A Study on Combustion Characteristics of Superadiabatic Combustor in Porous Media

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Because of the energy resource exhaustion, the aggravating environmental air pollution, the smoke phenomena and so on, the recent trends and targets in designing combustor are reduction of pollutant emissions and improvement of combustor efficiency. Therefore many combustion methods and emission control technologies have been proposed by many researchers through numerical and experimental analyses. One of the most available and effective combustion methods is the excess enthalpy combustion, so called, the superadiabatic combustion. In this study, the superadiabatic combustion with the reciprocating flow in a porous media has been investigated with the variation of equivalence ratio, flow velocity and reciprocating cycle time. In this system, the flow direction is reversed regularly by the solenoid valves. The results of this study show that the maximum gas temperature is remarkably higher than the theoretical adiabatic flame temperature and the emission characteristic is very excellent. The analyses reveal several attractive characteristics of the flame and the proposed idea is promising to burn mixtures of low heat content in a reciprocating type combustor. This combustor can be applied to the elimination of unburned compound, with more intensive and continuous study.

Key Words: Excess Enthalpy Combustion, Superadiabatic Combustion, Porous Medium, Adiabatic Flame Temperature, Ultra-lean Combustion

1. Introduction

Recently, due to the increased public awareness of pollutant emissions including VOC (Volatile Organic Compound) from combustion systems, it is much more demanding to incinerate the pollutants exhausted from various industries and the lean mixtures from mines and so on. These gases are usually diluted by the large amount of air and inert gases, therefore the large amount of external energy must be supplied to burn these low heat content gases when the conventional combustion methods are adopted. Therefore, many combustion methods and emission control technologies have been proposed and are being studied by many researchers around the world.

The most effective and convenient combustion method is the excess enthalpy combustion, initially proposed by Weinberg (1971). The principle of the excess enthalpy combustion is to recirculate the enthalpy of the hot combustion products to the cold reactants and the large excess enthalpy is produced at the head of the reaction zone. This method performed by Weinberg was to preheat the unburned mixture by the hot combustion products through a spiral quartz tube heat exchanger. In a series of analytical and experimental studies, Weinberg et al. (1974, 1986) have shown that the burning system has been successful to extend the ranges of flame stability and flammable limits. However, this heat recirculation method used some sort of extra heat exchangers by means of which the unburned mixture was preheated before it entered into the flame. This method was indirect in the sense that the heat was

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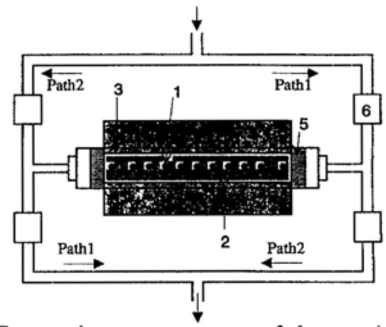
recirculated outside the flame and the heat recirculation was not efficient.

On the principle of the excess enthalpy combustion, many subsequent researches have revealed the potential of this combustion method to achieve the high combustion efficiency and the low pollutant emissions in the practical burner applications. Sato et al. (1979) proposed a simple method of producing the excess enthalpy flame by changing the internal structure of flame itself. The idea was just to insert a porous material of high thermal conductivity into the one-dimensional flame zone to recirculate the heat internally through the solid from the downstream high temperature region to the upstream low temperature region. The subsequent theoretical and experimental analyses revealed several attractive characteristics of the system, suggesting that the idea was a promising one for burning mixtures of low heat content in a simple combustion system (Hase et al., 1981 ; Kotanc and Takeno, 1982 ; Buckmoster and Takeno, 1983 ; Benbahani etal., 1984). Echigo et al. (1984, 1988) revealed the thermal structures in porous media with internal heat generation by analytical and experimental studies. It was shown that the remarkable energy recirculation was achieved from the burned gas to the preheat of unburned mixture by thermal radiation propagation against the flow direction. The porous medium of an appropriate optical thickness was very effective for energy conversion from the flowing gas enthalpy to the thermal radiation. They also revealed that the radiative absorptivity and the optical thickness of porous medium had a great effect on the thermal structures. In the present study, an experimental investigation on the superadiabatic combustion in the reciprocating flow type combustor with porous media has been performed over a wide ranges of parameters.

2. Experimental Apparatus and Methods

2.1 Experimental apparatus

The schematic diagram of the reciprocating flow type combustor and the specifications of combustor are shown in Fig. 1 and Table 1, respectively. The honeycombs were set up at the edges of the combustor for stability of the propane-air mixtures. The solenoid valves were installed at the inlet and outlet to change the flow direction regularly, equipped with a timer. The combustor is filled with the spherical type porous media which have high heat-conductivity and high heat-resistance for effective superadiabatic combustion. The physical properties of the porous media are shown in Table 2. For the measurement of temperature distributions in a combustor, eleven thermocouples were installed along the axial direction. The schematic diagrams of the thermocouples are shown in Fig. 2. The Pt



1 : Thermocouple 2 : Porous media 3 : Inner ceramic blanket
4 : Outer ceramic blanket 5 : Honeycomb 6 : Solenoid valve

Fig. 1 Schematic diagram of combustor.

Table 1 The compostion of superadiabatic reciprocation type combustor.

Material	Stainless steel
Structure	cylinder
length [mm]	384
diameter [mm]	61
thickness	6

Table 2 The physical value of porous media.

Material	Al_2O_3
Diameter [mm]	3.25
Porosity [%]	34
Boiling poink [K]	2305
Thermal conductivity [kcal/mh ℃]	31

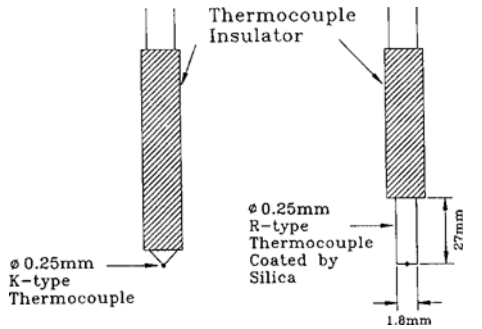


Fig. 2 Schematic thermocouple for temperature measurement of combustor

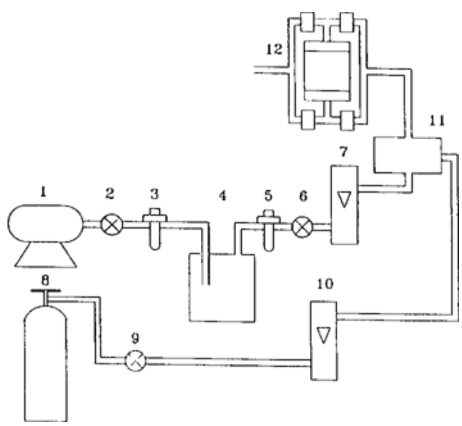


Fig. 3 Schematic diagram of experimental devices.

-Pt/13%Rh PR thermocouples (ϕ0.25mm) were installed at the middle part of the combustor with the equal distance of 12mm for the measurement of the high temperature gas. Seven thermocouples where located every 12mm in near by center of the combustor and others were located every 48mm. The chromel-alumel CA thermocouples (ϕ0. 25mm) were installed at the edges of the combustor with the equal distance of 48mm to check the reliability of the experiment.

Figure 3 shows the schematic diagram of experimental devices which mainly consist of air supply devices, fuel supply devices and the combustor. After the air from a air-compressor is regulated through the pressure regulator, the moisture and oil are eliminated by a air filter. The air then flows into the flow meter (TCP NP series purge meter, 2.5～25ℓ/min) after the moisture, oil and the pressure fluctuations are completely eliminated through the second air filter and a pressure regulator. The propane from the fuel tank passes through the pressure regulator and flows into the flow meter (KOFOLC NO. 9, 50～500 cc/min). Finally, the fuel and the air enter into the combustor after they are mixed in the pre-chamber by swirler.

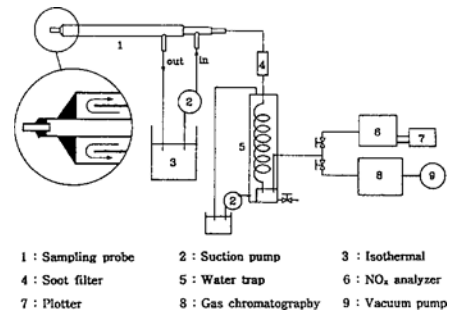


Fig. 4 Schematic diagram of gas sampling system.

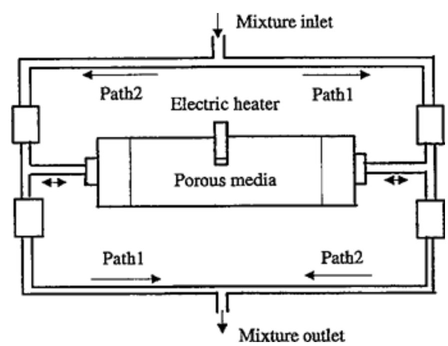


Fig. 5 Schematic diagram of flow control system.

The schematic diagram of the gas sampling system is shown in Fig. 4. The moisture in the exhaust gas is eliminated through the water trap after the soot is eliminated by a soot filter. Finally, the exhaust gases enter into the gas -chromatography (shimadzu, GC-7A) and the chemiluminescent NOx analyzer (model NA 510 -2, SA700).

2.2 Experimental methods

The schematic diagram of the flow control system is shown in Fig. 5. The unburned mixture which passes through the path 1 is ignited at the middle of the combustor by a electric heater. The exhaust gas is emitted to the outside through the path 1, in the meantime the porous media are preheated by the hot burned gas and release the radiative heat. After the flow direction is reversed by the solenoid valve, the unburned mixture passes through the path 2 and is preheated by the radiative heat emitted from the porous media. These procedures are regularly repeated with the change of flow direction by the solenoid valves.

The starting characteristics of the combustor are presented in Fig. 6. 11 thermocouples were installed along the axial direction and the temperature distributions at the inlet and outlet were measured. The electric heater was used as an igniter for stable flame in the combustor. In the first ignition, the conditions were settled as equivalence ratio 0.64 and mixture flow velocity 34 cm/s for the consistency of the experiments. After the flame becomes stable, the temperature at the edges becomes constant and it takes about 25 minutes. Therefore, all the experiments were implemented after 25 minutes from the ignition for consistency of the experiments.

3. Results and Discussion

3.1 Typical temperature distribution

The typical gas temperature distribution along the axial direction is shown in Fig. 7 at the conditions of equivalence ratio 0.26, flow reciprocating cycle time 45 seconds, mixture flow velocity 34 cm/sec. The highest temperature is obtained at the middle part of the combustor where the chemical reaction occurs and the temperature distribution is symmetric. The temperature at the edges of the combustor is almost room temperature due to the heat release from the hot burned gas to the porous media. The gas temperature at the middle part of combustor is larger than the theoretical adiabatic flame temperature (dotted line). It shows the superadiabatic combustion effect that appears due to the preheating of the reactant mixture. It means air-fuel mixture get the energy by the heated high conductivity porous medium.

3.2 Gas temperature distributions

The gas temperature distributions along the axial direction with the variation of equivalence ratio are shown in Fig. 8 at the conditions of flow reciprocating cycle time 45 seconds, mixture flow

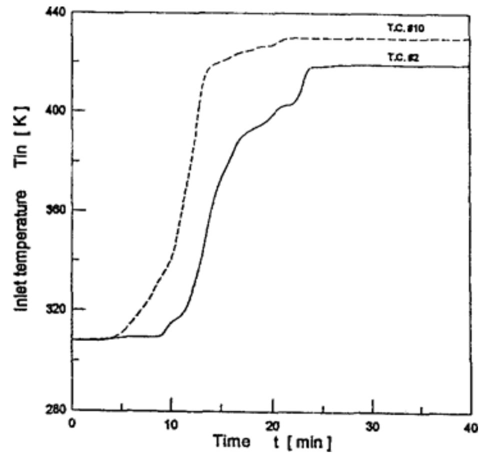


Fig. 6 Starting characteristics of combustion system.

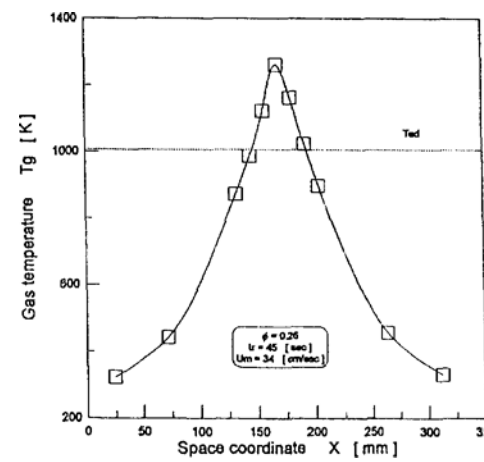


Fig. 7 Typical gas temperature distribution along space coordinate.

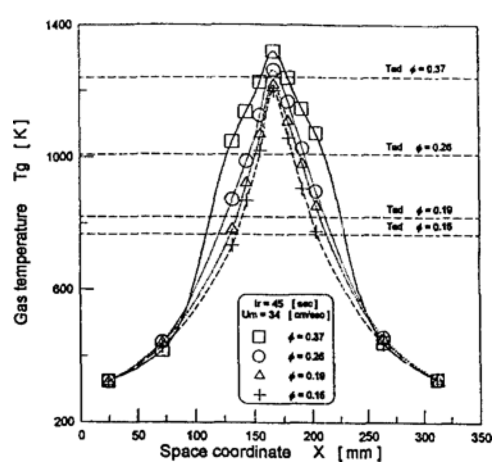


Fig. 8 Gas temperature distributions with various equivalence ratio along space coordinate.

velocity 34 cm/sec. At every equivalence ratio 0.15, 0.19, 0.26 and 0.37, the maximum gas temperature was higher than the theoretical adiabatic flame temperature. Also, the high temperature regions were extended with the increase of the equivalence ratio.

Figure 9 presents the gas temperature distributions along the axial direction with the variation of mixture flow velocity at the conditions of equivalence ratio 0.26, flow reciprocating cycle time 45 seconds. The length of the flame reduces as the flame front moves to the center of combustor. Therefore, the maximum gas temperature

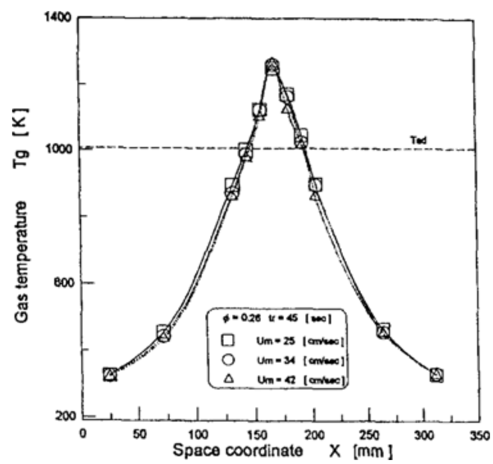


Fig. 9 Gas temperature distributions with various mixture flow velocity along space coordinate.

increased and the high temperature regions became narrow with the increase of the mixture flow velocity. However, the effect of mixture flow velocity was mimid, because it was porous media had some volume enough to prevent the flow velocity.

The gas temperature distributions along the axial direction with the variation of flow reciprocating cycle time are presented in Fig. 10 at the conditions of equivalence ratio 0.26, mixture flow velocity 34 cm/sec. The maximum gas temperature increased and the outlet temperature decreased slightly, with the diminution of flow reciprocating cycle time. The reduction of flow reciprocating cycle time resulted in the minimization of the heat loss, that is, the increased heat transfer to the porous media. Therefore, it is important to change the flow direction before the outlet temperature becomes relatively high.

3.3 Superadiabatic combustion characteristics

Figure 11 illustrates the characteristics of the superadiabatic combustion with the temperature difference between the maximum gas temperature and the theoretical adiabatic flame temperature. The experiments were implemented with the variation of equivalence ratio, flow reciprocating cycle time and the mixture flow velocity was settled as 34 cm/sec. The effect of the super-

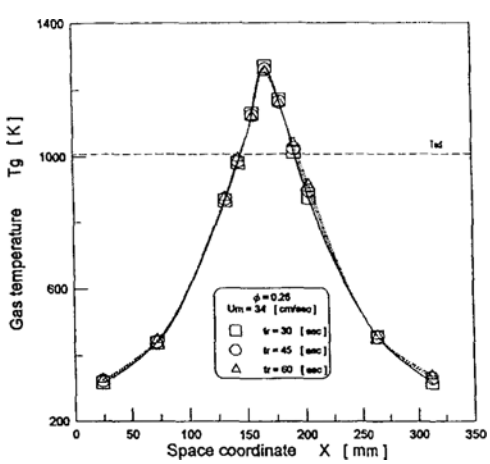


Fig. 10 Gas temperature distributions with various flow reciprocating cycle time along space coordinate.

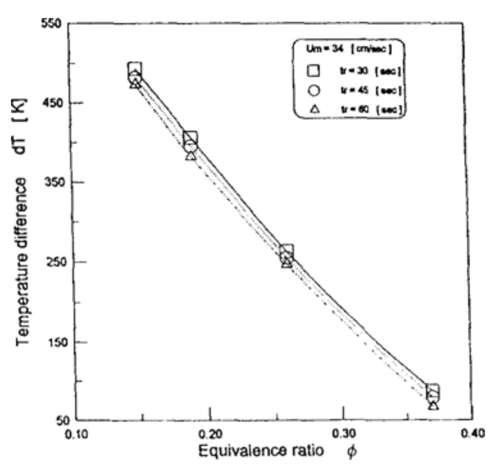


Fig. 11 Effect of superadiabatic combustion characteristics along equivalence ratio with various flow reciprocating cycle time.

adiabatic combustion was more obvious at relatively low equivalence ratio than that of at high equivalence ratio. From the above results, it can be inferred the possibility of the ultra-lean combustion.

The characteristics of the superadiabatic combustion with the variation of flow reciprocating cycle time and mixture flow velocity are presented in Fig. 12 at the condition of equivalence ratio 0.26. The superadiabatic combustion effect was slight with the reduction of flow reciprocating cycle time, that is, the decrease of the radiation heat loss from the combustor to atmosphere.

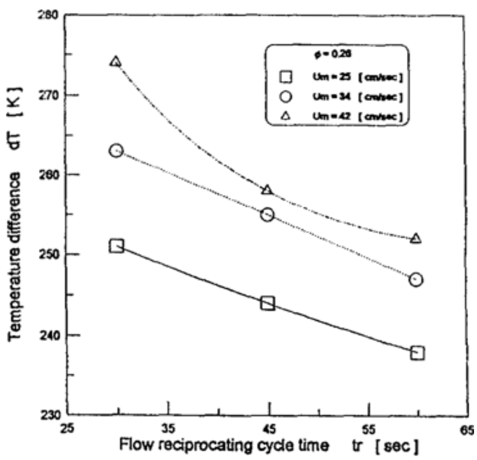


Fig. 12 Effect of superadiabatic combustion characteristics along flow reciprocaing cycle time with various mixture flow velocity.

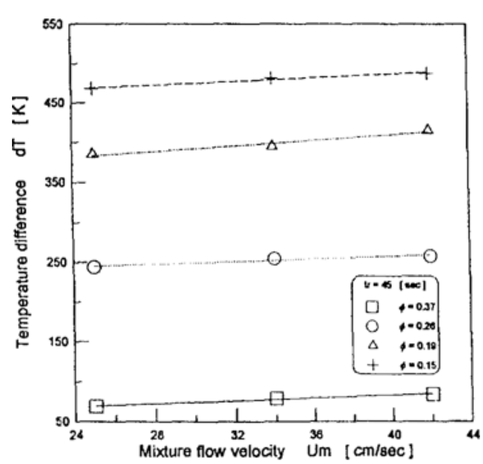


Fig. 13 Effect of superadiabatic combustion characteristics along mixture flow velocity with varlus equivalence ratio.

The superadiabatic combustion characteristics with the variation of mixture flow velocity and equivalence ratio are shown in Fig. 13 at the constant flow reciprocating cycle time 45 seconds. As the mixture flow velocity increases, the temperature difference between the maximum gas temperature and the theoretical adiabatic flame temperature increase slightly. The superadiabatic combustion effect appeared through the all ranges of mixture flow velocity in this experiment and it was more evident at low equivalence ratio than at relatively high equivalence ratio. It shows the possibility that this reciprocating type combustor can be applied to the energy and pollution related industries.

3.4 Exhaust characteristics

The NO concentration distributions with the variation of mixture flow velocity and equivalence ratio are shown in Fig. 14 at the constant flow reciprocating cycle time 45 seconds. As the equivalence ratio increases, the NO concentration increases due to the augmentation of the flame temperature. Although the flame temperature was high, the NO concentration was lower than 3.5 ppm through all the ranges.

Figure 15 shows the NO concentration distributions with the variation of flow reciprocating cycle time and equivalence ratio at the constant mixture flow velocity 34 cm/sec. The NO concen-

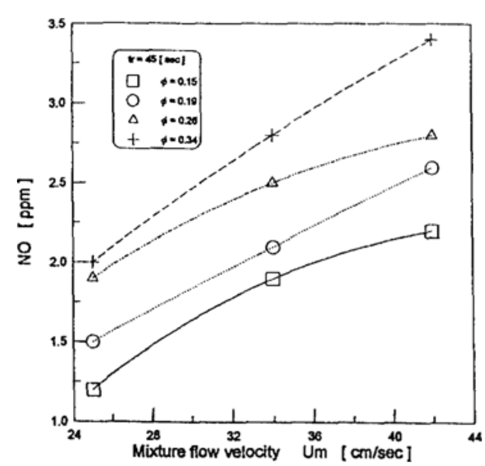


Fig. 14 NO concentration along mixture flow velocity with various equivalence ratio.

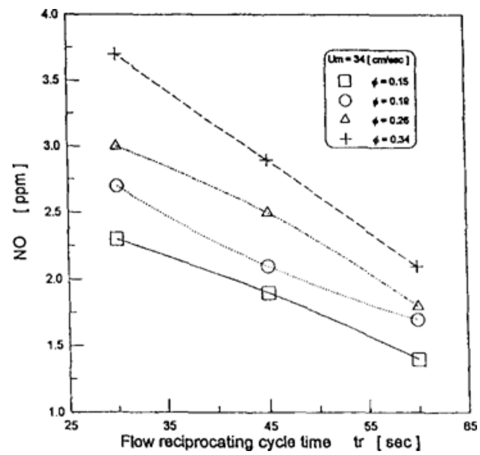


Fig. 15 NO concentration along flow reciprocting cycle time with various equvialence ratio.

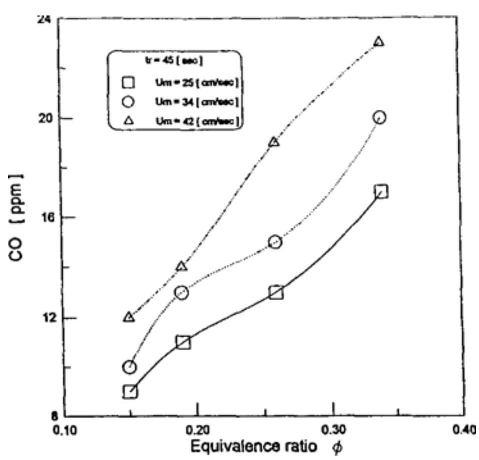


Fig. 16 CO concetration along equivalence ratio with various mixture flow velocity.

tration increased with the equivalence ratio and with the decrease of flow reciprocating cycle time. Although the NO concentration increased with the flame temperature, the overall exhaust characteristics were excellent.

Figure 16 shows the CO concentration distributions with the variation of mixture flow velocity and equivalence ratio at the constant flow reciprocating cycle time 45 seconds. The overall CO concentration was lower than 24 ppm and finely risen with the increase of equivalence ratio and mixture flow velocity, due to the increase of flame temperature.

The distributions of CO_2, O_2 and unburned

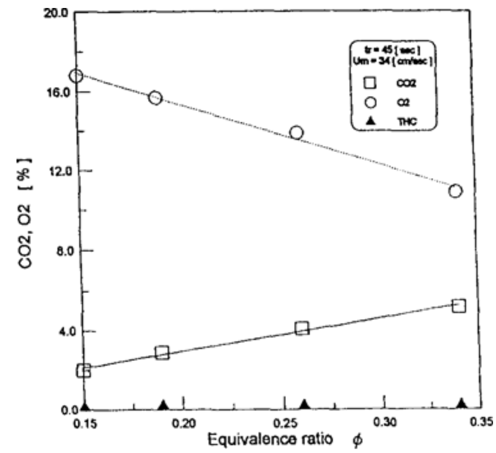


Fig. 17 Concentraion of the total hydrocabon and the other stable species along equivalence ratio.

hydrocarbon concentrations are shown in Fig. 17. The experiments were implemented with the variation of equivalence ratio and the flow reciprocating cycle time and the mixture flow velocity were settled as 45 seconds and 34 cm/sec, respectively. The overall concentrations of unburned hydrocarbon were lower than 1 ppm through all the ranges. The complete combustion through the porous media was observed from the concentrations of CO and unburned hydrocarbons.

4. Conclusions

In this study, the characteristics of superadiabatic combustion were discussed through the analysis of the flame structure in the reciprocating flow type combustor. The possibility of the ultra-lean combustion was also suggested. The flame structure, exhaust characteristics, etc. in the reciprocating flow type combustor were investigated with the variation of mixture flow velocity, equivalence ratio and flow reciprocating cycle time. The main results are as follows ;

(1) The superadiabatic combustion was observed, that is, the maximum gas temperature was higher than the theoretical adiabatic flame temperature.

(2) The superadiabatic combustion effect became obvious as the equivalence ratio and the mixture flow velocity increased and the flow

reciprocating cycle time decreased.

(3) In this study, the flammable limit of propane was extended to 0.15 by the preheat of unburned mixture through the internal heat recirculation of the reciprocating type combustor filled with porous media.

(4) The measured NO concentration was lower than 3.5 ppm through all the ranges and mainly affected on the flame temperature which depended on the equivalence ratio, the mixture flow velocity and the flow reciprocating cycle time. The concentrations of CO and unburned hydrocarbons were lower than 24 ppm and 1 ppm through all the ranges, respectively.

(5) The ultra-lean combustion is possible with the reciprocating flow type combustor which is used in this study, therefore this combustor can be applied to the energy and pollution related industries, that is, the elimination of lean and harmful pollutions, the reuse of exhaust energy, and so on.

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