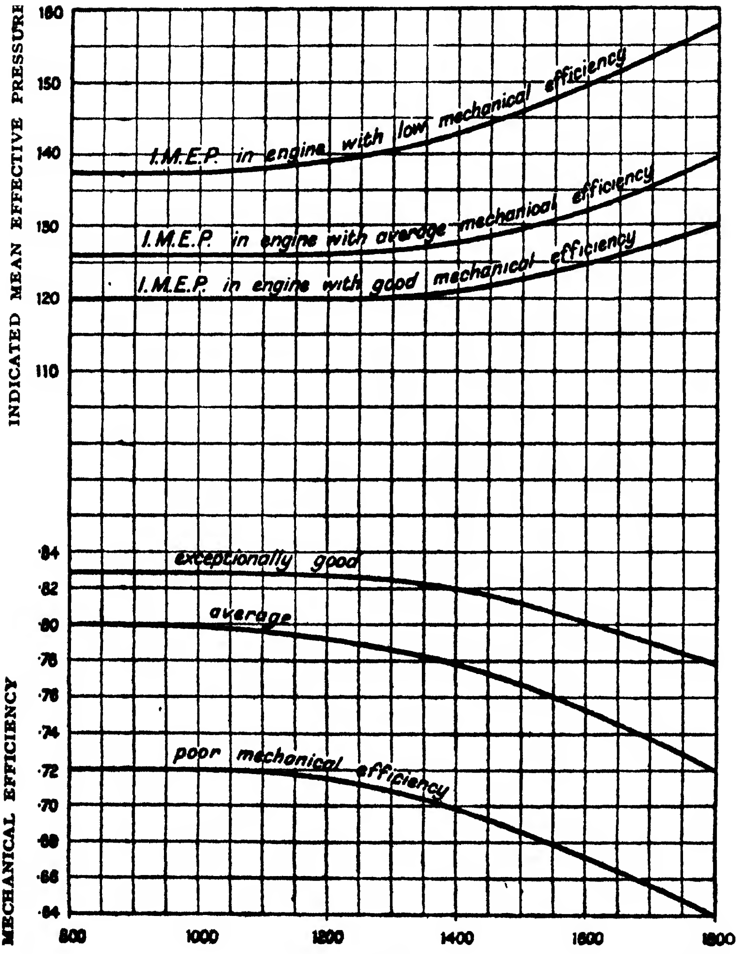

INDICATED MEAN EFFECTIVE PRESSURE
160
150
140
130
120
110
I.M.E.P. in engine with low mechanical efficiency
I.M.E.P. in engine with average mechanical efficiency
I.M.E.P. in engine with good mechanical efficiency
MECHANICAL EFFICIENCY
·84
·82
·80
·78
·76
·74
·72
·70
·68
·66
·64
exceptionally good
average
poor mechanical efficiency
800
1000
1200
1400
1600
1800
R.P.M.

FIG. 15

and curves in solid lines are for the engine with heavy mechanical losses.

The analysis of these results is most instructive. *It proves beyond doubt that the fuel consumption figures for brake horse-power can never be taken as an indication of the engine combustion efficiency, for the difference in mechanical efficiencies can upset the results in any direction.*

For example, it is hard to believe that curves 1, 4 and 8 are all for engines with the same fuel combustion efficiency, or that fuel combustion efficiency for curve 7 is much worse than in curve 1.

The explanation of it is very simple. For the same B.M.E.P. and low mechanical efficiency, a higher indicated mean effective pressure is necessary than is required for high mechanical efficiency, which, as can be found from Fig. 14, will again correspond to a considerably higher fuel consumption per indicated horse-power. Thus all the losses are being increased, resulting in accumulation of losses in the same direction.

On the other hand, an engine with high mechanical efficiency can develop the required horse-power at a moderate indicated mean effective pressure, corresponding to lower consumption figure. Thus the gains are again accumulating in all directions.

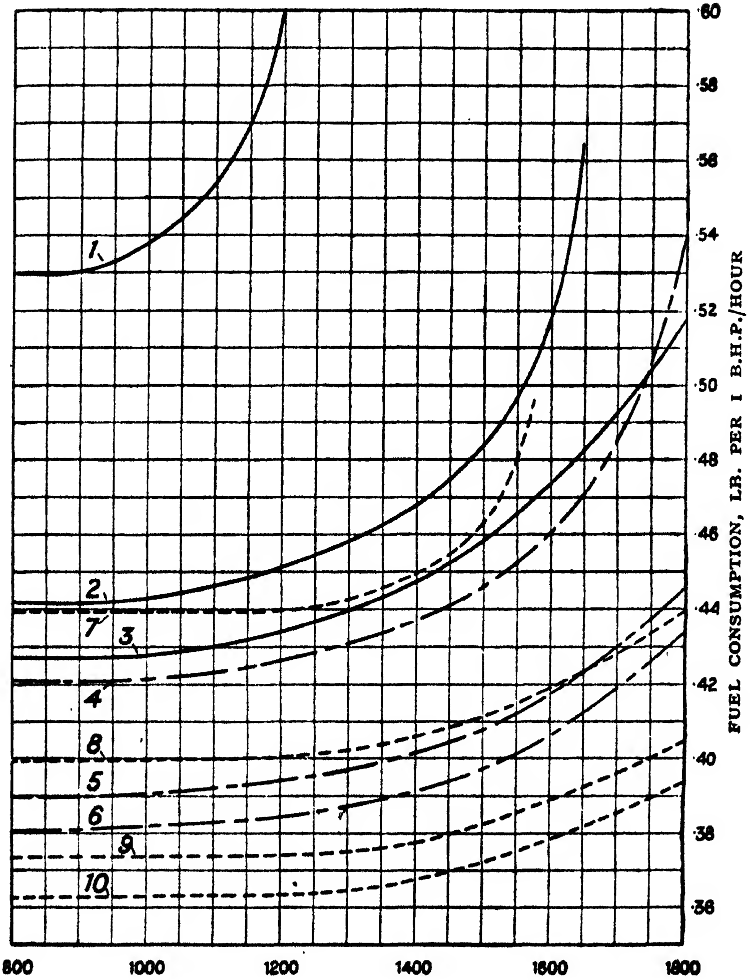

1
2
7
3
4
8
5
6
9
10
·60
·58
·56
·54
·52
·50
·48
·46
·44
·42
·40
·38
·36
FUEL CONSUMPTION, LB. PER I B.H.P./HOUR
800
1000
1200
1400
1600
1800
R.P.M.

FIG. 16

As can be seen from Fig. 13, mechanical efficiencies for different indicated mean effective pressures can differ even for the same speed as much as 60 per cent., whereas the difference between a good and a bad fuel combustion efficiency is no more than 15—16 per cent. in terms of fuel consumption, although it may differ a lot more in terms of clean exhaust and clean engines.

Consequently the mechanical efficiency is by a long way the more important factor in the overall engine efficiency, and it is not advisable to ignore it.

The engine builders usually state that as the buyers of engines are only interested in the engine B.H.P. output and the corresponding fuel consumption, they are not in the least interested in the engine's indicated fuel consumption.

This is a most unwise attitude both on the part of the engine buyers and builders, for the life of the engine depends more on the quality of its combustion than on its mechanical efficiency.

Low mechanical efficiency often is due to long and heavy pistons, too many piston rings, long bearings, etc., all of which can only contribute to long life of the engine, whereas bad fuel combustion efficiency often results in overheated pistons, carbonized piston rings, bad wear on cylinder liners, carbonized cylinders, contaminated lubricating oil, heavy wear on bearings, etc. Yet, two engines with very different characteristics like this, may easily have exactly the same fuel consumptions, even more, the engine with lower mechanical efficiency may have a worse fuel consumption per B.H.P. than an engine with a high mechanical efficiency and bad combustion efficiency, as can be illustrated by comparison between curves 3 and 8, or 2 and 7.

For these important reasons it is essential both for the engine buyer and the manufacturer to know the engine mechanical efficiency and its fuel combustion efficiency, and judge the efficiency of fuel injection and of the combustion chamber by the fuel consumption per indicated horse-power, which figures are free from influence of the mechanical efficiency. In other words, the fuel consumption figures per indicated horse-power give us a direct information on the engine's injection and combustion efficiency, which is the most important factor contributing to long life of the engine.

Fig. 17 shows how the mechanical efficiency of the three standard engines depends on the indicated mean pressure, at the same speed of 1000 r.p.m.

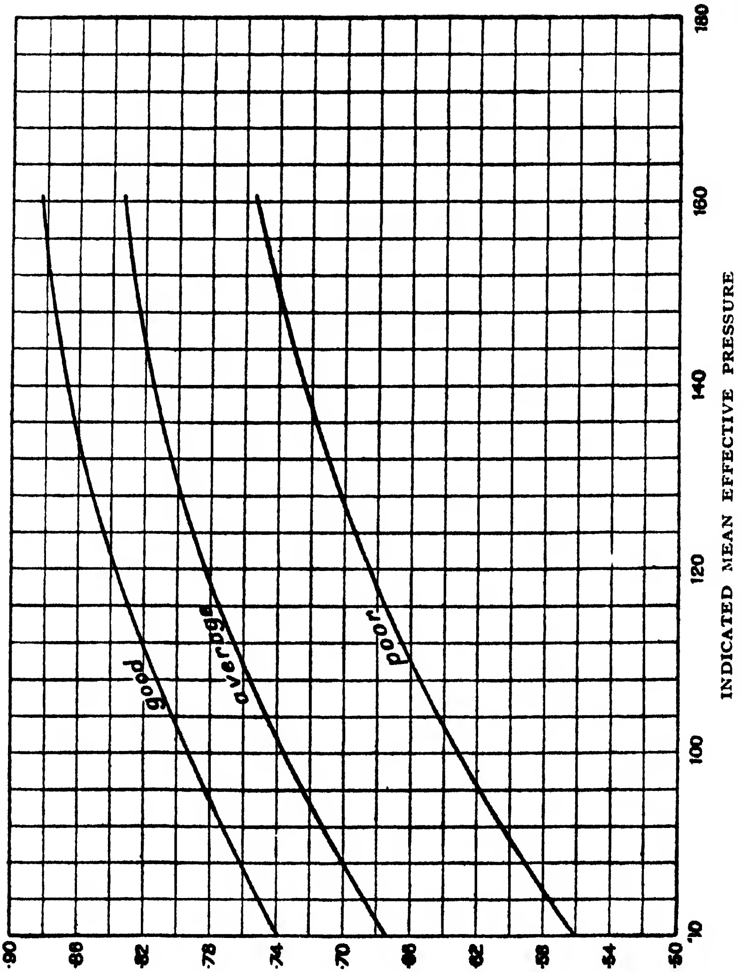


FIG. 17

Mechanical Efficiency

There are several satisfactory methods known for working out the mechanical losses of the engine, the best of all, in the writer's opinion, being by driving the engine with an electric motor of a known efficiency. Some electric dynamometers can also be used for this purpose.

It is always useful, however, to check the figures obtained from direct drive by indicator diagrams, if a reliable indicator is available.

CHAPTER VII

FUEL OIL AND ITS INFLUENCE ON SMOOTHNESS OF COMBUSTION, COMPRESSION RATIO, DIESEL KNOCK AND ITS CAUSES

ALTHOUGH detailed investigation of the desirable fuel characteristics is entirely out of scope of this work, it is necessary to mention some of the qualities in the fuel which are beneficial to long life and smooth engine running.

It is a well-known fact that the ignition of fuel does not take place immediately the fuel is injected, but a little time usually elapses before the injected fuel ignites. This brief period is usually called the "ignition lag." The length of this period depends on the complex composition of the fuel.

If this ignition lag had no other effect except to retard the ignition, all would be well, for then the effect of the retarded ignition could be cured by advancing the injection.

Unfortunately, however, fuels with a big ignition lag have a tendency to produce detonations.

The " detonation " is usually called a very quick rise in pressure, producing a heavy knock in the cylinder, akin to a hammer blow.

There are, of course, many degrees to the force of detonation.

It is considered that ignition lag is mainly responsible for detonation of fuel, which can be easily understood from the following argument.

When fuel is injected in atomized condition into highly compressed hot air, the tiny globules of fuel are first being heated by surrounding air, then the fuel they are composed of begins to vaporize, and eventually ignition of the fuel vapours and compressed air takes place at some point, accompanied by a quick rise in pressure and temperature.

If the ignition lag is short, then only a small portion of fuel will be injected before the ignition occurs, and further increase in pressure and temperature will take place only gradually, as fresh fuel is being injected. Consequently the rise in combustion pressure will be comparatively slow, and will not produce any effect of a sharp blow.

On the contrary, if the ignition lag is considerable, much fuel will be already injected before ignition takes place, and then all fuel vapours present in the cylinder will detonate, producing a quick rise in pressure.

The detonation of fuel has a detrimental effect on some working parts in the engine, such as bearings, pistons and rings, etc.

There is, however, another factor of at least equal importance for the elimination of diesel knock, which is the pressure and temperature of compressed air in the cylinder.

It is obvious, that by increasing both the pressure and temperature of compression the ignition of fuel can be considerably accelerated and consequently the ignition lag reduced for any fuel. This factor gives the designer a powerful weapon for fighting against the effect produced by a fuel with long ignition lag.

Generally speaking, by increasing the compression ratio in an engine with mechanical injection the designer will benefit in all directions. First of all the thermal efficiency of the engine will be increased, then starting will become much easier, and combustion much smoother. Notwithstanding the increased compression pressure, the maximum pressure in the cylinder will more likely be reduced, and the engine will become generally quieter in running.

For these reasons a high compression ratio can be generally recommended for all engines with mechanical injection of fuel, and particularly for small high speed engines.

For larger engines the compression ratio should be from about 14 : 1 to 14·5 : 1 and for small high speed engines about 15 : 1 or 16 : 1 depending on its size. Generally speaking, the smaller the engine, the higher should be its compression ratio for easy starting and quiet running and for cylinder diameters round about 3¼″ the compression ratio should be 18 : 1 or even 20 : 1.

How beneficial is the effect of the increased compression on the smoothness of combustion can be seen from the comparison of two indicator cards, Fig. 18, for compression ratio 14·5 : 1 and Fig. 19 for compression ratio 18 : 1, both for a small high speed engine of 3½″ bore × 5½″ stroke at about 1800 r.p.m., using the same fuel.

Both indicator cards are for about the same H.P., and it is interesting to note that the maximum pressure in Fig. 18 is about 1000 lb. per sq. inch with compression pressure about 570 lb. per sq. inch, and the maximum pressure in Fig. 19 is only 850 lb. per sq. inch with compression pressure about 650 lb. per sq. inch.

Increasing the compression ratio is beneficial not only for small engines, but also for large ones, and the effect it produces is always the same—the increased combustion efficiency, smoother running, lower maximum pressure, lower fuel consumption, easier starting, and often even increased power output.

The reason why a smaller engine requires a higher compression ratio is very simple—for a smaller engine has a much greater cylinder surface-to-volume ratio and consequently greater heat losses during compression, which must be compensated for by increasing compression ratio.

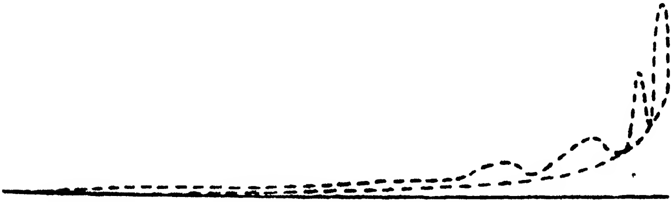


FIG. 18

FIG. 19

A very important point in connection with any fuel is apparently the sulphur content in it. It is not yet definitely known what effect sulphurous gas has on the life of the engine, but from some data available it is possible to suspect some connection between sulphur content in fuel and sludging of oil in the engine crankcase in case the combustion of fuel is not very clean.

It may be possible that soot suspended in the products of combustion in the cylinder acts as a catalyst, and converts the SO_2 from the combustion of sulphur into SO_3, which, combining with H_2O, forms sulphuric acid. The latter together with moisture condenses on all watercooled parts in the cylinder, assists corrosion of piston rings and cylinder liners, and eventually finds its way into the crankcase, contaminating the lubricating oil. This is, at the moment, however, only a conjecture, and some more experimental work will have to be carried out before more definite information bearing on this matter can be obtained.

It is a well-known fact that dirty exhaust is responsible for sludging of lubricating oil, and also for greater wear on cylinder liners, piston rings and bearings.

CHAPTER VIII

SURVEY OF EXISTING COMBUSTION CHAMBERS

IT is, of course, quite impossible to describe in this chapter all the existing combustion chambers used in various engines.

A small number of typical combustion chambers has been chosen, representing the various lines of the development of combustion chamber design. It is the intention of the writer to illustrate not the cylinder covers of any particular engines, but the general principles of the various designs.

Fig. 20 shows the modernized modification of the old Brons design. The main difference is that the pre-combustion chamber is arranged in the centre of the cylinder, and fuel is injected into it by an ordinary modern injector. A number of pre-combustion chambers like this are used now, all differing slightly in internal shape of pre-combustion chamber, otherwise very similar.

The general characteristic of this type is that the combustion chamber is divided into two parts, one into which the fuel is injected and where it explodes and the other where the combustion is completed. The volume

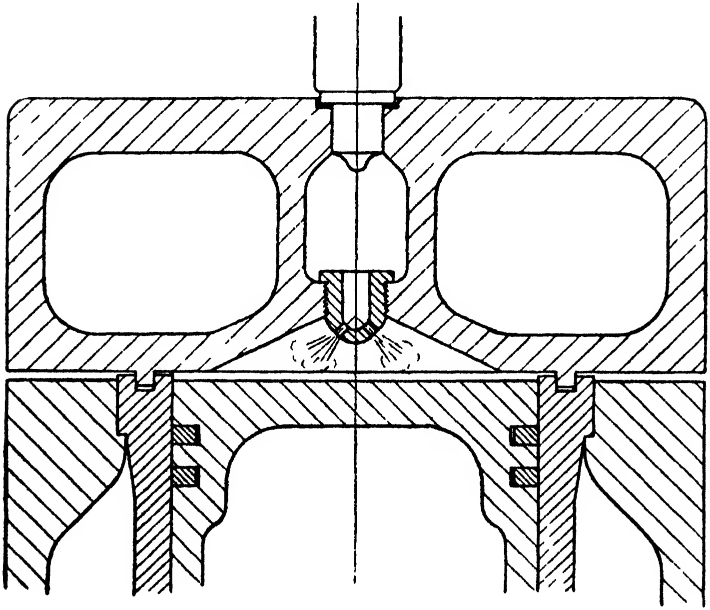


FIG. 20

of the pre-combustion chamber varies approximately from 1/5 to 1/6 of the total volume of the compression space.

Fig. 21 shows the so-called Acro-Bosch type pre-combustion chamber, which played a very considerable part in the beginning of the high speed diesel development. It is obviously a more modern development of the old Mietz and Weiss hot bulb type, shown

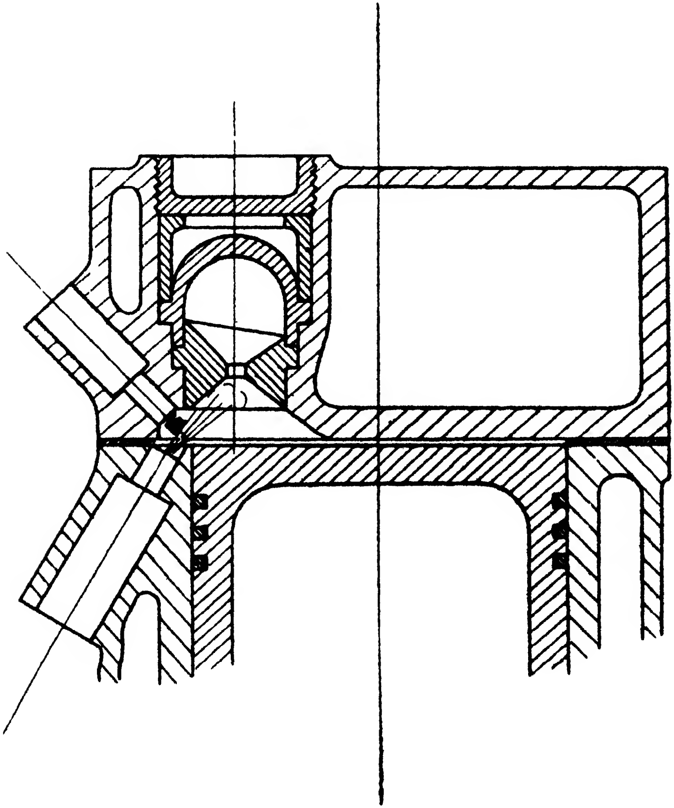


FIG. 21

in Fig. 2, with streamlined throat, improved injection and increased compression.

Fig. 22 shows the air-cell type, in which the fuel is injected into the main combustion chamber, and the baffle-plate with a hole in the middle in the piston top softens the shock of the combustion.

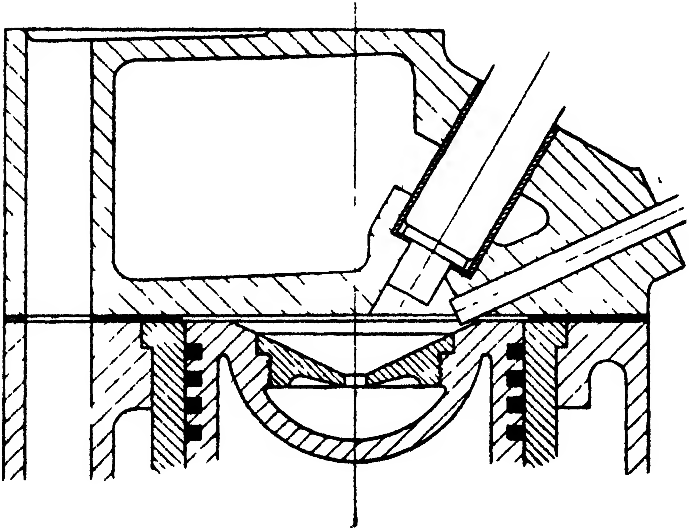


FIG. 22

Fig. 23 shows the well-known and very popular "Comet" type Ricardo combustion head. It is the modernized direct descendant of the type illustrated in Fig. 5, which goes back to 1909. A number of other similar type heads are also used under different names, all slightly differing in shape, like, for example, the one illustrated in Fig. 24.

The combustion chamber illustrated in Fig. 25 has the injector placed in a different

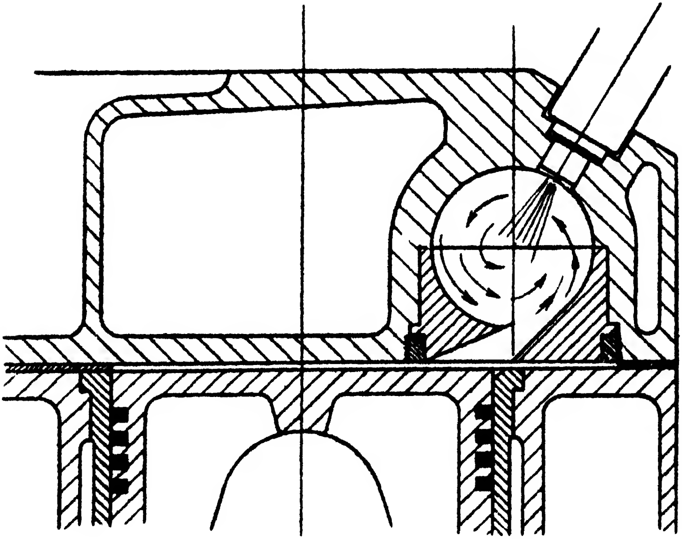


FIG. 23

position, with one jet injecting directly in the main combustion chamber. This makes it possible to start the engine up from cold without any heating plugs.

Fig. 26 illustrates the combustion chamber designed by Hesselman and is the first open type combustion chamber with a rotational turbulence produced by masked valves.

This design has been extensively imitated, and the rotational turbulence is used now in

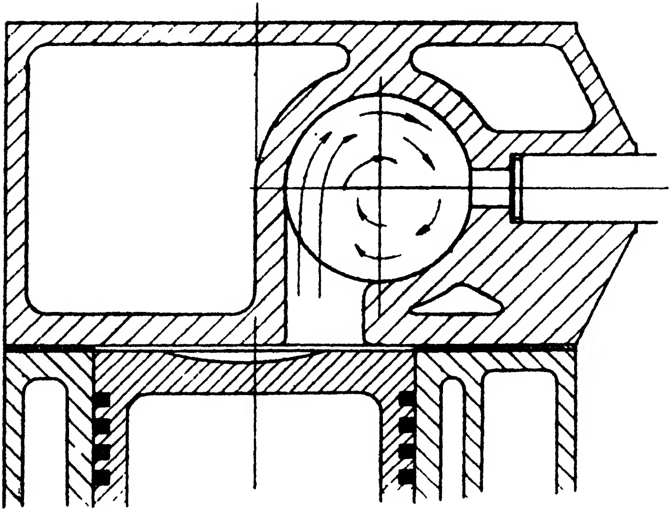


FIG. 24

a number of other open type combustion chambers.

Fig. 27 shows rather an original shape of combustion space with the single jet injection placed near its periphery. Rotational turbulence, of course, is necessary for this position of injector.

A somewhat similar type, but with the central injector is shown in Fig. 28.

Further development of the open type combustion chamber is shown in Fig. 29, as used in Armstrong-Saurer high speed engines. The rotational and radial turbulence in it combined to form a dual type turbulence in the combustion chamber.

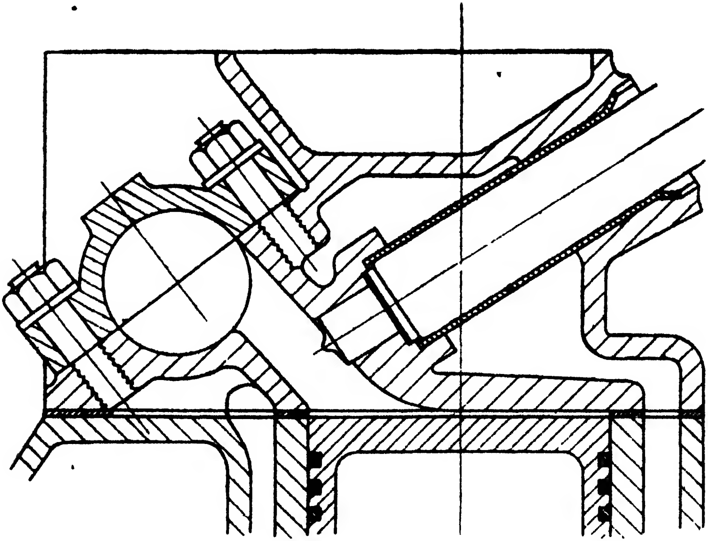


FIG. 25

In Fig. 30 is shown the adaptation of the combustion chamber as illustrated in Fig. 10

to four-cycle engines. It has a simple type low velocity turbulence.

Fig. 31 shows a partly conical, partly spherical type combustion chamber. The turbulence in it is not of a very suitable type.

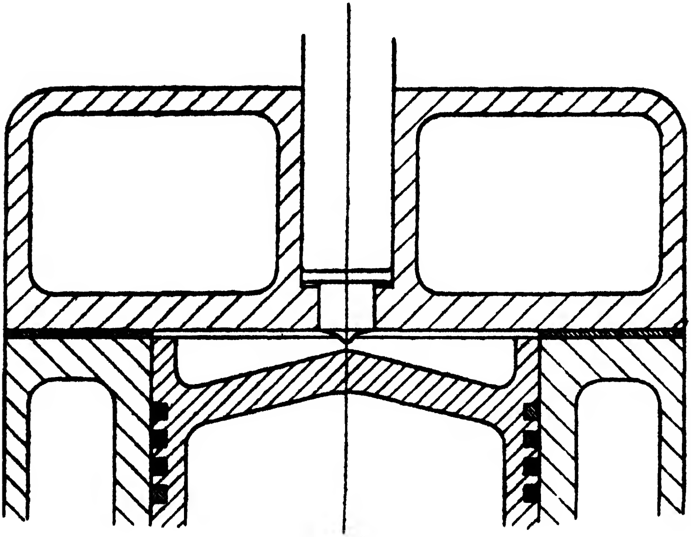


FIG. 26

Fig. 32 shows the latest development of the Hesselman's low pressure oil engine, with rotational turbulence and sparking plug ignition.

It is quite useless to give any data for fuel consumptions from published results for all these combustion chambers, for, as was fully discussed in the preceding chapter, the fuel consumption figures for brake horsepower do not give any indication as to the combustion chamber efficiency.

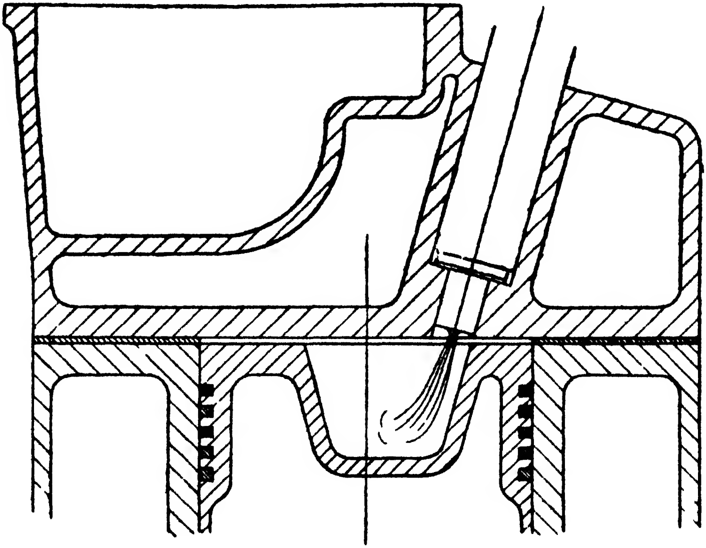


FIG. 27

It is only possible, therefore, to give some general figures and information relating to the general types.

All pre-combustion type chambers have the advantage that they do not require efficient fuel injection.

But they require heating plugs for starting instead, and the combustion is not perfect. Consequently the exhaust is not very clean, and the amount of soot deposited in the cylinders is considerably greater than it is with open type combustion chambers.

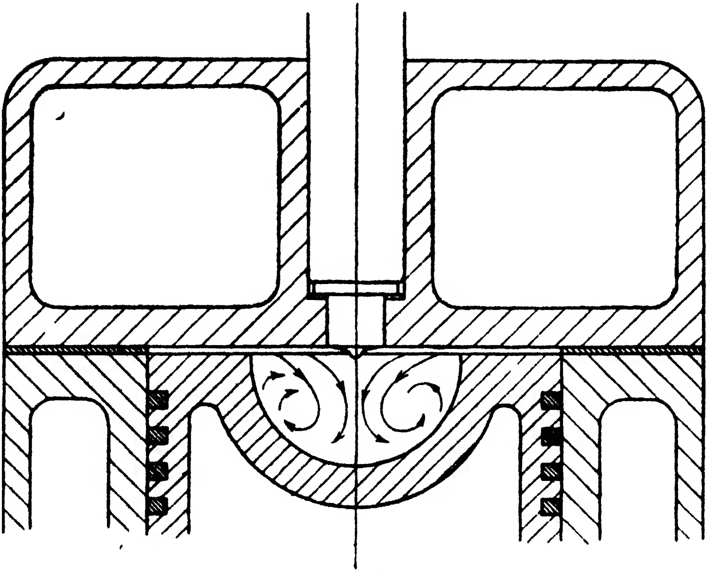


FIG. 28

The fuel consumption in pre-combustion

chambers probably varies on the average between ·35 to ·37 lb. per I.H.P./hour.

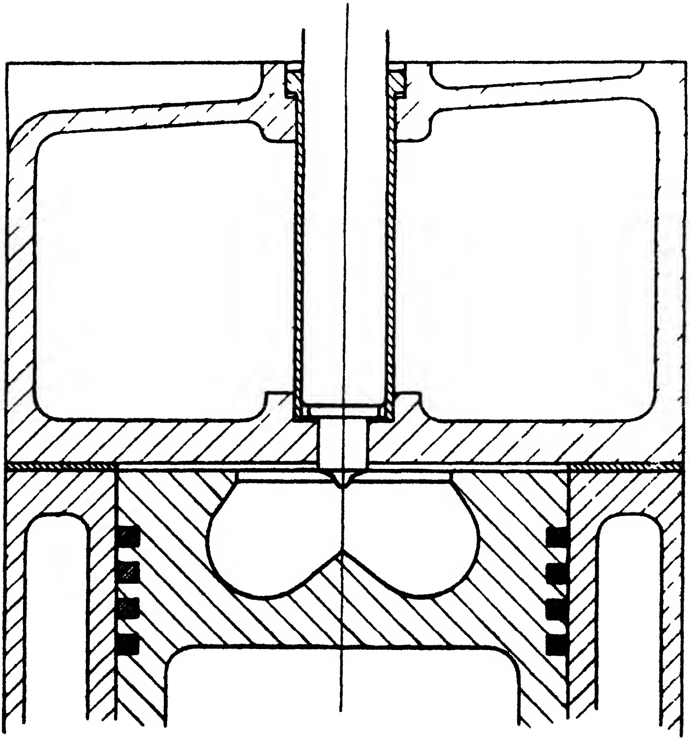


FIG. 29

The high velocity turbulence chambers as shown in Figs. 22, 24 and 25 stand about half-way between the pre-combustion chamber and the open combustion chamber types.

Although they also require heating plugs for starting from cold, the same as the pre-combustion chamber type, the combustion process in them is of a much higher efficiency and of a better quality. A reasonably clear exhaust can be obtained, and very high mean effective pressure.

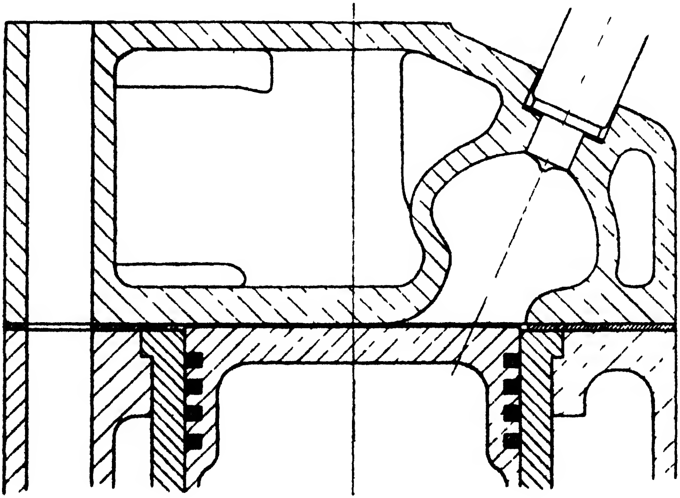


FIG. 30

The fuel consumption per indicated horse-power is probably of the order of ·32 lb. per I.H.P./hour, on the average.

The open type combustion chamber, like

the ones illustrated in Figs. 26 and 28, require very efficient fuel injection equipment and design to get the best out of them ; but this best is worth striving for.

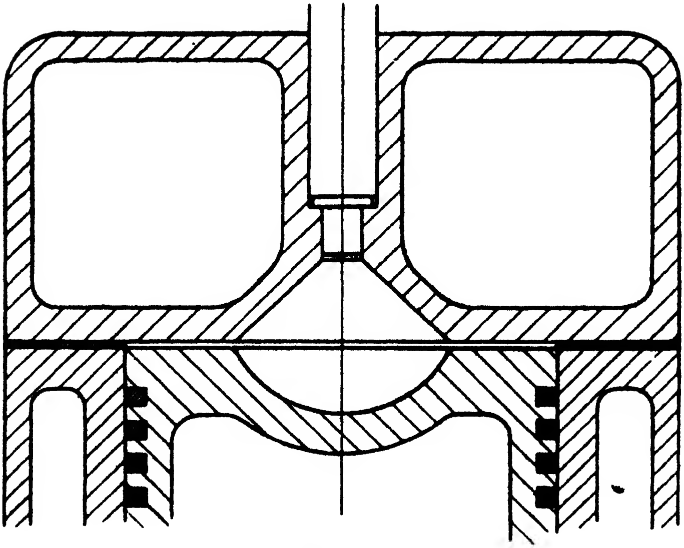


FIG. 31

Although they are not capable of very high indicated mean effective pressures, 110–120 lb. per sq. inch being probably the practical limit, a perfect combustion with invisible exhaust can be obtained in them particularly with rotational turbulence of moderate velocity with fuel consumption per indicated horse-power per hour as low as ·30 lb. per I.H.P./hour.

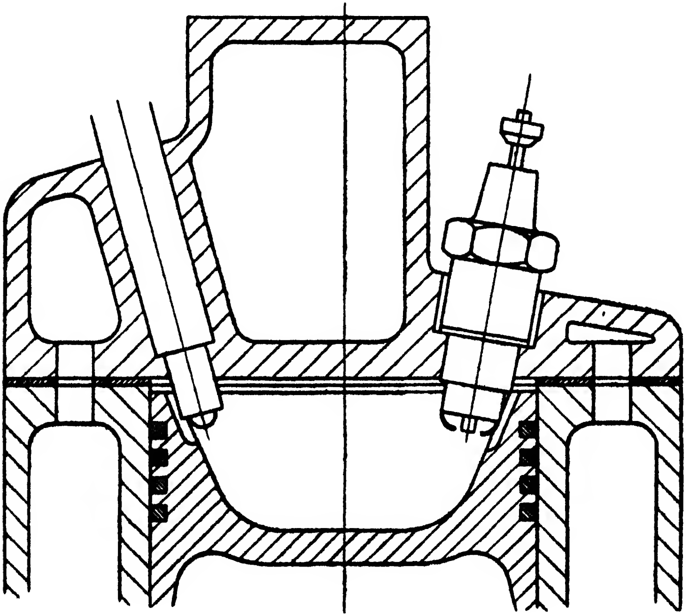


FIG. 32

Engines with open type combustion chambers, if well designed, remain very clean inside and have long life.

The other type of open type combustion chamber, illustrated in Figs. 8a and 8b, have all the advantages of the ordinary open type combustion chambers, and, moreover, they do not require any organized turbulence, and the fuel injection for them is easier to choose, for they have a very favourable natural turbulence. The combustion efficiency in them can also be very high, and the fuel consumption very low.

Combustion chamber of the open type, as illustrated in Figs. 10 and 30, are somewhat similar to the last discussed type, but are more rationalized, for their internal shape is specially designed to be particularly suitable for the normal natural turbulence in them and the shape of the spray.

The shape of the throat is also specially adapted to form an air stream of the most suitable shape, although this throat is very wide.

Very high mean effective pressures, invisible exhaust and low fuel consumption figures per I.H.P. have been obtained with this combustion space. The fuel injection arrangement is very easy with this combustion chamber, two or three jets usually being satisfactory. Some of the many advantages of this chamber are that it is entirely free from diesel knock and not so sensitive to change in injection advance as many of the other combustion space.

The starting of engines from cold is very easy with all open type combustion spaces.

The combustion chamber as shown in Fig. 3 is an example not to be imitated. It is subjected to detonations, as the air in the narrow conical portion around the jets is well cooled, and this lengthens the ignition lag. Besides, the natural turbulence in it is not of a suitable kind to promote the good distribution.

CONCLUSIONS

THE most desirable features to have in a rational combustion chamber :—

(1) The compressed air in it must be in one volume concentrated as much as possible.

(2) The natural turbulence in it should be of sufficient strength and correct direction, preferably *not* of a rotating kind.

(3) Its shape should suit the fuel jets.

(4) The air in it must not be cooled during compression by passing through narrow passages.

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