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On the Transmission of the Blow-Down Energy in the Exhaust System of a Diesel Engine*

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For the improvement of the performance of a turbocharged Diesel engine with a pulse system, the utilization of the blow-down energy is most essential. The present authors have carried out the experiment on the transmission of the blow-down energy with an air model simulating the exhaust system of an actual engine. By measuring the kinetic energy of the air ejected from the equivalent nozzle at the pipe end, several factors affecting the energy transmission—pipe length, pipe diameter, area-ratio of the nozzle to the exhaust pipe and exhaust pressure—were found out. Further the theoretical calculations were performed by means of the characteristic method. The resulting values agree almost exactly with the experimental ones, when the friction loss in the exhaust pipe is taken into consideration.

1. Introduction

For the improvement of the performance of a turbocharged Diesel engine, the reduction of the loss of the blow-down energy in the exhaust system is one of the most important problems. Hitherto many studies along this line have been carried out with practical engines, however the loss of the kinetic energy in the exhaust system seems to have hardly ever been clarified due to the difficulty of obtaining the individual effect of the exhaust system itself. The scavenging of the cylinder is a knotty problem relating to the engine speed, the nozzle area, the pressure change etc. Apart from this problem, the authors have carried out the experiment with an air model to investigate the individual effect of the exhaust system on the transmission of the blow-down energy, which constitutes the greater part of the input to the turbocharger.

2. Experimental apparatus and method

The schema of the experimental apparatus, which is similar to the exhaust system of a four cycle Diesel engine, is shown in Fig. 1. The exhaust valve *B* and the exhaust pipe *C* with the nozzle *D* equivalent to the exhaust-gas turbine are fitted to the cylinder *A*. The temperature and the pressure of the air supplied in the cylin-

der through the charge valve *E* are measured by the thermocouple and the pressure gauge *F* respectively. Opening the valve by the falling weight *G* with the contour of the exhaust cam, the air is ejected perpendicular to the cage *J* of the pendulum through the exhaust pipe and the nozzle. Thus the state similar to the pressure condition during the blow-down period in the practical engine is reproduced. The pressure changes at the positions *I*, indicated in the same figure, are recorded by using indicators of the photo-electric type. The momentum of the ejected air is measured by reading the maximum swing angle of the pendulum

A: Cylinder (140 ϕ ×200), B: Exhaust valve (Seat angle 30°, Diameter 53 ϕ), C: Exhaust pipe, D: Nozzle, E: Charge valve, F: Pressure gauge, G: Falling weight, H: Rocker lever, I: Connection for indicator, J: Cage, K: Knife edge, L: Index, M: Screen

Fig. 1 Schema of experimental apparatus

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with the index L . The opening velocity of the valve is controlled for the desired value by the height, from which the weight is dropped, and the movement of the valve is recorded by tracing the movement of the screen M fitted to the valve with the photo-electric method.

Besides the standard pipe of 53 mm in diameter, we chose two different diameters of 41.6 and 68 mm, which were fitted to the standard at the distance of 168 mm from the valve by gradually enlarging or reducing. The cage of the pendulum consists of a disc lined with felts and covered with wire gauzes, on which 69 aluminum rings are piled at intervals of 3 mm. The suitable weight is added to the cage to hold the maximum swing angle of the pendulum between 5° and 8° .

It has already been found out that the momentum of the ejected air is completely absorbed by such a cage as above mentioned⁽¹⁾. We confirmed that the influence of the cage on the condition of ejecting the air through the nozzle can be eliminated by making the distance between the nozzle and the disc of the cage longer than a certain value.

The air ejected perpendicular to the disc of the cage, after turning its direction vertically, flows out through the spaces between the aluminum rings. Assuming that the momentum of all the ejected air is instantaneously imparted to the cage, its momentum M is given by

$$M = \frac{1}{l} \sqrt{2WIh(1 - \cos \theta)} \quad (1)$$

where h : distance between the knife edge and the center of gravity of the cage, I : moment of inertia of the cage about the knife edge, l : distance between the knife edge and the center line of the nozzle, W : weight of the cage, θ : maximum swing angle of the pendulum.

As the weight of the ejected air G_z is equal to the decrease of air in the cylinder, the following equation is obtained.

$$G_z = \frac{p_{z0}V}{RT_{z0}} \left\{ 1 - \left(\frac{p_{ze}}{p_{z0}} \right)^{1/\kappa} \right\} \quad (2)$$

where p_{z0} : pressure in the cylinder at the moment of closing the valve, p_{ze} and T_{z0} : pressure and absolute temperature in the cylinder at the moment of opening the valve respectively, R : gas constant, V : volume of the cylinder, κ : ratio of specific heats.

By using the average velocity of the ejected air derived from Eqs. (1) and (2), the blow-down energy E at the outlet of the nozzle can be written as follows:

$$E = \frac{gWIhRT_{z0}(1 - \cos \theta)}{l^2 V p_{z0} \{1 - (p_{ze}/p_{z0})^{1/\kappa}\}} \quad (3)$$

where g : gravitational constant.

When the pressure in the cylinder drops from p_{z0} to p_{ze} by the ejection of air into the atmosphere, the theoretically available energy of the ejected air E_i is given as follows:

$$E_i = \frac{p_{z0}V}{\kappa - 1} \left[1 - \frac{p_{ze}}{p_{z0}} - \kappa \left(\frac{p_{ze}}{p_{z0}} \right)^{\kappa-1/\kappa} \left\{ 1 - \left(\frac{p_{ze}}{p_{z0}} \right)^{1/\kappa} \right\} \right] \quad (4)$$

where p_0 : atmospheric pressure.

We represented the degree of the transmission of the blow-down energy in the exhaust system by the transmission efficiency η defined as follows:

$$\eta = \frac{E}{E_i} \quad (5)$$

The transmission efficiency thus defined includes the efficiency of the nozzle.

3. Experimental results

(a) Effect of opening velocity of exhaust valve

The flow area of the exhaust valve calculated from the contour of the weight (statical) is compared with that measured by tracing the movement of the valve (dynamical) in Fig. 2. The rapid opening of the valve causes the throttling loss in the valve to decrease and thereby improves the transmission efficiency as shown in Fig. 3. A similar tendency is seen concerning the length of the exhaust pipe, however the rate of improvement becomes more remarkable as the area-ratio of the nozzle to the exhaust pipe increases. Since the duration of the valve opening is shortened with the increase of the opening velocity of the valve, the pressure in the cylinder at the moment of closing the valve becomes higher.

(b) Effect of area-ratio of nozzle to exhaust pipe

In Fig. 4, the transmission efficiency is shown against the area-ratio. With a short exhaust pipe, the pressure waves reflected from the nozzle reach

t : Time from beginning of valve opening

T : Duration of exhaust valve opening

Fig. 2 Flow area of exhaust valve

d_e : Diameter of exhaust pipe
 l_e : Length of exhaust pipe
 φ_n : Area-ratio of nozzle to exhaust pipe

Fig. 3 Influence of opening velocity of exhaust valve

Fig. 4 Influence of area-ratio of nozzle to exhaust pipe

the valve shortly after the beginning of the exhaust and exert large influence on the throttling loss in the valve, which represents the major part of the loss in the exhaust system. Therefore the effect of the nozzle area becomes more remarkable as the exhaust pipe is shortened. The compression waves reflected from the nozzle reduce the throttling loss, but the rarefaction waves increase it. Accordingly the transmission efficiency can be improved by reducing the nozzle area. With a long exhaust pipe, the pressure waves reflected from the nozzle do not reach the valve at an early stage of the blow-down, and exert no large influence on the throttling loss. Therefore the maximum transmission efficiency is obtained at the area-ratio of the nozzle, where the reduction of

Fig. 5 Influence of pressure in cylinder at beginning of exhaust

Fig. 6 Influence of length of exhaust pipe

the throttling loss caused by the reflection waves is offset by the increase of both the friction loss in the exhaust pipe and the loss due to the adverse flow into the cylinder. The area-ratio giving the maximum transmission efficiency becomes larger as the length of the exhaust pipe increases.

(c) Effect of pressure in cylinder at beginning of exhaust

Since the discharge from the cylinder is prolonged with the increase of the pressure in the cylinder at the beginning of the exhaust, the throttling loss is the more reduced by the reflection waves. Therefore the transmission efficiency is improved, while the pressure in the cylinder at the end of the ejection becomes higher as shown in Fig. 5.

(d) Effect of length of exhaust pipe

As shown in Fig. 6, the improvement of the transmission efficiency is gained with the shorter exhaust pipe and is more remarkable with the smaller area-ratio of the nozzle to the exhaust

Fig. 7 Influence of diameter of exhaust pipe

pipe. In the case of the large area-ratio the residual pressure in the cylinder falls under the atmospheric pressure due to the pulsation in the exhaust pipe.

(e) Effect of diameter of exhaust pipe

The transmission efficiency obtained by changing the diameter of the exhaust pipe, keeping that of the nozzle $d_n = 28$ mm, is shown in Fig. 7.

With the decrease of the diameter of the exhaust pipe the transmission efficiency increases, while the residual pressure in the cylinder hardly changes. In the case where the condition of reflection of the pressure wave at the nozzle is made identical by using the nozzle of a constant area-ratio, the transmission efficiency increases more rapidly with the decrease of the diameter of the exhaust pipe. Therefore, with respect to the transmission efficiency alone, it is desirable to reduce the diameter of the exhaust pipe. However the optimum diameter must be determined considering the flow resistance during the scavenging period.

4. Calculation of transmission efficiency by theoretical analysis

If the flow in the exhaust pipe is a one-dimensional transient one, the characteristic method gives a mathematically strict solution of this problem. Accordingly the analysis is carried out with the method developed by Jenny⁽²⁾, neglecting the change of entropy and using an average entropy.

The characteristic equations in the exhaust pipe are shown as follows :

$$\frac{d(x/l_e)}{d(ta_0/l_e)} = \frac{u}{a_0} \pm \left(\frac{p}{p_0}\right)^{\frac{\kappa-1}{2\kappa}} \dots\dots\dots (6)$$

$$d\left(\frac{u}{a_0}\right) = \mp \frac{2}{\kappa-1} d\left(\frac{p}{p_0}\right)^{\frac{\kappa-1}{2\kappa}} - \frac{\lambda l_e}{2d_e} \cdot \frac{u}{|u|} \cdot \left(\frac{u}{a_0}\right)^2 \left\{1 \mp (\kappa-1) \frac{u/a_0}{(p/p_0)^{\frac{\kappa-1}{2\kappa}}}\right\} d\left(\frac{ta_0}{l_e}\right) \dots\dots\dots (7)$$

where a_0 : sonic velocity for reference, d_e and l_e : diameter and length of the exhaust pipe respectively, p : pressure in the exhaust pipe, p_0 : pressure for reference (atmospheric pressure), t : time, u : velocity, x : distance measured along the exhaust pipe, λ : coefficient of friction of the exhaust pipe.

Further the following assumptions are made for the computation.

(1) The velocity in the cylinder is naught. (2) The flow area through the valve changes continuously from the cylinder to its minimum and the loss in this part is represented by the coefficient of velocity Ψ_v . (3) The pressure in the minimum cross-section at a subsonic velocity is equal to that in the exhaust pipe.

The following boundary conditions at the valve are given by solving the equations of continuity, energy and state.

For the sonic velocity in the minimum cross-section of the valve :

$$\varphi_v \Psi_v \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{2(\kappa-1)}} \cdot \frac{1}{\left\{1 + \frac{\kappa-1}{2}(1-\Psi_v^2)\right\} \frac{p}{p_z}} = \frac{u/a_z}{1 - \frac{\kappa-1}{2}\left(\frac{u}{a_z}\right)^2} \dots\dots\dots (8)$$

$$\frac{G}{a_z \gamma_z f_e} = \varphi_v \Psi_v \left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{2(\kappa-1)}} \cdot \frac{1}{1 + \frac{\kappa-1}{2}(1-\Psi_v^2)} \dots\dots\dots (9)$$

where a_z : sonic velocity in the cylinder, f_e : cross-sectional area of the exhaust pipe, G : flow rate, p_z and γ_z : pressure and specific weight of air in the cylinder respectively, φ_v : area-ratio of the effective flow area of the valve to the exhaust pipe.

For a subsonic velocity in the minimum cross-section of the valve :

$$\varphi_v \Psi_v \frac{\sqrt{\frac{2}{\kappa-1} \left\{ 1 - \left(\frac{p}{p_z} \right)^{\frac{\kappa-1}{\kappa}} \right\} \left(\frac{p}{p_z} \right)^{-\frac{\kappa-1}{\kappa}}}}{1 + (1 - \Psi_v^2) \left\{ \left(\frac{p}{p_z} \right)^{-\frac{\kappa-1}{\kappa}} - 1 \right\}} = \frac{\frac{u}{a_z}}{1 - \frac{\kappa-1}{2} \left(\frac{u}{a_z} \right)^2} \quad (10)$$

$$\frac{G}{a_z \gamma_z f_e} = \varphi_v \Psi_v \frac{\sqrt{\frac{2}{\kappa-1} \left\{ 1 - \left(\frac{p}{p_z} \right)^{\frac{\kappa-1}{\kappa}} \right\} \left(\frac{p}{p_z} \right)^{\frac{1}{\kappa}}}}{1 + (1 - \Psi_v^2) \left\{ \left(\frac{p}{p_z} \right)^{-\frac{\kappa-1}{\kappa}} - 1 \right\}} \quad (11)$$

Representing the loss in the nozzle by the coefficient of velocity Ψ_n , the boundary conditions at the nozzle are shown by the following equations.

For the sonic velocity in the outlet of the nozzle :

$$\frac{\varphi_n \Psi_n \left(\frac{2}{\kappa+1} \right)^{\frac{\kappa+1}{2(\kappa-1)}}}{1 + \frac{\kappa-1}{2} (1 - \Psi_n^2)} \left\{ \left(\frac{p}{p_0} \right)^{\frac{\kappa-1}{\kappa}} + \frac{\kappa-1}{2} \left(\frac{u}{a_0} \right)^2 \right\}^{\frac{\kappa+1}{2(\kappa-1)}} \left(\frac{p}{p_0} \right)^{-\frac{1}{\kappa}} = \frac{u}{a_0} \quad (12)$$

$$\frac{G}{a_0 \gamma_0 f_e} = \frac{u}{a_0} \left(\frac{p}{p_0} \right)^{\frac{1}{\kappa}} \quad (13)$$

where γ_0 : specific weight for reference, φ_n : area-ratio of the nozzle to the exhaust pipe.

For a subsonic velocity in the outlet of the nozzle :

$$\varphi_n \Psi_n \frac{\sqrt{\frac{2}{\kappa-1} \left\{ \left(\frac{p}{p_0} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right\} + \left(\frac{u}{a_0} \right)^2 \left(\frac{p}{p_0} \right)^{-\frac{1}{\kappa}}}}{1 + (1 - \Psi_n^2) \left\{ \left(\frac{p}{p_0} \right)^{\frac{\kappa-1}{\kappa}} - 1 + \frac{\kappa-1}{2} \left(\frac{u}{a_0} \right)^2 \right\}} = \frac{u}{a_0} \quad (14)$$

$$\frac{G}{a_0 \gamma_0 f_e} = \frac{u}{a_0} \left(\frac{p}{p_0} \right)^{\frac{1}{\kappa}} \quad (15)$$

The above given boundary conditions are applicable to the case, where the air flows from the cylinder to the exhaust pipe and further to the atmosphere. When the air flows in the opposite direction, the boundary conditions at the valve and the nozzle must be exchanged with each other.

The blow-down energy transmitted to the outlet of the nozzle E is obtained from the state at the inlet of the nozzle as follows:

$$E = \int \kappa f_e l_e p_0 \Psi_n^2 \cdot \frac{G}{a_0 \gamma_0 f_e} \cdot \left[\frac{1}{2} \left(\frac{u}{a_0} \right)^2 + \frac{1}{\kappa-1} \left\{ \left(\frac{p}{p_0} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right\} \right] d \left(\frac{t a_0}{l_e} \right) \quad (16)$$

Assuming $\Psi_v = \Psi_n = 0.98$, $\kappa = 1.4$, and $\lambda = 0.02$, one example calculated step by step with a graphical method is shown in Fig. 8. The results obtained are approximately agreeable with the experimental ones, when the friction loss in the exhaust pipe is taken into consideration, otherwise a considerable difference occurs.

Fig. 8 Comparison of transmission efficiencies

5. Conclusion

By measuring the transmission efficiency of the blow-down energy in the exhaust system with an air model, several factors affecting the energy transmission were investigated. The results obtained are summarized as follows:

(1) The opening velocity of the valve should desirably be as high as possible. However its effect becomes less as the area-ratio of the nozzle to the exhaust pipe decreases.

(2) Except for the case of a short exhaust pipe, there exists the area-ratio of the nozzle to the exhaust pipe, that makes the transmission

efficiency maximum. In the case of a short exhaust pipe, the transmission efficiency is always improved with the decrease of the area-ratio. However the minimum area-ratio is limited by the prolongation of the discharge period.

(3) The transmission efficiency is improved by reducing the diameter of the exhaust pipe.

(4) The results calculated by the characteristic

method agree almost exactly with the experimental ones, when the friction loss in the exhaust pipe is taken into consideration.

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Idling Knock in a Pre-Combustion Type Diesel Engine*

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The idling knock of a pre-combustion type Diesel engine is considered as one of the obstacles to its automotive use, especially, to a passenger car. The mechanism of the idling knock was investigated by observing the effects of various running conditions on the knock. The idling knock was found to be originated from a rapid pressure rise in the cylinder, which was caused by explosive combustion of fuel escaping from the pre-combustion chamber in incompletely mixed state. On the basis of these results, the favourable methods to reduce this knock were worked out.

The combustion noise of a high speed Diesel engine is considered as one of the obstacles to its automotive use. This trouble becomes especially severe at a low speed running of a pre-combustion type Diesel engine, which is called "idling knock". By utilizing the throttle type of injector, this obstacle can be removed tolerably well for the bus and truck, however, unsatisfactorily for the passenger car. Hitherto, very few had studied on this problem with the exception of H. Hoffmann, who stated in his work the idling knock would be originated from the cause other than conventional Diesel knock⁽¹⁾⁽²⁾. Then, the present authors examined the influence of various running conditions upon the idling knock, and attempted to investigate the mechanism of the idling knock with numerous indicator diagrams and to work out favourable methods to reduce the knock.

1. Experimental apparatus

A pre-combustion type Diesel engine manu-

factured by the Yanmer Diesel Engine Co. (Type ST-95) was used throughout this experiment. Its main dimensions are as follows :

Type	4 cycle, single cylinder, water cooled, vertical
Cylinder bore/stroke	95 mm ϕ /115 mm
Normal output	7 B.H.P. (at 1500 r.p.m.)
Compression ratio	19 : 1
Volume ratio of the pre-combustion chamber	34 %

Fig. 1 Cross-section through the combustion chamber

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