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Performance to be expected from a fuel-injection, spark-ignition, internal combustion engine

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PERFORMANCE TO BE EXPECTED FROM A FUEL-INJECTION, SPARK-IGNITION, INTERNAL COMBUSTION ENGINE

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Submitted in Partial Fulfillment of the

Requirements for the Degree of

Master of Science in Aeronautical Engineering

from

Massachusetts Institute of Technology

1933

Thesis
P6

Professor A. L. Merrill,

Secretary of the Faculty,

Massachusetts Institute of Technology,

Cambridge, Mass.

Dear Sir:

We herewith submit a thesis entitled "Performance to be Expected from a Fuel-Injection, Spark-Ignition, Internal Combustion Engine," in partial fulfillment of the requirements for the degree of Master of Science in Aeronautical Engineering from the Massachusetts Institute of Technology.

Respectfully yours,

ACKNOWLEDGMENT

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TABLE OF CONTENTS

I. Object and Scope of Investigation

II. Description of Apparatus

III. Preliminary Considerations

a. General

b. Review of pertinent work by others

IV. Method of Attack

V. Results of Preliminary Investigations

a. Valve Timing

b. Length of Intake Pipe

c. Fuel Injection System

VI. Results of Final Tests

VII. Conclusions

VIII. Illustrations and Data

IX. References.

I.

Object and Scope of the Investigation.

This investigation was undertaken to determine and formally record the performance of an internal-combustion engine with a fuel-injection system substituted for the usual carburetor. The title was chosen so as to cover any and all investigations which might conceivably be made as to the effect of any variables on the performance of the engine. The idea in mind was to start with the factors which would seem to have the greatest effects on performance and after these had been disposed of to continue with an investigation of factors whose effects on performance are of a secondary nature.

However, as the tests proceeded it became increasingly evident that the limited time at our disposal would not permit of a careful investigation of any of the secondary factors referred to above. Therefore, it was considered wise to investigate only those factors which exert a predominating influence on the performance of the engine.

The object then of our investigation is the determination of the best performance of a fuel-injection, spark-ignition engine as indicated by the usual curves of Brake Mean Effective Pressure and Brake Specific Fuel Consumption vs. fuel flow.

The scope of the investigation included the effect of the following variables on performance:

Valve Timing

Length of Intake Pipe

Type of Fuel Pump Operating Cam

Spark Timing

Fuel Injection Timing

After these variables were fixed at optimum values the performance of the engine was recorded at various compression ratios.

In all tests two radically different fuels were used, namely a gasoline fuel and a dehydrogenated heavy fuel known to the trade as a "safety fuel."

II.

Description of Apparatus.

Description of the N.A.C.A. Universal Test Engine. (Figs. 1 to 13 inclusive.)

The engine employed in our investigation was a 5 inch bore by 7 inch stroke single cylinder, four stroke cycle test engine developed by the National Advisory Committee for Aeronautics for laboratory research on internal-combustion engine problems. It is manufactured by the Allison Engineering Company of Indianapolis, Indiana and is equipped with a Sprague Electric Dynamometer manufactured by the General Electric Company. The dynamometer is rated at 45 Horse Power; speed 1200-2500 r.p.m.; 122 amperes and 250 volts.

The engine is arranged for variation over wide ranges of the compression ratio and lift and timing of both inlet and exhaust valves while the engine is operating. In our investigation the N.A.C.A. cylinder head was employed in the preliminary investigations, after which it was replaced by a head specially designed at, and for use in, the Aeronautical Engine Laboratory at the Massachusetts Institute of Technology This head will be termed the M.I.T. head thruout this thesis.

The engine was specially designed and constructed so that all variables not under investigation could be effectively controlled, while the variables under investigation could be changed quickly without changing other variables. Arrangements can be made for the connection of various accessories,

thus providing a unit suitable for carrying on a large variety of research problems with greater facility and with greater reliability of results than would otherwise be possible. The engine is of rugged design to insure long service without detailed attention. Ball and roller bearings are used in place of plain bearings to a large extent tending to stabilize and minimize bearing friction losses. The compression ratio may be changed from 4:1 to between 13:1 and 15:1, according to the value lift. The opening and closing adjustments of the inlet and exhaust valves have ranges of 50° each so that the valve opening periods may be increased a total of 100°.

The timing corresponding to the minimum position may be altered by changing the setting of the valve operating gear train in the usual manner. The inlet valve lift may be changed from 7/32 to 7/16 inch and the exhaust valve lift from 7/32 to 1/2 inch.

The mechanism for changing the compression ratio is one of several interesting features and is described first.

The compression ratio is varied by moving the cylinder unit vertically, the stroke of the piston remaining fixed, thus changing the clearance volume. The manner in which this is accomplished can be seen by examination of Figures 1, 2, and 9.

There are four main parts to the cylinder construction: the guide, jacket, barrel, and head. The jacket, barrel, and head form an assembled cylinder unit that is movable with respect to the crankcase and guide for the purpose of vary-

ing the compression ratio. The guide is bolted rigidly to the crankcase and provides the means for guiding the movable cylinder unit. The barrel fits within the jacket, which in turn fits within the guide.

Threads formed on the outside and bottom of the cylinder jacket engage an internally threaded ring, which surrounds the jacket and is restrained from moving vertically by the guide. Rotation of the ring will, therefore, cause the cylinder and head to be raised or lowered according to the direction of rotation, the jacket being prevented from rotating by means of a key fitted permanently in the cylinder jacket, but free to slide in a keyway cut in the cylinder guide. The ring is rotated by a worm meshing with teeth formed on the outside of the ring, a handwheel mounted on the same shaft with the worm providing manual control. The cylinder jacket is clamped in the guide to relieve the threaded ring from taking the explosion load while power tests are being made. For this purpose the cylinder guide is split vertically and is provided with means for clamping the guide around the jacket by the hand-lever located on the side of the guide and immediately above the worm and threaded ring construction shown in Figures 1, 2, and 9.

A counter is geared to the worm shaft to give the position of the cylinder and head with respect to the crankcase. The compression ratio is readily ascertained from the counter reading by use of a calibration curve obtained from actual measurement of the clearance volume at a number of different

positions of the cylinder. If it is desired to make a test at a definite compression ratio, the cylinder is moved to obtain the counter reading corresponding to this compression ratio.

The timing of the opening and closing of the valves is varied by the use of three-part cams (See figures 7,8,10,11, 12,13). The central part of each cam is fixed directly to the camshaft by means of splines while the two outside parts can be rotated independently with respect to the central part and to each other.

All three parts of the cam have the same contour so that when they are aligned they act as a single broad face cam. Displacements of the outside parts from the aligned position are made in opposite directions so that rotation of one of the outside parts varies the opening point while rotation of the other in the opposite direction varies the closing point. When both outside parts are displaced from the aligned position, the dwells of the three parts form one continuous dwell. The dwell of the movable cam parts must be at least equal to the angular variation of the parts from the aligned position. In this engine the dwell of the inlet cam is 50 crankshaft degrees and the dwell of the exhaust cam is 80 crankshaft degrees with both inlet and exhaust cams aligned. Figure 10 shows a camshaft assembly with the three cam parts nearly aligned. Figure 11 shows the three parts at their maximum displacement position. Figure 12 is a sectional view of the operating mechanism. Each outside part of the cams is formed on the end of a sleeve, A, splined internally with

helical splines. A bronze bushing, B, surrounding each sleeve, A, and pinned to the camshaft housing, C, retains the parts of the cams in their proper axial location on the camshaft and serves as a bearing for the camshaft unit. Another sleeve, D, is located between the camshaft and the sleeve that carries the outside cam part. This sleeve slides on straight axial splines on the camshaft, E, while at the same time external helical splines cut on the outside of, D, engage internal helical splines in A.

As a result of this spline construction axial movement of sleeve, D, will cause relative rotation between the camshaft E, and the sleeve, A. Since the center part of the cam, F, is splined directly to the camshaft, the outside part of the cam, A, will be rotated relative to the center part, F, and the valve opening or closing point will be changed according to which outside part is rotated.

Sleeve, A, and intermediate sleeve, D, rotate with the camshaft when the engine is in operation. Since it is desired to vary the valve timing while the engine is in operation, it is necessary to provide for the axial movement of sleeve, D, by some method that will permit its continued rotation during the adjusting operation. A third sleeve, G, provided with an internal flange, serves as a shifting collar for sleeve, D. Internal threads are provided on sleeve, G, which engage with threads formed on stationary sleeve, B, for a considerable portion of its length. Gear teeth cut on the outside of G mesh with a small hand-operated worm, H. Rotation of the worm causes G to move axially with respect

to the camshaft in turn shifting D on the camshaft and causing relative displacement between the parts of the cam whether the camshaft is turning or is stationary.

The cams operate the valve by direct action on rocker arms, each pair of valves being actuated by a single rocker arm.

Variation of valve lift is affected by changing the position of the rocker arm fulcrum by means of the mechanism shown in Figures 13. The fulcrum is formed by a small hardened pin bearing against a hardened plate fastened to the rocker arm, the line of contact between the pin and the plate forming the fulcrum axis.

The rocker arm is constrained from changing its position with respect to the valve and cams by trunnions machined on the outside of the rocker arm carried in blocks free to move in the rocker arm housing only, and parallel to the valve-stem axis. The hardened pin is mounted in a block that is movable with respect to the rocker arm housing. The fulcrum point is changed by turning a small handwheel mounted on a threaded shaft, the threads in turn producing linear movement of the block carrying the fulcrum pin.

All the handwheels controlling the valve functions are geared directly to revolution counters, the counter readings giving the value of the timing and lift of the valves.

The engine has heavy mounting flanges integral with and extending the full length of the cast-iron crankcase. This construction permits of flexibility in the mounting of

the engine on dynamometer equipment with the use of the usual parts of this equipment for the testing of engines and without alteration of the crankcase or provision of a special base.

The crankshaft is carried by two large ball bearings and one roller bearing. The roller bearing is located on the gear end of the crankshaft and is mounted directly in the main crankcase casting. The two ball bearings are located on the fly-wheel end of the crankshaft and are mounted in a cylindrical cast-iron cage which is mounted in and bolted to the main crankcase casting. Removal of this cage effects removal of the crankshaft and bearing assembly from the crankcase.

The cylinder construction has been partially described under the description of the compression ratio changing mechanism. A long annular space is formed between the jacket and the barrel which provides for circulation of the cooling water. The cylinder head has a cored water jacket with no internal communication with the cylinder jacket. This construction permits separate control of the flow of the cooling water to the cylinder and head and consequently permits maintaining different water temperatures for the two parts. The guide, jacket, and head are iron castings and the barrel is a steel forging.

Three openings are provided in the head for the insertion of spark plugs, fuel injection valves, pressure indicating devices, etc. These openings are provided with

standard metric spark plug threads. Two openings are located on opposite sides of the head; the third opening is located in the center of the head. Several threaded openings into the jacket and head are provided for connecting cooling water fittings.

Two camshafts are provided, one for the inlet valves, and the other for the exhaust valves. These two shafts are driven from a vertical shaft by helical gears. The vertical shaft is in turn driven by the crankshaft through bevel gears. The vertical shaft bevel gear is mounted directly on a relatively short hollow shaft having internal splines which engage splines on the lower end of the vertical shaft proper. This construction permits the upper shaft to move vertically with the cylinder. The crankshaft bevel gear furnishes also the drive means for the water and oil pumps.

The ends of both camshafts and the top end of the camdriving vertical shaft extend outside their housings to provide driving means for various auxiliaries, such as ignition timing mechanisms, indicators, etc.

The oil pump is a combined scavenging and pressure pump. The scavenged oil is carried off to an external reservoir. To insure a positive oil supply an additional motor-driven pump located at the external reservoir and shown in foreground of Figure 4 is used. The oil from the pressure side of the pump is carried through drilled passages in the base to the front of the engine. The path divides at this point, one path leading to the camshaft mechanism, where the flow

to the various parts is regulated by three sight-feed adjustable oilers, the other path leading to the crankshaft through a bronze casting that rides on the crankshaft. Passages are drilled in the crankshaft for the lubrication of the connecting rod bearing. The bearing in the crankcase, the piston pin, and piston are lubricated by oil thrown off the connecting rod.

The fuel flow was computed from the time required to use the known volume contained in a vertical gauge, which was calibrated at the beginning of this investigation.

The M.I.T. head used in our final investigation was of the conventional flat head type using four vertical poppet valves.

For use with fuel-injection the conventional carburetor was eliminated and a fuel-injection system substituted. A Bosch cam-operated plunger type pump was used (Figures 15, 16, and 17. It was driven from a camshaft of the engine through a gear coupling which permits adjustment of the injection timing at will. The duration of injection for this investigation varied from 19 to 25 degrees of crankshaft rotation, according to the injected fuel quantity. A pressure of about 6 inches of mercury was maintained in the fuel system on the low pressure side of the pump.

The pump consists of a cylinder a (Figure 15) and a piston or plunger b. The cylinder is closed at its upper end by a spring-loaded pressure valve c, from which the fuel line d leads to the injection valve. In the upper part of the

housing is a suction space which is connected with the fuel tank by means of the suction pipe <u>e</u>. Two small holes connect the suction space with the pressure space in the pump cylinder. The stroke of the plunger is constant. The upper edge of plunger <u>b</u> controls the beginning and the slanted groove controls the end of the fuel delivery. The end of the delivery is reached sooner for a small quantity of fuel than for a large quantity. This is brought about by turning the pump plunger into different positions.

The pump cylinder is enclosed by a bushing <u>f</u> to the upper end of which a gear segment is fastened. This segment in turn engages with a toothed rack <u>g</u>, which is actuated manually or may be moved by a micrometer from which readings may be recorded. At its lower end this bushing has two opposite slots, in which a cross arm of the piston is guided, the angular motion of the bushing, caused by sliding the control rod, being thereby transmitted to the plunger.

The operation of the plunger is shown in detail in Figure 15. In the lowest position of the piston, the two opposite ports are opened and the cylinder above the piston is filled with fuel. During the first part of the pressure stroke of the piston a small quantity of fuel is forced back into the suction space until the plunger closes both port holes. From then on, the fuel is put under pressure and the pump begins to force it through the check-valve and the fuel line into the injection valve. Delivery begins as soon as the plunger has covered the ports on the way up and ends as soon as the

sloping edge, indicated by the arrow, opens the port hole on the right-hand side and permits the fuel to escape from the pressure space above the plunger, through the groove in the plunger and the port, to the suction space.

In the two views at the left (Figure 17) the plunger is shown in the position for maximum delivery, in which the edge of the helical groove does not open the port hole at all. The next two views show the position of the plunger for medium delivery of fuel, and the one at the right shows the position when no fuel is being delivered.

As soon as the slanting edge of the groove in the plunger opens the port hole, the pressure in the pump cylinder is relieved. The pressure which still exists in the fuel line, together with that of the valve-spring, forces the pressure valve to its seat. The fuel line is now closed off from the pump cylinder until more fuel is delivered during the next working stroke. The check-valve, however, has another important task to perform. It is highly desirable to relieve the pressure in the fuel line in order to obtain a rapid closing of the injection valve, as otherwise dripping of fuel from the nozzle into the combustion chamber may occur. A special construction of the check-valve provides this pressure relief in an effective and reliable way, as shown in Figure 16. During the working stroke of the pump, the valve is raised from its seat and the fuel flows through the hollow stem and the two connecting holes into a ring groove, and from there to the fuel line. Adjoining the ring groove is a short

cylindrical surface forming a shroud, and above this is the valve head. When the bypass opens, the valve closes. In doing so, the receding valve stem causes an increase in the volume of the fuel line by an amount equal to the volume of the shrouded part of this valve stem. The fuel in the line is in this way suddenly relieved of its pressure, and rapid closing of the injection valve is effected.

Injection timing could only be changed, when using the N.A.C.A. head, by stopping the engine and changing the angular relation between the pump and crankshaft. A magneto type serrated coupling provided means for doing this.

When using the M.I.T. head, an arrangement similar to that used for changing valve timing in the N.A.C.A. head provided a means for changing injection timing quickly and by definite amounts.

Injection Valve or Nozzle.

The function of the injection valve is to inject the fuel into the cylinder of the engine and at the same time to give the jet or spray a shape to conform with the contour of the combustion chamber.

The injection valve used in our preliminary investigations for injection into the cylinder when equipped with the N.A.C.A. head, shown in Figure 18, is of a type developed in the Aeronautic Engine Laboratory at the Massachusetts Institute of Technology especially for investigations of this nature. With a high injection pressure, it has a high degree of atomization but very low penetration. The valve is spring-

loaded so that a fuel pressure of about 3000 lbs. per sq. in. is required to open it. The fuel, passing out with high velocity between the valve and its seat impinges almost perpendicularly upon the face of the hollow cone shown in the enlarged figure which serves to atomize the fuel very thoroughly, and at the same time to absorb a large portion of the velocity of the fuel particles. The resultant spray has the shape of a wide, short cone with an apex angle of about 120 degrees and is nearly solid except for a small hollow near the apex. The penetration was slightly less than one foot. This valve was used in the top hole with the N.A.C.A. head.

The valve used in the M.I.T. head was similar to the first valve with the single exception that the spray angle was about 90 degrees.

Intake Pipes.

Two types of intake pipes were used. In the preliminary investigations with the N.A.C.A. head a variable length seamless steel tube of 3 inch diameter was used, while in the final tests two 2-inch tubes were used as shown in Figures 3 and 5.

III.(a)

Preliminary Considerations

There are several advantages which the fuel-injection engine seems to possess over the usual carburetor engine. They may be listed as follows:

1. Better volumetric efficiency due to elimination of the pressure drop through the carburetor.

2. More even distribution of fuel between the various cylinders.

3. More positive delivery of fuel to the cylinder. That is, the fuel delivery is independent of the air flow and therefore of pulsations in intake system. There is no lag during accelerating period comparable to that existing in the carburetor system.

4. Heat of vaporization of fuel helps to cool residual products which may produce greater volumetric efficiency.

5. Due to simplicity of intake pipe, its form and length may be adjusted to take advantage of ramming action of incoming air.

As against these advantages there are the following disadvantages, or factors tending to offset the above tabulation.

1. The modern well designed carburetor has a very small pressure drop through it.

2. The carburetor has been developed to the point where the mixture ratio is almost perfectly controlled under all

conditions of operation.

3. The fuel-injection system is, if not more complicated than the carburetor, at any rate more extensive. It in general weighs more, and due to the greater number of moving parts, is inherently less reliable. Furthermore, it is in the experimental stage as contrasted to the carburetor which is the culmination of long experience.

4. Granting that distribution is poor in the carburetor engine, which is not a fault in carburetor design but in manifold design, the actual effective mixture in the cylinder may be anything when using an injection system if the injection nozzle design is poor. In other words, the whole success of the injection system depends on how effective the injection nozzle is in delivering the fuel into the cylinder so that the resulting fuel-air mixture is homogeneous and the ineffective fuel (unvaporized or unburned) is an absolute minimum.

5. Maintenance problems are apt to be acute in the case of the fuel-injection engine, especially as regards the carbonizing of injection valve and wear on delicate parts such as pump plungers.

The above enumerated factors should, of course, be thought of as applying to four-stroke cycle engines only, and in the case of the fuel-injection engine to one where injection is direct into the cylinder.

Injection systems have been and are being used in which the fuel is injected into the inlet manifold just outside the cylinder. Tests made in the Aeronautical Laboratory of the

Massachusetts Institute of Technology (Reference A) show that in general this is not as effective as injection into the cylinder. The object in injecting into the inlet manifold is to obtain a more uniform mixture than was thought possible with injection direct into the cylinder. Undoubtedly, the idea of injecting into a presumedly uniformly moving column of air conjures up a more thorough mixing than injection into the cylinder. However, the pulsations or harmonic flow in the air column (as contrasted with the ramming effect or resonant pulsation which at optimum conditions is in phase with the engine cycle) prevent the attainment of the ideal of a more or less uniform flow of air past the injection valve or nozzle. At best, injection into the inlet manifold is a poor compromise dictated by a basically poor injection valve design. Furthermore, it prevents taking advantage of, or at least partially nullifies the effect of vaporization in the cylinder in increasing the volumetric efficiency.

Based on the above considerations we choose to investigate the fuel-injection engine in which the fuel is injected directly into the cylinder rather than into the manifold. This seems the logical starting point. The problem as we saw it in the narrow sense was this: How may we get the optimum performance out of a given engine and with a given injection system, using fuel injection into the cylinder? The problem we set ourselves to solve neatly side-steps the all-important question of fuel pump and nozzle design. It was hoped to try some novel forms of simple injection valves or nozzles, but this we were unable to do due, as mentioned in the first section of this thesis,

to lack of time. However, the staff of the Aeronautical Engineering Department have been working on this problem for some years now, and we have had the benefits of their experience. So while our problem does not concern itself with the basic design of the injection system, it by no means follows that this all-important element of the engine has not received attention. As a matter of fact, we feel that we were very fortunate in being able to start with a well-proven and efficient injection system.

Some General Considerations in Regard to the Effect of Valve Timing on Engine Power.

There are four valve events in a complete cycle of the engine, namely intake valve opening and closing and exhaust valve opening and closing. With any given set of cams these events are measures of the time the valves are open and exert a great influence on the volumetric efficiency or breathing capacity of the engine.

In the ideal cycle the exhaust closes and intake opens at top dead center while the other two events take place at bottom center.

Due to the impossibility of instantaneous opening or closing of a valve and also to inertia effects in the charge itself, both of which may be grouped under the heading of "mechanical time factors," the time of occurence of the four events is shifted away from the ideal positions.

For any particular valve timing mechanical time factors such as maximum allowable accelerations on ramp, slope and

nose of cam govern the design of the cam itself. Of course the object here is to get the valve opened and closed in the shortest possible time, and to have it remain fully open the greatest possible time.

The other "mechanical time factor" influencing power is the inertia effects of the charge in its progress into, through and out of the cylinder. The effect of each valve event in relation to inertia effects will now be taken up in some detail. An arbitrary starting point in the discussion is the proper or optimum valve timing. Effect of changes in each valve event from this point are considered in what follows.

Inlet opening and exhaust closing are often considered together, the term "dwell" indicating that there is a period around top center during which both valves are closed. Likewise, the term "overlap" refers to a timing such that both valves are open during a period near top center. Formerly it has been argued that dwell was advantageous, but this view has been abandoned, and practically all modern engines use some overlap. Overlap is, of course, especially advantageous in supercharged engines. This discussion, however, should be thought of as applying primarily to unsupercharged engines.

Intake Opening.

If the inlet valve opens too early some of the exhaust gases will be blown out the inlet valve. This will dilute the incoming charge and may establish a back or counter flow in the intake pipe which will have to be overcome by piston "suction" before the next charge is drawn in. Both of these effects will, of course, reduce the volumetric efficiency and

horse-power of the engine.

If, on the other hand the inlet valve opens too late, inertia effects in incoming charge will reduce the volumetric efficiency. Also the overlap is decreased which will have the same effect, i.e. decreased power, but in this case due to lessened scavenging effects.

Exhaust Closing.

Early closing of exhaust valve reduces the overlap with the same results as mentioned above and a primary result, i.e. independent of considerations of overlap due to the retention of excess residual gases in the cylinder. This may be accentuated if the optimum timing takes advantage of exhaust pipe negative ram or suction effect.

Late closing of exhaust valve does not have as marked an effect as early closing does. This is believed to be due to two conflicting tendencies, a drop in power due to late closing divorced from overlap and perhaps an increase in power due to increased overlap.

Inlet Closing.

As the piston moves down, an air flow is established in intake system and a momentum acquired which is taken advantage of by closing well after bottom center. Too early a closing will take no or little advantage of this ram effect. Too late a closing may permit a back flow to be established and an escape of some of the charge which was rammed into the cylinder. Optimum inlet closing timing is usually quite critical in any engine but may become more critical if the intake system is such that large ram effects are present, as it is with certain

inlet pipe dimensions.

Exhaust Opening.

The optimum timing for this event is not sharp. Reasonable valves on either side show rather small changes in power. Most of the work of the cycle is done in the first 45 degrees or so of piston travel. However, these statements must be slightly modified when dealing with a fuel injection engine. Here the effect of changes from the optimum point is greater than in the carburetor engine. This would seem to indicate slower burning with higher temperatures and pressures later in the cycle.

Overlap Effects.

This is a combination of effects due to the two valve events of which it is composed. It is convenient sometimes to regard it as a separate consideration, or perhaps more accurately, to regard the combination of the two events rather then the separate effects.

Overlap provides a continuous passage from intake to exhaust pipe and permits of a scavenging action if a pressure differential sufficient to produce a flow exists. This pressure differential does exist, of course, if a supercharger is used, but it may also be produced by a proper combination of inlet and exhaust pipe dimensions, or if the exhaust conditions are fixed by proper intake system dimensions.

Effect of Speed on Valve Timing.

An "optimum valve timing" for one speed is not, of course, the optimum for any other speed. Variation in optimum timing with speed for a supercharged engine is less than for an unsupercharged engine. An intake system which will give maximum

ram at a certain speed will not in general be the most favorable at another speed, and this fact will be reflected in valve timing. A broad generalization as to the effect of speed on timing is: at increased speed close later and open earlier.

An additional factor in a consideration of valve timing in the case of the fuel-injection engine is the effect of fuel-injection timing. As far as we were able to determine, its effect is secondary. Properly it is believed that the problem should be considered the other way around, i.e. the effect of valve timing on injection timing. At our first series of tests using the N.A.C.A. head optimum injection timing was 2 degrees or so after exhaust valve closing, and reasonable variations in injection timing produced appreciable differences in power. However, with the M.I.T. head, injection timing optimum point was less distinct as judged from power output, but was definitely much nearer top center (about 5 degrees after usually).

Basic Consideration of Effect of Intake Pipe Dimensions on Engine Power.

In the fuel injection engine no intake system external to the cylinder is necessary. Any intake pipe which is used is extra and useless weight unless by its use the power of the engine is increased. On the other hand, with the carburetor engine an intake system must be used to connect the carburetor with the various cylinders. We are using an unsupercharged fuel-injection engine and the following considerations should be thought of as applying primarily to it or to a similar engine.

With no intake pipe in use the air drawn into the cylinder is supplied from an infinite reservoir, thus the variable orifice formed by the port and valve. No appreciable momentum is imparted to any air which is not drawn into the cylinder. With large valve area and reasonable engine speed it is essentially a displacement phenomenon, although in the usual engine wire-drawing effects do exist. The point is that the column of air which must be set in motion is of minimum length, and further, that resistance to flow due to viscosity is a minimum.

We now put on an inlet pipe which for simplicity we will assume has a straight line axis and unifrom diameter, the latter being determined by the size of inlet port. We have increased the total resistance of intake system to air flow by a definite amount. We have also increased the mass of air which must be set in motion for flow to take place into the cylinder. Furthermore, the total mass of air which must be set in motion is greater than the total mass which will be drawn into the cylinder (if the pipe length is greater than a certain minimum, which, in the case under consideration, it actually will be).

Thus for the initial cycle we impart momentum to the air in excess of the minimum required to fill the cylinder, i.e. there is residual energy in the air in the inlet pipe after the valve is closed in the form of velocity of the air column. If this residual energy is dissipated by friction before the next engine cycle takes place, there is, of course, a complete loss to the atmosphere of the engine power expended in im-

parting the excess momentum in the initial cycle. Thus during each cycle extra work will be done, and pumping loss will be higher than with no inlet pipe, from considerations of momentum of air alone, leaving out of consideration the increased friction due to longer passage.

The increased friction will, of course, reduce the volumetric efficiency of the engine.

From the above one thing is clear: that the excess energy stored in the air column must be of sufficient magnitude that it will not be dissipated before the next cycle. This means that there is a minimum pipe length below which no "carry-over" effects will be noted. This length is probably only slightly greater than the length necessary to contain a volume equal to the displacement of the cylinder.

It is realized that the carry-over energy is in the form of pressure waves. The wave length will be a function of the speed of the engine and the dimensions of the system. If these are correctly adjusted a resonant condition will be reached which will produce a ramming effect, i.e. an increase in volumetric efficiency.

Our problem is merely to find the resonant length of intake pipe as indicated by the power output of the engine, which under otherwise constant conditions is a measure of volumetric efficiency.

It is not considered necessary to go into a purely mathematical treatment since it does not give a definite answer, which is what we desire.

Fuel Injection System Variables.

These are:

1. Fuel injection valve.
2. Pump operating cam.
3. Injection timing.
4. Injection pressure.

No complete study is to be made of these variables, since a good deal of work on them has already been done in the Institute Aeronautical Laboratory. For our investigation they are secondary variables.

Two types of injection valves of proven efficiency are available. They differ only in the cone angle, a 120-degree angle, one which has been found to work best in the N.A.C.A. head, and a 90-degree cone valve suitable for use in the M.I.T. head.

Three cams are available which differ in slope and thus give various injection rates.

Injection timing will be varied to determine best setting.

Injection pressure is, of course, a very important factor in the fuel-injection system. Low injection pressures with good atomization is the goal in design. The question of how poor atomization characteristics will be put up with to achieve low pressures can only be answered by the individual engineer.

III.(b)

Pertinent Work by Others.

Inasmuch as the scope of this thesis is a continuance of work previously done and reported on from the Aeronautical Engine Laboratory at the Massachusetts Institute of Technology, rather complete reviews of those investigations are presented below:

Fuel Injection with Spark Ignition in an Otto-Cycle Engine, by C. F. Taylor, E. S. Taylor, and G. L. Williams. S.A.E. Journal - March, 1931.

This investigation was carried out in the Aeronautical Engine Laboratory at the Massachusetts Institute of Technology to determine the practical value of the use of a fuel-injection system in place of a carburetor on an Otto-cycle engine using spark ignition. The apparatus used in this investigation consisted essentially of the same apparatus used in our investigation,* except that the air flow was measured by means of a sharp-edged orifice; a 50 gal. drum was interposed between the orifice meter and the engine to reduce pulsations; and a different injection nozzle was employed.

A series of tests were run to compare the performance of an engine having (a) injection of fuel into the inlet pipe and (b) injection into the cylinder on the one hand, with the performance obtainable with the conventional carburetor on the other. Fuel injection, either into the inlet pipe or into the cylinder was found to be superior in performance to the usual

type of carburetion. The available power was increased by over 10 percent with injection into the cylinder, and substantially lower fuel consumption was obtained. The mechanical problems of injection were found simpler than those of a similar design for Diesel engines. Gasoline was used for most of the investigation, but comparative tests were made using fuel oil. Operation with fuel oil compared very favorably with gasoline operation at a compression ratio at which no detonation occurred with either fuel. However, it was found that the compression ratio had to be kept quite low on account of the poor antiknock tendencies of the fuel oil. The influence of injection rather than carburetion of the fuel, either gasoline or fuel oil, upon the highest compression ratio was insignificant.

All tests were made at full throttle with the load adjusted to maintain a constant speed of 1500 R.P.M. Compression ratios from 3:5:1 to 5:1 were used for these tests. Most of the tests, however, were made with a compression ratio of 4:3:1 with a constant spark advance of 28 degrees. The mixture ratio was varied for each injection arrangement from the lean limit to very rich and observations were taken during each run to determine the brake horse-power, fuel consumption and air consumption. Gasoline was the fuel used in these tests.

Tests were first made with a conventional carburetor to afford a basis of comparison. The next set of runs was made with the injection valve placed in position in the inlet pipe to inject the spray in the same direction as the incoming air for the first set of runs.

For the next group of runs injection was down the pipe against the air flow. The Bosch valve was used for these runs and the carburetor was left in place with its fuel jets closed. The fuel was injected into the inlet pipe during the suction stroke, beginning 30 degrees after top center and continuing for about 21 degrees of crankshaft rotation.

The performance curves from these tests show that there is little difference between the power obtained with the carburetor and that obtained with injection. Injection against the air flow seems to be slightly better than injection in the same direction as the air flow.

These curves indicate the fact that nearly full power is maintained with inlet pipe injection to a leaner limit than can be obtained with the carburetor. Further, the minimum specific fuel consumption with inlet pipe injection is considerably lower than that with the carburetor, while the minimum specific fuel consumption with inlet pipe injection is considerably lower than that with the carburetor.

Similar tests to those discussed above were repeated with the carburetor removed and replaced by a section of plain tubing.

The principal effect of removal of the carburetor is a large increase in volumetric efficiency with a corresponding increase in brake mean effective pressure. This gain in performance may be considered as a direct advantage of the manifold injection. The sharp break in performance at the lean limit is again clearly illustrated. Injection down the inlet pipe is consistently better than injection in the direction of

the air flow. The gain in maximum brake horse-power with injection into the inlet pipe is 6.5 percent. The corresponding gain in volumetric efficiency is 6.65 percent. This gain in volumetric efficiency seems to be due entirely to the removal of the carburetor.

It was found that the best power and economy is obtained by injection into the inlet pipe. This was thought to be due chiefly to the imperfect distribution of the fuel in the combustion chamber.

Further tests were made to determine the knocking characteristics of the engine with fuel injection into the cylinder by raising the compression ratio to the point of incipient detonation. During these tests the engine was throttled to give the same volumetric efficiency with fuel injection as with the carburetor. Little or no difference in the knocking characteristics was noted between introducing the fuel through the carburetor and injection through the injection valve.

Fuel oil injection was next tried with all test conditions identical except that the compression ratio was lowered to 3:5:1, which was the highest useful compression ratio for the fuel oil, to avoid detonation. Injection into the cylinder alone was tried, since attempts to operate the engine by injecting the fuel oil into the intake pipe were failures. The results of these tests were similar to those obtained by the use of gasoline. However, no comparisons of these tests with those obtained when using gasoline could be made due to the differences in the compression ratios.

A drop in volumetric efficiency was noted when the fuel oil

was used and this drop was extremely marked. This drop was explained by the fact that the volatility of the fuel oil was lower and therefore its rate of vaporization lower. This in effect corresponds to a late injection of gasoline in which case the cooling effect of the vaporization on the cylinder gases occurs too late to be of much benefit to the volumetric efficiency. This decrease in volumetric efficiency was accompanied by a proportional decrease in brake mean effective pressure. An increase in specific fuel consumption was deemed due to the lower volatility of the fuel, wherein the heavy ends did not vaporize so that the effective mixture ratio is somewhat leaner than the actual. Considerable trouble was experienced with fouling of the plugs during warming up of the engine, although it would start easily enough.

It was found that automobile spark plugs with their more exposed electrodes functioned better and this was deemed due not so much to the higher temperature of the exposed points as to the better mixture at the spark gap.

The following conclusions were drawn from the results of this investigation.

1. High pressure fuel-injection either into the inlet pipe or into the cylinder is superior to carburetion in respect to performance. The maximum available power may be increased by 7 to 11 percent, depending upon the method of injection. This increase in power is due partly to elimination of the pressure drop through the carburetor and partly to direct effects of the spray itself. The improvement in specific fuel consumption may be traced to more even distribution of

the fuel between successive cycles, and to a lesser extent to the higher mechanical efficiency which accompanies the gain in power.

2. The highest useful compression ratios, as fixed by detonation characteristics, are essentially the same with injection of the fuel as with carburetion, at the same volumetric efficiency.

3. The best power is obtained with injection of the fuel into the cylinder, and the best fuel consumption is obtained with injection into the inlet pipe against the air stream. It is expected that good turbulence would improve the fuel consumption with injection into the cylinder.

4. Fuel oil can be used as well as gasoline for injection into the cylinder except for the more severe limitations upon useful compression ratio fixed by its detonation characteristics and its tendency to foul the plugs during the warming up process.

Further Investigation of Fuel Injection in an Engine Having Spark Ignition, by E. S. Taylor and G. L. Williams. S.A.E. Journal - April 1932.

This investigation was a continuance of the work previously summarized. A four-cycle engine was used, but since the fuel was injected during the compression stroke, the results might be applicable to the two-stroke cycle without possibility of loss of fuel through the exhaust. Particular attention was paid to late injection or late ignition and stratification as a means of controlling detonation with high compression.

The fuels used were aviation gasoline, ordinary Diesel fuel and hydrogenated fuel oil, with the hydrogenated fuel

seemingly offering interesting possibilties. Directed turbulence was found to be essential for good distribution of the fuel in the cylinder and satisfactory operation of the engine.

The apparatus consisted of a 3 1/4 x 4 1/2 in. Cooperative Fuel Research engine, together with such other equipment required for a complete performance test. The injection valve was especially modified for this investigation since it was imperative to meet the requirements of late injection necessary for two cycle engines, and to fit the engine.

At the beginning of the investigation it became apparent that distribution of mixture in the combustion space was poor at the time of passage of the spark. This was shown by the erratic operation at late injection. Spark plugs with extended points were employed to determine whether the original plug was in an unfavorable position. The results of this test showed that without directed turbulence the distribution of the fuel was very poor and operation was uncertain regardless of the position of the spark plug.

A shrouded inlet valve was designed to cause rotary trubulence in the combustion space. This valve assembly was set up in such a way that the position of the shroud might be varied through a range of 90 degrees. Tests showed that the most consistent results were obtained with the valve placed so as to give full rotary turbulence while, better performance could be obtained with the shroud moved to a position where full rotary turbulence was not obtained.

All tests were made at full throttle with the load ad-

justed to maintain a constant speed of 1000 R.P.M. The injection timing was varied from about bottom center at the beginning of the compression stroke to as late as was consistent with steady firing. At each injection timing, the mixture ratio was varied from the lean limit to the very rich. The tests were made at a compression ratio of 5:1 except in tests of the effect of injection timing on detonation, during which the ratio was varied, and tests with fuel oil, in which the ratio was 4:1. The shrouded valve was in place and set to give maximum rotary turbulence for all tests recorded.

The first series of tests was made with aviation gasoline. The timing of the beginning of injection was varied from 30 degrees before bottom center to 90 degrees after bottom center on the compression stroke. The best performance as to fuel-air ratio, brake mean effective pressure and specific fuel consumption were plotted against injection timing, These curves were irregular, and this irregularity was probably due to the fact that the air from the unshrouded side of the inlet valve sweeps directly across the injection valve, cleaning away the spray as it comes from the valve and thus stratifying the fuel into the upper part of the combustion space. The inlet valve closes at about 30 degrees after bottom center, and the stream of very high velocity air from the inlet valve being stopped, the penetration of the spray increases, giving better distribution of the fuel through the combustion space and increased power. The minimum fuel consumption seems little affected in this range, the decrease in mechanical efficiency due to the decreased power apparently being balanced by increased thermal

efficiency due to stratification. However, the increasing pressure and temperature in the combustion chamber after the closing of the intake valve, resulting from compression, and the decreasing interval of time before the spark passes, apparently bring about poorer distribution of the fuel and a corresponding decrease in performance.

A plain 45 degree poppet-type injection valve with greater penetration and poorer atomization was tried over the same range of injection timings and the engine was found to be much less sensitive to the stratification effects. The results indicate that, for best performance, injection should begin about simultaneously with the closing of the inlet valve and that the performance obtained under these conditions is excellent.

Next, tests were run with a special hydrogenated aviation Diesel fuel of low volatility manufactured by the Standard Oil Company of New York. The results were similar to those obtained by the use of aviation gasoline, and the performance was slightly better with injection beginning at the closing of the intake valve. This fact suggests an important field of investigation for the use of hydrogenated fuels in injection systems for both two and four-cycle engines.

A brief investigation was then undertaken because of the possibilities of using higher compression ratios without detonation with late injection of the fuel.

The first set of runs in this series was made with aviation gasoline. With this fuel, incipient detonation occurred at a compression ratio of 5:5:1 with injection early, at the beginning of the suction stroke. Thereafter, the compression ratio

was increased by one-fourth unit increments while the injection timing was retarded to incipient detonation in each case.

The results show a slight gain in brake mean effective pressure over the normal range, while the fuel consumption improved over the normal range, reaching a minimum when the fuel was injected at about 80 degrees after bottom center, at which point the fuel consumption was 20% less than obtained under usual operating conditions.

In the next series of runs it was definitely established that suppressing detonation by retarding the spark offered a distinct advantage in maximum power over detonation control by retarding injection.

IV.

Method of Attack.

Given the problem as stated in the previous section: How may we so adjust the engine variables at our command so that we may get the optimum in performance from the given engine? This is, of course a problem which can only be answered by actual tests of the engine. It is an experimental investigation rather than a mathematical one. The solution is only to be arrived at by a series of test runs on the actual engine.

We decided on a standard speed for all tests of 1600 R.P.M. This choice of speed was based fundamentally on the usual operating speed of an aircraft engine of around 2000 R.P.M. The speed of 1600 R.P.M. was considered to be as close to this as we could safely get, having in view the type of test engine. The demands on the engine in general, and the fuel injection system in particular, are, of course, more severe at the higher speed, and we wished to approach the usual aircraft engine speed in order to simulate actual aircraft operating conditions.

The other basic variable we selected was the compression ratio. All preliminary work was carried out at a compression ratio of 5:1. Final runs were at various compression ratios.

Other variables at our disposal were the following:

1. Type of cylinder head
   a. N.A.C.A. Variable Valve Timing Head (Figure 1)
      As may be seen in the figure, this head is of

the conical type with valves set at an angle to the vertical. (For complete description of this head see Section II).

b. M.I.T. Flat Head (Figure 2)

In this head all valve axes are vertical (For complete description of this head see Section II).

2. Valve timing.

The N.A.C.A. head provided a method of studying the effect of various valve timings.

3. Inlet pipe.

Due to absence of a carburetor we were free to use any kind of inlet pipe we wished.

4. Fuel pump operating cam.

Three different types of cams, designed to give different injection rates were available.

5. Injection timing.

6. Spark timing.

7. Mixture ratio (As evidenced by various fuel flow rates).

8. Injection pressure.

9. Type of nozzle.

From a consideration of the object in view and the variables at our disposal it was decided that the investigation should be divided into two major parts: the first part to consist of all preliminary work necessary to insure that all variables at our disposal should be given proper values such that the combined effect would produce optimum results, as measured in terms of Brake Mean Effective Pressure and Brake Specific Fuel Consumption: the second part to consist of a formal series of runs

at various compression ratios at the optimum conditions as determined in the preliminary tests.

The first part was subdivided into three basic groups of tests.

1. Valve timing.
2. Length and diameter of intake pipe.
3. Fuel system variables.
    a. Injection pressure.
    b. Type of fuel pump operating cam.
    c. Type of fuel injection valve.

This program takes care of all variables at our disposal except the following:

1. Type of Head.

    This is not a variable in the same sense that the other factors are. The N.A.C.A. variable valve timing head was used for valve timing tests which were run more or less concurrently with the intake pipe tests. The M.I.T. head was used in all final tests.
2. Spark timing.
3. Injection timing.
4. Mixture ratio.

The effect of each of these was determined during each run of all series and adjusted for optimum conditions.

The mixture ratio was never measured directly as any reasonable orifice system we could set up would disturb the flow in intake pipe. Mixture ratio tests were plotted against fuel flow when adjusting for maximum power in individual runs.

To sum up, the method of attack was essentially this:

1. Preliminary runs to determine best combination of all variables.
2. Final runs at various compression ratios to record performance under optimum conditions.

V.

Results of Preliminary Investigations.

Valve Timing

The tests in connection with valve timing were made with the N.A.C.A. variable valve timing cylinder head in place. All other variables were brought to their optimum value and valve timing runs were made, varying one valve event at a time and using its optimum value in succeeding tests. This process was repeated, succeeding variations becoming smaller and smaller until it was certain that the optimum combination was reached.

It was found that the optimum valve timing was not appreciably affected by changes in intake pipe length. Conversely, the critical or optimum pipe length was not appreciably different at various valve timings.

Valve timing runs made with the N.A.C.A. head in place were used as a basis for the construction of valve cams for the M.I.T. head. It was found that the inlet cam already installed in the M.I.T. head could be used. A complete new exhaust camshaft was constructed and installed.

The N.A.C.A. head is equipped with Liberty engine valves while the M.I.T. head has the smaller Hispano Suiza or Wright valves.

The lift used in the N.A.C.A. head was .323 inches on

intake and .318 inches on inlet valve, while in the M.I.T. head the lift was .444 inches on both valves. The shape of the lift curve is not the same in the two heads as shown in Figure 19. The longer dwell of the N.A.C.A. cam is a more favorable factor than the higher lift of the M.I.T. cam. Furthermore, the valve arrangement in the N.A.C.A. head is more favorable than that of the vertical valve in the M.I.T. head.

These factors mentioned above resulted in a higher maximum power output of the engine when using the N.A.C.A. head than when using the M.I.T. head.

<u>Curves of Valve Timing Tests</u> (<u>Figures 23-24</u>).

These curves bear out and illustrate the basic facts discussed under Section III and indicate the method used in actually finding the optimum valve timing. In connection with these curves attention is invited to the marked increase in power with very large values of overlap.

<u>Length of Intake Pipe</u> (<u>Figures 20-21</u>)

Length of intake pipe vs. Brake Mean Effective Pressure was the basis for many runs. A typical curve is shown in Fig. 20. The characteristics of this curve, i.e. two peaks followed by a gradual fall with increasing length of intake pipe, were duplicated in all other curves of this type. The two peaks occurred at substantially (within 2 inches) the same intake pipe lengths when using either head and either one 3-inch or two 2-inch pipes. In the case of the N.A.C.A. head the outer or second peak was higher than the inner or first peak. With the M.I.T. head this condition was reversed. In both heads the peak was fairly critical.

The optimum length of intake pipe for the N.A.C.A. head was 74 inches using a single 3-inch pipe. For the M.I.T. head best results were obtained with two 2-inch pipes of 56 inches length.

Fuel Injection System Variables.

The injection pressure used in all tests was the same, or more correctly, the injection valve spring setting was constant. Opening pressure was approximately 3000 pounds per square inch. Lower pressures existed for smaller injected fuel quantities. Also, reduction in engine speed reduced the actual pressure in fuel line as measured by a balanced diaphragm indicator. Inasmuch as our Brake Specific Fuel Consumption when using "Safe-T-Fuel" was rather high, we did not reduce injection pressures, as we believed that to do so would further increase fuel consumption.

Injection Timing (Figure 22).

This was a secondary variable but was adjusted for optimum power. Using the N.A.C.A. head optimum injection timing appeared to be shortly after exhaust valve closed, both with safety fuel and gasoline. Using the M.I.T. head, optimum timing was about 5 degrees past top center on suction stroke with both fuels. This setting was not at all critical, little variation in power being noted with injection timing up to about 25 degrees after top center, but a distinct fall in power being evidenced with timing later than 35 degrees.

The timing referred to above is the nominal timing. Stroboscopic readings of actual injection lag indicated that injection starts 5 to 8 degrees after nominal timing.

Type of Fuel Pump Operating Cam.

Performance was distinctly best when using the cam which gave highest acceleration of plunger and hence highest injection rate.

Upkeep of Injection Valve.

This valve was taken out for inspection at the beginning of each day's operation. Carbon formation was quite slow when using gasoline fuel, but appreciably faster when using safety fuel. The usual deposit was a ring of carbon on injection valve face encircling the discharge orifice. In time this would interfere with proper spray characteristics unless carbon was removed. However, with quite an appreciable ring of carbon built up around orifice, perhaps to a height of .04 to .06 inches, no visible effect on the spray cone was noticed when fuel was ejected with valve removed from the cylinder. It appears that attention to the injection valve of the same kind and extent as is ordinarily given spark plugs is entirely sufficient.

VI.

Results of Final Tests.

Figure 31 sums the results of the final series of formal runs with all variables fixed as nearly as possible at their optimum values. As is apparent from examination of the curves, the maximum power increases and the fuel consumption decreases with increased compression ratio. The following table shows the results in condensed form:

Table I.

Fuel: Safety Fuel

Compression Ratio	Max. BMEP	BSFC at Max. BMEP	Min. BFSC
5:1	147	.635	.540
6:1	155	.560	.485
7:1	162.5	.560	.468
7.5:1	168	.540	.458
8:1	177	.540	.455

For the purpose of comparison, additional runs at 5:1 and 8:1 compression ratio were made on gasoline fuel, that used at 5:1 ratio being a mixture of automobile gasoline plus 20% benzol, while at the high ratio, a blend of domestic aviation gasoline plus 20% benzol and 5 cubic centimeters of ethyl fluid per gallon was used. The results of these runs are as follows:

Table II.

Fuel: Gasoline

Compression Ratio	Max. BMEP	BFSC at Max. BMEP	Min. BFSC.
5:1	148.5	.540	.481
8:1	184	.455	---*

Using safety fuel slight detonation was noticed during the first 7.5:1 compression ratio run in the middle range of mixture ratios. More noticeable detonation occurred during the 8:1 ratio run. As a consequence of this, the formal runs at these two ratios were made with slightly retarded spark, this being shifted from 25 1/2 degrees advance to 20 degrees for the 7.5:1 run and to 18 degrees for the 8:1 run.

As seen in the preceding tables, the power obtained using gasoline is slightly greater and fuel consumption is markedly less than when using safety fuel.

The greatly increased fuel consumption when using safety fuel is undoubtedly due to the relatively poorer vaporization characteristics of it as compared to gasoline. Evidently when safety fuel is injected a very appreciable portion of the charge is not effective during the combustion process and either leaks past the rings or is blown out the exhaust or both. This portion of the charge which is wasted is much greater in the case of safety fuel than in the case of gasoline, as evidenced by the marked increase in rate of crankcase dilution noticed when operating on safety fuel.

* No minimum BFSC was obtained due to severity of detonation as mixture ratio was reduced.

Starting the engine on safety fuel was not particularly satisfactory. Since all our tests were made at full throttle, no throttle valve was used and consequently no fine adjustments on air supply for starting were available. This factor contributed materially to the difficulty of starting on the heavy fuel. Although we have no evidence to support it, it is believed that starting in the usual service engine, which, of course, would be equipped with fine air throttle control, would not be difficult.

VII.

Conclusions.

1. The optimum valve timing is not appreciably affected by the length of intake pipe.

2. Large valve overlap is distinctly advantageous.

3. The optimum intake pipe length varies with the cross-sectional area, being shorter for smaller areas.

4. The optimum intake pipe length is quite critical.

5. For engines of this type the optimum injection timing appears to be early in the suction stroke.

6. Although the type of safety fuel used appears to have excellent anti-knock properties, the fuel consumption is higher than when using gasoline.

7. Satisfactory operation of an engine of this type on safety fuel requires a higher order of injection valve performance than required when using an ordinary gasoline fuel.

List of Illustrations.

Figure I. Transverse Section, N.A.C.A. Test Engine.

Figure II. Longitudinal Section, N.A.C.A. Test Engine.

Figure III. N.A.C.A. Engine, M.I.T. Head.

Figure IV. N.A.C.A. Engine, M.I.T. Head.

Figure V. N.A.C.A. Engine, M.I.T. Head.

Figure VI. N.A.C.A. Cylinder Head.

Figure VII. N.A.C.A. Cylinder Head, Side View.

Figure VIII. N.A.C.A. Cylinder Head, Front View.

Figure IX. Compression Ratio Adjusting Mechanism.

Figure X. Camshaft Assembly. Parts slightly displaced.

Figure XI. Camshaft Assembly. Parts fully displaced.

Figure XII. Cam Adjusting Mechanism.

Figure XIII. Rocker Arm and Housing.

Figure XIV. Cam and Follower.

Figure XV. Bosch Pump

Figure XVI. Pressure Relief Valve.

Figure XVII. Barrel with Plunger.

Figure XVIII. Injection Nozzle.

Figure XIX. Injection Nozzle.

Figure XIX(a) Comparison N.A.C.A. and M.I.T. Valve Lift Curves.

Figure XX. Effect of Length of Intake Pipe on Brake Mean Effective Pressure.

Figure XXI. Effect of Length of Intake Pipe on Brake Mean Effective Pressure Varying Quantity of Fuel Injected.

Figure XXII. Effect of Varying Injection Timing on Brake Mean Effective Pressure.

Figure XXIII. Effect of Valve Timing on Brake Mean Effective Pressure Using D.A.G.

Figure XXIV. Effect of Varying Valve Timing on Brake Mean Effective Pressure Using Safe-T-Esso.

Figure XXV. Power Output vs. Fuel Flow at Compression Ratio 5 Using Aviation Gasoline Plus 20% Benzol.

Figure XXVI. Power Output vs. Fuel Flow at Compression Ratio 5 Using Safe-T-Esso.

Figure XXVII. Power Output vs. Fuel Flow at Compression Ratio 6 Using Safe-T-Esso.

Figure XXVIII. Power Output vs. Fuel Flow at Compression Ratio 7 Using Safe-T-Esso.

Figure XXIX. Power Output vs. Fuel Flow at Compression Ratio 7.5 Using Safe-T-Esso.

Figure XXX. Power Output vs. Fuel Flow at Compression Ratio 8 Using Safe-T-Esso.

Figure XXXI. Effect of Compression Ratio on Brake Mean Effective Pressure and Brake Specific Fuel Consumption.

Figure XXXII. Typical Analysis of Safe-T-Fuel.

IX.

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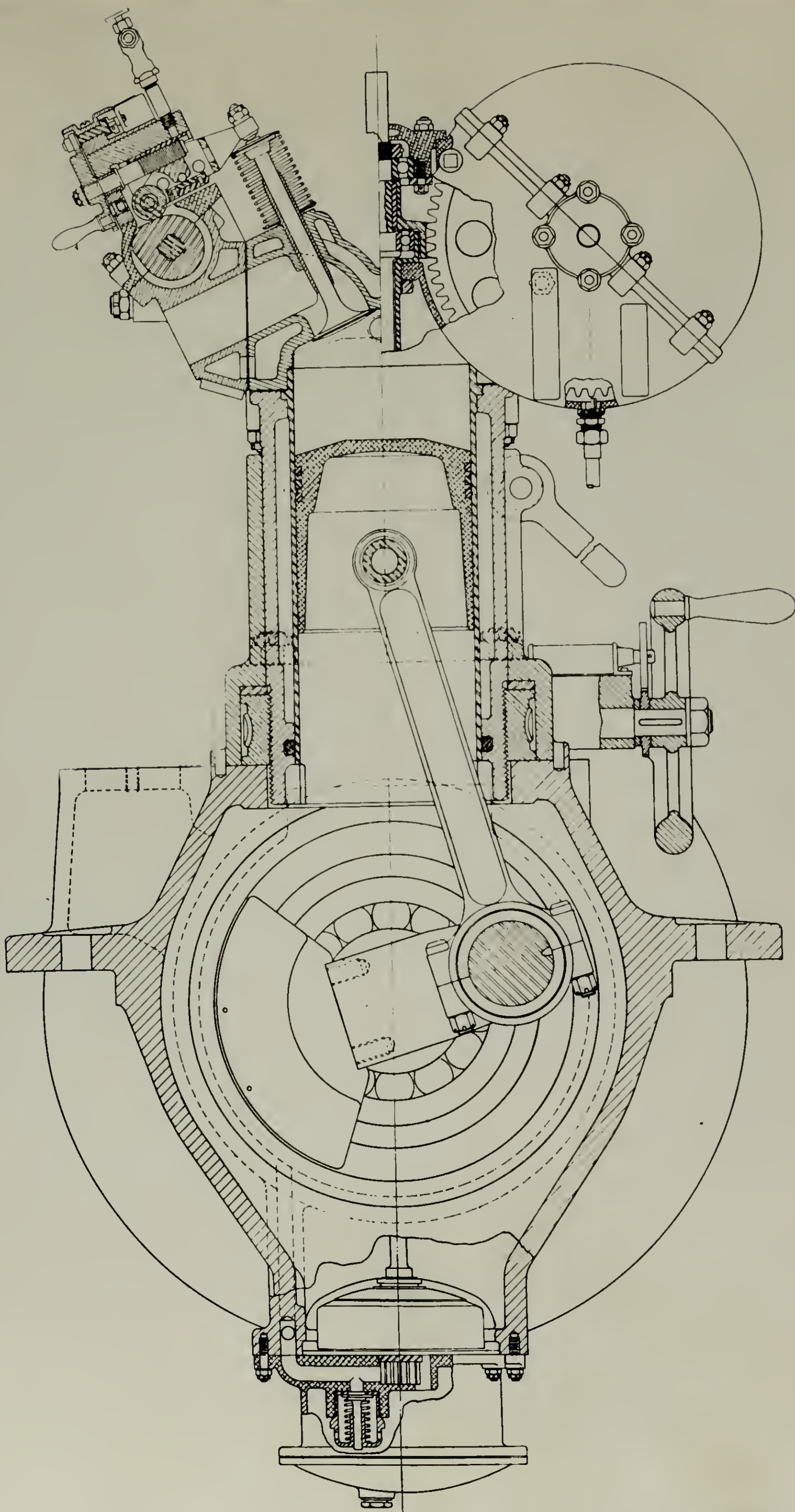


Fig 1.—Transverse section through single-cylinder test engine, showing adjustment for varying the compression ratio while running

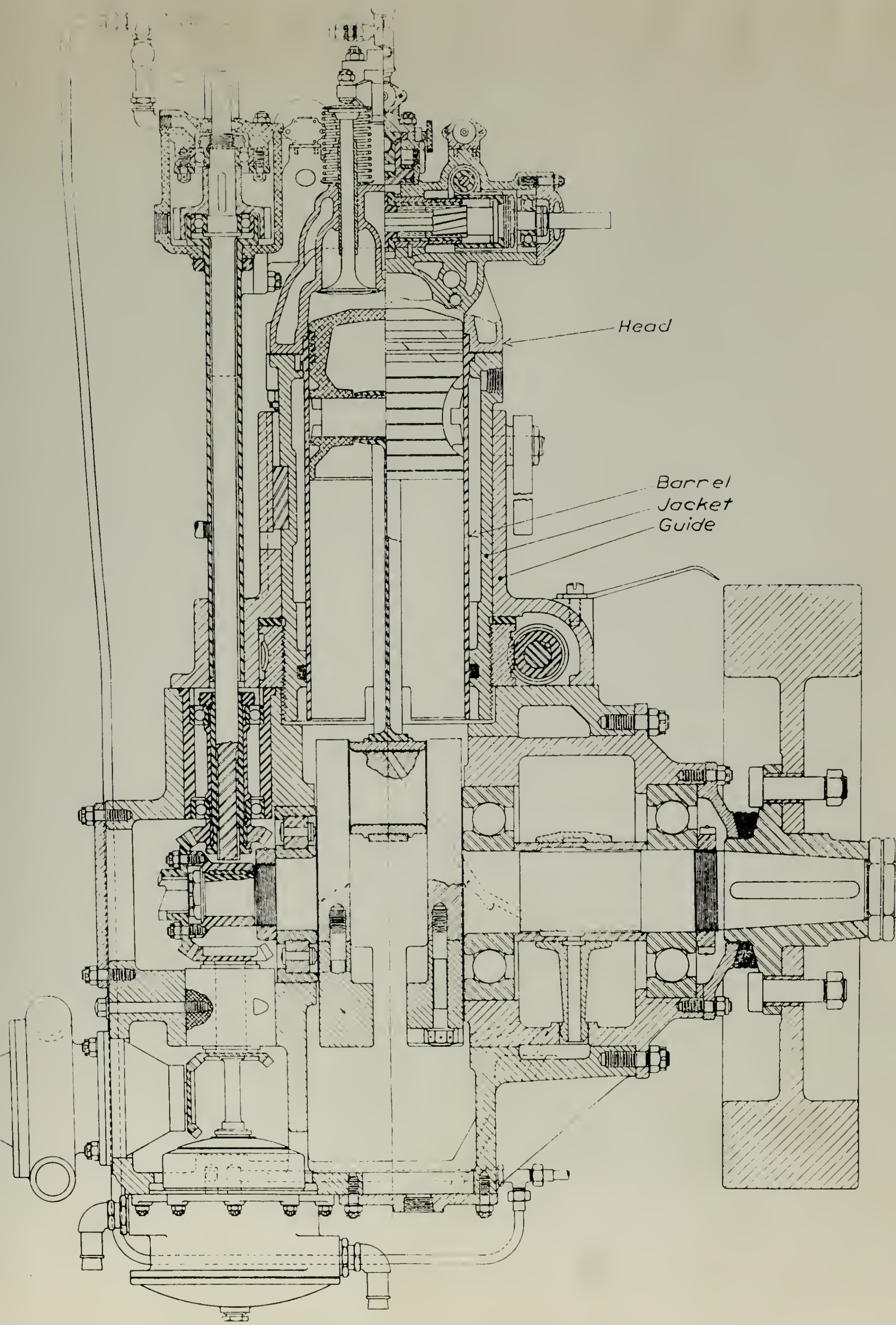


FIG. 2.—N. A. C. A. universal test engine—longitudinal cross section

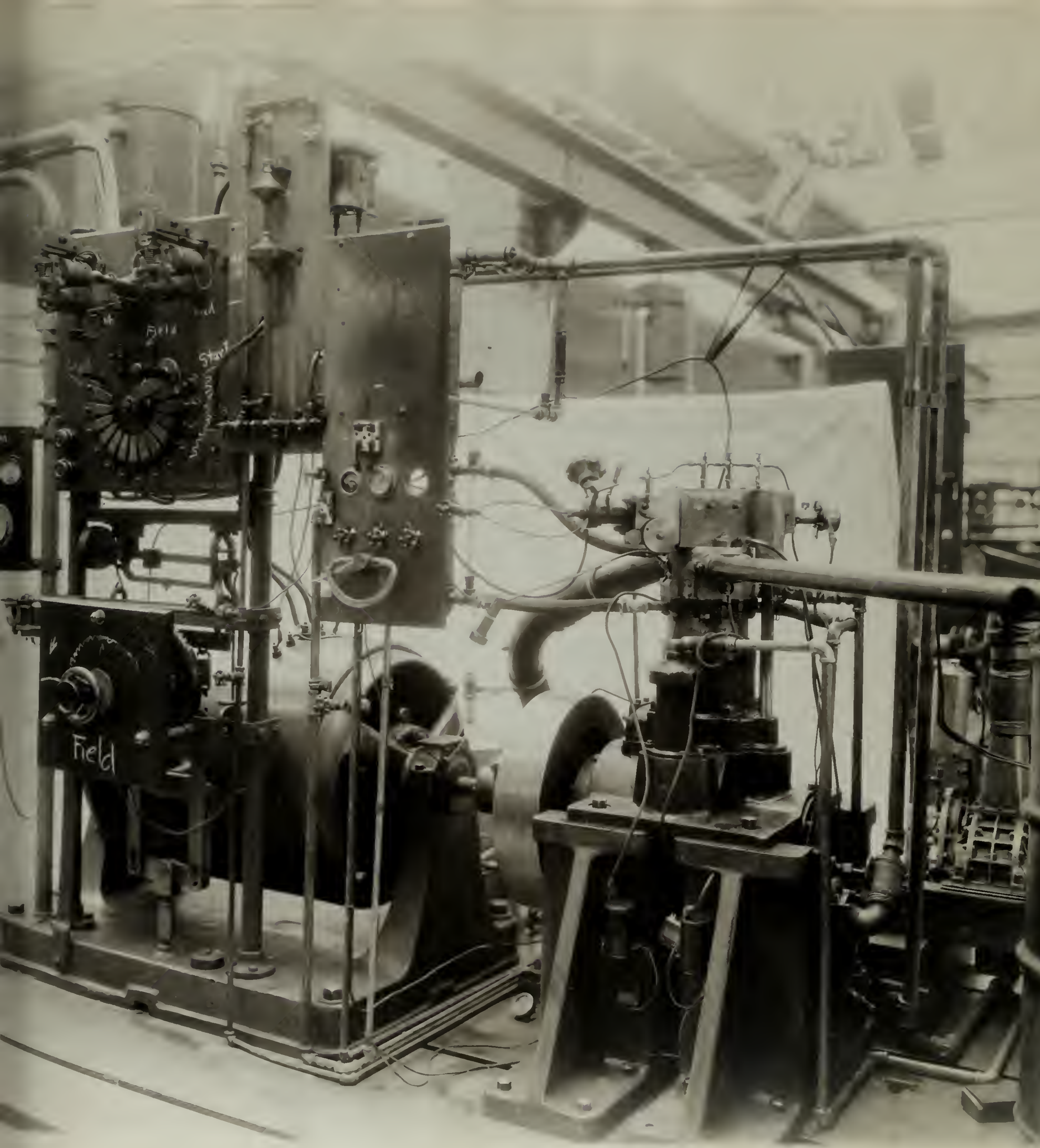


Figure 3. N.A.C.A. Engine, M.I.T. Head.

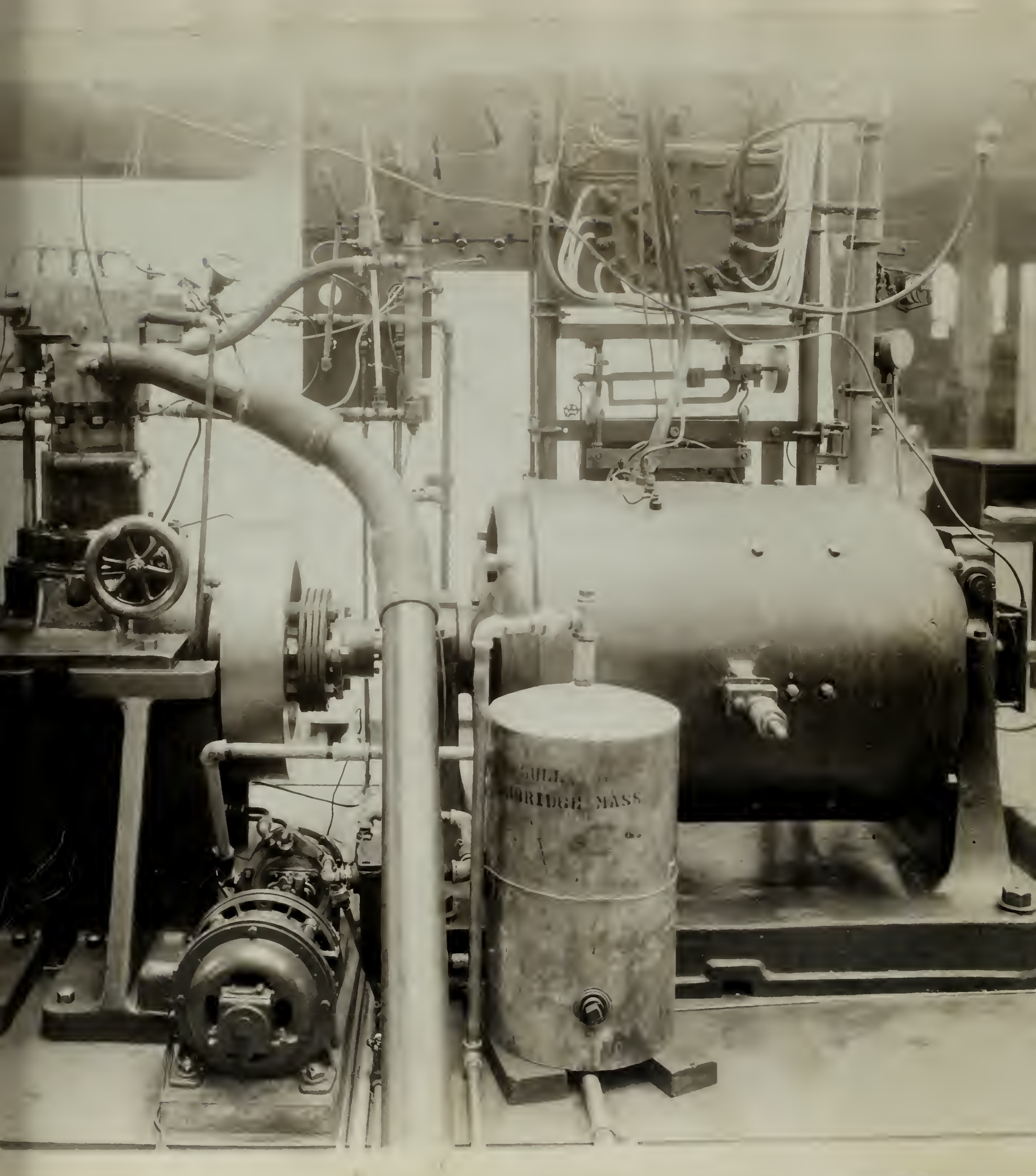


Figure 4. N.A.C.A. Engine M.I.T. Head.

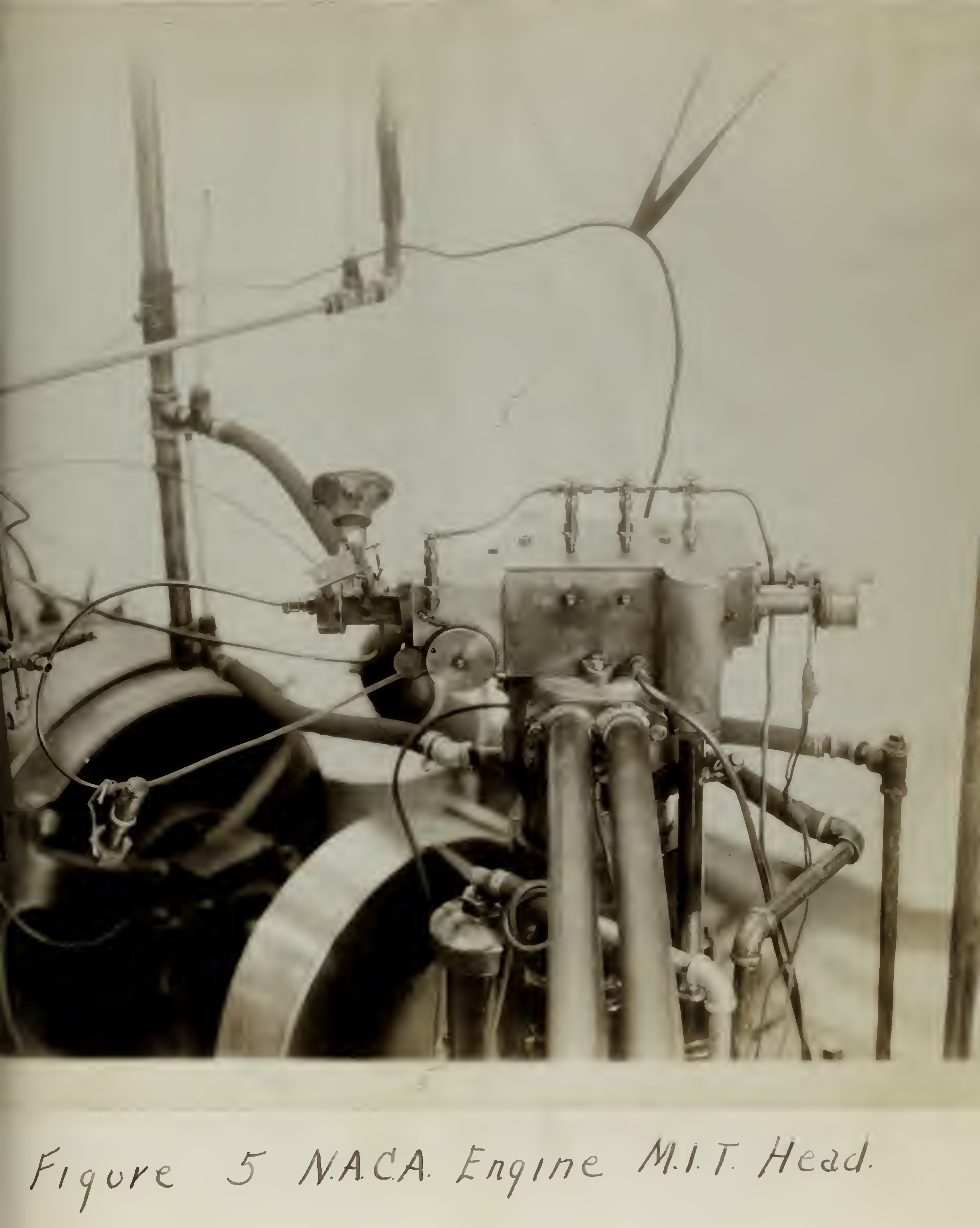


Figure 5 N.A.C.A. Engine M.I.T. Head.

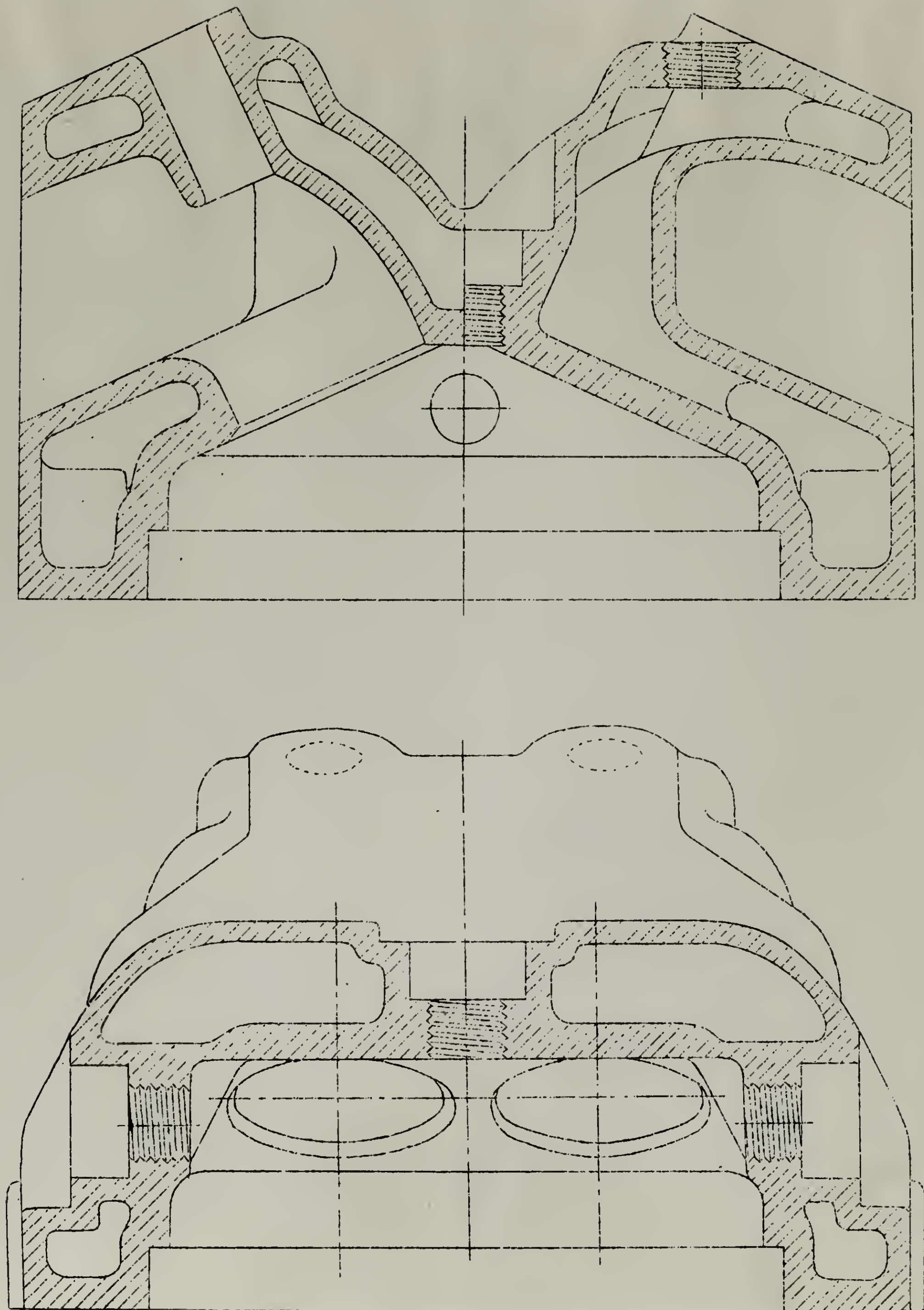


Figure 6. N.A.C.A. Head.

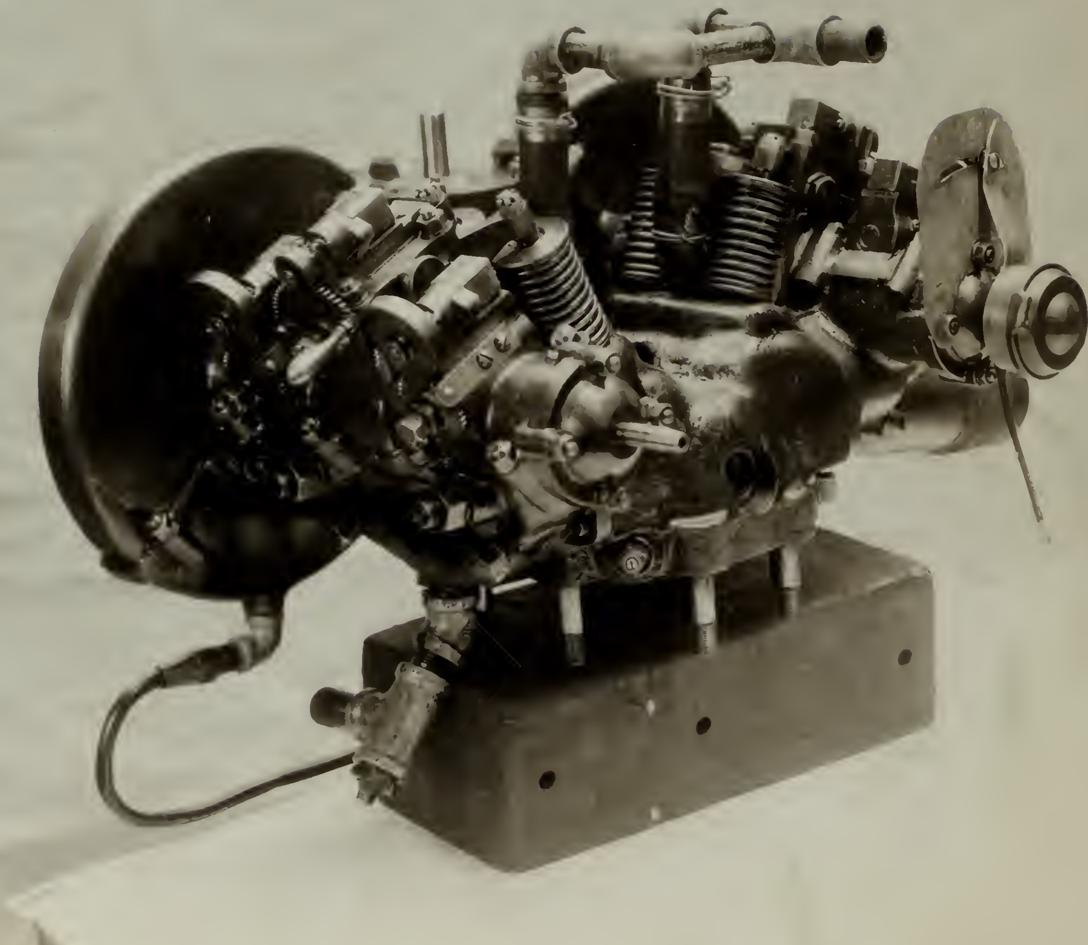


Figure 7. N.A.C.A. Head.

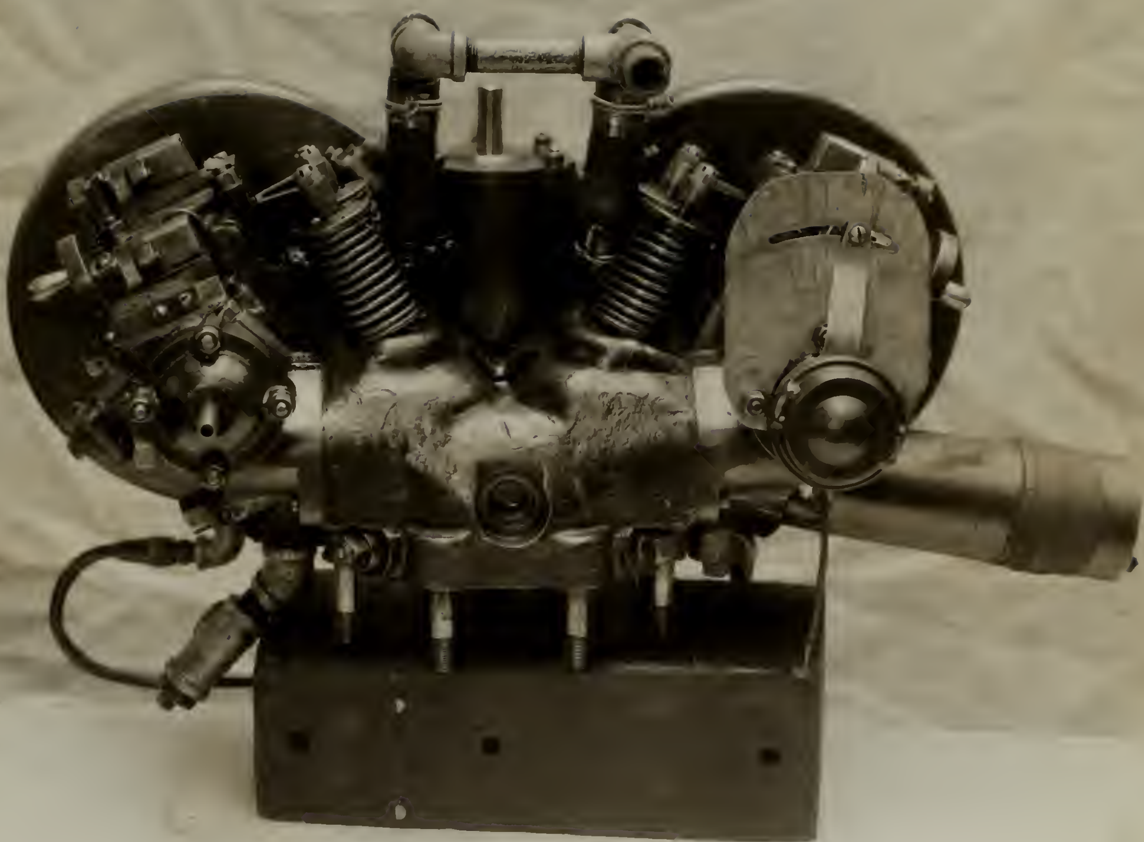


Figure 8. N.A.C.A. Head.

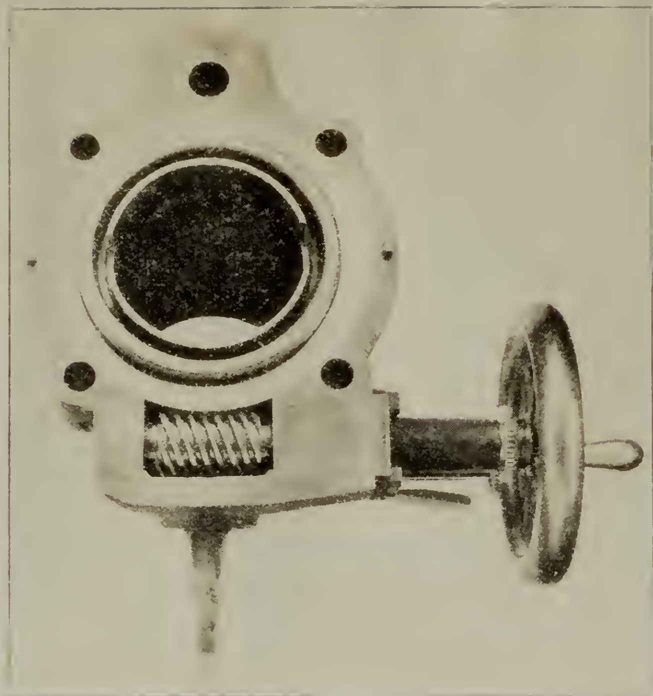


Figure 9. Compression Ratio Adjusting Mechanism.

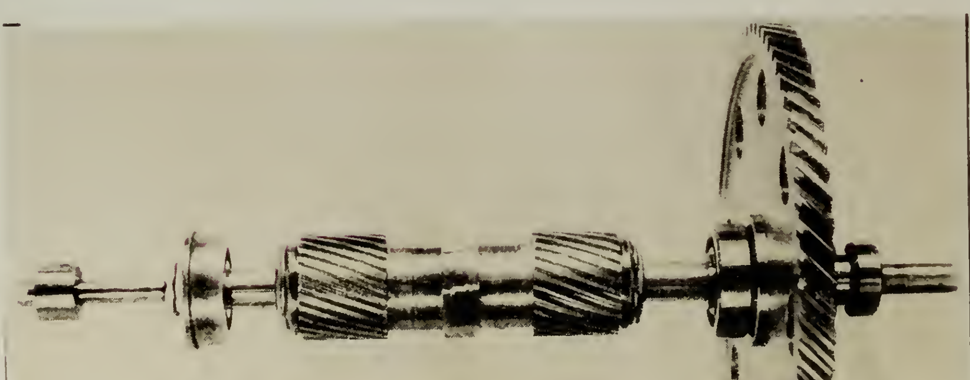


Figure 10. Camshaft Assembly. Parts Displaced Slightly

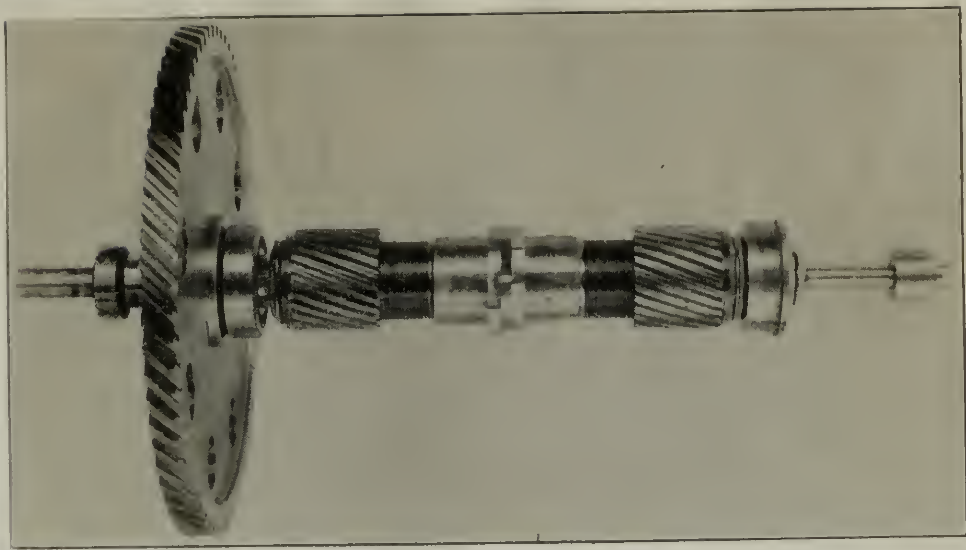


Figure 11. Camshaft Assembly. Parts fully displaced.

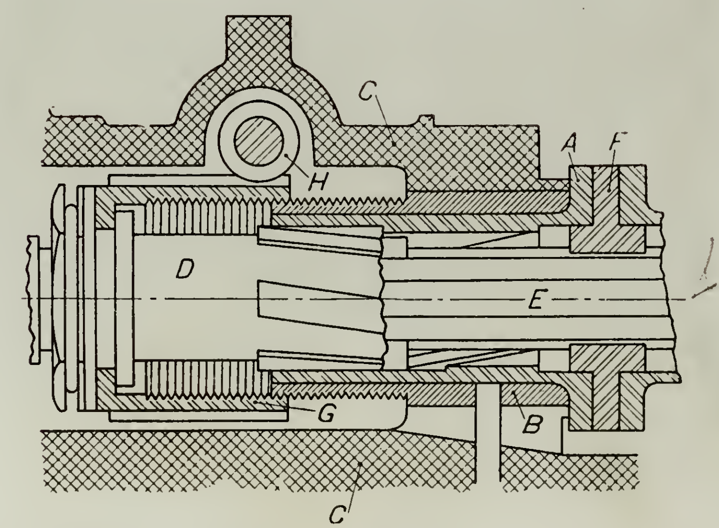


Figure 12 Cam Adjusting Mechanism.

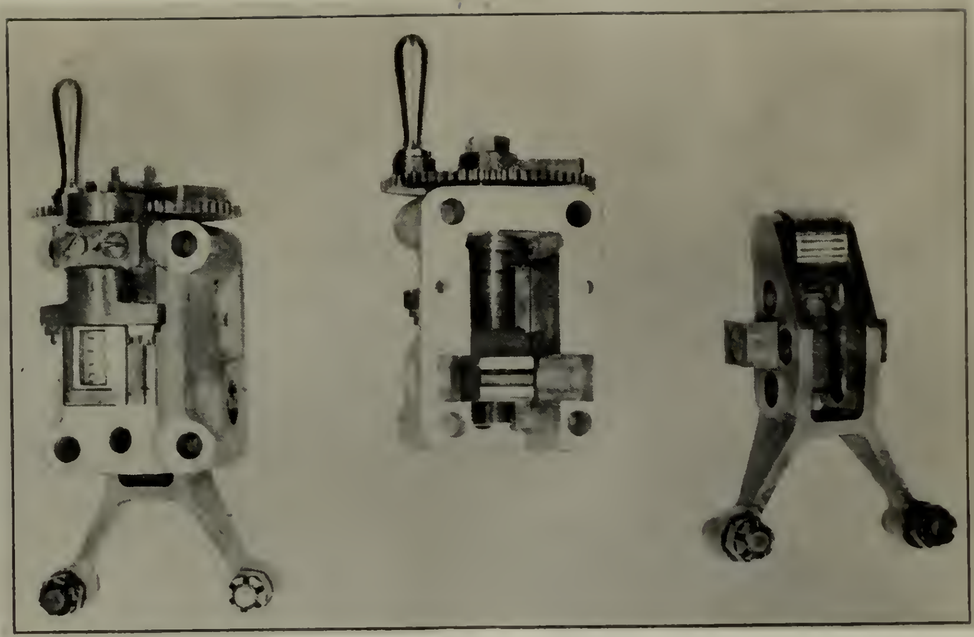


Figure 13. Rocker Arm and Housing.

Figure 14. Cam and Follower.

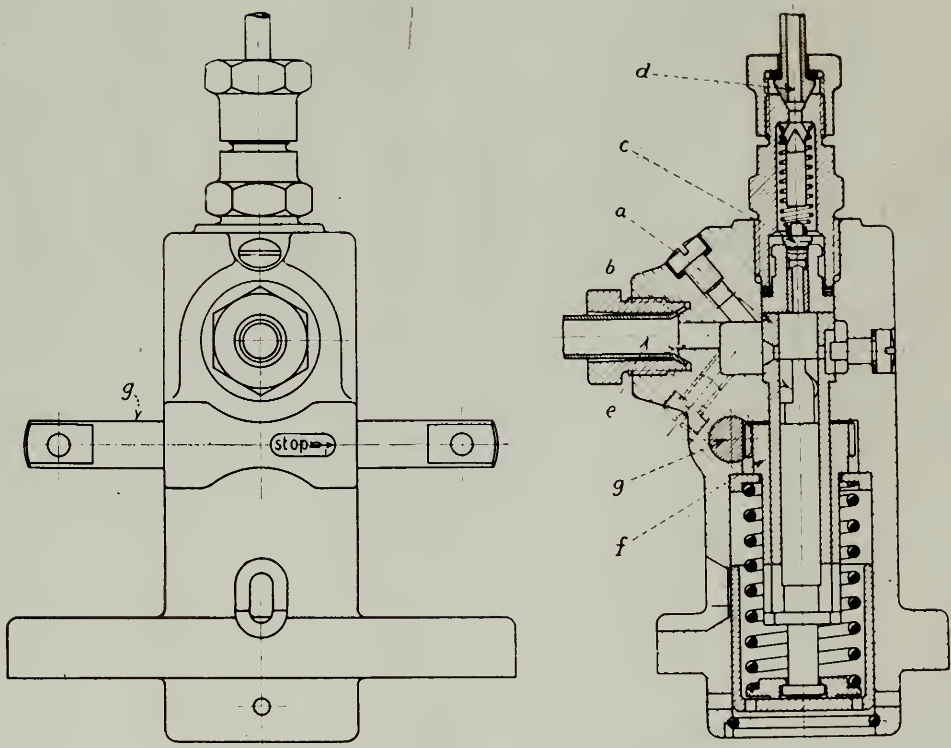


Figure 15 Bosch Pump.

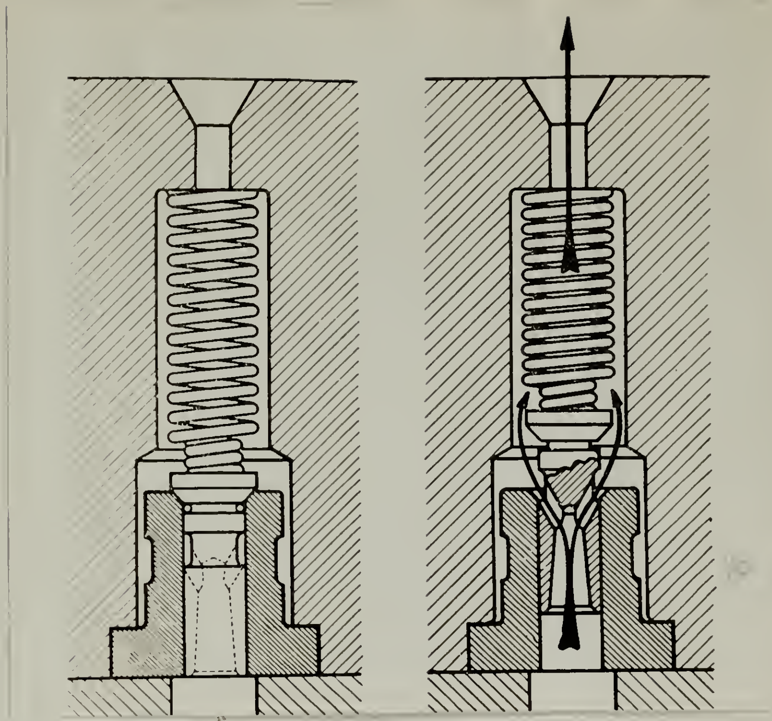


Figure 16. Pressure Relief Valve.

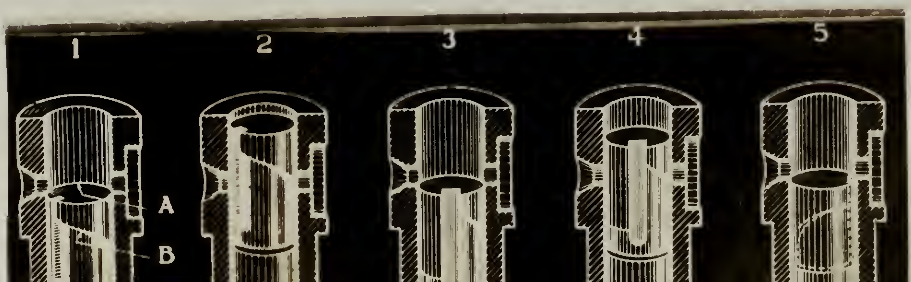


Figure 17. Barrel with Plunger.

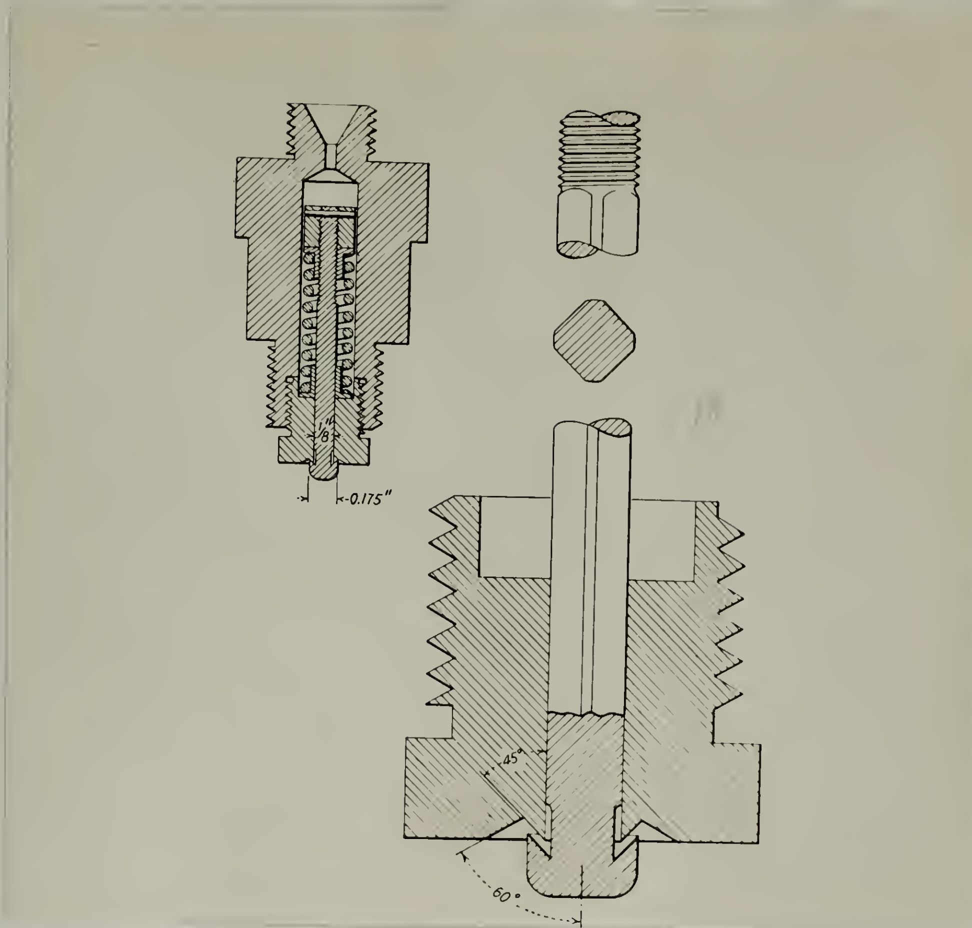


Figure 18 Injection Nozzle.

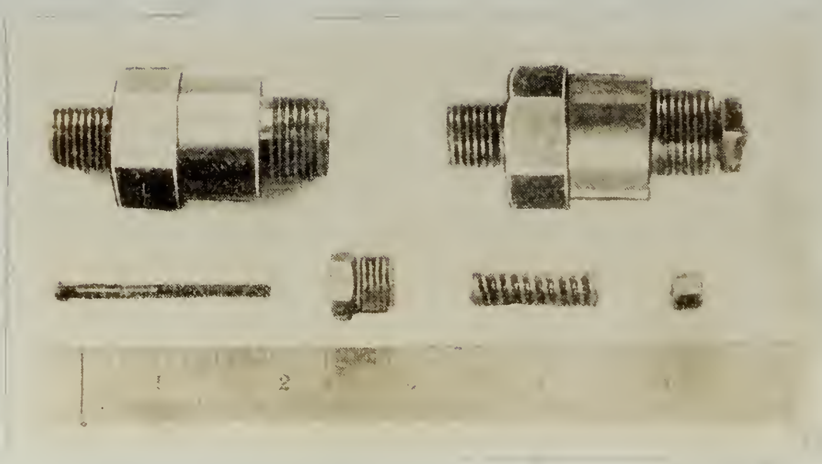


Figure 19. Injection Nozzle.

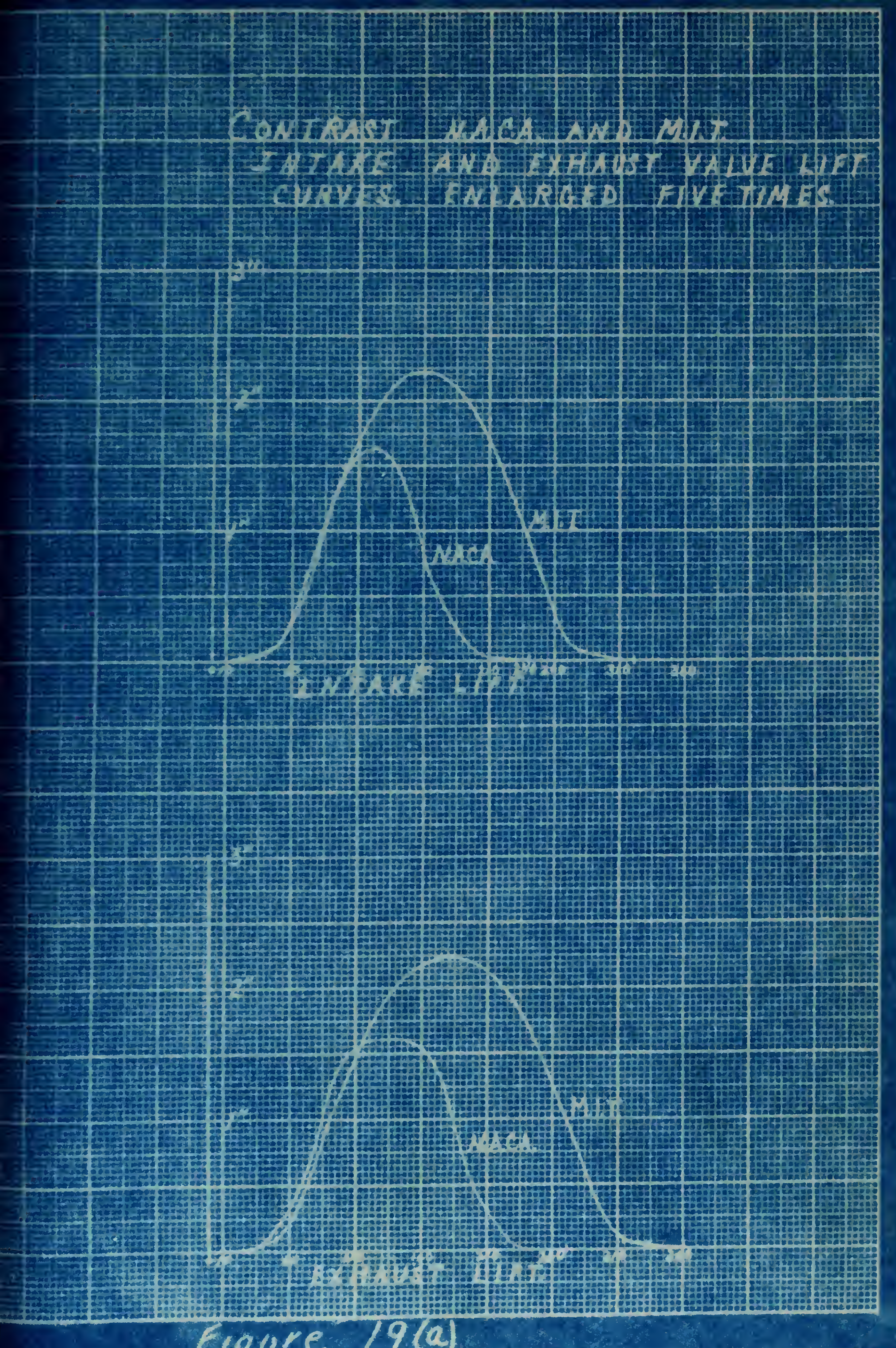

CONTRAST NACA AND M.I.T.
INTAKE AND EXHAUST VALVE LIFT
CURVES ENLARGED FIVE TIMES
M.I.T
NACA
INTAKE LIFT
M.I.T
NACA
EXHAUST LIFT

Figure 19(a)

Figure 20

Effect of Length of Intake Pipe on
Brake Mean Effective Pressure

Engine--C.A.E. [illegible] Head
Bore--Stroke--5 x 7
Displacement--137.4cu.in.
Compression Ratio--5
Engine R.P.M.--1600
Pump--Bosch Standard 7mm.
Nozzle--[illegible] (Low Penetration)
Fuel--[illegible]
S.G.--.[illegible] at 60°.
Injection--Cylinder
Cam--

Intake Opening--[illegible]
Intake Closing--[illegible]
Exhaust Opening--140
Exhaust Closing--35
Spark Adv.--25
Inj. Begin--B.T.C.--30
Fuel-Air--Best Power

Aero. Engine Laboratory

[illegible] March, 193[illegible].
Name-- Dixson; [illegible]; [illegible].

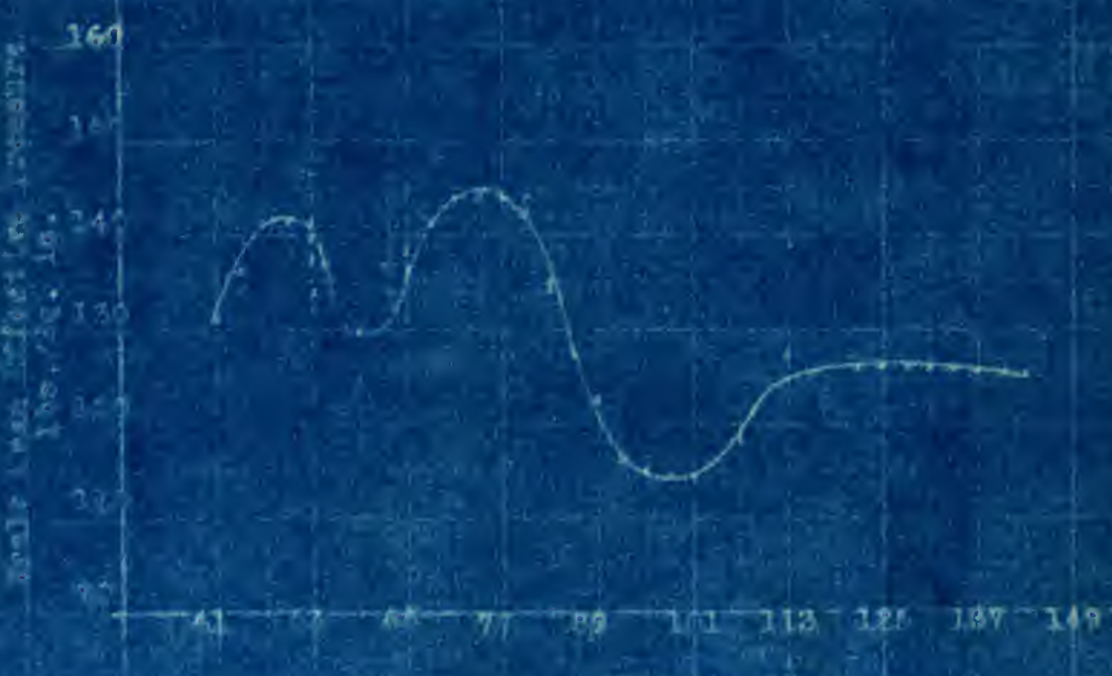


Figure [illegible]1

Effect of Length of Intake Line on Brake Mean Effective Pressure Varying Quantity of Fuel Injected

Engine--N.A.C.A. [illegible] Head	Intake Opening--[illegible]
Bore-Stroke-- [illegible] x 7	Intake Closing--[illegible]
Displacement-- [illegible].4 cu. in.	Exhaust Opening--[illegible]
Compression Ratio--[illegible]	Exhaust Closing--[illegible]
Engine R.P.M.--1600	Spark Adv.--[illegible]
Pump--Bosch Standard 1[illegible] mm.	Inj. Begin--[illegible]--[illegible]°
Nozzle--N.A.C.A. (Open Construction)	Fuel-Air--Variable
Fuel--[illegible]	
S.G.--.710 at 60°F.	
Injection--Cylinder	
Gas--	

Aero. Engine Laboratory

Research 193[illegible]
Made--[illegible]; [illegible]; [illegible]

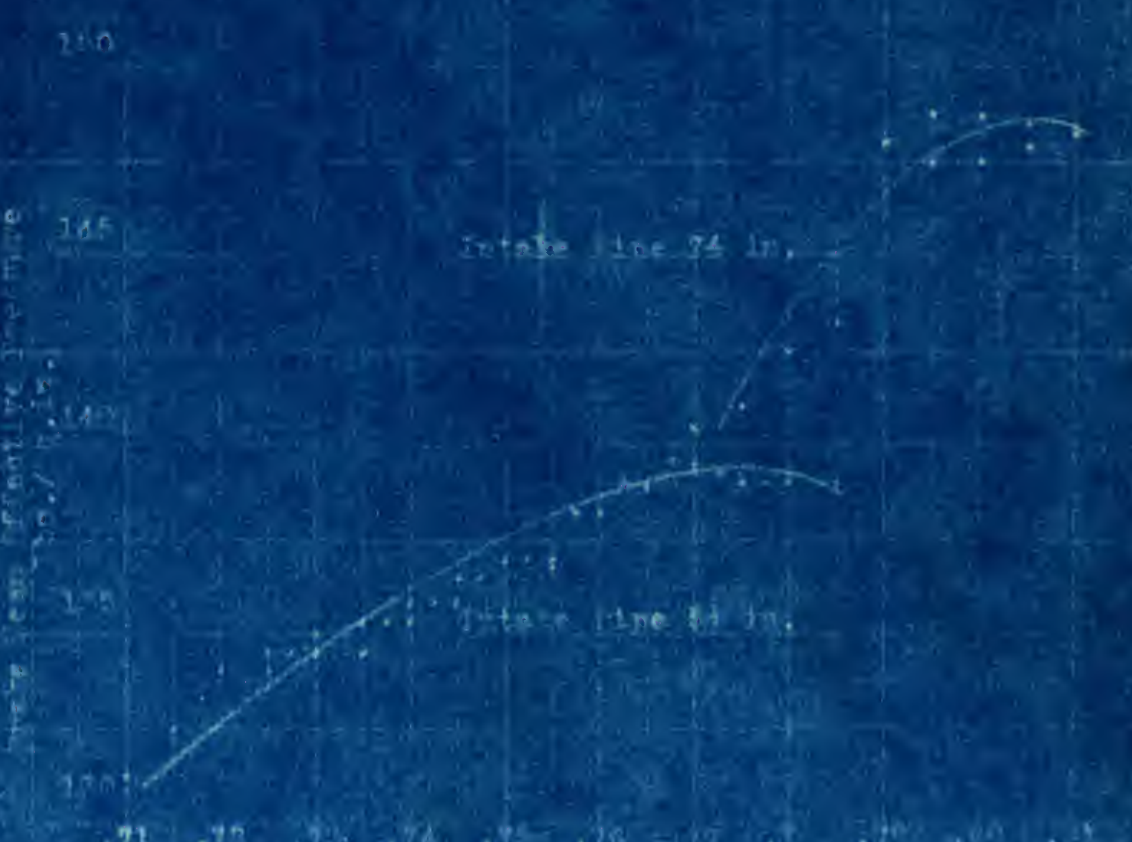


Figure 58

Effect of Varying Injection Timing on Brake Mean

Effective Pressure

Engine--N.A.C.A.(Kidd Head)
Bore-Stroke--5 x 7
Displacement--137.4 cu.in.
Compression Ratio--8
Engine R.P.M.--1500
Pump--Bosch Standard 10mm.
Nozzle--N.T.(low penetration)
Fuel--S.A.E.
S.G.--0.710 at 60°F.
Injection--Cylinder (3000#)
Gen.---

Intake Opening--15
Intake Closing--210
Exhaust Opening--130
Exhaust Closing--35
Spark Adv.-30°
Inj. Timing-Variable
Fuel-Air--Variable
Intake Pipe Lgth--72in.
Results Corr. to STD.

Aero. Engine Laboratory

17 March,1933
Name;--Dixton; Ellis; Rhoads.

Figure [illegible]

Effect of Varying Valve Timings on

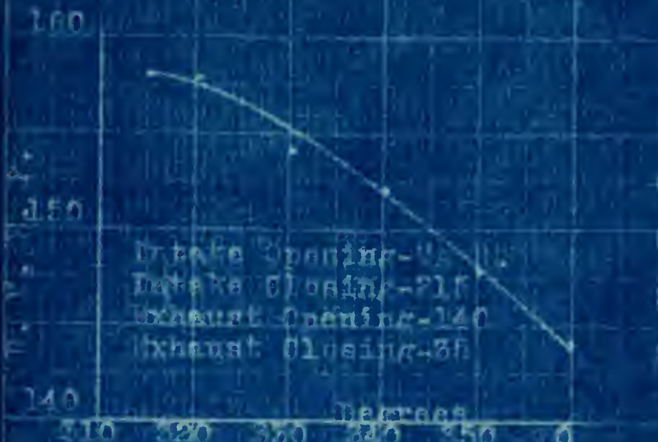
Brake Mean Effective Pressure

Engine--N.A.C.A. ([illegible] Head)
Bore-Stroke-- 5 x [illegible]
Displacement--13?.? cu.in.
Compression Ratio--[illegible]
Engine R.P.M.--1600
Pump--Bosch Standard 10mm.
Nozzle--[illegible] (Low Penetration)
Fuel--D.A.O.
S.G.--.71 at 60°F.
Injection--Cylinder(2000)
Cam--Medium

Intake Opening--
Intake Closing--
Exhaust Opening--
Exhaust Closing--
Spark Adv--22°
Inj.Begins--[illegible]
Fuel-Air--Best Power
Intake Pipe Lgth--74in.
Results--Corr.to STAND.

Aero. Engine Laboratory

16 March,193?.
Name--Dixton; Ellis; Rhoads.



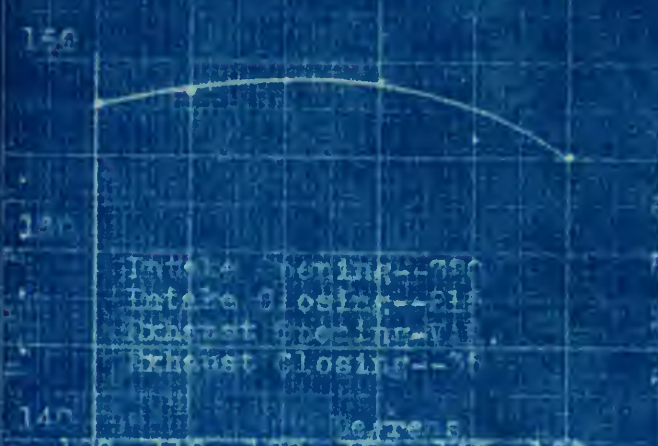


Figure 24

Effect of Varying Valve Timings on

Brake Mean Effective Pressure

Engine--[illegible] Head | Intake Opening--
Bore-Stroke-- [illegible] x 7 | Intake Closing--
Displacement-- 137.4 cu.in. | Exhaust Opening--
Compression ratio--[illegible] | Exhaust Closing--
Engine R.P.M.--1600 | Spark Adv.--25°
Pump--Bosch Standard [illegible]mm. | [illegible]
Nozzle--[illegible] Low Penetration | Fuel-Air--Best Power
Fuel--Safe-T-Esso | Intake Pipe [illegible]in.
S.G.--.865 at 60°F. | Results--Corr. to Stand.
Injection--Cylinder [illegible]
Cam--Medium

Aero. Engine Laboratory

[illegible] March, [illegible]
Name--[illegible]; [illegible]; Rhoads.

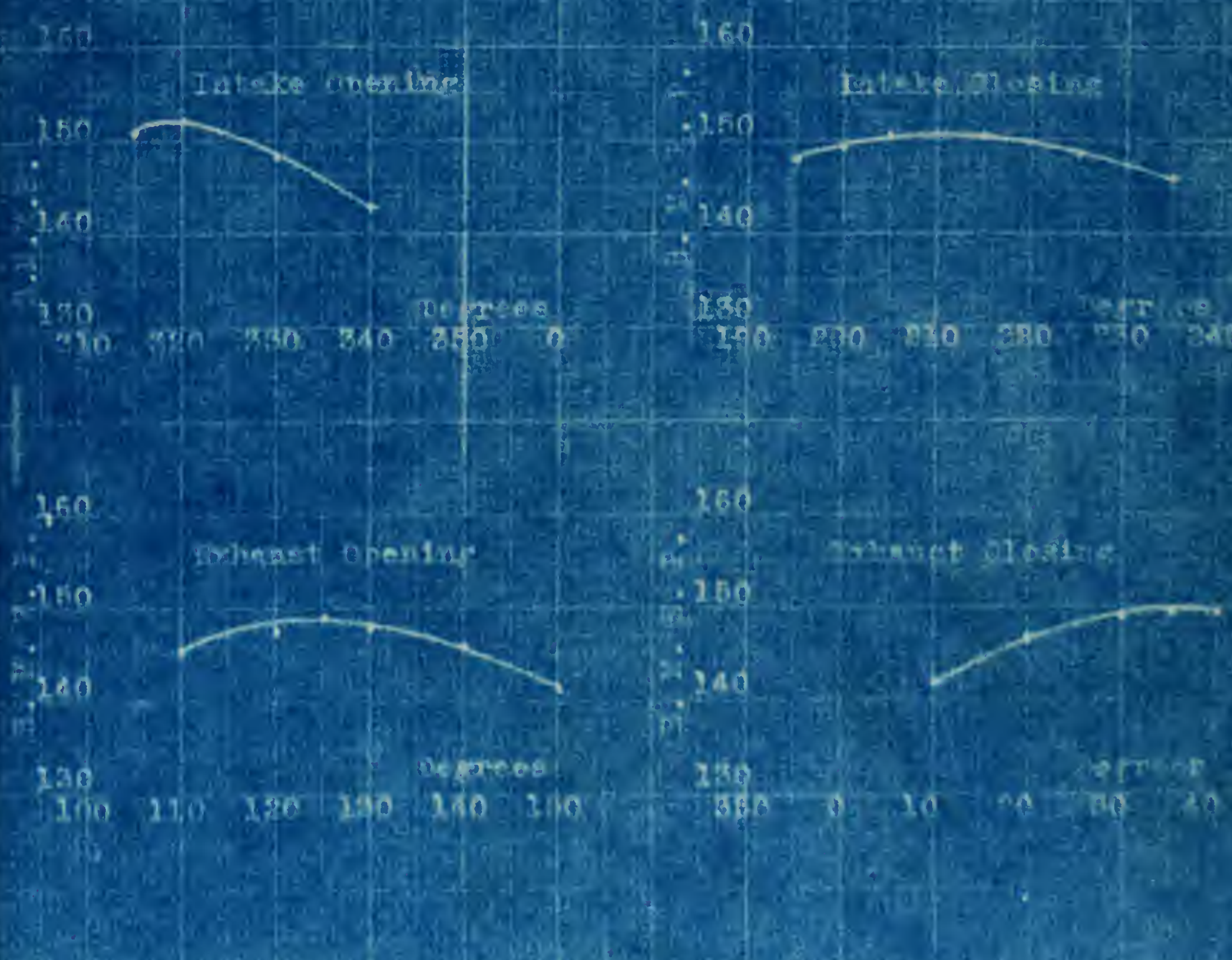


Figure 25

Power Output Vs. Fuel Flow at Various

Compression Ratios

Engine--N.A.C.A. (Pent Head)
Bore-Stroke--5 x 7
Displacement--137.4 cu.in.
Compression Ratio--6
Engine R.P.M.--1600
Pump--Bosch Standard 10mm.
Nozzle--N.A.C.A. (low penetration)
Fuel--Vic Gas - 20% Benzol
S.G.--.755 at 60°F.
Injection--Cylinder(3000#)
Cam--Bosch; High lift.

Intake Opening--30.5
Intake Closing--210
Exhaust Opening--140
Exhaust Closing--[illegible]
Spark Adv.--28.50
Inj. Begin--B.T.C.--30
Fuel-Air--Variable
Intake Pipe Length--[illegible]in.
Results Corr. to [illegible].

Aero. Engine Laboratory

27 April, 1933
Runs--Burton; Ellis; Rhoads.

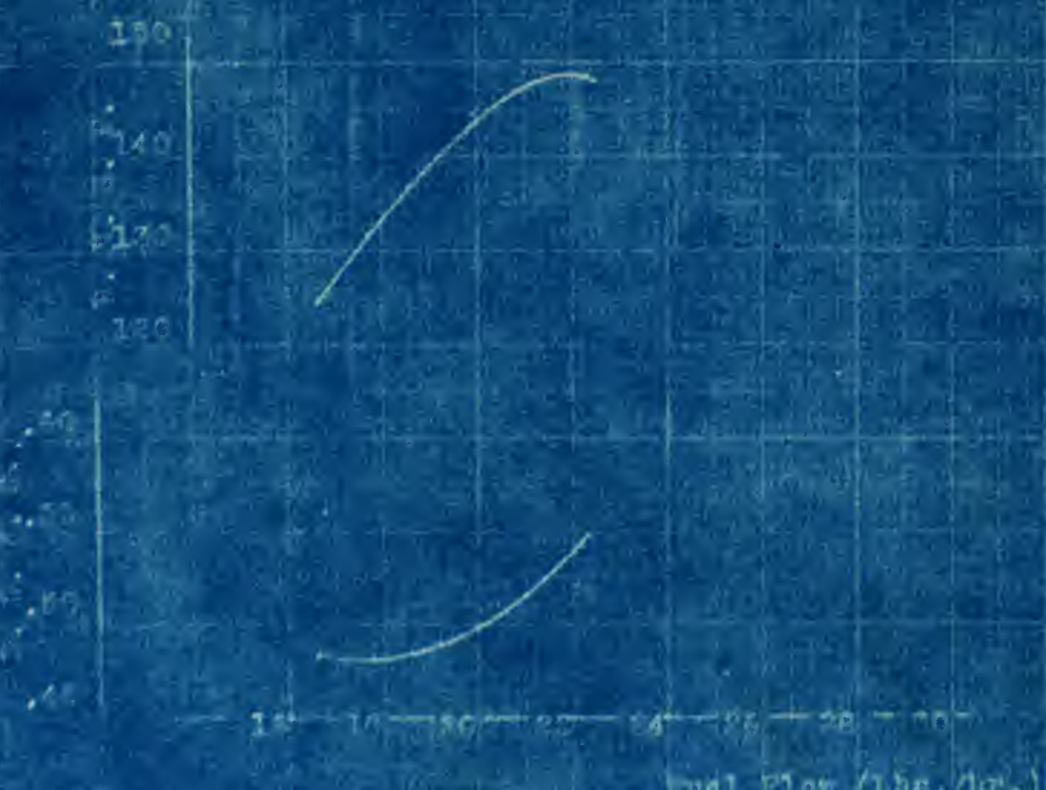


Figure 26

Power Output Vs Fuel Flow at Various

Compression Ratios

Engine--N.A.C.A. (MIT Head)
Bore-Stroke--5 x 7
Displacement--137.4 cu.in.
Compression Ratio--6
Engine R.P.M.--1600
Pump--Bosch Standard 10mm.
Nozzle--M.I.T.(Low Penetration)
Fuel--Safe-T-Esso
S.G.--.860 at 60°F.
Injection--Cylinder(2000#)
Cam--Bosch; High lift.

Intake Opening--21°
Intake Closing--21°
Exhaust Opening--140
Exhaust Closing--25
Spark Adv.--25.5°
Inj.Begin-B.T.C.--5°
Fuel-Air--Variable
Intake Pipe Lgth-56in.
Results Corr.to STAND.

Aero. Engine Laboratory

27 April,1933
Name--Pixton; Ellis; Rhoads.

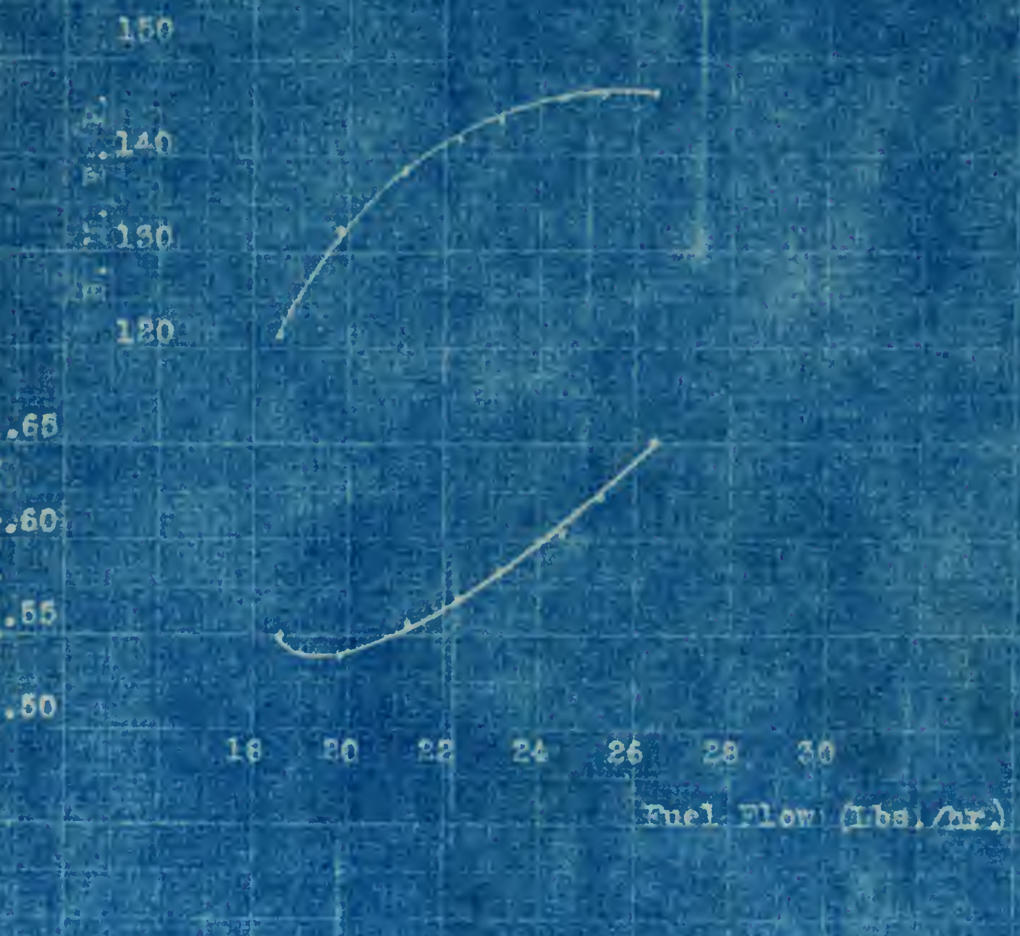


Figure 27

Power Output Vs Fuel Flow at Various

Compression Ratios

Engine-N.A.C.A. (MIT Head)
Bore-Stroke--5 x 7
Displacement--137.4 cu.in.
Compression Ratio--6
Engine R.P.M.--1500
Pump--Bosch Standard 10mm.
Nozzle--M.I.T.(low Penetration)
Fuel--Safe-T-Esso
S.G.--.865 at 60°F.
Injection--Cylinder(3000#)
Cam--Bosch; High lift.

Intake Opening--31 8
Intake Closing--21 0
Exhaust Opening--14 0
Exhaust Closing--35
Spark Adv.--2h. 50
Inj. Begin-A.T.C.--50
Fuel-Air--Variable
Intake Pipe lgth-56in.
Results Corr.to STAND.

Aero. Engine Laboratory

27 April,1933
Name--Pixton; Ellis; Rhoads.

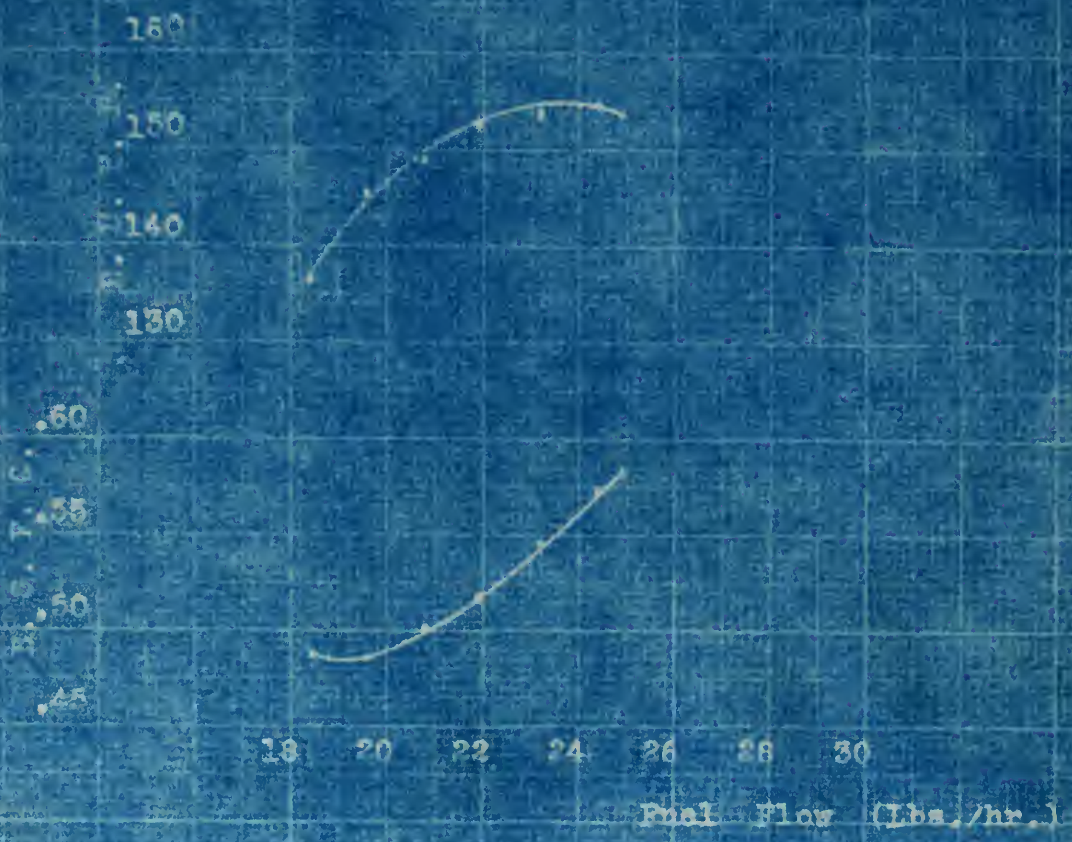


Figure 28

Power Output Vs Fuel Flow at Various

Compression Ratios

Engine-N.A.C.A. (MIT Head)
Bore-Stroke--5 x 7
Displacement--137.4 cu.in.
Compression Ratio--7
Engine R.P.M.--1600
Pump--Bosch Standard 10mm.
Nozzle--M.I.T.(low Penetration)
Fuel--Safe-T-Esso
S.G.--.865 at 60°F.
Injection--Cylinder(30004)
Cam--Bosch; High lift.

Intake Opening--315
Intake Closing--210
Exhaust Opening--140
Exhaust Closing--35
Spark Adv.--25.5°
Inj. Begin-A.T.C.--5°
Fuel-Air--Variable
Intake Pipe lgth-66in.
Results Corr.to STAND.

Aero. Engine Laboratory

27 April,1933
Name--Pixton; Ellis; Rhoads.

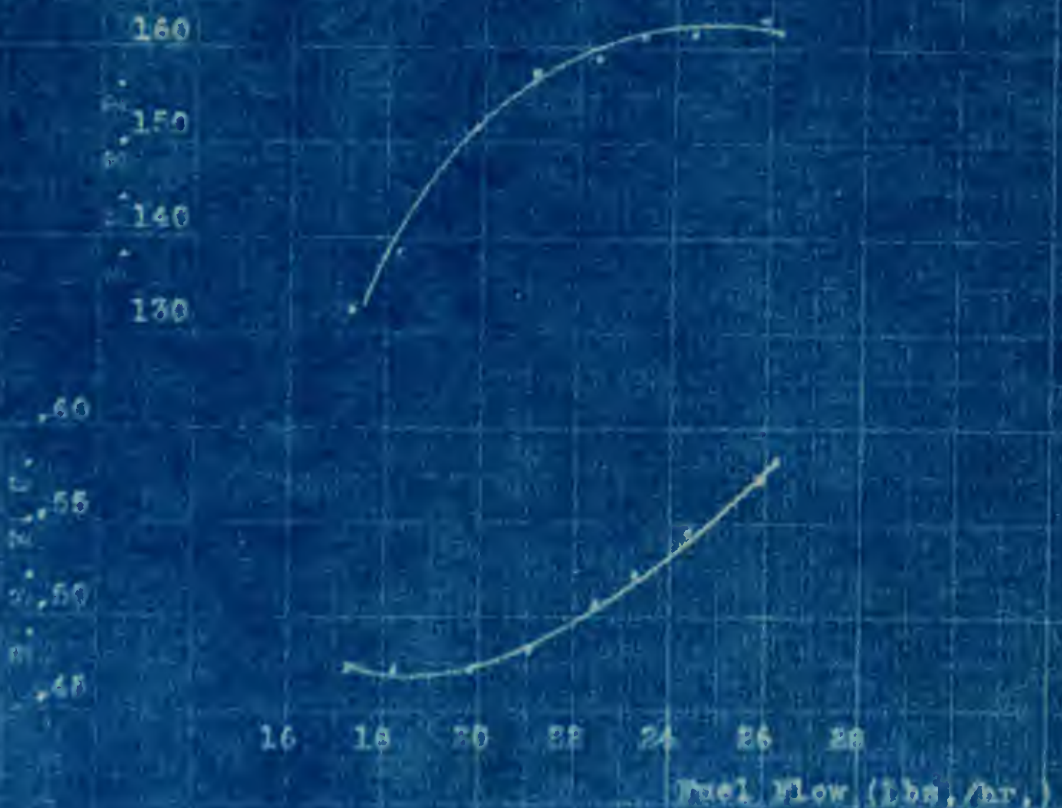


Figure [illegible]

Power Output vs Fuel Flow at Various Compression Ratios

Engine--[illegible] ([illegible] Head)
Bore-Stroke--[illegible] x 7
Displacement--137.4 cu.in.
Compression Ratio--7.[illegible]
Engine R.P.M.--1500
Pump--Bosch Standard 10mm.
Nozzle--[illegible] (Low Penetration)
Fuel--Safe-T-Fume
S.G.--.868 at 60°F.
Injection--Cylinder (4000#)
Cam--Bosch; High lift.

Intake Opening--[illegible]
Intake Closing--[illegible]
Exhaust Opening--[illegible]40
Exhaust Closing--[illegible]
Spray Ang.--2[illegible]
Inj. Begins [illegible]--5°
Fuel-Air--Variable
Intake Pipe [illegible]
Results Corr. to [illegible]

Aero. Engine Laboratory

28 April, 1933
Name--Pixton; Ellis; Rhoads.

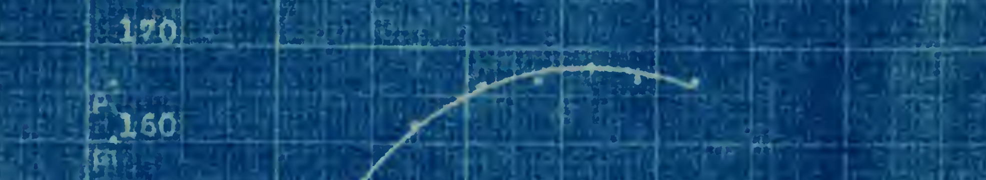


Figure 30

Power Output Vs Fuel Flow for Various

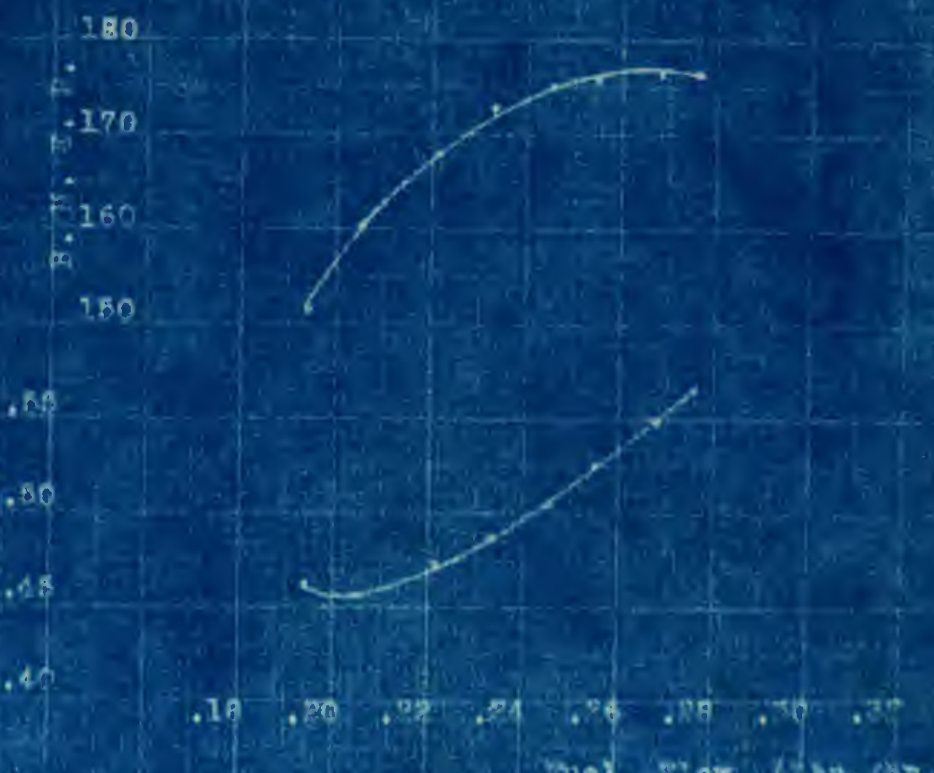
Compression Ratios

Engine--N.A.C.A.(MIT Head)
Bore-Stroke--5 x 7
Displacement--137.4 cu.in.
Compression ratio--8
Engine R.P.M.--1800
Pump--Bosch Standard 10mm.
Nozzle--[illegible] (Low Penetration)
Fuel--Safe-T-ess
S.G.--.862 at 60°F.
Injection--Cylinder (3000#)
Cam--Bosch: High lift

Intake Opening--15
Intake Closing--10
Exhaust Opening--140
Exhaust Closing--35
Spark Adv.--18°
Inj.Begin-A.T.C.--6°
Fuel-Air--Variable
Intake pipe lgth--55 in.
Results Corr.to [illegible]

Aero. Engine Laboratory

28 April, 1933
Name:-- Sixton; Ellis; Rhoads.



Effect of Compression Ratio on Brake Mean Effective
Pressure and Brake Specific Fuel Consumption

Engine [illegible] (N.P.S. Head)	Intake Opening [illegible]
[illegible]	Intake Closing [illegible]
Displacement [illegible] cu.in.	Exhaust Opening [illegible]
Compression Ratio—Variable	Exhaust Closing [illegible]
[illegible] R.P.M.—1600	Spark Adv. [illegible]
[illegible] 10mm.	[illegible]
[illegible]	[illegible]
Fuel—Safe-T-Gaso	[illegible]
[illegible]	[illegible]
Injection—Cylinder (3000#)	Results Corr. to [illegible]
[illegible]	

[illegible]

[illegible] April 1933
[illegible] Ellis; Rhoads.

Typical Analysis of Safe-T-Fuel.

Gravity	30
Octane No. at 300° Jacket Temperature	93
Color	+28
Flash	105°F.
Aniline Point	-28°F.(about)
Sulphur	0.019
% Hydrogen	10.7
% Carbon	89.3

<u>Saybolt Distillation</u>

Initial °F.	
5 %	312
10 %	324
20 %	330
50 %	348
90 %	372
95 %	380
Final	406

This material is produced by the hydrogenation of certain special stocks, followed by appropriate distillation to give a high flash point (105°F).

Fig. 32