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Experimental Study on Exhaust and Scavenging Process Regarding Turbocharging of Two-Cycle Diesel Engine**

By Iwao KOIZUMI**

A series of experiments on the discharge and scavenging phenomena of two-cycle diesel engine of uniflow type was carried out by means of a test rig, which faithfully represents the phenomena in a fired engine, only using the compressed air separately charging and boosting the cylinder from different pressure sources.

The experimental results show good agreement with those calculated and verify as a whole that the simple method of calculation proposed by the author may well be applicable to the practical purpose of evaluating the phenomena and the associated problems under design stage. At the same time, the results obtained with the test rig may be applied to presume the performance of the fired engine having the same exhaust and scavenging characteristics, though certain modifications are required according to the differences in temperature condition, etc.

1. Introduction

Assuming that the flow through the scavenging and exhaust system of a diesel engine is one dimensional, the boundary conditions at either end of the cylinder or exhaust pipe may be expressed by simple formulae, and the process of discharge and the pressure set-up in either of them may also be evaluated step by step between these two boundaries with the aid of a graphical method based on the simple theory of pressure wave propagation.

The more rigorous mathematical approach using characteristics has ever been made to such similar problems as of the non-steady flow in a single pipe⁽¹⁾⁽²⁾ or pipe system⁽³⁾, none the less, the author believes it worthwhile, as the practical method, to propose this method of an approximate

calculation, and to study its adaptability by comparing with the serial results of the experiment on a test rig, representing the exhaust as well as scavenging phenomena of a two-cycle diesel engine of uniflow type, only using the compressed air from separated sources.

Comparing physically this method with the rigorous solution, it may be explained that the velocity of flow is neglected against the propagation velocity of disturbance through the flow, and also that the change in entropy of the fluid caused by the intersection of waves is disregarded, both for the reason of practical purpose of approximation.

2. Boundary equations at exhaust valve, scavenging port, and at exhaust turbine

Boundary condition at either end of cylinder or exhaust pipe, such as turbine nozzle, exhaust valve or scavenging port may be derived, on the

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assumption that the flow through the boundary is steady, from the continuity and energy conservation between the two states before and after the boundary.

These equations are given in the simple form when properly abbreviated, wherein the pressure and the corresponding velocity of the fluid in cylinder as well as in exhaust pipe are expressed by dimensionless coefficients, y_z , U_z and y , U respectively.

(i) At exhaust turbine

Regarding the exhaust turbine as an equivalent nozzle having the effective area (coeff. φ_0), the velocity there is given in the following equation in case of subsonic flow: ($r=1.4$)

$$U = \sqrt{5} \varphi_0 \frac{\sqrt{y^2 - 1}}{y^5}$$

and for sonic flow:

$$U = \frac{1}{1.2^3} \varphi_0 y$$

(ii) At the exhaust valve

Considering that the flow through the exhaust valve has its throat or minimum cross sectional area (coeff. φ_a) before coming into exhaust pipe, the following equation is derived for the upstream before the throat, viz, in the cylinder:

$$U_z = C \cdot \frac{\varphi_a}{\varphi_z} \cdot y_z$$

where

$$C = \frac{M_c}{\left(1 + \frac{r-1}{2} M_c^2\right)^{\frac{r+1}{2(r-1)}}} = \frac{M_c}{(1 + 0.2 M_c^2)^3}$$

and that for down-stream after the throat, viz, in the exhaust pipe is:

$$U = \left(1 + \frac{r-1}{2} M_c^2\right)^{\frac{1}{2}} \cdot M_c \cdot \varphi_a \cdot y$$

$$= \sqrt{1 + 0.2 M_c^2} \cdot M_c \cdot \varphi_a \cdot y$$

Thus, these two equations obtained for up- and down-streams of the flow through the exhaust valve are related to each other by means of Mach number M_c at the throat. Therefore, when calculating the condition at the exhaust valve, we shall first assume a certain value for M_c . Then the boundary equations for up- and down-streams are evaluated and when coupled respectively with the other equation for the opposite end, the values $y_z U_z$ in the cylinder and $y U$ in the exhaust pipe can be obtained. And if the assumed value of M_c is correct, the following relation should be satisfied between calculated values of $y_z U_z$ and $y U$, viz.

$$\frac{y_z}{y} = \left(1 + \frac{r-1}{2} M_c^2\right)^{\frac{1}{2}} = \sqrt{1 + 0.2 M_c^2}$$

When the flow through throat becomes sonic,

respective equations for up- and down-streams are to be:

$$U_z = \left(\frac{2}{r+1}\right)^{\frac{r+1}{2(r-1)}} \cdot \frac{\varphi_a}{\varphi_z} \cdot y_z = \frac{1}{1.2^3} \cdot \frac{\varphi_a}{\varphi_z} \cdot y_z$$

$$U = \left(\frac{r+1}{2}\right)^{\frac{1}{2}} \cdot \varphi_a \cdot y = \sqrt{1.2} \varphi_a \cdot y$$

which show that the condition in the cylinder may be determined independently of that in the exhaust pipe, and the pressure limit when the subsonic flow starts, may be given by the following equation:

$$\left(\frac{y_z}{y}\right)^2 = \frac{r+1}{2} \cdot \frac{1}{1 + \frac{r-1}{2}} \cdot \left(\frac{2}{r+1}\right)^{\frac{r+1}{r-1}} \cdot \left(\frac{\varphi_a}{\varphi_z}\right)^2$$

$$= \frac{1.2}{1 + 0.07(\varphi_a/\varphi_z)^2}$$

(iii) At scavenging port

The approximate equation of subsonic flow into cylinder from the air chamber of constant pressure through air port may be written as follows:

$$U_{zs} = \sqrt{\frac{2}{r-1}} \cdot \frac{\varphi_s}{\varphi_z} \cdot y_s \cdot \left[\left(\frac{y_s}{y_z}\right)^2 - 1\right]^{\frac{1}{2}}$$

$$= \sqrt{5} \cdot \frac{\varphi_s}{\varphi_z} \cdot y_s \cdot \left[\left(\frac{y_s}{y_z}\right)^2 - 1\right]^{\frac{1}{2}}$$

When the air blows back from the cylinder into air chamber, a similar equation will be given as follows:

$$U_z = \sqrt{5} \cdot \frac{\varphi_s}{\varphi_z} \cdot y_z \cdot \left(\frac{y_s}{y_z}\right)^5 \cdot \left[1 - \left(\frac{y_s}{y_z}\right)^2\right]^{\frac{1}{2}}$$

It should be noted that the values of $\bar{a}_0$ and a_0 , the sound velocity under atmospheric pressure corresponding to the gas condition in cylinder and exhaust pipe respectively, and a_{0s} , the same for air flowing into the cylinder during scavenging period differ from each other and, none the less, vary by nature through the period of exhausting and scavenging. But as the means of calculation with such an accuracy as required for practical purpose, it may well be sufficient to assume the adequate mean value of a_0 exhaust pipe based on the actual data, regarding it unchanged during entire course of exhaust, while in the cylinder, constant value of $\bar{a}_0$ may hold good as far as the change there is regarded adiabatic. When scavenging starts and fresh air comes into cylinder, the condition of gas and air in the cylinder is substantially influenced by the mixing between them and consequently the values $\bar{a}_0$, a_0 , and a_{0s} have to be corrected step by step by the approximate calculation, with which the new temperature of gases in cylinder after mixing with such an amount of new charge which enters during a certain period is to be found. By this experiment of model test, however, where only the compressed

air is used, the calculation is carried out disregarding the change of the values, as the temperature variation is not influential during the whole period under observation.

3. Equation of wave

On the arrival of pressure wave at one end of the pipe, propagating without changing its magnitude through the flow starting from the other end, there is the following relation between velocity and pressure differences caused at the point:

$$-(u_A - u_B) = \pm \frac{2}{r-1} a_0 \left[\left(\frac{p_A}{p_0} \right)^{\frac{r-1}{2r}} - \left(\frac{p_B}{p_0} \right)^{\frac{r-1}{2r}} \right]$$

where positive sign represents the case of arrival at B when the disturbance propagates in the direction of flow from A , while negative sign is applied in case of arrival of wave at A , propagating against the flow direction from B .

Supposing that the change of condition is adiabatic, these relations between A and B are represented by two groups of intersecting straight lines with the inclination of $\pm 1/5$ ($r=1.4$) on $y-U$ plane.

While on the other hand, the values of pressure and velocity at either end should satisfy the relation determined by the boundary and, therefore, the condition at the end is given on the previous diagram as the intersecting point of the corresponding straight line with the respective boundary equation, thus the variation of the pressure and velocity at each end can be evaluated step by step as the result of the successive change due to the arrival of the pressure wave from the opposite end, if the initial condition at one end is given, where the wave begins to propagate. Fig. 16 is an example of the process, showing the calculation in the cylinder and the exhaust pipe on the case of the experiment corresponding to Figs. 11, 12, and 13.

As noted in this example, the same diagram may be obtained so far, when we start from the same initial condition, the same release pressure, for instance, and if the same boundary condition of valve opening or nozzle contraction is given after each return of the wave.

Therefore, between those engines of geometrically similar construction and running with the same piston speed under the same release and boost pressure, for instance, there hold the same scavenging or exhaust characteristics. The experimental results with a test rig on a certain constructional item may well be applicable to any different engine, so far as the above mentioned condition will be satisfied.

4. Equipment of the experiment

The test rig consists of an actual cylinder cover complete with valves and gears, a cylinder liner and a set of another cover with a sleeve valve, which closes the opposite end of the cylinder cover around scavenging ports. The scavenging port opening is controlled by the sleeve valve having the same timing as given by the moving piston in the actual engine.

The block is mounted on a scavenging box having an ample air space around it, with the necessary gears for exhaust and scavenging valve drive, thus ensuring the wanted opening and timing for the purpose, while the cylinder volume is kept practically unchanged.

The cylinder is charged with compressed air up to a certain pressure before exhaust process commences, through a charging valve on cylinder cover, which is kept closed during the whole period of discharge and scavenging of the cylinder.

Scavenging air is led from the large surge tank with big pipe connection to the scavenging box, so that the scavenging pressure may be kept constant, and the amount of air is measured through an orifice placed before entering the surge tank, where the flow is almost in steady state.

Thus the exhaust and scavenging process in a fired engine of two-stroke uniflow type can be faithfully represented in the simplest way with this test rig, and the measurement of the phenomena can be made easily with high accuracy and reliability, as the measuring pick-ups are entirely free from the difficulties in measurement due to high temperature gas, which are frequently encountered in case of the actual engine.

At the end of the exhaust pipe, a nozzle with suitable opening is fitted, which gives on the flow in the pipe a nearly equivalent effect to that of an exhaust turbine. The effects of the exhaust pipe length, the various bends and other constructional features of the pipe may also be examined experimentally with the test rig.

Pressure variation in the cylinder and in the exhaust pipe (at two points: one after the exhaust valve and the other before the nozzle), is recorded simultaneously, together with that in the scavenging box, against the crank angle base, on the cathode-ray oscilloscope of four-channel type. Figs. 1 and 2 show the construction and the arrangement of the test rig.

As noted in the boundary equations quoted, the effective area at the vena-contracta of the flow through the exhaust valve as well as the

scavenging ports, f_a and f_s (φ_a and φ_s in the form of coefficient), is to be measured at first experimentally at various openings. This is not only the function of geometric opening of the valve in question but also it is subject to those effects of up- and down-stream passage leading to and from the valve. Figs. 3 and 4 show the results obtained respectively with the exhaust valves and scavenging ports of the test rig.

According to our previous experiment with another valve, the results obtained under steady flow test through the fixed opening have agreed practically with the results obtained from the measurement of the cyclic release of pressure through moving valve driven similar to the engine condition. Although the effect of pressure ratio upon the measured effective area of the throat becomes noticeable at larger opening, it is little

Fig. 1 Cross-section of test rig

Fig. 2 Arrangement of test rig

while the opening is comparatively small. This nature gives favourable condition to our case, since the effective area may be well assumed to be independent of the release pressure in the cylinder.

Using the obtained value as given in Fig. 4

- f : Total geometrical area
- f_a : Total effective area (measured)
- n : Number of exhaust valve
- d : Diameter of exhaust valve
- h : Lift of exhaust valve

Fig. 3 Relation between area ratio and valve opening for exhaust valve

- f : Geometrical opening area (Effect of corner rounding neglected)
- F : Geometrical area of port
- f_s : Effective area (measured)

Fig. 4 Relation between area ratio and port opening

Exhaust valve : Fixed lift 4 mm
r.p.m. : 435
Boost pressure : 0.71 kg/cm² gauge
Fig. 5 Pressure in cylinder

Fig. 6 Effective opening diagram in the experiment
(No. 1 and No. 2 are the same in cam profile and only different in timing)

Fig. 7 Scavenging characteristics

for the scavenging port, which was measured under steady flow test, the pressure as well as the velocity variations in the cylinder were calculated according to the method developed, where the scavenging air only would be charged under constant boost pressure through the ports opened and closed by the moving sleeve valve, while the exhaust valves would be kept open for discharge. The results show fairly good agreement with the experimental measurements as shown in Fig. 5, and prove that the assumptions hitherto made for constructing the boundary equations are practically acceptable.

Boost pressure : 0.5 kg/cm² gauge
r.p.m. : 450
Valve timing : No. 1
Exhaust pipe : $l=680$, $d=127$
Nozzle area ratio : 0.317
Fig. 8 Pressure in exhaust pipe (No release from cylinder)

Fig. 9 Exhaust and scavenging characteristics of two different timings under various release pressure P_{20}

Fig. 10 Effect of release pressure on the scavenging quantity under various boost pressure

5. Scavenging characteristics

By running the test rig under a constant boost pressure, but without charging the cylinder with compressed air as in normal case, the relation between the boost pressure and corresponding flow quantity was examined. This test should show clearly the scavenging characteristics of the valve timing, unaffected by the pressure release of discharge or successive wave effect in the cylinder as well as in the exhaust pipe. Two different valve timings were selected, and the results show that the flow quantity is determined largely by the lap of exhaust and inlet openings, and that greater time area of the inlet port after the closure of the exhaust valve seems almost useless (Figs. 6 and 7).

The flow quantity is also influenced by the revolution of the test rig, but it seems to have two effects, one being the charge in cylinder per cycle, the other the amount of "blowing through"; the former is by nature relative to the revolution, while the latter is rather independent of it.

One of the examples of calculated and measured pressure variation in the exhaust pipe at exhaust valve end is illustrated in Fig. 8, in which the agreement between these values is obvious. The amount of flow quantity calculated also agrees well with the measured results.

Scavenging air quantity per unit time G_s may be expressed non-dimensionally as follows:

$$\frac{G_s}{f_z \cdot S \cdot \rho_s} = 2 \frac{u_z}{a_z} \cdot \frac{1}{\Delta T} \cdot \left[1 + \frac{r-1}{2} \left(\frac{u_z}{a_z} \right)^2 \right] \cdot \frac{p_z}{p_s}$$

Neglecting the terms of higher order of u_z/a_z , as the value is usually small enough (physically it means that the flow is regarded as isothermal and this may be the usual case for incoming air heated in the cylinder to a certain degree.), we have:

$$\frac{G_s}{f_z \cdot S \cdot \rho_s} = \frac{2}{\Delta T} \cdot \frac{p_0}{p_s} \cdot U_{zs} y_z^5$$

Total amount G_s^* entering into the cylinder during the period of m times of wave return is therefore given by

$$\frac{G_s^*}{f_z \cdot S \cdot \rho_s} = \frac{2p_0}{p_s} \int_0^m U_{zs} y_z^5 dm$$

6. Exhaust and scavenging characteristics

The process of exhaust and scavenging similar to that in a fired engine was experimented in series varying release and boost pressures by the test rig, and the results are shown in Figs. 9 and 10, which indicate the effects of release and boost pressures upon scavenging characteristics given by two kinds of valve timing. It is obvious in both cases that the "blow-back" from cylinder into scavenging box at the early period of scavenging is the cause of reduction in the air quantity. In this respect, the importance of pre-exhaust opening may be worthy of note, together with the lap area of exhaust and scavenging, in order to ensure a good scavenging under possibly low boost pressure.

Calculated scavenging air quantities corresponding to several cases of experiment agree well with the results measured directly. One case of the pressure variation in cylinder and in exhaust pipe, calculated and measured, is shown in Figs. 11, 12, and 13, while the calculated distribution of flow quantity was illustrated in Fig. 14, where the "blow-back" may be observed in one of the examples. This phenomenon was verified by the experiment in such a way that the pressure in scavenging box was measured after enclosing it with a blank flange at the pipe connection, while the test

Release pressure : 3.0 kg/cm² gauge
 Boost pressure : 0.5 "
 r.p.m. : 450
 Valve timing : No. 2
 Exhaust pipe : $l=680$, $d=127$
 Nozzle area ratio : 0.317

Fig. 11 An example of experiment : Cylinder pressure

Fig. 12 An example of the experiment (corresponding to Fig. 11): Pressure in exhaust pipe at exhaust out-let

Fig. 13 An example of the experiment (corresponding to Fig. 11): Pressure in exhaust pipe at nozzle end

Timing	No. 1	No. 2	
Release pressure	4.0	3.0	kg/cm ² gauge
Boost pressure	0.5	0.5	"
r.p.m.	450	450	

(Other particulars are the same as Fig. 11)
Fig. 14 Distribution of flow quantity (calculated) under two different valve timing

Fig. 15 Oscilloscopic record of pressure in scavenging box, showing reversal flow from cylinder

rig was run by charging the air in cylinder. Fig. 15 is an example of oscillograms taken in the test. This shows the equilibrium pressure in the scavenging box, where the in-flow amount from the cylinder due to "blow-back" may be balanced with the out-flow amount from the box into the cylinder while the scavenging ports are open, and also should be equivalent to the boost pressure at zero-flow, extrapolated from those data obtained under normal test as shown in Fig. 9. In other words,

the effective part of the boost pressure is reduced by this pressure head, when there occurs "blow-back" into scavenging box.

7. Conclusion

The experimental results as hitherto explained verify as a whole that the proposed process of calculation is well applicable for the practical purpose to the evaluation of the exhaust and scavenging phenomena and the associated problems at design stage. At the same time, it should be noted that the results obtained with the test rig can be used also for the presumption of the performance of fired engine having the same exhaust and scavenging characteristics, although some

amendments are required for different conditions of temperature, piston movement, etc. But such important problems, which would affect strongly the scavenging efficiency as well as scavenging air amount, in regard to the mixing of the fresh air with the residual gas in the cylinder, or the effect of cylinder heating on the incoming air, were intentionally disregarded in the experiment with this test rig, and we expect to take them up on some other occasion with a different idea.

Appendix: Notation

p_z, ρ_z, u_z, a_z : Conditions of gas in cylinder; representing pressure, density, velocity and acoustic velocity
 p, ρ, u, a : Condition of gas in exhaust pipe; do. do.

Fig. 16 An example of calculation (corresponding to Figs. 11, 12, and 13)

p_s, ρ_s, a_s	: Condition of air in scavenging box; representing pressure, density and acoustic velocity	f_z	: Cross-sectional area of the cylinder
$p_{zs}, \rho_{zs}, u_{zs}, a_{zs}$	: Condition of air after entering in the cylinder; representing pressure, density, velocity and acoustic velocity	f_R	: Cross-sectional area of the exhaust pipe
p_0	: Atmospheric pressure	G	: Mass flow per second
$\bar{a}_0$	: Acoustic velocity of the cylinder gas under atmospheric pressure and with the same entropy	S	: Distance between both ends of cylinder
a_0	: Acoustic velocity of the gas in exhaust pipe under atmospheric pressure and with the same entropy	ΔT	: Time duration of one wave return in cylinder $\left(= \frac{2S}{a_z} \right)$
a_{0s}	: Acoustic velocity of the air in cylinder under atmospheric pressure and with the same entropy	Dimensionless expressions of area, velocity and pressure are given as follows:	
f_a	: Throat area of flow through the exhaust valve	$\varphi_a = f_a/f_R, \varphi_0 = f_0/f_R, \varphi_z = f_z/f_R, \varphi = f_s/f_R$	
f_0	: Throat area of flow through the nozzle	$U = u/a_0, U_z = u_z/\bar{a}_0, U_{zs} = u_{zs}/a_{0s}$	
f_s	: Throat area of flow through the scavenging ports	$y = \left(\frac{p}{p_0} \right)^{\frac{\gamma-1}{2\gamma}}, y_z = \left(\frac{p_z}{p_0} \right)^{\frac{\gamma-1}{2\gamma}}, y_s = \left(\frac{p_s}{p_0} \right)^{\frac{\gamma-1}{2\gamma}}$	

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On the Ignition Quality of Normal Paraffins*

By Morio TSUGE** and Yüzō OKAMOTO***

A hydrocarbon fuel has its inherent ignition property definitely related with its chemical structure, and, for the family of normal paraffins, the characteristic values may be supposed to increase with the carbon number.

In this study, we investigated the nature of the inherent ignition property in such a way that the reductions in ignition lag in a CFR Diesel engine were measured when several kinds of normal paraffins higher than Cetane, liquid or solid, were mixed with or dissolved in a liquid paraffin base fuel to varied amount and the rate of ignition lag decrease was used for evaluation.

Kogasin I (nearly Octane) was adopted as the base fuel, and two higher liquid paraffins (C_{16} and C_{17}) and three solids (C_{19} , $C_{21.5}$, and C_{31}) were examined.

It was found that the inherent ignition property increases with the carbon number irrespective of whether liquid or solid paraffins. The characteristic values obtained for C_{17} , C_{19} , $C_{21.5}$, and C_{31} were 104, 115, 136, and 202 in terms of cetane value, respectively.

1. Introduction

It is generally accepted that the principal

factor concerned with ignition lag of hydrocarbon fuels is the velocity with which activated radicals are produced to form the centers of preflame reactions. Chain carriers in the preflame reaction may originate in thermal decompositions of hydrocarbon fuel molecules, and the generating

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