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Ignition Enhancement in a Two-Stroke Spark-Ignition Engine

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ABSTRACT

Conventional two-stroke spark-ignition (SI) engines have difficulty meeting the ignition requirements of lean fuel-air mixtures and high compression ratios, due to their breaker-operated, magneto-coil ignition systems. In the present work, a breakerless, high-energy electronic ignition system was developed and tested with and without a platinum-tipped-electrode spark plug. The high-energy ignition system showed an improved lean-burn capability at high compression ratios relative to the conventional ignition system. At a high compression ratio of 9:1 with lean fuel-air mixtures, the maximum percentage improvement in the brake thermal efficiency was about 16.5% at 2.7 kW and 3000 rpm. Cylinder peak pressures were higher, ignition delay was lower, and combustion duration was shorter at both normal and high compression ratios. Combustion stability as measured by the coefficient of variation in peak cylinder pressure was also considerably improved with the high-energy ignition system.

INTRODUCTION

The ignition system in an SI engine must provide sufficient voltage across the spark plug electrodes to set up a discharge and supply sufficient energy to the discharge to ignite the combustible mixture adjacent to the plug electrodes under all operating conditions. The magnitude of the energy required is influenced by the composition of the mixture and the flow field around the spark plug. It has been reported that the minimum ignition energy required increases with leaner fuel-air mixtures and with higher flow velocities near the spark plug [1].

The breaker-operated, magneto-coil ignition system is commonly used in small-capacity, carbureted, two-stroke SI engines (typically in motorcycles, mopeds, autorickshaws, lawn mowers, snowmobiles, chain saws, and outboard marine applications) because of its simplicity and independence of a battery or generator. However, one of its major limitations is a decrease in available voltage as the engine speed increases, due to limitations in the current-switching capability of the breaker system and decreases in the time available for energy storage in the primary coil. The breaker points are also subjected to both electrical and mechanical wear, which results in short maintenance cycles. As a result, the conventional magneto-coil ignition system is not effective when engines are operated at high compression ratios with lean fuel-air mixtures. The techniques [2] for dealing with this problem include use of (i) an extended-reach spark plug with a wide gap, (ii) dual spark plugs, (iii) multi-point and multi-electrode spark plugs, (iv) a platinum-tipped, thin-electrode spark plug, and (v) a high-energy spark-discharge ignition system.

The influence of the number and type of spark plugs, the ignition system, plug location, and ground electrode orientation have been studied in detail by Anderson and Asik [3,4]. A multi-point spark ignition system having with several spark gaps in the combustion chamber has been found to be one of the most effective techniques [5] for reducing the combustion duration and extending the lean misfire limit. Rado et al. [6] showed that the three-gap spark plug improves fuel economy and extends the lean misfire limit. Durbin and Tsai [7] compared the ten-gap multiple-electrode spark plug (total gap width of 10.2 mm) with the conventional spark plug. Platinum-tipped, thin-electrode spark plugs have been found effective because of their wider plug gap and the reduced voltage required for ignition under certain electrode-polarity conditions [8]. Plasma-jet ignition systems are being investigated to provide high temperatures that can ignite very lean fuel-air mixtures through both an increased formation of free radicals and a larger volume of initially burned mixture. It has been reported [9] that a plasma jet with an ignition energy as high as 14 times that of a normal spark has extended lean operation air-fuel ratio limits by about 24%. Such systems, however, have the disadvantage of higher power requirements, in that the relatively high electrical power for the ignition system must be drawn from the engine. Ignition enhancement techniques, therefore, remain attractive for the stabilized combustion of lean fuel-air mixtures at high compression ratios.

In the present study, the effects of spark plug type, spark gap, duration of spark, and energy of the ignition system were evaluated. Engine performance, lean combustion capability, cyclic variations, and exhaust emissions were determined with lean fuel-air mixtures and high compression ratios.

HIGH-ENERGY IGNITION SYSTEM

A circuit block diagram for the breakerless, high-energy, capacitive-discharge ignition (CDI) system that was developed in the present work is shown in Figure 1. A schematic diagram of the timing of pluses at various stages in the ignition circuit is illustrated in Figure 2. A fixed signal at 60° before top-dead-center (TDC) from an optical-trigger pulse generator was used as the triggering signal for the ignition system. This signal was converted to transistor-transistor logic (TTL) levels and was delayed using a digital monostable multi-vibrator. The timing of the spark was varied by changing this delay. The width of the delayed trigger signal was varied using another monostable multi-vibrator that varied the spark duration. An opto-isolator and CDI driver were introduced to isolate the TTL stage (low-voltage section) from the CDI unit (high-voltage section). This was to help prevent any damage to the ignition circuit by high-voltage spikes from the CDI unit. The resulting signal was used to trigger the CDI system. The start of the spark could be varied from 60° before TDC to TDC, and the spark duration could be varied by about 40° of crank angle.

CYCLIC VARIATION

Cycle-by-cycle (CBC) variation in the peak pressure development within the cylinder of an SI engine has long been recognized as a phenomenon of considerable importance [10]. An understanding of the nature and causes of the CBC pressure variation is necessary to stabilize the spark-ignition, lean-burn combustion process. In general, it has been shown that cyclic dispersion is dependent upon the homogeneity of the air-fuel mixture, the turbulence in the vicinity of the ignition process, residual exhaust gas mixing, and various ignition parameters [11]. Ignition and flame propagation greatly influence cyclic variability in lean-burn engines.

In an SI engine, flame initiation, which is a necessary condition for combustion, and early flame development, which greatly affects cyclic variation, are both significantly influenced by the spark discharge process. Combustion instability, as exhibited by CBC variations in the combustion pressure waveform is hypothesized [12] to be a result of two major factors: variability in the chemical and physical conditions near the spark plug at the time of ignition and flame propagation velocities early in the combustion process. With lean combustion, ignition parameters have an important effect on combustion stability.

In the present work, an attempt has been made to quantify the relationship(s) between ignition system parameters and CBC variations and thereby to understand their effects on engine-fuel consumption and emissions. The coefficient of variation (COV) in peak cylinder pressure is the primary variable measured in the present study for quantifying CBC variations. Because of the limited recording capacity (8 kB) and processing capabilities of the digital data acquisition system (DDAS) employed, combustion stability was measured in terms of the COV of peak pressure rather than the more common COV of indicated mean effective pressure (IMEP). The COV in peak pressure was calculated using the approach suggested by Amann [13] and is given below:

$$COV = \frac{\sqrt{\dfrac{\sum [\bar{P} - P_i]^2}{(N-1)}}}{P_i}$$

Where N = Number of cycles sampled
P_i = Peak pressure of each cycle
$\bar{P}$ = Average peak pressure

To analyze cyclic variations in peak pressure, it is enough to measure only specific portions of consecutive pressure-time signals. This can be done by repeatedly feeding clock pulses into the DDAS only during those portions of the cycle when the pressure-time signal is to be recorded. A digital data clock-pulse controller was designed and built to enable such a procedure. A schematic diagram of the clock-pulse controller is shown in Figure 3. Clock pulses from a function generator (the frequency was adjusted to obtain one or more samples per degree of crank angle) were fed into the DDAS via this controller. The controller was triggered during every cycle by a TTL-level pulse coming from an optical trigger pulse generator. Upon receiving the trigger pulse, a counter in the controller started counting the clock pulses until a set value (set by a thumb-wheel switch) was reached. After this the clock pulses were allowed into the DDAS and sampling commenced. Simultaneously, the number of clock pulses going to the DDAS were counted by a second counter, and the whole system was reset once the second counter reached a value set by a second thumb-wheel switch. With this, the clock pulses going to the DDAS, and hence the sampling, were stopped until the arrival of next trigger pulse during the following cycle. By adjusting the two thumb-wheel switches, it was possible to capture specific portions of the pressure-time signals. By keeping the sampling rate per cycle very small, a large number of such signals were recorded

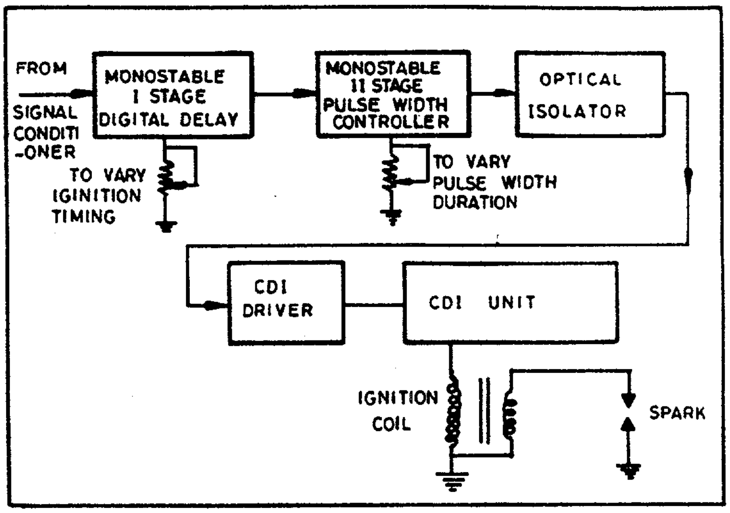

Fig. 1 Schematic diagram of the ignition system circuit.

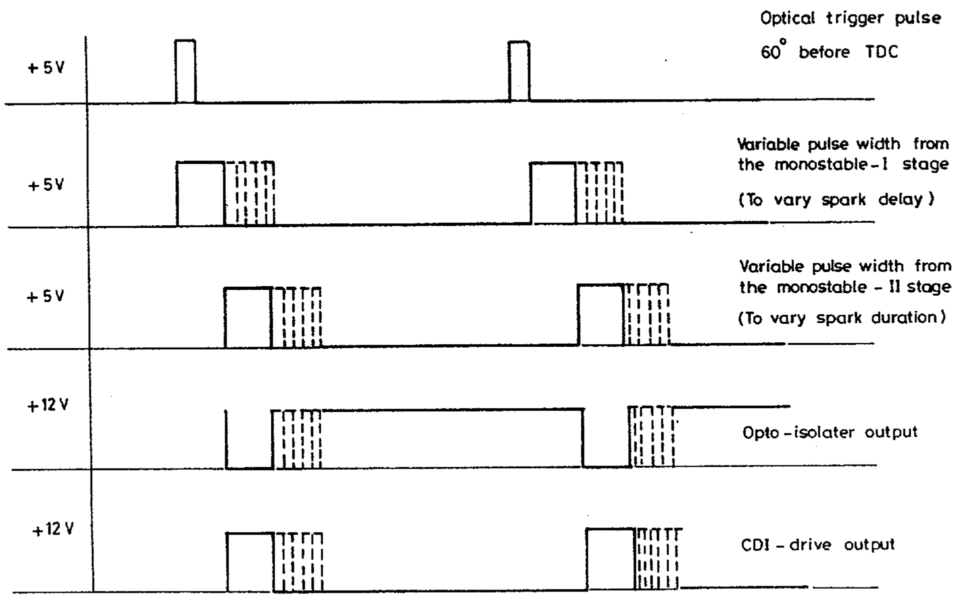

Fig. 2 Schematic timing diagram at various stages.

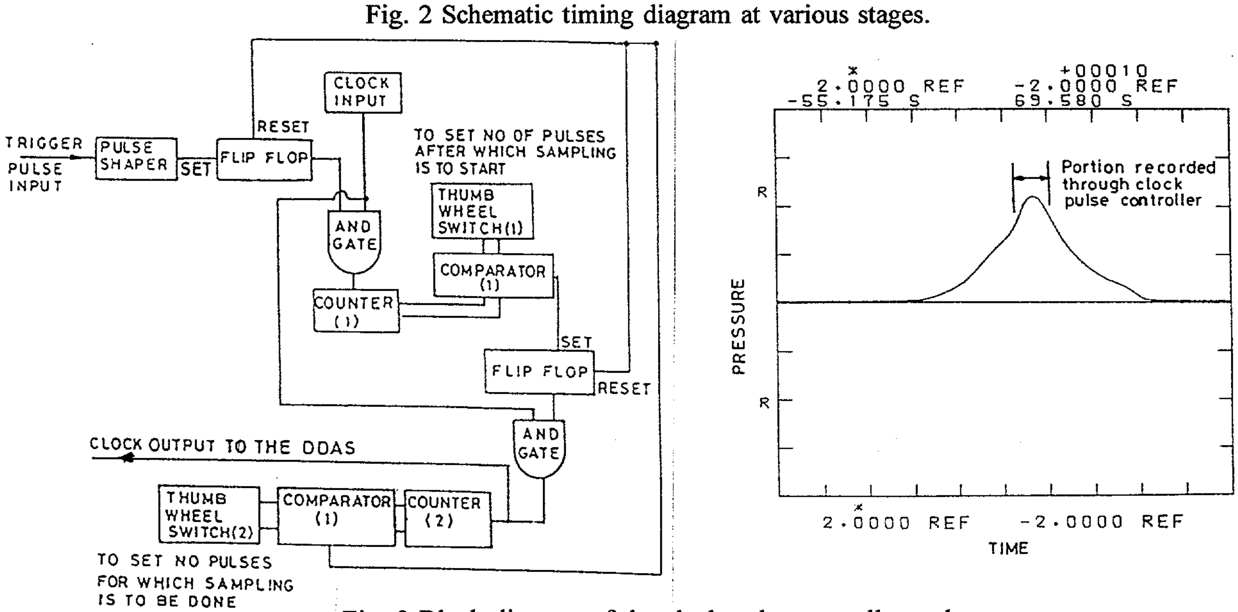

Fig. 3 Block diagram of the clock pulse controller and
a portion of the pressure-time signal recorded.

despite the limited data memory. The COV in peak pressure proved to be highly useful as a combustion diagnostic for the high-energy ignition system. Combustion stability and system performance were quantified using this parameter. Lower COV values indicate better combustion stability and hence improved performance.

EXPERIMENTAL PROCEDURE

A single-cylinder, small-capacity (150 cc, 4.3 kW at 5200 rpm), loop-scavenged, air-cooled, two-stroke SI engine was employed in the present investigations. This engine was coupled to an eddy current dynamometer for speed and torque measurements. An infrared gas analyzer was employed to measure HC and CO emission levels in the engine exhaust. Calibrated standard instrumentation was used for air- and fuel-flow rates. Cylinder pressure was measured by a flush mounted, piezoelectric pressure transducer. The pressure, crank angle, and spark signals were fed to the DDAS (IWATSU-SM 2100) for storage and analysis of the various combustion parameters. The DDAS was a two-channel signal analyzer with a simultaneous sampling and display capability. The salient features of this system are given in Appendix I. The maximum sample length was 4 kB per channel, which restricted the number of consecutive pressure-time traces that could be recorded and processed. The number of clock pulses allowed into the DDAS were varied using the digital data clock-pulse controller developed for the experiment. Sampling was done in the external clock mode with the TTL-level square pulses supplied by a function generator at a frequency that gave one or more samples per degree of crank angle. By suitable adjustment of the sampling rate (two thumb-wheel switches), selected portions of consecutive pressure-time signals were recorded.

The first stage of the experiment was conducted with the breakerless, high-energy ignition system using a normal spark plug at the standard compression ratio (7.4:1) and a fuel-jet size (standard) of 0.84 mm. Variable-load tests were carried out at constant engine speeds of 2000 and 3000 rpm. In the second stage of experiment, a platinum-tipped-electrode spark plug was used with the breakerless, high-energy ignition system at a compression ratio of 7.4:1 and a fuel-jet size of 0.84 mm. In the third stage, the compression ratio was increased to 9:1 and fuel-jet size was reduced to 0.80 mm (lean). The breakerless, high-energy ignition system and the platinum-tipped-electrode spark plug were employed in the third-stage tests. The spark plug gap was optimized for the best performance and the minimum spark advance for best torque (MBT) spark time was maintained throughout the experimentation.

RESULTS AND DISCUSSION

The variations in brake thermal efficiency with brake output at the constant engine speeds of 2000 and 3000 rpm are presented in Figures 4 and 5 for the high-energy CDI system, with and without the platinum-tipped-electrode spark plug.

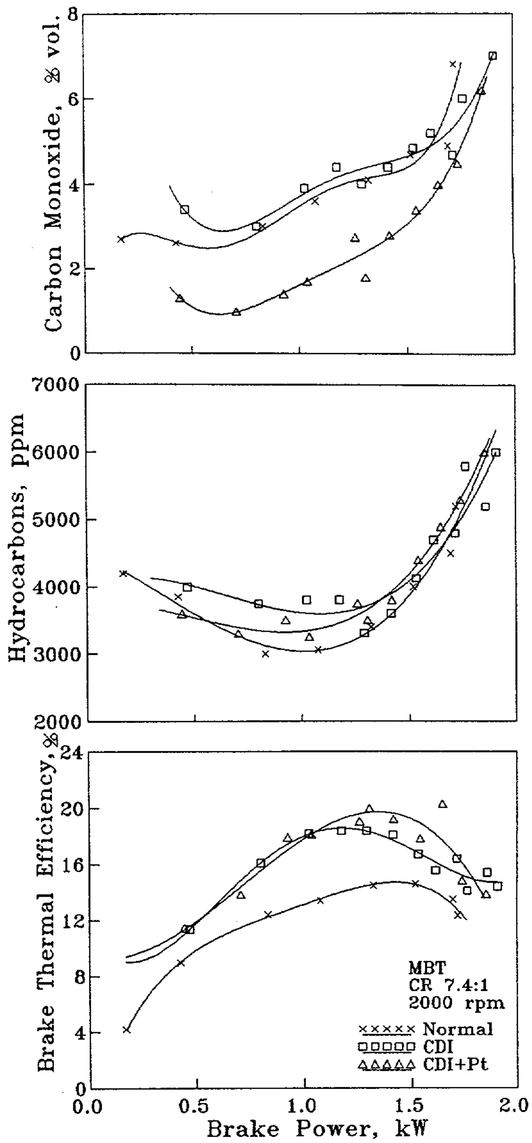

Fig. 4 Performance characteristics of the high-energy ignition system at the normal compression ratio and 2000 rpm.

The results were compared with those from a normal engine (magneto-coil ignition system with a standard spark plug and a compression ratio of 7.4:1). With the high-energy CDI system, there was a considerable improvement in the brake thermal efficiency over the entire range of engine operation. The platinum-tipped-electrode spark plug further improved the high energy CDI system's brake thermal efficiency. The drawbacks of the conventional, magneto-coil ignition system, such as irregular combustion and low spark energy, were eliminated with the high-energy ignition system, thereby improving engine performance. The maximum absolute brake thermal efficiency obtained at 2000 rpm, and 1.4 kW (Figure 4) increased from 14.5% for the normal ignition system to 18.1% for the high-energy ignition system and to 19.2% with platinum-tipped-electrode spark plug.

At low engine speeds, the conventional magneto-coil ignition system evidently produced a low voltage across the

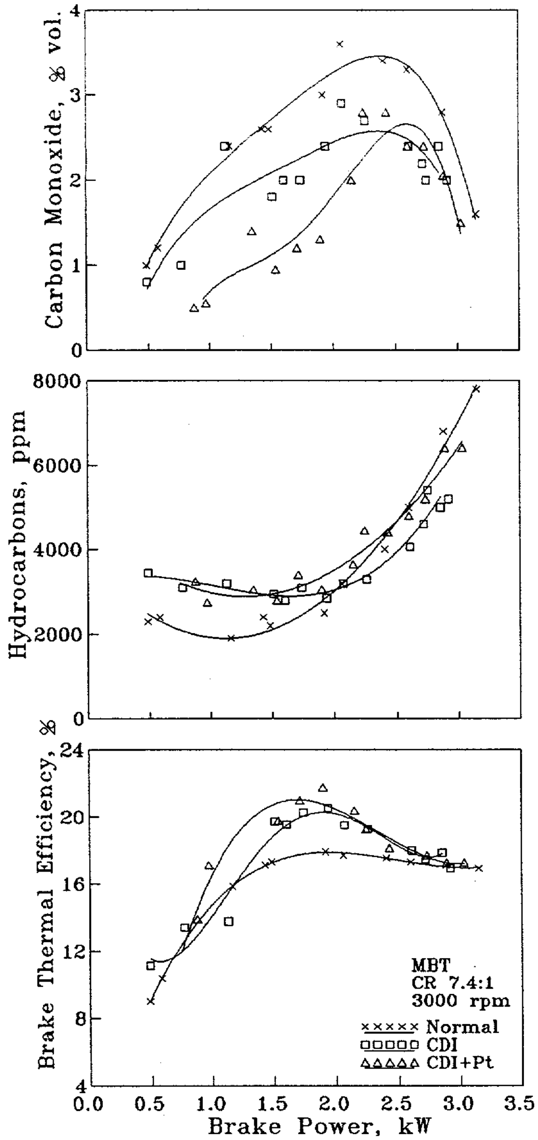

Fig. 5 Performance characteristics of the high-energy ignition system at the normal compression ratio and 3000 rpm.

spark plug, since the voltage supplied to the primary side of the ignition coil was low. In addition to the low ignition energy, conditions near the spark plug were unfavorable to igniting the air-fuel mixtures easily, due to the presence of a large amount of exhaust dilution. This was one of the main reasons for irregular combustion and misfiring at part loads [14]. On the other hand, the high-energy ignition system supplied a high ignition energy across the wide spark gap, and therefore burned the air-fuel mixtures effectively. Hence, an improvement in the brake thermal efficiency with the high-energy ignition system was observed, particularly at 2000 rpm, as is shown in Figure 4. Similar improvements were noticed with the high-energy ignition system at the other speed tested (Figure 5). The platinum-tipped electrode spark plug provided a larger spark gap because of its reduced electrode diameter. It required a lower breakdown voltage, which reduced electrode wear caused by capacitive discharge. Hence, the high-energy ignition system with platinum-tipped-electrode spark plug gave better performance than the normal ignition system.

The HC and CO emissions for the high-energy ignition system with both the normal spark plug and the platinum-tipped-electrode spark plug are shown in Figures 4 and 5 at the constant engine speeds of 2000 and 3000 rpm, respectively. A considerable reduction in CO emission was observed with the high-energy ignition system, and the reduction was especially pronounced when the platinum-tipped-electrode spark plug was used. The maximum reduction in CO emission was about 1.9% by volume at 0.8 kW and 2000 rpm (Figure 4) and about 1.6% by volume at 1.5 kW and 3000 rpm (Figure 5) for the high-energy ignition system with the platinum-tipped-electrode spark plug, when compared to the CO emission for the normal ignition system. These reductions in CO emission were possibly due to improved combustion resulting from a high spark-discharge energy. The presence of platinum in the electrode tip should also enhance the oxidation process, due to its catalytic effect. As a result, lower CO emissions were obtained. HC emissions differed marginally from those with the normal ignition system over most of the operating conditions, and were higher by about 400 to 600 ppm at part-load operation (Figure 4) with the high-energy ignition system.

The variations of combustion parameters, such as cylinder peak pressure, ignition delay, and combustion duration, with brake output are illustrated in Figures 6 and 7 at the constant engine speeds of 2000 and 3000 rpm, respectively. The results obtained with the high-energy ignition system (with and without the platinum-tipped spark plug) were compared to those obtained with the normal ignition system. High-energy ignition system decreased the ignition delay, shortened the combustion duration, and increased the peak cylinder pressure. The maximum increase in cylinder pressure was about 8 bar at 1.7 kW and 2000 rpm, and 7.5 bar at 3 kW and 3000 rpm. The corresponding reduction in ignition delay was about 9° and 2° of crank angle, respectively. The combustion duration was decreased by about 5° to 12° of crank angle, depending on the operating conditions. These improvements were possibly due to the superior ignition conditions, better flame initiation, and higher flame propagation speeds attained with the high-energy ignition system.

The variations in brake thermal efficiency and HC and CO emissions with brake power for both the high-energy ignition system with a platinum-tipped-electrode spark plug and the normal ignition system are shown in Figure 8 at a high compression ratio of 9:1, constant engine speed of 3000 rpm, and with a lean fuel-jet size of 0.80 mm. The results show that a considerable improvement in the brake thermal efficiency, particularly at higher brake outputs, was observed with the high-energy ignition system. The maximum percentage improvement in brake thermal efficiency was about 16.5% at 2.76 kW and 3000 rpm.

The conventional magneto-coil ignition system was generally not effective when the engine operated at higher compression ratios with leaner fuel-air mixtures. For an

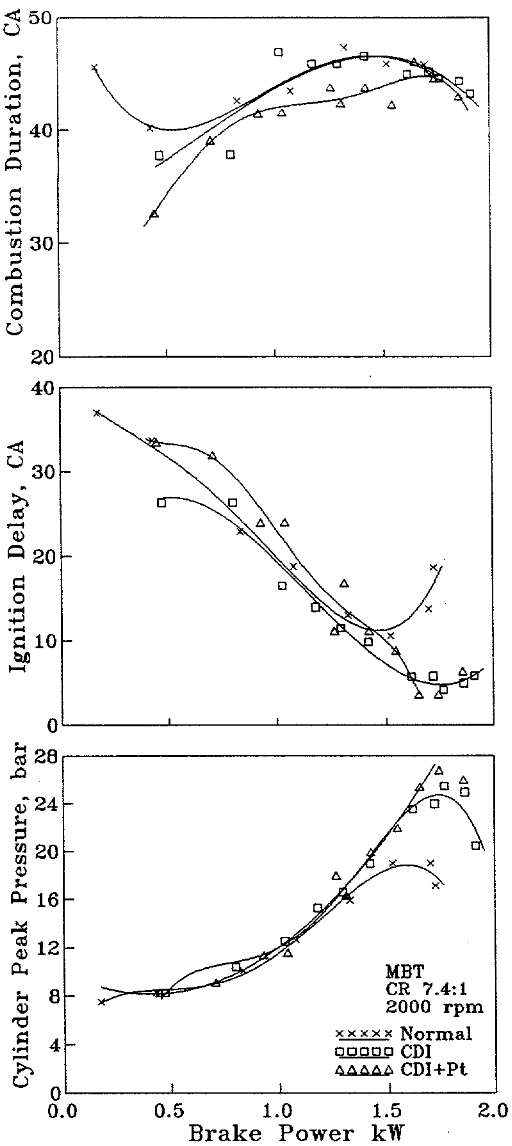

Fig. 6 Combustion characteristics of the high-energy ignition system at the normal compression ratio and 2000 rpm.

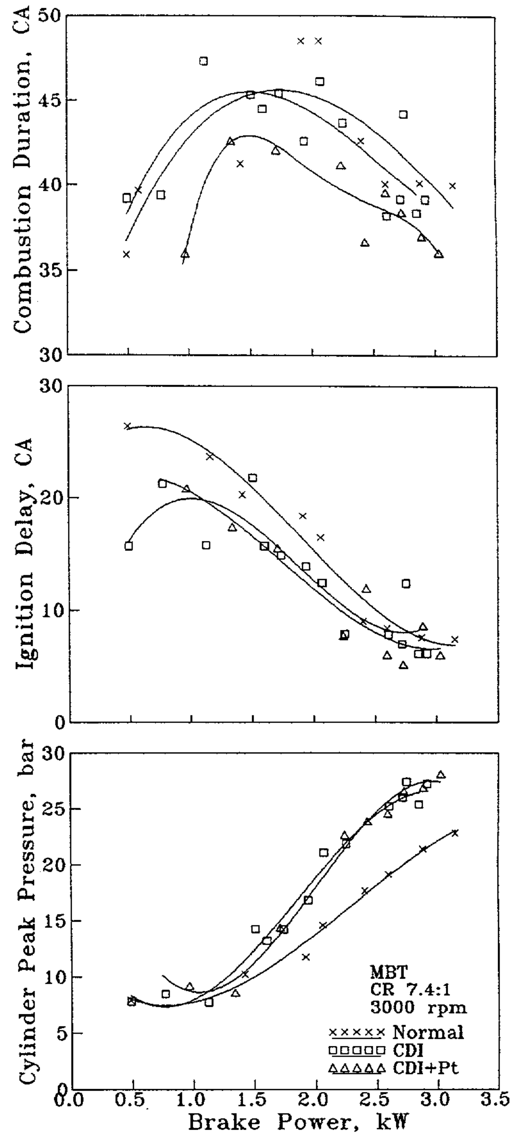

Fig. 7 Combustion characteristics of the high-energy ignition system at the normal compression ratio and 3000 rpm.

engine to run effectively under these conditions, the spark-discharge energy must be higher and spark duration must be longer or multiple spark must be used. These requirements were fulfilled to a large extent by using the high-energy ignition system. As a result, a considerable improvement in performance was observed with the high-energy ignition system over the entire range of engine operation.

The HC and CO emission characteristics for both the high-energy and normal ignition systems are shown in Figure 8 at a constant engine speed of 3000 rpm. The CO emissions were marginally higher with the high-energy ignition system. The increase in CO emissions was about 0.2% to 0.8% by volume for the range of engine operation tested. The HC emissions increased by about 50 to 200 ppm with the high-energy ignition system. This increase in HC emission was expected, due to bulk quenching of the gases at higher cylinder pressures. The wider spark gap between the electrodes significantly affected hydrocarbon emissions, particularly, when the mixture was leaner than the stoichiometric ratio [15]. The higher spark-discharge energy required the use of relatively large diameter electrodes and a larger spark gap between the electrodes. Hence, a careful electrode design and optimization of the gap between electrodes were necessary to reduce the HC emissions under lean fuel-air mixture operating conditions.

Figure 9 shows the variation in cylinder peak pressure, ignition delay, and combustion duration for both the high-energy and normal ignition systems at a constant engine speed of 3000 rpm (a high compression ratio of 9:1 and a lean fuel-jet size of 0.8 mm). Lower ignition delays were obtained with the high-energy ignition system over the entire range of engine operation tested. Cylinder peak pressures were higher and combustion duration was shorter with the high-energy ignition system. The maximum increase in cylinder peak pressure was about 5.0 bar at 2.5 kW and 3000 rpm. The pre-flame

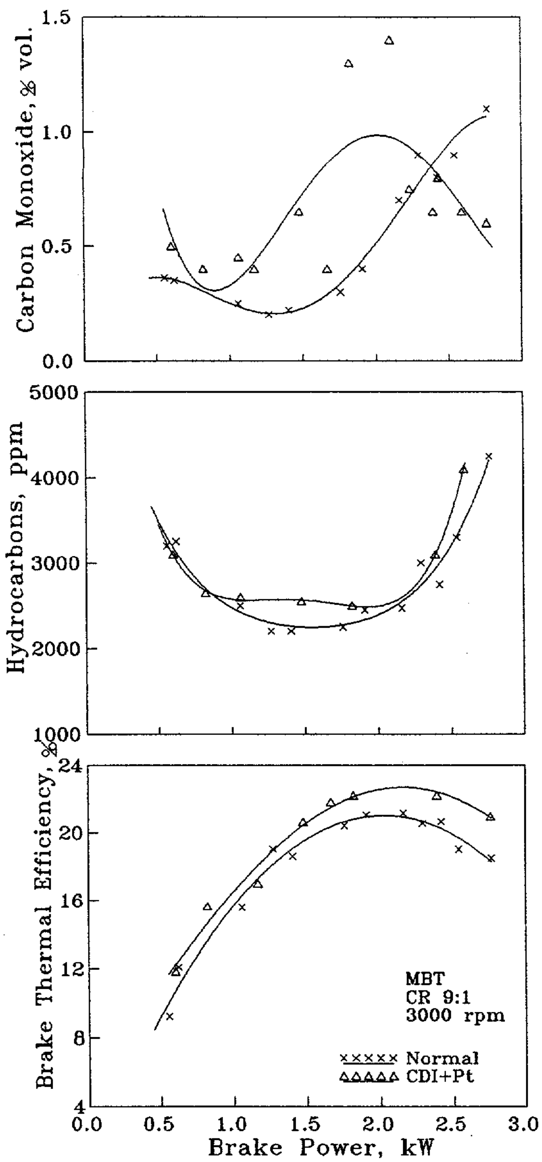

Fig. 8 Performance characteristics of the high-energy ignition system at a high compression ratio of 9:1.

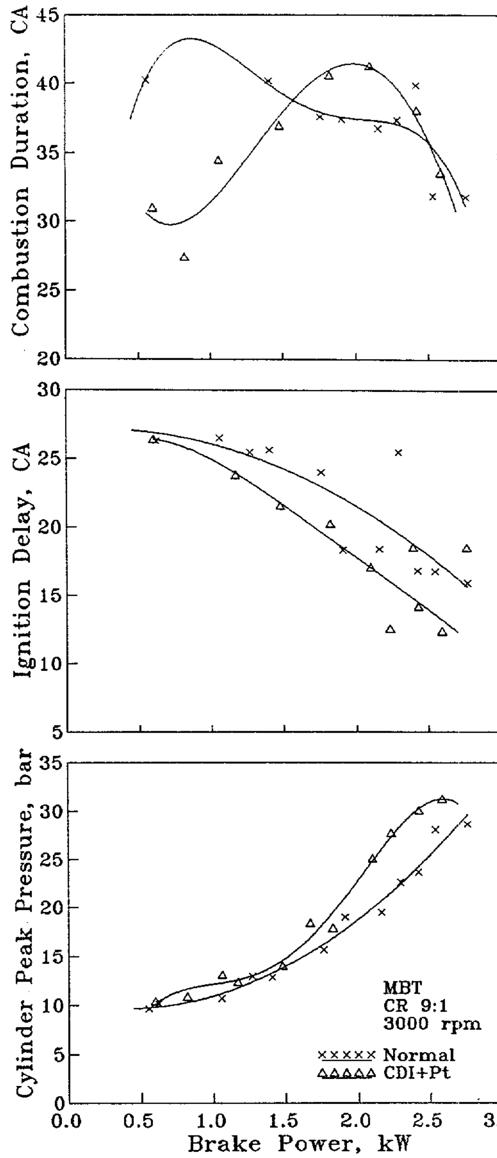

Fig. 9 Combustion characterstics of the high-energy ignition system at a high compression ratio of 9:1.

reactions in the vicinity of the spark plug were expected to be faster and to start much earlier, due to a transmission of high discharge energy from the spark to the lean mixtures. Higher flame propagation speeds and better combustion stabilization during the early period of the combustion were attributed to the observed combustion improvements with the high-energy ignition system.

The COV in peak pressure as functions of brake power for both the normal and high-energy ignition systems are shown in Figure 10 at the constant engine speeds of 2000 and 3000 rpm. The COV in peak pressure was generally higher at part loads with the normal ignition system. The large percentage of exhaust residuals at these conditions contributed to the high cyclic pressure variability [16]. However, combustion stability was improved with the high-energy ignition system because the high spark discharge effectively initiated the combustion process. As a result, the COV in peak pressure was considerably reduced over the entire range of engine operation. The improvement in combustion stability as measured in terms of the COV in peak pressure could also be noticed from the actual pressure-time traces (consecutive peak cylinder pressures) at various operating conditions, as illustrated in Figures 11 and 12, which indicate that the lower the COV in peak pressure, the higher the combustion stability and better the thermal efficiency become. When the COV in peak pressure decreased from 0.16 to 0.13, the corresponding absolute brake thermal efficiency increased from 19.2% to 22.3% at 1.97 kW and 3000 rpm with the high-energy ignition system, as compared to the normal ignition system.

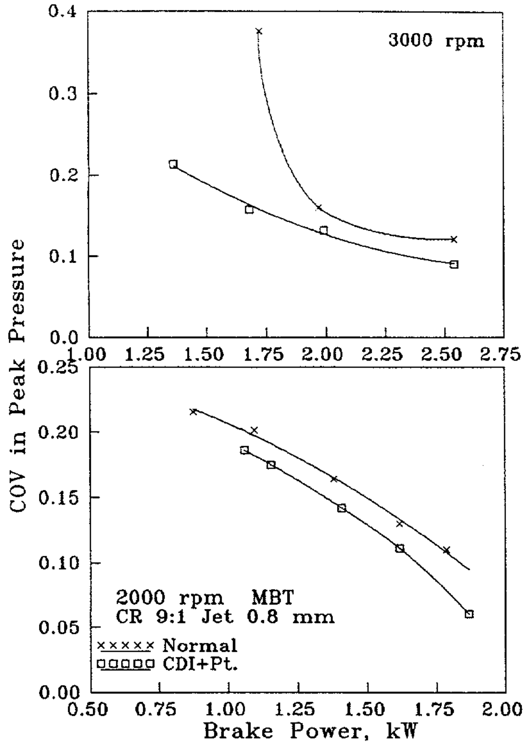

Fig. 10 COV in peak pressure with the high-energy ignition system.

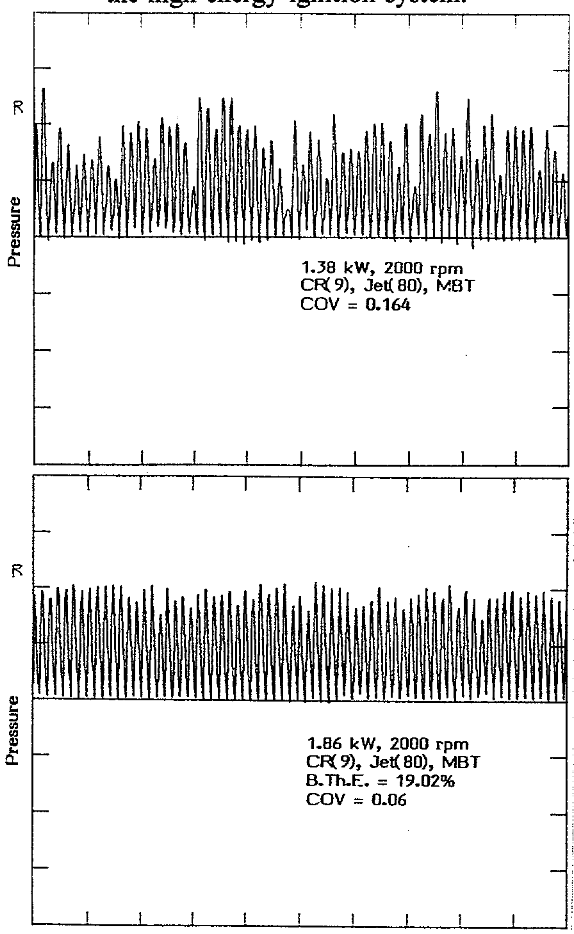

Fig. 11 Cycle-by-cycle variation of the cylinder peak pressure-time traces for the normal and high-energy ignition systems at 2000 rpm.

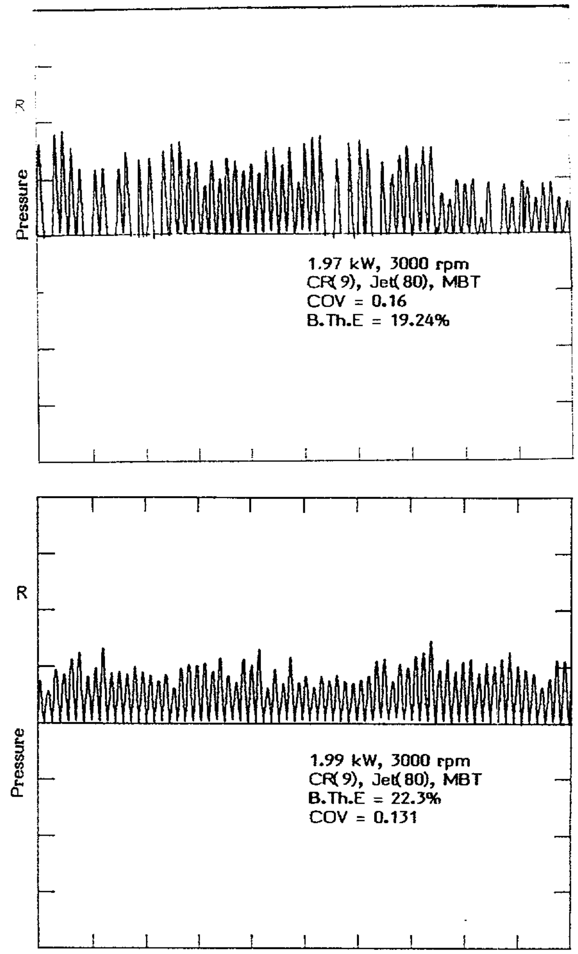

Fig. 12 Cycle-by-cycle variation of the cylinder peak pressure-time traces for the normal and high-energy ignition systems at 3000 rpm.

CONCLUSIONS

1. A breakerless, high-energy, capacitive-discharge ignition system and a platinum-tipped-electrode spark plug having a variable spark delay, spark duration, and spark gap were developed to overcome the drawbacks of the conventional, contact-breaker-points-operated, magneto-coil ignition system, and to allow operation at higher compression ratios with leaner fuel-air mixtures.
2. At a normal compression ratio of 7.4:1, the high-energy ignition system with the platinum-tipped-electrode spark plug increased the absolute brake thermal efficiency from 14.5% to 19.2% at 1.4 kW and 2000 rpm relative to the normal ignition system.
3. CO emissions were considerably reduced when the platinum-tipped electrode spark plug was employed with the high-energy ignition system. The maximum reduction in CO emission was about 1.9% by volume at 0.8 kW and 2000 rpm and 1.6% at 1.5 kW and 3000 rpm.

4. Higher peak cylinder pressures, lower ignition delays, and shorter combustion durations were observed when using the high-energy ignition system with the platinum-tipped-electrode spark plug. The maximum increase in cylinder pressure was about 7.5 bar. The ignition delay and combustion duration decreased by about 5° to 12° of crank angle, depending on the operating conditions.

5. The lean-burn capabilities at higher compression ratios were considerably enhanced by using the high-energy ignition system with a platinum-tipped-electrode spark plug. The maximum percentage improvement in the brake thermal efficiency was about 16.5% (at 2.7 kW, 3000 rpm, a high compression ratio of 9:1, and with a lean fuel-jet size of 0.8 mm) with the high-energy ignition, compared to normal ignition system.
6. Combustion stability as measured by the COV in peak cylinder pressure was found to be better with the high-energy ignition system. The COV in peak pressure decreased from 0.16 to 0.13 at 1.9 kW and 3000 rpm, with the corresponding improvement in the absolute brake thermal efficiency from 19.2% to 22.3% with the high-energy ignition system.

ACKNOWLEDGMENT

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REFERENCES

1. Bolt, J.A., and D.L. Harrington, "Effects of Mixture Motion Upon the Lean Limit and Combustion of Spark-Ignited Mixtures," SAE Paper 670467, 1967.
2. "Lean-Burn Ignition Systems," *Automotive Engineering*, Vol. 97, No. 2, pp. 141-145, 1989.
3. Anderson, R.W., and J.R. Asik, "Lean Air-Fuel Ignition System Comparison in a Fast-Burn Engine," SAE Paper 850076, 1985.
4. Anderson, R.W., and J.R. Asik, "Ignitability Experiments in a Fast-Burn, Lean-Burn Engine," SAE Paper 830477, 1983.
5. Narihiko, N., B. Toyokazes, and S. Yoshiaki, "Multi-Point Spark Ignition for Lean Combustion," SAE Paper 852091, 1985.
6. Rado, W.G., J.E. Amey, B. Bates, and A.H. Turner, "The Performance of a Multi-Gap Spark Plug Designed for Automotive Applications," SAE Paper 760264, 1976.
7. Durbin, E.J., and K.C. Tsai, "Extending the Lean Limit Operation of an SI Engine with a Multiple-Electrode Spark Plug," SAE Paper 830476, 1983.
8. Norihiko, N., K. Tatsuo, H. Masanori, and N. Takagi, "A New Platinum-Tipped Spark Plug Extends the Lean Misfire Limit and Useful Life," SAE Paper 830480, 1983.
9. Kupe, J., H. Wilhelmi, and W. Adams, "Operational Characteristics of a Lean-Burn SI Engine: Comparison Between Plasma Jet and Conventional Ignition System, SAE Paper 870608, 1987.
10. Martin, J.K., S.L. Plee, and D.J. Remboski, Jr., "Burn Modes and Prior-Cycle Effects on Cyclic Variations in a Lean-Burn Spark-Ignition Engine Combustion," SAE Paper 880201, 1988.
11. Young, M.B., "Cyclic Dispersion in the Homogeneous Charge Spark-Ignition Engine-A Literature Survey," SAE Paper 810020, 1981.
12. Patterson, D.J., "Cylinder Pressure Variations, A Fundamental Combustion Problem," SAE Paper 660129, 1966.
13. Amann, C.A., "Cylinder Pressure Measurement and Its Use in Engine Research," SAE Paper 852067, 1985.
14. Wakisaka, T., Y. Hamamoto, S. Ohigashi, T. Nabari, and M. Hasoi, "Limits of Flame Propagation in Two-Stroke Cycle Gasoline Engine," *Bulletin of the JSME*, Vol. 19, No. 136, pp. 1204-1211, 1976.
15. Quader, A.A., "What Limits Lean Operation in Spark-Ignition Engines-Flame Initiation or Propagation ?," SAE Paper 760760, 1976.
16. Soltau, J.P., "Cylinder Pressure Variations in Petrol Engines," *Proceedings of the I. Mech. E.*, No. 2, pp. 99-117, 1960-61.

APPENDIX I
Digital Data Acquisition/Analysis System

Make	IWATSU SM 2100 B (JAPAN)
No. of channels	2
Frequency range	0.2 to 100 kHz
Internal clock	0.2 to 100 kHz
External clock	DC to 100 kHz (TTL)
A/D converter	Sequential comparison
Resolution	12 bits
Sensitivity	0.1 to 200 V full scale
Filter range	10 Hz to 100 kHz
Word length	16 bits
Maximum data length	4 kB
Memory in BASIC mode	8 kB
Display	Cathode ray tube (CRT)
Storage	Floppy (5 1/4") disc
Scale	Linear and log

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