Turbulent Premixed Flames in Combustion Chamber under Engine-like Conditions

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ABSTRACT

A thick burning zone including unburned and burned gas is observed in the turbulent premixed flames under engine conditions. The turbulent burning velocity and burning zone thickness in the cylinder of the spark-ignition engine were compared with those in the constant volume vessels. The burning zone thickness has a good correlation with the value obtained by dividing turbulent burning velocity by laminar burning velocity under various conditions of pressure (atmospheric pressure to 1.6 MPa), unburned gas temperature (300 ∿ 620 K), fuel (propane or methane) and equivalence ratio (0.8 ∿ 1.4). By investigating the relation between Damköhler number and turbulence Reynolds number, it is confirmed that the regimes of the turbulent flames in the constant volume vessel are similar to those in the spark-ignition engine. Research in constant volume vessels is useful for the study of combustion processes in engine cylinder.

INTRODUCTION

In order to understand the combustion processes in spark-ignition engine cylinder and to predict it precisely, it is necessary to grasp the structure of turbulent premixed flames in detail. There are many studies about turbulent flame structures. Cross sections of the turbulent flames in spark-ignition engines are observed with a laser sheet[1-4]. The structures of the turbulent flames are investigated by measuring an ion current[5-8]. The spectrum for the roughness of the flame front by image analysis is compared with that of the turbulence characteristics[9]. A thick burning zone including unburned and burned gases is confirmed to exist in the turbulent premixed flames under the engine-like conditions. It is very interesting to study how this burning zone is characterized.

Regimes of the turbulent flames are approximately estimated by investigating the relation between Damköhler number and turbulence Reynolds number. It is shown that the combustion regimes in spark ignition engines are approximately located in the reaction sheet model[10,11].

Many investigators have studied turbulent premixed flames by using constant volume vessels because it is relatively easier to take measurement in these vessels. However, the experimental conditions seem to be different from higher pressure and higher unburned gas temperature in spark-ignition engines.

Turbulent burning velocity (relative entrainment speed against the average turbulent flame front) and burning zone thickness have been investigated not only in spark-ignition engine[12] but also in the constant volume vessels[13-15]. In this study, these results are compared in a very wide range of experimental conditions. Moreover, the burning zone thickness estimated from a phenomenological combustion model is also compared. The structure of the turbulent flame is also discussed using Damköhler number-Reynolds number relationship. The usefulness of constant volume vessel for combustion research in engine is examined through these discussions.

EXPERIMENTAL APPARATUS AND CONDITIONS

The turbulent burning velocity and the thickness of burning zone were determined in the single-event engine with swirl flow and also in the constant volume vessel of disk type with swirl flow.

Figure 1 shows a schematic diagram of the single-event engine[12]. The engine cylinder and the mixture tank are connected by a pipe. When the engine is driven by an electric motor, the fuel gas and air pass back and forth between the cylinder and the tank through the intake valve which is normally open in order to mix fuel gas and air thoroughly. After a homogeneous mixture is made, the intake valve is closed at a bottom dead center (BDC), after which the mixture in the cylinder is compressed and expanded repeatedly by the piston motion. The number of compression and expansion cycles after the valve closure is defined as N. During the intake stroke, a swirl flow is produced in the cylinder. There is no squish flow because the combustion chamber is of pancake type. The swirl flow decays with N. The mixture is ignited at the center of the combustion chamber by a pair of electrodes. The engine cylinder diameter and stroke are 78 and 85 mm respectively.

The compression ratio, ε, was 3.2. Propane was used as fuel and the equivalence ratio, ϕ, was 1.0. The engine speed was set at 600, 900, 1200 rpm while N was set at 1, 2 and 3. The measured turbulence intensity varied from 0.13 to 0.78 m/s as shown in Table 1.

Table 1 also shows the conditions measured in the constant volume vessel of disk type. As shown in Fig.2, the disk type of the combustion chamber is 125 mm in diameter and 35 mm deep[13,14]. The mixture is stored both in the combustion chamber (430 cm^3) and in the mixture tank (1920 cm^3) wherein initial pressures were 50 kPa and 300 (or 400) kPa respectively. These combustion chamber-mixture tank pressure conditions are therefore designated as 50-300 and 50-400. When the special valve is forced open during 200 ms, the homogeneous mixture flows into the combustion chamber in the direction tangential to the cylinder wall due to the pressure difference and shape of the special valve. The swirl flow decays with the lapse of time. The mixture is ignited at the center of the vessel. The flow was measured with LDA. Also here propane was used for the fuel and the equivalence ratios were set at 0.8, 1.0, 1.2 and 1.4. The turbulence intensity was up to 0.45 m/s.

Both experimental apparatus, namely, the single-event engine and the constant volume vessel have similar shapes with swirl flow at ignition time while both spark locations are also at the center of the combustion chambers. Figure 3 shows the experimental and the calculation conditions of the pressure and the unburned gas temperature in determining the burning velocity and burning zone thickness. As can be seen, experimental conditions vary widely.

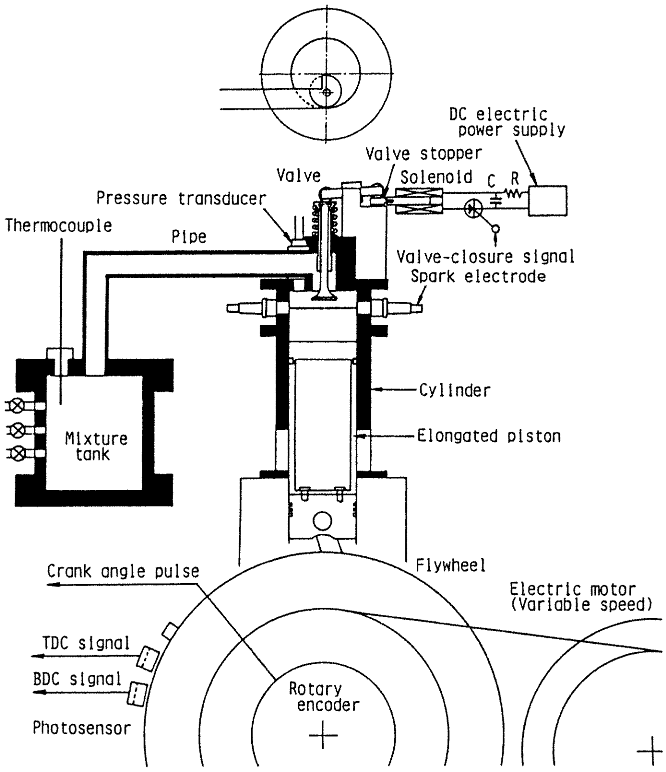

Fig.1 Schematic diagram of the single-event engine

Table 1 Experimental conditions

	SI Engine	Constant volume vessel of disk type
Dimension, mm	Bore =78 Stroke=85 (Compression ratio=3.2)	ϕ125 ×35
Mixture	C_3H_8 -air	
Spark location	Center of Combustion Chamber	
Equivalence ratio, ϕ	1.0	0.8, 1.0, 1.2, 1.4
Turbulence intensity, u', m/s	0.13~0.78	0 ~ 0.45
Unburned Gas Temperature, T_u, K	440 ~ 480	310~ 410
Pressure, P, MPa	0.46~0.61	0.26~0.74
Reference	12	13, 14

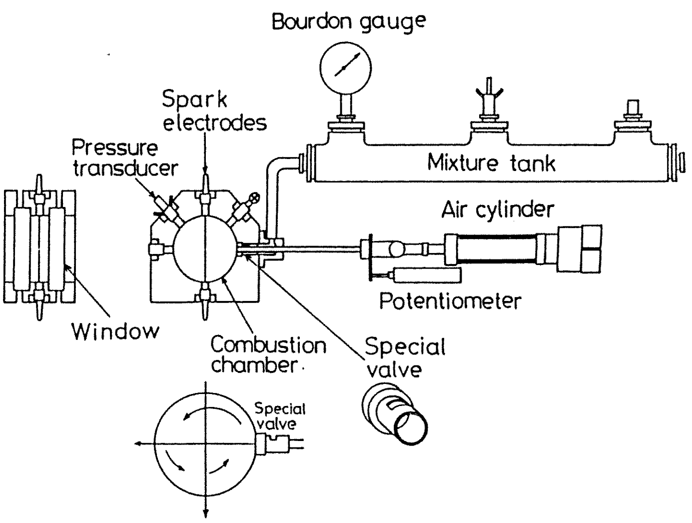

Fig.2 Schematic diagram of the constant volume vessel of disk type

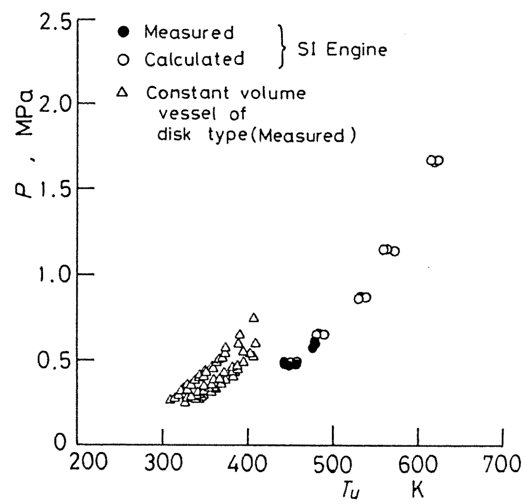

Fig.3 Unburned gas temperature and pressure

TURBULENT BURNING VELOCITY AND BURNING ZONE THICKNESS

Figure 4 shows an example of the ion current measured with a small ion-gap which was fixed in the single-event engine[12]. It has multiple peaks in the turbulent flame unlike in the laminar flame where it has only one peak. The flame front is generally defined as the boundary between burned and unburned gases or an "elementary combustion zone" where actual combustion occurs. Assuming that the appearance of one peak corresponds to the pass of the "elementary combustion zone" similar to the laminar flame, it is suggested that the burned and unburned gas clouds exist during the pass of the turbulent flame. Thus, it is considered that the turbulent flame has a certain width of the burning zone comprising burned and unburned gas clouds (see Fig.5).

Turbulent burning velocity, S_T, is determined by considering the mass balance of unburned gas in the turbulent flame as follows:

$$S_T=(\dot{M}_b/\rho_u+\dot{V}_{Bu})/A_T \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots(1)$$

where $\dot{M}_b$=rate of mass burned in cylinder, ρ_u= density of unburned gas, V_{Bu}= volume of unburned gas in the turbulent flame, A_T= average flame surface area. The flame arrival time was measured with a small ion-gap at many points in the combustion chamber. The average shape of turbulent flame was determined. Furthermore, by estimating the volume of laminar flame, V_L, of which the mass fraction burned is equal to the value for a turbulent flame, the value of V_{Bu} was obtained from the difference between V_T and V_L.

Assuming that the volume of burned gas in the burning zone of the turbulent flame is equal to the unburned one, the burning zone thickness, δ_T, is expressed as follows:

$$\delta_T = kV_{Bu}/A_e = k(V_T-V_L)/A_e \quad \cdots\cdots\cdots\cdots\cdots(2)$$

where k= constant(=2), A_e= surface area of an equivalent flame of which volume is V_L and V_T= volume of the turbulent flame enveloped with average surface area.

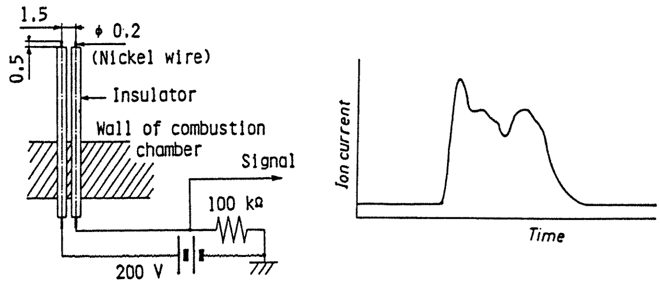

Fig.4 Ion-gap and an example of ion current

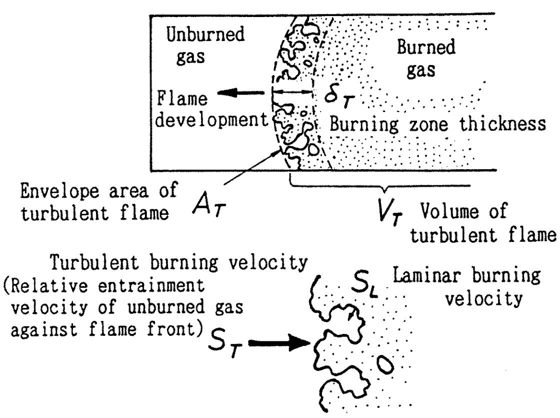

Fig.5 Schematic diagram of turbulent premixed flame in spark-ignition engine

Experimental Results

Both values of δ_T and S_T increased with increasing turbulence intensity[12-14]. The unburned gas entrained into the burning zone increases with turbulence intensity. However, the mean burning rate in the burning zone does not increase in proportion with the amount of the entrained unburned gas. Therefore, the burning zone thickness increases with turbulence intensity. The relationship between the burning zone thickness, δ_T, and the turbulent burning velocity, S_T, are shown in Fig.6 for the case with the test engine and the constant volume vessel of disk type. The burning zone thickness in lean (ϕ=0.8) or rich (ϕ=1.4) mixture is larger for the same value of S_T compared with other mixtures (ϕ=1.0, 1.2). Almost at all experimental conditions, the burning zone thickness in the test engine is smaller than those in the constant volume vessel. The laminar burning velocity in the engine is larger because of the higher temperature of the unburned gas. And therefore unburned gas in the burning zone burns faster. This is probably the reason why the burning zone thickness is smaller.

Figure 7 shows the correlation between δ_T and S_T/S_L. The triangles, △, in Fig.7 indicate the results obtained in the constant volume vessel with grid-stirred turbulence[15]. Methane was used as fuel and the equivalence ratio was varied from

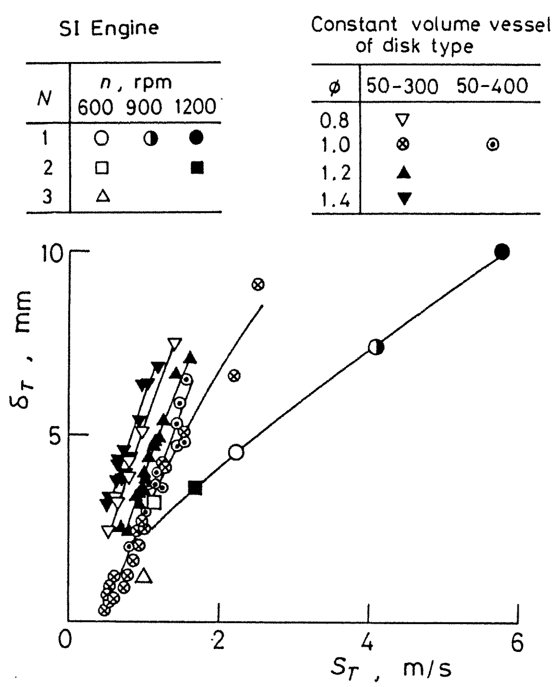

Fig.6 Relationship between burning zone thickness, δ_T, and turbulent burning velocity, S_T

0.8 to 1.3. The unburned gas temperature and pressure were near room temperature and atmospheric pressure respectively. The conditions in these experimental apparatus are very different, that is, wide ranges of turbulence intensity ($u' \leq 0.8$ m/s), unburned gas temperature (T_u = room temperature to 480 K), pressure (atmospheric pressure to 0.74MPa), and equivalence ratios; propane (ϕ= 0.8 ∿ 1.4), methane (ϕ= 0.8 ∿ 1.3). It is recognized from Fig.7 that there is a correlation between δ_T and S_T/S_L independent of the above experimental conditions.

Effect of Compression ratio on Burning Zone Thickness

The compression ratio in an actual engine is higher than 3.2 which is used in the single-event engine in the present study. The burning zone thickness for higher compression ratio is discussed by using the entrainment model for combustion[16]. This model is one of the phenomenological model which is proposed by Blizard and Keck[17] and extended by Tabaczynski et al.[18]. Although the basic concept of the model in the present study is based on their model, some parameters were modified. For example, the experimental results in the single-event engine were used for the turbulent burning velocity and the turbulence characteristics, that is, turbulence intensity and integral spatial scale. The calculated values of mass fraction burned have agreed well with the experimental data[16].

The burning zone thickness was calculated under various compression ratios (ε= 3.2, 4.0, 4.8, 6.0 and 8.0) and engine speeds (n=600, 900 and 1200 rpm) for the first cycle after the valve closure (N= 1). The calculated values of the burning zone thickness were larger than those obtained by measurement. Therefore, the estimated values of the burning zone thickness shown in Fig.8 are smaller than the calculated ones based on this combustion model by a factor of 0.8.

Figure 8 shows the relationship between δ_T and S_T/S_L estimated from the above calculation method. Under the condition of compression ratio, ε=3.2, the estimated values are in good agreement with the measured ones. In order to compare the calculated values with the measured ones, the calculated values shown in Fig.8 are chosen to be those in which mass fraction burned was about 5% as was the case in the experiment. Calculated values are compared with experimental data from the single-event engine under experimental conditions shown in Fig.8. It is seen that calculated and measured values are in good agreement as indicated by the solid line fitted to these points. On close examination of experimental data plotted in Fig.7 for all combustion chambers under the various conditions, this same solid line of Fig.8 also fits all these experimental data approximately. The relation estimated from the calculation agrees with that from the measurement for the range considered (see also Fig.7). It is worth noting that the experimental and calculation results of the burning zone thickness, δ_T, and S_T/S_L which may be represented by the solid lines in Figs.7 and 8, where determined over a wide range of unburned gas temperature (300 ∿ 620K) and pressure (0.1 ∿ 1.6 MPa) as shown in Fig.3. From the experimental results, the small values of δ_T can be determined even in the case of very weak turbulence wherein the structure of the flame is considered to be just like a weakly wrinkled laminar flame.

DISCUSSION ON COMBUSTION CONDITIONS

Figure 9 shows the relationship between Damköhler number and turbulence Reynolds number. The Damköhler number denotes the ratio of the eddy turnover time to the laminar flame residence time, the latter of which is characteristic reaction time at flame front. Turbulence Reynolds number is seen to be the ratio of the turbulent diffusivity to the laminar diffusivity or characteristic indicator of turbulence. The Damköhler number, Da, and turbulence Reynolds number, Re_L, are defined as follows:

$$Da = (L/u')/(\delta_L/S_L) \quad \cdots\cdots (3)$$

$$Re_L = u'L/\nu \quad \cdots\cdots (4)$$

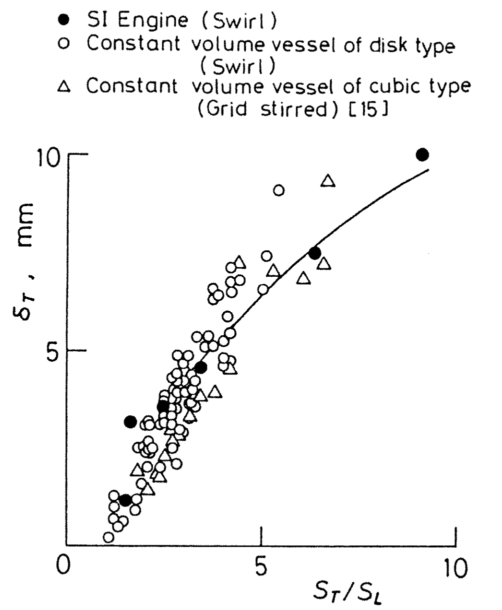

Fig.7 Relationship between δ_T and S_T/S_L

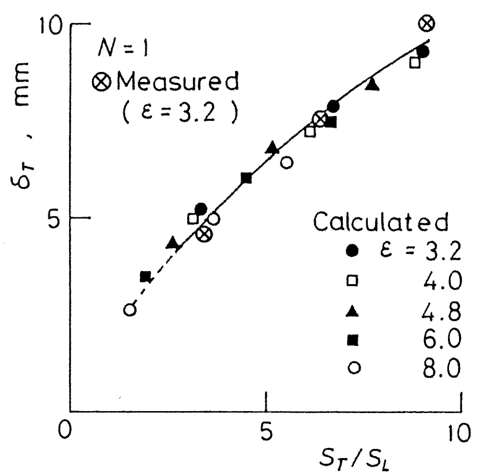

Fig.8 Relationship between δ_T and S_T/S_L (Higher compression ratio)

(Single-event engine, Experimental conditions; ε=3.2, N=1, n=600,900,1200 rpm $x \fallingdotseq 0.05$)

where L = integral spatial scale, u'= turbulence intensity, δ_L= characteristic thickness of preheat zone (presented as laminar flame thickness), ν = kinematic viscosity.

It is known that there are two regimes in the structure of premixed flames, namely, reaction sheet model and distributed reaction model regimes[19]. Kolmogorov length scale, η, is a measure of the size of eddies at which molecular viscosity becomes dominant. The reaction sheet model occurs when the Kolmogorov scale is much larger than the laminar flame thickness ($\eta/\delta_L \gg 1$) and the structure of the laminar flame is not affected by the turbulence so that the flame front is wrinkled as it goes over the turbulent eddies. On the other hand, the distributed reaction model occurs when the integral scale is much smaller than the laminar flame thickness ($L/\delta_L \ll 1$) and a laminar flame can not exist in the turbulent field. Both these model regimes are shown in Fig.9. The small eddies of unburned gas enter the flame front of "elementary combustion zone". However, the precise boundary has not been clarified yet and other regimes can be considered between the two. It is believed that even in the regime of reaction sheet model, the flame front is forced to become complex as the turbulence intensity increases. Then it can be imagined that unburned gas clouds are entrained into the burned gas and that they burn from their boundaries at the rate of laminar burning velocity.

The relationship between Damköhler number, Da, and turbulence Reynolds number, Re_L, is investigated in the single-event engine and the constant volume vessel of disk type. As shown in Fig.9, the experimental conditions in the engine are indicated by the double circle, ◎, and the black circle, ●. The double circle denotes the conditions of weak turbulence (n=600 rpm and N= 2 or 3) at ϕ=1.0. The black circle denotes the other experimental conditions of stronger turbulence at ϕ=1.0. Results of the present study indicate that the combustion conditions in the engine are similar to those in the constant volume vessel of disk type. The conditions in this study are within those indicated by Abraham et al.[10]. Therefore studies using constant volume vessels can be useful if the experimental conditions are well chosen.

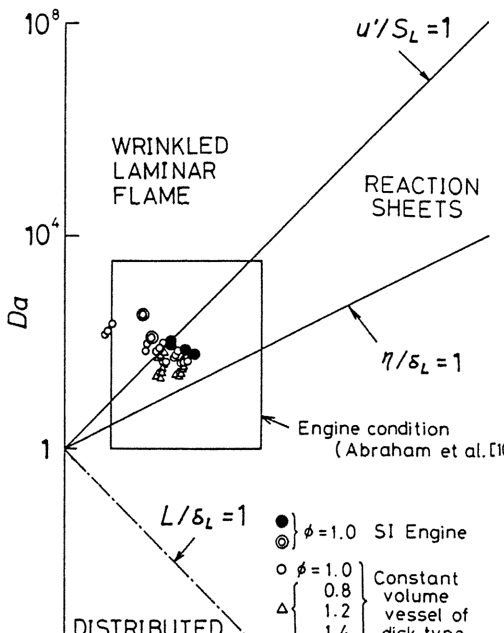

Fig.9 Regimes of the turbulent flames (Relationship between Damköhler number and turbulence Reynolds number)

Figure 10 shows an example of the relationship between Damköhler number and turbulence Reynolds number calculated by the entrainment model during the progress of combustion in the cylinder for the case with ε=3.2, n=900rpm and N=1. The mass fraction burned in the combustion chamber is denoted by x. Both values of Da and Re_L increase with x and the combustion regime moves in the direction of the reaction sheet model.

The effects of compression ratio and engine speed were also investigated. The relationship of Da and Re_L was calculated under several conditions and the results are shown in Fig.11. In these

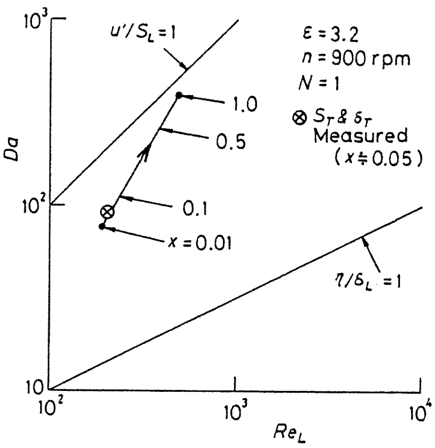

Fig.10 Relationship between Damköhler number and turbulence Reynolds number during combustion

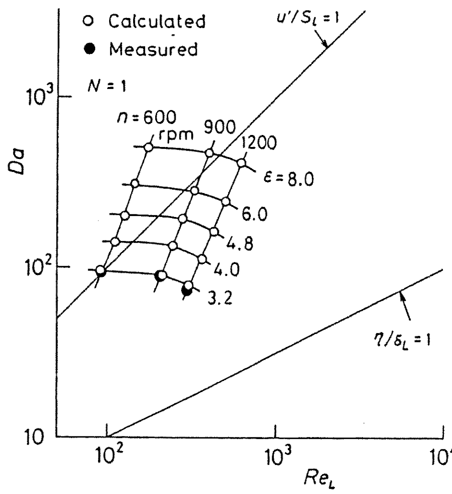

Fig.11 Relationship between Damköhler number and turbulence Reynolds number

calculations, the value of compression ratio was varied from 3.2 to 8.0, and the value of engine speed from 600 to 1200 rpm. As the compression ratio increases, the values of Da and Re_L increase and the combustion regime moves in the direction of reaction sheet model. As the engine speed increases, the value of Da decreases and the value of Re_L increases so that the combustion regime moves in the direction of distributed reaction model though the regime is still in the reaction sheet model.

CONCLUSIONS

The burning zone thickness and the turbulent burning velocity under the conditions of the single-event engine and the constant volume vessel have been discussed. Main results are as follows:

(1) The burning zone thickness, δ_T, increases with increasing turbulence intensity. There seem to exist a good relationship between δ_T and S_T/S_L under various conditions, that is, fuel (propane or methane), equivalence ratio (0.8 ∿ 1.4), pressure (0.1 MPa ∿ 1.6 MPa) and unburned gas temperature (300 ∿ 620 K).

(2) The burning zone thickness estimated under the conditions of compression ratio, ε =3.2 ∿ 8.0 by using the entrainment model for combustion show a good correlation with experimental values.

(3) Combustion research by using constant volume vessels is useful for contributing to the understanding of the combustion in actual engine when Damköhler number-turbulence Reynolds number conditions in the constant volume vessel are similar to those in the engine.

(4) When compression ratio increases, the turbulent flame structure moves in the direction of reaction sheet model. On the other hand, it moves in the direction of distributed reaction model as engine speed increases.

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