TECHNICAL MEMORANDUMS

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

No. 391

# EXPERIMENTS ON SELF-IGNITION OF LIQUID FUELS

By Kurt Neumann

From "Zeitschrift des Vereines deutscher Ingenieure," August 7, 1926.



Washington
December, 1926

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS.

TECHNICAL MEMORANDUM NO. 391.

# EXPERIMENTS ON SELF-IGNITION OF LIQUID FUELS.*

By Kurt Neumann.

The conversion of the chemical energy of the fuel into heat exhibits the same outward characteristics in airless-injection Diesel engines as in all internal combustion engines. Whatever the details of the process may be, three main divisions can be distinguished:

1. The time from the entrance of the fuel into the cylinder until ignition occurs;

2. The main combustion from the beginning of the pressure increase to the maximum combustion pressure;

3. The after-combustion during the expansion.

In order to limit the unavoidable heat loss during the chemical conversion to a minimum, it is endeavored to convert the chemical energy into heat energy as quickly as possible, since the heat loss (under otherwise like conditions) is proportional to the time consumed. The thermodynamic requirements must at times be restricted; for example, when the maximum pressure in the working process can not be allowed, for reasons of economy, to exceed a certain limit.

---

* "Untersuchungen über die Selbstzündung flüssiger Brennstoffe," a lecture delivered at the Diesel session of the Association of German Engineers. From "Zeitschrift des Vereines deutscher Ingenieure," August 7, 1926, pp 1071-1078.

The time within which the individual processes take place is of great importance for the working process. It is not sufficient therefore for the energy conversion to take place in the desired manner only, but it must also accord with the time intervals available in the functioning of the engine. The difficulties are very obvious, if we remember that these time intervals usually amount to only a few hundredths of a second.

If, therefore, we wish to take a deeper look into the cause and effect of the energy conversion in airless-injection engines, in order to obtain a criterion for the chosen working method, we must take into consideration the time relations between the individual processes. Observations of the engine alone do not suffice to obtain the necessary bases for a critical analysis of the processes. We could probably determine in this manner as to how correct and economical the method is, but we would seldom be able to draw conclusions regarding the time relations.

The course of every chemical reaction is governed by the speed of the reaction. It depends on the temperature and concentration of the components which take part in the reaction. This speed generally increases very rapidly with the temperature.

In oil engines, as distinguished from gas engines, the situation is rendered more complex by the fact that the outflowing fuel and air exist in two different phases. The rapidity of the conversion therefore depends on:

1. How quickly the materials pass from the liquid and gaseous phase to the boundary surface in which the reaction takes place;

2. How rapidly the chemical action proceeds;

3. How quickly the reaction products are removed from the reaction chamber.

The fuel, converted into drops by spraying, at first forms, with the combustion air, a non-homogeneous mixture, which absorbs heat up to the ignition point. If, at the same time, the fuel is well vaporized, the mixture rapidly assumes the homogeneous state, due to the easy diffusibility of vapors and air. At the instant of ignition, there is usually a "cloud." For the characterization of this "cloud," it is not sufficient to give the pressure, temperature and mixture ratio, as in the case of gases alone, but we must also know the size of the drops and the proportion of the fuel in the liquid and in the gaseous state.

It is already obvious that the conversion of energy in liquid fuels is much more complicated than in gases. The injection process and the volumetric concentration of the fuel depend on the state of motion of the combustion air, while the heat absorbed by the fuel depends on the thermal relations. In other words, both hydrodynamic and thermodynamic problems must be solved. Their solution is rendered difficult by the fact that the heavy oils are complex hydrocarbons, concerning the molecular construction of which we have no definite knowledge.

In order to obtain an insight into the processes which could be mathematically utilized, they had to be produced independently of the engine in a special experimental apparatus which would render it possible to vary the different conditions systematically. We utilized as explosion chamber a cylindrical, electrically heated bomb made from special steel by the Krupp Company (Fig. 1), into which the fuel was injected all at once. The initial pressure and temperature could be increased at will up to about 30 atm. (426.7 lb. per sq.in.), and 700°C (1292°F). The temperature was measured by thermocouples, and the curve of the pressure plotted against the time was recorded on a rapidly revolving sooted glass drum, on which a fuel-needle lift diagram and the vibrations of a tuning fork were simultaneously recorded, the latter being for the purpose of measuring the time. A fan was installed in the bomb, in order to impart a whirling motion to the air. All the details of the experimental apparatus will be described in the dissertation soon to be published by my assistant Hartner-Seberich, who performed the experiments.

The fuel was gas oil. In the different experiments, the initial pressure and temperature were measured both with and without the use of the fan. Fig. 2 is a pressure-time diagram made at 15 atm. (213.35 lb. per sq.in.) initial pressure, and 375°C (707°F). This was the first record of a combustion in which the chemical energy of the fuel was not released by an electric spark, but by the self-ignition of the fuel. The three principal parts of the process are shown in Fig. 2. The time from the beginning of the entrance of the fuel to the ignition (i.e., the ignition delay) is $z_s = 0.115$ second. The time from the beginning of the increase of the initial pressure $p_0 = 14.9$ atm. (211.93 lb. per sq.in.), to the maximum combustion pressure, $p_{max} = 30.7$ atm. (436.66 lb. per sq.in.), is $z_v = 0.02$ second. The ratio $z_s/z_v = 5.75$, according to which the ignition delay is here almost six times as large as the duration of the main combustion, illustrates the great importance of the processes preceding ignition.

Therefore, we first directed our attention to the discovery of the conditions which affect the ignition delay. In recent years important articles have appeared on the ignition processes in Diesel engines. On the basis of these articles it may now be safely assumed that the fuel undergoes certain physical and chemical changes before self-ignition. Too little attention has been paid, however, to the real cause of the changes, the transmission of heat from the overheated combustion air to the fuel mixture. Our experiments show conclusively that the ignition process is controlled by the transmission of heat from the air to the fuel. Our experimental apparatus was therefore so constructed at the outset as to simulate closely the actual engine processes, although this involved great experimental difficulties.

The application of thermodynamics to the processes during the ignition delay leads to a system of differential equations, from which we can calculate the temperature curve of the fuel in terms of the time from the beginning of the fuel injection to the ignition, if we assume that the fuel particles immediately take on the form of drops. A more thorough investigation shows that the fuel jet remains intact but a very short time.

The decisive equation reads symbolically

$$\frac{d^2 t}{dz^2} + P_1(z) \frac{dt}{dz} + P_2(z) = 0$$

This is a linear differential equation of the second order, whose coefficients are known functions of the time and contain all adequate values which characterize the transmission of heat. The solution is

$$t = \int e^{-\int P_1(z)dz} \left( \int P_2(z) \cdot e^{+\int P_1(z)dz} dz + C_1 \right) dz + C_2 \ [^\circ C]$$

or, after carrying out the integrations and determining the two integration constants,

$$t = t_0 + b_1 z + b_2 (1 - e^{-b_3 z}) \ [C], \text{ s. Fig. 3.}$$

We know that the temperature of the fuel increases with the time, according to an exponential function, from $t_0$ at its emergence from the injection nozzle. If the ignition temperature $t_s$ is reached, then the time elapsed up to that point represents the ignition delay $z_s$. The temperature rises rapidly or slowly according to the experimental conditions. The former corresponds to a shorter, the latter to a longer ignition delay. We have accurately measured ignition delays between 0.01 and 0.6 second. The known time functions $P_1(z)$, $P_2(z)$ and the constants $b_1$, $b_2$, and $b_3$ of the solution include the mixture ratio of fuel and air, the specific heat, the coefficient of heat conductivity, the heat of vaporization and the radii of the drops, to mention only a few quantities. The radii were computed, at the beginning of the jet disruption, from the surface tension and relative velocity, according to the directions of Triebnigg.* Although the functional structure of these constants is rather complicated, we can clearly recognize in them the effect of the different variables on the course of the fuel temperature during the ignition delay.

We have given special attention to the vaporization of the fuel during the injection, since opinions differ greatly on this point, they being all based only on rough estimates. The experiments rendered it possible to obtain the first quantitative data. Fig. 4 shows the results of two series of experiments, in which the initial pressure of the air in the cylinder was kept constantly at $p_0$ = about 8 atm. (113.78 lb. per sq.in.), and the initial temperature of the air was varied between 280 and 600°C (536-1112°F). The first series of experiments was performed with the air at rest; the second, with it in active turbulent motion. The ignition point was considered the lowest

*"Der Einblase- und Einspritzvorgang bei Dieselmaschinen," published by Julius Springer, Vienna, 1925.

temperature at which ignition occurred. In the mathematical computation, the ignition temperature was determined from the condition that the temperature of the air $\vartheta_0$ = ignition temperature $t_s$ for the ignition delay $z_s = \infty$. Under the chosen conditions at 8 atm., $t_s$ = 265 and 306°C (509 and 582.8°F).

The higher therefore the initial temperature of the compressed air lies above the ignition temperature $t_s$ of the fuel, the shorter the ignition delay. This depends, as can be demonstrated, on the rapid increase in the reaction speed with the rise in temperature. Without this influence, the rapid decrease in the ignition delay can not be explained alone by the physical effect of the heat transmission resulting from the difference in temperature between the air and the fuel.

Furthermore, all ignition delays between $z_s = 0$ (about) and $\infty$ can be obtained with or without air turbulence, but, in a lively air flow, the temperature range of the ignitions is considerably restricted, in that the ignition delay decreases far more rapidly with rising initial temperature of the air than when the air in the cylinder is at rest. The temperature range in the former case is only about 100°C (180°F), while, with the air at rest, the temperatures lie between 265 and 608°C, or a range of over 300°C (540°F).

The rapidity of heat transmission from the air to the fuel is increased by an active flow in the combustion chamber of a Diesel engine and the temperature of the fuel drops rises faster. The reaction speed also increases with increasing temperature. The objection is not justified that active commotion causes a greater heat loss to the cooling water. The fact is thereby overlooked that the effect of the reaction speed on the rapid conversion of chemical energy into heat, i.e., the obtention of shorter ignition delays, far outweighs the heat loss in strong turbulence due to a rapid rise in the temperature.

The turbulence, which is of fundamental importance for the course of the combustion, can therefore not be dispensed with, even for the ignition. Moreover, it tends to render the fuel mixture more homogeneous during the injection, which, under some circumstances, may cause simultaneous ignition at different points and thereby shorten the combustion.

It has often been denied that any vaporization of the fuel occurs during the process of injection. That this does occur, however, is demonstrated by the fact that the fuel drops, on entering the combustion chamber of the engine, find a space filled with highly heated air whose vapor density is zero at first, but which can, in the course of time, increase to the degree of saturation corresponding to the temperature.

The mathematical analysis of the vaporization process of heavy fuel oils (Figs. 5-7) here leads to the result that the radius r of the drops decreases according to an exponential function of the time

$$r = r_o\, e^{-\frac{k\, \mu_{1q}\, c_m''}{\gamma_{1q} \cdot r_o} z} \quad [m]$$

The vaporization speed is affected by the molecular weight $\mu_{1q}$, the saturation $c_m''$ of the vapor on the surface of the drops, the specific weight $\gamma_{1q}$, the radius $r_0$ of the drops at the beginning of the process and the vaporization constant $k$, which latter depends on the physical properties. In order to obtain quantitative results, we accordingly had to determine the molecular weight of the fuel, the course of the vapor tensions and the vapor constants, the latter by manometric methods.

Figs. 6-7 show the relative diminution in the volume, $\frac{V_0 - V}{V_0} 100$, of the drops at the instant of ignition with increasing ignition delay and rising temperature of the air. The quantity of fuel evaporated is accordingly proportional to the time. Relatively more fuel evaporates at lower temperatures of the air, due to the longer ignition delay. The evaporation rate is increased by turbulence, this being in accord with the principles of kinetics. In general, however, the quantity of fuel evaporated during the ignition delay was small. The maximum decrease in the volume of the drops was found to be about 5% with a long ignition delay and great turbulence. With the minimum ignition delay ($z_s \sim 0.018$ s, $\frac{V_0 - V}{V_0} 100 = 0.1\%$) no appreciable evaporation takes place.

The tests lead to the important conclusion that no preliminary evaporation of the fuel is necessary for producing the ignition. We will consider later, as to how much the progress of the combustion is affected by the degree of evaporation attained previously and during the combustion.

From the course of the fuel temperature $t = f(z)$, computed by means of the differential equation, and from the similarly computed weight loss of the drop by evaporation during the ignition delay $z_s$ the amount of heat can be determined which is required for heating the fuel drop to the ignition temperature and for the evaporation of the fuel up to this instant. Per unit weight this heat is

$$[Q]_0^{z_s} = \int_0^{z_s} \left(\frac{\partial Q}{\partial z}\right)_{air} dz + \int_0^{z_s} \left(\frac{\partial Q}{\partial z}\right)_{vap.} dz$$

or $$[Q]_0^{z_s} = c_{1q}\, b_2 \left(1 - e^{-b_3 z_s}\right) + \frac{3\ k\ \mu_{1q}\ q_v\ c_m''}{\gamma_{1q}\ r_o}\ z_s \left[\frac{\text{kcal}}{\text{kg.}}\right]$$

The first summand is the heat acquired by the drop through conduction. The second summand is the heat required for vaporization, which does not exceed 2% of the total heat absorbed, even in the most unfavorable case (Fig. 8). The heat required up to the ignition point, $[Q]_0^{z_s}$ = 129 kcal/kg (232.18 B.t.u.) is transmitted to the injected fuel at a rate proportional to the excess of the temperature $\vartheta_0$ of the air in the cylinder over the ignition temperature $t_s$ = 265°C (509°F) of the fuel. For this reason high temperatures of the air give short ignition delays. For experiment 1, $\vartheta_0 - t_s$ = 343°C (617.4°F) and $z_s$ = 0.0150 sec.

For experiment 7, $\vartheta_0 - t_s$ = 19°C ( 34.2°F) and $z_s$ = 0.3822 sec.

The second series of tests with air turbulence showed the same characteristic course and other values only for the temperature difference $(\vartheta_o - t_s)$ and the ignition delay $z_s$. It is nevertheless obvious as to how essential the effect of turbulence is on the heat transmission and thereby on the time required for ignition, if the mean coefficients of heat transmission from the air to the fuel drops during the ignition delay are calculated for both series of tests. Referred to unit weight,

$$\alpha_m = \frac{1.2 \times 10^3 \ \gamma_{fl} \ r_o \ [Q]_o^{z_s}}{(\ \ _o - t_m) \ z_s} \left[\frac{\text{kcal}}{m^2 h^o C}\right]$$

whereby the mean fuel temperature is

$$t_m = t_o + b_2 - \frac{b_2}{b_3 \ z_s}\left(1 - e^{-b_3 z_s}\right)[^oC]$$

Fig. 9 shows the rapid increase in the coefficient of heat transmission $\alpha_m$ with increasing air temperature. The same mean coefficient of heat transmission is obtained with turbulence at a much smaller temperature difference $\vartheta_o - t_s$, than when the air in the cylinder is at rest. For example, the temperature difference for $\alpha_m = 500 \ \frac{\text{kcal}}{m^2 h \ ^oC}$ with turbulence is only 70°C (136°F), but a difference of 224°C (403.2°F) is required for air at rest.

In the injection process in airless-injection engines, the rapidity of heat absorption by the fuel drops is of far-reaching importance. If we differentiate the expression obtained for the heat of ignition $[Q]_o^{z_s}$ according to the time, we immediately obtain the rate of heat absorption by the fuel during the ignition delay:

$$\frac{dQ}{dz} = c_{lq}\ b_2{}'\ e^{-\frac{C_1}{\theta_0} z} + \frac{3\ k\ \mu_{lq}\ q_v\ c_m{}''}{\gamma_{lq}\ r_o} \left[\frac{kcal}{kgh}\right]$$

If we at first disregard the second term, which corresponds to the vaporization and has but a slight effect, the rate of heat absorption is controlled by the temperature variable $b_2{}'$ of the fuel and by the absolute temperature $\theta_0$ in the cylinder at the beginning of the injection.

The temperature variable $b_2{}'$ is itself a function of the air temperature and, above all, also of the state of motion of the air. It increases rapidly with increasing turbulence. Hence the lower the ignition point $t_s$ of the fuel and the higher the rate of fuel absorption by the fuel, just so much smaller is the required ignition time

$$z_s = \frac{c_{lq}\ (t_s - t_o)}{\left|\frac{\partial q}{\partial z}\right|_0^{z_s}} [s].$$

In a Diesel engine, therefore, the ignition delay is inversely proportional to the temperature variable of the fuel, the temperature of the air and the turbulence of the flow in the combustion chamber at the beginning of the ignition.

The magnitude of the temperature variable depends chiefly on the degree that the temperature increase (resulting from the transmission of heat from the air to the fuel) is accelerated by the increasing reaction speed during the ignition delay. The rapid increase in the mean coefficient of heat transmission $\alpha_m$

with rising air temperature $\vartheta_o$, is therefore to be attributed not only to the physical process of heat transmission, but doubtless also to the chemical processes preceding ignition and especially to the increasing reaction speed. This explains the high value of $\alpha_m$, which otherwise in the transmission of heat from air to ordinary objects, would have an order of magnitude of only 4 to 10 $\frac{\text{kcal}}{\text{m}^2\ \text{h}\ {}^\circ\text{C}}$.

Fig. 10 shows how the individual quantities change from the beginning of the injection up to the instant of ignition. The gas oil is here injected at an initial temperature of 50°C (122°F) into quiet air at 364°C (687.2°F) and a pressure of 8 atm. (113.78 lb. per sq.in.). The ignition delay is 0.113 sec. Since $dQ/dz$ is variable during the ignition delay $z_s$, we can best get an idea of the rate of the heat absorption by the fuel by calculating the mean value of $\left(\frac{dQ}{dz}\right)_o^{z_s}$ and plotting this against $\vartheta_o - t_s$ (Fig. 11).

rom

$$\left|\frac{dQ}{dz}\right|_o^{z_s} z_s = \int_o^{z_s} \frac{\partial Q}{\partial z}\, dz$$

follows

$$\left(\frac{dQ}{dz}\right)_o^{z_s} = \frac{c_{1q}\ b_2}{z}\left(1 - e^{-\frac{C_1}{\theta_o} z_s}\right) + \frac{3\ k\ \mu_{1q}\ q_v\ c_m''}{\gamma_{1q}\ r_o} \left[\frac{\text{kcal}}{\text{kgh}}\right]$$

The values run similarly to those of the mean coefficient of heat transmission $c_m$, From an initial value of zero, $\left|\frac{dQ}{dz}\right|_o^{z_s}$ quickly increases, with increasing temperature, according to an exponential function, and all the more rapidly, the greater the turbulence.

The tests were also extended to higher initial pressures of the air in the cylinder, about 15 and 26 atm. (213.35 and 369.8 lb. per sq.in.).

Application to Airless-Injection Diesel Engines

The fact that the ignition processes are largely controlled by the transmission of heat from the air to the fuel must be expressed in the dependence of the ignition temperature on the determining conditions for the heat transmission, and all the clearer, the more the chemical changes recede in individual cases. In the engine, the air, highly heated by compression, imparts heat to the spherical fuel drops. According to our present knowledge of heat transmission, the heat absorbed by a drop must be a function of three index values:

$$Q = f\left(\frac{\alpha}{\lambda}\, r_0, \frac{\alpha z}{{r_0}^2}, \frac{r}{r_0}\right) \text{ [kcal]},$$

in which:

$\alpha$ is the coefficient of heat transmission;

$\lambda$ " " " " " conductivity of the liquid;

$r_0$ and $r$, the radii of the drops at the times 0 and $z$;

$\alpha = \lambda/c\gamma$, the temperature increase.

The experiments of Tauss and Schulte show that the ignition points of the heavy oils used in Diesel engines fall considerably as the pressure is increased.* This result can not be ex-

* Tauss and Schulte, "Ueber Zündpunkte und Verbrennungsvorgänge im Dieselmotor," Halle, 1924. Wilhelm Knapp, p. 49, "Zeitschrift des Vereines deutscher Ingenieure," 1924, p. 574.

haustive from the viewpoint of heat transmission. If we consider the second index value, it is obvious that the ignition point depends on the density of the air. In fact, it is then manifest that the ignition point does not depend on the pressure, as Tauss believes, but on the density of the air.

If, on the basis of the experiments of Tauss and Schulte, we determine the density

$$\gamma = \frac{10^4\ p}{29.3\ T_s} \left[\frac{kg}{m^3}\right],$$

belonging to every ignition temperature $T_s$ and to every pressure $p$, we can represent the dependence of the ignition point on the air density with great accuracy by an exponential formula

$$T_s = C\gamma^{-m} \ [°abs]$$

in which $C$ and $m$ are constants, that, for fuels composed principally of aliphatic hydrocarbons, fluctuate about a common mean value.

For the aliphatic series, we have:

| | C | m |
|---|---|---|
| Gasoline | 653 | 0.142 |
| Kerosene | 725 | 0.183 |
| Shale-tar oil | 767 | 0.169 |
| Lignite-tar oil | 692 | 0.148 |
| Mean | 709 | 0.160 |

Aromatic oils have somewhat different constants.

| | C | m |
|---|---|---|
| Gasoline | 1163 | 0.231 |
| Coal-tar oil | 977 | 0.193 |

Fig. 12 shows the dependence of the ignition point on the air density in accordance with the formula $T_s = 709\gamma^{-0.16}$. If the other physical constants of the heavy oils were taken into account, the spreading of the experimental points about the mean curve would be still less. It is perfectly evident, however, that the ignition point falls as the air density increases. The fall is considerable at first, but small subsequently. It can be expressed mathematically by the differential equation

$$\frac{d\,T_s}{d\,\gamma} = -\,11.32\gamma^{-1.16}\left[\frac{{}^{\circ}\mathrm{m}^3}{\mathrm{kg}}\right]$$

For

| | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| $\gamma$ = | 1 | 3 | 5 | 7 | 9 | 12 | 16 | kg/m³ |
| $\frac{d\,T_s}{d\,\gamma}$ = | -11.32 | -3.16 | -1.75 | -1.18 | -0.88 | -0.64 | -0.45 | °C m³/kg |

With an air density of 1 kg/m³ (.062428 lb./cu.ft.) the ignition point is accordingly lowered 11.3°C (20.34°F), but with an air density 9 and 16 times as great it is lowered only 0.88 and 0.45°C (1.58 and 0.81°F) respectively. If $\gamma$ exceeds 8 kg/m³ (.499 lb./cu.ft.), the ignition point sinks but very little. In Diesel engines $\gamma$ is always larger than 8 kg/m³ (.499 lb./cu.ft.). For final pressures of $p_k$ = 25 and 35 atm. (355.6 and 497.8 lb./sq.in.), $\gamma$ = 11.5 and 14.9 kg/m³ (.93 lb./cu.ft.).

For reaching the ignition temperature, no increase in the final compression pressure above 20 atm. (284.47 lb./sq.in.) would be necessary, were it not simultaneously required that the ignition delay be short. Our experiments show that the ignition delays are inversely proportional to the excess in the temperature of the air in the cylinder above the ignition point of the oil and to the velocity of the air flow in the cylinder. Our experiments with two mechanical injection engines and an ignition chamber engine show to what degree such engines conform to the calculation during the process of ignition.

In a 200 HP. Deutz 4-cylinder VM engine, the beginning of the injection at full load was changed between 47, 37, and 32° before the crank dead center. Fig. 13 shows the course of the pressure in the cylinder according to the indicator diagram, as plotted against the time. The final compression pressure $p_k$ was 27 atm. (384 lb./sq.in.) The temperature and density of the air during the compression was calculated on the assumption that, at the beginning, $t_1 = 80°C$ (176°F) and $p_1 = 1$ atm. (14.22 lb./sq.in.), and that the compression followed the law $pv^{1.35}$. Moreover, the ignition temperature was plotted against the air density on the basis of the exponential law $T_s = 709\gamma^{-0.16}$ °abs. From the indicator diagram it was evident that, with the latest possible beginning of the injection, the combustion began exactly at the dead center. The ignition delay was as follows

$$z_s = \frac{\alpha}{6\ n} = \frac{32}{6 \times 295} = 0.018 \text{ sec.}$$

In these comparisons, the ignition delay is assumed to be the time between the beginning of the injection stroke of the fuel pump and the visible pressure increase in the indicator diagram. In fact, on account of the compressibility of the fuel, the effect of the piping, the inertia of the indicator, etc., the ignition delay is really somewhat smaller, which, according to a communication from Dr. Mader, was confirmed by direct observation of the combustion chamber in the Junkers engine. The measured ignition delays of the engines are therefore to be regarded as upper limits.

By means of the measured ignition delay, it can be approximately established that, with the injection at 47 and 37° before the dead center, the ignition occurs at 15 and 5° respectively, before the dead center. The course of the air temperatures and the ignition temperatures $t_s$ during the compression gives for all Diesel engines an excellent piston position at which $\vartheta = t_s$, i.e., the two temperatures are equal. Beyond this piston position the temperature difference $\vartheta - t_s$ increases rapidly as the piston approaches the dead center. Ignition of the injected fuel can occur only when $\vartheta$ equals or exceeds $t_s$. This determines the earliest beginning of the injection.

The limit is reached for the tested engine at a crank angle of $\alpha = -49°$, or a piston stroke of 17%. During the ignition delay for the normal injection beginning $(\alpha = -47°)$ the temperature difference $(\vartheta - t_s)$ increases rapidly from 10°C (18°F) to 220°C (396°F). Ignition occurs at $p = 19.5$ atm. (277.35 lb./sq.in.) and $\vartheta = 458°C$ (824.4°F) shortly before the dead center and the combustion is consequently at approximately constant volume, which is expressed in the great pressure increase

$$\frac{p_{max}}{p_z} = \frac{46.5}{19.5} = 2.39.$$

The duration of the main combustion, from ignition to maximum pressure, is $z_v = \frac{15}{6 \times 295} = 0.0085$ sec. and the mean combustion-chamber volume during this time is

$$V_x = (\epsilon_o + \epsilon_x)\ Vh$$
$$= (0.0955 + 0.0045)\ 0.0275 = 0.002\ 75\ m^3.$$

If we assume that the combustion chamber is spherical and that the combustion proceeds radially from the center, we obtain, as the radius of the combustion chamber, $r = 0.087$ m (3.43 in.) and, as the combustion speed, $c = r/z_v = 10.2$ m (33.46 ft.) per second.

The following results were obtained by varying the injection time:

| | | | | |
|---|---|---|---|---|
| Crank angle in degrees | | -47 | -37 | -32 |
| Ratio of pressure increase | $\frac{p_{max}}{p_z}$ | 2.39 | 1.84 | 0.93 |
| Combustion time, sec. | $z_v$ | 0.0085 | 0.0102 | 0.0181 |
| Radius of sphere, m | r | 0.087 | 0.094 | 0.105 |
| Radius of sphere, in. | r | 3.435 | 3.701 | 4.134 |
| Combustion speed, m/s | c | 10.20 | 9.20 | 5.80 |
| Combustion speed, ft./sec. | c | 33.46 | 30.18 | 19.03 |

It is recognized that the answer to the question regarding the best injection timing depends on the ignition delay and on the time required for the combustion. High-compression engines require a shorter injection advance since, due to the greater temperature difference $(\vartheta - t_s)$, the reaction speed is great, while the ignition delay and combustion time are small.

The great reaction speed makes high-compression engines unsensitive to changes in the combustion chamber, atomization and load. The chemical situation passes rapidly through the energy conversion into the final stable condition. The more the final compression pressure is lowered, the greater the ignition delay and the longer the combustion time. If we wish to obtain high combustion pressure with small heat consumption, the injection must be much advanced, whereby the earliest limit of the injection is, however, very soon reached.

We see in what varying degree the necessary compression ratio in Diesel engines depends not on the temperature alone, but also on the time required for the processes which, in turn, depends on the reaction speed. This variation in the reaction speed manifests itself again during the combustion, in the pressure curve, in which no special importance attaches to combustion at constant pressure.

The greater the turbulence is in the cylinder during the ignition and combustion, just so much better is the heat transmission from the air to the fuel and just so much faster the combustion proceeds. This fact is turned to special advantage in the Junkers two-stroke engine ("Zeitschrift des Vereines deutscher Ingenieure," 1925, p. 1369). The air is set in motion in the combustion chamber through tangentially directed scavenge ports, the turbulence continuing during the injection and combustion. Due to the high final compression pressure $p_k = 38.5$ atm. (547.6 lb./sq.in.), the density $\gamma = 14.4$ kg/m³ (.899 lb./cu.ft.) and the temperature $\vartheta = 639°C$ (1182.2°F) of the air are high, but the ignition point $t_s = 190°C$ (374°F) is low. On account of the great temperature difference $(\vartheta - t_s)$, the reaction speeds must be high. The necessary consequences are a small ignition delay $z_s = 0.00486$ sec. and a high combustion pressure $p_{max} = 63.5$ atm. (903.18 lb./sq.in.). In order to obtain combustion at constant volume, the fuel is injected shortly before the dead center $(\alpha = - 11°)$.

Fig. 14 represents the ignition phenomena of the Junkers two-cylinder two-stroke engine 2 HK at full load. The test gave the especially remarkable result for two-stroke engines, that 37.4% of the energy in the fuel was transformed into useful work on the crankshaft.

Fig. 15 shows the results obtained with a Körting ignition-chamber engine, which follows another method of working. Its great advantage, as compared with other ignition-chamber engines, is due to the fact that it avoids uncooled walls in the ignition chamber and insets. The fuel, which is injected in a closed jet through the ignition chamber into the cooled channel between the ignition chamber and the cylinder, is heated by the highly compressed charging air the same as in air-injection engines. The heat transmission from the air to the fuel is so strongly assisted by the great relative velocity in the channel between the two, that the ignition delay remains small. Since only a portion of the fuel, corresponding to the quantity of oxygen present, can be burned in the ignition chamber, the combustion period is thus lengthened. For this reason, only slight pressure increases occur. A further advantage of the accelerated heat transmission in the channel and of the strong initial ignition in the chamber is the unsensitiveness of the engine to various fuels.

A comparison of the results of the three characteristic types of airless-injection engines (Table I and Fig. 16) shows that the ignition delay constitutes only a part of the phenomena which affect the heat conversion in Diesel engines. At full load all three engines show the same thermal efficiency of about 36%, although they function by different methods and under different compression pressures. The ignition delay and pressure in the engine depend on the difference between the temperature $\vartheta$ of the air in the cylinder and the ignition temperature $t_s$ of the fuel, which is also a function of the air density. The greater this temperature difference at the instant of ignition, the higher lies the ignition pressure $p_z$ and the smaller is the ignition delay $z_s$.

The low velocities of the air in the combustion chamber of the Deutz VM engine and the consequent small heat transmission from the air to the fuel drops are offset by the longer path imposed on the fuel drops by the shape of the hollow piston. This contributes to the maintenance of a hot core of air.

Körting and Junkers utilize high air velocities during the ignition for the transmission of heat. Körting generates it dynamically by pressure differences between the cylinder and ignition chamber; Junkers, by a corresponding introduction of the air into the cylinder. The indicator diagrams of the three engines (Figs. 17-19) show that the same excellent thermal efficiency can be attained by very different methods.

It is obvious that the fuel injection and combustion in airless-injection engines are so entangled at the last end that they can not be included, for all engines, in a single formula, covering all the relations. The investigations show, however, that the best principle for the constructor is to provide, through the shape and the method of functioning, that all quantities required for the heat transmission during the ignition and combustion be as high as possible. The means for accomplishing this are the enlargement of the surface of the fuel drops through finer atomization, a greater difference between the temperature of the combustion air in the cylinder and the ignition temperature of the fuel during injection, and greater turbulence in the combustion chamber. The expenditure of the requisite energy, which can not be generated by airless injection alone, in contrast with the air-injection method, is fully offset by the rapid conversion of the chemical energy of the fuel into heat.

The immediate loss in heat by transmission to the cylinder walls plays a subordinate role so long as, by correct heat transmission between the air and fuel, the reaction speeds are great enough to effect a rapid energy transformation. With the beginning of self-ignition, the combustion speed of the fuel mixture can be strongly accelerated only by a great excess of air and great turbulence.

TABLE

Results Obtained with Three Characteristic Types of Airless-Injection Diesel Engines.

| E n g i n e | | Deutz | Körting | Junkers |
|---|---|---|---|---|
| Final compression pressure, $p_k$ | lb./sq.ft.<br>atm. | 384.03<br>27.00 | 561.82<br>39.50 | 547.60<br>38.50 |
| Ignition pressure, $p_z$ | lb./sq.ft.<br>atm. | 277.35<br>19.50 | 455.15<br>32.00 | 547.60<br>38.50 |
| Max. combustion pressure, $p_{max}$ | lb./sq.ft.<br>atm. | 661.38<br>46.5 | 604.49<br>42.60 | 903.18<br>63.50 |
| Press. increment from combustion, $p_{max}/p_z$ | lb./sq.ft.<br>atm. | 33.99<br>2.39 | 19.06<br>1.34 | 23.47<br>1.65 |
| Air temp. at ignition, in | °F<br>°C | 856.4<br>458.0 | 1121.0<br>605.0 | 1182.2<br>639.0 |
| Ignition temp. of fuel, $t_s$ | °F<br>°C | 460.4<br>238.0 | 383.0<br>195.0 | 374.0<br>190.0 |
| Difference, $- t_s$ | °F<br>°C | 396.0<br>220.0 | 738.0<br>410.0 | 808.2<br>449.0 |
| Ignition delay, $z_s$ | sec. | 0.0181 | .00935 | .00486 |
| Air density at ignition, $\gamma_z$ | lb./cu.ft.<br>kg/m³ | .5306<br>8.5 | .7866<br>12.6 | .8989<br>14.4 |

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

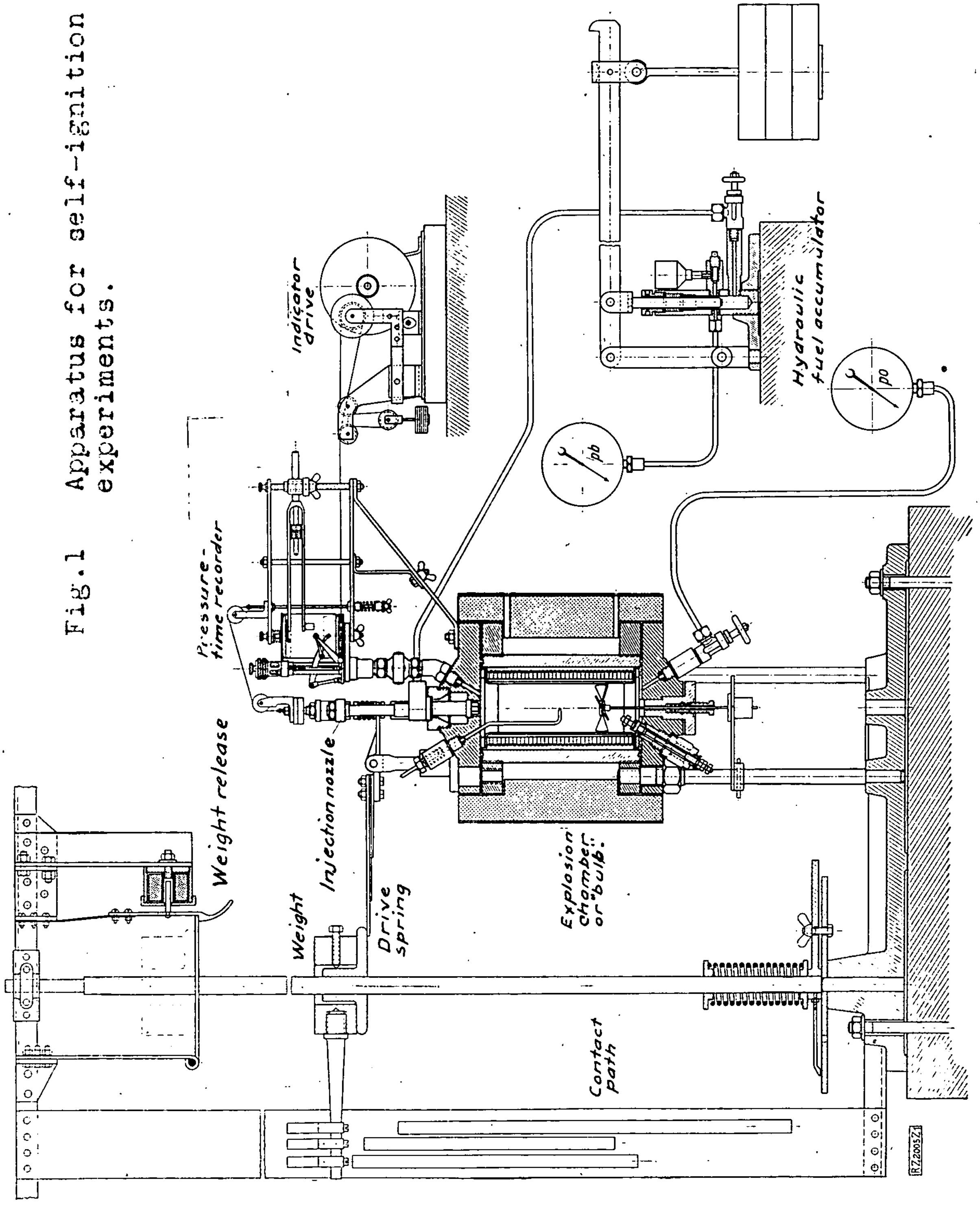



Fig.1 Apparatus for self-ignition experiments.

5466 A.S.

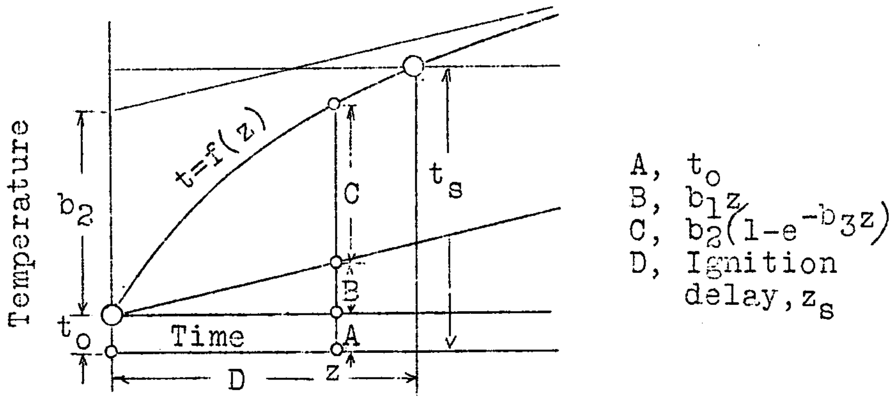
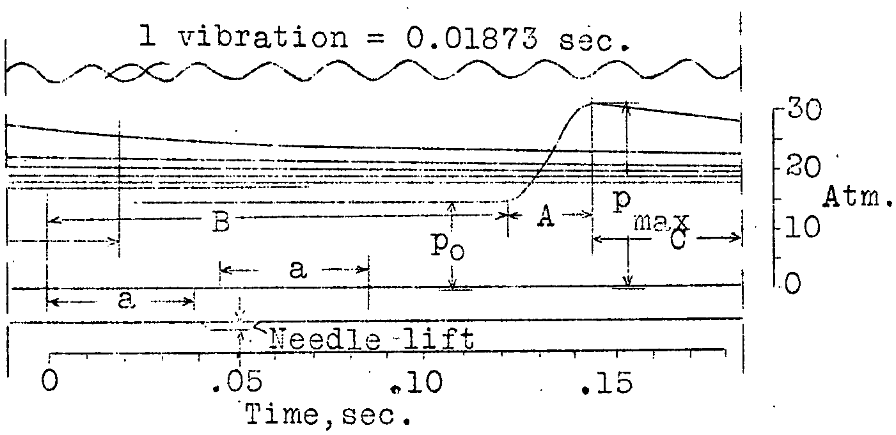



Fig.3 Fuel temperature curve from beginning of injection to ignition.




A, Main combustion
B, Ignition delay
C, After burning

Fig.2 Pressure-time diagram. Initial pressure, 15 atm.(213.35 lb./sq.in.); initial temperature, 375°C.(707°F)

Vaporization of injected fuel.

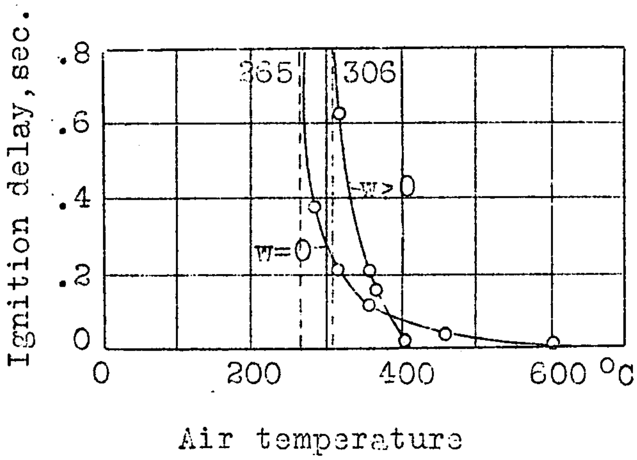



Fig.4 Ignition delay plotted against air temperature without (w=0) and with(w>0) turbulence at $p_o$=8 atm. (113.78 lb./sq.in.).

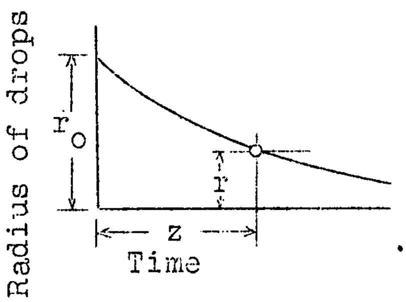



Fig.5 Decrease in radius of drops.

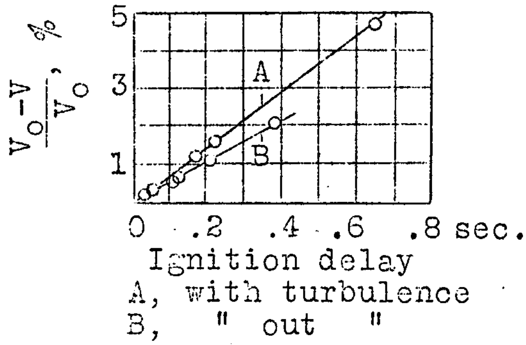



Fig.6 Decrease in vol.of drops plotted against ignition delay.

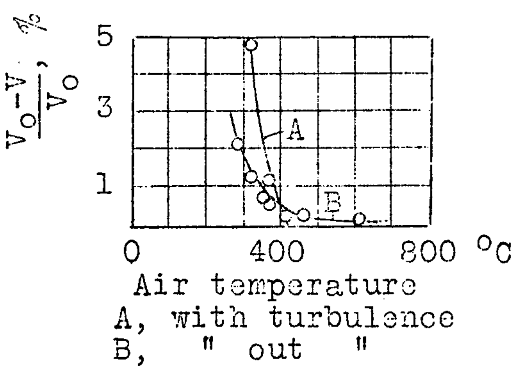



Fig.7 Decrease in vol. of drops plotted against air temperature.

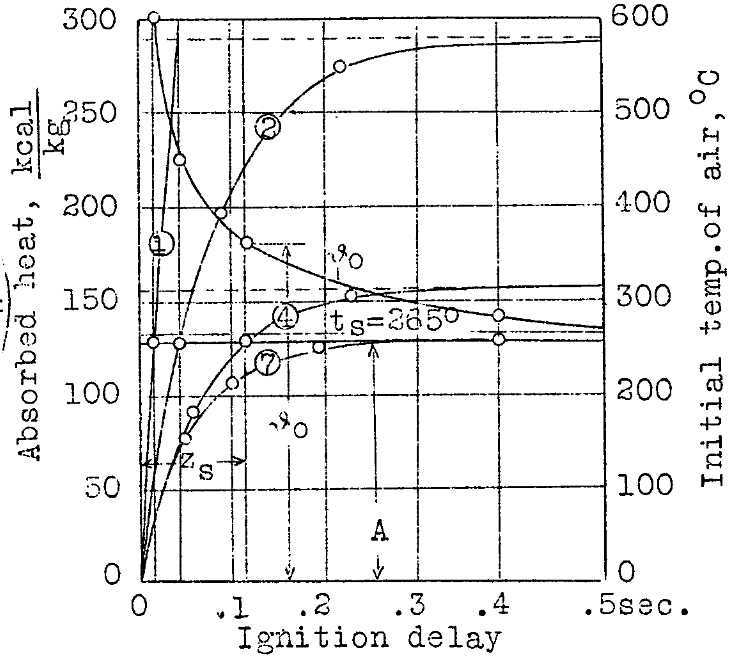



Fig.8 Heat absorption of fuel from beginning of injection to ignition. Initial pressure 8 atm.(113.78 lb./sq. in.). No turbulence.

A, with turbulence. $\alpha_m = 225e^{1.57 \, 10^{-5}(\vartheta_0 - t_s)^2}$
B, " out " $\alpha_m = 225e^{1.57 \, 10^{-4}(\vartheta_0 - t_s)^2}$

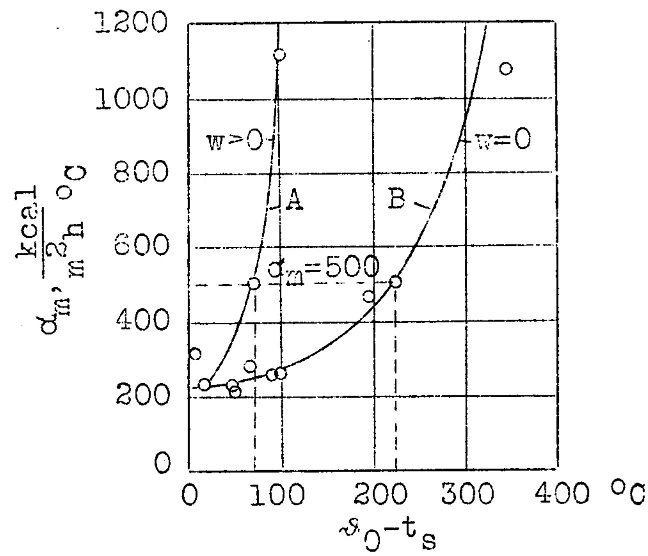
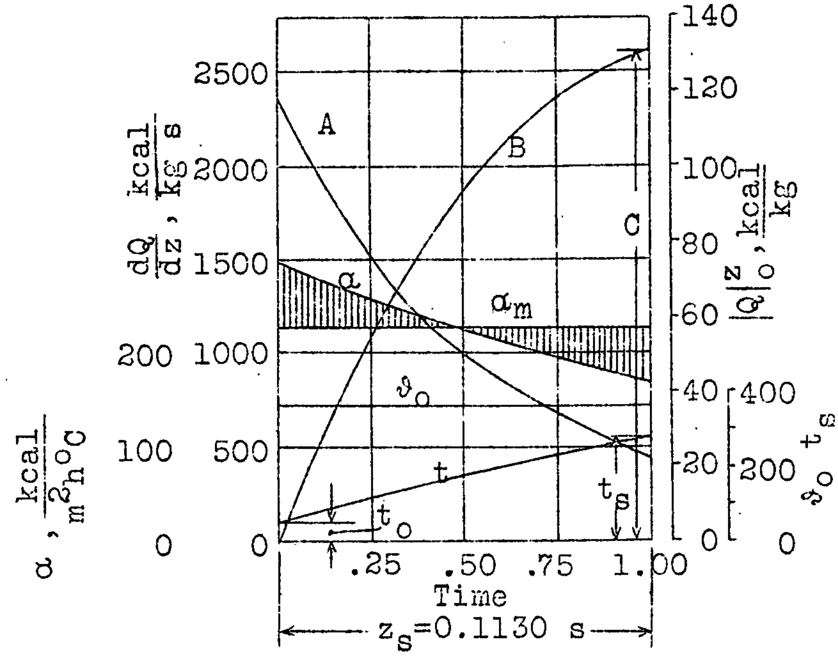



Fig.9 Mean coefficients of heat transmission from air to fuel drops during ignition delay.

$A = \frac{dQ}{dz}$

$B = |Q|_o^z$

$C = |Q|_o^{z_s}$

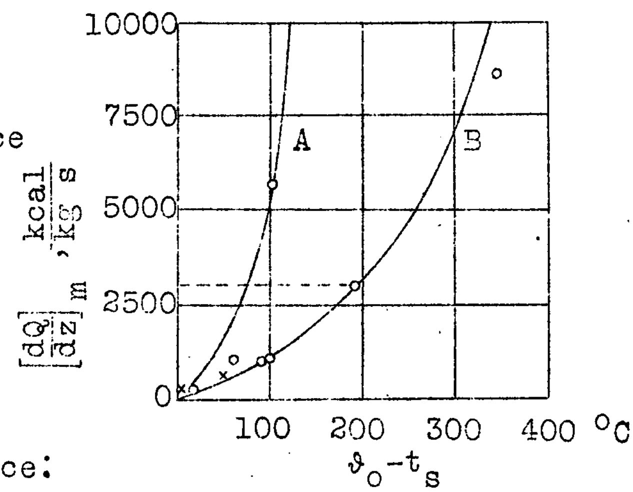


$t_s = 265^oC$, $|Q|_o^{z_s} = 129$ kcal/kg, $\alpha_m = 230 \ \frac{kcal}{m^2h^oC}$

Fig.10 Heat transmission plotted against time from beginning of injection to ignition.

A=With turbulence

B=Without turbulence.



Without turbulence:

$$\left[\frac{dQ}{dz}\right]_m = 1000\left[e^{0.00723(\vartheta_o - t_s)}-1\right]$$

With turbulence:

$$\left[\frac{dQ}{dz}\right]_m = 1000\left[e^{0.0188(\vartheta_o - t_s)}-1\right]$$

Fig.11 Mean heat absorption of fuel per unit of time during ignition delay-(With and withiut turbulence)

A = Range for Diesel engines.

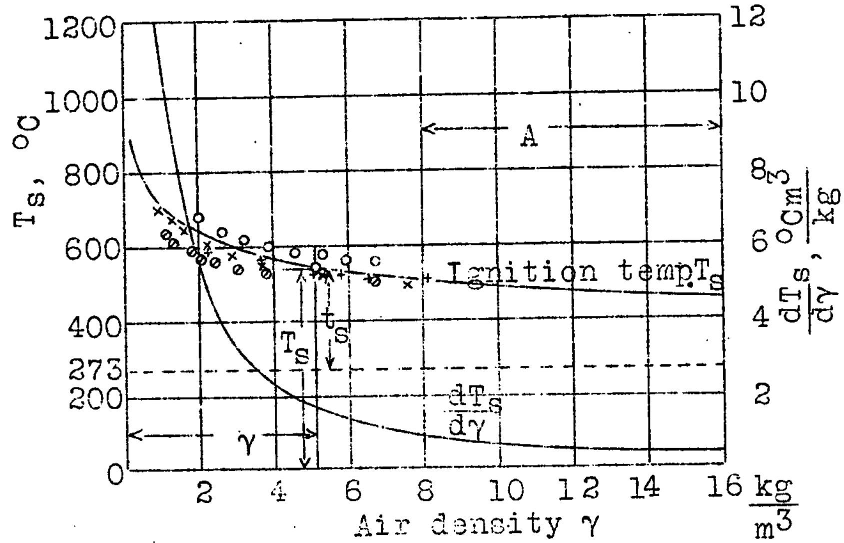



Fig.12 Self-ignition temperature plotted against air density.

A=Compression.
B=Expansion.
$C=z_{s_{III}}$
$D=z_{s_{II}}$
$E=z_{s_I}$
$F=p_{max_I}=46.5$ atm.
$G=p_{max_{II}}=33$ atm.
$H=p_{max_{III}}=27$ atm.
J=Ignition temp.
K=Air temp.
L=Pressure p.
M=Density γ
N=220°
O=7.5 atm.
P=3.75 $\frac{kg}{m^3}$
$Q=\gamma_k=11.1\ \frac{kg}{m^3}$
$R=z_I$
$S=z_{II}$
$T=z_{III}$
$U=p_k=27$ atm.
$V=\vartheta_k 558°$

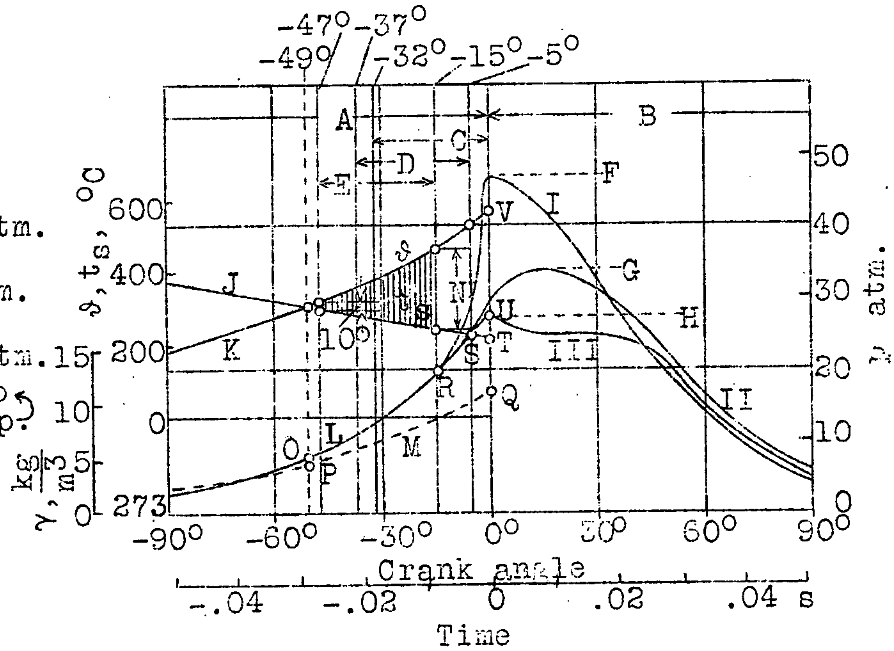



Fig.13 Ignition phenomena in a mechanical - injection engine with variable injection timing.

Stroke, 450mm (17.72in.), bore, 280mm (11.02 in.)

R.P.M. 295, $N_e$ = 202 HP., ignition delay $z_s$ =0.018 sec.

A=Compression.
B=Expansion.
C=Air temp. $\vartheta$
D=Ignition temp. $t_s$
E=Pressure $p$
F=Density $\gamma$
G=351°
H=449°
J=$z_s$
K=$\vartheta_k$=639°
L=$p_k$=38.5 atm.
M=$\gamma_k$=14.4 $\frac{kg}{m^3}$
N=$p_{max}$=63.5 atm.
O=$\frac{p_{max}}{p_z}$=1.65
P=293°

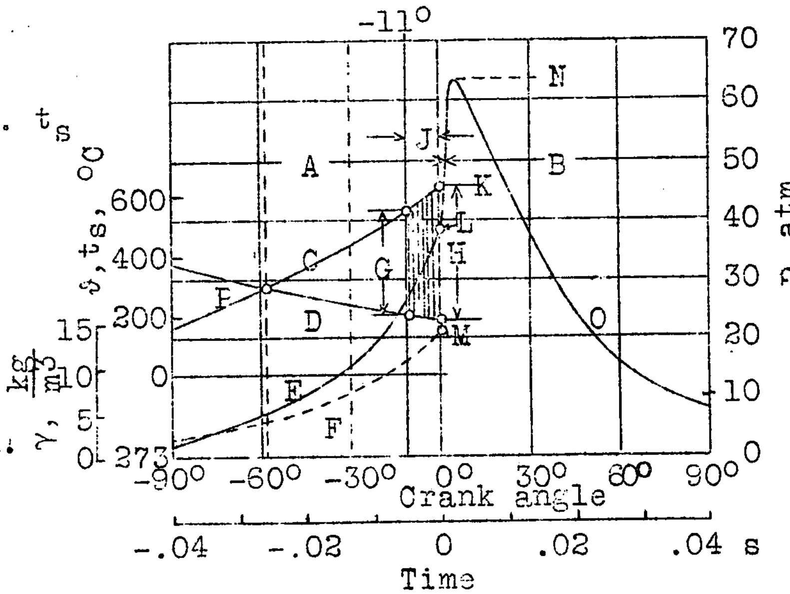



Fig.14 Ignition phenomena in a Junkers 2-cyl. 2-stroke engine 2HK. Stroke, 560mm (22.05 in.); bore, 160mm (6.3 in.); R.P.M. 377; $\alpha$ = -11° before dead center; $N_e$ = 121 HP.; ignition delay $z_s$ = 0.00486 sec.

A=Compression.
B=Expansion.
C=Air density $\vartheta$.
D=Ignition temp. $t_s$
E=Pressure p.
F=Density $\gamma$.
G=339°
H=410°
J=$z_s$
K=$\vartheta_k$=645°
L=$p_k$=39.5 atm.
M=$\gamma_k$=14.7 $\frac{kg}{m^3}$
N=$p_{max}$=42.6 atm.
O=$\frac{p_{max}}{p_z}$=1.34
P=300°

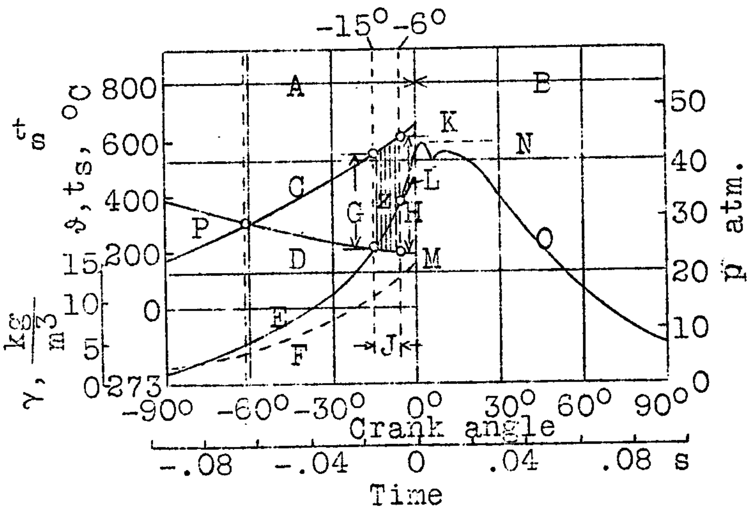



Fig.15 Ignition phenomena in a Körting two-cylinder ignition-chamber engine LT47. Stroke, 850mm (33.46 in.); bore, 495mm (19.49 in.); R.P.M.160.7; $\alpha$ = -15° before dead center; $N_e$ = 300 HP.; ignition delay $z_s$ = 0.00935 sec.

A = Air temp. $\vartheta$

B = Ignition pressure $p_z$

C = Ignition temp. $t_s$

D = Ignition delay $z_s$.

E = $\eta t_e = 0.36$

F = $\vartheta = t_s$

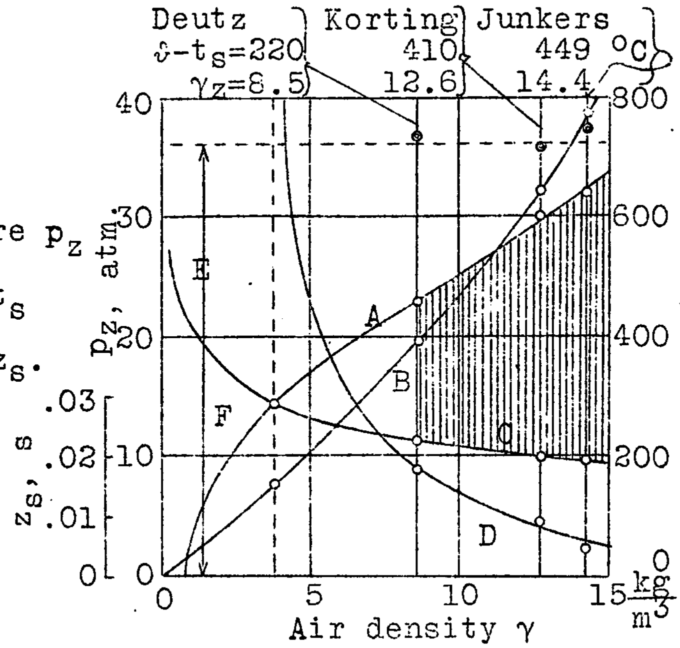



Fig.16 Ignition phenomena in three characteristic types of airless-injection Diesel engines (full load).

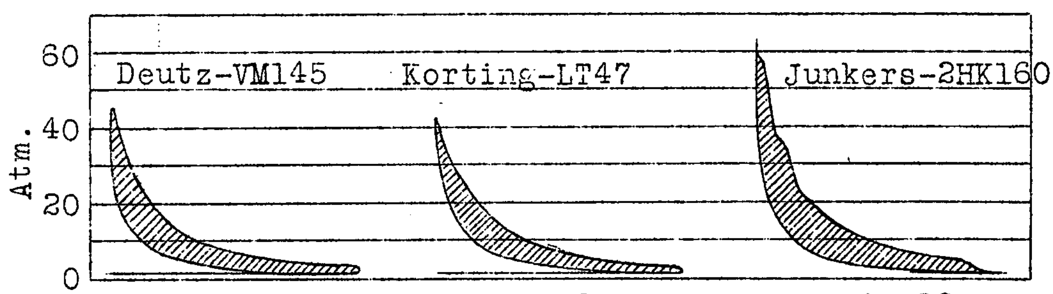



Fig.17          Fig.18          Fig.19

Indicator diagrams of three characteristic types of airless-injection Diesel engines.

| | | |
|---|---|---|
| $\frac{N_e}{i}$=50 HP. | $\frac{N_e}{i}$=150 HP. | $\frac{N_e}{i}$=60 HP. |
| $\frac{s}{d}$=1.6 | $\frac{s}{d}$=1.72 | $\frac{s}{d}$=3.5 |
| n =300 R.P.M. | n =160 R.P.M. | n =375 R.P.M. |

$N_e$= HP. i=No. of cyl. s=stroke. d=cyl. diam. n=R.P.M.