On the Ineffective Angle of Suction and Exhaust Valvet in Internal Combustion Engines

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N. Sawa*, K. Fukusima** and M. Sawa***

Abstract

To examine the effects of various factors on the ineffective angle or the corresponding valve lift, the authors carried out some experiments with two model pieces of piston head and two regular engines.

Some conclusions reached are summarized as follows:

a) It is possible to estimate the inrffective angle and the corresponding lift by the analysis of pressure diograms as by the measurements of air flow.

b) The ineffective angle and lift increases in proportion to the engine speed but it decreases inversely with the pressure differe nce, which shows a good coincidence with the computed results.

c) It the ineffective valve lift is used instead of the ineffective angle, it is not necessary at all to consider the sort of valve lift curve.

d) The minimum ineffective angle and lift remains contant invariably regardless of the engine speed and the pressure difflence.

1. Introduction

At a certain crank angle just after opening or just before closing of poppet or piston valves (Suction or exhaust) in an internal combustion engine, there is practically little sution or exhaust action, so that such a crank angle is termed an ineffective angle and the corresponding valve lift is called an ineffective lift.

To examine the effects of the engine speed, the valve lift curve changed by the tappet clearance and pressure difference between suction and ,exhaust sides on the ineffective angle or lift, the authors carried out some experiments with two model pieces of piston head fixed at the cylinder barrel of four-stroke cycle engine as a simple case, and successively other experiments with two regular engines of four-stroke and two-styoke cycle type.

2. Experimental apparatus and method

2.1 Measurement of air flow (Model piece of piston)

The authors have used an air-cooled, single cylinder, four-stroke cycte gasoline engine, in which an exhaust valve and two suction valves are all driven by two overhead cam-shafts, but one of suction valves is not actuated to simplify the experiments. Some dimensions of the engine are discribed in Table 1 and the

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Table 1. Dimensioes of test engine

		4 cycle engine	2 cycle engine
Cylinder volume V_h		125 cc	50 cc
Cylinder bore×Stroke		60 mm ϕ×44 mm	40 mm ϕ×39.8 mm
Clearance volume V_c		24.2 cc	8.3 cc
Suction valve	Diameter	27 mm	14 mm
	Maximum lift	5.5 mm	(Inlet pipe dia.)
Exhaust valve	Diameter	32 mm	20 mm
	Maximum lift	5.5 mm	(Exhaust pipe dia.)

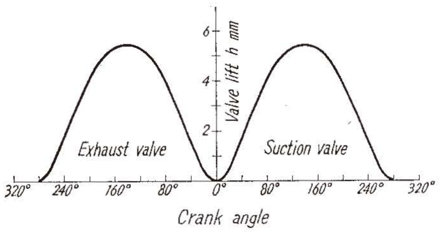


Fig. 1. Valve lift curve

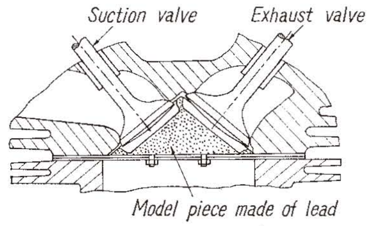


Fig. 2. A model piece of piston head fixed

valve lift curve is shown in Fig. 1. Two model pieces (A) and (B) made of lead are fixed at the position of top dead center as shown in Fig. 2. In Fig. 3 which shows the general layout of testing apparatus with the model piece of piston, an air flow-meter ② of venturi type is connected to the suction port of the test engine ① through a surge tank ③, and another tank ④ and a vaccum pump ⑤ are placed in series at the exhaust side to provide a given pressure difflence which is measured by a manometer ⑥. To prevent their wearing, both cam shafts are lubricated with oil fed by an oil pump ⑦ and the engine is cooled by a fan.

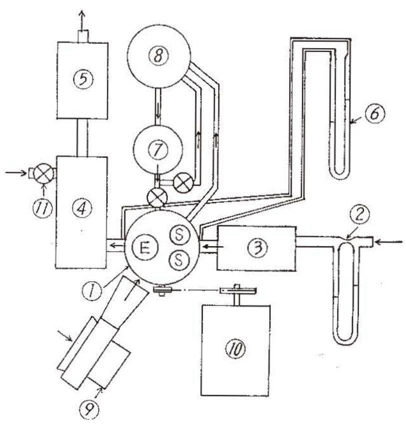


Fig. 3. General layout of testing apparatus

① Test engine ② Flow meter
③④ Surge tank ⑤ Vacum pump
⑥ Manometer ⑦ Oil pump
⑧ Oil tank ⑨ Cooling fan
⑩ Dynamometer ⑪ Cock

In the first place, the valve overlap ($\varDelta\theta$) between suction and exhaust valve and the tappet clearance ($\varDelta T$) for both valves are adjusted to give the values shown in Table 2 and Table 3, and then both cam shafts are driven by a variable speed motor ⑩. On the other hand, the pressure difference ($\varDelta P$) is controlled by a cock ⑪ on the tank ④. Then the amounts of air flow at a given $\varDelta\theta$ and $\varDelta T$ are measured changing the engine speed (N) and the pressure difference ($\varDelta P$).

Table 2. Valve overlap angles tested for model (A)

$\Delta\theta$ crank angle	$-15°$	$9°$	$28°$	$54°$	$78°$	$102°$
$\Delta h \times 10^{-2}$ mm	0	6	40	111	185.5	257
ΔT mm	4/100	4/100	7/100	6/100	6/100	6/100

Table 3. Valve overlap angles tested for model (B)

ΔT=4/100mm	$\Delta\theta$	$-15°$	$9°$	$33°$	$57°$	$81°$	$105°$
	$\Delta h \times 10^{-2}$mm	0	6.0	41.2	111.0	185.0	256.8
ΔT=7/100mm	$\Delta\theta$	$-20°$	$4°$	$28°$	$52°$	$76°$	$100°$
	$\Delta h \times 10^{-2}$mm	0	3.0	40.0	109.0	185.0	255.0
ΔT=15/100mm	$\Delta\theta$	$-5°$	$19°$	$43°$	$67°$	$91°$	—
	$\Delta h \times 10^{-2}$mm	0	31.0	98.2	173.1	245.0	—

2.2 Measurement of pressure variation

2.2.1 Case of model piece of piston

To obtain the ineffective angle by another method, that is, the analyzing of cylinder pressure, a pressure indicator of electric capacity type is inserted into the cylinder head and two indicators of the same type are also installed near the cam-shafts respectively to indicate the valve timings accurately.

Thus some pressure variations in cylinder are recorded on a magnetic oscillograph through an amplifier of the direct current type changing the engine speed (N) and the pressure difference (ΔP) in the same manner as in the case of air flow measurements.

2.2.2 Case of regular piston head

In place of the model piece, the regular piston is inserted into the cylinder and driven by the crank shaft. Besides, when the valve overlap is changed, the valve timing i.e. suction closure (S.C.) and exhaust opening (E.O.) are also altered inevitably. Thus it is very difficult to estimate the ineffective angle or lift by the air flow method for model pieces.

Considering the good agreement between the results for model pieces obtained by the two different method as mentioned above, in the case of regular piston, the ineffective angles are determined by means of the pressure diagrams at a constant tappet clearance (ΔT) and several valve overlaps ($\Delta\theta$).

Accordingly a surge tank with a flow-meter of round nozzle type is connected with the suction pipe directly and some pressure variations at the suction and exhaust ports as

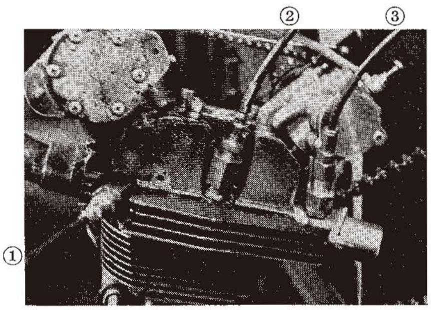


Fig. 4. Pressure indicator
① Suction port ② Cylinder head
③ Exhaust

well as in cylinder were picked up by three indicators of capacity type as shown in Fig. 4 and recorded simultaneously on a magnetic oscillograph set changing the engine speed (N) and overlap angle ($\varDelta\theta$).

3. Experimental results and Consideratiosn

3.1 Results obtained by air flow method

The amounts of air flow (Q) measured by the venturi-meter are plotted against the engine speed (N) Figs. 5 and 6 as the typical example in the case of model piece (A) and (B). By comparing the both figures, it is easily seen that the air amount for model piece (B) is larger than that for (A) and both curves show dilierent tendencies extremely at the higher value of the valve overlap. Such facts seem to be caused by the difference of throttling area between each model piece and the cylinder head for air flow, in other words, the clearance volume of model (B) is passably larger than that of (A).

Since the air amount (Q) includes not only the air blow-by ($\varDelta Q$) through the valve overlap but the air displaced by the clearance volume, the former is given by

$$\varDelta Q = Q - Q_{x^\circ} \quad \cdots\cdots (1)$$

where Q_{x° is the air amount measured when there is no valve overlap. Then a ratio of the air volume blow-by per cycle $\varDelta Q/(N/120)$ to clearance volume (V_c) is

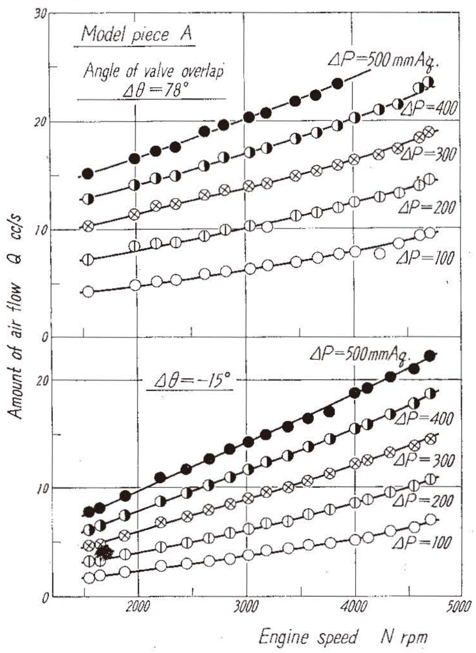


Fig. 5. Amount of air flow measured for model (A)

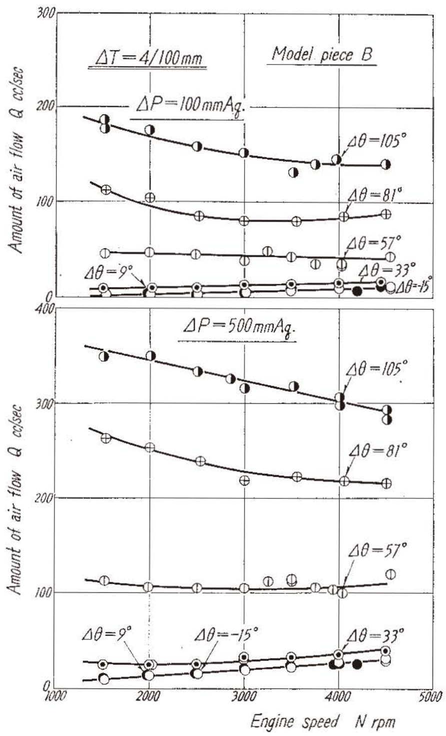


Fig. 6. Amount of air flow measured for model (B)

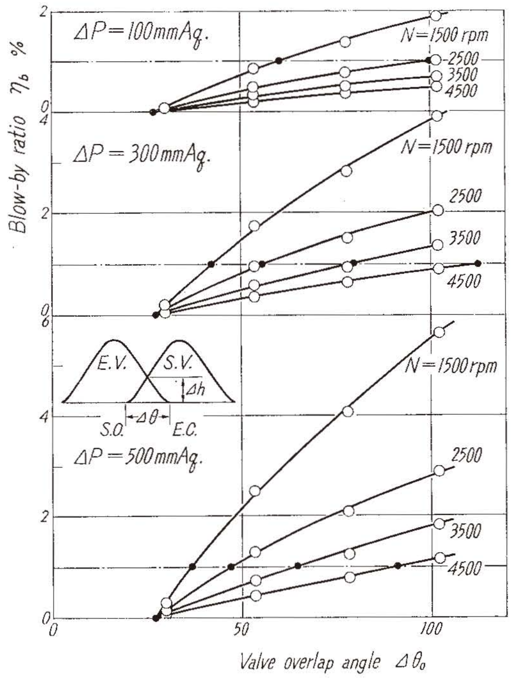


Fig. 7. Relation of blow-by ratio and value overlap angle (Model piece A)

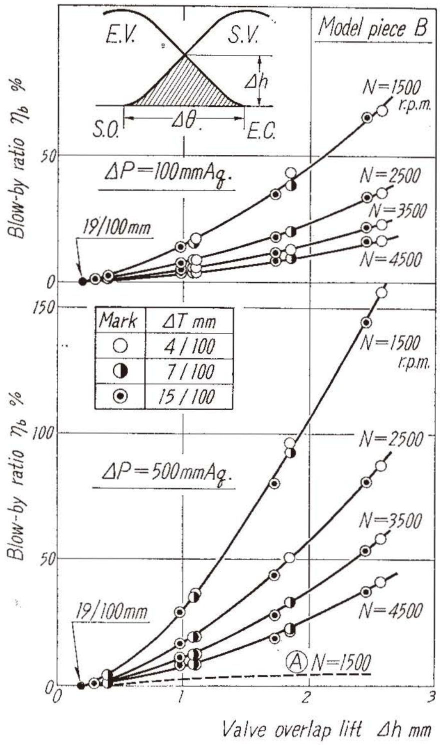


Fig. 8. Relation of blow-by ratio and valve overlap lift Model piece (B)

termed the blow-by ratio and denoted by

$$\eta_b = \{\varDelta Q/(N/120) \div V_c\} \times 100\% \quad \cdots\cdots\cdots\cdots\cdots\cdots \quad (2)$$

In Fig. 7 showing the relationship of the ratio (η_b) and the valve overlap ($\varDelta\theta$) for the model piece (A), it is noticed that the blow-by ratio (η_b) increases inversely with the engine speed (N), but all curves converge on a point $\eta_b=0$, where the minimum angle of valve overlap remains constant value $\varDelta\theta_0 \doteqdot 29°$. Such overlap angle seems to be so much affective by the sort of valve lift curve and the tappet clearance ($\varDelta T$), so that it is converted with the valve lift shown in Table 2 to the corresponding valve lift, some experimental results of the model piece (B) for various tappet clearances are compared in Fig. 8, and thus it is easily seen that if the valve lift is used instead of the valve angle, then it is not necessary at all to take account of the difference of tappet clearance and the kind of valve lift curve.

Assuming that there is no air flow practically when the ratio η_b is less than 1% for model piece (A), it seems that the ineffective angle $\varDelta_0$ equals to a half of the overlap angle $\varDelta\theta_0$ at $\eta_b=1\%$.

Fig. 9 presents such ineffective angle ($\varDelta_0$) as functions of the engine speed (N) and includes also some values obtained by C. F. Mucklow[1], which shows a good coincidence with the author's results.

According to Fig. 10 converted to the ineffective lifts (Δh_0), it is noticed that each values increase in proportion to the blow-by ratio (η_b) and engine speed (N), decrease inversely with the pressure difference, but all curves of the ineffective lift focus on a certain point, that is, the ineffective lift at $\eta_b=0$ is 32/100 mm for model piece (A).

Similarly, in the case of model piece (B) shown in Fig. 6 and Fig. 11, all curves of the air blow-by ratio η_b or the ineffective lift converge on a point $\eta_b=0$ as well as in the case of model piece (A), where the minimum overlaplift diminishes to 19/100 mm from the value 32/100 mm for model piece (A) shown in Fig. 8.

Such diminution appears to depend largely on the increase of throttling area for air flow in the clearance volume as mentioned above.

If it is considered that the air flow in the valve seat can be approximate to the steady lamminar flow in clearance[2], in the case of constant flow velocity, for example, as exhaust opening, the amount of air flow (V_0) during from the opening to a given crank angle (θ) is given from the general formula of valve lift due to the flank as follows.

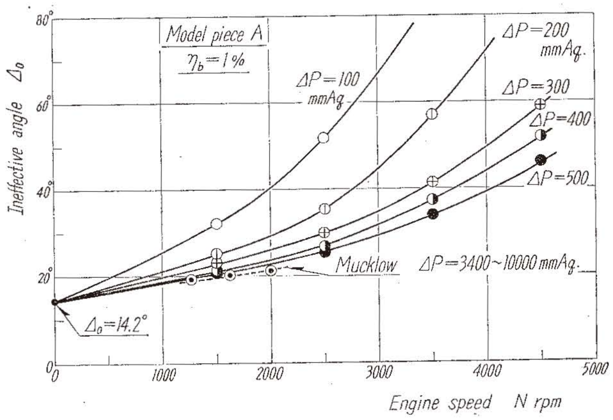


Fig. 9. Ineffective angle estimated by air flow method (Model piece A)

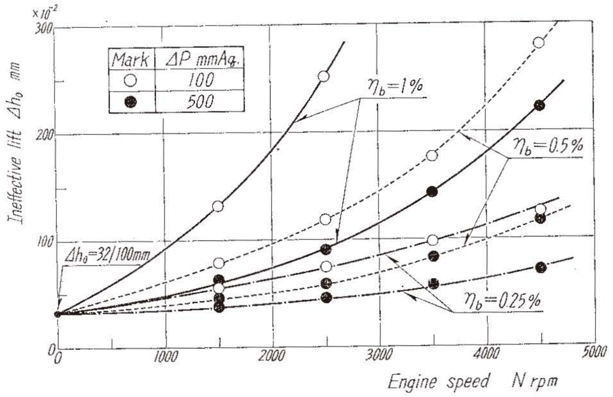


Fig. 10. Ineffective lift estimated by air flow methed (Model peice A)

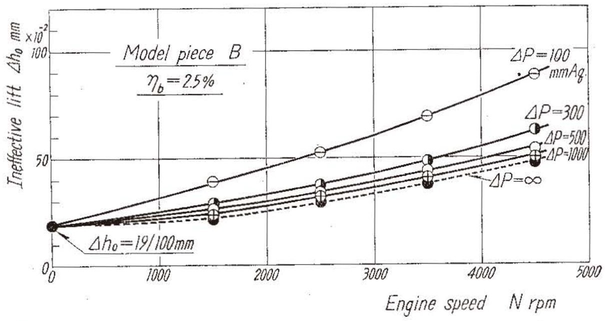


Fig. 11. Ineffective lift estimated by air flow methop

For tangential cam with roller follower as shown in Fig. 12

$$V_0=30k_1\cdot k_2\cdot\cos\alpha\cdot C\cdot\frac{D^2}{N}\cdot\left\{2\log\cdot\tan\left(\frac{\pi+\theta}{4}\right)-\theta\right\} \cdots\cdots\cdots\cdots (3)$$

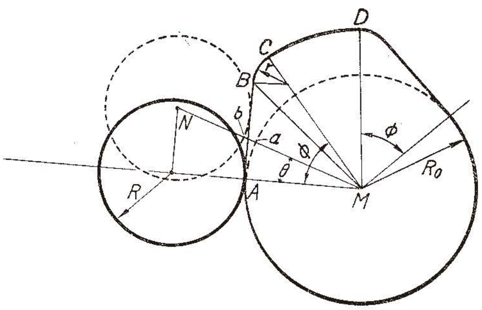


Fig. 12. Tangential Cam with rolloer follower

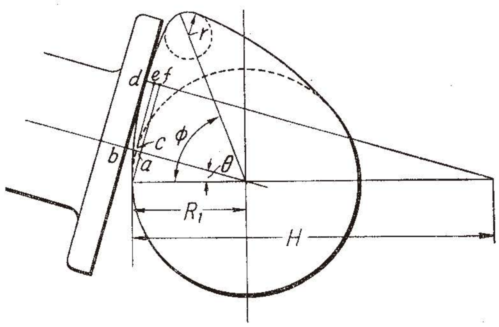


Fig. 13. Convex cam with mushroom follower

for convex cam with mushroom follower as shown in Fig 13

$$V_\theta = 30k_1 \cdot k_2' \cdot \cos\alpha \cdot C \cdot \frac{D^2}{N} \cdot \left\{\theta - 2\sin\frac{\theta}{2}\right\} \cdots (4)$$

for constant accelerative cam with roller follower as shown in Fig 14

$$V_\theta = \frac{\pi}{6} \cdot k_1 \cdot \cos\alpha \cdot C \cdot K \cdot \frac{D}{\omega^3} \cdot \theta^3 \; \cdots\cdots \; (5)$$

where $k_1 = d/D$, $k_2 = RD$, $k_2' = H/D$, D is cylinder bore, α is angle of valve seat, C is constant velocity of air flow, K is constant acceleration, N is engine speed, ω is angular velocity, θ is crank angle.

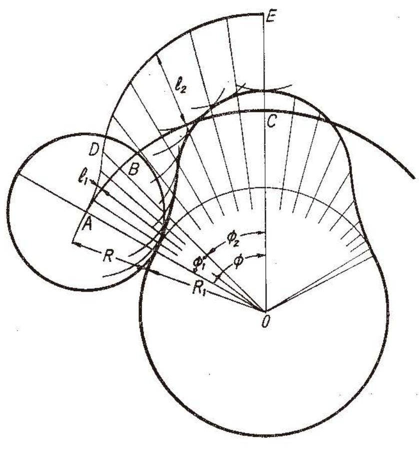


Fig. 14. Constant acceleration cam with roller follower

Secondly, the amount of air flow (V_θ) in the case of constant pressure difference (ΔP) is given respectively as follows

$$V_\theta = \frac{5}{2} \cdot k_1 \cdot k_2^3 \cdot .\cos^3\alpha \cdot \frac{\Delta P \cdot D^4}{h_1 \cdot \mu \cdot N} \left\{\sec\frac{\theta}{2} \cdot \tan\frac{\theta}{2} + \log\left(\sec\frac{\theta}{2} + \tan\frac{\theta}{2}\right) - 6\tan\frac{\theta}{2} + 6\log \cdot .\tan\left(\frac{\pi+\theta}{4}\right) - \theta\right\} \; \cdots\cdots \; (3')$$

$$V_\theta = \frac{5}{2} \cdot k_1 \cdot (k_2')^3 \cdot \cos^3\alpha \cdot \frac{\Delta P \cdot D^4}{h_1 \cdot \mu \cdot N} \left\{\frac{5}{2}\theta - \frac{15}{2}\sin\frac{\theta}{2} + \frac{3}{2}\sin\theta - \frac{1}{6}\sin\frac{3}{2}\theta\right\} \; \cdots\cdots\cdots \; (4)'$$

$$V_\theta = \frac{\pi}{672} \cdot k_1 \cdot \cos^3\alpha \cdot K^3 \cdot \frac{\Delta P \cdot D}{h_1 \mu \omega^7} \cdot \theta^7 \; \cdots\cdots\cdots \; (5)'$$

Supposing that there is no air flow practically when the air flow volume (V_θ) is less than a given value as mentioned above, it is clear from the equations that the ineffective angle or lift increases together with the engine speed (N), the size of engine (D) and decreases inversely with the pressure difference, which shows a good agreement qualitatively with the experimental results shown already in Figs. 8, 9, 10 and 11.

Moreover, all curves of the air flow ratio (V_θ/V_{30}) computed by the equation show a nearly coincidence with some experimental values obtained by air flow method as shown in Fig. 15. Accordingly the effect of such engine factors as D, N, ΔP etc. on the ineffective angle and lift can be also estimated roughtly, especially that for the constant acceleration cam with roller follower are easily calculated from Eq. (5) and Eq. (5)'.

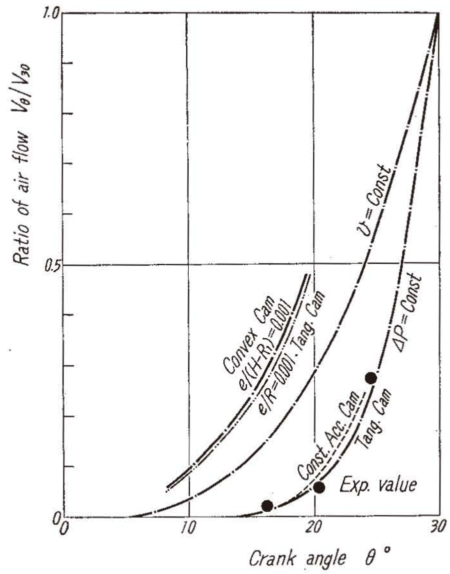


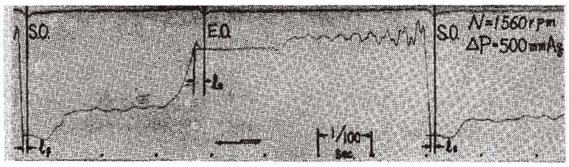
Fig. 15. Relation between Ratio of air flow and Crank angle

3.2 Results obtained by pressure diagrams

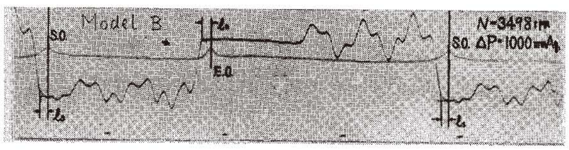
3.2.1 Case of model pieces of piston

Two typical pressure diagrams shown in Fig. 16 include the variation of cylinder pressure and also the marks of valve timing (S.O. and E.O.) for model pieces (A) and (B).

On the diagrams, it is easy to measure some lags, denoted by $l_s$or l_e, between the actual pressure change and the nominal valve openings, and calculate the ineffective angle with such lags, and finally convert it again to the ineffective lifts, which are shown in Fig. 14 and Fig. 15.



Model piece (A), $\Delta T=9/100$ mm, $\Delta\theta=-44°$



Model peice (B), $\Delta T=9/100$ mm, $\Delta\theta=-27°$

Fig. 16. Pressurdiagrams for model piece (A) and (B)

In the figures, all curves show a good concurrence with those obtained by the air flow method shown already in Figs. 10 and 11, and further it is noticed that the minimum ineffective lifts, 30/100 mm for model (A) and 18/100 mm for model (B), which were calculated from the pressure diagrams, are nearly equal to the values, 32/100 mm for model (A) and 19/100 mm for model (B), which were estimated by the air flow method.

Considering the good agreements, it is verified experimentally that it is possible to estimate the ineffective lift by means of the pressure diagrams instead of the measurement of air flow.

3.2.2 Case of regular piston head

When a regular piston is inserted, it is very difficult to estimate the ineffective angle or lift by the air flow method as mentioned already, while it is confirmed that both methods, i.e., air flow and pressure diagrams, indicate the same results successfully, so that some pressure variations at suction and exhaust ports were recorded simultaneously and are shown in Fig. 19 as a typicalexample.

The lag (l_s or l_e) between the pressure change and the nominal valve timing (S.O. or E.O.) in Fig. 19 is converted to the in effective lifts by the same procedure, and the latter is plotted against tne engine speed in Fig. 20 as a parameter of ${}_{\alpha}\theta_{\beta}$, where α is a crank angle of suction opening before top dead center ($T.D.C.$) and β is an angle of exhaust closure after $T.D.C.$ and then the valve overlap ($\varDelta\theta$) is equal to the sum of α and β.

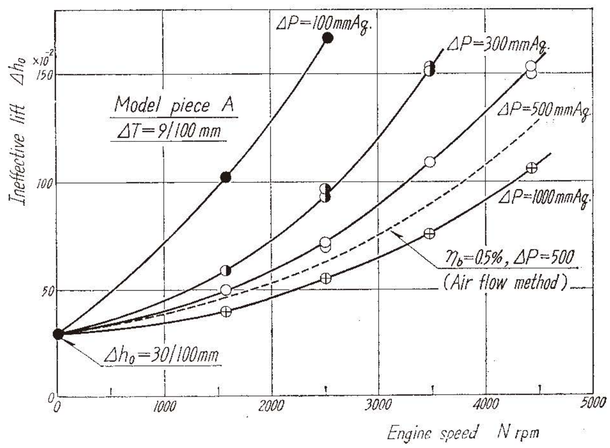


Fig. 17. Ineffective lift calculated from pressure diagram (Model piece A)

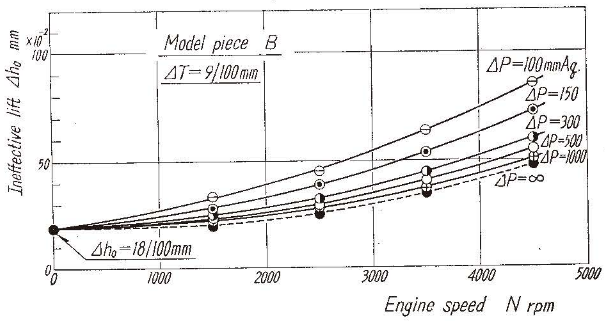


Fig. 18. Ineffective lift calculated from pressure diagram

In the figure, the ineffective lift increases with the engine speed in the same manner as shown in Figs. 10, 11 and 17 for the model pieces, however the minimum ineffective lift 10/100 mm, seems to be fairly smaller than thatfor model piece

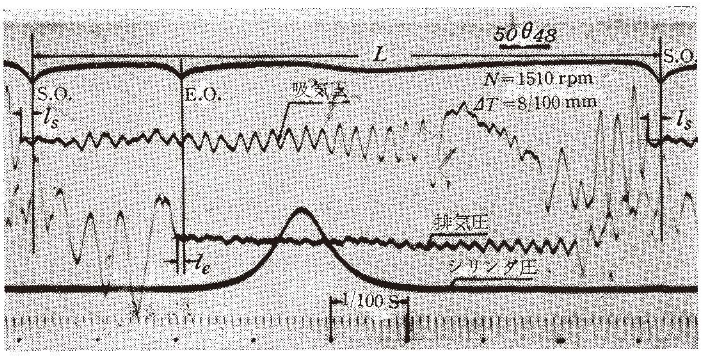


Fig. 19. Pressure diagrams for regular piston (4 cycle engine)

shown in Table 4, because the clearance volume is not only larger than those for model pieces, but also the throttling area for air flow through the clearance gap increases largely by the piston displacement.

If it is considered that the pressure difference between cylinder and suction or exhaust port is inversely proportional to the overlap $(\alpha+\beta)$, in Fig. 20 the pressure difference affects the ineffective lift similarly in the case of model pieces. Since the pressure difference at exhaust opening is usually

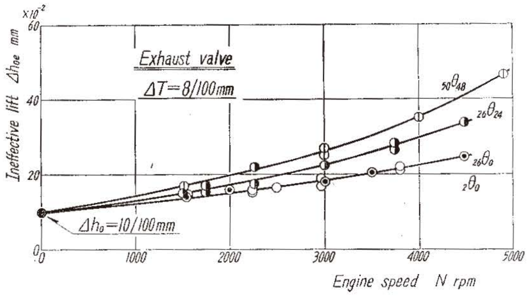


Fig. 20 (a). Ineffective lifts calculated from pressure diagrams (Regular piston, cycle)

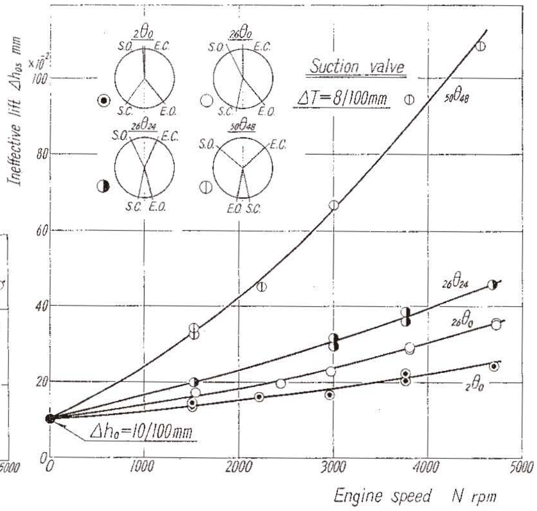


Fig. 20 (b). Ineffective cifts calculated from pressure diagrams (Regular piston, 4 cycle)

Table 4. Comparison of minimum ineffective lifts

Piston head		Model piece A	Model piece A	Regular piston 4 cycle	Regular piston 2 cycle
Minimum value of ineffective lift Δh_0 mm	Air flow method	32/100	19/100	—	—
	Pressure diagram method	30/100	18/100	10/100	7/100

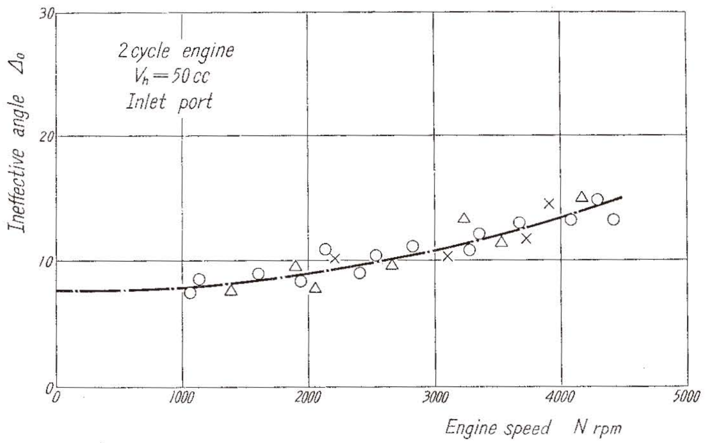


Fig. 21. Ineffective angle calculated from pressure diagrams (Regular piston, 2 cycle)

larger than that at suction opening, the ineffective lift at exhaust valve ($\varDelta h_{0e}$) is smaller than that at suction valve ($\varDelta h_{0s}$) as shown in Fig. 20.

In the next place, the ineffective angles at the port for two-cycle engine obtained from the pressure diagrams are plotted against the engine speed in Fig. 21.

Since the ports are usually opened and closed by the movement of piston, the lift of port opening at any crank angle can be determined by the piston displacement and the minimum value converted to the ineffective lift is shown in Table 4.

The minimum ineffective lift for two-cycle engine seems to be smaller than that for the poppet valves in four-cycle engine, however the ineffective angle at usual engine speed for two-cycle engine is nearly 10°.

4. Conclusion

The results reached are summarized as follows:

a) It is possible to estimate the ineffective angle and lift not only by the measurements of air flow, but also by the analysis of pressure diagrams.

b) The ineffective angle and lift increases with the engine speed and decreases inversely with the pressure difference, which shows a good agreements with the calculated results.

c) The minimum ineffective angle and lift remains constant invariably regardless of the engine speed and the pressure difference, and the minimum value increases with the air flow resistance.

d) It the ineffective lift is used instead of the ineffective angle, it is necessary at all to consider the effects of tappet clearance and the design of the cam and cam followers.

e) When using a regular piston, the nominal valve timing must be corrected by the ineffective lift, which is at least larger than 10/100 mm for the suction process and increases with the engine speed.

f) In practice, the ineffective lift at exhaust opening is considerably smaller than that at the suction process, because the pressure difference at the former is extremely larger than at the latter.

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