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By Dr. Buchner

PART II

A DISCUSSION OF THE SEMI-DIESEL PRINCIPLE AND ITS APPLICATION TO VARIOUS TYPES OF SOLID-INJECTION ENGINES

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McKechnie's indicator diagrams of 1910 differ but little from the constant-pressure diagrams of Diesel engines with air injection. This is comprehensible, since, at that time, the constant-pressure line was still considered an essential characteristic of the Diesel engine and McKechnie did not contemplate changing the principles and limits of the Diesel process, his idea being only to replace the air-injection method by simple mechanical injection ("solid injection"), while retaining the high compression pressure unchanged.

During the last decade, the question of the most favorable combustion line has often been the subject of both theoretical and practical investigations. Thereby freer and deeper conceptions of the form of the combustion line have been evolved and thus it has come about that the indicator diagrams

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of the so-called "solid-injection" Diesel engines, now gradually coming into publicity, differ greatly from the diagrams of the original Diesel engines and often remind us strongly of the diagrams of explosion engines. They have decidedly sharp points and only a slight development in the direction of the constant-pressure line, though the latter increases somewhat as the load is increased. Unlike the well-known indicator diagrams of explosion engines, the compression pressures are higher and the pressure increase (the ratio of the maximum pressure to the compression pressure) is less than in explosion engines, though it is likewise affected by the engine loading. It seems entirely appropriate to designate such engines, whose cycles lie between those of the older explosion engines and those of the original Diesel engines (constant-pressure engines) and whose indicator diagrams are accordingly referred to as "mixed diagrams," as "semi-Diesel engines," after the precedent of the Americans, English and French. It must be remembered, however, that they constitute an engine type with flexible limits, in which the hot-bulb engines can be included, as is often done by Americans.

Fig. 16 shows indicator diagrams of the Deutz Engine Works, Körting Bros. Co., and Ruston & Hornsby. For the Deutz VM and the Körting engines, so-called "shifted diagrams" are given alongside the regular indicator diagrams. In the Deutz VM, the maximum pressure is 38-40 kg/cm³ (540-569 lb./sq.in.) at a compression pressure of 25 kg/cm² (355.6 lb./sq.in.). The Körting engine shows a compression pressure of about 39 kg/cm² (554.7 lb./sq.in.) and a maximum pressure of 44 kg/cm² (625.8 lb./sq.in.). The Ruston & Hornsby engine shows, at idling speed, a maximum pressure of 38 kg/cm² (540 lb./sq.in.) and, at normal load, a maximum pressure of 48 kg/cm² (682.7 lb./sq.in.). Since there are no "shifted diagrams," the compression pressures can only be read approximately on the vertical scale.

No further information on these three engines is necessary at this point, for these diagrams are only intended to help to explain the nature of high-speed semi-Diesel engines, which are closely related to these engines. Although these engines are distinguished by a very favorable rate of fuel consumption, the reference to the solid injection and a certain form of nozzle affords no adequate explanation of this fact, but their success is largely due to the fact that technicists have learned to control the character of the combustion line and to determine the most favorable values for the beginning and end and during the course of the fuel intake, which values, however, are not always easy to maintain in actual engine operation under greatly varying conditions.

Prof. Hawkes, of Armstrong College in Newcastle-on-Tyne, has demonstrated, on a 300 HP. engine with reciprocating pistons, that the fuel consumption is decidedly affected by the shape of the indicator diagram and the latter, in turn, by the temporal course of the fuel delivery ("Engineering," 1920, No. 2, p. 786). Both of the shifted diagrams in Fig. 17 again show the ascent of the combustion curves in comparison with the compression curves, at which the fuel is delivered. The motion of the fuel valve is also represented and emphasized by the hatching. Fuel cams of various shapes were tried, but the engine was always adjusted to a maximum pressure of 44 atm. by a simple shifting of the cams. Everything else proceeds from the following table.

Experiments of Professor Hawkes.

Fuel Texas oil	Diagram I	Diagram III
Fuel valve I		
Advance	10.0°	18°
Retard	28.5°	51°
Fuel valve 2		
Advance	5.5°	9°
Retard	38.0°	41°
Effective HP.	296	300
R.P.M.	296	300
Mean effective pressure	6.0	5.9
Fuel consumption per HP_e/hr.	212	238

Diagram III, with slowly opening fuel cams, shows a slow combustion in comparison with diagram I. The fuel consumption was greater and the exhaust was slightly smoky.

Dr. Heidelberg's preliminary experiments with the Deutz VM engine also show that much depends on the adjustment of the fuel cam. He found that the beginning of the fuel delivery has an especially marked effect on the combustion line in the direction of constant or explosive pressure. An advance angle of 40-45° produced explosive combustion, while 25-27° gave constant-pressure combustion. The first indicator diagram in Fig. 18 was made at an advance angle of the fuel valve of 25°, while the second diagram was made, with the same cam, for an advance angle of 45°. In both cases the fuel pressure was about 280 atm. and the compression pressure about 35 atm. The indicator diagram for the first case shows constant-pressure combustion, both the ignition pressure and the compression pressure being 35 atm. The diagram for the second case clearly indicates an explosion which reached a maximum pressure of 57 atm. In the first case, the fuel consumption was 204g (7.2 oz.) per HP_e/hr. for $n = 280$ and $N_e = 50$ HP. In the second case, it was 180g (6.4 oz.) per HP_e/hr. for $n = 218$ and $N_e = 48.5$ HP. It is therefore absolutely necessary to determine the most favorable angle of advance for all high-speed semi-Diesel engines for all revolution speeds and loads and, if possible, to provide for the automatic adjustment of the most favorable angle for any kind of engine, independently of the engine driver.

According to Dr. Heidelberg, the magnitude of the compression pressure in explosive combustion affects the fuel consumption less than in constant-pressure combustion. This fact seems very significant, for it opens up the most promising prospects, that the compression pressure in high-speed semi-Diesel engines can be reduced to a magnitude practicable for high-speed engines and that the total weight of the engine can be reduced and its life increased. It is advisable, however, not to go too far with the explosive combustion. After having fortunately overcome the prejudice that the constant-pressure line is the only goal worth striving for, we must beware of the danger of going to the other extreme. "The middle course is the best," as Daedalus told his son Icarus.

In Dr. Heidelberg's article "Ueber Einspritz- und Verbrennungsvorgänge in kompressorlosen Dieselmotoren" (On Injection and Combustion Processes in Solid-Injection Diesel Engines) ("Zeitschrift des Vereines deutscher Ingenieure," 1924, p. 1047 ff), the experiments on the shape of the combustion chamber and on the magnitude of the injection pressure appear especially important and directive. No one who has anything to do with Diesel can afford to disregard the results of Dr. Heidelberg's researches. Some of these results have been taken from Dr. Heidelberg's original diagrams and are shown in Fig. 19.

At the top of Fig. 19, in the middle, there is shown the final form of the combustion chamber with a No. 1 piston head, which furnished the most favorable fuel consumption in the experiments. On the left and right are diagrams of the piston heads Nos. II and III, which proved less favorable. The heavy lines I, II, III represent the fuel consumption in grams per HP_e/hr. The effective HP. varies between 20 and 65 and the mean effective pressure between 1.84 and 5.9 kg/cm² (83.9 lb./sq.in.). Piston I furnishes, at about 50 HP. and a mean pressure of 4.7 kg/cm² (66.8 lb./sq.in.), the most favorable fuel consumption of but little over 180g (6.4 oz.) per HP./hr. The final compression pressure was 32 atm. in all three cases. The "nozzle b", shown in the middle diagram, was used. It is a needle valve, which is provided with three spraying holes behind the valve seat. The fuel pressure in the nozzle was 200 atm., and the maximum combustion pressure ("ignition pressure") under full load was 38-40 atm. at an advance angle of 40° for the pump cam.

The fine lines ascending from left to right represent the exhaust temperatures obtained with the piston heads II and III and the "nozzle b". All the exhaust temperatures, which were observed immediately after the exhaust valve, fall inside the strip bounded by these lines. This strip is a proof of the "cold combustion" in solid-injection engines, for the low mean temperature.

You have all heard the objection that fuel-injection engines would not last long on account of the high pressures in the fuel pump and injection nozzle. This objection will soon be overcome by the further development of high-pressure fuel pumps and injection nozzles. It does not matter that not every engine works is now in position to make such pumps and nozzles, which belong in the realm of skilled mechanics, since, for this purpose, there is necessary, among other important things, the education and training of a special class of workmen and engineers. In any event, one thing is certain that we Germans have here an open field which is not closed to us by the Treaty of Versailles. So let us grasp it!

Moreover, I might refer those who consider untenable the high pressures connected with fuel injection, to no less a person than Dr. Forster, the famous shipbuilder, who delivered a lecture in this room before the "Schiffbautechnische Gesellschaft," on experiments on recent ship lines and expressed himself to the effect that, in order to make progress, we must, with full knowledge, study the extremes. It is a comparatively simple matter for the practical constructor to make a compromise, but the knowledge of the extremes must first be acquired.

The value of the fuel-injection method becomes the more manifest, the greater the fuel-injection pressure employed. This is clearly demonstrated by the lower diagram in Fig. 19, in which the effect of the pump pressure on the specific fuel consumption is represented. The diagram begins at the left with an injection pressure of about 70 atm. and extends to 300 atm. It does not, therefore, reach the maximum pressures with which McKechnie worked. For a given engine power (In the diagram, N_e = 52 HP.), the following peculiar course of the fuel-consumption curves is shown. As the fuel pressure increases, there first occurs a great improvement in the fuel consumption. Then, with a further increase in the fuel pressure, no appreciable improvement follows and then, following this "insensitive zone," there occurs a still further diminution of the fuel consumption.

Dr. Heidelberg offers no explanation for this remarkable phenomenon. Perhaps light may be shed on it by a hypothesis at which I arrived in considering the problem of how to separate the fuel jet into the smallest possible drops. Among other things, I found that the fuel jet must enter the compressed air at a velocity greater than that of sound. The splendid photographic researches of E. Mach on the behavior of the air in the vicinity of a moving bullet demonstrated experimentally the sudden increase in the coefficient of resistance ζ_o on exceeding the velocity of sound. Siacci found a decrease in ζ_o for a further increase in the speed of the bullet beyond the velocity of sound (Compare Hans Lorenz, "Technische Wärmelehre," 1904, p. 147).

The calculation of the velocity of sound by Laplace's formula $c = \sqrt{\frac{\kappa p}{s}}$, in which c denotes the velocity of sound, κ the exponent for adiabatic change of state, p the pressure and s the specific gravity (Compare E. Grimschl, "Lehrbuch der Physik," 1923, p. 683), gave extraordinarily high fuel pressures, which are much greater than the ones employed by Dr. Heidelberg. For a compression pressure of 22 atm. and an air temperature of 70°C (158°F) at the beginning of the compression, a sound velocity of nearly 590 m (1936 ft.) per second (in air compressed at 22 atm.) was found, which would give a fuel pressure of at least 1500 atmospheres.

The action of the fuel jet on the surrounding air is, however, not exhausted with the assumption that the fuel jet enters the compressed air like a bullet and that the individual particles formed by the breaking up of the fuel jet are also comparable to bullets. Since the fuel jet penetrates the hot compressed air in a cold condition and is ignited like a match on a rough surface, the bullet theory must be supplemented by the assumption that the bullet is a cold bearer, which takes heat from its surrounding medium. It is obvious that LaPlace's formula does not apply to this case. The assumption on which it rests, that the propagation of a longitudinal wave in a gas is an adiabatical process must be replaced by the general assumption of a polytropic change of state connected with the progress of the longitudinal wave.

Hence we have the formula $c = \sqrt{\frac{np}{s}}$ for the velocity of sound, n being the exponent of the polytropic curve. Just as, in the case of a bullet moving at about the velocity of sound, there is quite an extensive region in which the air resistance is very great, in like manner, according to the above hypothesis, critical conditions can also arise for a more extensive pressure and velocity region in the case of fuel jets. The circumstance that the region of insensitiveness is greater for Dr. Heidelberg's three-hole nozzle than for a one-hole nozzle, is not opposed to this, since the heat-transmission surface for a triple jet is nearly 45% greater than for a simple jet injecting the same amount of fuel. The velocity of sound depends, however, not only in a high degree on the initial temperature of the compressed air and on the compression pressure, but also on the temperature of the injected fuel, especially on all the factors affecting heat transmission.

The insensitive region for the three-hole nozzle, in Dr. Heidelberg's experiments, lies between 140 and 260 atm., which corresponds for gas oil, to theoretical fuel velocities between 180 and 245 m (590-804 ft.), as calculated according to Dr. Kuehn's formula $w_{th} = 15.16\sqrt{P}$ m/sec., the discharge coefficient being disregarded. The insensitive region of the simple nozzle lies between 210 and 260 atm., and the corresponding theoretical velocities being 220-245 m (722-804 ft.) per second. If these velocities are sound velocities, they must have polytropic curves whose exponents are between 0.133 and 0.346 for the three-hole nozzle and between 0.198 and 0.246 for the simple nozzle. From the above consideration, it follows at least that the heat exchange between the fuel jet and the compressed air exerts a decided influence on the atomization process and that, therefore, the temperature of the fuel jet and of the hot air must be taken into account in all fuel-injection experiments.

The Augsberg-Nuremberg Engine Works was induced to hollow out the piston heads in their vertical solid-injection four-stroke-cycle engines (cf. Fig. 37, Part III - Technical Memorandum No. 358). Regarding this form of piston, Professor Nägel expresses himself as follows: "Its success was largely due to the perception that the injection of the fuel into the cylinder under a pressure of several hundred atmospheres necessitates for its atomization a space of sufficient length in the direction of the jet. If the jet strikes the piston head, a portion of the fuel is precipitated and the reflected particles of fuel have largely lost their kinetic energy and therefore escape further atomization. On the other hand, a jet of small cross section, injected under great pressure, seems to be largely atomized by friction with the surrounding compressed air. This fact induced both the engine works mentioned to give their piston heads the form of a hollow hemisphere. In this hollow hemisphere the fuel jet is dissipated into a fuel cloud which, due to the cessation of the cold expanding injection air, is certain to ignite at a moderate compression ratio and (considering the smallness of the tested cylinders) to effect an excellent thermal outcome in mechanical work (cf. "Dieselmaschinen," Publication Office of the V.D.I., Berlin, 1923, p. 29, par. 2)."

The Deutz Engine Works seems, according to an article "Fort mit dem Compressor" (Away with the Compressor), to hold the deflecting action of the piston on the fuel as not entirely superfluous, but rather as an effective means for cooling the piston. On Page 2 of the above-mentioned article, it says: "The assumption that most of the fuel particles reach the piston head in a liquid state is clearly indicated by its unchanged appearance after long periods of operation. Heat is continuously abstracted from the piston head, which is threatened with overheating, by the evaporation of the fuel drops. It is regularly cooled by a fine spray before being again subjected to the effect of another combustion. Since the cooling spray and the heat requirement change approximately in proportion to the increasing load, we here have an almost perfect means for cooling the cylinders. It is not strange, therefore, that much larger cylinders can be used in solid-injection engines without piston cooling than in air-injection engines." Lower down on the same page it says: "The retort-like shape of the piston head in the 'Deutzer VM' favors the retention of heat by the air in the depression in the piston head, so that the fuel jet first strikes the hottest portion of the air." The Deutz Engine Works accompanies these statements with two diagrams (Fig. 20) intended to illustrate the atomization of the fuel jet and the temperature distribution in the piston head. The representations of Prof. Nägel and of the Deutz Engine Works are of general importance as regards the problematical individual engines.

On the one hand, the pure spraying action and, on the other, the dissipation of the fuel jet, in which the impact on the more or less heated walls assists the atomization, constitute two important boundaries between which the development of Diesel engines has progressed and is still progressing. Recently, however, ground has been gained for the view that the contact of the fuel with any metal surface, even when wet with lubricating oil, only causes imperfect combustion and should therefore be avoided as much as possible. In fact, an air envelope which encloses the combustion zone on all sides offers a suitable means for restricting to a minimum the formation of products of incomplete combustion and the direct transmission of heat to the metal walls, even in much-divided combustion chambers.

Chemists and engineers agree that the completeness of the combustion in carburetor engines is due to the perfectly uniform distribution of the fuel in the air in a definite proportion. Since homogeneous fuel mixtures can best be produced with gases and vapors, they can also be most readily and completely burned to carbon dioxide and water, only a slight excess of air being required. On the contrary, the situation is much less favorable in the mixtures of fuel and air with which we have to do in Diesel engines, the mixtures being less homogeneous and differing more in the size of the fuel drops. There is therefore a less capacity for perfect combustion than in gas and vapor mixtures. The combustion is less rapid and requires a greater excess of air. All these disadvantages increase with the mean size of the drops. With the ignition, however, there begins automatically a more rapid formation of vapor in the neighboring layers of the mixture. To a certain degree, the flame automatically generates the fuel mixture in which it advances and contributes, though in an imperfect manner, to the homogeneity of the mixture. The flame spreads more rapidly in the portions of the combustion chamber where the fuel mixture is richer than where it is poorer. Thus the fuel drops in the richer portions are not allowed time enough for complete combustion. Consequently, the combustion will be complete only in the poorer portions of the mixture, while in the richer portions, products of incomplete combustion will be formed or the fuel will remain unburned, namely, where too large drops are present in too large numbers.

Technical engineers are not willing to accept these conditions as unalterable, but are seeking some practicable remedy. Just as, in carburetor engines, the air stream in the intake pipes, adjusted at certain negative pressures and velocities, has proved to be the best means for distributing the fuel most uniformly in the mixture, so air currents and eddies are also utilized in many ways in solid-injection Diesel engines for equalizing the mixture, even after ignition. This method has already met with such practical success, that many have reached the conclusion that solid injection under high pressure, with its sensitive nozzles and fuel pumps (or, in other words, the fine atomization of the fuel when first introduced), is unnecessary and that the solid-injection process can be employed with a comparatively coarse spray. Instead of the hitherto widespread deplorable lack of appreciation of the processes of flow in the combustion chambers of Diesel engines, much interest is now being manifested in these phenomena. There is still a partial lack in accurate information, so that mistakes are yet possible. It has always been possible, however, to obtain practical results by systematically altering the air currents.

Thus Hesselman determined the effect of the circulation of the air around the cylinder axis on the radially sprayed fuel (Hesselman, "Hochdruckölmotor mit Einspritzung des Brennstoffes ohne Druckluft," Z.d.V.D.I., 1923, pp. 658-662. For a translation of Hesselman's article, see N.A.C.A. Technical Memorandum No. 312). Since he used only a simple, centrally located injection nozzle, he was compelled by the diameter of the combustion chamber to work with round jets. The number of holes in the nozzle was varied between 4 and 8, the best results being obtained with 5 holes. The striking or carrying distance of the individual fuel jets, aside from their round cross section, is probably also somewhat increased by the fact that the piston is raised a little in the middle, whereby, toward the end of the compression stroke, the air in the cylinder axis is forced outward.

By turning the stem of the air-intake valve, which is provided with a deflecting screen (Fig. 21, lower right), a definite direction, with reference to the neighboring portions of the cylinder wall can be imparted to the air flow during the injection stroke, thereby changing its initial velocity (which is approximately the component of the air-intake velocity falling in the direction of the cylinder circumference). The diagram shows how the fuel consumption is affected by the adjustment angle of the intake valve. It is obvious that the fuel consumption is most favorable when this angle is 75°. From this fact, Hesselman concludes that the air charge during the injection process must describe the angle at the center included between two adjacent fuel jets and estimates the circumferential velocity at 8 m (26 ft.) per second. Therefore, this does not represent the maximum circumferential velocity which

Hesselman could obtain in his experiments. The maximum circumferential velocity moreover, increased the fuel consumption by about 30%.

Hesselman's engine works with a compression of 28 atm. Hesselman assumes, however, that larger engines could work with 25 atm. compression and perhaps with still lower compressions in warmer regions.

The ignition point affected the fuel consumption in Hesselman's experimental engine only in a subordinate degree. The ignition point was so adjusted that the engine worked without vibration and gave mixed indicator diagrams. Hesselman could discover no relation between the advance injection angle and the revolution speed of the engine.

While Hesselman was determining the retardation of the ignition with reference to the injection point, with the aid of a deflecting plate placed under the mouth of the injection nozzle, he was also able to determine that the ignition begins considerably sooner with a smaller number of holes than with a larger number. The condensation of the oil, as regards space, and its greater concentration in the chemical sense also hastened the beginning of ignition in his experiments, or lessened the so-called "ignition shift" (designated by Hesselman as the "crank angle γ"), a very important fact, especially for high-speed engines. Hesselman established an angle of 3.5° (corresponding to 1/700 second), as the minimum value for the "ignition shift," though ordinarily this angle is larger (corresponding to 1/500 - 1/400 sec.).

In connection with one of the many fuel pumps devised by Hesselman, two important tasks connected with the injection process may be mentioned: first, how a very sudden beginning of the combustion, combined with heavy shocks, is to be avoided and, secondly, how the after-dripping of the injection nozzles can be reduced to a minimum.

Although changes in the magnitude of the injection pressure during the process of injection need not be condemned on principle (because they cause the fuel jet to "breathe," i.e., to facilitate scattering laterally to its axis), special caution must, however, be exercised with respect to those parts of the pressure curve within the pump diagram, where the pump pressure, necessary for the most favorable fuel velocity, has not been attained or has already fallen below. In order to assure the requisite freedom of motion for the beginning and ending of the injection process, Hesselman added a second piston to the pump shown in Fig. 22. It is located in a chamber immediately adjoining the pressure valve of the pump and has a smaller stroke volume than the main piston. Like the latter, it is operated by a cam in such manner that it is driven forward at the beginning of the injection process, then remains stationary during most of the injection process and, at the conclusion of the injection process, it is returned by a spring to its original position.

Since the curves of the cam can be changed independently of one another and the total fuel delivery is composed of the individual action of the pump pistons, there are many possibilities to alter both the ascending and the descending curve of the pump pressure diagram and to determine the most favorable course for both. Hesselman states that it is desirable to keep the fuel pressure small at first, to let it increase gradually and, at the end, to let it fall rapidly. This is similar to the fuel-injection process in air-injection Diesel engines, for which a gradually ascending but rapidly descending cam has been found to give the best results.

The close connection between the pump and the injection must, however, not be lost sight of. It is therefore advisable to proceed with caution in generalizing from the above statement of Hesselman.

In the designing of solid-injection engines, some foreigners have adopted devices which at first seem strange to the German way of thinking. They deserve serious consideration, however, since their success cannot be contested and no conclusion of their development is yet visible. I am thinking of the engines of Price and Banner. Americans are proud of these types, as Englishmen are of the Scott-Still engines. They think, not entirely without reason, that something original has sprung into existence on American soil.

The nature of Price's engine is best understood by comparison with its forerunners, the American hot-bulb engines, especially with those of the De La Vergne Company with hot bulb located laterally and perpendicularly to the axis of the cylinder, with which Price, from his employment as engineer for this company, was thoroughly familiar, even with respect to the injection nozzle. For lack of time, I cannot here go into the history of the origin of Price's engine, but must confine myself to a brief explanation of one engine made by the Price Engine Company of Philadelphia (Fig. 23).

First of all, it is noticeable that a flat, cooled plate, or "heat shield," is placed between the working cylinder and the cylinder head in which the inlet and exhaust valves are situated. In its inner frontal surface, engaging in the cylinder liner, it has a circular opening whose cross section is about one-fifth of the piston area, while on its outer surface in contact with the cylinder head, it has an oblong opening, which is composed of a large central circle and two smaller lateral circles, whose centers lie in the axes of the inlet and exhaust valves. Adjacent to the openings in the frontal surfaces there are short cylinder surfaces which, in turn, are connected with one another by conical surfaces. In this manner, two opposite conical antechambers with inclined axes are formed, which adjoin a diffuser with a circular neck.

The heat shield, like the cylinder, is provided with channels through which the cooling water flows close to all the heated surfaces. The ignition of the fuel is therefore far more dependent on the temperature of compressed air than on the temperature of the metal walls.

A spraying nozzle is located in the longitudinal axis of each one of the two antechambers. The conical jets produced by them have a medium apex angle, so that the direct wetting of the antechamber walls by the jets is avoided. Inside the jets the fuel is uniformly divided, according to Price, into a very fine spray, for the "uniformity of the jet" is, so Price maintains, especially important for the satisfactory operation of the engine. For this purpose, the nozzles have inner spindles provided with spiral grooves and the injection channel preceding the mouth of the nozzle has a length which is a multiple of its diameter, the diameter of the nozzle opening being 0.036" for an engine of 19" cylinder diameter, 24" stroke and 235 R.P.M. The fuel is injected into both antechambers with quite a large advance. The real reason for this is probably to be found in the comparatively low compression of 11 - 13.5 atm. at normal load and in the strong scattering action of the nozzles. The injection begins when the engine is overloaded, at an angle of 52° before the dead center and ends at 2° after it. At normal load, it begins earlier and ends later.

The two sprays encounter each other about in the cylinder axis and since, on account of the inclination of their axes, they possess a small downward velocity component in the cylinder axis, they meet the air stream generated by the diffuser during the compression stroke and mix with it principally in the marginal region produced by the cylindrical neck. It seems to be worthy of note that this neck is short and angular and that the working piston, at the end of the compression stroke comes very close to the surface of the heat shield, so that all but a small fraction of the air charge is driven through the neck and strong eddies are generated in the air stream on both its inner and outer surfaces.

At the beginning of the combustion, the burning gases press from both sides toward the cylinder axis and, since they suffer many changes in their direction of motion and mean cross sections, on the way to the working cylinder, thus generate, between the neighboring stream filaments, strong displacements and eddies, which assist the combustion. The fuel consumption of 190 g (6.7 oz.) per HP_e/hr. (converted to a lower heat value of 10,000 heat units per kilogram at an engine power of 100 HP.) seems very favorable. The combustion takes place, as shown by the original indicator diagrams, in the form of a simple explosion. At a compression pressure of 11 atm. the combustion pressure was 27 atm. (Fig. 25, right).

This engine also resembles an explosion engine in two other respects. The air charge introduced during the suction stroke, which, moreover, helps to expel the exhaust gases from the combustion chambers, can be regulated by an air shutter. Then, however, there is installed in the axis of the cylinder head an auxiliary spark plug, which is set in operation when the compression pressure, in starting from the cold condition, is not sufficient to effect automatic combustion of the fuel mixture, and which remains in operation until the engine has become sufficiently heated. This spark-plug is a proof that an explosive mixture is formed by the two spraying nozzles in conjunction with the diffuser and that the air flow produced by the neck forces this mixture against the cylinder head and the spark plug.

As follows from the original indicator diagrams, spontaneous combustion often begins at a considerable distance from the dead center, which might give rise to heavy shocks in the driving gear, a disadvantage which also occurs occasionally in other solid-injection Diesel engines, when an explosive drive, dependent on spontaneous combustion, is produced in too pure a form.

Banner's engine, built by the Falk Corporation in Milwaukee (Fig. 24) forms a fine companion piece to Price's engine. Banner's engine has a mixed indicator diagram, half Otto and half Diesel. Viewed externally, it is similar to Price's engine, in that it has the same arrangement of the inlet and exhaust valves in the cylinder head, a heat shield, two opposite combustion chambers, two injection nozzles and a diffuser common to both combustion chambers, but the shapes of the combustion chambers, fuel jets and diffuser neck are logically adapted to one another. The combustion chambers are shallow with rectangular cross sections; the fuel jets are flat and fan-shaped; the neck is narrow and rectangular. Not only has more pains been taken to prevent the fuel from coming in contact with the walls of the combustion chambers, but the pure explosion drive is intentionally left, for the uniform distribution of the fuel in the combustion air, which Price thought desirable, can no longer be the question. The flat fuel jets and their enveloping vapors are, instead, bedded in air on all sides until the combustion begins and are first dispersed and mixed with the air as a result of the combustion and during their passage through the diffuser. This is shown in the mixed form on the indicator diagram. For an engine of 19" cylinder bore, 28" stroke and 200 R.P.M., the indicator diagram shows a pressure of 27 atm. at the end of the compression (Fig. 25). The first partial combustion, during which about 24% of the heat contained in the total fuel charge is liberated, raises the pressure to 39 atm. (1.45-fold), which is maintained, during a piston stroke of 6%, to the beginning of the expansion, while the remaining 70% of the total heat in the fuel is being liberated.

In Banner's engine, the injection begins 8.5° before the dead center. At this instant the air in the combustion chambers is moderately agitated and there is no further inflow of the combustion air from the working cylinder through the diffuser. The diffuser, therefore, has relatively little effect on the formation of the mixture in its inception during the upward stroke of the piston (in any event less than in Price's engine), while it is of decisive importance, during the downward stroke of the piston, for the further mixing of the air with the partially burning and partially unburned fuel charge.

Since the diffuser neck is just as wide as both combustion chambers, but is narrow in the direction of the axis of the two jets, there is (in comparison with the circular cross section of the diffuser neck in Price's engine having the same cross-sectional area) an increased marginal action whose influence extends at least as far as the middle of the rectangular cross section.

Banner's engine is therefore characterized by an excellent utilization of the air charge and functions without smoking, with an air excess of 10-20%. It can, therefore, stand much overloading. Its mean indicated pressure, under normal load, is 7 atm./cm² (99.56 lb./sq.in.) and its mean effective pressure is 6.3 atm./cm² (89.6 lb./sq.in.). According to recent news from America, Banner has succeeded in obtaining mean pressures up to 11 atmospheres, a result of especial importance for automobile engines, because he did not have to make use of the piston cooling, which was originally provided, and was also able to dispense with scavenging, for which he had likewise made provision in his experimental engine. The heat shield, which helps to protect the piston against the high combustion temperatures, is rightly entitled to its name. This latest important result was obtained by a modification of the combustion chambers as shown in Fig. 24, which facilitates their scavenging and improves the action of the diffuser. The form and arrangement of the injection nozzles, which belong to the type shown in Fig. 9 (Part I - Technical Memorandum No. 356), were also improved. According to Banner, the shape of the indicator diagram is closely related to the short injection period and the small injection angle, the shape of the fuel sprays, combustion chambers and diffuser. We are not informed as to whether Banner (like Heidelberg and Hawkes) has investigated the effect of the injection angle and of the shape of the cam on the process of combustion.

Translation by Dwight M. Miner,
National Advisory Committee
for Aeronautics.

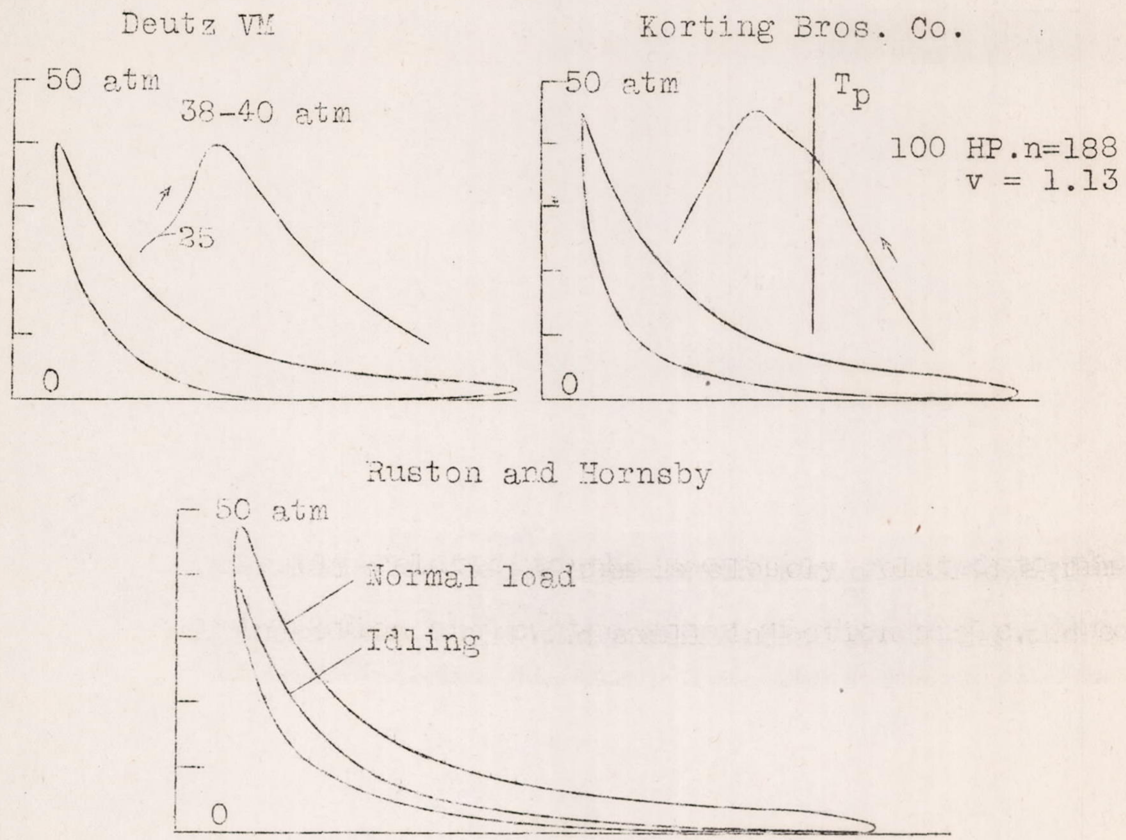
Deutz VM
50 atm
38-40 atm
25
0
Korting Bros. Co.
50 atm
T_p
100 HP.n=188
v = 1.13
0
Ruston and Hornsby
50 atm
Normal load
Idling
0

Fig.16

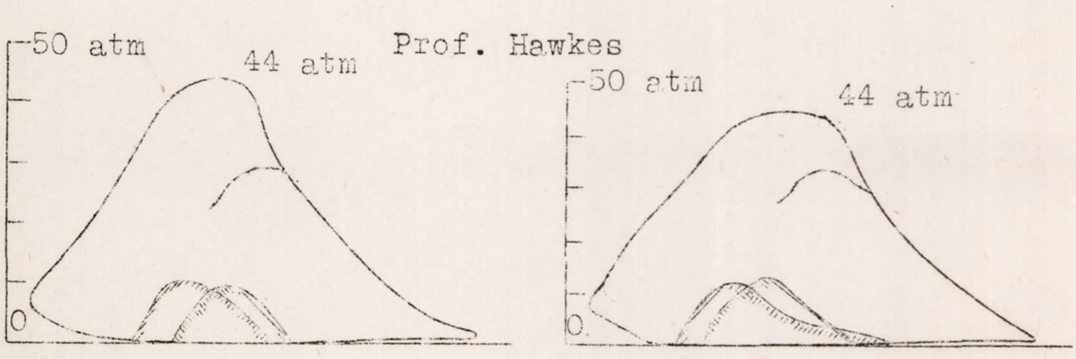
Prof. Hawkes
50 atm
44 atm
0
50 atm
44 atm
0

Fig.17

Dr. Heidelberg - "Deutz"

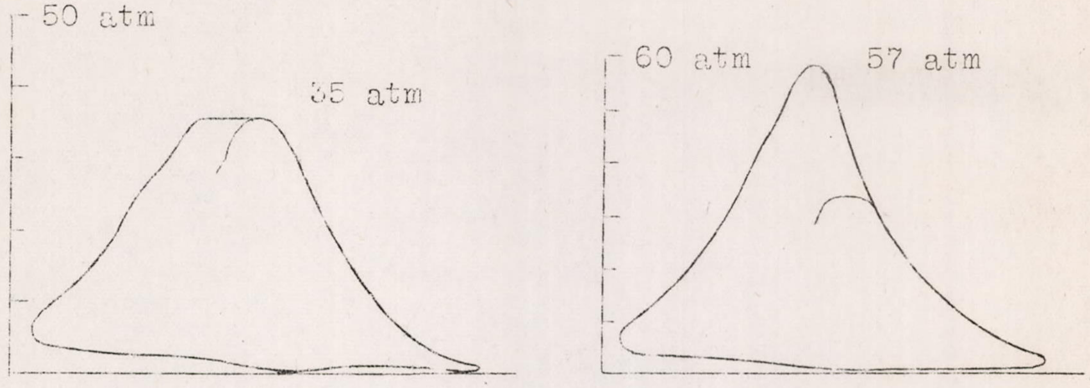

Fig.18

Banner Price

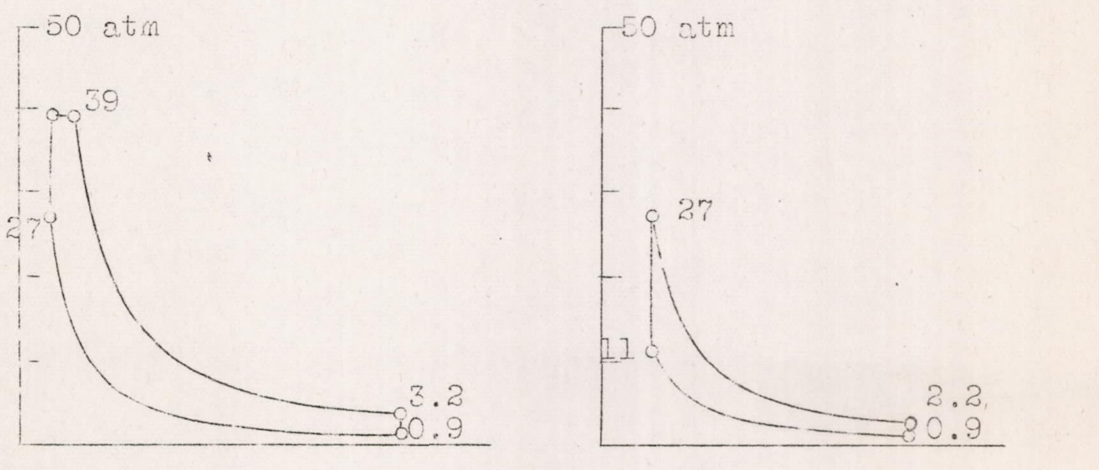

Fig.25

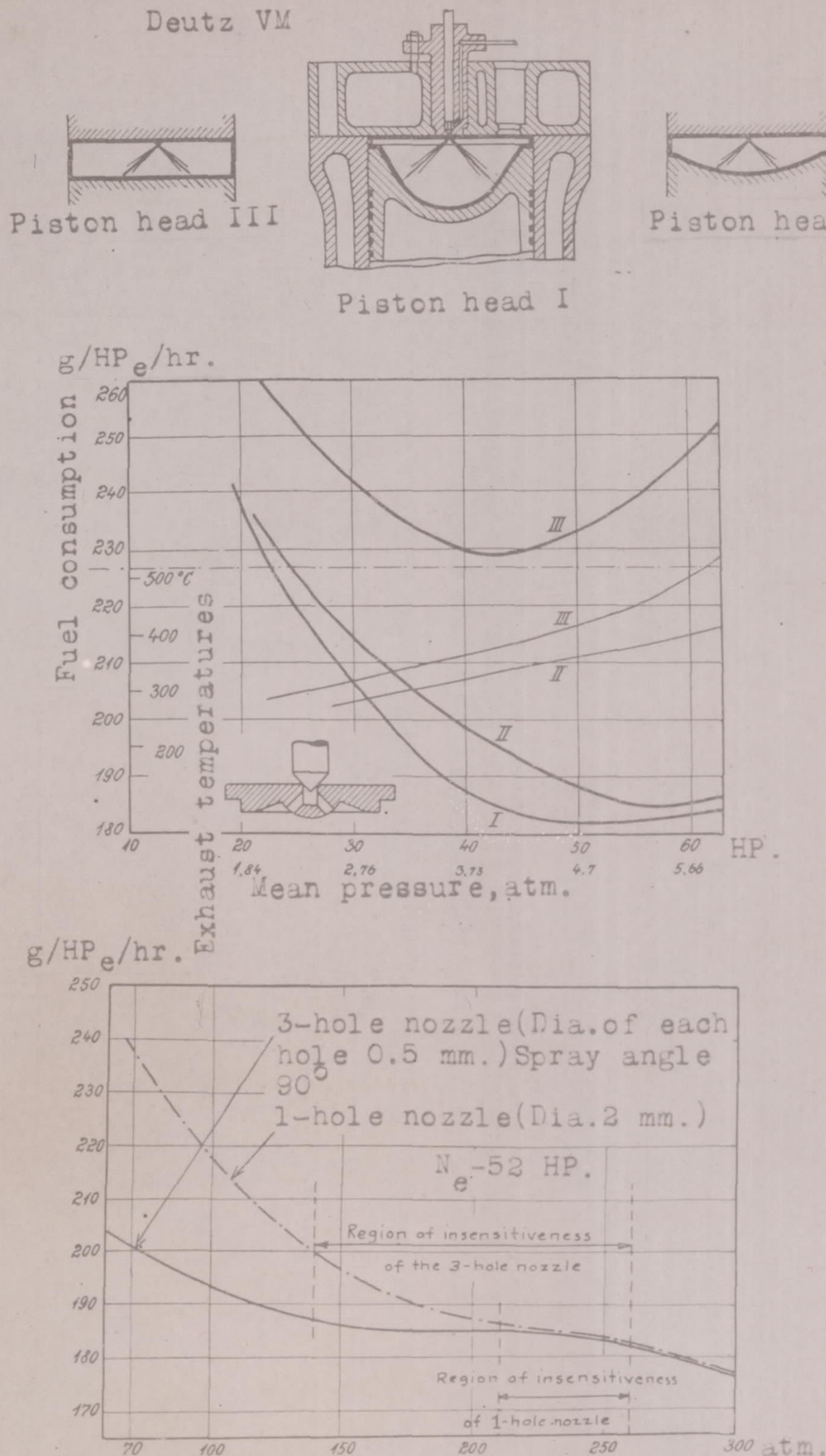
Deutz VM
Piston head III
Piston head II
Piston head I
g/HPe/hr.
Fuel consumption
Exhaust temperatures
500°C
400
300
200
260
250
240
230
220
210
200
190
180
III
III
II
II
I
10
20
30
40
50
60
HP.
1.84
2.76
3.73
4.7
5.66
Mean pressure, atm.
g/HPe/hr.
250
240
230
220
210
200
190
180
170
3-hole nozzle(Dia.of each hole 0.5 mm.)Spray angle 90°
1-hole nozzle(Dia.2 mm.)
Ne-52 HP.
Region of insensitiveness of the 3-hole nozzle
Region of insensitiveness of 1-hole nozzle
70
100
150
200
250
300
atm.

Fig.19

Deutz VM

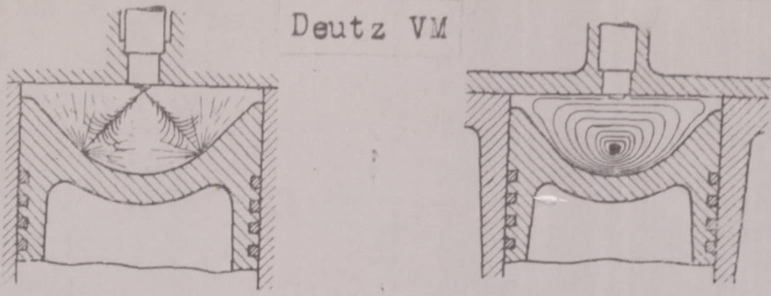

Fig.20

Hesselman

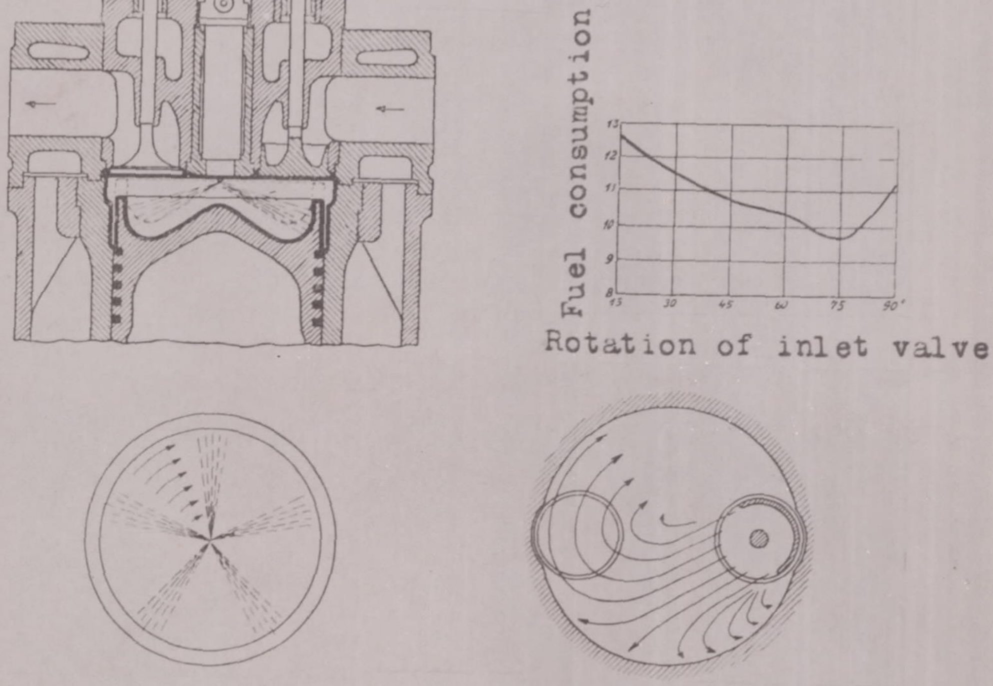

Fig.21

Hesselman

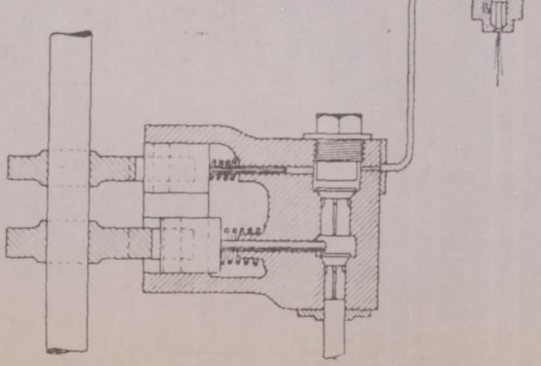

Fig.22

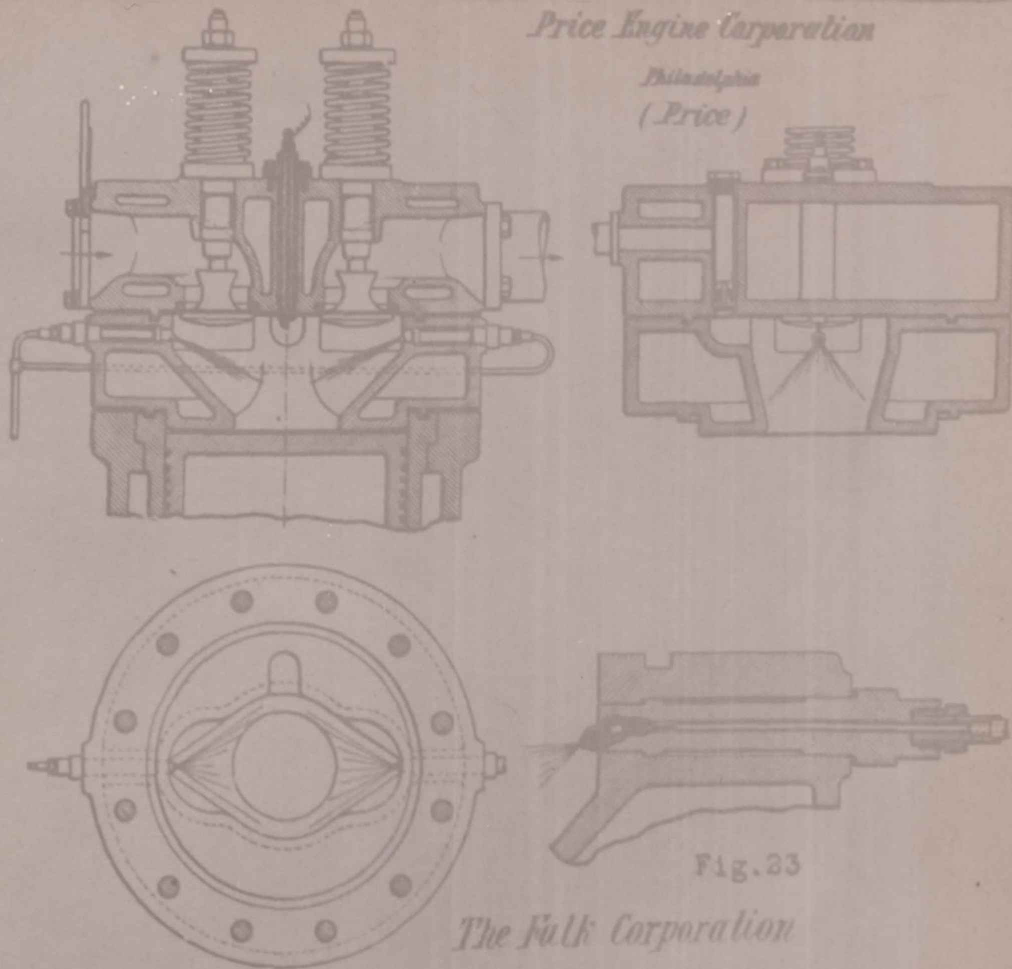
Price Engine Corporation
Philadelphia
(Price)

Fig.23

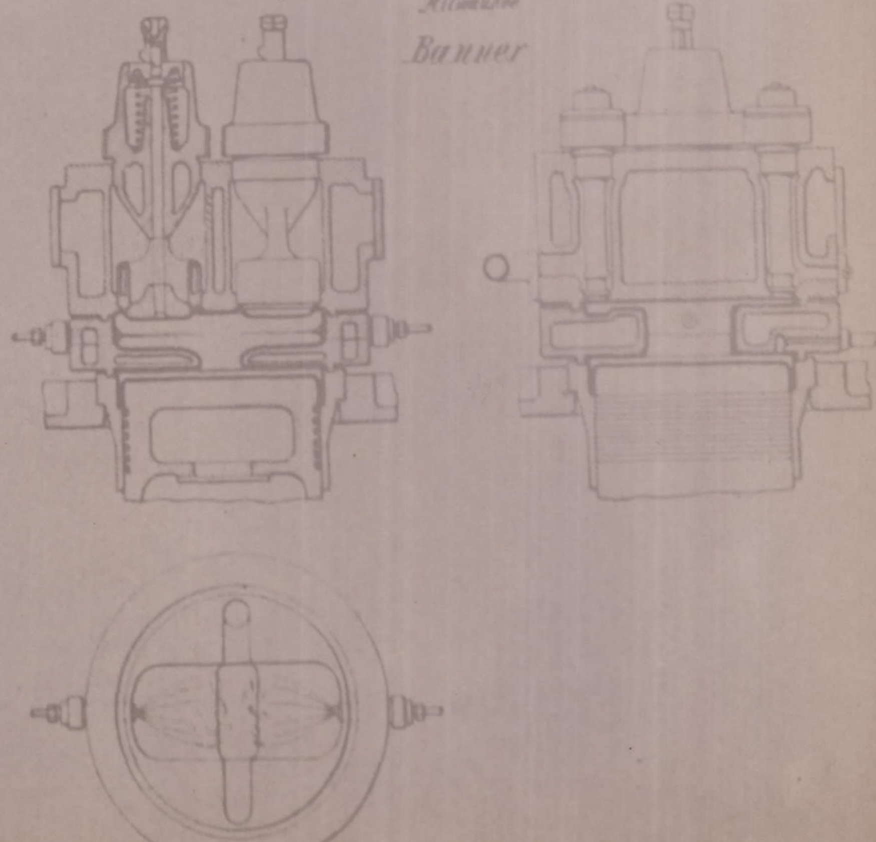
The Falk Corporation
Milwaukee
Banner

Fig.24