621.436.068

A Method of Measuring the Exhaust-Gas Energy Available for Turbocharging of a Diesel Engine*

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The measurement of the exhaust-gas energy is essential for the design of turbochargers. However, in a pulse system its direct measurement is next to impossible. The present authors have devised a simple method for this purpose. A part of the exhaust gas is by-passed to expand down to the atmospheric pressure through a small measuring nozzle similar to the turbine nozzle. The momentum of by-passed gas is measured as the thrust on a cage, built of piles of aluminum rings and a disc lined with wire gauzes, and the flow rate, by means of an orifice. Thus the mean ejecting velocity of the gas from the turbine nozzle can be calculated. It has been confirmed that the exhaust-gas energy can be determined by the above mentioned method with tolerable accuracy.

1. Introduction

In order to combine properly exhaust-gas turbochargers and a Diesel engine or to investigate the efficiency of the turbine, the measurement of the available exhaust-gas energy to be utilized in the turbine is one of the principal problems. In the turbocharged engine with a pulse system, in which most of the available energy is given out during the blow-down period and the flow in the exhaust system is pulsating, the direct measurement of the available energy is impossible without the measurement of the instantaneous variations of temperature, flow rate, and pressure of exhaust gas. The measurement of these items is difficult, especially the temperature. However, by ejecting a part of the exhaust gas into the atmosphere through a nozzle similar to the turbine's and measuring its mean momentum and flow rate to calculate the mean ejecting velocity, it is possible to estimate easily the available energy regardless of the variations of the above items. In this paper, a simple method mentioned above has been investigated.

2. Method of measurement

The schematic diagram of the measuring

apparatus is shown in Fig. 1. The nozzle *A* equivalent to a turbine nozzle is fitted to the exhaust pipe *B* (53 mm in diameter) of a single-cylinder four-cycle Diesel engine (140 mm bore $\times$ 200 mm stroke). A part of the exhaust gas, which is by-passed through the intake *C* set parallel to the exhaust pipe and the by-pass pipe *D*, is ejected through the measuring nozzle *E* perpendicularly against the cage *F* in the receiver. The cage is built of a disc (120 mm in diameter) lined with 23 sheets of wire gauze, on which 6 aluminum rings (120 mm in outside diameter and 80 mm in inside diameter) are piled and fixed with the gap of 1 mm and, in addition, 32 aluminum rings with the gap of 2 mm. It is so devised as to swing on the shaft *G* held with two ball bearings at both its ends. The ejected gas escapes through the gaps between the aluminum rings at a low velocity and entirely at right angles to its original jet direction after giving its total kinetic energy to the cage. By measuring the thrust on the cage with the balance *H* through the arm *I* fitted at an end of the shaft for the reading, the mean momentum of the gas ejected is obtained. After the pressure fluctuation is damped in the receiver, the mean flow rate of gas ejected is measured by the orifice *J*. The back pressure at the outlet of the measuring nozzle is somewhat higher than the atmospheric one because of the orifice. However, this

* Received 11th September, 1958.

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pressure is so adjusted as to become the turbine back-pressure. This apparatus is made adjustable by the balance weights K so that the center of gravity of the total assembly, composed of the cage, the arm, the shaft, and the balance weights, may coincide almost exactly with the center of the shaft and only the slight weight of the cage may be imposed on the balance table. It is possible to detect the thrust as small as 1 gr.

The available exhaust-gas energy per unit weight E is expressed as follows:

$$E = \frac{1}{2g} \left(\frac{Tg}{G} \right)^2 = \frac{gT^2}{2G^2} \dots\dots\dots (1)$$

where g is the gravitational acceleration, G the flow rate of gas ejected, and T the thrust on the cage.

Being induced from the mean momentum and the mean flow rate, Eq. (1) will give a somewhat smaller value than the true value, which should be obtained through the measurement of the varying temperature, etc. This difference, however, seems to be slight, and in single-stage impulse turbines the available energy is transmitted to the impeller as the momentum. It may be regarded as appropriate to calculate the available energy from the mean momentum.

In order to produce the same state of gas at the inlet of the measuring nozzle as that at the inlet of the turbine, the measuring nozzle must be made similar to a nozzle, which may be substituted for the turbine. Without consideration of the losses from the intake to the nozzles, the conditions to produce the dynamically similar state are indicated as follows:

(1) The length from the intake to the turbine nozzle should be equal to that from the intake to the measuring nozzle.

(2) The area-ratio of the turbine nozzle to the exhaust pipe and that of the measuring nozzle to the by-pass pipe should be the same.

The difference between the available energies at the turbine nozzle and at the measuring nozzle, which is caused by a different area-ratio, is investigated in the case where the pressure wave given by Eq. (2) propagates towards the turbine nozzle in the exhaust pipe and is reflected as shown in Fig. 2.

$$\left. \begin{aligned} (p/p_0)^{(\kappa-1)/2\kappa} &= 1 + 0.06 \sin \alpha t, & u/a_0 &= 0.3 \sin \alpha t & \text{for } 0 \leq t \leq \pi/\alpha \\ (p/p_0)^{(\kappa-1)/2\kappa} &= 1, & u/a_0 &= 0 & \text{for } t > \pi/\alpha \end{aligned} \right\} \dots\dots\dots (2)$$

where a_0 is the sonic velocity for reference, p the pressure in pipes, p_0 the pressure for reference (atmospheric pressure), t the time measured from the moment the pressure wave reaches the intake, u the velocity of exhaust gas, α a constant, and κ the ratio of specific heats.

The analysis is carried out by the characteristic method under the assumptions given as follows:

- (1) The losses and the changes of entropy in the exhaust and by-pass pipes are neglected.
- (2) Both the pressures at the outlets of the nozzle and of the measuring nozzle are equal to the

A: Nozzle, B: Exhaust pipe, C: Intake, D: By-pass pipe, E: Measuring nozzle, F: Cage, G: Shaft, H: Balance, I: Arm, J: Orifice, K: Balance weight

Fig. 1 Schematic diagram of the measuring apparatus

Fig. 2

atmospheric pressure.

(3) The pressure wave propagates at the constant velocity a_0 .

The characteristic equation for both the exhaust and the by-pass pipes is given by

$$d\left(\frac{u}{a_0}\right) = \mp \frac{2}{\kappa-1} d\left(\frac{p}{p_0}\right)^{(\kappa-1)/2\kappa} \dots\dots\dots (3)$$

The boundary conditions at the nozzles are given by solving the equations of continuity, energy, and state. In the case where the exhaust gas flows out through the nozzle, they are written as follows.

For the sonic velocity at the outlet of the nozzle:

$$\frac{\varphi\psi\left(\frac{2}{\kappa+1}\right)^{(\kappa+1)/2(\kappa-1)}}{1+\frac{\kappa-1}{2}(1-\psi^2)} \left\{ \left(\frac{p}{p_0}\right)^{(\kappa-1)/\kappa} + \frac{\kappa-1}{2} \left(\frac{u}{a_0}\right)^2 \right\}^{(\kappa+1)/2(\kappa-1)} \left(\frac{p}{p_0}\right)^{-1/\kappa} = \frac{u}{a_0} \dots\dots\dots (4)$$

where φ is the area-ratio of the nozzle to the exhaust pipe (or of the measuring nozzle to the by-pass pipe) and ψ the velocity coefficient of nozzle.

For a subsonic velocity at the outlet of the nozzle:

$$\varphi\psi \frac{\sqrt{\frac{2}{\kappa-1} \left\{ \left(\frac{p}{p_0}\right)^{(\kappa-1)/\kappa} - 1 \right\} + \left(\frac{u}{a_0}\right)^2 \left(\frac{p}{p_0}\right)^{-1/\kappa}}}{1+(1-\psi^2) \left\{ \left(\frac{p}{p_0}\right)^{(\kappa-1)/\kappa} - 1 + \frac{\kappa-1}{2} \left(\frac{u}{a_0}\right)^2 \right\}} = \frac{u}{a_0} \dots\dots\dots (5)$$

The flow rate of exhaust gas through the nozzle is given by

$$\frac{G}{a_0\gamma_0 f} = \frac{u}{a_0} \left(\frac{p}{p_0}\right)^{1/\kappa} \dots\dots\dots (6)$$

where γ_0 is the specific weight of exhaust gas for reference and f the cross-sectional area of the exhaust or by-pass pipe.

When the exhaust gas flows adversely through the nozzles, the boundary conditions for a subsonic velocity at the outlet of the nozzle are given as follows:

$$\varphi\psi \frac{\sqrt{\frac{2}{\kappa-1} \left\{ 1 - \left(\frac{p}{p_0}\right)^{(\kappa-1)/\kappa} \right\} \left(\frac{p}{p_0}\right)^{-(\kappa-1)/\kappa}}}{1+(1-\psi^2) \left\{ \left(\frac{p}{p_0}\right)^{-(\kappa-1)/\kappa} - 1 \right\}} = \frac{u}{a_0} \frac{1}{1 - \frac{\kappa-1}{2} \left(\frac{u}{a_0}\right)^2} \dots\dots\dots (7)$$

$$\frac{G}{a_0\gamma_0 f} = \varphi\psi \frac{\sqrt{\frac{2}{\kappa-1} \left\{ 1 - \left(\frac{p}{p_0}\right)^{(\kappa-1)/\kappa} \right\} \left(\frac{p}{p_0}\right)^{1/\kappa}}}{1+(1-\psi^2) \left\{ \left(\frac{p}{p_0}\right)^{-(\kappa-1)/\kappa} - 1 \right\}} \dots\dots\dots (8)$$

provided that the pressure at the outlet is equal to that in the pipe

The boundary conditions at the intake is induced under the assumption that the state of gas in the exhaust pipe is not affected by that in the by-pass pipe.

For flowing into the by-pass pipe:

$$\frac{2}{\kappa-1} \left(\frac{p_g}{p_0}\right)^{(\kappa-1)/\kappa} + \left(\frac{u_g}{a_0}\right)^2 = \frac{2}{\kappa-1} \left(\frac{p_g}{p_0}\right)^{(\kappa-1)/\kappa} + \left(\frac{u_g}{a_0}\right)^2 \dots\dots\dots (9)$$

For flowing out from the by-pass pipe:

$$\frac{2}{\kappa-1} \left(\frac{p_g}{p_0}\right)^{(\kappa-1)/\kappa} + \left(\frac{u_g}{a_0}\right)^2 = \frac{2}{\kappa-1} \left(\frac{p_g}{p_0}\right)^{(\kappa-1)/\kappa} \dots\dots\dots (10)$$

where suffixes e and g refer to the states in the exhaust and by-pass pipes respectively.

The available energy E transmitted to the outlet of the nozzle is expressed as follows:

$$E = \frac{\int \frac{a_0^2 G}{2g} \psi^2 \left[\left(\frac{u}{a_0}\right)^2 + \frac{2}{\kappa-1} \left\{ \left(\frac{p}{p_0}\right)^{(\kappa-1)/\kappa} - 1 \right\} \right] dt}{\int G dt} \dots\dots\dots (11)$$

Using $\kappa=1.4$, $\psi=0.98$, and $\varphi_g=0.3$, the effect of the area-ratio φ_g of the measuring nozzle to the by-pass pipe is calculated. Fig.3 indicates the variations of the available energy E_e and E_g transmitted to the outlets of the nozzle and the measuring nozzle respectively. It is evident that E_g becomes larger with

the increase in the area-ratio of the measuring nozzle φ_g . Fig. 4 shows the available energies for the total exhaust gas which flows out through the nozzle and the measuring nozzle respectively. In a fairly long by-pass pipe, the difference between the available energy E_e and E_g caused by the different area-ratios of the measuring nozzle is considerably large.

3. Calibration of apparatus

It has already been found out that the momentum of gas ejected is completely absorbed by such a cage as previously described⁽¹⁾. In order to take off the influence of cage on the condition of ejecting the exhaust gas, the proper distance is necessary between the measuring nozzle and the bottom of cage. Using pulsative flows produced by motorizing the Diesel engine and steady ones by the compressed air, the effect of the distance is investigated. As shown in Fig. 5, the thrust of ejected air becomes constant beyond a certain distance, being not affected by the position of cage. In our measurement, this distance has been fixed at 20

Fig. 3 Variations of available energy E_e and E_g in the nozzle and the measuring nozzle respectively

Fig. 4 Effect of area-ratio φ_g of the measuring nozzle to the by-pass pipe

mm.

The diameter of by-pass pipe is required to be as small as possible, so as not to disturb the flow state in the turbine. However, there exists a restriction, because of the increase in the error of measurement and in the loss in the by-pass pipe and the measuring nozzle. In Figs. 6 and 7, the loss in the by-pass pipe and the measuring nozzle is shown in the ratio of the thrust measured in the steady flow of air to that calculated without consideration of the loss.

Under the constant area-ratio of the nozzle to the exhaust pipe, the effect of the area-ratio of the measuring nozzle to the by-pass pipe is investigated. As shown in Fig. 8, there exists no remarkable difference in the available energies measured at various area-ratios. The pressure pulsation in the exhaust pipe is usually not so

Fig. 5 Effect of position of the cage on the thrust of exhaust gas ejected

Fig. 6 Loss in the by-pass pipe and the measuring nozzle

Fig. 7 Loss in the by-pass pipe and the measuring nozzle

Engine speed: 800 r.p.m., Length of exhaust pipe: 1.44 m, Length and diameter of by-pass pipe: 0.25 m × 12.7 mm, $\varphi_e = 0.28$

Fig. 8 Effect of area-ratio φ_g of the measuring nozzle to the by-pass pipe

rapid. Even if the positive pressure wave is produced by the blow-down during the period of 40 degrees in crank angle, the value of $\alpha l / \pi a_0$ attains only 1/17 (4/4 load). Referring to Fig. 4, it is evident that the conditions of similarity for the measuring nozzle need not be so strictly considered for the small value of $\alpha l / \pi a_0$.

Fig. 9 shows a comparison of the available energies at the respective outlets of the nozzle and the measuring nozzle. The thrust of exhaust gas ejected through the nozzle is directly measured by a cage of pendulum type, which is hung before the nozzle. The flow rate through the nozzle is measured from the fuel and air consumptions of the engine. The loss in the by-pass pipe and the measuring nozzle, which is considerably large as already shown in Figs. 6 and 7, disappears by the loss in the exhaust pipe and the nozzle. Therefore, there is no conspicuous difference between both available energies measured.

4. An example of measurement

The relation between the mean pressure at

Engine speed: 800 r.p.m., Length of exhaust pipe: 1.44 m, Length and diameter of by-pass pipe: 0.25 m × 12.7 mm

Fig. 9 Comparison of available energies E_e and E_0 at the nozzle and the measuring nozzle respectively

Engine speed: 800 r.p.m., Load: 4/4, Length of exhaust pipe: 1.44 m, Length and diameter of by-pass pipe: 0.25 m × 12.7 mm, $\varphi_g = 0.28$

Fig. 10 An Example of measurement

the inlet of the nozzle and the available exhaust-gas energy is investigated with a single-cylinder Diesel engine by changing the area of nozzle. For measuring the energy, the constant area of measuring nozzle is used, because the influence of the area-ratio of the measuring nozzle is slight. The engine is charged with a Roots blower independently driven. For a given mean pressure at the inlet of the nozzle, the available energy becomes lower with the increase in the charging pressure because of the drop of the exhaust-gas temperature as shown in Fig. 10.

5. Conclusions

For the measurement of the exhaust-gas energy available to the turbine of a pulse system, a simple method was proposed. The experiment was carried out with a running engine. It was confirmed that the available energy could easily be determined by using a small measuring nozzle.

The accuracy of measurement is not remarkably affected by the similarity of both nozzles, if the pressure pulsation is not rapid. However, further research seems to be necessary, because the characteristics of the exhaust-gas turbine is not

exactly represented by a nozzle.

Reference

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621.432.4.052

Effect of Crankcase Volume on the Delivery Ratio of a Crankcase-Scavenged Two-Cycle Engine*

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In order to increase the delivery ratio and the scavenge pressure in a crankcase-scavenged two-cycle engine, the crankcase volume must be made as small as possible. However, there exists the constructional restriction. On the other hand, the delivery ratio is considerably affected by pressure fluctuations in the exhaust and inlet systems.

The present authors have experimentally investigated the effects of the crankcase volume and of the exhaust and inlet systems on the delivery ratio. It was confirmed that the drop in the delivery ratio caused by the increase in the crankcase volume could be fairly well compensated by tuning the exhaust and induction pipes. Finally they found out the conditions to improve vastly the delivery ratio by the properly tuned exhaust and induction pipes.

1. Introduction

In order to improve the performance of a crankcase-scavenged two-cycle engine, increasing the air charge is one of the principal problems. For this purpose, the crankcase volume must be made as small as possible. However, its minimum value is limited from the constructional point of view. On the other hand, it is a wellknown fact that the air charge is considerably affected by the pressure fluctuations and can be easily made larger than the piston displacement by the properly tuned exhaust and induction pipes.

Changing the crankcase volume and the length of exhaust and induction pipes, the present authors have experimentally investigated the effect of the crankcase volume on the delivery ratio and the effect of the exhaust and induction systems on the compensation for the drop in the delivery ratio caused by the increase in the crankcase volume. Further the experiment was carried out to find the conditions, under which the use of the

properly tuned exhaust and induction pipes brings a remarkable gain in the delivery ratio.

2. Experimental apparatus and method

A crankcase-scavenged two-cycle gasoline engine has been used throughout this experiment. Its main dimensions are as follows:

Type	Schnürle-scavenged, air-cooled	
Cylinder bore and stroke	65 mm × 60 mm	
Normal output	5.5 PS (at 3 600 r. p. m.)	
Effective stroke in crankcase/stroke	0.475	
	duration of open port (symmetrical timing)	time area
Inlet port	132 deg	564 cm ² deg
Scavenge port	111.4 deg	325 cm ² deg
Exhaust port	145.2 deg	924 cm ² deg
Diameter of induction pipe	29 mm	
Diameter of exhaust pipe	40 mm	

The schematic diagram of the experimental arrangement is shown in Fig.1. The discharge end of the exhaust pipe is inserted into the expansion chamber, whose volume is about 200-times the piston displacement. The exhaust gas is discharged out of the room through the expansion

* Received 12th December, 1958.

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