counterrotating vortices in regular rows and the heat transfer is controlled by such convection as increases with the rotational speed. The transition from laminar to vortex region is decided only by the Taylor number. The critical Taylor number is 41.

2) When $T_a < 41$, the heat transfer by the natural convection due to the temperature difference of both walls is superposed accompanying the increase of the Rayleigh number ($G_r \cdot P_r$), and for the oil this effect is fairly great. When $(G_r \cdot P_r) > 10^4$, the data are correlated by the following equation:

$$N_u = 0.11(G_r \cdot P_r)^{0.29}$$

But for the air the Rayleigh number is small, therefore the natural convection offers no problem.

3) When $T_a > 41$, the overall heat transfer coefficients is correlated by the experimental equation,

$$N_u = 0.21(T_a^2 \cdot P_r)^{1/4}$$

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Studies on the Penetration of Fuel Spray in a Diesel Engine*

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Regarding the penetrating distance of fuel spray in a diesel engine, the old theory dealing with the motion of a fuel droplet in still air is recognized not to coincide with the actual phenomenon in a diesel engine because of the extremely small size of atomized fuel droplets and the very high density of gas in cylinder. In this paper, the characteristics of spray penetration are discussed from the viewpoint of momentum theory based on the idea that the air induced into a fuel jet stream makes a kind of mixed gas together with fuel droplets. According to the results of experiment, the authors confirmed that the theory was satisfactory, that there existed the simple relations among several dimensionless numbers which indicate the effect of various factors on the spray penetration, that there is a close relationship between the spray cone angle and penetration, and others.

1. Introduction

Regarding the penetration of fuel spray in a diesel engine, various studies have been made by such investigators as Sass⁽¹⁾, Miller & Beardsley⁽²⁾, Schweitzer⁽³⁾ etc. The theoretical aspect of the conventional studies has been based on the idea that fuel spray moves in still air. However, in the case of such minute oil droplets as of diesel engine fuel spray with the initial diameter of 10

to 20 microns, if the theories were satisfactory, the velocity of the oil droplets injected in the combustion chamber should decrease suddenly and the droplet should not penetrate. However, actual experimental results do not show such sudden decrease in velocity. Therefore, taking up the momentum theory in which the mass of air is taken into account, the authors have examined the characteristics of the spray penetration.

2. Momentum theory

The resistance of the oil droplets R is given by the following Eq. (1).

$$R = \frac{\rho_a}{2} C_D \frac{\pi}{4} D^2 V^2 \dots\dots\dots (1)$$

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where D is the average diameter of the oil droplets injected from a fuel nozzle orifice, V is the relative velocity of the oil droplet and the air composing the fuel spray, C_D is the coefficient of resistance and ρ_a is the air density.

Accordingly, if ρ_f is the density of the oil droplets and t is the time after injection begins, then the equation of motion is expressed by

$$-\frac{\pi}{6} D^3 \rho_f \frac{dV}{dt} = \frac{\rho_a}{2} C_D \frac{\pi}{4} D^2 V^2 \quad \dots\dots\dots (2)$$

If $V = V_0$ at $t = 0$, the solution of the equation is given by

$$V = \frac{V_0}{1 + V_0 \kappa t} \quad \dots\dots\dots (3)$$

where

$$\kappa = \frac{3\rho_a C_D}{4\rho_f D} \quad \dots\dots\dots (4)$$

The Eq. (3) is a typical one which has been given so far⁽¹⁾. When the Eqs. (3) and (4) are examined, C_D is a function of R_e , and C_D increases as R_e decreases as shown in Fig. 1. Accordingly, as V decreases and the diameter D decreases due to the evaporation of oil droplets, C_D increases. Besides, since κ is inversely proportional to D , it seems that V decreases acceleratedly in the engine cylinder. Actually if the calculation is made on the assumption that the formula (3) is satisfactory, V at the time when the oil droplet reaches near the piston crown of an ordinary type engine becomes several tens of microns per second. So, let us make the following suppositions.

Supposition 1

In the case of fuel spray in a diesel engine, whose oil droplet diameter is very small and when the density of air in the cylinder is high, after a little penetration from the fuel nozzle, the relative velocity of oil droplets and the air composing fuel spray can be neglected.

Supposition 2

The moving mass of the fuel jet at the fuel nozzle outlet is transferred to the mixture of the air and oil droplets composing the fuel spray.

Let us consider the penetration of fuel spray on the basis of both suppositions.

Now, as shown in Fig. 2 we consider the cylindrical inspection surface which is the same in diameter as the sectional surface of fuel spray, and which is located at the distance of x from the fuel nozzle. In this case, it is known that the direction of an air stream induced into the fuel spray from the surrounding air zone is normal to the cylindrical surface, and therefore there is no component in x direction. So, it is only the momentum of fuel jet M_0 passing the left end surface that is led into the interior bounded by the inspection surface, and it is only the momentum of the mixture of air and fuel M_x passing the sectional surface at x that goes outside the cylindrical inspection surface. Since the air pressure inside the cylinder is considered to be uniform, the following equation can be obtained,

$$M_0 = M_x \quad \dots\dots\dots (5)$$

M_0 and M_x are given by the following Eqs. (6) and (7).

$$M_0 = c \frac{\pi}{4} d^2 \rho_f V_0^2 \quad \dots\dots\dots (6)$$

$$M_x = w \left[\left\{ \pi \left(x \tan \theta + \frac{\sqrt{c}}{2} d \right)^2 w - \frac{\pi}{4} d^2 c V_0 \right\} \rho_a + \frac{\pi}{4} d^2 c V_0 \rho_f \right] \quad \dots\dots\dots (7)$$

where d is the diameter of the fuel nozzle, V_0 is the jet velocity at the fuel nozzle outlet, c is the coefficient of contraction, the coefficient of velocity is 1 and 2θ is the cone angle of fuel spray.

From Eqs. (5), (6), and (7), the following equation is obtained,

$$\left(\delta \tan \theta + \frac{\sqrt{c}}{2} \right)^2 w^2 + \frac{c}{4} V_0 (\sigma - 1) w - \frac{c}{4} V_0^2 \sigma = 0 \quad \dots\dots\dots (8)$$

where δ is the ratio of penetration x and the diameter of fuel nozzle d , and σ is the ratio of density between fuel oil and air.

Fig. 1 Coefficient of resistance as a function of Reynolds number R_e , for sphere

Fig. 2 Model of fuel spray

The ratio between S shown in Fig. 3 and the fuel nozzle d is generally within the ranges of 300 to 400. As the ratio has little relation to the dimension of the engine, δ is used instead of x directly in the Eq. (8). From the Eq. (8) the velocity of the fuel spray w is obtained,

$$w = \frac{cV_0}{8} \left\{ \frac{\sqrt{(\sigma-1)^2 + \frac{16\sigma}{c} \left(\delta \tan \theta + \frac{\sqrt{c}}{2} \right)^2} - (\sigma-1)}{\left(\delta \tan \theta + \frac{\sqrt{c}}{2} \right)^2} \right\} \dots\dots\dots (9)$$

From $w = dx/dt = (d\delta/dt)d$, the following differential equation is obtained,

$$\frac{d\delta}{dt} = \frac{cV_0}{8d} \left\{ \frac{\sqrt{(\sigma-1)^2 + \frac{16\sigma}{c} \left(\delta \tan \theta + \frac{\sqrt{c}}{2} \right)^2} - (\sigma-1)}{\left(\delta \tan \theta + \frac{\sqrt{c}}{2} \right)^2} \right\} dt \dots\dots\dots (10)$$

since it can be assumed that fuel spray is almost steady during the most part of the injection period, V_0 , σ , $\tan \theta$, and c can be deemed to remain constant under a certain condition of injection. Accordingly Eq. (11) can be obtained by putting in Eq. (10) $x=0$, i.e., $\delta=0$ at $t=0$

$$\frac{1}{4 \tan \theta} \left(\frac{\sigma-1}{\sigma} \right) \left[\left(\frac{\sqrt{c}}{2} + \delta \tan \theta \right) \left\{ \sqrt{1 + \frac{16\sigma}{c(\sigma-1)^2} \left(\frac{\sqrt{c}}{2} + \delta \tan \theta \right)^2} + 2 \right\} - \frac{\sqrt{c}}{2} \left\{ \sqrt{1 + \frac{4\sigma}{(\sigma-1)^2}} + 2 \right\} \right] + \frac{\sigma-1}{4} \sqrt{\frac{c}{\sigma}} \ln \left| \frac{\frac{4}{\sigma-1} \sqrt{\frac{\sigma}{c}} \left(\frac{\sqrt{c}}{2} + \delta \tan \theta \right) + \sqrt{1 + \frac{16\sigma}{c(\sigma-1)^2} \left(\frac{\sqrt{c}}{2} + \delta \tan \theta \right)^2}}{\frac{2}{\sigma-1} \sqrt{\sigma} + \sqrt{1 + \frac{4\sigma}{(\sigma-1)^2}}} \right| = V_0 \frac{t}{d} \dots\dots\dots (11)$$

This is the equation showing the relation between the penetration of the fuel spray and condition of injection based on the momentum theory. And as stated later the cone angle can be considered to be the function only of σ . Though the Eq. (11) is much complicated, since σ is from 40 to 60 in the actual engine and since δ has a large value of 300 to 400 as already stated except the part very close to the fuel nozzle, the following equation can be considered to be correct in obtaining the approximate value.

$$\sqrt{\frac{1}{\sigma c}} \delta^2 \tan \theta \doteq V_0 \frac{t}{d} \dots\dots\dots (12)$$

This equation can also be expressed as follows:

$$x \doteq \left(\frac{2c \Delta p}{\rho_a} \right)^{0.25} \left(\frac{td}{\tan \theta} \right)^{0.5} \dots\dots\dots (13)$$

where Δp is the effective pressure of the injection and $V_0 = (2\Delta p/\rho_f)^{0.5}$. Fig. 5 shows the relation between δ and $V_0 t/d$ calculated in consideration of the relation between σ and 2θ on the basis of the data of Sass given in Fig. 4. As seen in Fig. 5 the difference between the Eqs. (11) and (12) is small. The difference of Eq. (12) and (13) from

Fig. 3 Sectional view of combustion chamber of diesel engine and variation of air distribution

Fig. 4 Spray cone angle as a function of air density in the combustion chamber (quoted from the data of Sass)

Fig. 5 Dimensionless number of spray penetration δ vs., dimensionless number of injecting condition $V_0 t/d$ as function of 2θ and σ

the conventional equation of a single oil droplet is that, according to the former, the diameter of the oil droplet has no relation with penetration but depends upon the cone angle of the fuel spray. This is very important. If a single oil droplet moves in still air as assumed in the conventional theory, the sizes of the oil droplet diameter should have a direct relation with penetration. On the contrary, the cone angle of the fuel spray is not directly related to penetration, while it has a close relation with penetration in the momentum theory.

Following the above conception, let us examine the excess air coefficient of the fuel spray.

The weight of the air flowing per second in the fuel spray at the section, $\delta = x/d$, is given by the following equation.

$$G_a = \pi d^2 g \rho_a \left[w \left(\delta \tan \theta + \frac{\sqrt{c}}{2} \right)^2 - \frac{c}{4} V_0 \right] \quad (14)$$

Quantity of the fuel oil in the spray flowing at the same section in constant regardless of δ and then it is obtained by the following equation,

$$G_f = \frac{\pi}{4} c d^2 V_0 g \rho_f \quad (15)$$

Hence,

$$\lambda = \frac{1}{L_{th}} \left(\frac{G_a}{G_f} \right) = \frac{1}{L_{th}} \left\{ \sqrt{\frac{1}{4} \left(\frac{\sigma-1}{\sigma} \right)^2 + \frac{4}{c\sigma} \left(\delta \tan \theta + \frac{\sqrt{c}}{2} \right)^2} - \left(\frac{\sigma+1}{2\sigma} \right) \right\} \quad (16)$$

where the theoretical air quantity is L_{th} , and the excess air coefficient is λ . In this case as in the case of penetration, where $\sigma = 40 \sim 50$ and δ is very large, the approximate value of λ is obtained by the following equation,

$$\lambda \approx \frac{2\delta \tan \theta}{L_{th} \sqrt{c\sigma}} = \frac{2 \tan \theta}{L_{th} \sqrt{c}} \sqrt{\frac{\rho_a}{\rho_f}} \frac{x}{d} \quad (17)$$

Fig. 6 shows the results of the calculation of λ made in accordance with the Eqs. (16) and (17).

In the Eq. (17), λ is proportional to x , and so what is meant by $x_{\lambda=1}$ is that the penetration, where λ reaches 1, can be considered to be the minimum penetration necessary for the complete combustion of the fuel oil in the spray. The distribution of air in the combustion chamber is given by the curve shown in Fig. 3. However, as to the direction of fuel oil injection, we have been given so far only the point near the maximum figure of this curve⁽⁴⁾, and, as to the penetration, no reliable basis for designing has been given.

Fig. 6 Dimensionless number of spray penetration δ vs. excess air coefficient of spray mixture λ as function of 2θ and σ

According to their theory, however, there should be such relation between the distribution of air and $x_{\lambda=1}$ as is the most suitable for the effective absorption of the fresh air.

Considering two similar diesel engines, the ratio of $x_{\lambda=1}$ and the cylinder diameter D_c can be taken as the value which represents the similarity of the distribution of the fuel spray. Then, the coefficient of the relative distribution φ , the ratio of $x_{\lambda=1}$ and D_c , is given by the following equation,

$$\varphi = \frac{x_{\lambda=1}}{D_c} = \left(\frac{d}{D_c} \right) \frac{L_{th}}{2 \tan \theta} \sqrt{c \frac{\rho_f}{\rho_a}} \quad (18)$$

Since φ is proportional to d/D_c , if the similarity of the distribution of the fuel spray is made the first condition in designing, d can be determined proportionally to D_c . Fig. 7 indicates the statistics of actual cases, which also shows that d is in proportion to D_c . The most suitable point actually determined from experience can be explained by the Eq. (18).

3. Experimental equipments and methods

In the light of the previous experiments, the spray penetration, x , is proportional to the square root of time t , and so by adopting the value K defined in the Eq. (19), the above theory can be expressed as the Eq. (20); and the Eq. (21) can be obtained by means of dimensional analysis.

$$K = \frac{x}{\sqrt{t}} \quad (19)$$

$$K=f(M_0, \rho_a, \mu_a) \dots\dots\dots(20)$$

$$\frac{K}{\sqrt{\frac{M_0}{\mu_a}}} = g\left(\frac{M_0 \rho_a}{\mu_a^2}\right) \dots\dots\dots(21)$$

Namely, the Eq. (21) shows that the fuel spray penetration is influenced by the relation between the two non-dimensional numbers, $K/(M_0/\mu_a)^{0.5}$ and $M_0 \rho_a/\mu_a^2$.

In the case of the research, to change M_0 three kinds of nozzle orifices having the diameters of 0.31, 0.34 and 0.38 mm were used, and the effective injection pressure was changed within the range of 400 to 750 atm. In order to change μ_a , the gas temperature was changed from the room temperature to 600°C; and to change ρ_a the gas pressure was changed from the atmospheric pressure to 24 atm. at the normal temperature. Fig. 8 shows a schematic diagram of the testing equipments. For the experimental reason, two methods were adopted. One is to raise the temperature at the atmospheric pressure and the other is to raise the pressure at the normal temperature. A high speed camera, "Fastax", was used for taking a picture of fuel spray. And the

Fig. 7 The relation between fuel nozzle orifice diameter vs cylinder bore determined from experience

Fig. 8 Schematic diagram of testing equipments

camera speed reached 7000 frames per second. Fig. 9 shows a series of pictures taken and Fig. 10 is the picture showing the testing equipments.

4. Results of experiments and analysis

Fig. 11 shows the results of the experiments. As seen from Fig. 11, the relation between $K/(M_0/\mu_a)^{0.5}$ and $M_0 \rho_a/\mu_a^2$ is shown by one unified curve, and accordingly the expected results could be obtained.

Stated below are the results of their studies of the gas viscosity effect in the cylinder. By using V_0 , d , and ρ_f instead of M_0 in Eq. (20), the relations shown by Eq. (22) can be obtained.

$$K = \sqrt{V_0 d} \phi\left(\frac{\rho_a}{\rho_f}, \frac{V_0 d \rho_f}{\mu_a}\right) \dots\dots\dots(22)$$

The Eq. (22) means that the non-dimensional number $K/(V_0 d)^{0.5}$ is considered to be a function of two non-dimensional numbers, ρ_a/ρ_f and $V_0 d \rho_f/\mu_a$, and that the effect of gas viscosity μ_a in the cylinder is related in the form of $V_0 d \rho_f/\mu_a$.

In Fig. 12 showing the relations between $K/(V_0 d)^{0.5}$ and $V_0 d \rho_f/\mu_a$, it has been found that $V_0 d \rho_f/\mu_a$ has scarcely any effect upon $K/(V_0 d)^{0.5}$. And as seen from Fig. 13 (a) in which is shown the analysis of the data on the test made by Miller and Beardsley⁽²⁾ about the effect of injection pressure upon the spray penetration at the room temperature and in the air with 21 atm., and also in which the changes of spray penetration 0.006 second after the injection are plotted in the form of relations between $x/V_0^{0.5}$ and V_0 , V_0 has scarcely any effect upon $x/V_0^{0.5}$. And Fig. 13 (b) shows in the form of relation between $x/d^{0.5}$ and d , that a similar analysis has been made of the data on the experiments made by A. E. G.⁽¹⁾ Co. concerning the effect of the nozzle orifice diameter. As seen from Fig. 13, d has scarcely any effect upon $x/d^{0.5}$. And exactly the same results have been obtained from the analysis of some other data. Anyhow, it is inconceivable that the non-dimensional number $V_0 d \rho_f/\mu_a$ has an effect upon the other non-dimensional number $K/(V_0 d)^{0.5}$, and it has been found that the effect is negligible. Therefore, the Eq. (23) will hold good.

$$\frac{K}{\sqrt{V_0 d}} = \phi\left(\frac{\rho_a}{\rho_f}\right) \dots\dots\dots(23)$$

Fig. 14 shows the comparison between the results of our test and the results of A. E. G.⁽¹⁾ Co.'s experiments. And as shown in Fig. 14, the above results are very similar to each other in the relation between $K/(V_0 d)^{0.5}$ and ρ_a/ρ_f in spite of the difference in nozzle orifice diameter, injection

Fig. 9 High-speed photograph of fuel spray

Fig. 10 Testing equipment

Fig. 11 Dimensionless number of spray penetration $K/(M_0/\mu_a)^{0.5}$ as a function of dimensionless number $M_0\rho_a/\mu_a^2$ (test result) Fig. 12 Dimensionless number of spray penetration $K/(V_0d)^{0.5}$ as a function of dimensionless number $V_0d\rho_f/\mu_a$ (test result)

(a) Effect of injection velocity (The result of analysis of the data by Miller and Beardsley)

(b) Effect of nozzle orifice diameter (The result of the data from A.E.G.)

Fig. 13 Effect of injection condition

Fig. 14 Dimensionless number of spray penetration $K/(V_0 d)^{0.5}$ as a function of ρ_f/ρ_a

Fig. 15 Spray cone angle 2θ as a function of ρ_f/ρ_a

pressure etc. Therefore, it is considered that Fig. 14 will serve as data for designing. Next, as to the cone angle of fuel spray, the Eq. (24) can be established by changing the Eq. (12) obtained from the theory of momentum.

$$2\theta = 2 \tan^{-1} V_0 \frac{t \sqrt{\sigma c}}{d \left(\frac{x}{d} \right)^2} = 2 \tan^{-1} \frac{\sqrt{c \rho_f / \rho_a}}{\left(\frac{K}{\sqrt{V_0 d}} \right)^2} \quad (24)$$

Fig. 15 can be obtained from the calculation of 2θ by putting in the Eq. (24) the value of $K/(V_0 d)^{0.5}$ obtained from the experiment of spray penetration as the function of ρ_f/ρ_a . On the other hand, the following Eq. (25) can also be obtained by making a dimensional analysis of 2θ in the same way as that of spray penetration,

$$2\theta = F \left(\frac{\rho_f}{\rho_a}, \frac{V_0 d \rho_f}{\mu_a} \right) \quad (25)$$

As a result of the experiments, it has been found that air viscosity has scarcely any effect upon 2θ like in the case of the spray penetration; 2θ can be determined by the following equation,

$$2\theta = G \left(\frac{\rho_f}{\rho_a} \right) \quad (26)$$

As seen from Fig. 15 in which the results of the experiments on 2θ are plotted against ρ_f/ρ_a , the results are in agreement with the results of calculation by Eq. (24). Accordingly, the theory of momentum has been found correct and the close relation between the spray penetration and the cone angle of fuel spray has also been clarified.

5. Conclusion

The following conclusion can be drawn from the above-mentioned theory and experiments.

i) In the case where the diameter of oil droplets is extremely small like the fuel spray in an actual diesel engine and also where the air density in the cylinder is very high, the relative velocity between the oil droplets and the air contained in a fuel spray is negligible, and the fuel spray can be recognized as mixture.

ii) Theory of momentum for the spray penetration holds good, and the effects of the

factors which influence the penetrating distance can be expressed by relatively simple relations consisting of non-dimensional numbers.

iii) The distance of spray penetration is proportional to square roots of each of nozzle orifice diameter, injection velocity and time after injection begins, and also is influenced by the density of gas in cylinder, but the effect of gas viscosity on the penetrating distance which remained uncertain is very little.

iv) The spray cone angle is influenced by the density of gas in the combustion chamber, and is closely related to spray penetration.

v) Excess air coefficient of spray mixture increases in proportion to the penetrating distance, and the ratio between cylinder bore and such a penetrating distance at excess air coefficient $\lambda=1$

gives a base of similarity of fuel spray distribution in cylinder. In the design, accordingly, nozzle orifice diameter may be determined only in proportion to cylinder bore.

The above theory, however, is applicable to the fuel spray characteristics of the large and medium sized diesel engines only, and will not be applicable to the fuel spray in the burners for the gas turbine and boiler, or the small sized diesel engine.

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Scavenging the 2-Stroke Diesel Engine

(Effect of Inlet Port-Angle on Scavenging Process of a Through Scavenging System)*

By Shunichi OHIGASHI**, Yukio KASHIWADA***,
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One of the important problems involved in the design of the port arrangement of a through scavenged 2-stroke Diesel engine is that of determining the entry angle of inlet ports. This paper presents the results investigated the effect of port-angle on the scavenging process inside the cylinder by using a new electronic method in model cylinders of a through scavenging system.

Introduction

Every designer of 2-stroke Diesel engines is always faced with the problem of how to determine the entry angle of the inlet ports in a through scavenging system. As to this problem some useful informations were obtained, which have already been reported. As an example, a detailed static investigation to find the accurate picture of air-flow inside the model cylinder of a through scavenging system under the simplified conditions of steady flow was carried out and the most

favourable port-angle was determined by W.H. Percival⁽¹⁾.

However, the question always arises as to whether the flow picture under such a steady-flow condition is the same as in a running engine. In order to make clear by what port angle the cylinder can be scavenged most effectively, some investigations as to how the residual gas in the cylinder is displaced by a fresh charge must be carried out under the actual conditions in a running engine.

Some fundamental research as to this respect was carried out using several new methods by W. Rizk⁽²⁾, and many useful results about the flow configuration in the inlet jet were obtained. However, no quantitative results exist regarding the effect of altering the port-angles on the rate

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