

Extension of Lean Burn Range by Intake Valve Offset

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ABSTRACT

Using a 109.2 cm³, four-stroke, single-cylinder, two-valve gasoline engine, improvement of fuel economy by extension of lean burn range has been attempted with invented way to intensify tumble flow from a simple mechanical arrangement. With a part of the intake valve was jutting out beyond the perimeter of the cylinder bore, the masking effects from the valve recess on top of the cylinder sleeve created a strong tumble flow, which enabled lean burn at an air fuel ratio leaner than the conventional design by two points. The motorcycle equipped with this engine attained better fuel economy by 5.7% to the base model when measured in Indian Driving Cycle (IDC). The outward-laid intake valve also increased the clearance from the exhaust valve, which enabled use of a large-diameter intake valve to minimize the reduction of maximum power.

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INTRODUCTION

In recent years, it is sought to reduce the CO₂ emission to cope with global warming. In the field of mobility, an improvement of fuel economy therefore has been a key theme for researches. Motorcycles are under the same demand, and various measures to improve fuel economy have been researched. The lean burn technology is one of the effective measures to improve fuel economy in gasoline fueled engines. Currently, in motorcycle category, motorcycles having small displacement engines share approximately 90% in the world.

In India, small motorcycles have a large market and the lean burn has been commonly applied as a principal technical solution for fuel economy improvement. For further improvement, extension of lean burn range is required. It has been known that such options as the multi-spark ignition system⁽¹⁾ which strengthens ignition, the application of sub combustion chamber⁽²⁾, the in-cylinder turbulence to accelerate combustion, etc. are effective for the said purpose. For example, with respect to the in-cylinder turbulence, researches have been conducted using a tumble-generator valve⁽³⁾ or a swirl-generator valve⁽⁴⁾. Application of such a device, however, requires additional parts and control devices.

In the meantime, researches have been conducted for other options such as reshaping of intake port⁽⁵⁾ and use of masking⁽⁶⁾ to intensify tumble flow from a simple mechanical arrangement without using additional parts. However, such an option tends to increase resistance in the intake passage, causing reduction of intake air volume. There have been only a few cases of researches on how the fuel economy improves from intensified tumble flow.

In this research, we have defined the partially open throttle tumble ratio as a new indicator of intensity of tumble flow, and clarified the relationship between the tumble flow intensity and the fuel economy improvement through the basic experiments using the stationary tumble generator flap (hereafter referred to as "flap") of various sizes. The tumble flow was further intensified to extend the lean burn range by the use of masking effects from the intake valve offset to improve fuel economy, and at the same time the intake valve diameter was increased to prevent lowering of maximum power resulting from the reduced intake air volume.

IDENTIFICATION OF LEAN-BURN LIMIT OF TEST ENGINE

First of all we have determined operating conditions to apply to assessment of test engine as well as identifying lean-burn limit of the test engine.

MAJOR SPECIFICATIONS OF TEST ENGINE

The commercially-available engine for small motorcycles was used for the tests. Table 1 shows the major specifications of the test engine. The engine was air-cooled, four-stroke, single-cylinder, two-valve gasoline engine with the displacement of 109.2 cm³. The fuel was supplied from the carburetor. It should however be noted that in the tests to alter the air fuel ratio (A/F), the pressure in the carburetor float chamber was regulated to control A/F. A rotary encoder was set on the crankshaft to obtain crank angle signals of 360 pulses per revolution. For measurement of the pressure in the cylinder, the piezoelectric pressure transducer (Kistler 6041A) was used. The engine was linked with the dynamometer via the transmission system, and controlled at constant engine speeds and loads. The tests by the unit engine were conducted at a constant transmission ratio.

Table 1. Specifications of experimental engine

Engine type	Air cooled, four-stroke, single-cylinder, gasoline engine
Bore × Stroke [mm]	50.0 × 55.6
Displacement [cm ³]	109.2
Compression ratio	9.0
Valve diameter [mm × number]	Intake : 25.5×1 Exhaust : 21.0×1
Valve lift [mm]	Intake : 5.8 Exhaust : 5.6
Ignition system	Electronically controlled capacitor discharge ignition
Fuel system	Carburetor
Maximum brake power output	6.27 kW at 8000 r/min
Maximum brake torque	8.72 Nm at 6000 r/min

CONDITIONS OF TEST OPERATIONS

The typical operating conditions to apply to assessment of the test engine were determined as per the following. Fig. 1 shows Indian Driving Cycle (IDC). Fig. 2 shows frequencies of appearance of engine speed and throttle opening in IDC. Except for idling, the engine speed was frequent at 3750 r/min, and the throttle opening was frequent at 13 to 15 degree.

The combination of engine speed at 3750 r/min and throttle opening at 15 degree (BMEP 400 kPa), which was the frequently-used condition in IDC and was considered as the most appropriate condition for testing, was defined as typical operating condition.

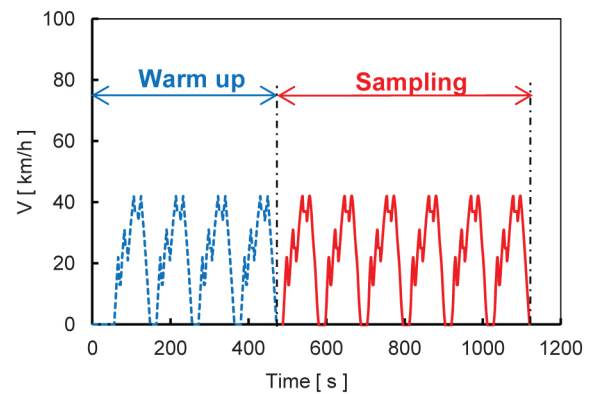


Fig. 1. IDC

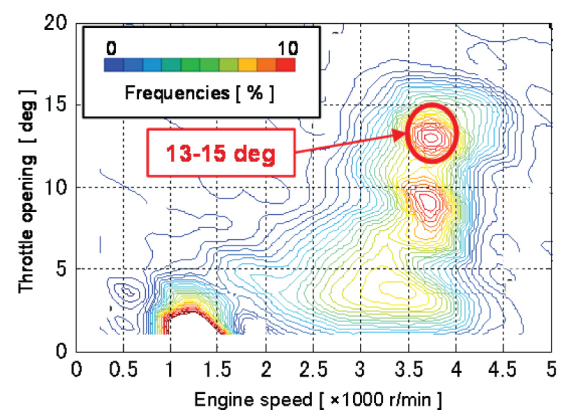


Fig. 2. Frequencies of appearance of engine speed and throttle opening in IDC

LEAN-BURN LIMIT OF TEST ENGINE

The relation between the setting area of A/F and the lean burn limit was investigated to the test engine. Fig. 3 shows the frequencies of appearance of engine speed and A/F in IDC. Except for idling, A/F of 16.0 to 16.5 most frequently appeared.

Fig. 4 shows the Brake Specific Fuel Consumption (BSFC) and the Coefficient Of Variation of Indicated Mean Effective Pressure (COV_{IMEP}) when the A/F was altered under the typical operating conditions. The BSFC became smallest at the A/F of 16.5. The test engine was already set at that A/F in the typical operating condition. The COV_{IMEP} at the A/F of 16.5 was 5%. As an increase of COV_{IMEP} occurred if the A/F was further shifted to lean side, a further reduction of BSFC was thought difficult. To further lower BSFC, an extension of lean burn range by an increase of turbulence in the cylinder, etc. was necessary. In this research, we focused on intensifying of tumble flow.

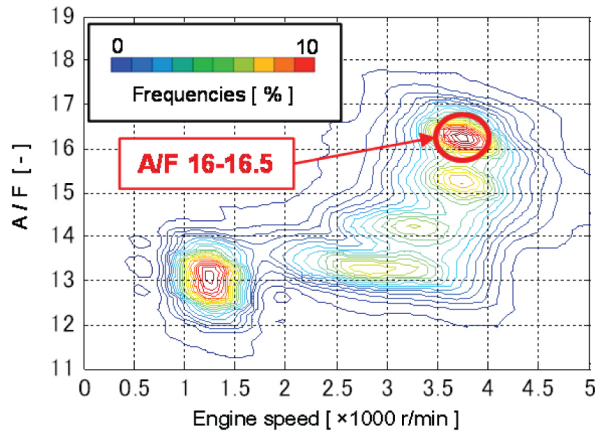


Fig. 3. Frequencies of appearance of engine speed and A/F in IDC

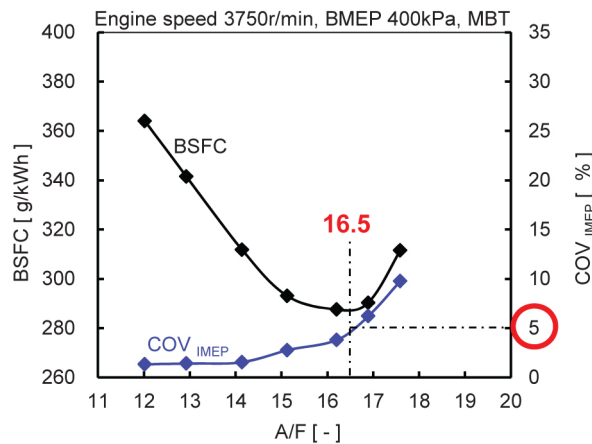


Fig. 4. BSFC and COV_{IMEP}

INDICATOR OF TUMBLE FLOW INTENSITY AND PROSPECTIVE OUTPUT IN CONSTANT FLOW TESTS

In this research, the tumble ratio (TR) was used as the indicator of intensity of tumble flow. Also, as the indicator of prospective power output, the integrated value of effective intake valve opening area over the crank angle (Z_i) was used. These indicators were measured in the steady flow rig tests.

DEFINITION OF TUMBLE RATIO

The TR is expressed by the ratio of the tumbling angular velocity (ω_R) of the air in the cylinder against the angular velocity of crankshaft (ω_E) when the charging of mixture is completed, and defined by the equation (1), (2) and (3)(7). In this research, since TR was estimated by the relative value instead of an absolute value, volumetric efficiency (η_v) was simply treated as 1.

$$TR = \frac{\omega_R}{\omega_E} = \frac{L_d \eta_v \int_{\alpha_1}^{\alpha_2} C_f N_r d\alpha}{(\int_{\alpha_1}^{\alpha_2} C_f d\alpha)^2} \quad (1)$$

$$L_d = \frac{BS}{n_v D_v^2} \quad (2)$$

Where,

TR : Tumble ratio

ω_R : Tumble angular velocity

ω_E : Angular velocity of crankshaft

L_d : Engine shape factor

η_v : Volumetric efficiency

α_1 : Crank angle at inlet valve opening

α_2 : Crank angle at inlet valve closing

C_f : Flow coefficient

N_r : Non-dimensional rig tumble

B : Cylinder bore

S : Engine stroke

n_v : Number of intake valves

D_v : Intake valve diameter

N_r is expressed by the ratio of tumbling angular velocity against the flow velocity in the intake port.

$$N_r = \frac{\omega_R B}{V_0} = \frac{8M}{\dot{m} V_0 B} \quad (3)$$

Where,

V_0 : Theoretical intake flow velocity

M : Torque measured by tumble flow meter

$\dot{m}$: Air mass flow rate through port

TR was assessed by the following two methods, actual measurements in the steady flow rig and the CFD analysis which simulated steady flow test.

Fig. 5 shows the test equipment used for the steady flow tests. Table 2 shows the conditions of the steady flow tests. In this research, TR was also assessed at partial throttle openings. As the intake resistance increases at a partial opening, the TR lowers. It was therefore necessary to improve accuracy of TR measurement. The compact impulse tumble meter for exclusive use was prepared, and the experiment was conducted.

Fig. 6 shows the CFD analysis model used for the TR calculation. Table 3 shows the calculation conditions of CFD analysis.

Table 2. Conditions of steady flow test

Valve lift [mm]	1, 2, 3, 4, 5, 6
Throat diameter [mm]	0.535, 1.31, 2.75
Pressure of surge tank B (ΔP_2) [kPa]	5
Load cell rated capacity [N]	0.5
Rated torque [Nm]	5×10^{-3}

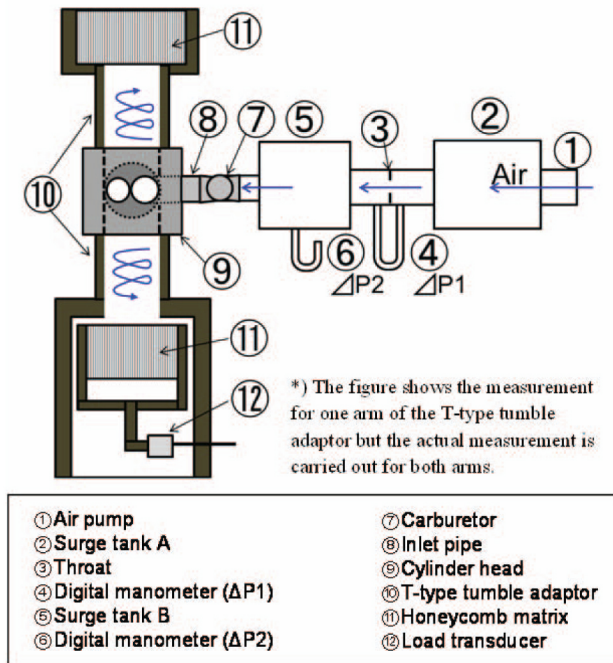


Fig. 5. Flow test rig

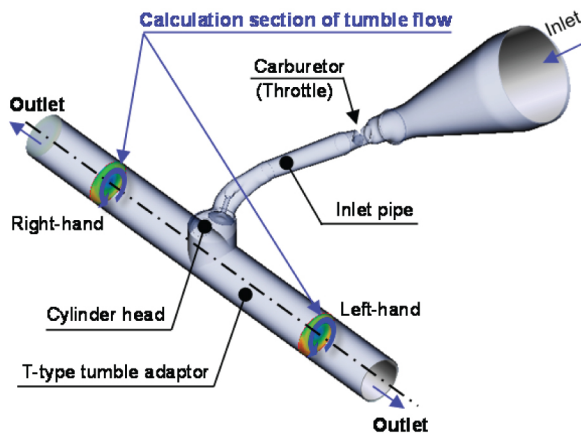


Fig. 6. CFD model for tumble ratio calculation

Table 3. CFD conditions for steady flow test

Software name	SCRYU / Tetra
Solution type	Steady state flow
Valve lift [mm]	1, 2, 3, 4, 5, 6
Pressure difference of Inlet and outlet [kPa]	5
Cell type	Tetrahedron
Boundary cell type [mm × layers]	Prism 0.3 × 2
Number of cells	4,500,000
Differencing scheme	MUSCL scheme
Pressure velocity coupling	SIMPLEC
Turbulent model	Standard k-ε

PROSPECTIVE POWER OUTPUT INDICATOR

The factor most influential for the engine power output is the amount of air drawn into the combustion chamber in a cycle. As an indicator for assessment of the amount of intake air, Z_i in the steady flow tests is effective, and defined by the equation (4). On the other hand, Z_i is defined by the equation (5) at the CFD analysis⁽⁸⁾. And Fig. 7 shows conceptual diagram of Z_i . In this research, Z_i was used as an indicator of prospective power output.

$$Z_i = \int_{\alpha_1}^{\alpha_2} Z d\alpha = \int_{\alpha_1}^{\alpha_2} \beta A \sqrt{\frac{\Delta P_1}{\Delta P_2}} d\alpha \quad (4)$$

Where,

Z_i : Integrated value of effective intake valve opening area over the crank angle

Z : Effective intake valve opening area

β : Flow coefficient of the throat

A : Cross section of the throat

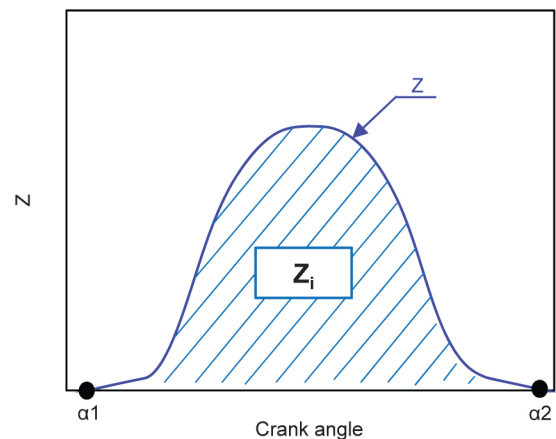
ΔP_1 : Pressure difference of the throat

ΔP_2 : Pressure of the surge tank B

$$Z_i = \int_{\alpha_1}^{\alpha_2} Z d\alpha = \int_{\alpha_1}^{\alpha_2} C_f A_v d\alpha \quad (5)$$

Where,

A_v : Intake valve opening area

Fig. 7. Definition of Z_i