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Simulation of Unadiabatic Intake Flow of 4 Cycle Engine*

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In this paper an equation for an unadiabatic mass flow rate through the nozzle is derived by application of polytropic analysis.

The author performed experiment to estimate validity of the flow equation, and found difference in flow coefficient between adiabatic and unadiabatic flows. This difference can be analyzed by laminar boundary layer theory which considers temperature distribution.

Simulation of intake process of a 4 cycle engine was done by solving this flow equation and energy equation in a cylinder by Runge-Kutta-Gill method.

Heat transfer at poppet valve affects flow rate only when valve lift is small, so it can change the total intake flow rate by a few percent. But, heat transfer in a cylinder can change the total intake flow rate by 10 or 20%, because it affects the cylinder pressure which governs the pressure difference between before and behind poppet valve.

1. Introduction

Volumetric efficiency has a great influence on power of internal combustion engine. We have many excellent engines, but don't have the method by which we can compute volumetric efficiency accurately.

Considering the influence which the heat transfer from valve and cylinder wall to intake gas has on volumetric efficiency, two factors can be recognized:

(1) Flow equation through intake valve (heat transfer from valve to gas can affect volumetric efficiency through this factor)

(2) Cylinder pressure (heat transfer from cylinder wall to gas can affect volumetric efficiency through this factor)

Concerning the first point, Kapadia⁽¹⁾ experimented on the effect of heat transfer on steady flow through a poppet valve. His experimental data are valuable, but he did not analyze his results.

Concerning the second point, Borman⁽²⁾ evaluated Cook's, Whitehouse's and Huber & Brown's papers about cycle simulation and he made simulation and experiment of Compression-Ignition engine. In his paper he computed volumetric efficiency using the traditional adiabatic mass flow equation.

In this paper, the author derived an unadiabatic mass flow equation from the fundamental equation of thermodynamics and evaluated this equation through experiment. Using this flow equation, the author

made simulation of intake process and studied the influence which the heat transfer from engine wall to intake gas has on volumetric efficiency.

2. Notations

e : internal energy
 i : enthalpy
 v : volume
 q : heat transfer rate
 s : entropy

In these notation, a big letter means an absolute value of the system and a small letter means the value per unit weight.

p : pressure
 T : temperature
 c_v : constant volume specific heat
 c_p : constant pressure specific heat
 c_n : polytropic specific heat
 κ : specific heat ratio
 n : polytropic index
 R : gas constant
 A : ratio of heat to work
 W : mass flow rate
 G : cylinder gas weight
 α : flow coefficient
 a : cross section area of nozzle
 L : slit length
 L' : modified slit length
 w : velocity
 g : acceleration of gravity
 δ : boundary layer thickness
 SD : slit distance

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Subscripts

- 1: before nozzle
- 2: behind nozzle
- ad: adiabatic
- pol: unadiabatic (when heat transfer exists)
- a: atmosphere
- t: temperature

3. Unadiabatic mass flow equation

Traditional mass flow equation through the nozzle is derived on the assumption that the process of change between before and behind nozzle is adiabatic. So we cannot use the traditional mass flow equation when heat transfer from nozzle wall to gas exists. In this chapter, the relation between polytropic index and heat transfer rate is described on the assumption that specific heat is constant in the process of change between before and behind nozzle. We can get an unadiabatic mass flow equation by the integration of one-dimensional steady flow momentum equation in this relation.

3.1 Fundamental equation

We can make discussion here on two assumptions. The one is that there are no temperature and velocity distributions, that is, gas has no viscosity and thermal conductivity of gas is indefinitely large. The other is that gas is an ideal one.

The first law of thermodynamics per unit weight is

$$de = c_v dT = -Apdv + dq \quad (1)$$

Using average specific heat (that is polytropic specific heat), dq in the right side is

$$dq = c_n dT \quad (2)$$

We can integrate Eq. (1), using Eq. (2) and equation of state. Then the polytropic index is

$$n = \frac{c_p - c_n}{c_v - c_n} \quad (3)$$

Using Eq. (3), we can get temperature variation in the polytropic process. Then integral of Eq. (2) is

$$q = c_n \left\{ \left(\frac{p_1}{p_2} \right)^{(c_v - c_p)/(c_p - c_n)} - 1 \right\} T_1 \quad (4)$$

In Eq. (4), if pressure ratio is known we can compute c_n from given q , and then n from Eq. (3). We must solve Eq. (4) numerically and can get a numerical solution by a short run of computer. You will find description of the method of numerical computation in appendix A. Neglecting the potential energy and friction forces, one-dimensional steady flow momentum equation is

$$\frac{1}{g} d \left(\frac{w^2}{2} \right) = -v dp \quad (5)$$

Now we can compute n from Eqs. (3) and (4). Equation (5) can be integrated by polytropic process. Then unadiabatic mass flow equation is

$$W_{pol} = \alpha_{pol} a \sqrt{\frac{2gn}{n-1} \frac{p_1}{v_1} \left(\left(\frac{p_2}{p_1} \right)^{2/n} - \left(\frac{p_2}{p_1} \right)^{(n+1)/n} \right)} \quad (6)$$

There are two differences between adiabatic and unadiabatic mass flow equations. Specific heat ratio (κ) in adiabatic mass flow equation is put to polytropic index (n) and unadiabatic flow coefficient is different from adiabatic one. The latter is because the velocity distribution is different between two flows. You will see this point explained in Chapter 4.

Now explanation of the first point is done using the i - s chart in Fig. 1. In adiabatic flow, the variation through the nozzle is 1→2 and integral of momentum equation through the nozzle is equal to the enthalpy change through the nozzle. In unadiabatic flow, when gas through the nozzle is heated, the variation through the nozzle is 1→3 or 4, and when gas through the nozzle is cooled, the variation is 1→5. The variation 1→3 means that in spite of heat addition the temperature decreases owing to expansion. In this case c_n is negative [see Eq. (2)]. The variation 1→4 means that temperature increase by heat addition is larger than temperature decrease by expansion and in consequence the temperature increases. In every case, integral of momentum equation is equal to the sum of enthalpy change and heat transfer rate.

3.2 Numerical solution of the mass flow equation

Now we can compute unadiabatic mass flow through the nozzle by Eqs. (3), (4) and (6). Figures 2 and 3 are examples of air flow.

Figure 2 shows the relation between heat transfer rate and polytropic index. Heat transfer ratio given on the abscissas of Fig. 2 and 3 is the ratio of heat transfer rate to adiabatic enthalpy change $[(i_1 - i_2)$ in Fig. 1]. In heat transfer ratio, zero means adiabatic, positive means heat addition, and negative means heat subtraction. In Fig. 2 pressure ratio through the nozzle does not have large influence on the rela-

Fig. 1 Explanation by i - s chart

tion, especially in positive n region.

Figure 3 shows how the flow rate changes when heat transfer exists. The abscissa is the heat transfer ratio mentioned above and the ordinate is the ratio of unadiabatic to adiabatic mass flow rate under the parameterized pressure ratio on the assumption that flow coefficients are same between two flows. c in Fig. 3 means choke state. According to detailed computation, velocity change by heat transfer is rather small, but temperature change by heat transfer is large and density change is equally large. Then mass flow rate increases by heat addition and decreases by heat subtraction.

For example, we assume that pressure ratio through the nozzle is 0.8 and heat which is twice the adiabatic enthalpy change is transferred from nozzle wall to gas. From Fig. 2 polytropic index is 0.8 and from Fig. 3 mass flow rate is 8.6% smaller than adiabatic flow of 0.8 pressure ratio. According to detailed computation, the velocity at the nozzle is 195 m/sec which is faster by 6 m/sec than in adiabatic flow, and temperature at the nozzle is 32°C which is higher by 33°C than in adiabatic flow.

In this example we neglect velocity and temperature distribution. The difference between adiabatic and unadiabatic in this point is expressed as flow coefficient. The author will explain it in next chapter because it largely depends on flow path.

4. Experiment and consideration by laminar boundary layer model

In Chapter 3 the viscosity and thermal conductivity of gas are neglected. Using Eq. (3), we

Fig. 2 Relation between transferred rate and polytropic index

must consider these factors in flow coefficient. The experiment in this chapter is done about this point and it simulates the intake valve of a 4 cycle engine.

4.1 Apparatus

The apparatus is shown in Fig. 4. Two dimensional slit nozzle made of bakelite in a square flow path of acryl simulates an axially symmetric three dimensional nozzle at intake valve of a 4 cycle engine. The slit is 10 mm in axial length and 60 mm in width. A stainless steel foil which is 50 μ in thickness is pasted on the slit nozzle and is heated directly by electricity.

In the experiment, with the pressure difference constant between before and behind nozzle, how the mass flow rate changed by heat addition was measured. The slit distance was set at 0.7, 1.2, 2.0, 2.6 mm. The ratio of pressure before and behind the slit nozzle was set in twelve steps of 0.96 to 0.72.

Thermal conduction in transverse direction can be neglected in this stainless steel foil because it is very thin and heat transfer from the slit wall takes place under the condition of constant heat flux. There are thermocouples at the points in slit in Fig. 4. Heat flow which transfers to bakelite is measured to be below 1% of heat addition by electricity.

4.2 Result

In the experiment the ratio of unadiabatic to

Fig. 3 Variation of flow rate in weight by heat transfer

Fig. 4 Outline of experimental apparatus

adiabatic mass flow rate was measured. Then we could compute the ratio of unadiabatic to adiabatic flow coefficient by adiabatic and unadiabatic flow equations [that is traditional flow equation and Eq. (6)].

The result of experiment for slit distance 1.2 mm is shown in Fig. 5. The abscissa is the difference between slit wall temperature and ambient temperature and the ordinate is the flow coefficient ratio. Various symbols mean pressure ratios through the nozzle shown in Fig. 5. At every pressure ratio the flow coefficient ratio changes similarly in accordance with the difference between slit wall temperature and ambient temperature.

Flow coefficient is decided chiefly by boundary layer near slit wall and exfoliation. Both are subject to Reynold's number. As described in section 3-2, velocity does not change largely with heat addition. So Reynold's number is subject to viscosity depending upon temperature which is represented on the abscissa in Fig. 5. This is the reason why the flow coefficient ratio changes similarly at every pressure ratio.

For various slit distances experimental results were arranged as shown in Fig. 5. Figure 6 summarizes these results. In Fig. 6 the variation of flow coefficient ratio is in inverse proportion to slit distance. Flow coefficient ratio is subject to the ratio of $(\delta_{pol} - \delta_{ad})$ in Fig. 7 to slit distance; $(\delta_{pol} - \delta_{ad})$ means difference in boundary layer thickness or exfoliation

Fig. 5 Variation of flow coefficient ratio by heat addition

Fig. 6 Experimental results in various slit distances

quantity between unadiabatic and adiabatic flow. Then Fig. 6 means that variation in $(\delta_{pol} - \delta_{ad})$ with slit wall temperature does not depend upon slit distance.

4.3 Analogy between slit and flat plate

The flow through slit nozzle is very complicated, but only to compute the variation from adiabatic to unadiabatic flow we can perhaps make analogy between slit and flat plate because the variation does not depend on slit distance, as we know from Fig. 6. The analogy is symbolized in Fig. 7. We can compute the variation in boundary layer thickness with heat addition when flat plate is laid in a parallel laminar flow.

In this respect the author referred to the paper written by Chapman & Rubesin⁽³⁾. Computing the boundary layer thickness at slit nozzle by using the analogy between slit and flat plate, the author introduced a modified slit length (L' in Fig. 7). It means that after the slit δ continues to grow because the gas behind the slit which doesn't have large velocity pulls the gas in the center which has large velocity. L' is decided by the next equation.

$$\left. \begin{array}{ll} w < w_0 & L' = L \\ w > w_0 & L' = \frac{w}{w_0} \times L \end{array} \right\} \dots\dots\dots (7)$$

(w : velocity at the end of the slit nozzle)

From experiment w_0 is 80 m/sec and the pressure ratio through the nozzle is 0.96. Qualitatively speaking, Eq. (7) means that L' becomes long in proportion to an increase of velocity at the end of the slit nozzle.

Now we can compute δ_{ad} or δ_{pol} in Fig. 7 and so we can also compute flow coefficient ratio by the method described in appendix B. Using this flow coefficient ratio in Eq. (6) and the traditional adiabatic mass flow equation, we can compute the ratio of unadiabatic to adiabatic mass flow rate.

Figure 8 is the comparison of computation with experimental result. The abscissa is heat flux from slit wall to gas. This is a case of 1.2 mm slit distance and 0.8 pressure ratio through the nozzle. In every case of slit distance and pressure ratio, the author could draw same figure as Fig. 8. In Fig. 8 tendency of accordance between computation and experiment is

Fig. 7 Analogy between slit and flat plate

average.

After all it was shown that the ratio of unadiabatic to adiabatic flow coefficient could be computed by introduction of modified slit length (L') on the assumption that the analogy between slit and flat plate in parallel laminar flow held good.

4.4 Flow equation in large slit distance

We assumed in derivation of Eq. (6) that there is no temperature distribution in vertical direction of slit wall, but there is temperature distribution as shown in Fig. 9.

If we assume that the flow through slit is laminar, the temperature distribution will be as described by a solid line. Actually the temperature distribution is as described by a dotted line because the flow is turbulent. Then it is possible in large slit distance that the gas in the center of slit is not heated. For reference, the laminar temperature boundary layer thickness at the end of slit is 0.25 mm when the velocity through slit is 80 m/sec. This becomes thinner with an increase of the velocity through slit. Thus the gas flowing in the center of slit can change adiabatically between before and behind slit in spite of heat addition when slit distance is larger than a certain fixed value. The fixed value should be decided by experiment because it depends largely on flow path.

Fig. 8 Comparison of computed flow quantity by laminar parallel flow model with experiment

Fig. 9 Temperature distribution at the end slit nozzle

In this way the unadiabatic mass flow equation in wide nozzle becomes as follows. It only differs in flow coefficient from the adiabatic mass flow equation

$$W_{pol} = \alpha_{pol} a \sqrt{\frac{2gK}{K-1} \frac{p_1}{v_1} \left\{ \left(\frac{p_2}{p_1} \right)^{2/K} - \left(\frac{p_2}{p_1} \right)^{(K+1)/K} \right\}} \quad (8)$$

So the unadiabatic mass flow equation is Eq. (6) in narrow nozzle and Eq. (8) in wide nozzle.

In this experiment the boundary slit distance between Eqs. (6) and (8) is about 2 mm. Figure 10 is the comparison of computations by Eq. (6) and (8) with experimental result in 2.6 mm slit distance. The spots represent experimental results. The solid line represent the computation by Eq. (6) and the dotted line does the one by Eq. (8). It can be judged that unadiabatic mass flow equation is Eq. (8) in 2.6 mm slit distance.

4.5 Comparison with Kapadia's experiment

Kapadia experimented on the effect of heat transfer on steady flow through a poppet valve. His experiment is summarized in Fig. 11. Pressure differences through a poppet valve are 40 and 20 inches in water head. The poppet valve is heated by 300 watts electrical heater.

Using his data, the author computed the rela-

Fig. 10 Comparison of computed flow quantity with experiment when slit distance is large

Fig. 11 Comparison with Kapadia's experiment

tion between flow reduction and valve lift by the method described in this paper. But the author couldn't compute completely because his detailed data were not available.

In computation the length of valve sheet (L_v in Fig. 11) corresponded to slit length (L) and valve sheet temperature was selected in three values as shown by three lines in Fig. 11. In small valve lift experimental results have a tendency to coincide with computation of higher slit wall temperature. And in large valve lift they have a tendency to coincide with computation of lower one. On the whole we can find similarity in the tendency of variation between Kappadia's experiment and author's computation.

5. Simulation of intake process

Borman⁽²⁾ did cycle simulation using the traditional adiabatic flow equation. In this chapter simulation of intake process using the unadiabatic mass flow equation is explained.

The simulation model is summarized in two equations.

(1) Energy equation in cylinder

$$dE = d(Ge) = -ApdV + dQ + hdG \dots\dots\dots (9)$$

Under the assumption that the gas is ideal, Eq. (9) can be transformed into Eq. (10), using the equation of state.

$$\frac{dT}{dt} = - \left(\frac{AR}{c_v} \frac{1}{V} \frac{dV}{dt} + \frac{1}{G} \frac{dG}{dt} \right) T + \frac{c_p T_a}{c_v G} \frac{dG}{dt} + \frac{1}{c_v G} \frac{dQ}{dt} \dots\dots\dots (10)$$

(2) Unadiabatic mass flow equation at an intake valve. Eqs. (6), (8)

When motoring is simulated, the traditional adiabatic mass flow equation is used. These two equations were solved by Runge-Kutta-Gill method. Table 1 and Fig. 12 show the results of simulation. The timing of initiation of intake process was fixed at twenty degrees after the top dead center. Engine speed was set at 2 000, 4 000 and 6 000 rpm. The volumetric efficiency was computed on next three cases.

(a) Motoring (it means no heat addition to gas)

(b) Heat transfer at the intake valve is considered. In this case, unadiabatic mass flow equation at the intake valve is used in the application of Kap-

padia's heat transfer equation at the intake valve backface and under the assumption that valve sheet temperature is 400°C constant. To estimate effect of unadiabatic mass flow equation, the volumetric efficiency by adiabatic mass flow equation was computed using the cylinder pressure of this case. The values in parenthesis are these results.

(c) In addition to heat transfer at the intake valve, heat transfer in cylinder is considered by Borman's computed data using Annand's⁽⁴⁾ type heat transfer equation.

As a result the difference between the volumetric efficiency by unadiabatic mass flow equation and the one by adiabatic mass flow equation is only a few percent because volumetric efficiency depends on intake gas quantity at large valve lift which is affected little by consideration of heat transfer in mass flow equation as described in Fig. 11. On the contrary, heat transfer from cylinder wall to gas changes cylinder pressure as described in Fig. 12 and volumetric efficiency becomes smaller by 10~20 percent than the one when it is not considered.

6. Conclusions

(1) Unadiabatic mass flow equation through the nozzle was derived. The following can be shown by using unadiabatic mass flow coefficient.

a) The whole gas at the nozzle is heated or cooled by heat transfer from nozzle wall. (Narrow nozzle)—Eq. (6)

b) Only the gas near the nozzle wall is heated or cooled by heat transfer from nozzle wall. (Wide nozzle)—Eq. (8)

(2) The difference in flow coefficient of slit nozzle between adiabatic and unadiabatic slit walls could be estimated by the difference in the laminar boundary layer thickness between adiabatic and unadiabatic flat plates in parallel laminar flow by introducing modified slit length.

(3) It was shown by simulation of intake

Fig. 12 Effect of heat transfer to cylinder pressure

Table 1 The valuation of volumetric efficiency by heat transfer from engine wall

Revolu- tion rpm	Motor- ing	Heat transfer near intake valve	Heat transfer in cylinder
2 000	87.0	84.6(87.1)	59.7
4 000	83.3	81.4(83.9)	71.4
6 000	76.8	75.4(76.8)	69.4

Fig. 13 Outline of numerical computation

process that the heat transfer near the intake valve could affect volumetric efficiency by a few percent, but heat transfer from cylinder wall could affect volumetric efficiency by 10 or 20% due to the change of cylinder pressure.

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Appendix A

In Eq. (4), if pressure ratio is known, c_n can be computed from given q as follows. Equation (4) can be divided into next two equations.

$$y_1 = \frac{q}{c_n T_1} \quad (11)$$

$$y_2 = \left(\frac{p_1}{p_2} \right)^{(c_v - c_p)/(c_p - c_n)} - 1 \quad (12)$$

When c_n is taken on abscissa and y on ordinate, these two equations are drawn in Fig. 13. Then the solu-

Fig. 14 Trapezoid type velocity distribution

tion of Eq. (4) is obtained as the point of intersection of two lines in Fig. 13. The value of c_n at the point of intersection can be determined by the method shown in Fig. 13 as a stepped line. We need only a few seconds to get the solution from a digital computer.

Appendix B

If adiabatic and unadiabatic boundary layer thicknesses at the end of slit nozzle are known, the flow coefficient ratio is computed as follows on the assumption that the velocity distribution is trapezoid.

In Fig. 14 the adiabatic velocity distribution at the end of slit nozzle is trapezoid ABEF and the unadiabatic velocity distribution is trapezoid ABGH under the condition as follows:

$$CE = \delta_{ad}, \quad CG = \delta_{pol} \quad (13)$$

Then adiabatic and unadiabatic flow coefficients are;

$$\alpha_{ad} = \frac{\Delta ABEF}{\Delta ABCD} \quad (14)$$

$$\alpha_{pol} = \frac{\Delta ABGH}{\Delta ABCD} \quad (15)$$

Dividing Eq. (15) by Eq. (14) and using Eq. (13)

$$\frac{\alpha_{pol}}{\alpha_{ad}} = \frac{\Delta ABGH}{\Delta ABEF} = \frac{SD - \delta_{pol}}{SD - \delta_{ad}}$$

See reference (3) as for the method of computation of δ_{ad} and δ_{pol} .

References

- (1) Kapadia, D.N., et al., *SAE Paper*, No. 670479 (1968).
- (2) Borman, G.L., et al., *SAE Trans.*, Vol. 74 (1966), p. 560.
- (3) Chapman d.r. and Rubesin, M.W., *Jour. Aeron. Sci.*, 16 (1949-7), p. 547.
- (4) Annand, W.J.D., *Proc. Inst. Mech. Engr.*, Vol. 177, No. 86 (1963), p. 973.
- (5) Gomei, Y., *M.S. Thesis, University of Tokyo* (in Japanese), 1970.

Discussion

T. OGURI (Yokohama National University):

The author said in Chapter 5 that heat transfer in cylinder is considered by Borman's computed data using Annand's type heat transfer equation. But Annand's type heat transfer equation is applicable only to compression and expansion process. How did the author think to apply Annand's equation to intake process?

Authors' closure

In this paper the author used the heat transfer quantity from cylinder wall to gas in intake process which was computed by Borman applying Annand's type heat transfer equation. Borman's computation and experiment can be summarized as follows. (See reference (2) as to the detail)

Annand's equation is a pipe-flow type heat transfer equation considering radiation from burning gas to cylinder wall. Using this equation, Borman did the cycle simulation and experiment. He took the coefficients in Annand's equation such that heat transfer quantity per one cycle in the cycle simulation might be equal to the one per one cycle which was released to coolant water in the experiment. It is not stated in reference (2) but the author thinks that Borman neglected the radiation from wall to gas in intake process as well as in compression process.

Thus Borman obtained the heat transfer quantity between cylinder wall and cylinder gas per crank angle

through one cycle. The author used Borman's heat transfer quantity per crank angle in intake process. It is not accurate quantitatively because the engine simulated in this paper is different from Borman's engine. But it may be called a method by which the influence of heat transfer in cylinder on the volumetric efficiency is considered.

The difference between two engines is as follows

	Borman's engine	Engine in this paper
Cylinder diameter	105 mm	70 mm
Cylinder stroke	110 mm	76 mm
Compression ratio	16	8.6