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Valve Timing Control by a Hydraulic Tappet with a Leak Hole*

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For a high-speed engine, valve timing is desired to be automatically controlled by engine speed to secure a good performance over a wide range of speeds. For this purpose an attempt was made to advance valve-closing and delay valve-opening automatically according to reduction of engine speed, by means of a hydraulic tappet with a hole for leakage of oil. The valve lift was measured on a motored engine so as to examine the relation between the diameter of the leak hole and valve timing. The result shows that the leak hole on the plunger controls valve-closing time with less effect on valve-opening, while the one on the cylinder wall influences only valve-opening time. Further, the valve motion was numerically solved by an original theory, which gave a fairly good estimation of valve lift. However, at low speed operation with this tappet, valve seating velocity becomes so high that a special cam profile which has low velocity over a range of closing time is desirable.

1. Introduction

It is desirable for today's conventional valve mechanism of an internal combustion engine to have variable valve timing corresponding to engine speed so as to secure a good performance over a wide range of speeds. As one of the possible ways to this, an attempt was made by means of a hydraulic tappet with a hole for leakage of oil to advance valve-closing and to delay valve-opening automatically with reduction of engine speed.

This paper shows experimental results on the

device and gives numerical solution of the valve motion.

2. Valve timing and engine performance

The air capacity of a four-stroke engine is greatly affected by inlet-valve timing and inlet- and exhaust-pipe systems. It is a well-known fact that, at high engine speed, late inlet-valve closing allows more flow into the cylinder. Fig. 1 shows the relation between volumetric efficiency and inlet-valve closing time obtained by the calculation of the exhaust and inlet processes without dynamic effects by means of a digital computer*****.

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***** Outline of the calculation is given as follows.

[Nomenclature] A : thermal equivalent of mechanical work kcal/kg-m, c_p, c_v : specific heats at constant pressure and constant volume kcal/kg-deg, f : valve flow area m^2 , μ : valve flow coefficient, G : weight of gas kg, n : engine speed rpm, p : pressure ata, R : gas constant kg-m/kg-°K, T : temperature °K, V : volume m^3 , κ : ratio of specific heats, θ : crank angle degree. Subscripts a, s and z refer to outflow gas from cylinder, inflow gas to cylinder and gas in cylinder, respectively.

It is assumed that the inflow gas instantly makes a heat exchange with the gas in the cylinder, but does not with the cylinder wall. Then the equation for energy balance in the cylinder is given as

$$c_v d(G_s T_s) + A p_s dV_s = c_p T_s dG_s - c_p T_s dG_a \quad (a)$$

From the ideal gas equation of state

$$\frac{dG_s}{G_s} = \frac{dp_s}{p_s} - \frac{dT_s}{T_s} + \frac{dV_s}{V_s}$$

Applying the above equation to Eq. (a), we obtain

$$\frac{dp_s}{d\theta} = \frac{\kappa}{V_s} \left\{ R \left(T_s \frac{dG_s}{d\theta} - T_s \frac{dG_a}{d\theta} \right) - p_s \frac{dV_s}{d\theta} \right\} \quad (b)$$

Conservation of mass in cylinder gives

$$dG_s = dG_z - dG_a$$

Combining this equation with Eq. (a), we have

$$\frac{dT_s}{d\theta} = \frac{RT_s}{p_s V_s} \left\{ (\kappa T_s - T_z) \frac{dG_s}{d\theta} - T_s (\kappa - 1) \frac{dG_a}{d\theta} \right\} - (\kappa - 1) \frac{T_s}{V_s} \frac{dV_s}{d\theta} \quad (c)$$

Outflow and inflow rate through the valves are written as

$$\frac{dG_s}{d\theta} = \frac{\mu_s f_s}{6n} \phi(p_s, p_s) p_s \sqrt{\frac{2g}{RT_s}} \quad (d)$$

$$\frac{dG_a}{d\theta} = \frac{\mu_a f_a}{6n} \phi(p_s, p_a) p_s \sqrt{\frac{2g}{RT_s}} \quad (e)$$

where

$$\phi(p_s, p_a) = \left(\frac{2}{\kappa + 1} \right)^{\frac{1}{\kappa - 1}} \sqrt{\frac{\kappa}{\kappa + 1}} \text{ for } \frac{p_s}{p_a} \leq \left(\frac{2}{\kappa + 1} \right)^{\frac{\kappa}{\kappa - 1}}$$

$$= \sqrt{\frac{\kappa}{\kappa - 1} \left\{ \left(\frac{p_s}{p_a} \right)^{\frac{2}{\kappa}} - \left(\frac{p_s}{p_a} \right)^{\frac{\kappa + 1}{\kappa}} \right\}} \text{ for } \frac{p_s}{p_a} > \left(\frac{2}{\kappa + 1} \right)^{\frac{\kappa}{\kappa - 1}}$$

and $\phi(p_s, p_a)$ will be derived by substituting p_s and p_a for p_s and p_s in $\phi(p_s, p_s)$ respectively.

From Eqs. (b), (c), (d), and (e), there will be obtained differential equations for p_s and T_s .

Supposing the cylinder pressure at exhaust blow-down to be 3.5 ata, the temperature 1000°K, inlet and back pressure atmospheric, inlet temperature 288°K, and exhaust temperature to be 740°K, p_s and T_s were solved numerically by means of a digital computer.

These solutions and Eqs. (d) and (e) give after integration the weight of trapped air and hence the volumetric efficiency.

In the calculation, the inlet-valve opening time is fixed, while the valve-closing time is variable maintaining a similar lift-curve as shown in the lower section of Fig. 1. The result shows that the maximum volumetric efficiency accompanies the earlier valve-closing time at the lower engine speed. If valve-closing time were adjusted along the broken line, i.e. the locus of the maximum efficiency point, the best volumetric efficiency would be obtained at each engine speed. Another point in favor of the advanced valve-closing is that it raises the effective compression ratio. Further, when the exhaust-valve opening is delayed at lower speed, the effective expansion stroke can be prolonged. These effects bring an increase in torque and a decrease in fuel consumption at lower speed. In addition, the highest compression ratio due to the earliest inlet-valve closing at starting exerts a good effect on cold starting.

In the case of supercharged engines, reduction of valve overlap at lower speed has an effect of saving the scavenging air.

As for a uniflow-scavenged two-stroke engine with overhead exhaust valves, to delay valve-opening time reduces blow-down loss* at low speed

cylinder bore: 88 mm, stroke: 96 mm, pressure at exhaust valve opening: 3.5 ata, temperature at exhaust valve opening: 1000°K, inlet and back pressure: atmospheric pressure, inlet temperature: 288°K, exhaust gas temperature: 740°K

Fig. 1 Effect of inlet valve closing time on volumetric efficiency and effective compression ratio

* Difference between the work done on a piston by adiabatic expansion from the exhaust-valve opening time to b.d.c. and the work actually done during the same period.

reducing an excessive blow-down area. When valve time-area is also reduced, scavenging pressure is raised and accordingly the relative cylinder charge is increased.

3. Construction and motion of the hydraulic tappet with a leak hole

As shown in Fig. 2, a leak hole 2 is bored on plunger 1 for leakage of oil. When cam 3 starts to lift cylinder 4, the pressure in cavity 5 increases and the oil leaks through the hole. But in the beginning, the cam velocity is still small and therefore the oil force in the cavity plus the force of assembling-spring 6 is not large enough to lift the plunger against the initial force of a valve spring plus the gas force on the valve face. When the cam velocity becomes large enough, the oil force overcomes the forces and then opens the valve. Valve-opening time becomes later as the engine speed decreases, for the cam velocity is in proportion to the engine speed. While the valve is open, the sum of the oil force in the cavity and the assembling-spring force balances that of the valve spring force, the inertia force of the valve train and the gas force on the valve face. At the same time the oil continues to leak out through the hole. Therefore the plunger is pushed down inside the cylinder corresponding to the volume of leakage. Since the leakage depends

1 Plunger	6 Assembling spring
2 Leak hole	7 Non-return valve
3 Cam	8 Side leak hole
4 Cylinder	9 Supply opening
5 Cavity	10 Push rod

Fig. 2 Construction of the hydraulic tappet with a leak hole

mainly on the duration of leaking, the valve lift decreases with decreasing engine speed and accordingly the valve-closing time advances. When the valve returns to the valve seat, the plunger stops supported by the pressure in the cavity, while the cam continues to descend. Then non-return valve 7 opens and permits the entry of oil. Thus the cylinder is forced down by the feed pressure and the assembling-spring force, following the cam profile.

4. Measurement of the valve lift

The hydraulic tappet was installed on the inlet-valve system of a four-stroke diesel engine. The valve lift was measured on a motoring test by means of a capacitance type displacement transducer, which is schematically shown in Fig. 3. A 2.5 mm diameter steel wire is screwed into the top of the valve stem as the earth pole of the detective capacity.

Fig. 4 shows two examples of records of the valve lift and velocity at the speed of 1000 rpm and

Fig. 3 Measuring apparatus of valve lift

Fig. 4 Records of valve lift and velocity (diameter of the leak hole 0.3 mm)

3000 rpm. The leak hole was 0.3 mm in diameter and 3 mm in length. For comparison of the valve lifts at various engine speeds, the records are plotted together in Fig. 5 (a). It is observed that the valve lift decreases and its maximum point advances as the engine speed is reduced. The broken line shows the valve lift of a mechanical tappet for comparison with the former. Fig. 5 (b) shows the valve lifts of the tappet with a 0.35 mm diameter hole. The valve lifts are smaller than the former ones, as of course would be expected, for a larger hole allows more oil leakage.

Fig. 6 shows the relation between valve timing and engine speed supposing the ineffective valve

Fig. 5 Valve lifts at various engine speeds

lift to be 0.3 mm, provided exhaust and inlet valves are as large as 34ϕ and 40ϕ in diameter respectively, on reference to Asanuma-Sawa's⁽¹⁾ experimental value of 0.32 mm on an undriven engine with overhead valves which have 32ϕ - and 27ϕ -diameter for exhaust and for inlet. As shown in Fig. 6, valve-opening time delays and valve-closing time advances with decreasing engine speed. As the engine speed increases, the valve timing curves approach the broken lines which are the ones of the mechanical tappet, but never reach the broken lines because of the leakage even at high engine speed. To compensate for the leakage at the maximum speed in order to secure the valve timing of the mechanical tappet, it is necessary to add some lift to the cam extending the total cam angle.

The valve timing could be adjusted by the diameter of the hole, but there was less effect on the valve-opening time, compared with that of the valve-closing time; for example, the change in the former was only about 5 degrees, while the change in the latter was almost 20 degrees. It is evident that the change in valve-opening could be increased by making a larger hole. But on the other hand, a larger hole gives too earlier valve-closing on account of much more leakage. To avoid the above inconsistency, a side leak hole 8 on the cylinder wall, shown in Fig. 2, is available instead. It closes by the plunger edge on the down stroke and then allows no more leakage, thus giving Fig. 7 in the case of the side leak hole amounting to 0.75 mm in diameter and 1.5 mm in distance l from the plunger edge at the initial position (See Fig. 2). The ascending curve varies with the engine speed, but the descending curve is constant without oil leakage. Accordingly, the valve-closing remains unchanged. On the contrary, the valve-opening time is changed almost 15 degrees between 950 rpm and 80 rpm. As a result, the leak hole on the plunger is available to control valve-closing with less effect on valve-opening and the side leak hole has the effect of changing only valve-opening. Consequently, combination of both may serve the purpose of giving desired

valve timing.

In the experiment above described, no consideration about oil temperature has been made, though in a practical operation the oil temperature varies according as the engine starts or is running. A change in the oil temperature affects the viscosity of oil and therefore varies its leakage. To examine this effect, the tappet was tested at different temperatures. Fig. 8 shows the relation of the valve timing to the engine speed at high temperatures ($86 \sim 89^\circ\text{C}$) and low temperatures ($44 \sim 47^\circ\text{C}$). Little difference is observed except a slight delay of the valve-closing at low speed, caused by decreasing oil temperature.

5. Numerical solution of the valve motion

Nomenclature

- F_v : equivalent initial force of valve spring at cam kg
- F_p : initial force of the assembling-spring of the tappet kg
- f_l : area of the leak hole cm^2
- f_p : area of the plunger face cm^2
- h_c : cam lift cm
- h_p : plunger lift cm
- K : bulk modulus of oil kg/cm^2
- k_v : equivalent spring rate of valve spring at cam kg/cm

Fig. 7 Valve lift of the tappet with a side leak hole

Fig. 6 Valve timing change according to engine speed

Fig. 8 Influence of oil temperature on valve timing

- k_p : spring rate of the assembling-spring kg/cm
 m : equivalent mass at cam, including valve, part of valve spring, rocker arm, push rod, plunger and oil in the plunger kg sec² cm⁻¹
 n : engine speed rpm
 P_z : gas force acting on valve face kg
 p_c : pressure in the cavity atg
 p_p : pressure in the plunger atg
 t : time sec
 V_c : initial volume of the cavity cm³
 γ : specific weight of oil kg/cm³
 θ : crank angle degree
 μ_l : flow coefficient of the leak hole

Fig. 9 shows a dynamically equivalent system for this valve train. It is treated as an equivalent system at a cam, and elastic deflection of the system is disregarded.

Taking no account of the leakage of oil through the gap between the plunger and the cylinder, the rate of a volume reduction of the oil is equal to the difference between the rate of a decrease in volume of the cavity and the rate of flow out of the leak hole. Thus, we have the relation

$$\frac{dp_c}{dt} \frac{\{V_c - f_p(h_c - h_p)\}}{K} = f_p \left(\frac{dh_c}{dt} - \frac{dh_p}{dt} \right) - \mu_l f_l \sqrt{\frac{2g}{\gamma} (p_c - p_p)}$$

Applying $d\theta = 6ndt$, we obtain an equation for $dp_c/d\theta$:

$$\frac{dp_c}{d\theta} = \frac{K}{(V_c/f_p - h_c + h_p)} \left\{ \frac{dh_c}{d\theta} - \frac{dh_p}{d\theta} - \frac{\mu_l f_l}{6nf_p} \sqrt{\frac{2g}{\gamma} (p_c - p_p)} \right\} \dots \dots \dots (1)$$

Before the plunger moves, $h_p = 0$ and $dh_p/d\theta = 0$,

Fig. 9 Dynamically equivalent system for the valve train

where Eq. (1) can be solved with the initial condition $p_c = p_p$ for a given lift curve. When the sum of $f_p p_c$ thus obtained and the assembling-spring force ($k_p h_c + F_p$) overcomes the initial force of valve spring F_v combined with the gas force on the valve face P_z , i.e. $f_p p_c + k_p h_c + F_p > F_v + P_z$, the plunger starts to open the valve. The equation of motion of the plunger is given by

$$m(d^2h/dt^2) + k_p h_p + F_v + P_z = f_p p_c + k_p (h_c - h_p) + F_p$$

and is rewritten, related to crank angle θ , as

$$\frac{d^2h_p}{d\theta^2} = \frac{1}{(6n)^2 m} \{ f_p p_c - (k_v + k_p) h_p + k_p h_c - F_v + F_p - P_z \} \dots \dots \dots (2)$$

Substituting v_p for $dh_p/d\theta$, we obtain first order differential equations for p_c , v_p and h_c , which can be solved by numerical integration.

Computation was made with KDC-II digital computer (HITAC 5020) of the Kyoto University Computation Center, and corresponding to the valve system including the hydraulic tappet shown in the preceding section. The following values were provided:

$$\begin{aligned} F_v &= 42.2 \text{ kg}, & F_p &= 2.0 \text{ kg}, \\ f_p &= 2.27 \text{ cm}^2, & k_v &= 63.8 \text{ kg/cm}, \\ k_p &= 2.7 \text{ kg/cm}, & m &= 0.585 \times 10^{-3} \text{ kg sec}^2 \text{ cm}^{-1}, \\ V_c &= 1.92 \text{ cm}^3, & \gamma &= 0.922 \times 10^{-3} \text{ kg/cm}^3 \end{aligned}$$

K was estimated to be $2.0 \times 10^4 \text{ kg/cm}^2$. Pressure p_p was derived from the experiment, regarded as

Fig. 10 Computed valve motion (diameter of the leak hole 0.3 mm)

equal to the feed pressure in the external oil supply pipe, as follows; 3.4 atg at 1000 rpm and 4.5 atg at 3000 rpm. Flow coefficient μ_l was measured by a steady flow test with water instead of oil, to simplify of the experiment, at the mean Reynolds number of the leaking oil. The flow coefficient of 0.3 mm diameter hole was 0.49. The gas force P_g was neglected, for it was small during the intake process.

Fig. 10 shows the computed valve lift, velocity, acceleration and pressure in the cavity. For comparison, the computed and measured lifts are shown together in Fig. 11, where fairly good agreement between them is seen.

It is noticeable that the valve velocity and acceleration curves pulsate with a frequency of about 1500 c/sec in the beginning of the motion. This oscillation is attributed to the compressibility of oil, because Eq. (1) does not give a periodic solution without any left term.

The records of valve velocity shown in Fig. 4 also contain such oscillation accompanied by another oscillation due to elasticity of the valve train which

Fig. 11 Comparison of computed and measured valve lifts

was disregarded in the calculation.

The pressure in the cavity also oscillates with the same phase as that of valve acceleration, because the inertia force of the valve system directly acts on the cavity. Naturally, the shape of pressure variation at high speed is remarkably different from the one at low speed on account of difference of the inertia force of the valve train.

As observed in both the measured and calculated valve velocities, a high seating velocity characterizes this tappet and hinders its practical application. One of the solutions is to reduce cam velocity at the end portion of the valve event so as to restrict the valve seating velocity to less than that of conventional engines at the maximum speed. It will not be impossible to design such a cam profile.

6. Conclusions

For improvement in performance of variable speed engines, a hydraulic tappet with a hole for leakage of oil has been introduced. It advanced valve closing and delayed valve opening automatically with decreasing engine speed. The relation between the hole diameter and valve timing was established from the measurement of the valve lifts for several kinds of diameters of the hole. The experimental results showed that the leak hole on the plunger controls valve closing with less effect on valve opening, while the one on the cylinder wall influenced only valve opening. Further, the valve motion was analyzed by numerical solution, which gave a fairly good estimation of the valve lift.

Reference

- (1) T. Asanuma and N. Sawa: *Trans. Japan Soc. Mech. Engrs.*, Vol. 24, No. 144 (1958-8), p. 607.