Ignition and Exhaust Emission Characteristics of Spray Combustion in a Pre-Chamber Type Vortex Combustor

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ABSTRACT

The lean ignition limit, the lean blowout limit and the exhaust emission characteristics of spray combustion have been investigated experimentally using a pre-chamber type vortex combustor developed for a 300KW large-bus gas turbine engine. It has been verified that these depend on the spray characteristics of the fuel injector and the air flow pattern or the distribution of air in the chamber.

Ignition succeeds through three processes. The first step is the formation of a flame kernel near the sparking ignitor, the second step is the propagation of the flame kernel into a flame holding region, and the last step is the formation of a rotating flame in that region. The lean blowout limit of the rotating flame depends on the air flow pattern in the pre-chamber when the air temperature in the combustor inlet is under 470K, while a constant fuel-air ratio of less than 0.001 is maintained at 470K and above. With no or a little secondary air, the NOx emission index does not increase in proportion to the fuel-air ratio, because both the gas temperature and residence time decrease due to the radiative heat loss caused by soot formation and reduction of a recirculation region in the main-chamber.

These phenomena were evaluated with 3 dimensional numerical simulations taking account of spray combustion, soot formation, the extended-Zeldovich thermal NO formation and radiative heat loss.

NOMENCLATURES

G = flow rate
H = area of air path into the combustion chamber
h = combustor inlet temperature dependent coefficient
\( \phi \) = mole fraction of exhaust gas
K = absorption coefficient
\( K_c \) = gas thermal conductivity
k = turbulent kinetic energy
L = latent heat of vaporization
l = equivalent beam length of cell
M = mole weight
m = mass
N = number density of soot particle
\( N_u \) = Nusselt number
n = number density of soot radical nucleus
P = pressure
Q = lower heating value
R = gas constant
r = stoichiometric oxygen requirement to burn unit mass of fuel or soot
Sh = Sherwood number
\( S_D \) = source rate of \( \phi \)
SMD = Sauter Mean Diameter
T = temperature
U = representative mean air velocity
\( u, v, w \) = velocity
V = combustion chamber volume
x = position
Y = mass fraction
z = combustion pressure dependent index
\( \Gamma_D \) = effective diffusion coefficient of \( \phi \)
\( \epsilon \) = dissipation rate of turbulent kinetic energy
\( \epsilon_r \) = emissivity
\( \rho \) = density
W = surface oxidation rate of soot particle
\( \theta \) = spray cone angle

SUBSCRIPTS

a = air
b = boiling point
c = combustion
cr = critical point
d = droplet

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INTRODUCTION

A regenerative gas turbine engine for vehicle application has been developed, which is advantageous for its large power-weight ratio, low vibration and noise, and its inherent ability for multifuel operation. The most noteworthy of all is, however, that the specific fuel consumption will exceed that of a diesel engine due to high turbine inlet temperature of the regenerative gas turbine cycle by the use of a heat resisting ceramic material. On the other hand, various investigating subjects on the gas turbine combustors must be resolved for practical use.

1. Rapid response to frequent acceleration and deceleration.
2. Satisfaction of exhaust emission standards. For example, the long-term NOx standard for heavy duty vehicles over 2.5 tons in Japan is below 4.5 g/kwh. It would be necessary for a combustor to achieve a mean NOx emission index of less than about 10 g/kg-fuel, assuming that the thermal efficiency of a gas turbine engine is 20%.

In regard to the abatement of exhaust emission, various prevapor-premixed-lean-combustors[1] and catalytic combustors[2] have been reported. These combustors need to install a variable geometry in order to stabilize the combustion. However, it is difficult to realize adequate endurance and reliability, considering the required specifications for vehicle engines. Therefore, the development of a simpler combustor is desirable for gas turbine combustors.

We have been attempting to apply a pre-chamber type vortex combustor[3] to a 300KW large-bus gas turbine engine[4] shown in Fig. 1. Spray combustion using this combustor has some advantages in ignitability and flammability required for a vehicle engine, although it is also considered to have disadvantages in the abatement limit of NOx emission index[5].

In this paper, the characteristics of spray combustion are investigated, carrying out a series of combustor rig tests and numerical simulations. First, ignition and flame stability are evaluated under non-heated air conditions. Then, the possibility of abatement for exhaust emission index is studied to make spray combustion lean in fuel-air ratio under heated air conditions.

APPARATUS

Combustion Test Rig

Figure 2 shows the schematic diagram of the combustor test rig. The air temperature in the combustor inlet is heated to over 800K by an electric heater. The air pressure is supplied to over 0.5 MPa.

Combustor

Figure 3 shows the pre-chamber type vortex combustor[6] using a GT31 engine whose specification is listed in Table 1. Combustion air is supplied from three parts; pre-chamber (primary), main-chamber (secondary) and downstream of main-chamber (dilution). The size and area of the air path are shown in Table 2.
The combustion intensity amounts to about \(4 \times 10^7\) kcal/m\(^3\)-h-at at a maximum acceleration load operation. Three types of combustors were used for comparison of combustion characteristics; COMBUSTOR-I has no secondary air path in the main-chamber, COMBUSTOR-II, -III and -III2 have many circular holes on the main-chamber wall, and COMBUSTOR-III has two pairs of opposed circular pipes on the main-chamber wall cooled by air.

The charged energy of the ignition unit is about 4 joules. A surface discharged electric ignitor is operated at 10 Hz and not less than 18 KV.

### Table 2: Size of pre-chamber type vortex combustors

<table>
<thead>
<tr>
<th>Diameter</th>
<th>Volume</th>
<th>Area of Air Path</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Combustor I</td>
</tr>
<tr>
<td>Pre-Chamber</td>
<td>2d/3</td>
<td>0.2V</td>
</tr>
<tr>
<td>Main-Chamber</td>
<td>d</td>
<td>0.6V</td>
</tr>
<tr>
<td>Dilution</td>
<td>2d/3</td>
<td>0.2V</td>
</tr>
<tr>
<td>Total</td>
<td>—</td>
<td>1.0V</td>
</tr>
</tbody>
</table>

Fuel Injector

Figure 4 shows two types of fuel injectors used in this study. INJECTOR-A and INJECTOR-A1 are a duplex swirl type injector and INJECTOR-B is a six-hole-plain-jet airblast type injector. For fuel, JIS No.2 gas oil is used, that is, H/C = 1.9 and \(\Phi = 10240\) kcal/kg. As shown in Fig. 5, the spray cone angle of INJECTOR-A is larger than that of INJECTOR-A1. The minimum SMD of INJECTOR-A and -A1 is 70±5 \(\mu\)m and 90±5 \(\mu\)m, respectively. The spray cone angle and SMD of INJECTOR-B is 70±5 degrees and less than 50±5 \(\mu\)m when the flow rate and the pressure ratio of the atomizing air is set to 2±0.1 g/s and 1.2, respectively.
DESCRIPTION OF NUMERICAL SIMULATION

A multi-dimensional simulation code[7] has been developed at TOYOTA Central R&D Labs., Inc. This code "FIRE3D-GT" is based on treatments of gas phase, fuel spray, combustion, NOx formation, soot formation and radiation[8].

Gas Phase

The gas flow is treated as a three-dimensional, unsteady and compressible viscous flow. The law-of-the-wall and standard k-E turbulence models are used. The governing equations are expressed as follows: the transport equations (1), the equation of state (2), and the conservation equation of species (3). \( \phi \) represents one of the variables, \( Y_i \), \( u \), \( v \), \( w \), \( e \), \( k \) and \( C \).

\[
\frac{\partial (\rho \phi)}{\partial t} + \nabla \cdot (\rho \phi \mathbf{v}) - \nabla \cdot (\mathbf{v} \phi) = S_{\phi} + S_{d\phi}
\]  

(1)

\[
P = \rho R T
\]  

(2)

\[
\sum Y_i = 1
\]  

(3)

Fuel Spray

The behavior of fuel spray is treated by the Discrete Droplet Model (DDM)[9]. In the model, since it is difficult to calculate the motions of all droplets, droplets with a similar motion are combined into one droplet group, and one droplet motion in it is calculated by the following equations.

\[
\frac{dm_d}{dt} = \rho_d \mathbf{v}_d \cdot \mathbf{u}_d + \mathbf{u}_d \cdot \nabla \mathbf{v}_d
\]  

(4)

\[
\frac{d(nu_u)}{dt} = -\frac{1}{\beta} \frac{d}{dt} \rho_d (\mathbf{v}_d - \mathbf{u}_d) \cdot (\mathbf{u} - \mathbf{u}_d)
\]  

(5)

\[
\frac{dm_d}{dt} = -\frac{dD}{dP} \mathbf{P}_f \mathbf{v}_m \left( \frac{n_{\text{fuel}}}{n_{\text{air}}} \right) \mathbf{I}\left( \mathbf{P}_f - \mathbf{P}_m \right)
\]  

(6)

Combustion

The combustion of fuel vapor is treated as a one-step reaction. The reaction rate is given by a combination of two equations representing a chemical reaction and a mixing of fuel and oxygen, as shown below.

\[
\frac{dp_f}{dt} = -A_0 n_{\text{fuel}} n_{\text{oxygen}} \exp \left( -\frac{E_c}{RT} \right)
\]  

(7)

\[
\frac{d\left( \frac{m N}{T} \right)}{dt} = \frac{C_{\text{eff}} c^p_s c^p_o \mathbf{v} \exp \left( \frac{E_a}{RT} \right)}{4} + \frac{\sum_{\text{soot concentration}}}{\text{soot concentration}}
\]  

(8)

NOx Formation

Most of NOx formed in the combustor is considered to be thermal-NO. The formation rate has been often calculated by empirical equations.

Since the operating conditions of practical combustors widely change, it is inadequate to apply the equations to their NOx formation.

In this study, the chemical equilibrium composition of ten species, \( O_2 \), \( N_2 \), \( CO_2 \), \( H_2O \), \( H \), \( H_2 \), \( O \), \( N \), \( OH \) and \( CO \), is calculated in each cell, and the NO formation is treated as the extended-Zeldovich mechanism expressed as the following reactions.

\[
\begin{align*}
N_2 + O & \rightarrow NO + N \\
O_2 + N & \rightarrow NO + O \\
N + OH & \rightarrow NO + H
\end{align*}
\]  

(9)

Soot and luminous flame is formed in practical diffusion combustors. The radiative heat loss decreases the burnt gas temperature. Since thermal-NO formation is strongly dependent on gas temperature, it is necessary to estimate the soot concentration and the radiative heat loss from soot.

Soot Formation [10]

The soot formation is treated as a two-step process of radical nuclei formation and subsequent soot particle formation, based on the Tesner model[11]. Their formation rates are given below:

\[
\frac{dn}{dt} = n + (T - G)n - G
\]  

(10)

\[
\frac{dN}{dt} = (A_0 - B_0)n
\]  

(11)

Coefficients \( G \) and \( B \) which are constant in the original Tesner model are modified according to the density change of the gas. The following equation by Farmer [12] is also used to supplement equation (12) in its temperature deficiency.

\[
\frac{d\left( \frac{m N}{T} \right)}{dt} = \frac{C_{\text{eff}} c^p_s c^p_o \mathbf{v} \exp \left( \frac{E_a}{RT} \right)}{4} + \frac{\sum_{\text{soot concentration}}}{\text{soot concentration}}
\]  

(12)

The equation giving a smaller formation rate is selected for each cell.

The soot oxidation rate is calculated similarly to the combustion rate of fuel vapor. The following equation is used, which is a combination of surface reaction[13] and mixing rate[14].

\[
\frac{d\left( \frac{m N}{T} \right)}{dt} = -\min \left| \frac{\mathbf{v}_d S_{\text{fuel}}}{2} + \mathbf{v}_d S_{\text{oxygen}} \left( \min \left| 1, \frac{P_{\text{fuel}}}{P_{\text{air}}} \right| \right) \right|
\]  

(13)

Radiation

The heat loss of each cell is treated by a 4-flux model. The temperature, species and soot concentration distributions are interpolated to orthogonal data and only two couples of heat fluxes in the radius direction are calculated. The emissivity of the burnt gas is expressed as follows[15][16].

\[
\epsilon_r = \sum_{\text{soot concentration}} \left( 1 - \exp (K_{\text{soot}}) \right)
\]  

(14)

A and \( K \) are calculated from the temperature and soot concentration.
IGNITION LIMIT

Ignition limits were measured by the following test procedures.

1. The combustor inlet temperature and pressure are kept to 290K and 0.1 MPa, respectively. Ignition is attempted at various values of overall fuel-air ratio with constant air mass flow.

2. The ignitor is sparked and the time until ignition is measured.

3. Ignition is confirmed from the end view of combustor. The flow rates were recorded when the fuel spray was ignited within 5 sec including the time lag of 1 sec for spray injection. A representative mean air velocity is calculated by equation (16). This measurement uncertainty is estimated to be less than 5%.

\[ U = \frac{4R \cdot Ta \cdot G_0}{5\cdot Pa - d^2} \text{ (m/s)} \]  

Effect of Spray

The ignition limit using INJECTOR-A is shown in Fig. 6 in comparison with that of INJECTOR-Al. The ignition limit of INJECTOR-A which has a wider spray cone angle expands further to the less fuel-lean side than that of INJECTOR-Al.

Effect of Air Flow

Figure 6 shows that ignition is impossible over a representative mean air velocity of 8 m/s, irrespective of the fuel-air ratio. To analyze this fact, numerical simulations for the combustion chambers were attempted. Figure 7 shows the results calculated by "FIRE3D-GT." With a velocity of 6 m/s, which is an ignitable condition, there is little up-stream flow from the main-chamber to the pre-chamber. A pathline started from a spark point swirls and moves gradually into the vortex center region formed in the pre-chamber. With a velocity of 14 m/s, which is a non-ignitable condition, there is a clear up-stream flow from the main-chamber to the pre-chamber. A pathline started from the spark point swirls along the pre-chamber wall.

From the observations and the above simulations, ignition process is considered to consist of the following three steps.

1. A flame kernel is formed near the sparking ignitor.

2. The flame kernel swirls and propagates into the vortex center region which is a flame holding region, as illustrated in Fig. 8.

3. The flame kernel develops a small rotating flame in the downstream region of the injector. The small rotating flame overlaps the next propagating flame kernel generated by the following ignition. The repetition of the steps forms a rotating flame in the pre-chamber.

At a fuel-air ratio of below 0.026, ignition is impossible with INJECTOR-Al with a narrow spray cone angle for the reason that a flame kernel is not formed near the ignitor because of scarce fuel supplied near the ignitor. At over a representative mean air velocity of 8 m/s, a flame kernel disappears without propagating into the vortex center region because of a high-speed barrier in the pre-chamber, even when a flame kernel is formed. Under a representative mean air velocity of 2 m/s, the flame oscillates unstably. This may result from the interaction between air stream and fuel injection.
LEAN BLOWOUT LIMIT

Lean blowout limit was measured by the following procedures.

1. Combustor inlet pressure is kept to 0.15 MPa. Under constant combustor inlet temperature and air mass flow, the overall fuel-air ratio is decreased slowly till the flame blows out.

2. The flow rates at the blowout are recorded.

Effect of Air Distribution

The influence of the distribution of air on the lean blowout limit at a constant combustor inlet temperature of 290K was investigated using COMBUSTOR-I, and COMBUSTOR-II, -II1, and -II2 with INJECTOR-A. The open area of primary air was 45% for COMBUSTOR-I, 41% for combustor-II, 33% for COMBUSTOR-II1 and 27% for COMBUSTOR-II2. As shown in Fig. 9, the blowout limit seems to extend further to the fuel lean region with decrease in the open area of primary air. Actually, however, the introduced velocity of primary air in the above 3 types of combustors is almost the same value, except for COMBUSTOR-II2 with half open area of primary air. Therefore, the velocity in the pre-chamber is very important to stabilize the flame.

Effect of Air Temperature

The influence of combustor inlet temperature, 290K, 370K and 470K, on the lean blowout limit is compared in Fig. 10. The limit for INJECTOR-A is different from that for INJECTOR-A1 under 470K, while it maintains a constant fuel-air ratio of less than 0.001 at 470K and above, irrespective of the representative mean air velocity. Lean blowout limits are affected by the fuel spray because it is hard to vaporize at lower temperatures. The stability of INJECTOR-A was increased, because a wider spray cone angle decreased the spray injection velocity. Although combustor inlet temperature for the regenerative gas turbine is over 650K at the idling conditions, it is under 470K during the time between ignition and idling in which the regenerator is not heated. Therefore, it is necessary to evade lean-blowout at the temperatures under 470K.

EXHAUST EMISSION

Exhaust emissions were measured by the following procedures.

1. With constant combustor inlet pressure, temperature and air mass flow, the overall fuel-air ratio is increased by step from 0.002 until 0.017.

2. Exhaust gases are sampled from downstream of the combustor exit. The distance from the exit is 7 times the combustor exit diameter.

3. The sampling gases are introduced through a heating line controlled at 464±6 K to a gas analyzer, which measures each concentration of CO, CO2, HC and NO/NOx after water vapor is removed. The emission index of each gas is calculated by equation (17). This measurement uncertainty is estimated to be less than 8%.
Case of COMBUSTOR-II

Figure 11 shows the combustion inefficiency and the NOx emission index against overall fuel-air ratio, in the case of COMBUSTOR-II with INJECTOR-A. The combustion inefficiency decreases and the NOx emission index increases in proportion to the fuel-air ratio until 0.007. However, it is interesting that they keep a constant value between fuel-air ratios of 0.007 and 0.012. Over 0.012, combustion inefficiency decreases again. These tendencies in the combustion inefficiency and the NOx emission index against fuel-air ratio are the same regardless of combustor inlet temperature.

The measured and the calculated NOx emission indexes are compared in Fig. 12. Good agreement in the tendency of NOx emission index is obtained between the calculation with consideration of radiation and the measurement. In an actual combustor, a luminous flame which looks brilliant yellow-white is formed. Therefore, the radiation from radiative flame must be taken into consideration in the calculation.

Calculations for COMBUSTOR-II were carried out for fuel-air ratios of 0.004, 0.008 and 0.017, whose results are shown in Fig. 13. At F/A = 0.004, combustion almost finishes in the pre-chamber, because of a relatively small amount of spray for a lot of primary air. At F/A = 0.008, a part of the flame reaches the main-chamber. Since the flame is constricted by the orifice at the exit of the pre-chamber, the temperature near the center region of the flame is rather high. At F/A = 0.017, the flame extends into the main-chamber. The high temperature region widely spreads into the center region of the main-chamber where the concentration of soot is highest, considering the soot formation and oxidation shown in Fig. 14. The highest temperature is lower and its region is narrower than those of the simulation without radiation. This is because much of the radiation flux from soot results in a loss of the energy released from combustion. The concentration of NO is also highest in the same region as shown in Fig. 15.

Since the up-stream region near the center region in the main-chamber is rather small due to restriction of the gas flow by an orifice between the pre-chamber and the main-chamber, the burnt gases pass through the high temperature region with short residence time. Because of the above two results, the NOx emission index above F/A = 0.007 does not increase in proportion to the fuel-air ratio. It appears that secondary air does not induce a decrease in temperature and the concentration of soot for COMBUSTOR-II.
Comparison of NOx Emission Index

The NOx emission indexes of COMBUSTOR-I, COMBUSTOR-II and COMBUSTOR-III using INJECTOR-A are compared in Fig. 16. The tendencies of the emission indexes change at a fuel-air ratio of about 0.007 for all the combustors.

The NOx emission index for COMBUSTOR-II is almost the same as that for COMBUSTOR-I between fuel-air ratios of 0.002 and 0.007. Above 0.007, that for COMBUSTOR-II is smaller than that for COMBUSTOR-I which has no secondary air. The NOx emission index for COMBUSTOR-III is smallest, due to the effect of secondary air introduced through two pairs of opposed circular air pipes in the main-chamber. The use of INJECTOR-B in place of INJECTOR-A for COMBUSTOR-III produces an even smaller NOx emission index, due to the effect of good spray dispersion in the pre-chamber. Incidentally, the NOx emission index also decreases when the combustion becomes bad. Then, the NOx emission indexes against inefficiency are compared in Fig. 17. Although the NOx emission index for COMBUSTOR-I...
and -II tend to decrease in the region of less than some inefficiency, that for COMBUSTOR-III increases against inefficiency. The figure shows that COMBUSTOR-III with INJECTOR-B has the smallest NOx emission index when the combustion inefficiency is the same. Fig. 18 shows the calculation for COMBUSTOR-III at F/A = 0.017 in comparison with COMBUSTOR-II. The secondary air introduced through air pipes reaches the center region of the main-chamber in COMBUSTOR-III and decreases the temperature rise.

Effect of Temperature and Pressure

The emission indexes of exhaust CO, HC and NOx depend on combustor inlet temperature and pressure. Table 3 shows coefficient h and index z of temperature and pressure dependences, respectively, using COMBUSTOR-II with INJECTOR-A, when the emission index of each gas is expressed by equation (18).

\[ EI/EI_{ref} = (E/E_{ref})^2 \cdot \exp[h \cdot (T - T_{ref})] \]

(18)

The temperature dependency of NOx is \(2 \times 10^{-3}\). This coefficient is smaller than that of literature[15], that is \(4 \times 10^{-3}\). This dependency is suitable for the regenerative gas turbine, which tends to increase the combustor inlet temperature in order to achieve its higher efficiencies. The pressure dependency of NOx is 0.6 at a full load (F/A = 0.015). This is almost the same as that of literature, 0.5. The NOx emission index of the present regenerative gas turbine engine for vehicle application under a rated load operation (Ta = 850K, Pa = 0.6 MPa) is almost 10 g/kg-fuel, which may conform the long-term NOx standard for heavy duty vehicles in Japan.

Table 3 Dependence of exhaust emission index on temperature and pressure; using COMBUSTOR-II with INJECTOR-A

<table>
<thead>
<tr>
<th>F/A</th>
<th>NOx</th>
<th>CO</th>
<th>HC</th>
<th>1-(n_b)</th>
<th>NOx</th>
<th>CO</th>
<th>HC</th>
<th>1-(n_b)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.005</td>
<td>3x10^{-3}</td>
<td>-2x10^{-3}</td>
<td>8x10^{-3}</td>
<td>-5x10^{-2}</td>
<td>0.3</td>
<td>-1</td>
<td>-1</td>
<td>-1</td>
</tr>
<tr>
<td>0.010</td>
<td>3x10^{-3}</td>
<td>-7x10^{-3}</td>
<td>9x10^{-3}</td>
<td>-8x10^{-2}</td>
<td>0.3</td>
<td>-2</td>
<td>-1</td>
<td>-1.5</td>
</tr>
<tr>
<td>0.015</td>
<td>2x10^{-2}</td>
<td>-8x10^{-3}</td>
<td>7x10^{-2}</td>
<td>-6x10^{-2}</td>
<td>0.6</td>
<td>-3</td>
<td>-0.6</td>
<td>-1.5</td>
</tr>
</tbody>
</table>

SUMMARY

Influence factors of the lean ignition limit, the lean blowout limit and the exhaust emission characteristics have been investigated for spray combustion using pre-chamber type vortex combustors under fuel lean operations.

1. The lean ignition limit depends on the spray characteristics of the fuel injector and the air flow pattern in the pre-chamber, which affect the formation of a flame kernel near the sparking ignitor and its propagation into a flame holding region forming a rotating flame in the pre-chamber.

2. The lean blowout limit varies largely with the combustor inlet temperature. It also depends on the air flow pattern in the pre-chamber under a combustor inlet temperature of 470K, while a constant fuel-air ratio of less than 0.001 is maintained at 470K and above.

3. In the developed pre-chamber type vortex combustor, combustion almost finishes in the pre-chamber at a low fuel-air ratio. However, at a higher fuel-air ratio, combustion in the main-chamber becomes dominant. Therefore, the tendencies of the emission indexes of exhaust CO, HC or NOx change at a certain fuel-air ratio where the flame stretches from the pre-chamber into the main-chamber.

4. With no or a little secondary air introduced into the main-chamber, the NOx emission index is constant against overall fuel-air ratio at an air temperature of 640K and a pressure of 0.15 MPa. This tendency is explained by the following two reasons: One is that gas temperature rise is restricted by radiation heat lost from soot formed near the center region in the main-chamber. The other is that the residence time of burnt gases is shortened by the decrease of up-stream region.

5. As the result of the restriction in gas temperature rise by effective introduction of secondary air, the NOx emission index decreases further by about 30%.

6. The NOx emission index decreases even more by about 10%, as the result of improvement in spray dispersion, that is, changing from a swirl type injector to an airblast type injector.

REFERENCES


