ANALYSIS OF JAPANESE AIRCRAFT ENGINE

by

Dean Bedford, Jr. and Herbert LaGrange Wolfe

Submitted in Partial Fulfillment of the Requirements for
the Bachelor of Science Degree in Mechanical Engineering
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Cambridge, Mass.
16 January, 1947.

Professor Joseph S. Newell
Massachusetts Institute of Technology
Cambridge, Massachusetts.

Dear Sir:

A thesis entitled "Analysis of Japanese Aircraft Engine" is herewith submitted in partial fulfillment of the requirements for the degree of Bachelor of Science in Mechanical Engineering.

Respectfully,

ACKNOWLEDGMENT

The authors of this paper wish to thank Professor J. V. D. Eppes, their thesis advisor, for his valuable contribution of time and information which made the completion of this paper possible. They also desire to express their sincere appreciation of the aid and contributions rendered by Professor C. F. Taylor, Professor W. A. Leary, Mr. Cyrus H. Kano, and the staff of the Sloan Laboratory. Thanks are also due to Mr. Stephen W. Plimpton for the use of the facilities of the Plimton Photo Lab.

Contents

Purpose

It is the purpose of the authors of this paper to examine the design and constructional features of an experimental Japanese aircraft engine and to evaluate any new or unusual features found therein.

This subject has been selected since it lies in the major field of study of the investigators and provides an opportunity for them to put into practice the theory learned in classroom work.

To the best knowledge of the authors there has been no previous work in this country on this design of Japanese aircraft engine. For this reason no data has been available from other sources pertaining to this particular design.

Introduction

This paper is intended not as a detailed description of the design and construction of the engine under investigation, but rather as a preliminary survey of such a design, seeking to discover and evaluate any unusual features. In this manner it is hoped that future research on the subject may be facilitated, since the authors of this paper have attempted to indicate the new features contained in this engine design and to analyze them with a view toward the desirability of their further investigation.

Body of Thesis

The engine upon which these investigations were performed was delivered to the Sloan Laboratory by the United States Army, being one of a number of Japanese engines shipped to this country for study. Prior to arrival at the laboratory the engine had suffered considerable damage, either in shipment or at the hands of Japanese or American troops. This damage, which resulted in the destruction and loss of many of the control mechanisms, and the lack of suitable testing facilities, which precluded operating the engine, greatly interfered with the results obtained from the investigations.

The engine is laid out in a vertically opposed, twelve cylinder arrangement, six cylinders in a bank. Cooling is by means of air, ducted around the cylinders by means of a series of sheet metal baffles. Operation is on the four stroke cycle, with provision for forced scavenging of the cylinders with a large amount of air. It is this provision for forced scavenging which is the most interesting feature of the design. Induction, scavenging, and exhaust are controlled by single sleeve valves, much of the same pattern as has been used by the Bristol Aircraft Company for their series of air cooled radial engines. Fuel distribution is by means of a manifold injection system, incorporating an injection pump and nozzles discharging into a common manifold. Figure 1(a), showing the engine prior to any dissassembly, indicates the general layout of the engine and accessories.

Since the unusual features of this engine are contained in the power and induction systems this paper will concern itself primarily with these components, detailing other portions of the engine only as they affect these components.

The Cylinder:

The cylinder is of conventional design for the use of single sleeve valves. The head, of cast aluminum, has a long barrel with two rings, extending below the top of the sleeve, the rings effecting a seal for the gases of combustion. A separate cast aluminum spacer is provided to position the head properly in the sleeve and to provide additional cooling area. The cylinder head and spacer are shown in figure 2.

The cylinder barrel, (figure 3), is of cast bronze construction, the bronze providing a material with a good coefficient of heat transfer and sufficient strength to prevent pulling out of the cylinder head studs under the imposed loads. Ports for inlet, scavenging, and exhaust are cast integral with the barrel. Those sections of the barrel in which it was impossible to cast fins due to the proximity of ports are provided with aluminum finned sections screwed to the main casting.

The Valve Mechanism:

The sleeve is machined from steel, the inside surfaces being finished to 130 mm. bore to take the piston. Five ports are provided to control inlet, scavenging, and exhaust, four ports being arranged at the top of the sleeve and one near the bottom to control the lower exhaust valve. Figure 4 shows the sleeve.

The sleeve valve is driven by means of a countershaft carrying bevel gears which mesh with gears on short shafts which extend through the crankcase wall and carry on their ends the eccentric and ball and socket joint which actuates the valve. The countershaft is shown in figure 5.

The countershaft is driven off the crankshaft by a train of spur gears which drives a planetary reduction gear system at crankshaft speed. This planetary system in turn drives the countershaft at half crankshaft speed and provides, by means of a worm gear, a method of changing the valve timing while the engine is in operation. Individual timing of separate sleeves is accomplished by rotating the bevel gears on the countershaft, these being secured by keys which may be placed in any one of a number of keyways, thus giving precise adjustment. This individual adjustment necessitates the removal of the timing gear case. The crankshaft drive gear and the planetary system for the lower countershaft is shown in figure 6. Figure 7 shows the spur gear train which is operated by the crankshaft gear. This photograph, taken from the front of the timing gear case, shows in the upper left hand corner the planetary system for the countershaft, clearly illustrating the method of splining this to the countershaft. In the lower right hand corner is shown the stationary gear of the planetary system and the worm for its adjustment. The operating handle and locking plate for this control on the upper bank may be seen in the top center of the photograph.

The Piston and Connecting Rod:

The piston is of cast aluminum construction of conventional design, the top being perfectly flat. Seven rings are provided, four compression rings and one wiper ring at the top of the piston and two oil rings just above the skirt. Holes are drilled through the piston wall at this point to control lubrication.

The wrist pin, of machined steel construction with aluminum end caps, is of free floating design, the aluminum caps preventing the pin from scoring the sleeve. The piston and wrist pin assembly is illustrated in figure 8.

The connecting rod assembly is of conventional fork and blade design. Construction is of steel with bronze bearing inserts. A high degree of finish has been achieved in the manufacture of this assembly. Figure 9 illustrates the connecting rod assembly. At the bottom of this photograph is also shown the eccentric and ball and socket which actuates the sleeve valve.

The Crankshaft:

The crankshaft is of the design normally used in six cylinder engines with throws at 120 degrees. Due to the inadvisability of completely dismantling the engine no further investigation was made of the crankshaft and its components.

Supercharger and Induction System:

The supercharger and drive mechanism form part of the accessory drive case which is directly attached to the rear of the crankcase. The supercharger is driven off the crankshaft by means of a spur gear train at 8.5 times engine speed. The drive shaft runs concentrically with the impeller, the impeller shaft being supported on bearings on the drive shaft. Figure 10, showing the front of the blower case, illustrates the impeller and diffuser, together with the splined end of the drive shaft which engages with the crankshaft. The rear view of the casting,(figure 11), shows the supercharger gearing and, in the lower part of the casting, the drive for the magnetos (the magnetos were removed during disassembly prior to this photograph). The fittings on the side of the blower casting, which are apparent in the photograph, are evidently intended to bleed off air at supercharger pressure to control the fuel pump and to actuate cockpit instruments.

The impeller itself, (figure 12), is of aluminum construction of a high degree of workmanship and finish. The design is of the usual pattern for centrifugal compressors, the row of holes shown in the illustration being provided to equalize the pressure on front and rear of the impeller and prevent excessive thrust on the bearings. Lubrication for the impeller shaft is provided from the crankshaft lubrication system, the end of the impeller shaft being fitted with seal rings to prevent oil leakage. These rings may be seen in figure 10.

The diffuser is of machined steel, fastened to the blower casing by means of screws. As with the other parts of the supercharger the diffuser shows a high degree of finish.

Figure 13 shows the front of the rear casing of the engine, illustrating the drives for various accessories such as lubricating oil pumps, many of which were not mounted when the engine was delivered. The rear of this casing carries the mountings for the accessories such as the fuel pump and had been damaged before the engine was received at the laboratory.

Induction of the air to the cylinders is by means of parallel manifolds running alongside each bank of cylinders, which are connected to the scavenging ports of the cylinders by short connectors set at right angles to the main manifold. These connectors contain small throttles which enable the scavenging to be shut off at low inlet pressures, and are evidently operated in conjunction with the main throttle on the supercharger inlet, the exact connection being uncertain due to the damaged condition of the engine. The main manifolds are connected together at the forward end of the cylinder bank and thence are connected to the injection manifold. The injection manifolding system will be detailed in the section dealing with the fuel pump. The induction system may be clearly seen in figures 1(a) and 1(b).

<u>The Injection System:</u>

The injection system consists of a twelve cylinder injection pump, the pump control mechanism, the injection nozzles, and two manifolds, one for each bank of cylinders, into which the nozzles discharge.

The injection pump is of cylindrical pattern, the design largely following that of Friedrich Deckel, A. G., of Munich. The pump is driven at half engine speed, the individual pistons being actuated by means of a wobble plate. Control of the amount of the fuel delivered per stroke is accomplished in the same manner as in the Deckel pump by means of a rack which controls the cutoff point, thereby limiting the effective length of the stroke.

The pump control mechanism, which determines the fuel delivery of the pump under operating conditions, is actuated by air at manifold pressure. The power to move the pump rack is supplied by oil pressure, the oil possibly being supplied from the lubricating oil pump of the engine. A diagram of this oil system,(figure 14), is included in the appendix. Figure 15 shows the control assembled to the pump.

The control mechanism disassembled from the pump is shown in figure 16. The bellows case appears in the upper right hand corner of the assembly and the linkage between the pilot valve and the pump rack may be seen in the top center of the photograph. This linkage is principally to control the travel of the main actuating piston, which is connected to the lower part of the pump rack. The main actuating piston and piston rod may be clearly seen in the left center of figure 17 which illustrates the control partially disassembled. The method of attaching this piston to the pump rack is uncertain as this part of the mechanism was missing when the unit was received. The actuating piston appears also in the photograph of the lower section of the control mechanism housing (figure 18).

The upper half of the control housing, a bottom view of which is shown in figure 19, contains the pilot valve and its actuating system. This system consists of a brass bellows and a piston connected to the pilot valve through a lever system. This lever system appears in the right half of the illustration, the upper and longer lever being attached to the bellows and the lower lever to the piston. The arrangement is such that, from all indications, the bellows merely compensates for the temperature of the air bled from the manifold, the main actuation of the pilot valve being accomplished by air pressure impinging on the piston.

The control mechanism is also fitted with an emergency control device to prevent fuel from being entirely cut off should the actuating oil pressure fail. This consists of a spring loaded piston actuating a stop lever. Under normal operating conditions oil pressure acting on the piston forces the stop out of contact with the pilot valve - pump rack linkage. Upon release of the oil pressure the spring forces the stop into contact with the linkage and prevents the rack from being moved into a position of fuel cutoff. This device may be seen on the left side of the control mechanism in figure 15. Since the oil pressure was shut off when the photograph was made, the stop is in the up, or emergency operation, position.

A series of tests was run on the injection pump and control mechanism with a view towards determining the manifold pressures used by the engine. Air pressure and oil pressure to actuate the controls were supplied from outside sources, the pump being driven by an electric motor. This test set-up is illustrated in figure 20. Since the pump control is actuated by manifold pressure the maximum manifold pressure used could be determined by noting the pressure which produced maximum fuel delivery and a curve of fuel delivery versus manifold pressure could be constructed. Using these figures and assumed fuel-air ratios, the operating conditions of the engine could be more accurately estimated.

Due to the damaged condition in which the engine was received it was impossible to obtain results from this series of tests. The linkage which moved the pump rack was missing and some damage had been done to the control mechanism, with the result that it would not give consistent control. Attempts to rig a substitute operating linkage failed and it was deemed impractical to continue this line of investigation further.

However, some runs were made with the control mechanism removed and a micrometer used to position the pump rack. From these runs a curve of pump delivery versus rack setting was obtained, which is shown in figure 21. Assuming the engine operated at 2400 rpm., the maximum pump delivery, using the figures for air consumption obtained from the supercharger calculations, gives a fuel-air ratio of .146. This reasonable figure serves to verify the results obtained for the supercharger performance.

Fuel from the injection pump is delivered through high pressure lines to the injection nozzles which are positioned in a common inlet manifold for each bank of cylinders. Air is introduced into this manifold at the forward end, directly from the scavenging manifold. A throttle valve is placed in the manifold at this point so that the pressure and amount of air delivered may be controlled. This was obviously introduced in the design for experimental purposes. This manifolding system may be seen on the assembled engine in figure 1(a) and disassembled in figure 1(c).

Since the injection manifolds are located on the opposite side of the engine from the cylinder inlet ports, the inlet pipes run from the manifolds through the crankcase and thence upwards to their connection with the inlet ports. These pipes form an intergral part of the crankcase, with the exception of the upper portion, running from the base of the cylinder barrel to the inlet ports, which is a separate section of tubing. Seven inlet pipes are provided, five of which are divided at their upper extremities to connect with two adjacent cylinders, and two, those located at the front and rear of the cylinder bank, which serve only one cylinder.

Six injection nozzles of standard pattern are provided in each injection manifold. These are located directly opposite the manifold openings of the first six inlet pipes. Provision had been made for the location of an additional nozzle to serve the last inlet pipe in the bank. Evidently this additional nozzle opening was provided to allow the placement of the six nozzles opposite the forward six inlet pipes or the after six, whichever gave the best performance.

Discussion of Results

From the foregoing description the experimental nature of this design is immediately apparent. Basically, the design of the engine appears to have been carefully considered but little attention has been paid to the requirements of maintenance. Certain assemblies, such as the lubricating oil pump, which require frequent removal for service or replacement, have been located with little regard for such operations. Such details would have to be considerably modified prior to large scale production if the design were to compare favorably with current domestic products.

This design contains two features which are of major interest; the high degree of forced scavenging used on a four stroke cycle and the unusual method of fuel injection. The former will be discussed with a view towards determining the reason for the use of such a system and whether the additional power and complication required thereby was justified. The latter will be considered in such a way so as to determine its effectiveness over that of more conventional systems of fuel distribution.

Several reasons have become apparent for the application of such a method of forced scavenging to a four stroke engine. The primary result of the application of such a system would be to completely remove residual gases from the cylinder. This would result in an increase in volumetric efficiency and a consequent raising of the maximum power which could be obtained from a given throttle setting. A secondary result of such scavenging would be to cool the cylinder walls, thus preventing the formation of localized hot spots. This would remove one of the causes of detonation and permit the use of lower fuel-air ratios and fuels of poorer anti-detonant properties than are normally required by aircraft engines of this capacity.

To achieve the maximum benefit of the forced scavenging as regards the removal of residual gas the placement and timing of the valves must be such as to insure a free and unobstructed flow of exhaust gases and scavenging air during the initial period of scavenging. In this design a free flow of gases has been achieved by placing an exhaust port immediately above the position occupied by the piston at the bottom of the stroke. This results in a sensibly uniflow movement of the exhaust gases during the blowdown period. Reference to the valve timing diagram, (figure 22), will reveal that this port is opened considerably in advance of the exhaust ports placed in the more conventional position at the top of the cylinder, thus permitting the uniflow movement of the gases to continue undisturbed during the period of high cylinder pressure.

After the initial period of blowdown the cylinder must next be cleansed of those residual gases which did not escape in the initial period. In this design such has been accomplished by next opening the upper exhaust ports, followed shortly by the opening of the intake side scavenging port. This results in a flow of scavenging air across the upper part of the cylinder, removing the residual gas which is normally entrapped in the compression space. It will be noted that the bottom exhaust port remains open for a time after the scavenging port is opened, thus permitting the scavenging air to accelerate the removal of gases from the lower half of the cylinder. Piston movement during this period is relatively small, hence the major part of the removal of exhaust gases is accomplished in this interval by the blowdown process and the scavenging air rather than by movement of the piston.

As the piston moves upward on the exhaust stroke the top ring closes the bottom exhaust port, the upper exhaust ports remaining open. This creates a flow of scavenging air directly across the top of the cylinder, removing the exhaust gases forced up ahead of the moving piston. When the piston has completed approximately half the stroke the exhaust side scavenging port is opened, introducing an additional flow of scavenging air and creating a turbulence which tends to sweep the remaining residual gas from any dead spaces on the oposite, or inlet, side of the cylinder.

In view of the large scavenging air flow existing when both inlet and exhaust side scavenging and upper exhaust ports are open it would be inadvisable to open the intake ports until relatively late in the cycle due to the excessive losses of fresh charge which would occur. It will be seen from the valve timing diagram that such a late opening has been incorporated in this design.

During the downward movement of the piston on the inlet stroke the scavenging ports remain open, thus the cylinder is being supplied with both pure air and fuel-air mixture. The location of the ports is such that in all probability sufficient turbulence exists in the cylinder to give good mixing of the fuel-air mixture with the scavenging air. Since the major part of the air is supplied by the scavenging system it is reasonable to assume that a very rich mixture is supplied to the inlet ports in order to maintain a combustible mixture in the cylinder. As may be seen in the valve timing diagram the scavenging ports close completely at approximately the same time. This leaves 31 degrees during which the inlet ports deliver mixture with no additional air from the scavenging ports. As has been noted previously this mixture supplied by the inlet ports is necessarily very rich. This gives rise to the probability that some stratification of the charge in the cylinder is obtained. Such stratification would be desirable since a richer, and more easily ignited, mixture would exist at the top of the combustion chamber near the spark, permitting the use of a leaner overall fuel-air ratio. Greater economy of operation would therefore result.

From the foregoing discussion it may been seen that the secondary result of the application of such a forced scavenging system, namely the cooling of the cylinder walls, has been achieved by the use of such a large amount of scavenging. As has been noted previously this would eliminate one of the causes of detonation, thus permitting the use of lower fuel-air ratios and poorer grade fuel, again resulting in better economy of operation.

The provision for the delivery of such a large amount of air to the engine for scavenging purposes necessitates a supercharger of greater size than that which would be required by the same engine without the scavenging feature. Applying empirical formulae, (see the calculations in Appendix IV and the plot of Supercharger Pressure Ratio vs. Engine Speed, figure 23), to the data obtained from the supercharger it has been found that a blower of more than twice the capacity required without scavenging is used in this design. Thus, at an engine speed of 2400 rpm., it was calculated that the supercharger delivers 275 pounds of air per minute at a pressure ratio of 1.89. At this speed, assuming a volumetric efficiency of 100%, the engine requires 108.1 pounds of air per minute, leaving 166.9 pounds of air per minute to be used for scavenging purposes. Converting the above into horsepower, the blower uses a total of 289 HP., 114 HP. of which is required to supply air to the engine and 175 HP. to provide the additional air for scavenging.

The second unusual feature of this engine design is the fuel injection and the inlet manifold system. As was noted previously the injection pump and nozzles are of conventional design and thus warrant no consideration here. However, the inlet manifold system is of unique design and it is this feature which will next be discussed.

As was mentioned before, each inlet pipe supplies two cylinders, with the exception of the two pipes at the ends of the bank, which supply only one apiece. Each of these pipes, excepting the rearmost pipe on the bank, has a nozzle located directly in line with its opening into the injection manifold. Thus, since the individual injection nozzles are timed to the inlet period of only one cylinder, only that cylinder to which it is timed will receive fuel directly from a nozzle. The adjacent cylinder, served by the same inlet pipe, will receive through that inlet pipe during its inlet period the mixture existing in the common injection manifold at that time, a mixture necessarily much leaner than that supplied directly by a nozzle. This cylinder, however, receives directly from the adjacent nozzle, this system being continued throughout the cylinder bank.

In light of the short period during which the inlet ports are open it will be seen that a relatively large port area is required. The achievement of this port area, keeping the dimensions of the individual ports within reasonable limits, necessitates the provision of two inlet ports. Possibly the manifolding system used was the best design available to fulfill these conditions without overcomplication.

Conclusions

The investigators have drawn several conclusions regarding the efficacy and justification of the unusual features of this engine design.

From considerations of power alone, it has been found, utilizing the figures for required supercharger power and an estimate of the performance of the engine, (see calculations, Appendix IV), that an increase in volumetric efficiency of the engine of approximately 14% is required to justify the use of such scavenging. As noted, these figures are based on the assumption that the addition of scavenging increases the volumetric efficiency to 100%. However, considering the large amount of scavenging used, it is entirely possible that the volumetric efficiency may be increased to near the theoretical maximum of 118%, (r/r-1), by the removal of residual gas from the compression space. Based on these figures the theoretical increase required to justify the scavenging is calculated to be 13%, while the actual increase over operation without scavenging will greatly exceed this figure, possibly being as high as a 28% increase, (see calculations, Appendix IV). Throughout this discussion negligible pressure losses across the valves have been assumed, a reasonable assumption since intake is through four large ports and high flow coefficients are normally obtained with the use of sleeve valves. However, to this consideration of power requirements the benefits derived from the cooling effects of the scavenging must be added. Taking into account the large amount of scavenging used, the investigators have concluded that such a system is justified, giving at least the required increase of volumetric efficiency in addition to increasing the internal cooling by an appreciable amount.

The principal fault of the manifolding would seem to be the provision for one of the cylinder inlet pipes taking mixture from the manifold, rather than the mixture supplied by the nozzle. Due to the length of the manifold and the location of the nozzles it may be assumed that the fuel-air ratio existing in the manifold will vary considerably, This will introduce an appreciable variation in the fuel-air ratio received by the various cylinders. Therefore it is difficult to justify the use of such a manifold design and in all probability a system of individual inlet manifolds, such as that used in the R. A. E. - Hobson injection system, would give more satisfactory distribution of the fuel to the cylinders.

Suggestions for Future Investigation

In the opinion of the investigators the results of this paper have shown the need for further research on the forced scavenging system. This research would best take the form of an actual operation of such a scavenging system, utilizing a single cylinder removed from the engine mounted on a suitably constructed crankcase. In this manner verification of the analysis contained in this paper could be made and quantitative data obtained for a wide range of operating conditions.

A metallurgical analysis might also be made on the operating parts of the engine, special attention being paid to the composition, mechanical properties, and heat transfer coefficients of the bronze used for the cylinder barrels.

APPENDIX I

Engine Data

Name: Unknown (Japanese)

Type: Vertically opposed in-line, air cooled, manifold injection, four stroke, forced scavenging, sleeve valve.

No. of Cylinders: 12, in two banks of 6 each.

Bore: 5.12 inches (130 mm.).

Stroke: 5.51 inches, (140 mm.).

Displacement: 1365 cubic inches (22.4 liters).

Compression Ratio: 6.5: 1

Propeller Reduction Gear Ratio: 1.95: 1

Estimated BHP: 1000 at 2400 rpm.

Direction of Rotation: Crankshaft - Counterclockwise.
(from anti-prop. end) Prop. Shaft - Clockwise.

Valve Gear: Crank operated sleeves actuated through bevel gears on counter shaft. Valve timing can be adjusted individually or as a unit controlling all six valves on a bank. (Two inlet ports, two scavenging ports, and three exhaust ports per cylinder)

Firing Order: 1-9-5-12-3-8-6-10-2-7-4-11

After	7-8-9-10-11-12	Forward
End	6-5-4- 3- 2- 1	End

Overall Dimensions: Length: 72 inches (not including fuel pump or prop shaft).
Height: 47 inches.
Width: 22 inches.

Valve Timing:

	Open	Close
Intakes..................	8^{o} ATC	62^{o} ABC
Bottom Exhaust...........	64^{o} BBC	67^{o} ABC (by top ring)
Top Exhaust..............	20^{o} BBC	55^{o} ATC
Intake Side Scavenging...	17^{o} ABC	10^{o} BBC (completely at 12^{o} AB)
Exhaust Side Scavenging..	75^{o} BTC	31^{o} ABC

Supercharger Gear Ratio: 8.5:1

Supercharger Data:

Type: Single stage, centrifugal, direct geared, aluminum alloy impeller.

No. of Impeller Blades: 12

No. of Diffuser Blades: 13

Impeller Blade Entrance Angle: 60^{o}

Dimensions: Overall impeller diameter - 10 3/16 inches

Discharge Opening: 3 7/8 inches

Intake Opening: 6 1/4 inches with 2 1/2 inch shaft.

APPENDIX II

Material Analysis.

Material Analysis

Crankcase: Aluminum Alloy.

Cylinder Barrels: Cast Bronze.

Cylinder Spacer: Aluminum Alloy.

Cylinder Head: Aluminum Alloy, (with two steel compression rings).

Pistons: Aluminum Alloy.

Sleeves: Steel.

APPENDIX III

Calculations.

Calculations

Computation of supercharger pressure ratio:

The plot of supercharger pressure ratio (fig. 21) was obtained using the following formula from "Superchargers for Aircraft Engines", R.G. Standerwick and W. J. King.

$$P_2/P_1 = \left(1 + \frac{\eta V^2}{6083\,T_1}\right)^{3.534}$$

V = tip speed of impeller

T_1 = Inlet temperature to supercharger

Assuming:

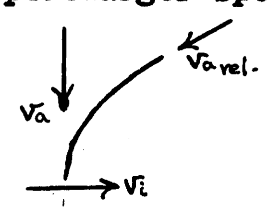
$\eta = .75$

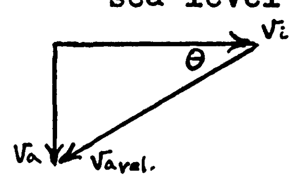
$T_1 = 520^{\circ}R$

Calculation of supercharger air delivery:

Engine speed = 2400 r.p.m.

Supercharger speed = 20,400 r.p.m. (Assumed designed speed sea level conditions)





$\theta = 30^{\circ}$

$V_i = 2\pi rN = 2 \times 20{,}400 \times \pi \times \frac{3.156}{12}$

$V_i = 33{,}800$ Ft./min.

$V_a = V_i \tan 30^{\circ} = 33{,}800 \times .577 = 19{,}500$

$M_a = V_a A_i \rho_{ai}$

$A_i = \frac{\pi(6.32^2 - 2.5^2)}{4 \times 144}$

$A_i = 0.184$ Ft.^{2}

$\rho_{ai} = .0765$ #/Ft.^{3}

$M_a = 19,500 \times 0.184 \times .0765$

$M_a = \underline{275 \text{ lbs./min.}}$

$$Y = (P_2/P_1)^{\frac{k-1}{k}} - 1 = (1.89)^{.286} - 1 = 1.20 - 1 = .20$$

$$(T_2 - T_1)_{actual} = \frac{T_1 Y}{\gamma} = \frac{520 \times .20}{.75} = 139^{o}F$$

$T_2 = 659^{o}R$

$\text{Power}_{s.c.} = M\ J\ C_p\ (T_2 - T_1)/\gamma$

$\text{Power}_{s.c.} = 275 \times 778 \times .240(139)/33,000 \times .75$

$\text{Power}_{s.c.} = \underline{289 \text{ HP.}}$

__Air Requirements of engine:__

Engine speed = 2400 r.p.m.

Assuming volumetric efficiency = 100%

$M_a = V_d \times \frac{N}{2} \times \rho_2$

$M_a \ \frac{1365}{1728} \times \frac{2400}{2} \times .114 = 108.1 \text{ lbs/min.}$

$M_a = \underline{108.1 \text{ lbs/min.}}$

__Power required for scavenging:__

$$\text{Power} = P_{s.c.} \times \frac{M_{a\ scav}}{M_{a\ total}}$$

$$\text{Power} = 289 \times \frac{275 - 108.1}{275} = 289 \times \frac{166.9}{275}$$

Power = __175 HP. for scavenging__

Power = __114 HP. to supply air for engine__

Estimated power of engine:

$$\text{Power} = M_a \times F/A \times E_c \times \eta \times J \qquad \eta = .60\ \eta_{I.C.}$$

$$\text{Power} = 108.1 \times .08 \times 18,900 \times .318 \times 778/33,000$$

$$\text{Power} = 1230 \text{ HP.}$$

Increase of volumetric efficiency to jusify scavenging:

$$\frac{1230-175}{1230} = \frac{e_1}{e_2} = \frac{1055}{1230}$$

$$e_1 = .86$$

$$e_2 - e_1 = \underline{14\%}$$

$$e_{possible} = \frac{r}{r-1} = \frac{6.5}{5.5}$$

$$e_{possible} = 118\%$$

Maximum fuel-air ratio at 2400 r.p.m.:

$$M_a = 108.1 \text{ lbs/min.}$$

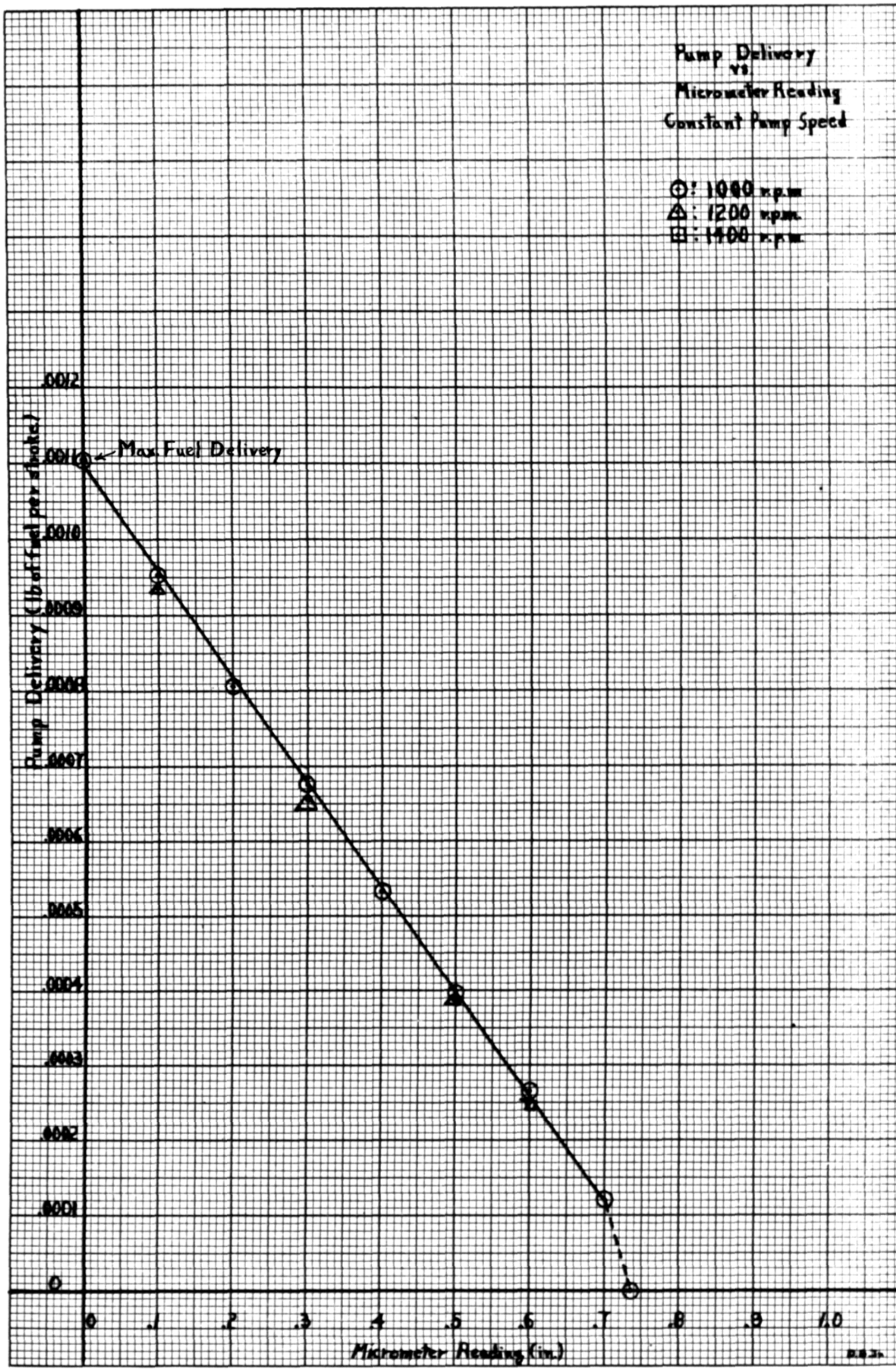
$$M_f = 15.9 \text{ lbs/min. (from graph)}$$

$$F/A = \frac{M_f}{M_a} = \frac{15.9}{108.1}$$

$$F/A = .146$$

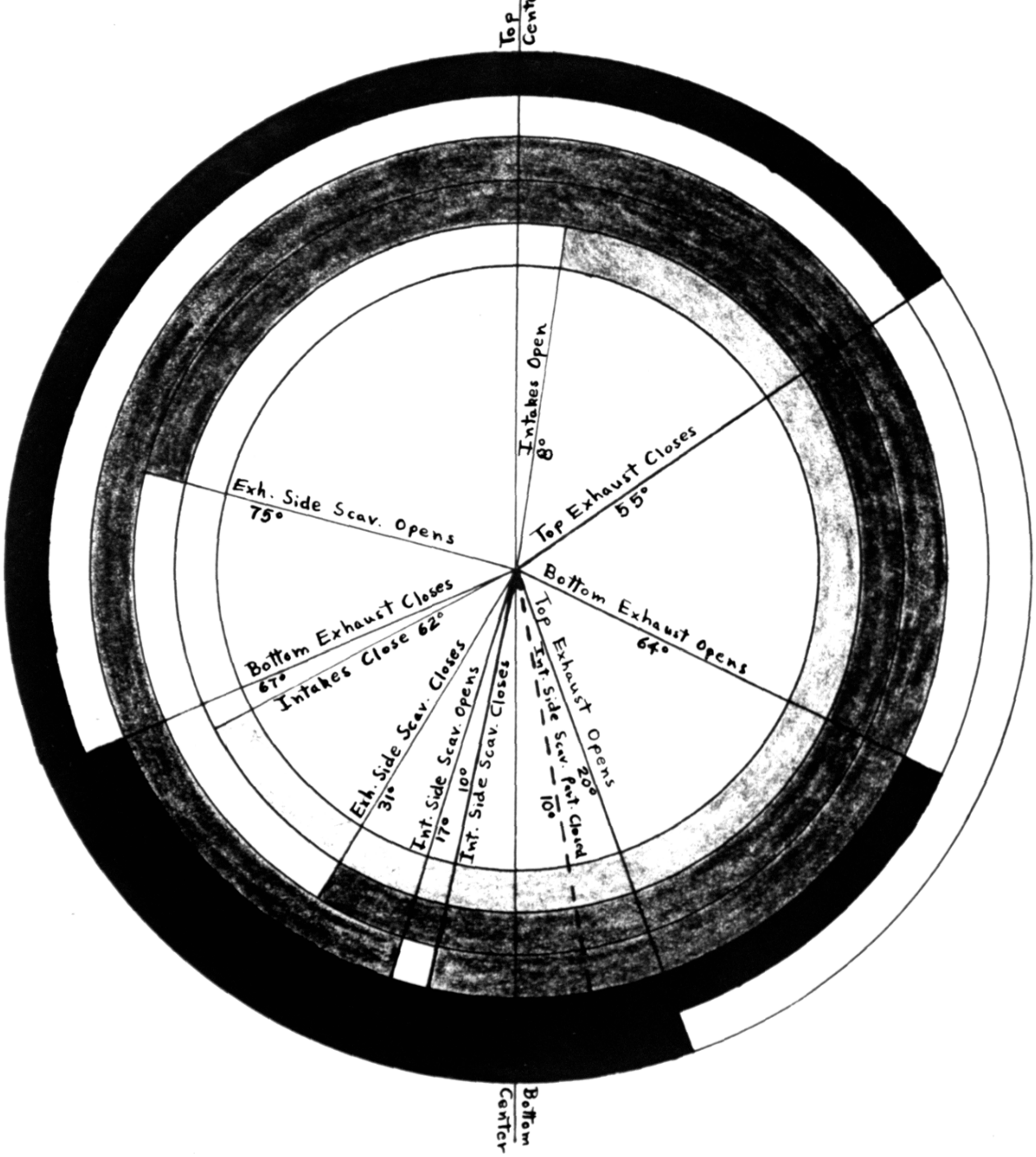
APPENDIX IV

Graphs



Pump Delivery vs. Micrometer Reading
Constant Pump Speed
: 1000 r.p.m.
: 1200 r.p.m.
: 1400 r.p.m.
Max. Fuel Delivery
Pump Delivery (lb. of fuel per stroke)
.0012
.0011
.0010
.0009
.0008
.0007
.0006
.0005
.0004
.0003
.0002
.0001
0
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
Micrometer Reading (in.)

Fig. 21.



Valve Timing Diagram.

Fig. 22.

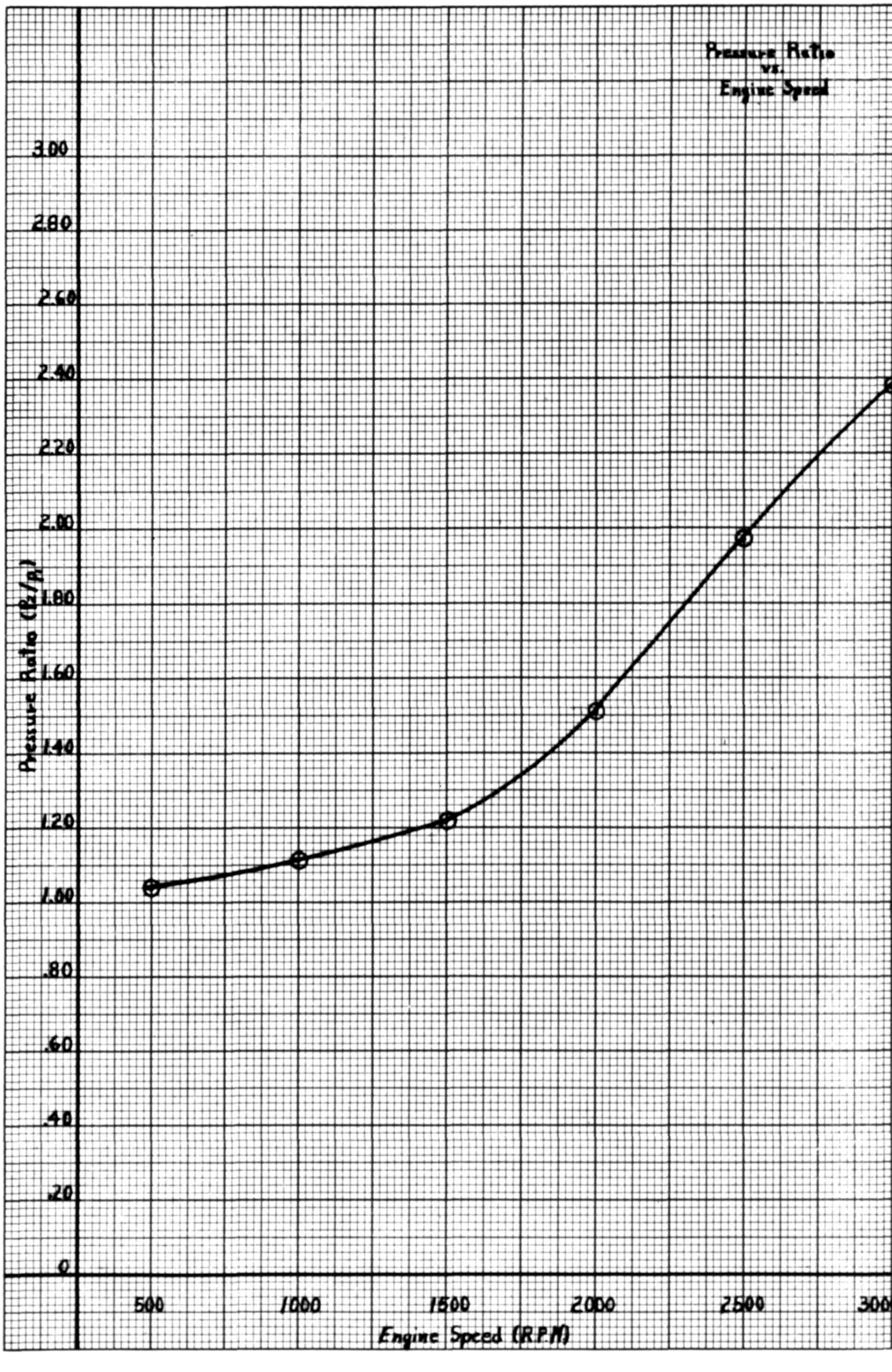

Pressure Ratio
vs.
Engine Speed
3.00
2.80
2.60
2.40
2.20
2.00
1.80
1.60
1.40
1.20
1.00
.80
.60
.40
.20
0
Pressure Ratio (P2/P1)
500
1000
1500
2000
2500
3000
Engine Speed (R.P.M)

Fig. 23.

APPENDIX V

Drawings and Photographs.

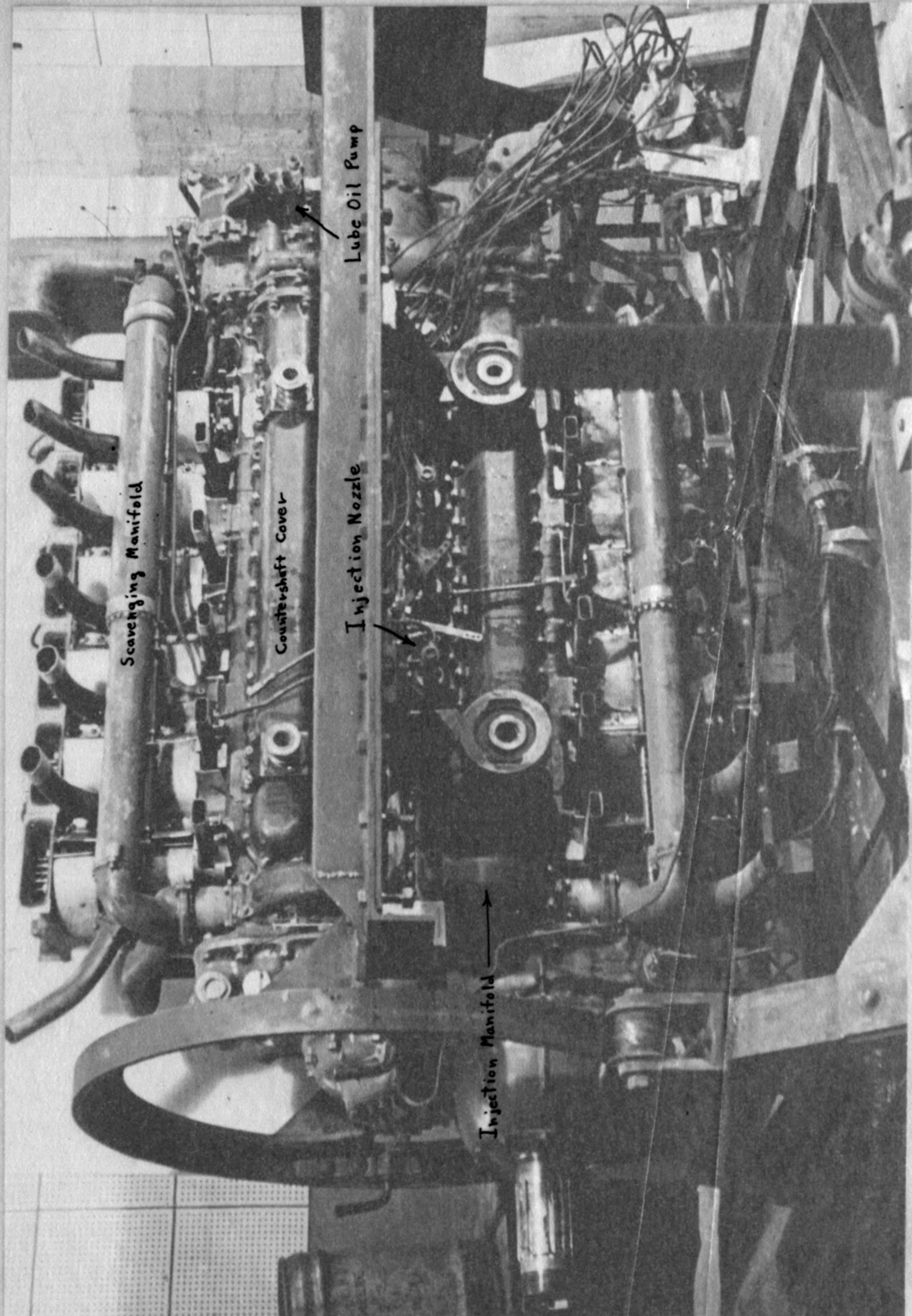


Fig. 1 (a): Exhaust side view of engine.

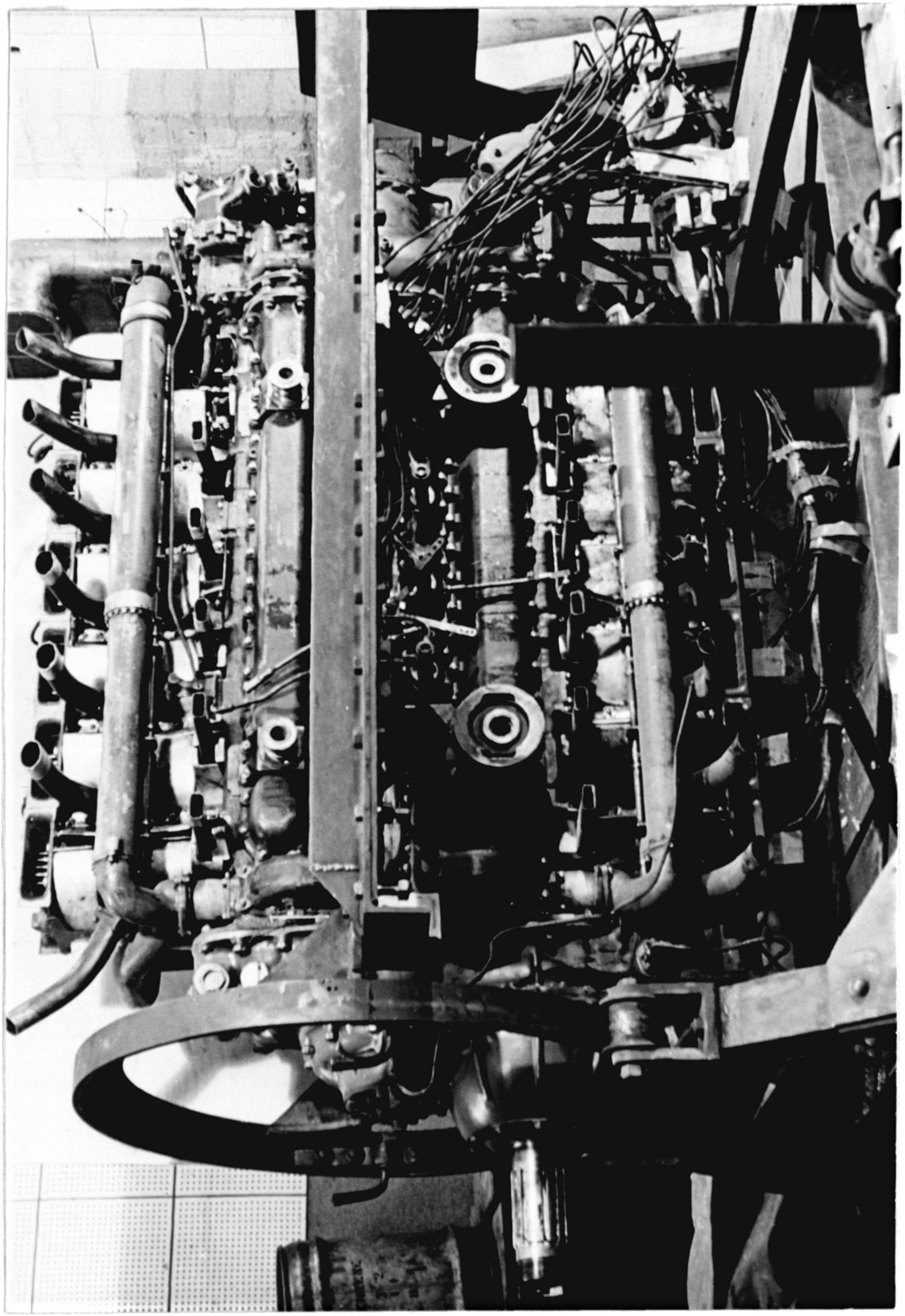


Fig. 1 (a): Exhaust side view of engine.

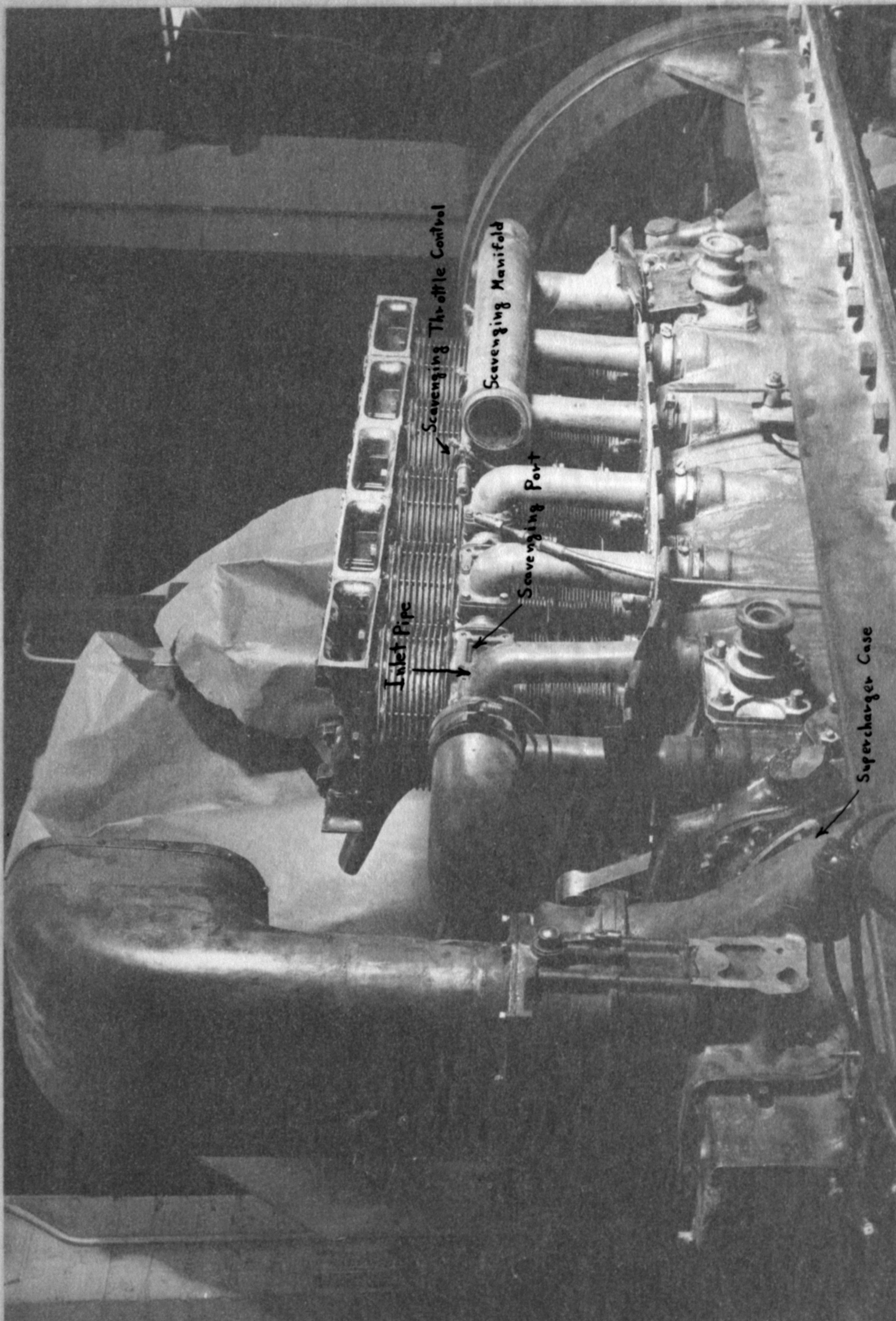


Fig. 1 (b): Inlet side, lower bank.

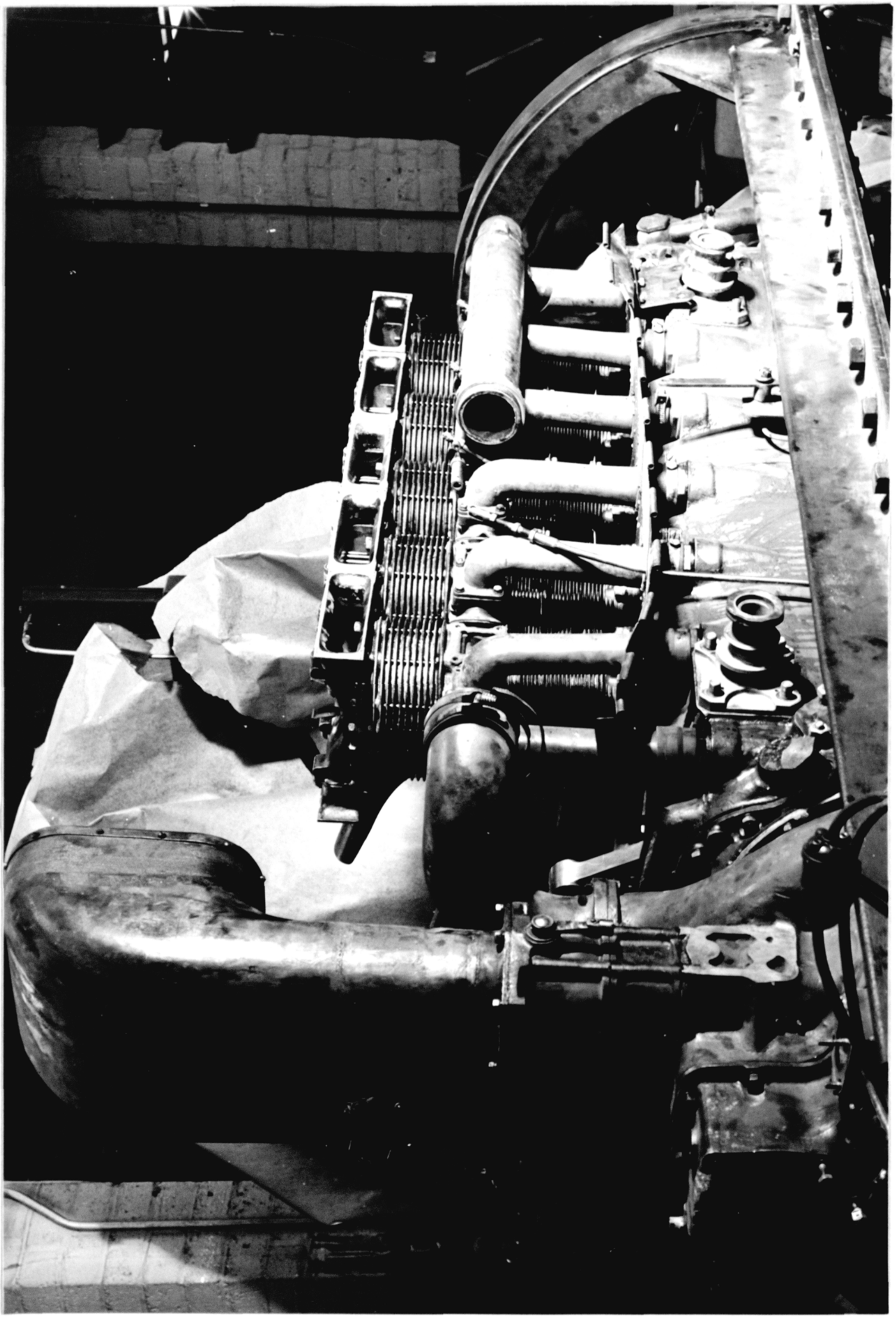


Fig. 1 (b): Inlet side, lower bank.

Fig. 1 (c): Manifold cover removed (inlet pipes).

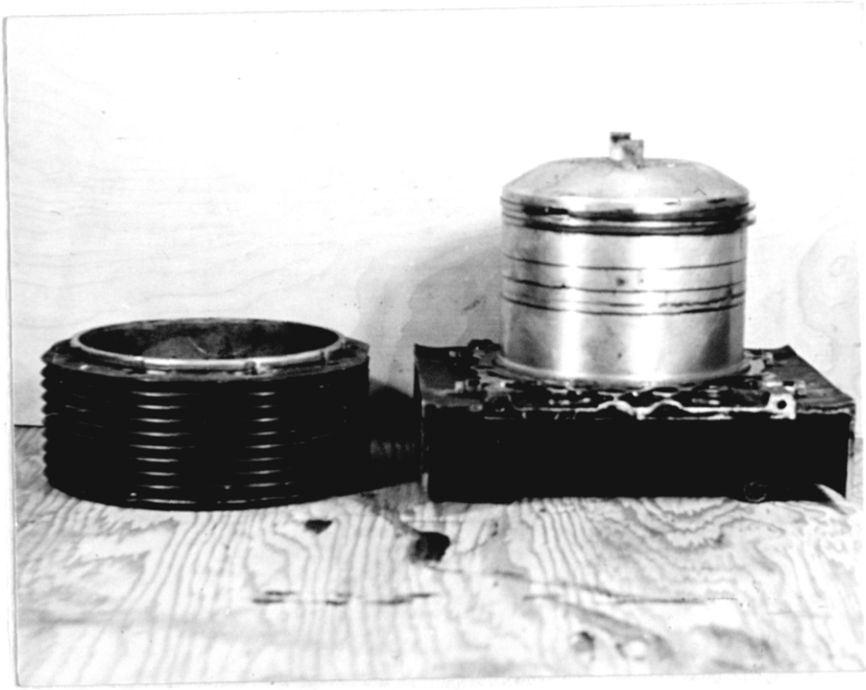


Fig. 2: Cyl. head and spacer.

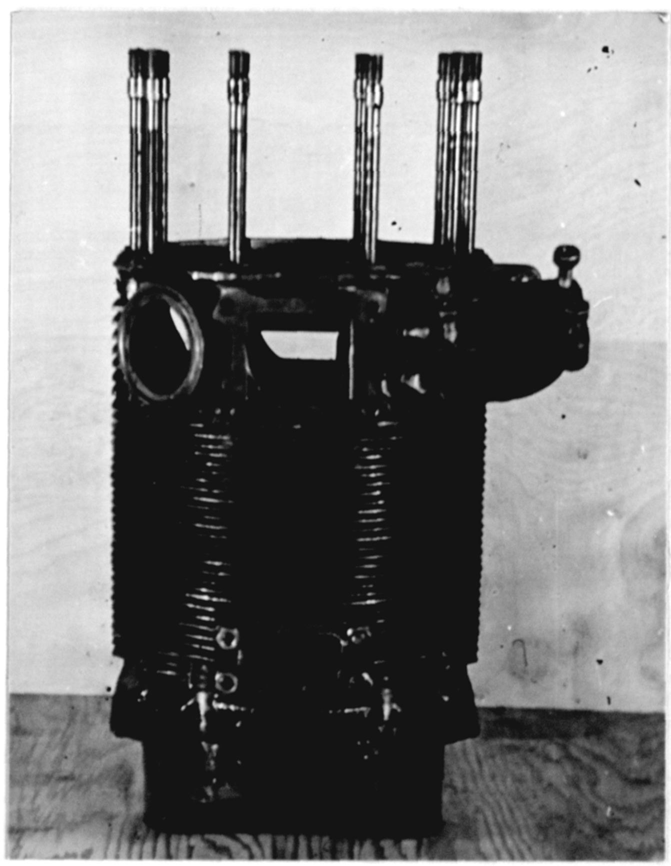


Fig. 3: Cylinder barrel.

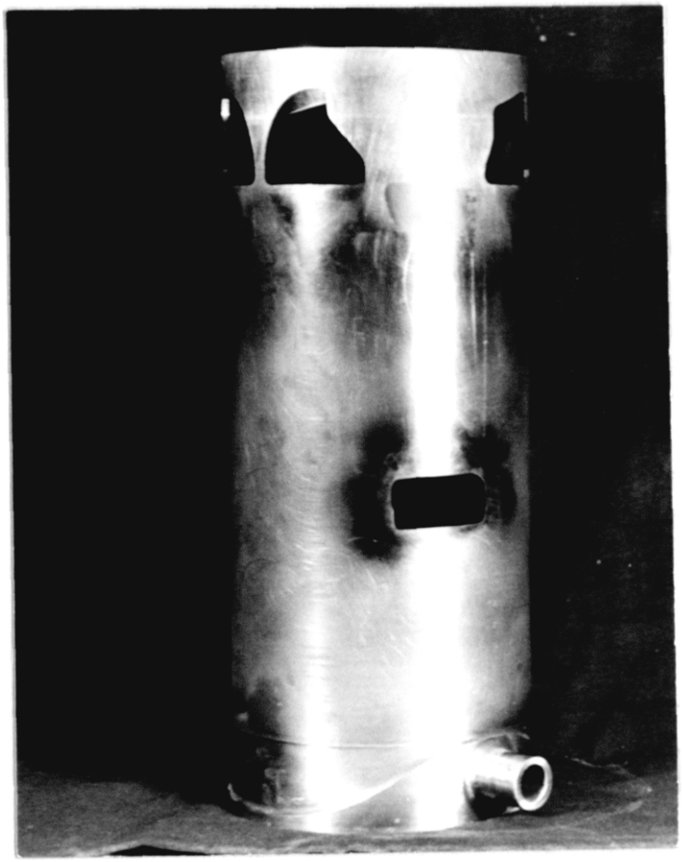


Fig. 4: Sleeve

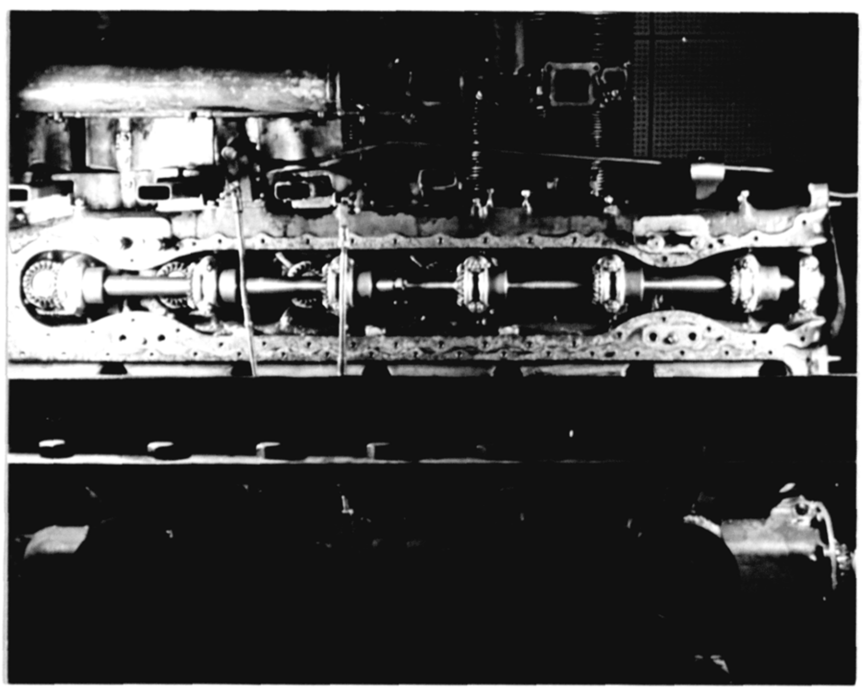


Fig. 5: Countershaft

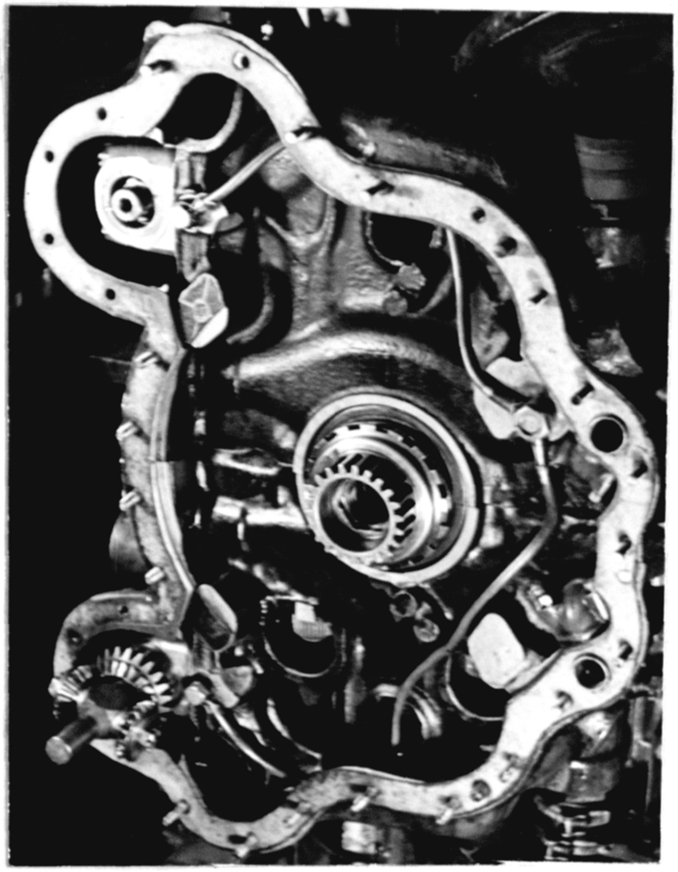


Fig. 6: Rear view of crankcase

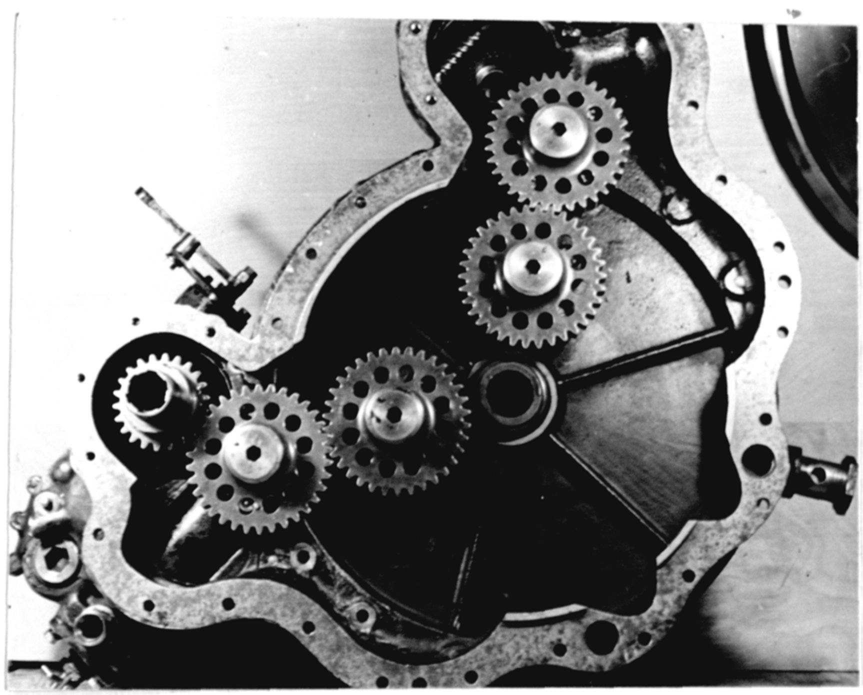


Fig. 7: Front view of timing gear case

Fig. 8: Piston.

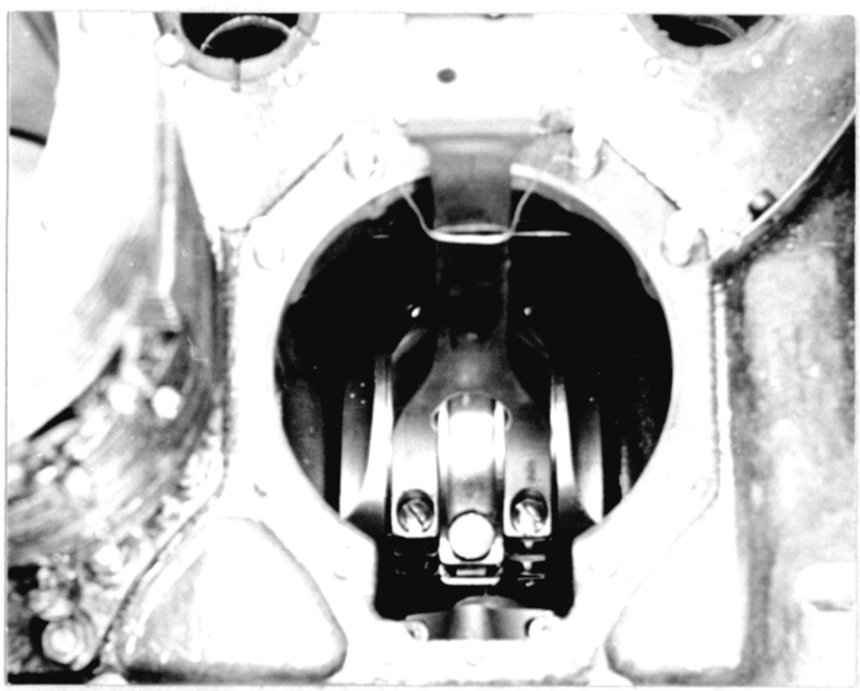


Fig. 9: Connecting rod.

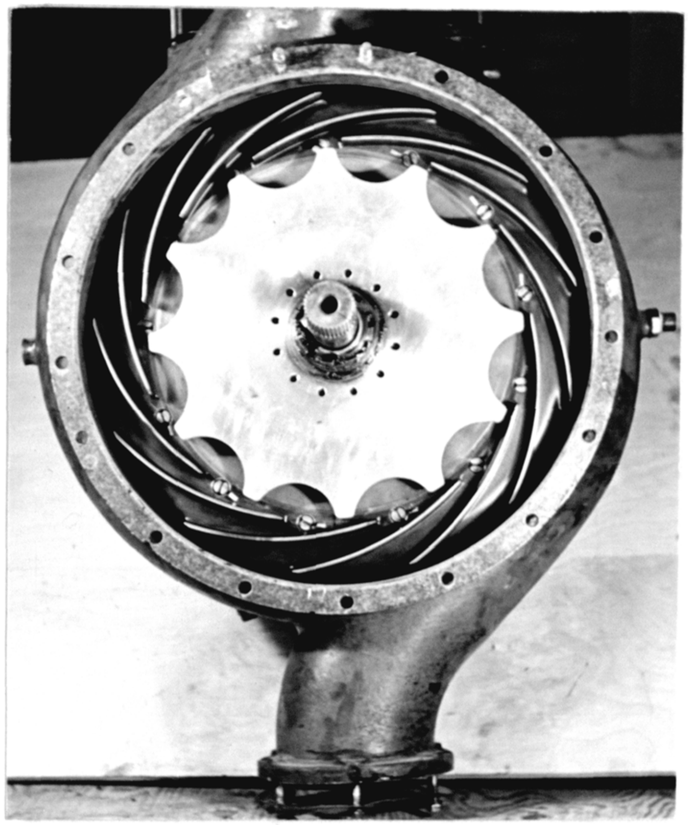


Fig. 10: Front view of supercharger

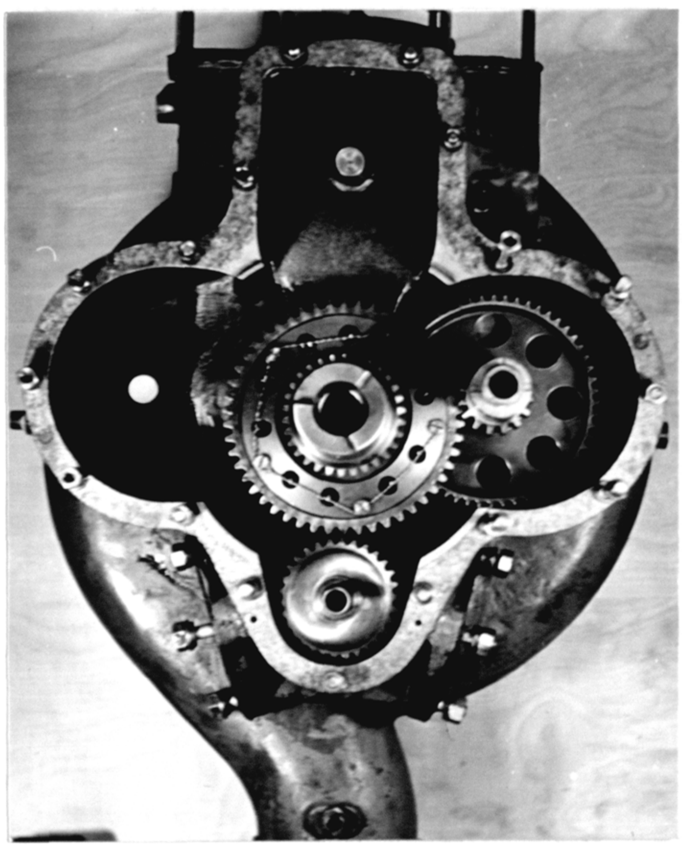


Fig. 11: Supercharger drive.

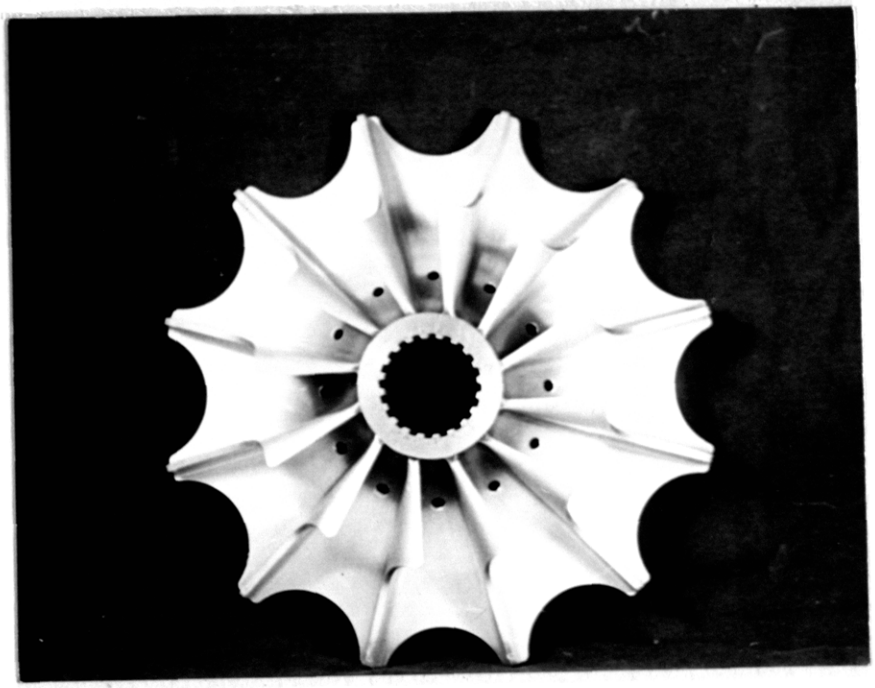


Fig. 12: Impeller

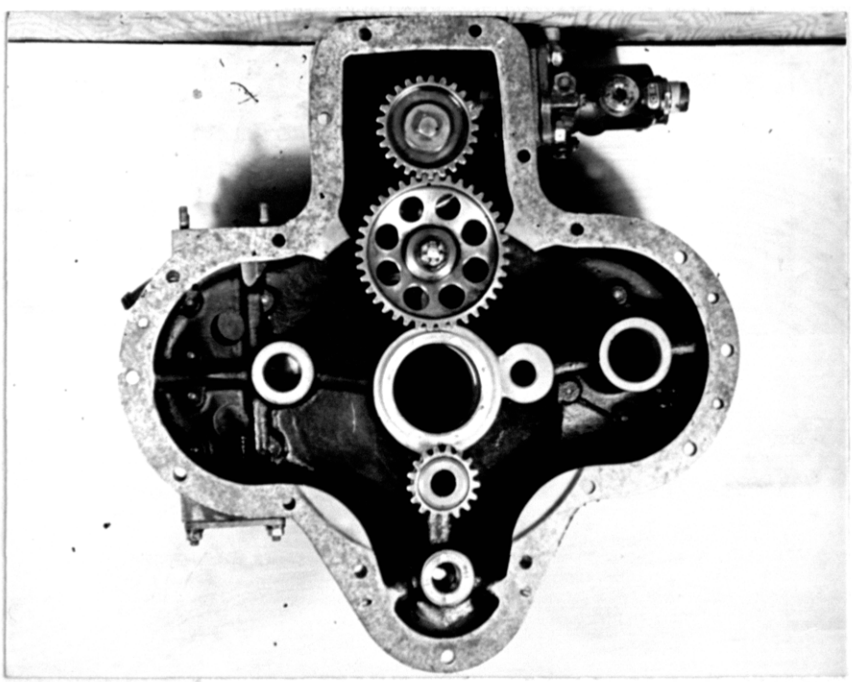
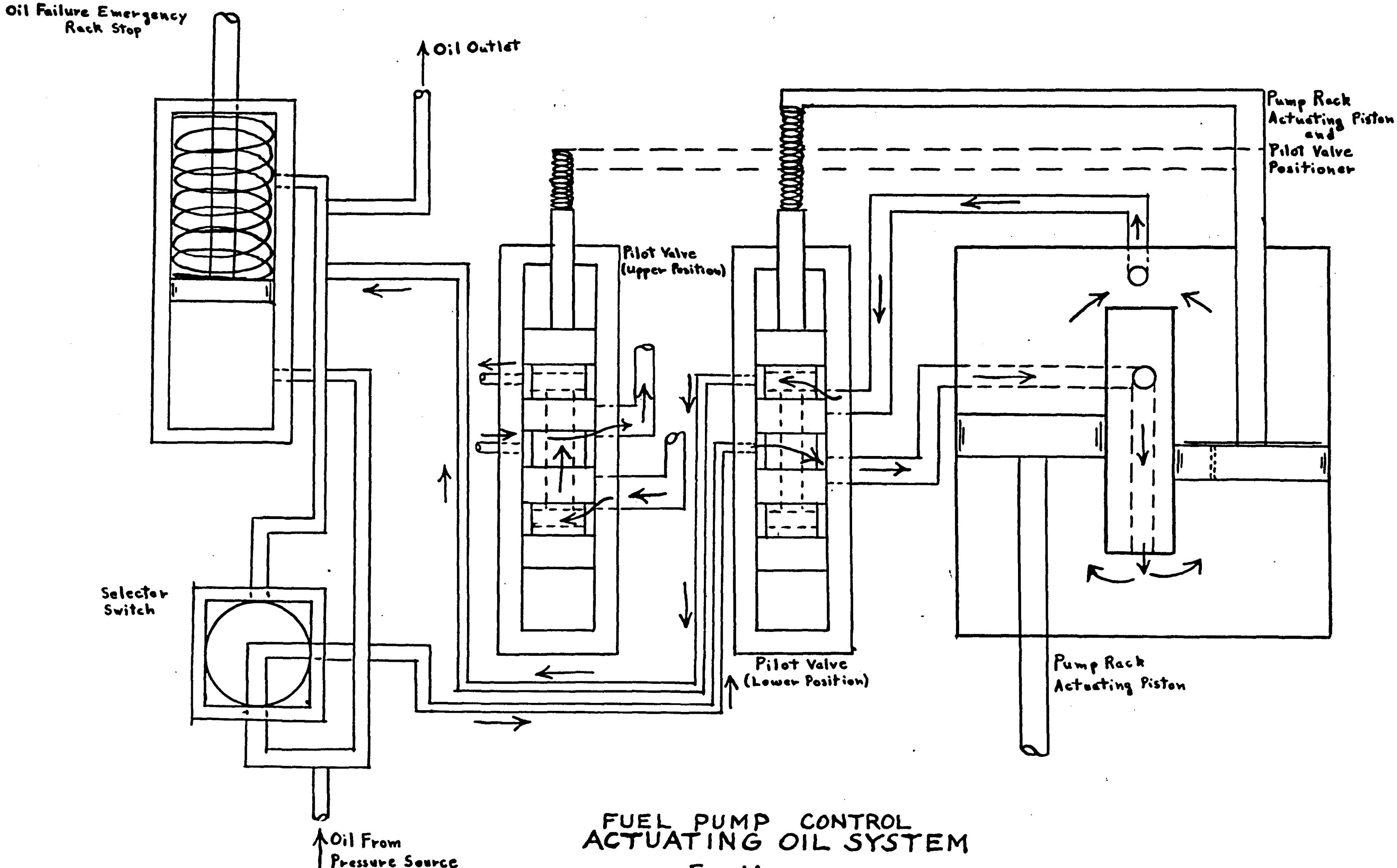


Fig. 13: Front view of rear casing.



FUEL PUMP CONTROL
ACTUATING OIL SYSTEM
Fig. 14.

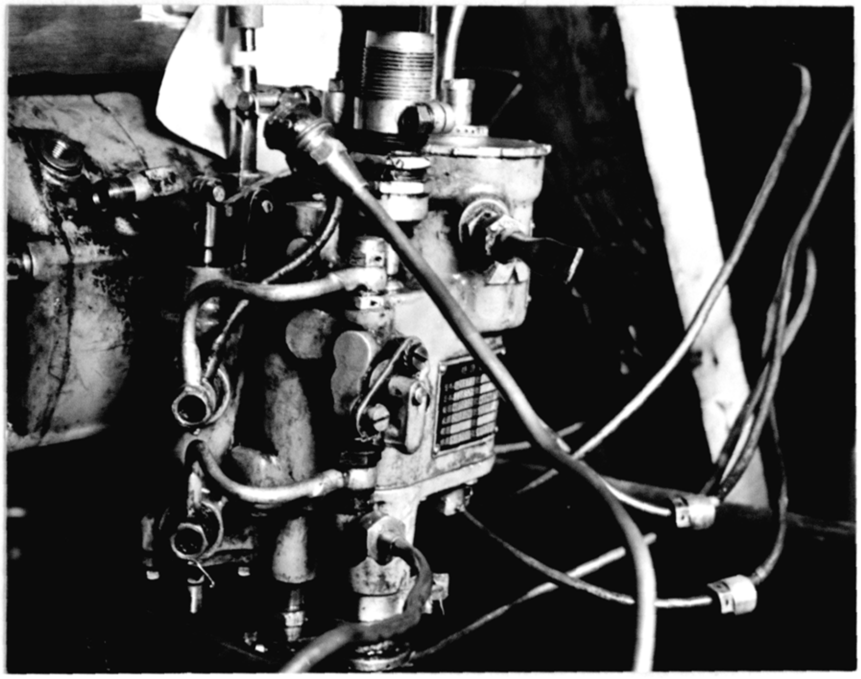


Fig. 15: Fuel pump control.

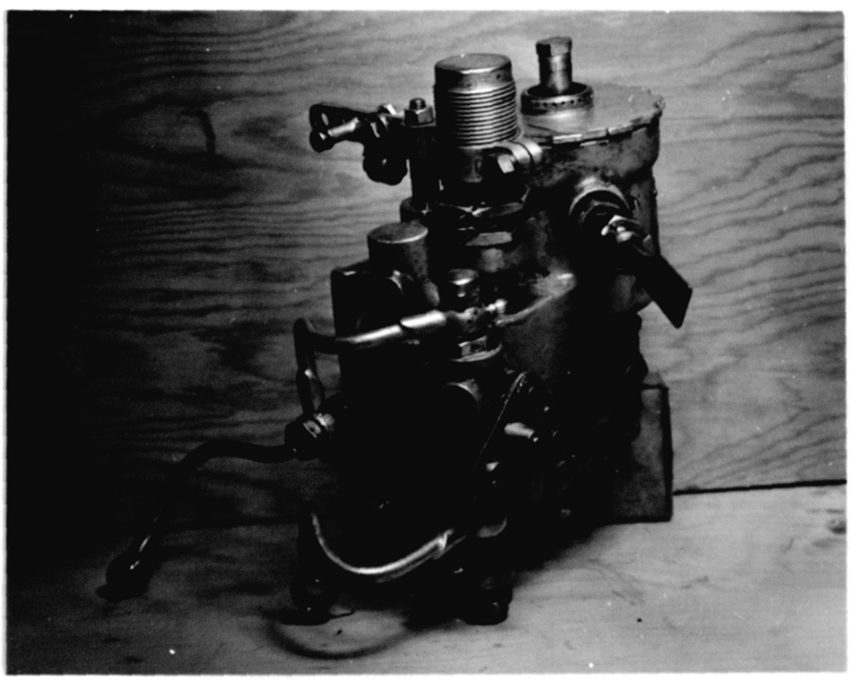


Fig. 16: Fuel pump control.

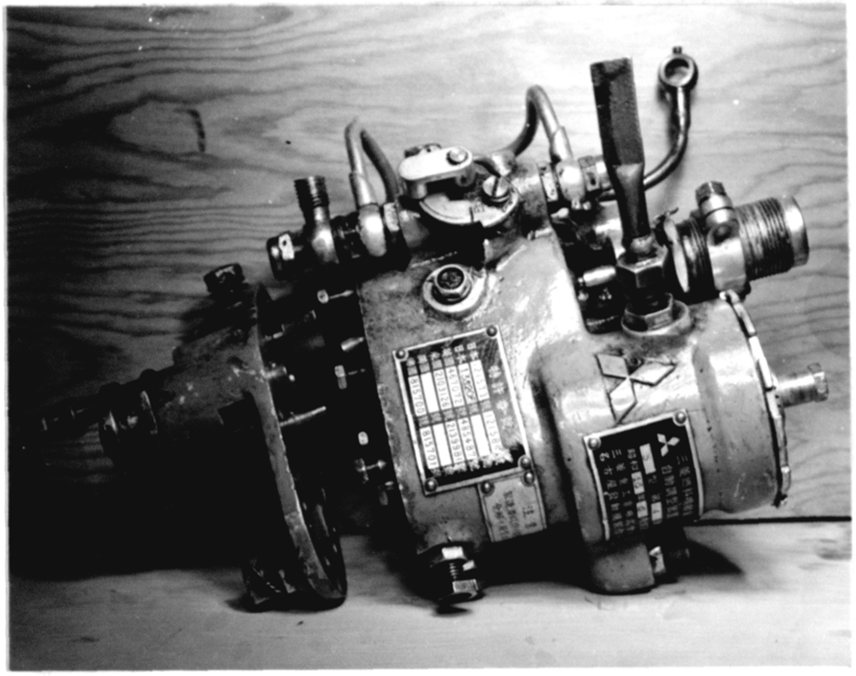


Fig. 17: Fuel pump control.

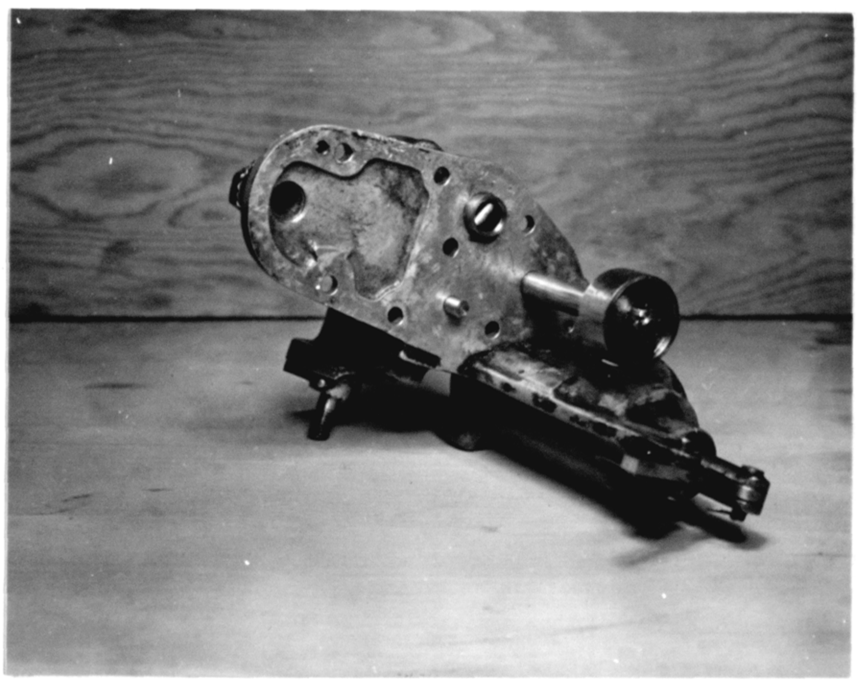


Fig. 18: Lower half, fuel pump control.

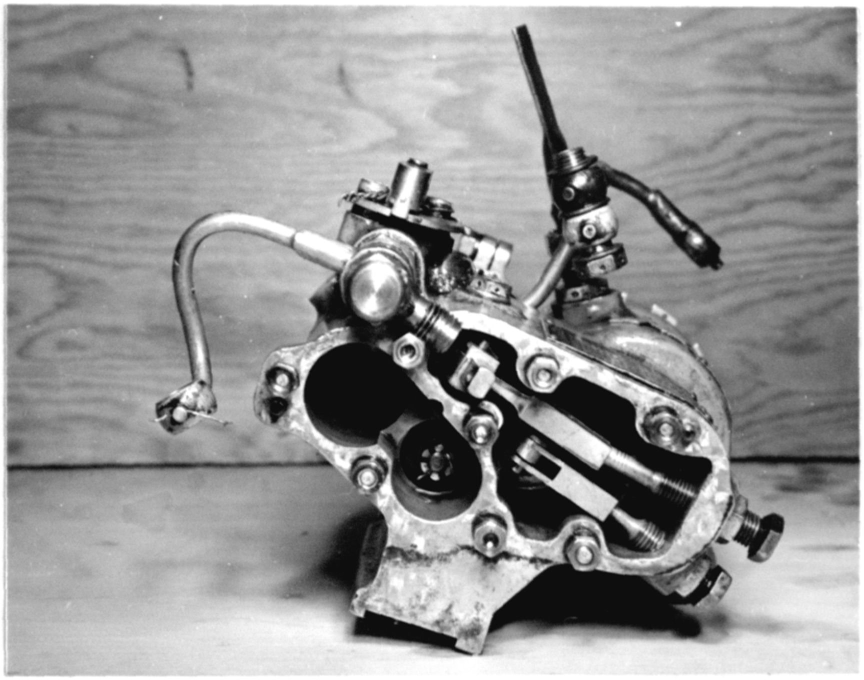


Fig. 19: Upper half, fuel pump control.

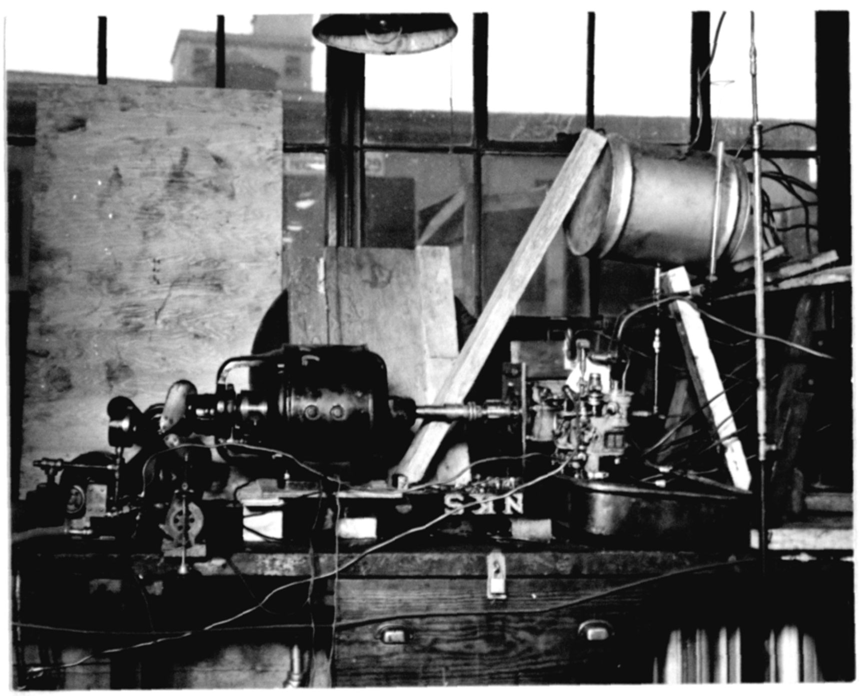


Fig. 20: Fuel pump test apparatus.

APPENDIX VI

Bibliography

Bibliography

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Wright Field, Dayton, Ohio.