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A Study of a Direct-Injection Stratified-Charge Rotary Engine for Motor Vehicle Application

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ABSTRACT

A study of a direct-injection stratified-charge system (DISC), as applied to a rotary engine (RE) for motor vehicle usage, was undertaken. The goals of this study were improved fuel consumption and reduced exhaust emissions. These goals were thought feasible due to the high thermal efficiency associated with the DISC-RE. This was the first application of this technology to a motor vehicle engine. Stable ignition and ideal stratification systems were developed by means of numerical calculations, air-fuel mixture measurements, and actual engine tests. The use of DISC resulted in significantly improved fuel consumption and reduced exhaust emissions. The use of an exhaust gas recirculating system was studied and found to be beneficial in NOx reduction.

INTRODUCTION

The need to continually improve the fuel efficiency of IC engines has resulted in many novel innovations. In regards to rotary engines (RE), active research has been undertaken on a direct-injection stratified-charge system (DISC). These studies were performed on engines designed for industrial or aircraft applications. The DISC-RE has a high thermal efficiency which has been previously studied [1,2]. The goal of our study was to research the suitability of using this technology for motor vehicle usage. This is an area in which no research has been done. This study produced the successful development of a gasoline-fueled DISC-RE (Figure 1). This engine demonstrated improved fuel consumption, and lowered exhaust emissions performance.

In this paper we will report our examination of ignition and stratification technology. We will also discuss the related comprehensive evaluations in terms of the development progress. Table 1 highlights the specifications of our developmental test unit DISC-RE.

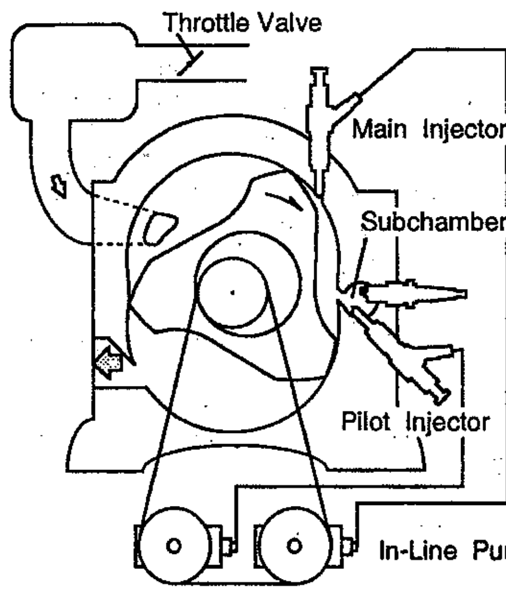


Figure 1. Schematic Diagram of Rotary Engine with Direct Injection Stratified Charge (DISC-RE).

Table 1. Engine Specification.

Engine Type	Wankel Type Rotary Engine Natural Aspiration
Displacement	654 cc x 2 Rotor
Compression Ratio	9.7 : 1
Port Timing	IO / IC = 32°(ATDC)/30°(ABDC) EO/EC = 75°(BBDC)/48°(ATDC)
Fuel Injection	In-Line Pump , Hole Nozzle

DEVELOPMENT TARGETS

The advantage of the DISC system is that it raises the thermal efficiency of the engine. This improvement is mainly a result of three factors.

(1) A reduction in pumping losses as the engine has no throttle intake valve.

(2) A decrease in heat loss as a result of localized combustion.

(3) A suppression of the unburned fuel loss as a result of localized combustion.

These factors have resulted in a high thermal efficiency for industrial or aircraft DISC-RE systems. When the DISC system is applied to a motor vehicle engine, this creates a non-throttle engine. This is most effective for thermal efficiency, but reduces the temperature of the exhaust gas. This temperature decrease is detrimental to the purification of the exhaust gas when using a catalytic converter. The most practical method for raising the exhaust gas temperature is to throttle the intake air. We strove to develop an DISC-RE system for motor vehicle usage by throttling the intake air. Our goal was to obtain ideal DISC combustion over a wide operating range. We theorized that although the effect of factor "(1)" would decline somewhat, the effect of factors "(2)" and "(3)" would increase, owing to the high surface volume ratio inherent to the RE. The intake air throttling has the additional advantages of reducing both vibration and noise.

The most serious trouble arising from the throttling of intake air is unstable ignition or combustion caused by local-richness. Overcoming this problem is the key for applying the DISC system to motor vehicle usage. We thus searched for a stable ignition system and ideal stratification system that would not be influenced by driving conditions.

TECHNOLOGY FOR STABLE IGNITION

First, we made a search for an ignition system that realized stable ignition even while throttling. This study was performed without a main injector shown in Figure 1, because the main injector prevented us from making clear the ignition performance.

SUBCHAMBER IGNITION SYSTEM - We are defining the subchamber as the area around the spark plug. The subchamber was designed to secure an air-fuel mixture suitable for ignition around the spark plug. Figure 1 shows the nozzle and spark plug configuration. Figure 2 shows details of the subchamber compared with the spray ignition system used for industrial DISC-RE applications [1,2]. The nozzle is placed such that the injected fuel will reach the main combustion chamber without colliding with the walls of the subchamber. The connecting

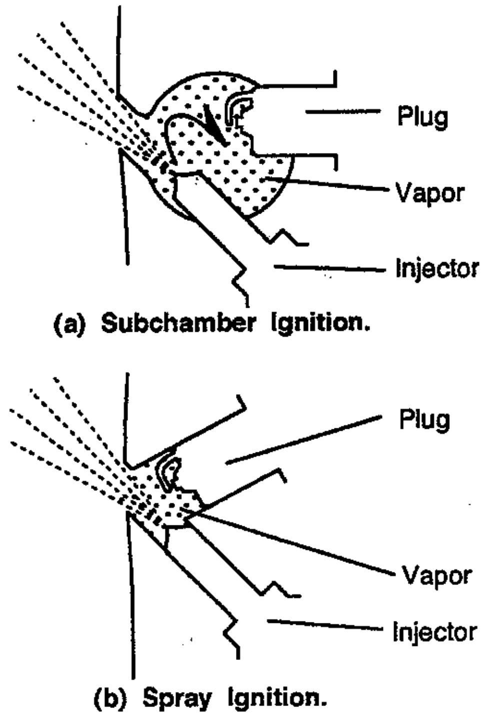


Figure 2. Ignition Method.

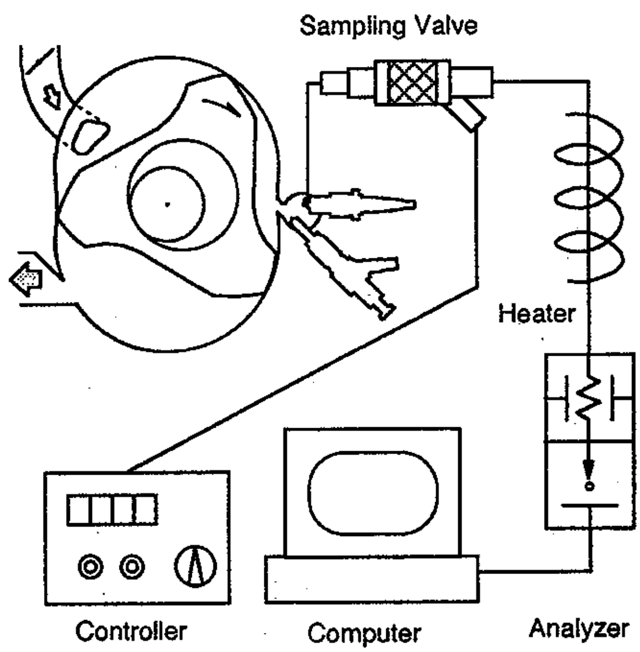


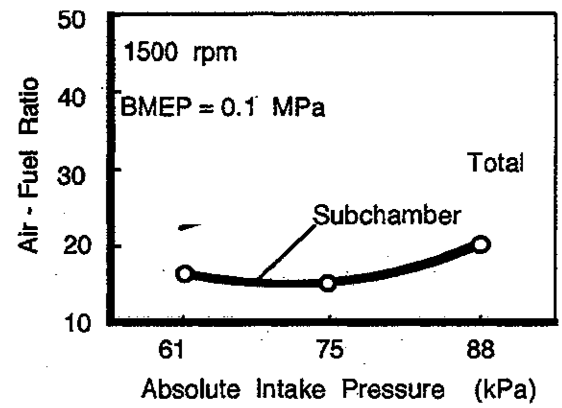
Figure 3. Gas Sampling Test Apparatus.

hole between the main chamber and subchamber is throttled to control the flow. With these specifications, we aimed to achieve the functions described below. Although the majority of the *injected fuel goes into the main combustion chamber,* the small diameter of the connecting hole causes fuel vapor and a small amount of fuel droplets to remain in the subchamber. Having completely gaseous fuel around the spark plug, results in continuous stable ignition without the air-fuel ratio being influenced by the fuel injection rate, injection timing, or inner pressure of the combustion chamber. The fuel injected into the main combustion chamber is stratified in the rotor recess and burned by flames from the subchamber. This results in stratified combustion.

MEASURING THE AIR-FUEL RATIO IN THE SUBCHAMBER OF AN ACTUAL ENGINE - To verify the ignitibility of the subchamber ignition system, we measured the air-fuel ratio in the subchamber of an actual engine.

Measurement Devices and Methods - To analyze the combustion chamber gas, a Mazda-developed rapid-response analyzer [3] was used. This analyzer is based on a quadruple mass spectrometer. The test configuration is shown in Figure 3. With this analyzer, a sample of the air-fuel mixture is taken by an electro magnetic valve. This sampling valve opens at a designated time (2 ms before ignition) of engine rotation. The sample is passed through the heated introduction tube to the measurement device. The quantities of fuel and air are measured and the resultant air-fuel ratio is displayed on the computer monitor.

Measurement Results - Figure 4 shows the air-fuel ratio of an air-fuel mixture sample in the subchamber that was taken from the spark plug, compared with that of a sample obtained from the exhaust gas. Intake pressure is absolute. This test was performed at an engine speed of 1500 rpm. Under a brake mean effective pressure (BMEP) of 0.1 MPa, variations in the intake pressure produce a smaller fluctuation of the air-fuel ratio in the subchamber than the total air-fuel ratio. When the air density is high, the elevated drag force increases the amount of fuel remaining in the subchamber. The leanness that would otherwise result from the increased air in the chamber due to increased density is thus suppressed. Conversely, when the air density is low, the decrease in drag force increases the amount of fuel reaching the main combustion chamber. This results in the richness of the air-fuel mixture being suppressed. In a similar manner the air-fuel ratio fluctuated less than would be expected from air density alone when the injection quantity was varied under a constant intake pressure of 88 kPa. The air-fuel ratio in the subchamber was only minimally affected by intake pressure and load, thus



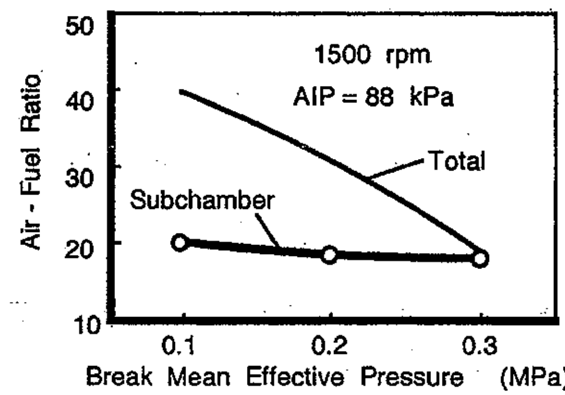


Figure 4. Air - Fuel Ratio in Subchamber.

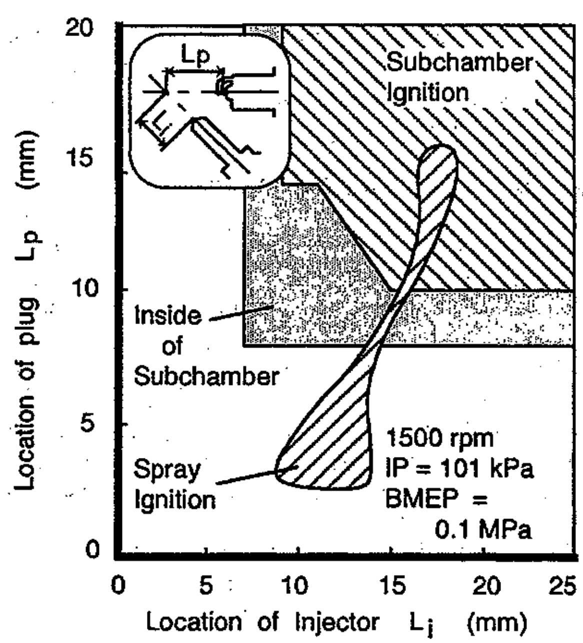


Figure 5. Ignition Performance - Influence of Injector and Plug Locations.

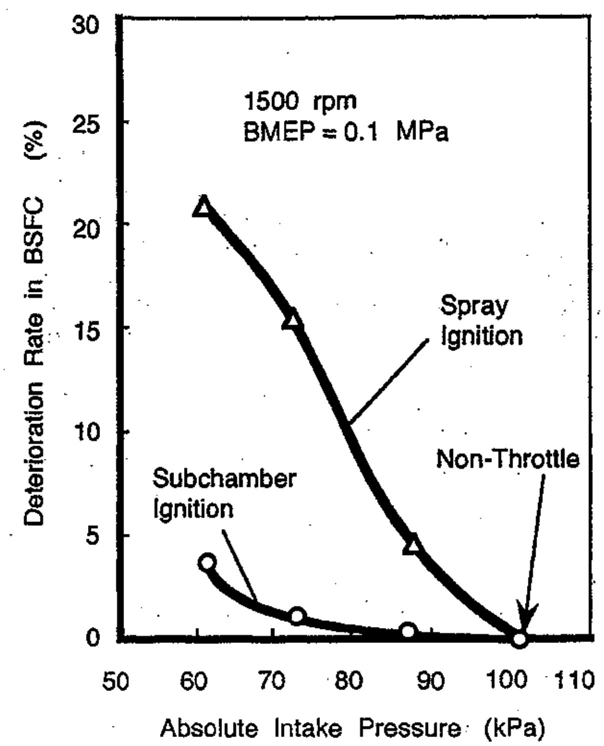
stable ignition performance could be expected.

IGNITION PERFORMANCE - During ignition performance testing, we studied the spray ignition system, as shown in Figure 2 (b). We realized that the locations of the injector nozzle and spark plug would influence the ignition performance. We examined the effect of changing Li ; the distance between the imaginary intersection of the injector nozzle and the spark plug, and the tip of the nozzle, and Lp ; the distance between that same imaginary intersection and the spark plug electrode. Li and Lp are shown in Figure 5.

Figure 5 also shows a map of the non-misfiring regions. As the figure indicates, the ignition region of the spray ignition system is very narrow. This results in the combustible mixture portion being severely confined. In case of the subchamber ignition system, the majority of the air-fuel mixture is ignitible. Differences between the two ignition systems are chamber shape and chamber volume. But change of Lp causes change of chamber volume. Therefore we suppose that the shape of chamber is more important for the holding effect of the gaseous fuel.

Ignitability of both system during throttling was evaluated by comparing changes in specific fuel consumption. Figure 6 presents a comparison of such changes with and without intake air, throttling. The horizontal axis represents absolute intake pressure. Specific fuel consumption was kept essentially constant with the subchamber ignition system. On the other hand, fuel consumption deteriorated rapidly with intake throttling using the spray ignition system.

Shown in Figure 7 are the experimental results of the effects of subchamber volume. Similar to the case in Figure 6; changes in specific fuel consumption occurred while throttling the intake air. Tests were conducted using three different subchamber volumes; 5 cc, 12 cc, and 16 cc. These volumes exclude the volume of the connecting hole. The shape of this connecting hole was held constant. In the 5 cc subchamber, specific fuel consumption deteriorated rapidly as intake pressure declined. In the 12 cc subchamber, specific fuel consumption deteriorated remarkably under intake pressure of less than 75 kPa. No rapid deterioration within the range of the test conditions was noticed for the 16 cc subchamber. Based upon these results, we determined that at least a 12 cc subchamber was necessary to sufficiently demonstrate the holding effect of the gaseous fuel. At the same time, taking into account of the fact that the smaller the subchamber's volume, the better its scavengeability, reliability, and compressibility, we used a 12 cc subchamber for all subsequent tests.



Figre 6. Ignition Performance - Subchamber Ignition vs Spray Ignition.

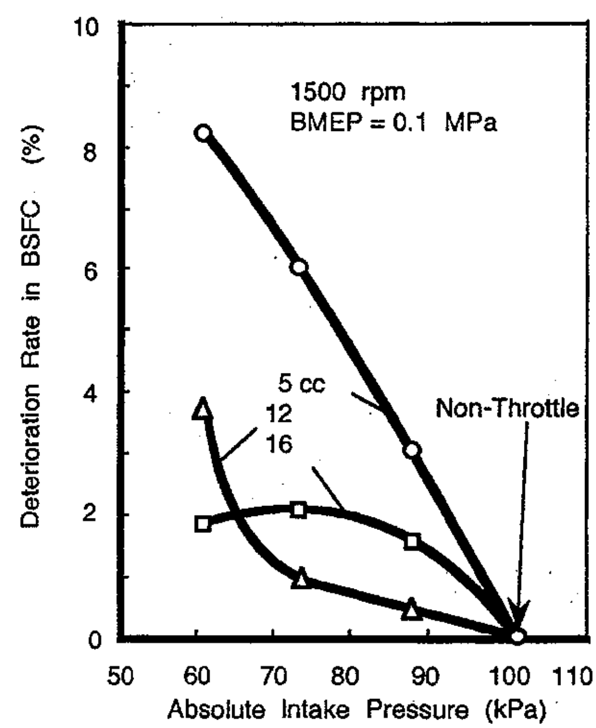


Figure 7. Subchamber Volume Effects - 5cc vs 12cc vs 16cc.

MAIN-FUEL STRATIFICATION TECHNOLOGY

Next, we optimized the location of the main fuel injector to improve air utilization in the combustion process. Figure 8 shows both the upstream and downstream injection positions which were investigated. This study was performed without the subchamber ignition system to make clear the degree of mixture stratification by means of the main injection.

FUEL SPRAY BEHAVIOR PREDICTIONS - Air utilization is influenced by the fuel trajectory and fuel evaporation characteristics. The behavior of the fuel spray was predicted through calculations on both the upstream and downstream injectors.

Theoretical Calculations - The principle factors controlling the behavior of the fuel spray within the combustion chamber are as follows.

- In-chamber flow.
- Interaction between droplets.
- Secondary air flow caused by spray movement.

Although these factors are highly complicated, we made simplified, two-dimensional calculations to predict the relative tendencies of the two injection methods. For in-chamber flow, we assumed that the flow rate was equal to the speed of the rotor wall surface. The flow direction was determined by the direction of rotor wall surface movement and by the tangential direction of the rotor housing trochoids. Assumptions made in regards to evaporation were as follows.

- The fuel droplets were true spheres.
- The inside temperature distribution was even.
- Diffusion occurred only through differences in concentration.

The equation for the motion of the liquid droplets is expressed below.

$$m\frac{d^2X}{dt^2} = F_x \quad , \quad m\frac{d^2Y}{dt^2} = F_y \qquad (1)$$

Where,

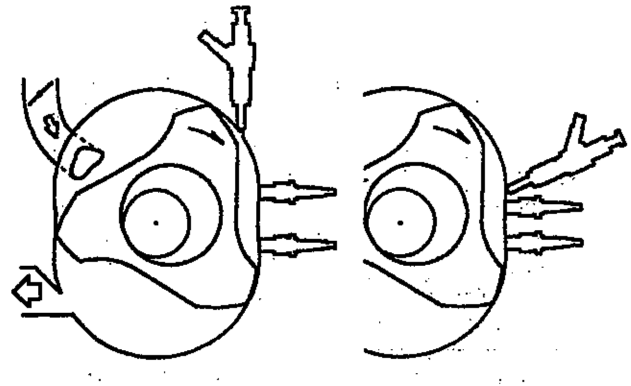
m	;	Mass of droplet.
X,Y	:	Cartesian coordinates of the droplet.
t	;	Time.
F_x	:	X component of drag forces.
F_y	:	Y component of drag forces.

The drag forces, Fx and Fy, are expressed in the following equations.

$$F_x = F\cos\theta \quad , \quad F_y = F\sin\theta \qquad (2)$$

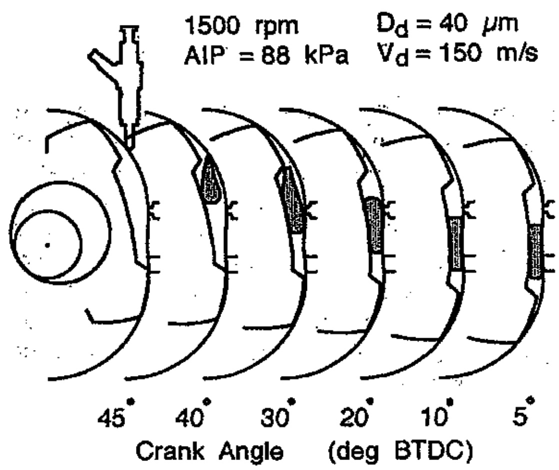
$$F = \frac{\pi D_d^2 \rho_a C_{dm}}{8}\{(V_{ax}-V_{dx})^2+(V_{ay}-V_{dy})^2\} \qquad (3)$$

$$\theta = \tan^{-1}\left(\frac{V_{ay}-V_{dy}}{V_{ax}-V_{dx}}\right) \qquad (4)$$

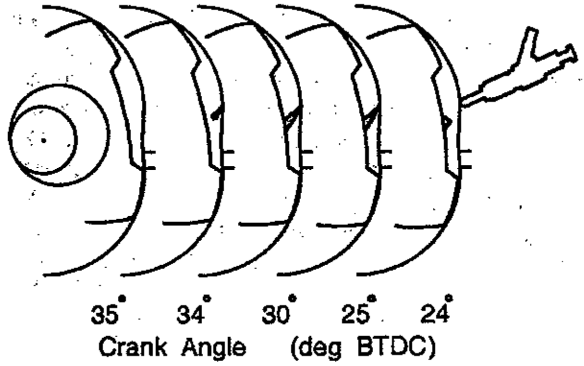


(a) Upstream Injection. (b) Downstream Injection.

Figure 8. Location of Main Injector.



(a) Upstream Injection.



(b) Downstream Injection.

Figure 9. Spray Trajectory Comparison - Upstream vs Downstream Injection

Where,

D_d : Diameter of droplets.
ρ_a : Density of the air.
V_{ax} : X component of the air velocity.
V_{ay} : Y component of the air velocity.
V_{dx} : X component of the droplet velocity.
V_{dy} : Y component of the droplet velocity.
C_{dm} : Drag coefficient of the evaporating droplets.[4]

Droplet vapor mass per unit of time and temperature are expressed in the following equations using evaporation ratio Ke.

$$\frac{dm}{dt} = -\frac{\pi}{4} D_d \rho_d K_e \qquad (5)$$

$$K_e = \frac{4\,\rho_m\,\delta\,Sh}{\rho_d} \ln\left(\frac{1-\omega_{fg}}{1-\omega_{fl}}\right) \qquad (6)$$

$$T_d = T_a - \frac{H}{C_{pm}} \exp\left(\frac{\rho_d C_{pm} K_e}{4\,Nu\,\lambda_m}\right) \qquad (7)$$

where,

ρ_m : Density of the air-fuel mixture.
ρ_d : Density of the droplet.
λ_m : Thermal conductivity of the air-fuel mixture.
C_{pm}: Specific heat of the air-fuel mixture.
ω_{fg} : Vapor mass fraction in the bulk.
ω_{fl} : Vapor mass fraction at the droplet surface.
δ : Diffusibility.
T_a : Temperature of the air.
T_d : Temperature of the droplet.
H : Latent heat of evaporation.
Sh : Sherwood number.[6]
Nu : Nusselt number.[6]

These equations were used to calculate the evaporation ratio of a single fuel droplet in the chamber. The predicted evaporation ratio was larger than that of the actual fuel droplets, but was acceptable for relative comparisons. Oversimplified, the change in fuel droplet momentum is the same as the change in in-chamber flow momentum. To increase our accuracy, in terms of trajectory, the secondary air flow caused from the fuel injection movement was taken into account when determining the change in fuel droplet momentum.

Calculation Results - Figure 9 shows the spray trajectory under the following conditions.

- Engine rotation = 1500 rpm.
- Intake pressure = 88 kPa
- Initial diameter of droplet = 40 μm.
- Initial injection velocity = 150 m/s.

In the downstream injection, the injected fuel collides with the rotor with no time to be caught by the flow. The time between the start of injection, until all of the injected fuel has collided with the rotor, is so short (13° as measured on the output shaft) evaporation

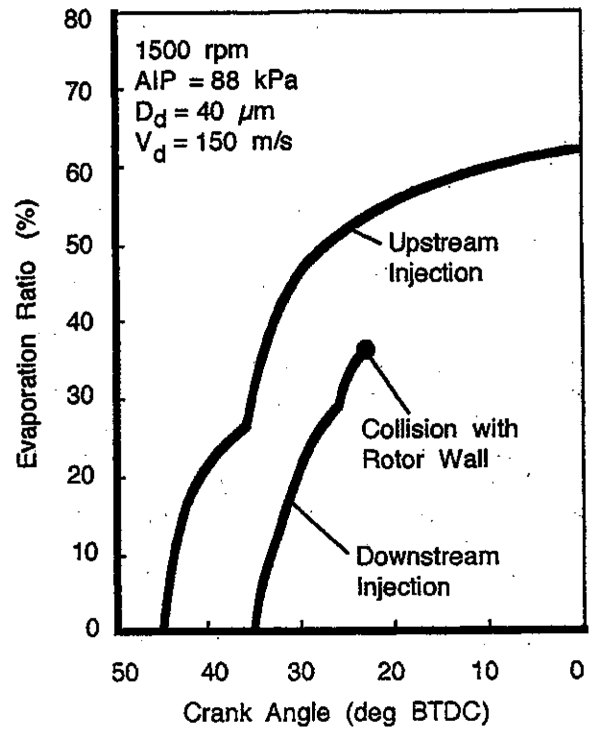


Figure 10. Evaporation Ratio of Fuel During Flight - Upstream Injection vs Downstream Injection.

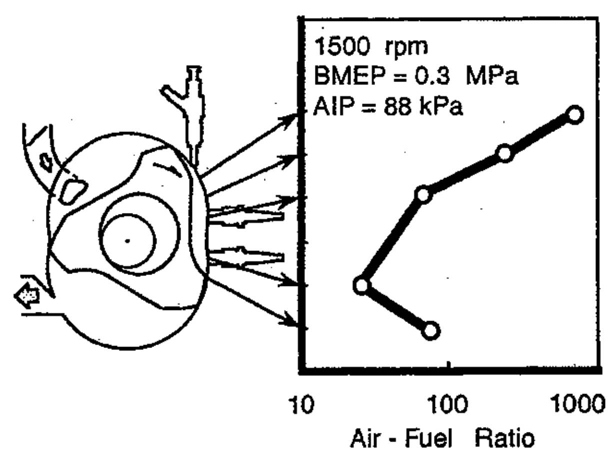


Figure 11. Fuel Distribution in Main Chamber of Upstream Injection.

during flight is not expected. In the upstream injection, we anticipated that the injected fuel would fly to the nearby TDC, resulting in a considerable advantage for vaporization.

Figure 10 shows the evaporation ratio of injected fuel during flight. With downstream injection, only about 38 % of the fuel evaporated in the time between the start of injection until the collision. Little evaporation after the collision was expected. In contrast, with upstream injection, a total of about 60 % of the injected fuel evaporated. As upstream fuel injection is characterized by a wide flight range and advantageous evaporation, it is considered suitable for fuel stratification under throttling.

MEASURING FUEL DISTRIBUTION IN THE COMBUSTION CHAMBER - Having determined the upstream injection method to be advantageous by our previous calculations, the fuel distribution in the main combustion chamber was quantified. This measurement was via the same procedure used to measure the air-fuel ratio in the subchamber (described previously). A 1.5 mm-diameter sampling hole in the trochoid surface was used for measurements.

Figure 11 shows measurement results with an engine speed of 1500 rpm, and BMEP = 0.3 MPa. This figure demonstrates that upstream fuel injection stratifies the fuel on the leading side of the combustion chamber. As it is difficult to burn fuel on the trailing side of the combustion chamber on an RE, stratification on the leading side is ideal. We thus determined that upstream fuel injection would enable an ideal fuel distribution with efficient stratified combustion even while throttling.

COMPREHENSIVE PERFORMANCE

Performance characteristics were evaluated in the test engine equipped with a subchamber ignition system and an upstream fuel injection system as shown in Figure 1. This allowed for excellent ignitability while throttling, as well as ideal stratification.

FUEL CONSUMPTION AND EMISSIONS PERFORMANCE - Figure 12 shows the relationship between the absolute intake pressure and exhaust temperature of the engine operated at 1500 rpm. As can be seen in the figure, the lower the intake pressure, and the greater the load, the higher the exhaust temperature. The activation temperature required for a catalytic converter is 250 ℃. This is the lowest possible temperature for exhaust gas purification. Figure 12 shows that the intake pressure for securing an exhaust temperature of 250 ℃, was 80 kPa at BMEP = 0.1 MPa, 95 kPa at 0.2 MPa, and unobtainable at 0.3 MPa.

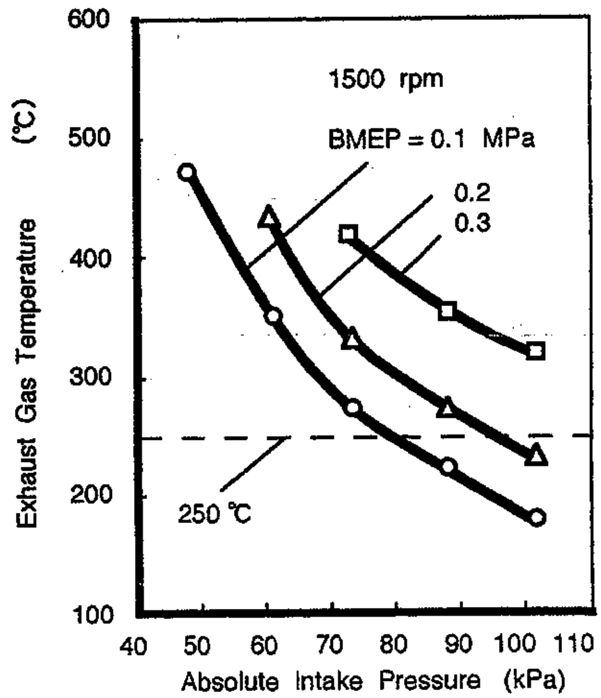


Figure 12. Influence of Intake Pressure on Exhaust Gas Temperature.

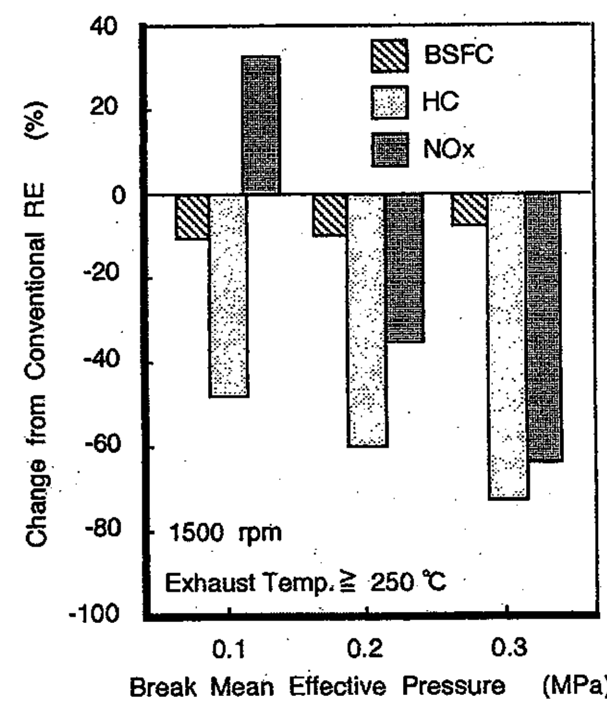


Figure 13. Fuel Consumption and Exhaust Emissions Performance - Comparisons with Conventional RE.

Based upon those values, the specific fuel consumption and emissions were then measured while maintaining an exhaust gas temperature of 250 ℃ and over. These experimental results were compared with those of a conventional premixed charge RE. These results are shown in Figure 13. They show an improvement in fuel consumption of about 11 % was obtained, even under a light load. An average improvement in fuel consumption of about 10 % was indicated. In terms of emissions performance, drastic improvements in HC and NOx emissions were obtained except under light loads. The average HC emissions improvement was about 60%. This is contrary to the general tendency for HC emissions to increase with stratified combustion. This is due to the major reduction of unburned fuel losses on the combustion wall, especially along the trailing side. This is a unique tendency of the RE, which has a high surface-volume ratio. The NOx emissions were improved as much as 63 %. However, they deteriorated under light load. We suppose the reason is as follows. The subchamber is spherical, besides it keeps completely gaseous fuel. It leads to higher combustion temperature and more NOx emissions than conventional RE. On the other hand, the mixture in the mainchamber include many droplets. It leads to slower combustion and less NOx emissions than conventional RE. The amount of NOx emissions likely depends on the combustion ratio between the mainchamber and the subchamber. The subchamber combustion ratio at light loads is higher than that at heavy loads, because quantity of pilot fuel is constant under whole operating conditions.

IDLING CONDITION PERFORMANCE - Next, the idling condition performance was evaluated. As the fuel injection rate is extremely low in idling, the main fuel injection was cut off, and only the subchamber fuel was injected. Figure 14 shows exhaust gas temperatures and comparisons of specific fuel consumption with those of a conventional RE at 800 rpm. The horizontal axis represents absolute intake pressure. Although we assumed that the increased intake air throttling would increase the pumping losses, fuel consumption was improved to a maximum of 23 %. This improvement was due to the lean air-fuel mixture in the main chamber becoming rich when the intake air was throttled. This richened air-fuel mixture is more suitable for combustion. However, exhaust gas temperatures only reached 170 ℃, even when the intake air was throttled at an intake pressure of 48kPa, the point at which misfiring occurs. To meet this problem, an optimized fuel stratification under even lower intake pressure conditions is needed. This will allow improved engine operation under even lower intake pressure conditions. This is a primary task for

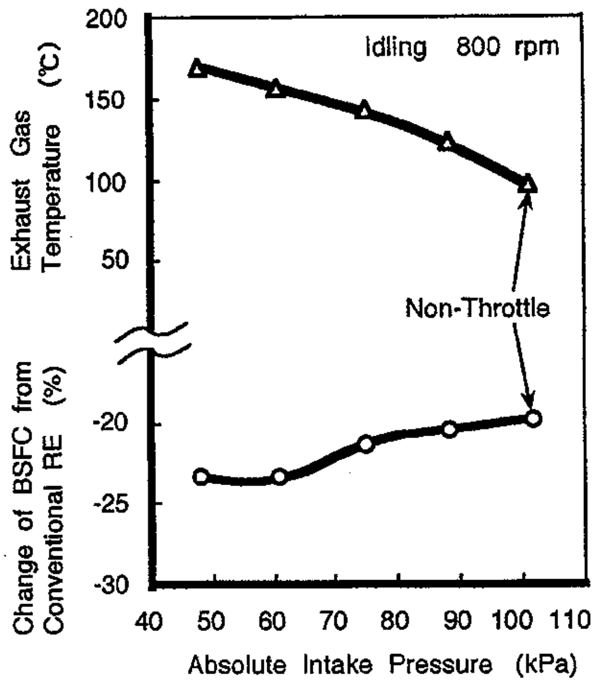


Figure 14. Idling Condition Performance.

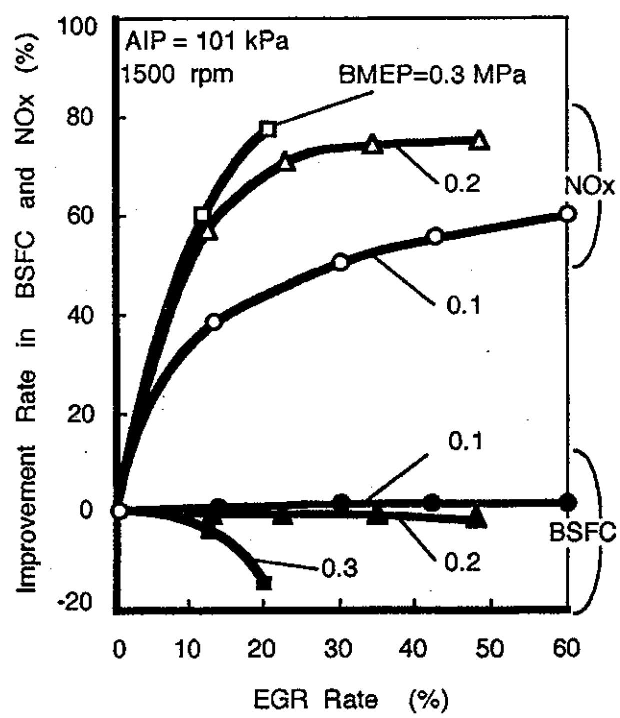


Figure 15. Exhaust Gas Recirculation (EGR) Comparison of Fuel Consumption and NOx effects.

future studies.

REDUCTION OF NOx EMISSIONS - It is difficult to reduce NOx with the catalytic converters for lean combustion engines using current technology. Therefore, we sought to suppress NOx emissions within the combustion process. To achieve this goal, we investigated the possibility of reducing NOx emissions with an exhaust gas recirculation (EGR) system.

Figure 15 shows the rate of improvement in brake specific fuel consumption and NOx emissions as a function of EGR rate. Under an EGR rate of 20 %, and a BMEP = 0.3 MPa, NOx emissions were reduced 79 %, but specific fuel consumption increased 16 %. With the BMEP = 0.2 MPa, and an EGR rate of 50 %, NOx emissions were reduced 76 %, and the specific fuel consumption remained nearly constant. At a BMEP of 0.1 MPa, NOx was reduced 60 %, and a 2.2 % improvement in specific fuel consumption was observed. We thus confirmed that an EGR system was a possible method to reduce either NOx emissions, or specific fuel consumption under a specific operating range.

CONCLUSIONS

We strove to create a motor vehicle DISC-RE with an acceptably wide range of stratified combustion by throttling under light load conditions. Combining our numerical calculations, fuel distribution measurements, and actual engine tests, led us to the following results.

(1) A subchamber ignition system was developed that resulted in stable ignition, even when throttling.

(2) An upstream injection system was developed that realized ideal stratified combustion. This system was successful due to a high air utilization rate in combustion. This air utilization was maintained even while throttling.

(3) By combining the subchamber ignition system with the upstream injection system, we improved both fuel consumption and exhaust emissions under a wide operating range. This was done while preventing exhaust gas temperatures from decreasing excessively under light loads.

(4) An exhaust gas recirculating system proved extremely effective for reducing NOx emissions.

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