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DEVICES TO IMPROVE THE PERFORMANCE OF A CONVENTIONAL TWO-STROKE SPARK IGNITION ENGINE

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ABSTRACT

This paper presents research efforts made in three different phases with the objective of improving the fuel economy of and reducing exhaust emissions from conventional, carbureted, two-stroke spark ignition (SI) engines, which are widely employed in two-wheel transportation in India. A review concerning the existing two-stroke engine technology for this application is included. In the first phase, a new scavenging system was developed and tested to reduce the loss of fresh charge through the exhaust port. In the second phase, the following measures were carried out to improve the combustion process: (i) using an in-cylinder catalyst, such as copper, chromium, and nickel, in the form of coating; (ii) providing moderate thermal insulation in the combustion chamber, either by depositing thin ceramic material or by metal inserts; (iii) developing a high-energy ignition system; and (iv) employing high-octane fuel, such as methanol, ethanol, eucalyptus oil, and orange oil, as a blending agent with gasoline. Based on the effectiveness of the above measures, an optimized design was developed in the final phase to achieve improved performance. Test results indicate that with an optimized two-stroke SI engine, the maximum percentage improvement in brake thermal efficiency is about 31%, together with a reduction of 3400 ppm in hydrocarbons (HC) and 3% by volume of carbon monoxide (CO) emissions over the normal engine (at 3 kW, 3000 rpm). Higher cylinder peak pressures (3-5 bar), lower ignition delay (2-4 °CA), and shorter combustion duration (4-10 °CA) are obtained. The knock-limited power output is also enhanced by 12.7% at a high compression ratio (CR) of 9:1. The proposed modifications in the optimized design are simple, low-cost, and easy to adopt for both production and existing engines.

BACKGROUND

The conventional, carbureted, two-stroke SI engine has a number of potential advantages over the equivalent four-stroke engine; these include higher specific power output, compactness, simple construction, lower production and maintenance costs, lower brake-specific NO_x emissions, lower engine friction, and reduced part-load pumping losses (Kenny et al. 1992). As a result of these advantages, the two-stroke SI engine is widely used in mopeds, motorcycles, three-wheeled autorickshaws, chainsaws, snowmobiles, lawn mowers, and outboard marine applications. In India, the two-wheeled vehicles powered by two-stroke SI engines are numbered around 17 million, compared to only 2.8 million passenger cars powered by four-stroke SI engines. By the year 2000, two-wheelers are projected to number around 35 million; in comparison, passenger cars are expected to number only about 4 million. Consequently, the two-wheelers are estimated to consume 60 to 65% of total gasoline in India, and their share may increase even further in the future (Kumar et al. 1994).

Despite various advantages and widespread applications, the conventional two-stroke SI engine has a number of distinct disadvantages compared to the four-stroke SI engine. These include (i) high specific fuel consumption, (ii) high unburned hydrocarbons and carbon monoxide emissions in the engine exhaust, and (iii) abnormal or irregular combustion at very light loads. Short-circuiting of the fresh charge through the exhaust port and the presence of a large percentage of exhaust residuals in the engine cylinder are primarily responsible for these drawbacks. The loss of fresh charge typically varies from 15 to 40% of the fuel supplied, and the residual burned gases may constitute about 20 to 40% of the total cylinder content at the end of the scavenging process,

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depending on the scavenging characteristics of the engine and its operating conditions (Kollman et al. 1973). The presence of a high amount of exhaust residuals inside the cylinder reduces the speed of flame propagation, leading to poor combustion (Wakisaka et al. 1976). Since it is very expensive to replace two-stroke with four-stroke SI engines, research efforts are necessary to improve the fuel economy and exhaust emission characteristics of the present carbureted, two-stroke SI engines with minimum modifications to the existing engine structure.

OBJECTIVES

The major objectives of the present work are to improve the fuel economy and to lower the exhaust emissions of a conventional, carbureted, two-stroke SI engine. In order to meet these objectives, research efforts are focused on reducing the short-circuiting of fresh charge through the exhaust port and on improving the combustion process. This work was carried out in three different phases:

1. Development of a new scavenging system to reduce fresh charge loss during the scavenging process.
2. Improvement in the combustion process, by the following methods.
   - (i) Using in-cylinder catalysts
   - (ii) Providing moderate thermal insulation of the combustion chamber
   - (iii) Developing a high-energy ignition system
   - (iv) Employing high-octane fuel blends
3. Development of an optimized design for two-stroke SI engines by combining the above modifications in order to obtain the best possible fuel economy with minimum exhaust emissions.

The following review provides a brief summary of the literature concerning the methods used to meet the above objectives. These approaches are broadly classified into two categories, improvements in scavenging and improvements in the combustion process.

IMPROVEMENTS IN THE SCAVENGING PROCESS

Research and development efforts made so far to reduce scavenged-through and other fuel losses may be categorized as fuel injection, premixed fuel-air scavenging, and scavenging with stratified charging systems.

Fuel Injection

Fuel injection on closing the exhaust port is considered to be a promising technique to eliminate the loss of fuel due to short-circuiting. Many investigations in this regard have employed high-pressure, mechanical or electronic fuel-injection systems (Sato et al. 1987, Douglas et al. 1982). In the fuel-injected system, the most important problem is to obtain adequate turbulence from the air scavenging process for good fuel-air mixing prior to combustion. This problem is particularly severe in the case of the cylinder head injection system. The development of an Orbital two-stroke engine (Automotive Engineering 1986) with pneumatic fuel injection has attracted considerable attention. In this system, compressed air is employed to atomize and induct a metered quantity of fuel into the engine cylinder. Pulsed compressed air from a piston pump operating at 4 to 7 bar is used to sweep the fuel, metered by another pump into the engine. Atomization of the fuel is claimed to be exceptionally fine, with over 95% of the droplets having a size below 10 microns. Another system, developed by Duret et al. (1988), uses a crankcase rather than an air pump to provide low-pressure air at 1.4 to 1.5 bar for assisting fuel atomization. The fuel-air mixture is injected into the engine on intake valve opening; a lower fuel consumption (less than 275 g/kWh) is realized in this system over the Orbital engine. Nevertheless, the high cost of fuel injection systems restricts extensive application in small two-stroke SI engines. Therefore, the methods to reduce or eliminate short-circuiting of fresh charge are aimed towards finding a system with minimum mechanical complexities and low-cost, and continuing emphasis is being placed on attempts to scavenge the engine cylinder with fresh air along with the usual provision of charging from the carburetor.

Premixed Fuel-Air Scavenging

The conventional two-stroke engine scavenging system primarily falls in the premixed category, where the premixed fuel-air mixture entering the cylinder through the transfer ports sweeps the burnt products out. Gas flow in the intake, transfer, and exhaust passages is unsteady and controlled by wave actions. There are several suggestions made in the literature to overcome the problem of gas oscillations in the intake system, such as fitting a branched pipe or resonator in the intake (Sawa et al. 1977, Hata et al. 1981a), tuning of intake, optimization of crankcase volume and port timings (Nagoa et al. 1967), use of boost ports (Wallace et al. 1969), and fluid diodes (Nagoa et al. 1971, Sher 1982). On the exhaust side, improvements in engine performance are achieved through positive exhaust flow control devices, such as variable-area exhaust port (Nomura et al. 1985, Sher et al. 1990), butterfly valve (Tsuchiya et al. 1980), and rotary valves (Hata et al. 1981b). By tuning the inlet and exhaust systems, significant benefits are realized over only a narrow range of engine operating conditions. Hence, the use of external devices is found to increase the mechanical complexity without a sufficient improvement in the fuel economy over the entire range of engine operation.

Scavenging with Stratified Charging Systems

Attempts have been made to scavenge the cylinder with air alone while still introducing the fuel by means of the carburetor. This process has been termed "scavenging by stratified charging" (Blair 1990). In an early attempt, Batoni (1978) developed a stratified charging concept in an opposed-piston two-stroke engine, showing considerable improvement in the fuel consumption and exhaust emissions with no loss in power or torque. Nevertheless, there is a considerable sacrifice of mechanical simplicity in this arrangement. Onishi et al. (1984) proposed a multilayer-stratified scavenging (MULS) system. In this method, mixture stratification is achieved by separating the mixture generated by the carburetor into a rich mixture and a lean mixture between the inlet manifold and the scavenging ports, and by finally controlling the scavenging flows. In the system developed by Hill et al. (1983), air is supplied to the crankcase while the rich air-fuel mixture from the carburetor is supplied to a long passage connecting the crankcase and the cylinder through a reed valve. In the arrangement of Ramesh et al. (1985), the primary circuit supplies the air-fuel mixture through the carburetor to the crankcase, while the secondary circuit supplies air alone to the upper ends of the transfer ducts through reed valves, allowing charge stratification. As is evident from these investigations, there is a trend towards development of relatively simple systems to carry out most of the scavenging, either with air alone or with a very lean fuel-air mixture to reduce the scavenged-through losses.

IMPROVEMENTS IN THE COMBUSTION PROCESS

The engine combustion process is always central to improvements in emissions and fuel economy. From a thermodynamic standpoint, it is difficult to achieve the constant volume requirement for better performance in two-stroke cycle operation due to the presence of large exhaust residual contents (Onishi et al. 1979). To achieve smoothness in engine operation and to compensate for the fresh charge loss, rich mixtures are utilized. However, this results in higher CO emissions and other combustion-related problems due to poor vaporization of the fuel. It is thought that the difficulties of two-stroke engine combustion can be reduced more effectively through stabilized combustion of lean mixtures. This objective is difficult to attain unless improvements in the mechanism of the conventional combustion process within two-stroke cycle engines are possible (Tsuchiya et al. 1983). This is an area where much effort is currently expended. In this respect, such concepts as catalytic combustion, thermal insulation of the combustion chamber, ignition enhancement, and high-octane fuel blends have received consideration.

Catalytic Combustion

The use of an in-cylinder catalyst for the improvement of combustion has been proposed by many investigators, mostly for diesel engines (Gaffney et al. 1980, Siegla et al. 1982). The objective of introducing a catalyst in the combustion chamber is to allow heterogeneous oxidation on the surface of the catalyst. Heterogeneously catalyzed gas-phase combustion was first demonstrated by Pfefferle (1978). A catalytic engine of Thring (1980) was one of the first to use a platinum wash coat wire mesh in both direct- and indirect-injection diesel engines. The potential improvement in the lean blow-out limits due to catalytic effects was established by Karim et al. (1986). Rychter et al. (1981) implemented this catalytic combustion technique in a lean-burn SI engine to activate the charge prior to ignition. Noble metal catalysts, such as platinum, have been extensively studied for the thermal and chemical enhancement of the gas-phase ignition chemistry for a variety of combustion-system applications (Bond 1974). Platinum, however, is an unlikely candidate for engine applications due to its high cost. Metal oxide catalysts typically have catalytic ignition temperatures significantly higher than those of noble metals, indicating the lower chemical activity of the former. However, the catalyst's activity is not necessarily a limiting factor under the expected mass-transport-limited conditions, so the effectiveness of metal oxide catalysts could be as great as that of noble metals. Among the metal oxides, nickel oxide, chromium oxide, manganese oxide, and vanadium oxide are potential catalysts for hydrocarbon oxidation (Blazowski et al. 1975, Lee et al. 1990).

Thermal Insulation of the Combustion Chamber

The concept of thermal insulation of the combustion chamber has been more widely investigated in diesel engines (Hoag et al. 1985, Kobori et al. 1992) than in SI engines (Assanis et al. 1987, 1990). The application of thermal insulation in SI engines has been hampered by the possibility of charge auto-ignition and knock or pre-ignition resulting from increased component temperatures. The approach favored by many investigators has been to apply thick, plasma-sprayed ceramic coatings or a very high degree of insulation by an air gap and/or nimonic materials (Woschini et al. 1987). Kamo et al. (1989) investigated the merits of thin ceramic coatings of the order of 0.5 mm (as against thick coatings of a few millimeters). Such coatings have been used satisfactorily in the gas turbine industry for the turbine blades, stator vanes, and combustor parts. Analytical studies (Assanis et al. 1987), including transient heat-transfer effects, suggest that the surface cyclic temperature swings around the mean temperature in the same manner for both thin and thick coatings. This suggestion opens the possibility of employing thin thermal barrier coatings for gasoline engine application. The works of Assanis et al. (1990, 1992) dealing with the use of thin ceramic coatings in SI engines are important, being among the few investigations available.

Two-stroke SI engines are normally supplied with rich mixtures (air-fuel ratio between 10.5 and 15.0) to compensate for the loss of fresh charge and exhaust dilution effects. This results in a heterogeneous mixture, and much of the fuel is carried to the engine in the form of large droplets, in particular at part-load and low-speed operating conditions. In such situations, a considerable quantity of heat is required to completely vaporize the fuel. External heating of the mixtures has been suggested (Hughes 1976) to improve the vaporization characteristics, but this affects the trapped charge density. The complexity of the systems is an added disadvantage with two-stroke engine applications. On the other hand, providing a hot combustion chamber will result in better vaporization of the rich mixtures. This hot combustion chamber can be obtained by providing moderate thermal insulation of the combustion chamber to retain the heat generated from combustion.

Ignition Enhancement

The magneto-coil ignition system is commonly used in two-stroke SI engines because of its simplicity and independence of a battery or generator. However, its major limitations are (i) decrease in available voltage as engine speed increases, due to limitations in the current switching capability of the breaker system, and (ii) less time available to build up the primary-coil stored energy. The breaker points are also subjected to both electrical and mechanical wear, which results in short maintenance intervals. As a result, the conventional magneto-coil ignition system is not effective if the engine is to operate at higher compression ratios and with leaner fuel-air mixtures. The techniques for achieving this include (Automotive Engineering 1989) use of (i) extended reach spark plug with wide gap, (ii) dual spark plugs, (iii) multi-point and multi-electrode spark plugs, (iv) thin-electrode, platinum-tipped electrode spark plug, and (v) high-energy spark discharge ignition system.

The influence of various types of spark plugs, the number of spark plugs, the ignition system, plug location, and ground electrode orientation are studied in detail by Anderson and Asik (1985, 1987). A multi-point spark ignition with several spark gaps in the combustion chamber is one of the most effective techniques for reducing the combustion duration and extending the lean misfire limit. Rado et al. (1976) showed that the three-gap spark plug improves the fuel economy and extends the lean misfire limit. Durbin and Tsai (1983) compared the ten-gap multiple-electrode spark plug (total gap width of 10.2 mm) with the conventional spark plug. Thin-electrode, platinum-tipped spark plugs have been found effective because of the reduced voltage required for ignition under certain electrode polarity conditions and their larger plug gap (Nakamura et al. 1983). Plasma-jet ignition systems are being investigated (Kupe et al. 1987) to provide higher temperatures with both an increased formation of free radicals and a larger initially burned mixture volume to ignite very lean fuel-air mixtures. It is reported that a plasma jet with ignition energy as high as 14 times that of a normal spark has extended lean operation air-fuel ratio limits, by about 24%. These systems have the disadvantage of higher power requirements, drawing from the engine to provide the required electrical power for the ignition system. Ignition enhancement techniques, therefore, become attractive for exploitation with lean fuel-air mixtures and at higher compression ratios.

High-Octane Fuel Blends

Present-day gasoline fuel as such cannot obtain the potential advantages of lean combustion because of the rapid decrease in flame speed with leaner mixtures and its narrow flammability limits (Hansel 1971). Methanol has been found to be a more suitable fuel, with relatively good lean-combustion characteristics, compared to gasoline and other hydrocarbon fuels. There has been sufficient evidence (Most et al. 1975, Katoh et al. 1986) to show that methanol has wider flammability limits and higher flame speeds under lean conditions than does gasoline. Nevertheless, many technical difficulties associated with the use of methanol for commercial application in vehicles are yet to be overcome. Until then, the use of gasoline-methanol blends is one alternative available for immediate use. Such blends have improved octane quality, lean limits, knock-limited compression ratio, thermal efficiency, power output, and NO_x emissions. However, some undesirable effects result from blending methanol into gasoline, namely phase-separation, increased vapor pressure, and emissions of unburned methanol, particularly at higher methanol concentrations (Wigg 1974).

Besides methanol and ethanol, some reports (Takeda et al. 1980, 1984) suggest that eucalyptus oil and orange oil possess high octane values and are also good potential alternative fuels for SI engines. Eucalyptus oil is extracted from eucalyptus leaves. The main ingredient is 1.8 cineol ($C_{10}H_{18}O$). Orange oil can be extracted from the peel of orange fruits. The main ingredient of orange oil is d-limonene ($C_{10}H_{16}$). The high octane values of eucalyptus and orange oils increase the octane value of the blend when added to low-octane gasoline. Their physical and chemical properties are similar, and they are easily miscible with gasoline. Eucalyptus oil is also an effective co-solvent to minimize the phase-separation problems of alcohol-gasoline blends.

OVERVIEW OF THE PREVIOUS WORK

The results obtained during the first two phases of the described work, including development of a new scavenging system, various in-cylinder catalysts, moderate thermal insulation of the combustion chamber, high-energy ignition system, and different high-octane fuel blends, are presented in the references (Poola et al. 1993, 1992, 1994a, 1994b, 1994c, respectively). A brief summary of the research efforts and key findings is provided in the following sections.

Development of a New Scavenging System

This system is based on the principle of scavenging with a stratified charging process. In the improved design, two reed valves are fitted at the upper ends of the transfer ducts connecting the crankcase and the cylinder, as shown in Figure 1. Atmospheric air enters the crankcase through the carburetor along with the fuel in the conventional way, as well as through the two reed valves fitted at the transfer ports. When the piston moves from BDC to TDC, the pressure inside the crankcase is below atmospheric pressure and air from the atmosphere gets into the crankcase through the above-mentioned three paths. When the air flows through the reed valves, the fuel-air mixture present in the transfer ducts as a result of the previous cycle is forced into the crankcase and air takes its place. This process results in either a partial or a complete filling up of the transfer ducts by pure air. During the downward stroke of the piston, pressure builds up in the crankcase and in the transfer ducts as soon as the inlet ports and the intake to the crankcase are closed. Hence, the reed valves are also closed. When the piston descends further, the transfer ports open and the pure air or very lean fuel-air in the transfer ducts enters the cylinder first. This air pushes the exhaust gases out and becomes the main component to be short-circuited into the atmosphere through the exhaust port. The fuel-air mixture, which follows the air, is retained inside the cylinder to a larger extent. As a result, the amount of fresh charge loss during the initial scavenging period is reduced; hence, improved fuel economy and lower hydrocarbon emissions could be obtained.

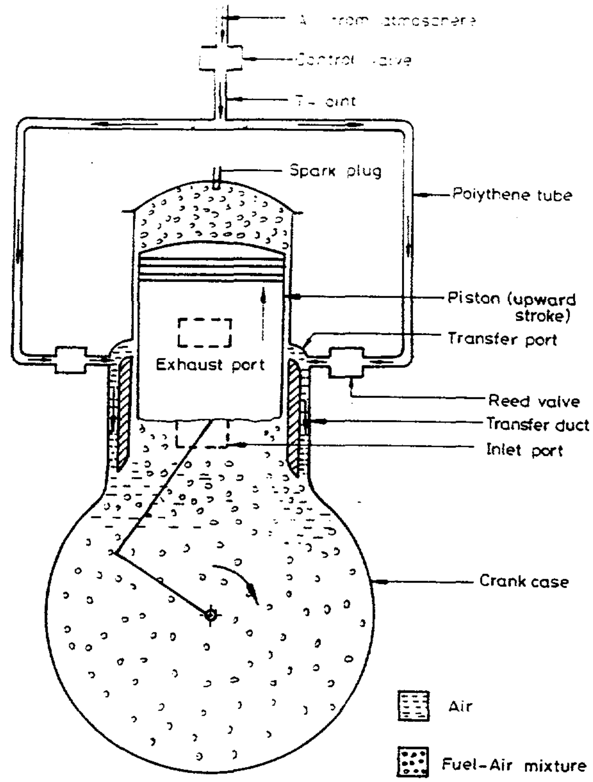

Fig. 1 Schematic diagram of the position of the extra reed valves at the transfer ducts

Experimental investigations were carried out to optimize this new scavenging system by using both atmospheric air and compressed air through the extra reed valves. The effects of engine capacity, load, speed, area of opening of the reed valve chamber, and secondary air flow rate through the reed valves were investigated in detail. The amount of secondary air entering through the reed valves should be such that it displaces more exhaust gases to reduce fresh charge loss, and while doing so, it should have minimal effects on the mixture quality of the trapped charge in the cylinder. It has been shown (Poola et al. 1993) that for large (250-cc) and medium (150-cc) capacity engines, a higher secondary air flow rate, achieved by regulating compressed air flow (part of the cooling air from the blower of the engine), is desirable. For example, the percentage improvement in brake thermal efficiency increases from 7.2 to 16.96% (at 2.8 kW, 3000 rpm), when reed valves permit atmospheric air and optimum compressed air, respectively, compared to the normal engine. For a very small capacity (55-cc) engine, there must be a control on the quantity of secondary air from the atmosphere itself in order to obtain the maximum benefits with regard to fuel economy and HC emissions. The HC emissions in the exhaust arise due to mixture short-circuiting, combustion of fuel-lubricant blend, trapping of unburned charge in the crevices, and flame-quenching. However, a major contribution of exhaust hydrocarbons comes from short-circuiting during the scavenging period of the engine cycle (Poola et al. 1994d). Reduction in fresh charge loss will directly influence the HC emissions. Hence, maximum reduction in HC emissions was obtained at optimum secondary air flow rate through the reed valves. When secondary air gets inside the transfer ducts through the reed valves, it not only fills the transfer ducts, but it also mixes with the fresh mixture in the crankcase and makes the mixture lean to a certain extent. Also, when the secondary air participates in scavenging, the amount of exhaust residual content decreases and hence, a larger amount of fresh mixture fills the cylinder after exhaust-port closure. As a result, combustion is more complete, and lower CO emissions could be obtained. The reduction in CO emissions obtained is in the range of 0.5 to 4.5% by volume, depending on the operating conditions and secondary air flow rate through the reed valves.

Use of In-Cylinder Catalysts

In order to investigate the application of the concept of catalytic combustion, various catalysts, such as copper, chromium, and nickel, were coated on the combustion chamber wall for determining their effects on engine performance, combustion, and emissions characteristics. The effects of lean fuel-air mixtures and higher compression ratios were studied with the best catalyst among all the catalysts tested. These catalysts were deposited on the piston top and cylinder head surface, using a standard electroplating process. Copper was coated by using a cyanide copper bath (Poola 1993), which produces a coating of porous nature and a fine deposition of metal on the surface. The above catalytic coatings were not tested for durability. Since the catalysts are present in the form of thin coatings (about 20-60 microns), both their durability and the effect of carbon deposits on the catalyst surface are likely to be questioned. Nevertheless, this work was aimed solely at finding the relative effectiveness of various in-cylinder catalysts for engine applications.

It has been shown (Poola et al. 1992) that coating the combustion chamber wall with such catalysts as copper, chromium, and nickel has yielded definite improvement in the fuel economy and reduction in CO emissions. These improvements were expected due to better combustion as a result of catalytic surface pre-reactions in the fuel-air charge prior to ignition. It is postulated that catalyst on the surface of the combustion chamber helps to enhance the gas-phase chemical activity prior to ignition by releasing intermediate species, as well as by thermal activation of the adjacent combustible mixture (Rychter et al. 1981), thus leading to better combustion. The early rise in heat release during the delay period and higher heat release rates during combustion (Poola 1993) provides ample evidence of the catalytic pre-flame reactions with hydrocarbon mixtures. It was found that catalytic activation is more effective with lean fuel-air mixtures and at higher compression ratios. Among the different catalysts investigated, copper was found very effective in reducing both HC and CO emissions, and brake thermal efficiency was also improved. At a high CR of 9:1 and with a lean mixture (A/F=15.7), copper catalyst increases the absolute brake thermal efficiency from 17.7% to 22.8%, decreases HC emissions from 3200 to 2300 ppm, and lowers CO emissions from 3.6 to 0.25% by volume when compared to the normal engine (CR=7.4, A/F=13.2) at 2 kW, 3000 rpm. Ignition delay was lower, combustion duration was shorter, and cylinder peak pressures were higher with the copper catalyst at higher compression ratios and with leaner fuel-air mixtures. Knock-limited power output also increased by about 12% at a high CR of 9:1 in the presence of copper catalyst.

Moderate Thermal Insulation of the Combustion Chamber

Moderate thermal insulation of the combustion chamber could be achieved by fixing an insert made of a low-thermal-conductivity material or by depositing a low-thermal-conductivity ceramic material on the standard aluminum-silicon alloy combustion chamber components. The insulation material should preferably have a coefficient of thermal expansion close to that of the parent components, so that thermal stresses in the interface for a given temperature increase will be minimal. With such considerations, moderate thermal insulation of the combustion chamber was provided by the following two methods:

1. Composite piston and composite cylinder head
2. Thin ceramic coating on the combustion chamber walls

Niresist, a form of cast iron with high nickel content, was selected to make the composite components. The construction of the composite piston and composite cylinder head are shown in Figure 2. Niresist material has a thermal expansion coefficient of the same order as that of aluminum alloy, and it has a much lower thermal conductivity than aluminum. Among the various ceramic materials, partially stabilized zirconia (PSZ) has excellent toughness, hot strength, thermal shock resistance, and low thermal conductivity. It has been widely used as a thermal barrier coating in the combustion chambers of diesel engines. The combustion chamber surface was coated with the PSZ ceramic material to a 0.5-mm thickness, by using plasma

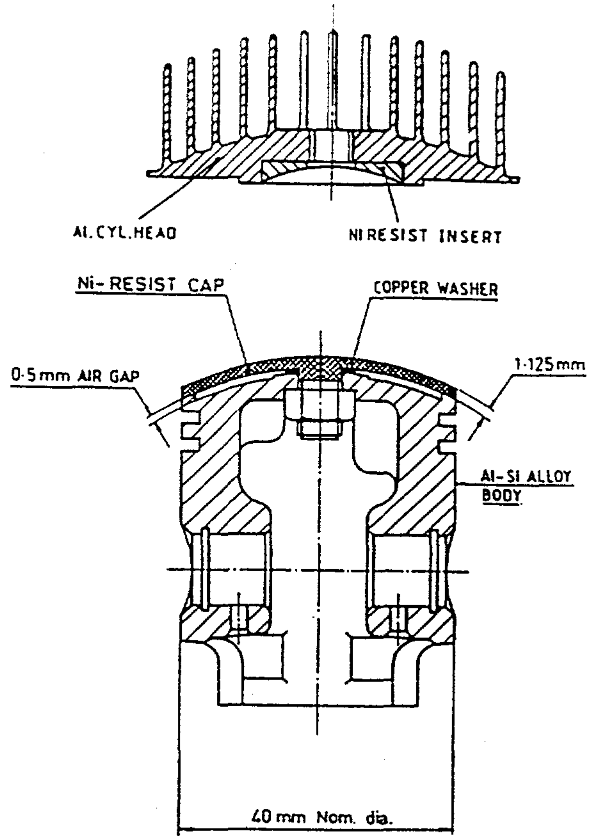

Fig. 2 Piston top and cylinder head insulated by niresist inserts

Table 1. Specifications of the test engines

Specifications	Engine 1	Engine 2
Make	BAJAJ	TVS
No. of cylinders	One	One
Bore (mm)	57	41
Stroke (mm)	57	42
Displacement volume (cc)	145.55	55.45
Compression ratio	7.4:1	8:1
Rated power (kW)	4.2 @5200 rpm	1.7 @5000 rpm
Cooling medium	Air-cooled	Air-cooled
Mixture induction	Rotary disc system	Reed valve
Scavenging system	Loop (Schnurle)	Loop scavenging
Ignition system	Magneto-coil	CDI

spraying technique. These two types of insulated combustion chambers were tested in a single-cylinder, two-stroke SI engine of 55-cc displacement volume. The details of this Engine-2 are listed in Table 1. The concept of moderate thermal insulation of the combustion chamber will also be well-suited to such high-latent-heat fuels as methanol, which suffers from the disadvantages of poor vaporization characteristics and associated combustion problems at part-load operation. Experiments were carried out with the above two types of moderate thermal insulation of the combustion chamber, using both gasoline and methanol as fuels.

Test results using gasoline or methanol as fuels (Poola et al. 1994a) indicate that CO emissions are significantly reduced, by about 3 to 4% by volume, with the thin ceramic-coated combustion chamber because of the relatively leaner fuel-air operation. The maximum percentage improvement in brake thermal efficiency, 15.4%, is obtained at part load (0.8 kW, 3000 rpm) with the composite components (compared to a normal engine using gasoline fuel). With the insulated combustion chamber, fuel vaporization is expected to be better, resulting in better combustion, lower combustion heat-transfer losses, fewer gas-to-wall heat interchanges, and lower flame-quenching effects. Between the two types of insulated combustion chambers tested, the fuel economy improvement of the ceramic-coated combustion chamber is less significant than that for the composite-component combustion chamber, in particular at higher outputs and at higher speeds, due to mild knock and loss of brake power. In comparison, the thin-ceramic-coated combustion chamber is better suited for high latent fuels, such as methanol, to obtain the potential benefits with regard to engine thermal efficiency, CO emissions, lean combustion capability, and knock-free operation compared to gasoline. Durability of ceramic coatings is less satisfactory than that of composite components. The advantage of the ceramic-coated combustion chamber, however, is that it is easier to coat ceramic in the combustion chamber than to make composite components. A limited, 50-hour endurance test revealed that there was no appreciable wear or deposit formation for both types of insulation.

Development of a High-Energy Ignition System

A compact, breakerless, high-energy electronic ignition system with variable spark timing has been developed and tested along with a platinum-tipped-electrode spark plug. A digital electronic, clock pulse controller circuit was designed and built to capture nearly 100 consecutive cylinder peak pressures with the available digital data-acquisition system (DDAS) (Iwatsu Signal Analyzer), in order to examine the cyclic variations. The coefficient of variation (COV) in peak pressure was calculated based on the approach suggested by Amann (1985) to quantify the combustion stability. Because of the limited memory (8 kB), recording, and processing capabilities of the DDAS employed, the combustion stability could be measured in terms of COV of peak pressure rather than the more common method of COV of indicated mean effective pressure (IMEP). The engine performance, lean combustion capability, cyclic variations, and exhaust emissions were determined with this system at higher compression ratios and with leaner fuel-air mixtures.

With the high energy ignition system, combustion stability is improved due to high spark discharge, which initiates the combustion process more effectively at higher compression ratios and with leaner fuel-air mixtures compared to the normal ignition system. A comparison of consecutive cylinder peak pressure-time traces is presented in Figure 3 for both normal and high-energy ignition systems, at an engine speed of 2000 rpm. The lower the COV of peak pressure, the higher the combustion stability is and the better the thermal efficiency, in particular at part loads and low-to-medium speed range, with the high-energy ignition system, where the conditions (such as high exhaust residuals, slower flame propagation velocities, and more gas-to-wall heat interactions) are usually unfavorable to ignite lean mixtures with a normal ignition system. Figure 3 shows that the COV of peak pressure decreases from 0.164 to 0.06 at 1.38 kW, 2000 rpm, due to the high-energy ignition system. At a high CR of 9:1, with lean fuel-air mixture (A/F=15.2), the maximum improvement in brake thermal efficiency obtained with this high-energy ignition system is 16.5% (at 2.76 kW, 3000 rpm) compared to a magneto-coil ignition system.

The Influence of High-Octane Fuel Blends

The high-octane fuels, such as orange oil, eucalyptus oil, methanol, and ethanol, were blended separately with gasoline in the proportion of 20% by volume, and their fuel economy, emissions, and combustion characteristics were evaluated at higher compression ratios. The combined proportions (by volume) of the four fuel blends tested were as follows:

1. 20% Orange oil (Or) + 80% Gasoline (G) + 3.5% SAE30
2. 20% Eucalyptus oil (Eu) + 80% Gasoline + 3.5% SAE30
3. 20% Methanol (M) + 75% Gasoline + 5% Eucalyptus oil + 2.8% SAE30 + 0.7% Castor oil
4. 20% Ethanol (E) + 20% Eucalyptus oil + 60% Gasoline + 2.8% SAE30 + 0.7% Castor oil

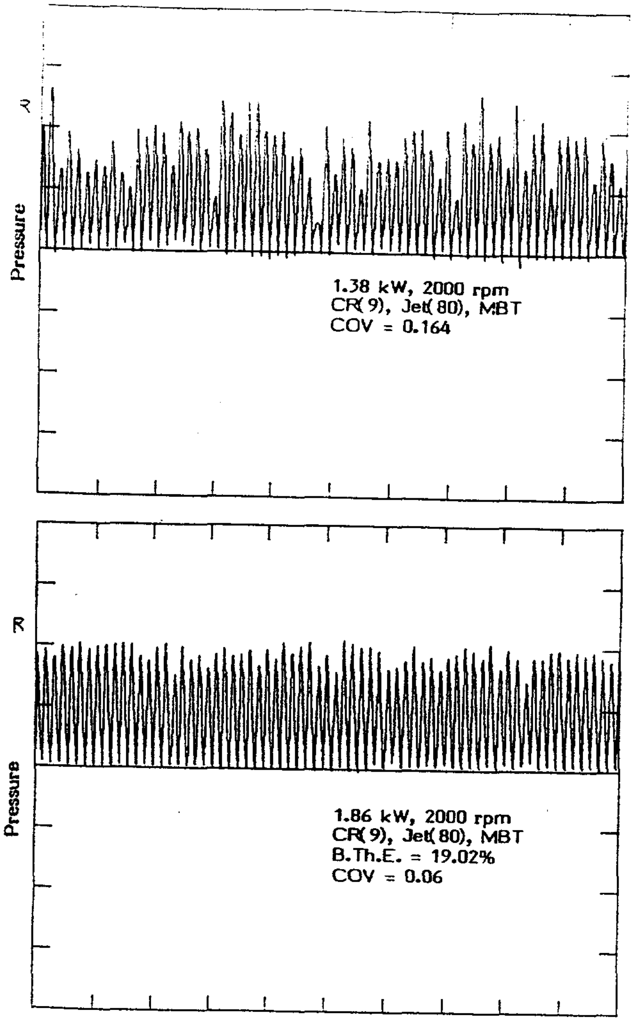

Fig.3 Cycle-by-cycle variation of the cylinder peak pressure-time traces for normal and high-energy ignition systems

Castor oil was used as a lubricant in the case of methanol/ethanol blends, and standard SAE 30 oil was employed in the case of eucalyptus oil and orange oil. Eucalyptus oil has been considered as an additive to lower the phase separation problems associated with the alcohol-gasoline blends. It was found that for blends of 20% methanol and 20% ethanol with gasoline, the percent volume of eucalyptus oil required to prevent phase separation was about 5% and 20% by volume, respectively, at room temperature (28°C).

Based on the test results (Poola et al. 1994c), high-octane fuels, such as methanol, ethanol, eucalyptus oil, and orange oil, can be blended at about 20% by volume with low-octane commercial gasoline to increase the octane quality of the fuel blend, thereby increasing the knock limit at higher compression ratios. The extension of knock limit at a high CR of 9:1 with various fuel blends was in the following descending order: methanol blend-eucalyptus oil blend-ethanol blend-orange oil blend. Among all the fuel blends tested, methanol blend and eucalyptus oil blend provide a potential for higher brake thermal efficiencies concomitant with lower exhaust emissions, extension of knock limit, and knock-limited power output at higher compression ratios; these blends are worthy of considering for two-stroke SI engines.

Comparisons were made at two typical operating conditions to examine the effect of different modifications incorporated during phases 1 and 2 of the investigations with regard to brake thermal efficiency, HC and CO emissions, and combustion parameters. Figures 4 to 9 illustrate these comparisons. Figure 4 indicates that the new scavenging system and the devices incorporated to improve the combustion process increase the brake thermal efficiency more or less in the same proportion compared to the normal engine. Although this result cannot be considered to imply these benefits over the entire range of engine operation, similar improvements are evident for a particular range of engine operation. This might help in considering a suitable device for a particular application. With respect to CO emissions, there is considerable variation among the different modifications incorporated, as illustrated in Figure 5. It can be seen that the thin ceramic-coated combustion chamber with methanol fuel produces the least CO emissions compared to the normal engine and other methods tested. Tests with the thin ceramic-coated combustion chamber (Poola et al. 1994a) also indicate that CO emissions are less than 0.5% by volume over the entire load range at 3000 and 4000 rpm for both gasoline and methanol as fuels. The reduction in CO emissions can be correlated with the extension of lean mixture operation with each of these modifications. However, in general the CO emissions are reduced by about 30 to 50% with all the modifications possibly due to improved combustion as well as lean mixture operation. The variation in HC emissions with different modifications is depicted in Figure 6. This figure shows that the high-octane fuel blend (80%G+20%Eu) had the lowest HC emissions compared to other modifications. It is difficult to interpret the HC emissions data because they are evolved from different sources. For example, with the new scavenging system, HC emissions are reduced as a result of lower fresh charge losses. On the other hand, a thin ceramic-coated combustion chamber helps to lower flame quenching effects and permits lean fuel operation, and lower HC emissions are obtained as a result. All the measures incorporated produced lower HC emissions compared to the normal engine. However, quantitative differences exist because of the differences in the combustion mechanism associated with these devices. Figures 7 through 9 demonstrate the combustion characteristics (peak cylinder pressure, ignition delay, and combustion duration) with

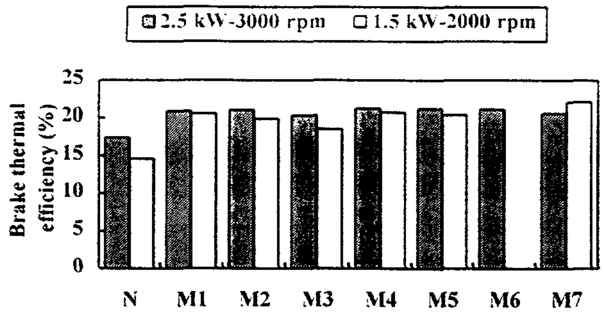

Fig. 4 Comparison of brake thermal efficiency with different modifications

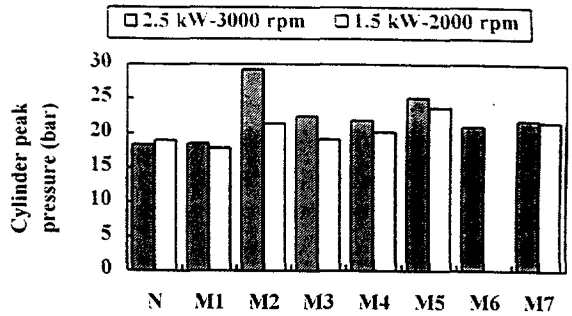

Fig. 7 Comparison of cylinder peak pressures for different modifications

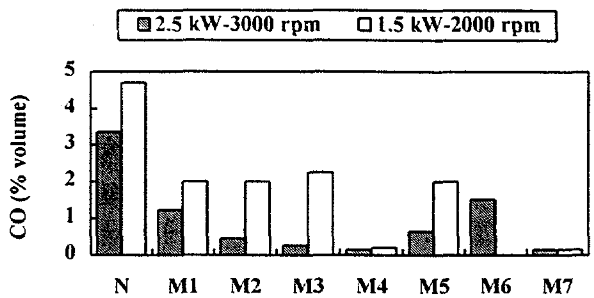

Fig. 5 Comparison of carbon monoxide emissions for different modifications

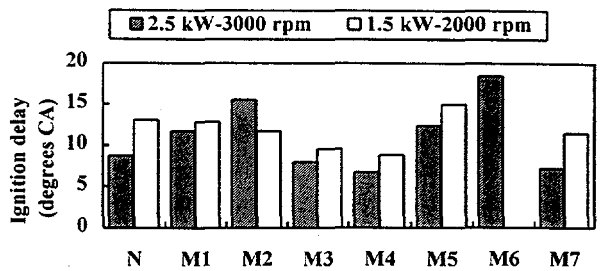

Fig. 8 Comparison of ignition delay characterstics for different modifications

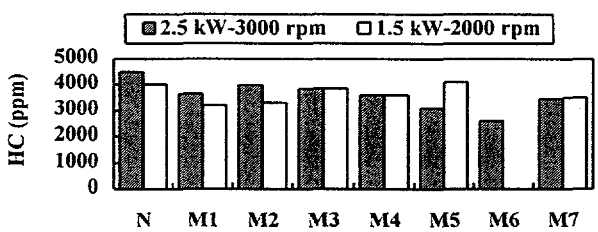

Fig. 6 Comparison of hydrocarbon emissions with different modifications

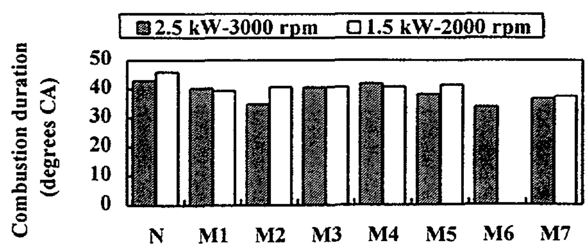

Fig. 9 Comparison of combustion duration for different modifications

N: Normal engine; gasoline fuel; CR=7.47
M1: New scavenging system with optimum secondary air-flow rate; gasoline fuel; CR=7.47
M2: Copper coated combustion chamber; lean fuel jet (0.8-mm); gasoline fuel; CR=9
M3: Thin-ceramic coated combustion chamber; gasoline fuel; CR=7.47
M4: Thin-ceramic-coated combustion chamber; methanol fuel; CR=7.47
M5: High-energy ignition system; platinum-tipped electrode spark plug; lean fuel jet; gasoline fuel; CR=9
M6: High-octane fuel blend(80%G+20%Eu); fuel jet 0.84-mm; CR=9
M7: Optimal design; high octane fuel blend; fuel jet size 0.84-mm; CR=9

Table 2. A Comparison of two configurations of the optimal design with the normal engine

Component	Normal engine	Optimal two-stroke SI engine	
		Configuration 1	Configuration 2
Piston	Standard	Copper coated	Niresist insert with copper-coated
Cylinder head	Standard	Copper coated	Copper-coated
Compression ratio	7.47:1	9:1	7.47:1
Fuel jet size	0.84 mm	0.80 mm	0.84 mm
Ignition system	Magento-coil	High-energy ignition	High-energy ignition
Contact breaker points	Yes	No	No
Ignition timing	Fixed (21 bTDC)	MBT	MBT
Reed valves at the transfer ducts	No	Yes	Yes
Fuel	Gasoline	Gasoline/Methanol/80G+20Eu	Gasoline/Methanol
Spark plug	Standard	Platinum-tipped electrode	Platinum-tipped electrode

different modifications. These comparisons, in turn, indicate the in-cylinder combustion behavior of each of the modifications. All the devices tended to produce higher cylinder peak pressures and relatively shorter combustion duration. It can be seen that the high-octane fuel blend had the highest ignition delay, but the corresponding combustion duration was lower and peak cylinder pressure was higher when compared to other modifications. On the other hand, the thin ceramic-coated combustion chamber with methanol fuel produced the lowest ignition delay among all the modifications. These comparisons may help to understand the differences in combustion characteristics associated with the various modifications tested.

PRESENT WORK

In the present work (phase 3), an optimal design for the two-stroke SI engine is being developed by incorporating the various methods to improve scavenging and combustion processes. Various modifications to the two-stroke SI engine and use of newer concepts have resulted in widely varying improvements in the fuel economy and exhaust emissions. The optimal design can be evolved by incorporating only those modifications that give the maximum benefits with regard to brake thermal efficiency and exhaust emissions from the various methods investigated. Two different configurations were formulated to evaluate the optimal design. The proposed modifications in each of the configurations, along with the normal engine, are compared in Table 2. A comparative study of these two configurations was carried out to evaluate the optimal design with regard to fuel economy and exhaust emissions. In order to examine the influence of fuel variations with this optimal design, experiments were carried out using a high-octane fuel blend (80% gasoline + 20% eucalyptus oil), pure methanol, and gasoline as fuels; comparisons have been made with the normal gasoline-fueled engine. The results obtained from these tests are discussed in detail in this paper.

EXPERIMENTAL PROGRAM

A single-cylinder, air-cooled, loop scavenged, carbureted, two-stroke SI engine, the details of which are given (as Engine 1) in Table 1, was used for the experimental study. This engine was coupled to an eddy-current dynamometer for speed and torque measurements. Calibrated, standard instrumentation was used to measure fuel and air flow (turbine-type) rates. The cylinder pressure was measured by a flush-mounted piezo-electric pressure transducer (Kistler). The cylinder pressure, crank angle, and spark signals were fed to the DDAS for storage and subsequent analysis of various combustion parameters. HC (n-hexane equivalent) and CO emissions were measured on a volume basis by an infrared gas analyzer (Horiba, Mexa 324FB). Since the engine exhaust consists of a large percentage of unburned hydrocarbons, care was taken to prevent any condensation losses by providing a heater on the sample line to the exhaust analyzer. HC emissions data are obtained using the NDIR technique, since this was the only equipment available for the measurements in the author's laboratory. It is currently recognized that for an engine operating on gasoline/methanol, the FID technique is more appropriate for measuring total hydrocarbons. Many researchers (Shynichi et al. 1971, Nuti et al. 1985) have suggested the use of multipliers, in the range of 1.7 to 2.3, to account for the partial measurements from the NDIR technique. However, the exact multiplier should ideally be derived from both the methods for a given set of instruments and the range of HC concentrations. In the absence of such a facility, the exact value of the multiplier for representing the total hydrocarbons is difficult to determine. The present approach is considered reasonable for demonstrating the qualitative trends of the HC emission levels for both gasoline and methanol fuels for all the experiments.

The lubricating oil used was SAE 30, mixed in the proportion by volume of 35 cc to 1000 cc of gasoline. Carburetor jet sizes having diameters of 0.98 to 0.80 mm were available for tests. However, a jet size of 0.84 mm was employed as a standard for the normal engine, based on the exhaust emission levels and air-fuel ratio (10.5 to 14.0). A fuel jet size of 0.8 mm (leanest), employed for lean-burn experiments, gave an air-fuel ratio on the order of 13.0 to 17.5, depending on the operating conditions. The range of air-fuel ratios obtained with different fuel jet sizes and at various operating conditions tested is presented in Figure 10. For methanol fuel operation, a jet size of 1.4 mm was employed to deliver the same output as that of the gasoline fuel. Castor oil, 3.5% by volume, was added to the methanol fuel for lubrication. Experiments were conducted with variable loads at constant engine speeds of 2000 and 3000 rpm, with commercial gasoline as a fuel. Throughout the experiments, the ignition timing was maintained at minimum spark advance for best torque (MBT) settings.

Heat-Release Analysis

A simple method of heat-release analysis suitable to the available DDAS was developed in this work. The present method of heat-release analysis involves the computation of the polytropic indices of compression and expansion for obtaining the rate of heat release, assuming the cylinder charge to be an ideal gas. Applying the first law of thermodynamics to the cylinder contents, the heat release (Q_c) can be expressed as:

$$\Delta Q_c = P \, \Delta V \left(\frac{\gamma}{\gamma-1}\right) + \left(\frac{1}{\gamma-1}\right) V \, \Delta P + \Delta Q_h \quad (1)$$

Where
- γ = ratio of specific heats
- V = cylinder volume
- P = cylinder pressure
- Q_h = heat transferred

If γ is replaced by a polytropic exponent k, then the heat-transfer effect also will be included in the above equation. Equation 1 can be rewritten to represent the heat-release rate as follows:

$$\frac{\Delta Q_c}{\Delta\theta} = \left(\frac{k}{k-1}\right) P \frac{\Delta V}{\Delta\theta} + \left(\frac{1}{k-1}\right) V \frac{\Delta P}{\Delta\theta} \quad (2)$$

where θ = crank angle.

The instantaneous value of k was obtained by applying the following equation for polytropic compression or expansion between successive points:

$$P_1 V_1^K = P_2 V_2^K \quad (3)$$

which gives

$$k = \frac{\ln(P_1/P_2)}{\ln(V_2/V_1)} \quad (4)$$

Hence, knowing the instantaneous cylinder pressure and corresponding cylinder volume, the heat-release rate can be computed.

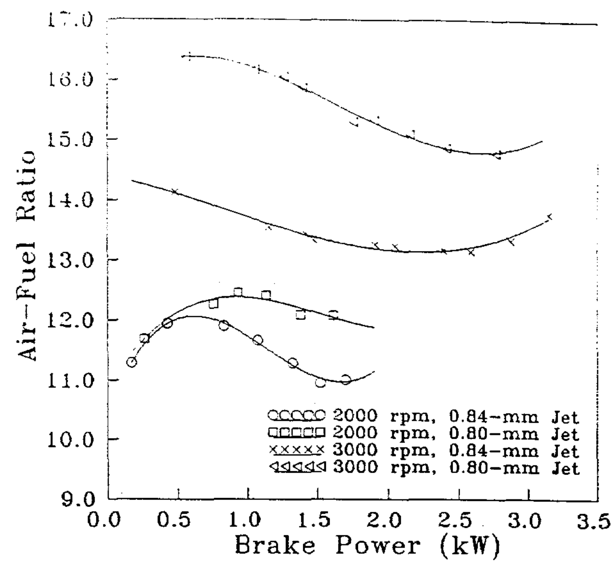

Fig. 10 Variation of air-fuel ratio with brake power for different fuel jet sizes

RESULTS AND DISCUSSION

The test results obtained with the two configurations evolved for the development of an optimal design are compared in this section. Figure 11 shows the variation of brake thermal efficiency and HC and CO emissions with brake power at a constant engine speed of 3000 rpm for the two different configurations. Gasoline fuel was employed in both the cases. It can be seen from Figure 11 that the brake thermal efficiency improves for both the configurations compared to the normal engine. These improvements are significant in the entire load range for configuration 1 and over the part-load to medium-load range for configuration 2. At the wide-open-throttle condition, configuration 1 increases the absolute brake thermal efficiency from 16.9 to 18.6%, whereas the efficiency for configuration 2 drops down to that of the normal engine. The deviation between the two configurations at higher outputs is possibly attributable to the combustion chamber surface temperatures and the increased weight of the piston. For both configurations, the improvement in the brake thermal efficiency is significant up to medium-load range and is attributed to the lower fresh charge losses during scavenging and improved combustion. At higher outputs, thermal insulation of the combustion chamber (configuration 2) leads to a drop in the trapped charge density, due to the higher temperature of the combustion chamber surface; this might counteract brake thermal efficiency improvements with configuration 2.

From the emissions point of view, it is observed that configuration 1 reduces the HC emissions consistently in the entire load range, whereas configuration 2 shows a marginal reduction compared to the normal engine. The maximum reduction in HC emissions obtained with configuration 1 is

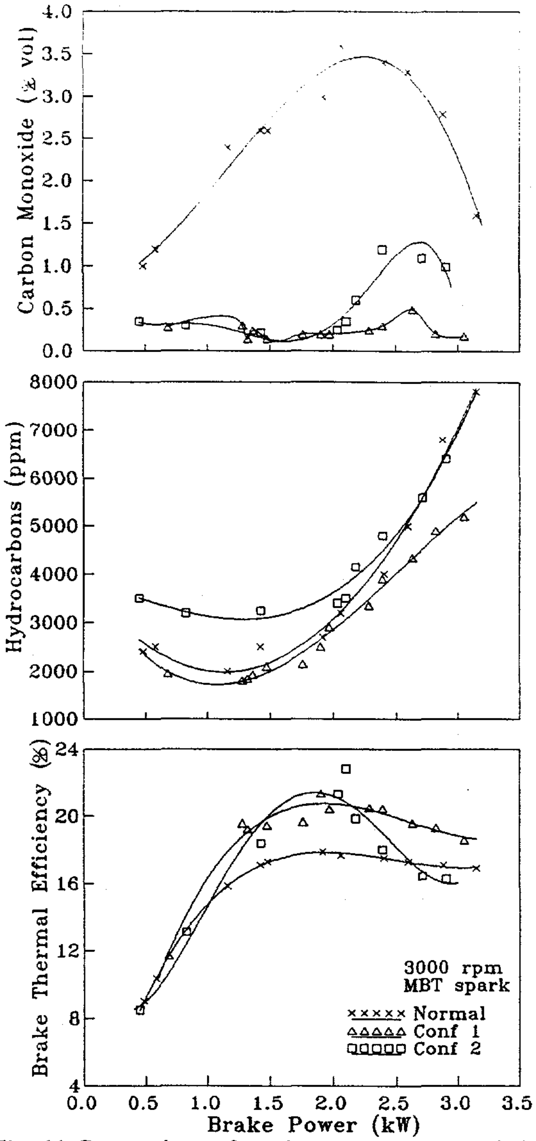

Fig. 11 Comparison of performance characteristics for the two configurations of the optimized design

about 2800 ppm (at 3 kW) compared to the normal engine. CO emissions are considerably lower with both configurations over the entire load range compared to the normal engine. The maximum reduction in CO emission is 3.2% and 2.2% by volume (at 2.4 kW) with configuration 1 and configuration 2, respectively. At higher outputs, CO emissions are relatively higher with configuration 2 when compared with configuration 1, possibly due to dissociation effects at elevated combustion chamber temperatures.

The cylinder peak pressure, ignition delay, and combustion duration characteristics for both configurations of the optimal design at a constant engine speed of 3000 rpm are depicted in Figure 12. Configuration 1 produces consistently higher cylinder peak pressures, lower ignition delay, and combustion duration over the entire load range compared to the normal engine. On the other hand, configuration 2 shows marginal benefits in combustion characteristics over the normal engine. Similar differences in brake thermal efficiency, emissions, and combustion characteristics were noticed at other speeds tested. From these test results, it is clear that configuration 1 is better than configuration 2 with regard to brake thermal efficiency and HC and CO emissions. Henceforth, configuration 1 will be referred to as the "optimal design" of the two-stroke SI engine based on the present studies. The results obtained with this optimal-design engine are discussed below.

The variation in brake thermal efficiency and exhaust emissions characteristics of the optimized two-stroke SI engine at constant engine speeds of 2000 and 3000 rpm, respectively, are presented in Figures 13 and 14. In order to examine the effect of fuel variations, a high-octane fuel blend (80%G+20%Eu), methanol, and gasoline are employed as fuels in the optimal design and comparisons are made with respect to the normal engine operating on gasoline fuel. It is observed that there is a significant improvement in the brake thermal efficiency over the entire test range of engine operation for the optimized design with all three fuels when compared to the normal gasoline-fueled engine. For example, brake thermal efficiency of the optimized design increases from 14.6 to 22% for both the high-octane fuel blend and methanol fuel, and from 14.6 to 20.7% with the gasoline fuel at 1.5 kW, 2000 rpm, when compared to the normal engine. These substantial improvements in the brake thermal efficiency are obtained with the optimized design as a result of such combined effects as reduction in fresh charge loss during the scavenging process with the new scavenging system, chemical activation of the charge before ignition due to catalytic prereactions, better flame initiation due to high-energy ignition with a wide, platinum-tipped-electrode spark gap, and high octane quality of the fuel blend/methanol fuel. Improvements are also obtained by virtue of efficient lean fuel-air operation and by utilizing the high octane value of methanol/high-octane fuel blend at a high CR of 9:1. There is no drop in power output or knock with the optimized design due to the 9:1 CR, compared to normal engine operation at a CR of 7.4:1. The increase in CR with the optimized design directly influences the gain in cycle efficiency. The effect of increasing CR with lean mixtures permits a more effective utilization of available energy, and reduced residual exhaust gases, leading to higher flame speeds and higher expansion rates (Marsee et al. 1977). Lean mixtures permit the use of higher compression ratios without knock. However, the increase in CR is often limited by the extent of lean mixture operation without misfiring, HC emissions, frictional and heat-transfer losses, and octane requirements of the fuel. The problems associated with higher compression ratios can be reduced to a certain extent

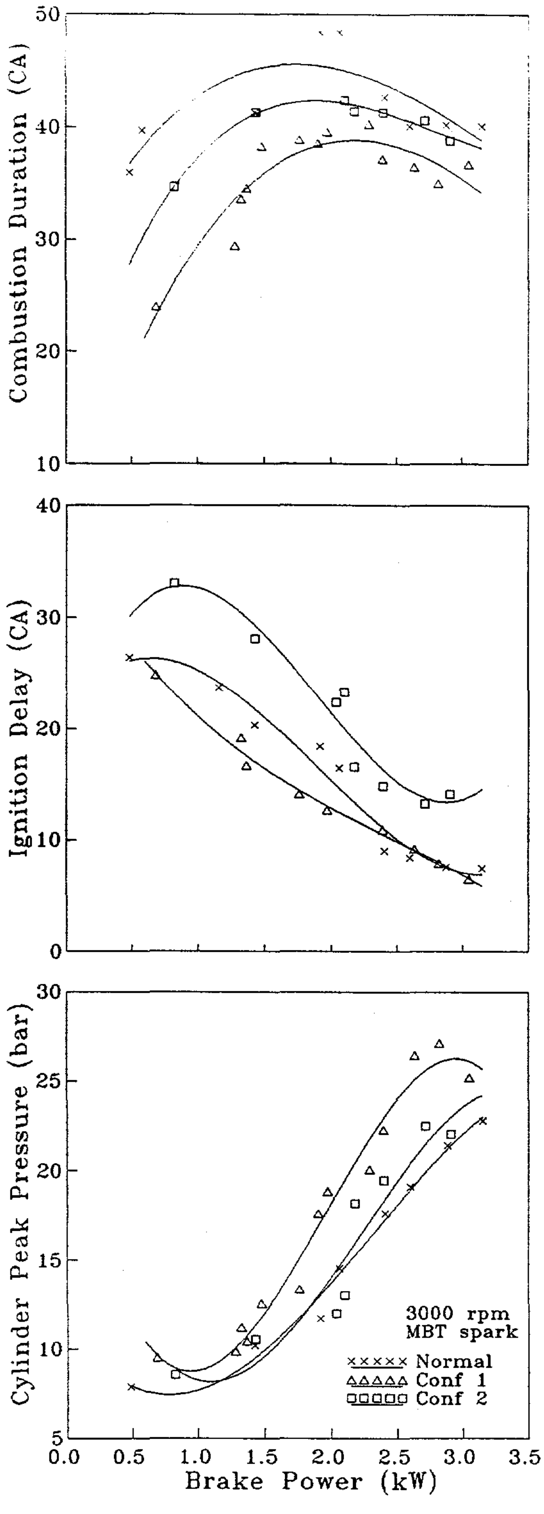

Fig. 12 Comparison of combustion characteristics for the two configurations of the optimized design

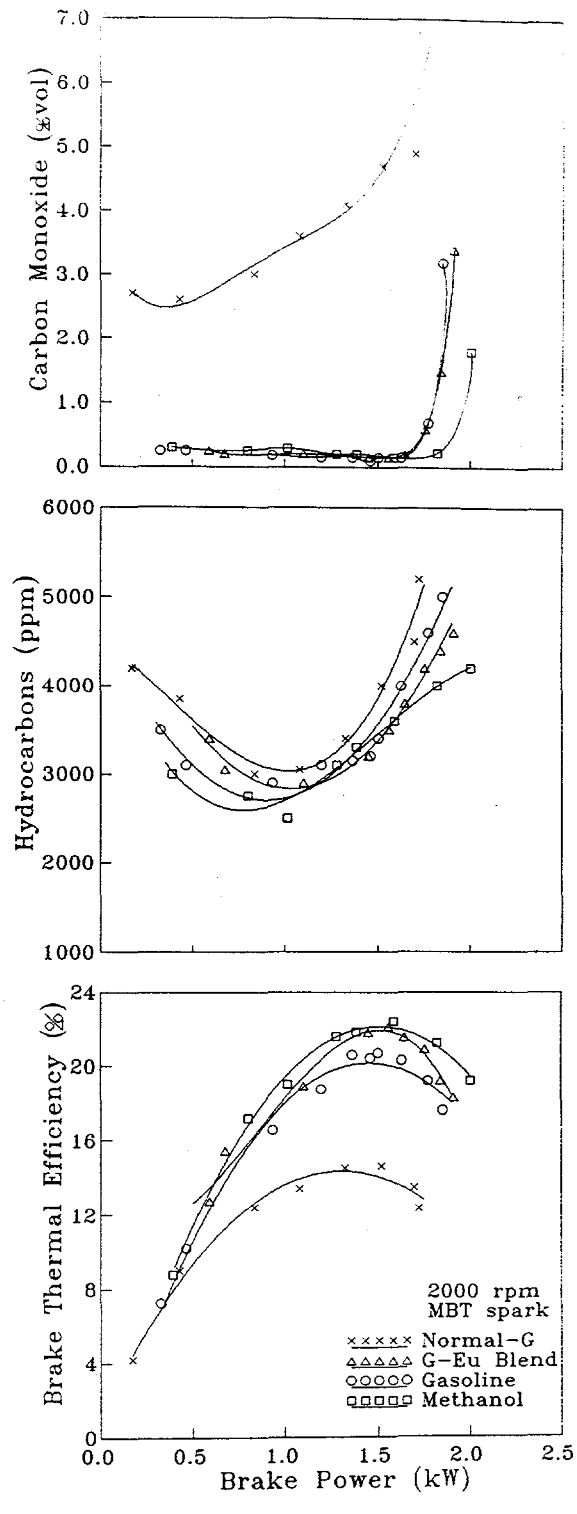

Fig. 13 Performance characteristics of the optimized design with three different fuels at 2000 rpm

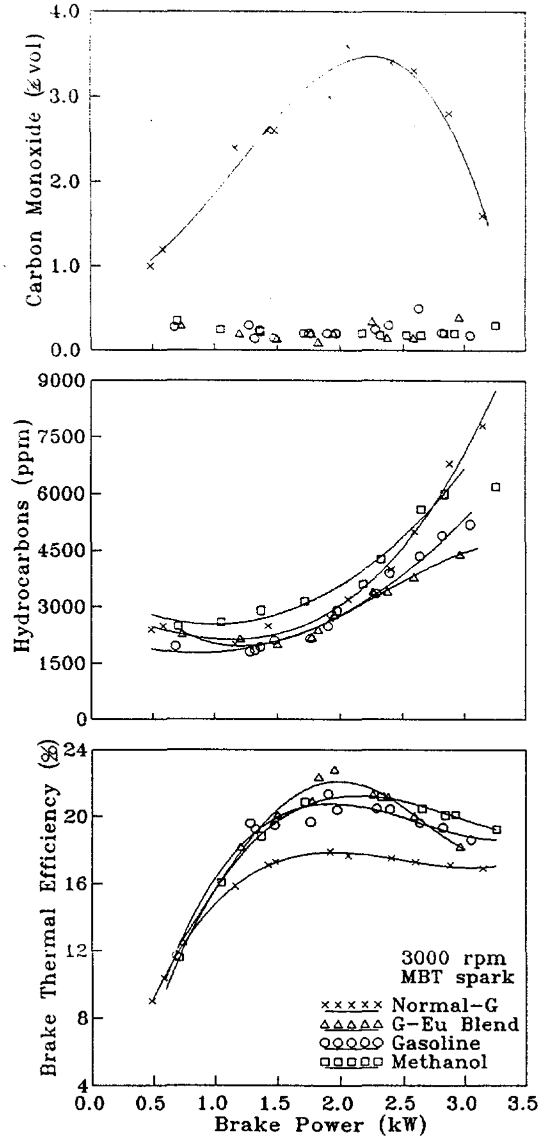

Fig. 14 Performance characteristics of the optimized design with different fuels at 3000 rpm

with the help of high-octane fuels. The superior fuel characteristics and better lean combustion characteristics of both the high-octane fuel blend and methanol fuel, compared to gasoline fuel, contribute to an increase in power output in the optimized design at a CR of 9:1. At a CR of 9:1, the knock-limited power output is increased by 12.7% with the optimized design using high-octane fuel blend compared to the normal gasoline-fueled engine. In the case of methanol fuel, its higher latent heat of vaporization cools the air entering the engine much more than in the case of gasoline, and this cooling effect increases the air density and mass flow. This increase in trapped charge density will compensate for the loss in power output at higher compression ratios. Among the three fuels, the performance with high-octane fuel blend is similar to that with methanol fuel, and both are superior to gasoline fuel. These performance differences among the three fuels reflect the influence of fuel properties in the optimized design.

Further insight into the optimal design with fuel variations can be provided through the exhaust gas analysis. From Figures 13 and 14, it is observed that the HC emissions are generally lower with the optimized design, in particular at higher outputs, for all three fuels tested. HC emissions are reduced by about 800 to 3400 ppm over those of the normal engine, depending on the operating conditions and the fuel employed. These substantial reductions in HC emissions are obtained due to lower fresh charge losses, leaner fuel-air mixture conditions, and improved combustion achieved by incorporating several approaches in the optimized design. As expected, HC emissions also vary with the type of fuel employed. Among the three fuels, the high-octane fuel blend shows reduction in HC emissions consistently over the entire range of engine operation. With methanol fuel, HC emissions are marginally higher compared to the normal engine under part-load conditions. The differences in HC emissions could be explained by variations in the hydrocarbon combustion mechanism with different devices. The reduction in HC emission that could be obtained due to lower fresh charge losses with the new scavenging system is perhaps counterbalanced by the relatively poor vaporization characteristics of the methanol fuel under part-load conditions. Nevertheless, in the regions where combustion chamber surface temperature is high enough to provide better vaporization, the reduction in HC emission is considerable. Generally, the HC emissions will be higher at higher compression ratios, due to lower exhaust gas temperatures. Hence, the net reduction in HC emissions with the optimized design is not as significant as expected.

The CO emissions are considerably reduced over the entire range of engine operation with the optimized design, using any of the three fuels. The reduction in CO emissions is consistent with all three fuels, about 0.7 to 3.8% by volume at various operating conditions, over those of the normal engine. In the optimized design, perhaps all the modifications incorporated are contributing towards relatively lean fuel-air operation and improved combustion. For instance, the new scavenging system not only minimizes the short-circuited fuel but also reduces the exhaust residuals in the cylinder, thereby overcoming the problems of slow and sluggish combustion normally present with high exhaust dilution. Similarly, the in-cylinder catalyst and high-octane fuel blend enhance the flame propagation velocities. The high-energy ignition system initiates combustion even with weak mixtures and stabilizes combustion in the initial stages of flame propagation. Hence, the combined effects of the various modifications in the optimized design result in lower CO emission levels over the entire range of engine operation.

Figures 15 and 16 illustrate the variation of cylinder peak pressure, ignition delay, and combustion duration with brake power at constant engine speeds of 2000 and 3000 rpm, respectively, for the optimized design. It can be observed from these figures that the combustion characteristics are improved considerably with the optimized design--higher cylinder peak pressures, lower ignition delay, and shorter combustion duration in particular--using high-octane fuel blend and methanol as fuels. The cylinder peak pressures are higher by about 5 to 6.5 bar at 2000 rpm and by about 4 to 6 bar at 3000 rpm for both high-octane fuel blend and methanol fuel. Similarly, ignition delay is reduced by about 3 to 8 degrees crank angle, and combustion duration is shortened by about 5 to 12 degrees crank angle, depending on the operating conditions, for both high-octane fuel blend and methanol as fuels. The reduction in ignition delay and combustion duration are possibly attributable to the better pre-flame reactions and higher flame propagation velocities of the lean mixtures obtained with various measures incorporated to improve the combustion at a CR of 9:1 in the optimized design. In addition to the effects of in-cylinder catalytic pre-reactions and the high-energy ignition system, the effects of fuel type also play an important role. For example, both methanol fuel and high-octane fuel blend have wider flammability limits and higher burning velocities at lean-mixtures, which directly influences ignition delay and combustion characteristics. As a result, lower ignition delay and shorter combustion duration are obtained with the optimized design compared to the normal engine. These improvements in combustion characteristics are in accordance with the brake thermal efficiency benefits obtained as a result of improved scavenging and combustion processes. To understand the complete combustion behavior of the optimized design with different fuels, more in-cylinder studies (beyond the scope of the present work) are necessary.

Among the three fuels tested with the optimized design, the high-octane fuel blend is superior compared to methanol and gasoline fuels, taking into account all the performance and emissions parameters. The superior characteristics of the high-octane fuel blend in the optimized design are also demonstrated through the comparison of heat-release rate and cumulative heat-release patterns over those of the normal engine at constant engine speeds of 2000 and 3000 rpm in Figure 17. The cumulative heat-release and heat-release rate patterns for the optimized design differ considerably from those of the normal engine, in particular at 2000 rpm. Heat-release patterns are sharp and close to TDC, indicating the higher heat-release rates. The peak rate of heat release is also higher for the optimized engine, substantiating the claims of improved combustion and higher brake thermal efficiencies over the normal engine. The brake thermal efficiency, HC, CO, and combustion characteristics are compared with all the modifications against the optimum design operating on high-octane fuel blend at two different representative operating conditions in Figures 4 to 9. It is evident from these figures

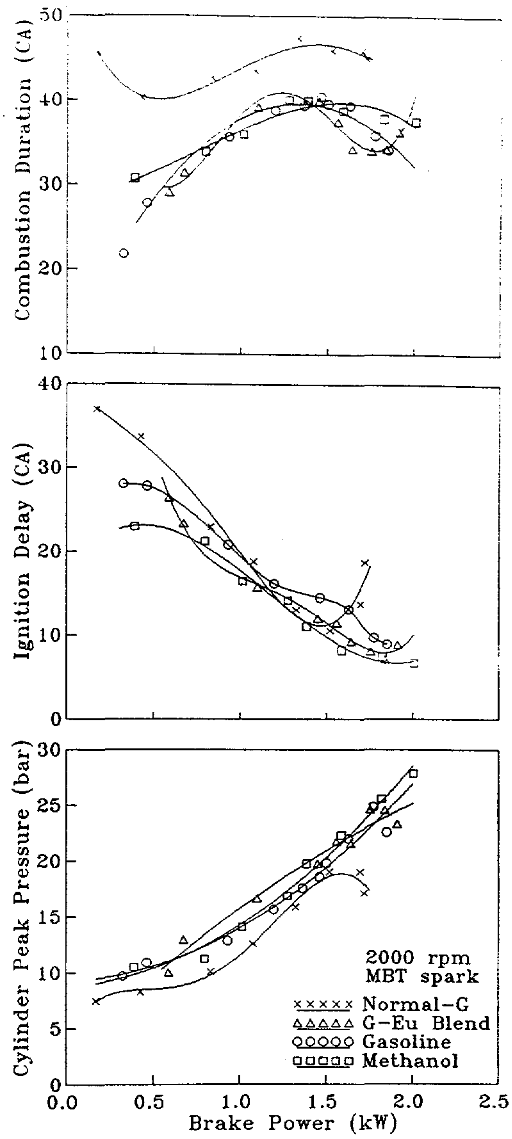

Fig. 15 Combustion characteristics of the optimized design with three different fuels at 2000 rpm

that, by combining certain devices with their optimum control parameters, an optimum design could be evolved. However, their effects are not cumulative in the optimized design, for several reasons. This is partly due to higher heat-transfer losses; also, some of the concepts might be redundant for producing a desired effect at a given operating condition in the engine cylinder. The optimized design, at typical operating conditions, has beneficial effects with regard to brake thermal efficiency, HC and CO emissions, and combustion characteristics compared to each of the devices tested.

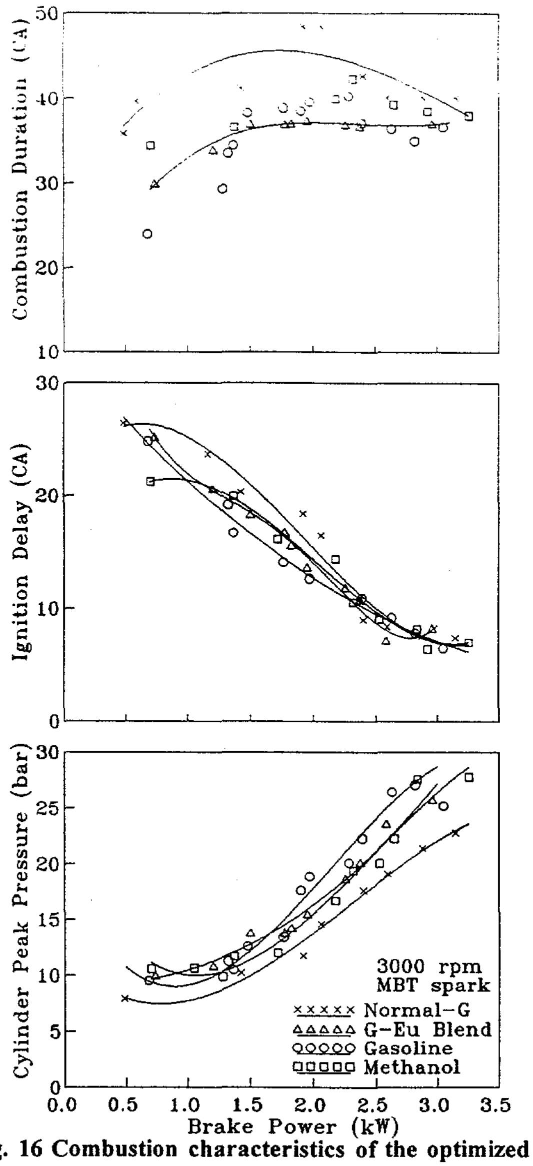

Fig. 16 Combustion characteristics of the optimized design with three different fuels at 3000 rpm

The optimized design of the two-stroke SI engine that was developed in the present work demonstrates improved fuel economy, lower exhaust emissions, better combustion characteristics, and multi-fuel capability compared to a conventional, gasoline-fueled engine. Many two- and three-wheeled vehicles in city driving conditions operate at part-load to medium-load, and the optimized design has shown significant benefits in this power range. Apart from the benefits listed, the design modifications proposed in the present work are simple, free from mechanical complexities, low-cost, and easy to adopt for both production and existing engines.

CONCLUSIONS

Based on the experimental investigations on a single-cylinder, carbureted, two-stroke SI engine with various measures incorporated to improve the fuel economy and to reduce exhaust emissions, the following conclusions are drawn:

1. Between the two types of optimal configurations developed, the engine with the following design features was found to be superior with regard to brake thermal efficiency, combustion, and exhaust emission characteristics and is denoted as the "optimal design".
   (i) Two extra reed valves at the transfer ducts, with an optimal secondary air flow through the reed valves.
   (ii) Combustion chamber surface coated with copper catalyst; a breakerless, high-energy, capacitive-discharge ignition system with platinum-tipped-electrode spark plug; and high-octane fuel blend (80% gasoline + 20% eucalyptus oil) or methanol as a fuel.
   (iii) A high compression ratio of 9:1 and lean fuel-air mixtures (A/F = 13.0 to 17.5).
2. With the optimized design, the absolute brake thermal efficiency increases from 14.6 to 22.1% at 1.5 kW, 2000 rpm, and from 17.7 to 23.3% at 2 kW, 3000 rpm, using high-octane fuel blend or methanol as a fuel, over the normal gasoline-fueled engine. There is no power loss or knock while operating at a CR of 9:1. The maximum reduction in HC and CO emissions is in the range of 1000 to 3400 ppm and 3 to 4% by volume, respectively, depending on the operating conditions and fuel employed, with the optimized design compared to the normal gasoline-fueled engine. These substantial improvements in brake thermal efficiency and exhaust emissions are obtained due to improved scavenging and lean combustion capabilities with the devices incorporated in the optimized design.
3. Cylinder peak pressures are higher by about 3 to 5 bar, ignition delay is lower by 2 to 4 degrees crank angle, and combustion duration is shorter by about 4 to 10 degrees crank angle, depending on the operating conditions and the fuel employed, in the optimized design over the normal engine due to higher burning velocities and improved combustion. The cumulative heat-release and heat-release rate patterns are sharp and close to TDC and are considerably higher than those of the normal engine.
4. The optimal two-stroke SI engine evolved from the present work is fuel-efficient, has lower exhaust emissions, and has a potential to apply to existing two- and three-wheeled vehicles in India.

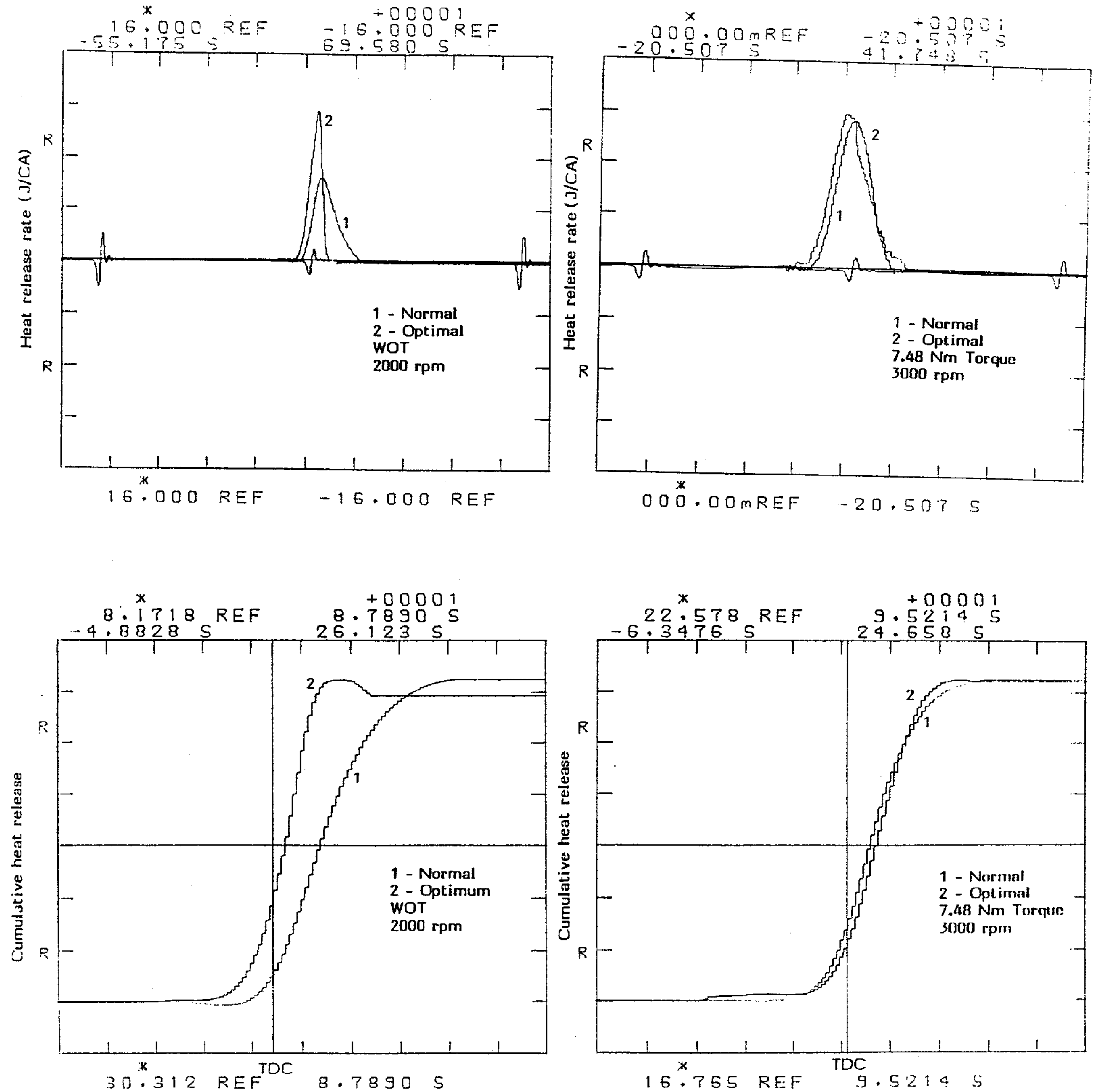

Fig. 17 Heat-release rate and cumulative heat-release patterns for normal and optimized engines

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