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HYDROGEN ASPIRATION IN A DIRECT INJECTION TYPE DIESEL ENGINE—ITS EFFECTS ON SMOKE AND OTHER ENGINE PERFORMANCE PARAMETERS

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Abstract—An experimental study was undertaken to investigate the possibility of reducing diesel particulates in the exhaust by aspirating small quantities of gaseous hydrogen in the intake of a diesel engine. A single cylinder, direct injection type diesel engine was used in the experiments. Hydrogen flow rates equivalent to about 10% of the total energy substantially reduced smoke emissions at part loads. At the full rated load, reduction in smoke levels was limited; this is believed to be due to the lower amounts of excess air available in the cylinder. It was found that the engine thermal efficiency was dependent on the portion of hydrogen energy, out of the total input energy, supplied to the engine. There was no significant change in the hydrocarbon emissions but oxides of nitrogen in the exhaust increased with an increase in hydrogen energy.

NOMENCLATURE

A/F	air–fuel ratio (kg/kg)
$\dot{E}_H$	rate of hydrogen energy supply (KJ/s)
$\dot{E}_t$	rate to total energy supply (KJ/s)
θ_{peak}	angle when peak pressure occurs (deg)
ϕ	fuel air equivalence ratio ($\phi < 1$ is lean)

INTRODUCTION

Hydrogen has been considered as a fuel for internal combustion engines since the early part of this century but only in recent years has there been a strong, revived interest in this area. Hydrogen has been adopted to spark ignition engines as reflected in literature indicating a number of projects where hydrogen has been successfully used in experimental vehicles powered by spark ignition engines. But investigations into the use of hydrogen as a primary or supplementary fuel for compression ignition engines have been limited due to the high auto ignition temperature of hydrogen, as shown in Table 1. Hence, ignition of hydrogen–air mixture by compression alone is generally considered to be difficult [1] unless engines with very high compression ratios are used [2]. In some of the recent studies [3–6] several researchers have used glow plugs, spark assist or pilot injection of diesel fuel to achieve reliable combustion of hydrogen in compression ignition engines.

Most of these studies were directed at developing and evaluating different methods of using hydrogen in compression ignition engines. Very little work has been done to determine if the presence of small quantities of hydrogen in direct injection type (DI) diesel engines influence particulate emissions in the exhaust.

Due to higher compression ratio, diesel engines have higher thermal efficiency than spark ignition engines and there are indications that future generation of DI diesel engines would show at least 10% gain in efficiency over the indirect injection type diesel engines [7]. The performance of a DI diesel engine is strongly determined by the fuel properties and by the manner in which fuel is introduced in the combustion chamber, which, in turn, depends on the spray characteristics and how well they are matched with the air motion in the cylinder. Matching of spray with the air movement over a wide range of engine operating conditions is difficult and hence naturally aspirated DI diesel engines emit higher smoke levels than an indirect injection type at the corresponding operating conditions [7].

The chemical kinetics of the formation and burning of soot in a diesel combustion process is highly complex and there is limited understanding of the basic processes involved in its chemistry. At present, it is considered that air–fuel ratios, temperature and residence time may be only some of the factors that influence the amount of smoke in a diesel exhaust [8, 9]. Experimental evidence indicates that oxygen, in sufficient quantity, suppresses soot formation. Such reduction of smoke emission may occur either by interfering with the polymerization process or by directly burning soot particles in the presence of sufficiently high temperature

Table 1. Comparison of fuel properties

	Hydrogen	Diesel
Density at ambient (kg/m^3)	0.081	824
Flammability in air (% by vol.)	4–75	—
Stoichiometric A/F (mass)	34.3	14.9
Heat of combustion (lower, MJ/kg)	119.98	42.36
Auto-ignition temperature (k)	845.1	524.5
Maximum flame speed (m/s)	2.92	0.38

[10, 11]. It is also considered that OH radicals may help to remove soot on one hand but they also play an important role in the dehydrogenation process which leads to soot formation. It would appear that the presence of small quantities of gaseous hydrogen, which has higher flame temperature and wider flammability limits, may help to burn part of the soot formed early in the combustion process. At high loads where fuel spray may penetrate near the periphery of the cylinder walls, presence of gaseous hydrogen may help to burn carbon formed in that region. Sampling of combustion gases in the cylinder of a DI diesel engine has shown soot levels near the walls to be higher than expected [11]. Some of the earlier work done on fumigating small amounts of volatile fuels to the intake air of diesel engine indicated that exhaust smoke levels can be reduced by such an approach [12].

The main aim of this investigation was to determine if small quantities of gaseous hydrogen supplied during the intake process can help to reduce smoke generated by a DI diesel engine. The study also considered the effects of such hydrogen fumigation on the emissions of oxides of nitrogen (NO*x*), unburned hydrocarbons (HC) and other combustion related problems.

EXPERIMENTAL

Apparatus

A single cylinder, naturally aspirated, four stroke DI engine was used in this study. The engine had a bore of 84 mm, a stroke of 78 mm and a compression ratio of 17.4. The fuel injection system of the engine comprised of a plunger type pump with an injector having three spray holes, each 0.25 mm in diameter. The injection needle lift pressure was set at 20 MPa.

Diesel fuel No. 2 (cetane No. 44–46) was supplied to the injection system while hydrogen gas of 99.95% purity was fed to the intake manifold, about 10 cm from the inlet valve. Hydrogen flow rate was controlled by a fine metering valve and the flow rate was measured by a calibrated flow meter maintained at a constant temperature of 25°C. An air-cooled pressure transducer was installed in the cylinder head; the output of the transducer-amplifier system was displayed on a storage oscilloscope along with the timing marks generated by the flywheel. The fuel injection timing was kept constant, at 22° BTDC, in all these experiments.

Samples of exhaust gas were withdrawn at bout 15 cm downstream from the exhaust valve and were supplied to a chemiluminescent NO–NO*x* analyser and a flame ionization HC detector. In addition, a Bosch smoke meter was used to measure smoke levels in the exhaust. The pressure of the exhaust gas from the engine was reduced to near atmospheric before supplying to the smoke meter. The probe from the smoke meter was located about 30 cm downstream from the exhaust valve.

Methods

Two different types of tests were conducted during this investigation. In the first set hydrogen flow rate was maintained constant while the diesel fuel flow rate was increased to increase the engine output at constant engine speed.

In the second set the engine was again run at a constant speed of 40 rev/s but the hydrogen flow rate was varied. In these tests two initial loads, corresponding to zero hydrogen flow rate, were selected: full rated load of the engine and 82% of the full rated load. There were some variation in the engine output as different hydrogen rates were supplied to the engine. At the full rated load the air–fuel ratio in the engine with diesel fuel operation alone, was typically 16.5. As hydrogen flow rate was increased, there was some reduction in air flow rate. This would have lowered the air–fuel ratio for the diesel fuel alone had the latter flow rate been kept constant. In order to make comparison of particulate emission meaningful, diesel fuel flow rate was adjusted so that the engine produced maximum torque at the constant speed. Thus, at full rated load, there was a small increase in the ratio of total mass of air to the total mass of diesel fuel. But the equivalent air–fuel ratio, based on the total energy equivalent to the diesel fuel energy, either remained constant or decreased as the hydrogen flow rate was increased. The maximum hydrogen flow rates used in this set of tests comprised about 14% of the total energy at full rated load and about 17% at 82% full rated load.

RESULTS AND DISCUSSIONS

Generally the engine performance and combustion related phenomena showed similar patterns at both the engine speeds, 40 and 35 rev/s. Hence, most of the results shown in this paper are for 40 rev/s engine speed;

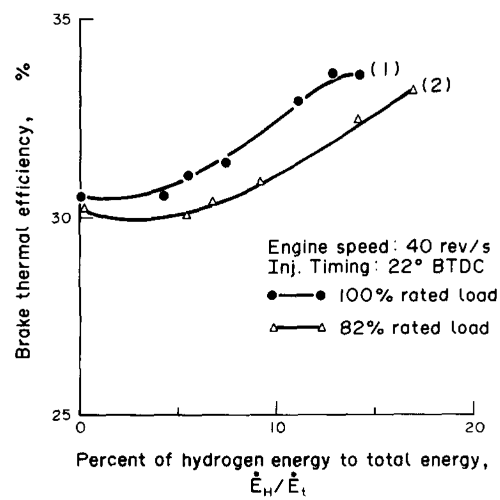

Fig. 1. Engine speed 40 rev/s, injection timing 22°BTDC. (1) 100% rated load, (2) 82% rated load.

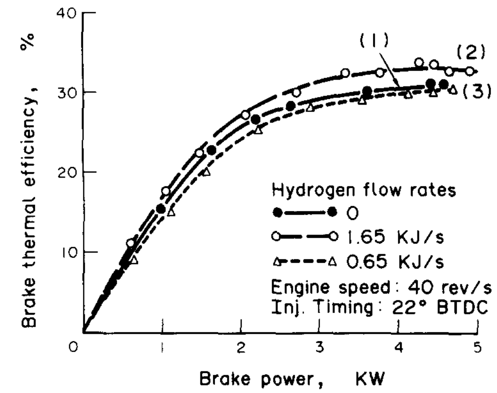

Fig. 2. Engine speed 40 rev/s, injection timing 22°BTDC. Hydrogen energy flow rates (1) 0, (2) 1.65 KJ/s, (3) 0.65 KJ/s.

the analysis pertaining to the results at 40 rev/s can be easily extended to the engine performance at 35 rev/s.

The effects of hydrogen aspiration on the engine brake thermal efficiency are shown in Fig. 1. In general, the efficiency steadily increased as the portion of hydrogen energy increased at both the power levels. At the lowest hydrogen fuelling rate, the engine efficiency either decreased or remained almost constant relative to the baseline operation, i.e. when no hydrogen was supplied to engine. The premixed hydrogen fuel to air equivalence ratios (ϕ) at these fuelling rates were extremely low, typically $\phi \sim 0.03$, which may have caused the fuel to burn in a very erratic manner. Combustion of hydrogen air mixtures at such low hydrogen fuel concentrations is dependent on the local temperature around parcels of fuel mixtures; higher temperature in the combustion chamber can thus aid in burning very lean mixtures of hydrogen. At 82% full load, the overall temperature in the combustion chamber is lower than full rated load operation. As a result, mixtures containing very low hydrogen content would burn better as full rated load than at part load operation. This may be the reason why the brake thermal efficiency decreased, relative to the baseline operation, when hydrogen fumigation rate was low.

That the rate of hydrogen energy supply can alter the engine thermal efficiency can be seen in Fig. 2. When the rate was 0.65 KJ/s the resulting efficiency was consistently lower than the baseline value. On the other hand increasing the rate to 1.65 KJ/s resulted in higher thermal efficiency over all the load range. Similar results were observed at 35 rev/s engine speed. The dependence of thermal efficiency on the portion of hydrogen energy supplied to the engine has also been observed in another investigation using diesel engine [6]. It may appear that the engine efficiency can be improved by increasing the percentage of hydrogen energy. But the results at full rated load indicate the possibility of either levelling off or a decrease in the efficiency curve for further increase in hydrogen flow. A problem of practical importance in a hydrogen fumigated diesel engine is the magnitude of peak cylinder pressure and the knocking associated with the high rates of pressure rise. Figure 3 shows how peak cylinder pressure and the crank angle when peak pressure occurs are influenced by the portion of the hydrogen energy in the mixture. Peak pressure increased sharply beyond about 11% of hydrogen in the mixture at full rated load. At the same time, the time for the occurance of peak pressure decreased from the baseline value, resulting in high rate of pressure rise and detectable knocking.

Acoustic noise levels were measured in the test cell at two different locations, about two meters, from the engine. These measurements showed a sharp increase in the level when the fuel mixture contained more than 11% of hydrogen energy, as shown in Fig. 4. It is interesting that the peak cylinder pressures for mixtures containing less than 6% hydrogen energy was higher than the baseline value but it occured late so that the average rate of pressure rise was almost the same as for the baseline operation.

Late occurence of peak cylinder pressure at low rates of hydrogen energy supply is believed to be due to delayed burning of hydrogen in the combustion

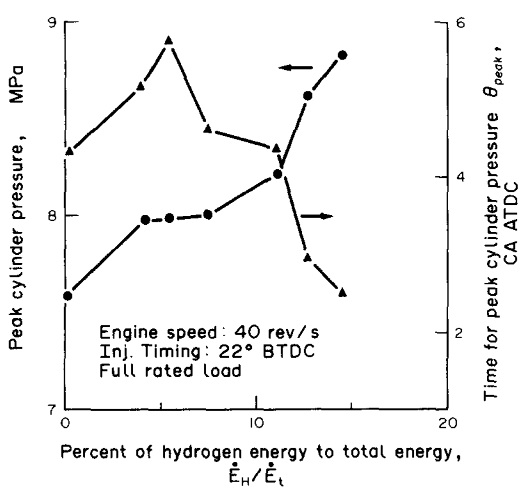

Fig. 3. Engine speed 40 rev/s, injection timing 22°BTDC, full rated load.

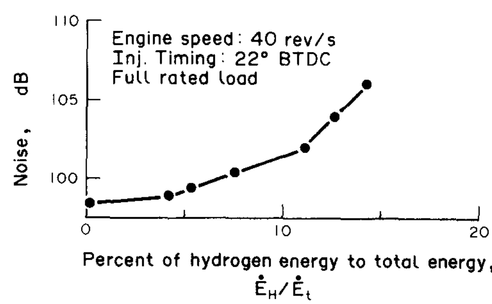

Fig. 4. Engine speed 40 rev/s, injection timing 22°BTDC, full rated load.

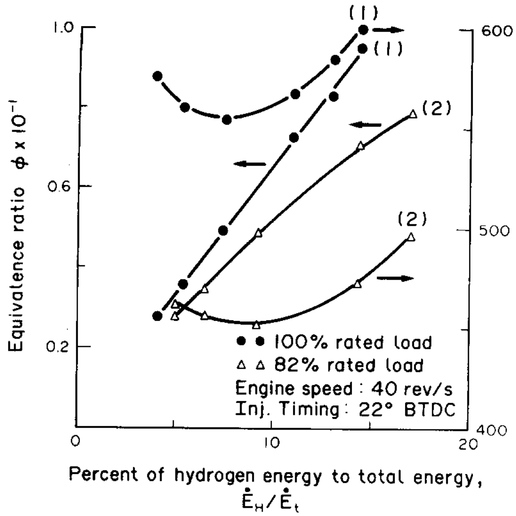

Fig. 5. Engine speed 40 rev/s, injection timing 22°BTDC. (1) 100% rated load, (2) 82% rated load.

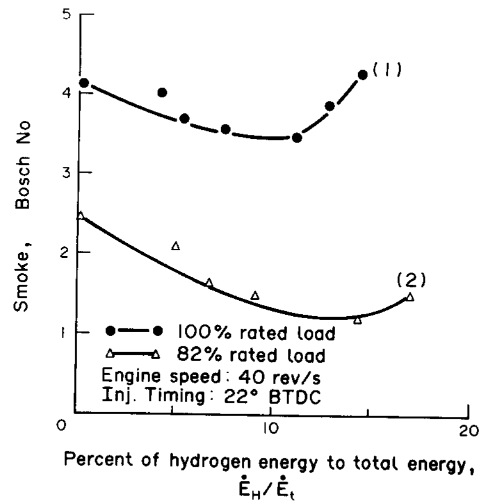

Fig. 6. Engine speed 40 rev/s, injection timinig 22°BTDC. (1) 100% rated load, (2) 82% rated load.

chamber. As stated earlier, the equivalence ratios for the premixed hydrogen–air mixtures used in this study were leaner than the lean flammability limit for hydrogen at ambient conditions. Thus, all the hydrogen inducted into the cylinder may not burn completely or may burn late during the expansion process. This explains partly why the thermal efficiency at part loads was lower than the baseline value when the hydrogen fuelling rate was low. No measurement of hydrogen concentration in the exhaust was made but the exhaust gas temperature, measured with a calibrated thermocouple probe, indicated the possibility of late burning when the hydrogen energy supply was low, as shown in Fig. 5. Comparison of Figs. 3 and 5 shows that the minimum exhaust temperatures were obtained when the time for the occurence of the peak cylinder pressure with hydrogen were almost identical to the baseline value. Increasing the portion of hydrogen increased the exhaust temperature due to rapid combustion and higher flame temperature for hydrogen for a given equivalence ratio. Figure 5 also shows the variation of ϕ for the premixed hydrogen–air mixtures. The equivalence ratios for low hydrogen flow rates were lower than the corresponding value for the lean flammability limits of hydrogen at the ambient conditions.

The effects of hydrogen fumigation on the exhaust smoke levels at 40 rev/s are shown in Fig. 6. For both the loads the smoke levels start to decrease as hydrogen content is increased up to a point; further increase in hydrogen was found to adversely affect the smoke density. At part load, reduction in smoke levels was much higher than at full load operation. As the hydrogen flow rate was increased, the flow rate of air inducted into the cylinder decreased slightly at the full rated load. A small reduction in air-flow rate would not be too severe at part loads but at full load, where air utilization is a critical factor in a DI diesel engine, it is certainly undesirable. This may be the reason why smoke density decreased substantially at part loads than at full rated load [10].

The kinetics involved in the formation and burning of soot in a diesel combustion process is highly complex and there is limited understanding of the chemistry involved in these processes. However, it has been shown that smoke levels of liquid hydrocarbons decrease as H/C ratio of the fuel is increased [13, 14]. Although gaseous hydrogen supplied to the engine will not show the same chemical behavior as that contained in a liquid hydrocarbon molecule, hydrogen fumigation increased the overall H/C ratio in the cylinder. Such an increase appears to result in soot reduction. It is also possible that presence of hydrogen inhibits dehydrogenation and polymerization steps essential in soot formation mechanism. But this does not explain why smoke levels were adversely affected when the hydrogen energy supply was increased unless the diesel fuel was deprived of sufficient oxygen by the combustion of hydrogen fuel.

To investigate this, air–fuel ratios for the diesel fuel alone were estimated on the assumption that hydrogen fuel burned immediately after ignition delay at the rich or the lean flammability limits, shown in Table 1. The values of the air–fuel ratios thus calculated are shown in Fig. 7 for the two engine outputs. It is apparent that if all the hydrogen were to burn at the lean limit, the diesel air–fuel ratio would decrease sharply with an increase in hydrogen flow rate. This should result in a sharp increase in smoke density with some degradation in the engine efficiency. On the other hand, if all the hydrogen were to burn at the rich limit, full load air–fuel ratios for the diesel fuel would be almost constant due to small variations in diesel fuel flow rate. At part load operation where the excess air ratio was relatively

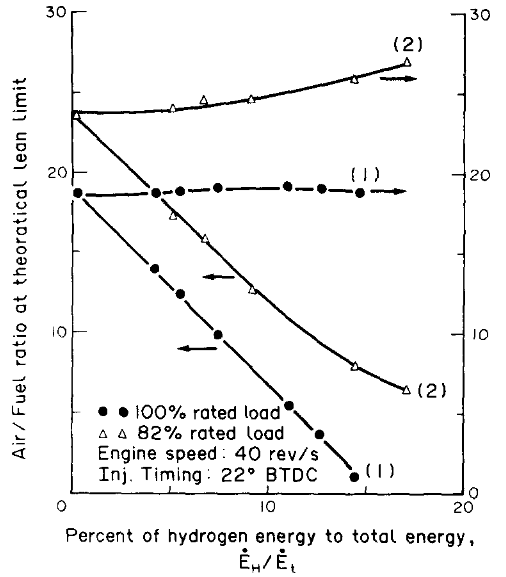

Fig. 7. Engine speed 40 rev/s, injection timing 22°BTDC. (1) 100% rated load, (2) 82% rated load.

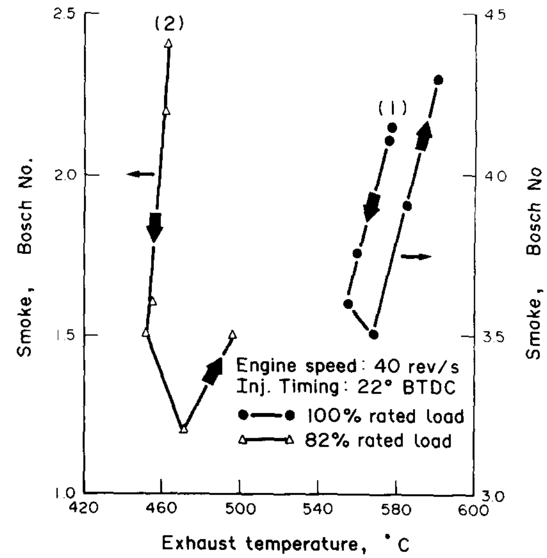

Fig. 8. Engine speed 40 rev/s, injection timing 22°BTDC. (1) 100% rated load, (2) 82% rated load.

higher than at full load, calculated air–fuel ratios increased particularly at high hydrogen flow rates; yet the smoke levels showed a reversed trend.

The combustion of hydrogen in the engine can take place even beyond the flammability limits shown in Table 1 due to high temperatures in the combustion chamber. Studies in Egerton [15] have shown that the lean limit for hydrogen–air can change depending on the mixture temperature; it was indicated that the final flame temperature should be at least 1000 K for the mixture to sustain combustion. As the mixture temperature prior to its combustion increases the flammability limits will be extended accordingly. Thus the overall air–fuel ratios for the diesel fuel may be quite different than those shown in Fig. 7.

There are indications that the temperature of gases inside the combustion chamber has some influence on the amount of soot generated [9]. Generally, high average cylinder gas temperature can be expected when the exhaust gas temperature is low. Figure 8 correlates the exhaust gas temperature and the smoke density for the two loads. It is intriguing that minimum smoke density was achieved when the exhaust temperature was around the minimum value. It is also interesting that this condition also corresponds to the region where the time for the occurence of peak cylinder pressure with and without hydrogen was almost identical.

Based on the results shown in Figs. 6–8, it seems that the aspiration of small quantities of hydrogen in a DI diesel engine can help to reduce smoke density. The optimum ratio of hydrogen to the total energy delivered to the engine should be such that the time for the occurence of peak cylinder pressure remains unaltered in relation to the baseline conditions.

Figures 9 and 10 show the effects of constant rate hydrogen energy flow on the exhaust smoke at engine speeds of 40 and 35 rev/s, respectively. The reduction in smoke levels at low hydrogen fuelling rates was almost negligible and part of this came about by some reduction in diesel fuel flow rate. But as hydrogen flow rate was increased substantial reductions in smoke density were achieved. In fact, at part loads, smoke levels decreased by over 50%; only at the full rated load was the gain limited due to some of the probable reasons described earlier.

The effects of adding different amounts of hydrogen

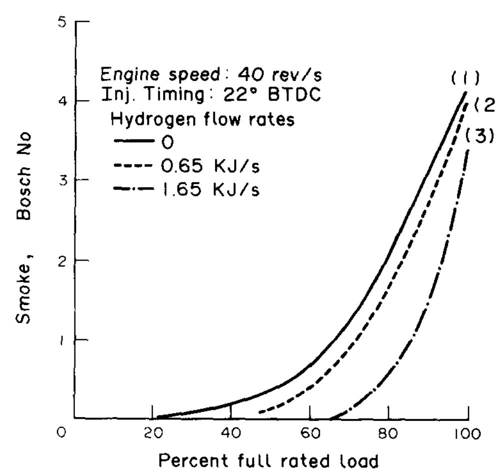

Fig. 9. Engine speed 40 rev/s, injection timing 22°BTDC. Hydrogen energy flow rates (1) 0, (2) 0.65 KJ/s, (3) 1.65 KJ/s.

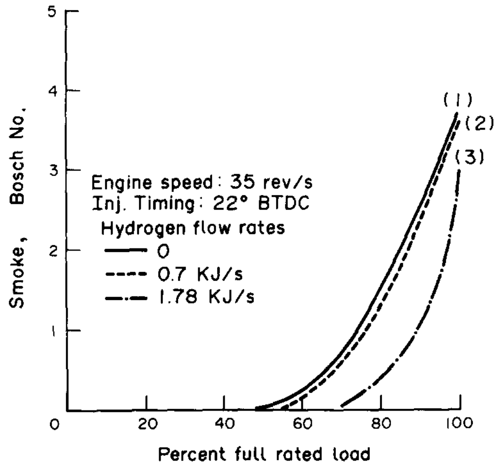

Fig. 10. Engine speed 35 rev/s, injection timing 22°BTDC. Hydrogen energy flow rates (1) 0, (2) 0.7 KJ/s, (3) 1.78 KJ/s.

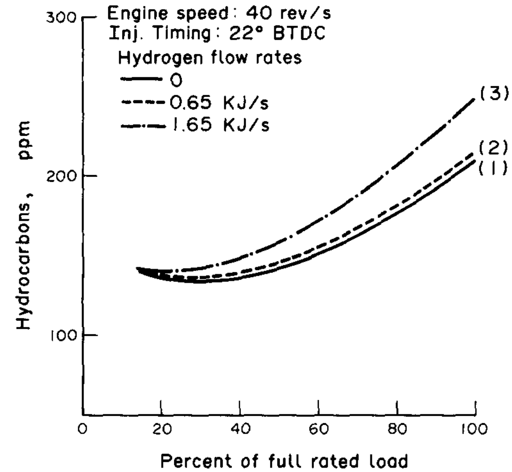

Fig. 12. Engine speed 40 rev/s, injection timing 22°BTDC. Hydrogen energy flow rates (1) 0, (2) 0.65 KJ/s, (3) 1.65 KJ/s.

on the exhaust NOx and HC emissions are shown in Fig. 11 for the two loads. Addition of hydrogen led to some increase in HC emissions at both the loads. This arises due to faster burning of hydrogen leaving the liquid fuel injected towards the end of injection period deficient in oxygen. Figure 12 shows how hydrocarbon emissions were influenced when the engine was run on fixed rate of hydrogen energy. At light loads there is hardly any difference in HC emissions with or without hydrogen but at high loads the effects of hydrogen addition are quite clear. This is certainly true as the hydrogen flow was increased. At low flow rates, such as 0.65 KJ/s, the deficiency in oxygen due to hydrogen combustion is minimum. Moreover, hydrogen does not burn as fast when its content in the mixture is low. Even at 1.65 KJ/s of hydrogen energy flow rate the absolute

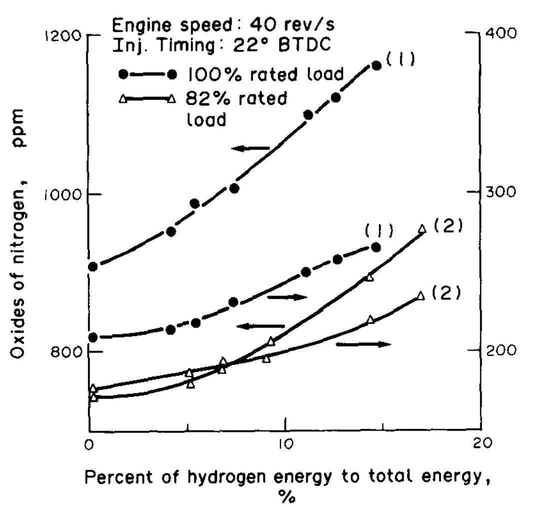

Fig. 11. Engine speed 40 rev/s, injection timing 22°BTDC. (1) 100% rated load, (2) 82% rated load.

increase in hydrocarbon emissions at full load was about 50 ppm.

Oxides of nitrogen increased faster than hydrocarbons as hydrogen content was increased, as shown in Fig. 11. NOx formed in a DI diesel engine depends on the local oxygen atom concentration, which is a function of the concentration of oxygen molecules and local temperature. Measurement of temperature in the combustion chamber of a DI diesel engine has shown the flame temperature for diesel fuel to be slightly lower than that for the hydrogen fuel at the same equivalence ratio [16, 17]. Addition of hydrogen can lead to higher local temperature earlier in the expansion process, resulting in rapid NOx formation rate for the fixed injection timing. This is true even at 82% full load, where the addition of hydrogen showed a similar increase in NOx even though the excess air ratio was higher than at full load operation.

Figure 13 shows the effects of constant hydrogen fuelling rate on NOx emissions. NOx levels did not change much from the baseline value for loads of up to 50% full rate load due to the presence of very lean mixtures in the combustion chamber. But as the load increased, local temperature as well as the overall mixture strength increased resulting in increased NOx formation. It may be possible to obtain NOx emissions at levels corresponding to the baseline value by retarding the injection timing and taking advantage of the fast burning of hydrogen mixtures. The variation in injection timing and its effects on NOx and smoke was not investigated in this study.

CONCLUSIONS

Small quantities of hydrogen supplied in the intake of the engine reduced smoke levels at part load operation while at the full rated load the reduction was limited. The best percentage of hydrogen energy for

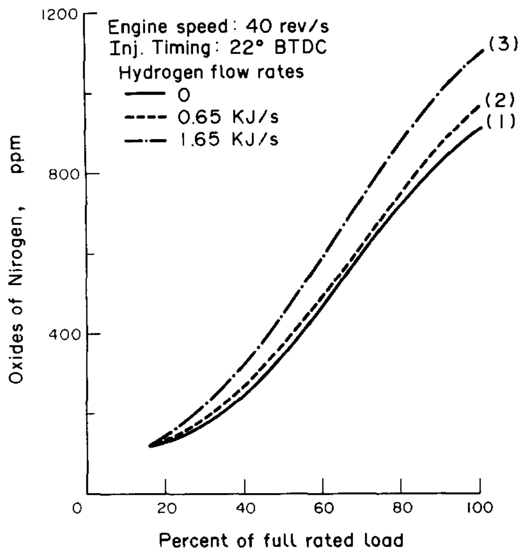

Fig. 13. Engine speed 40 rev/s, injection timing 22°BTDC. hydrogen energy flow rates (1) 0, (2) 0.65 KJ/s, (3) 1.65 KJ/s.

reducing smoke was found to be such that the time for the occurence of peak cylinder pressure should be about the same as the baseline value, if the engine is optimized with respect to the occurence of peak cylinder pressure. In the present study, optimum hydrogen percentage for smoke reduction was found to be between 10 and 15% of the total energy. Under optimum conditions, as much as 50% reduction in smoke levels were achieved at part load operation while at the full rated load the reduction was about 17%. The results of this study showed that minimum smoke density occured when the exhaust gas temperature was around the minimum value for a given engine load and speed.

Very low hydrogen flow rates had adverse effects on the engine thermal efficiency by marked improvements in efficiency were achieved by increasing the percentage of hydrogen supplied to the engine. Hydrocarbon emissions were not affected much by hydrogen fumigation but oxides of nitrogen in the exhaust increased with an increase in hydrogen energy, particularly for engine loads over 50% of the full rated load.

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