Improving the performance of biogas fuelled SI engines for rural use

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Abstract

Performance and combustion characteristics of a portable engine-alternator set fuelled with stimulated biogases are presented. It is shown that optimisation of the ignition timing can lead to substantial increases in maximum power. Thus for a gas of composition 70% CH_4/30% CO_2 the advancement of ignition from its original value of 14° before TDC (fixed, intended for petrol fuelling) to 29° before TDC results in an 11% increase in maximum power. With a gas of 45% CO_2 content the corresponding figures are 30% and 32°. Increasing the compression ratio (CR) from its original value of 6,4 to the region of 7 to 8 leads to a 3% increase in maximum power. Further raising of the CR results in decreased output. Supplementation of biogas fuels with petrol is very effective in combating the poor combustion, harsh running and low power output that are obtained with gases of high CO_2 content. Such dual fuelling is beneficial with biogases of all qualities and at all gas/petrol ratios, but particularly so with small quantities of petrol and very poor gases.

Background and aims

The work presented here is a continuation of a previous study, published in this journal, on the biogas-fuelling of a small spark ignition (SI) engine-alternator set [1]. The study was aimed specifically at rural African situations in which complex machinery, modifications and operating methods are to be avoided. These constraints continue to hold for the present study.

The composition of biogas usually lies within the following ranges: 50 to 70% methane (CH_4), 25 to 45% carbon dioxide (CO_2), 1 to 5% hydrogen (H_2), 0,3 to 3% nitrogen (N_2) and various minor impurities, notably hydrogen sulphide (H_2S); (all gas compositions in volume percent, throughout this paper).

The 'raw' gas may be 'scrubbed' to free it of CO_2, but this is generally too expensive for developing technology applications. The present work thus concentrates on raw biogas fuel.

Previous work

The previous study [1] employed a small side-valve engine of fixed ignition timing, coupled to an alternator. The engine was fitted with a standard venturi gas-fuel adaptor upstream of the liquid carburettor, but was otherwise unmodified. (The hardening of valves and valve seals, recommended in some quarters [2], was omitted with no resultant damage in the short term.) Biogases were simulated by mixtures of pure CH_4 and CO_2; the H_2 component, though considered to be of possible importance in view of its high burning velocity, was excluded. (A later study [3] showed this omission to have been of no serious consequence: maximum power output increased by less than 1% for each 1% H_2 content of the gas.) The mixture strength for each fuel was adjusted, as would be expected of the rural operator, by feel and ear. The resulting power outputs thus did not necessarily reflect the maximum capabilities of the unit, but rather those that could be reasonably expected in practice.

Maximum power was found to be 17% lower on fuelling with pure CH_4 rather than with petrol. Increased CO_2 content of the gas led to further losses with a 28% loss (compared with pure CH_4) at 41% CO_2. These findings were explained essentially in terms of the successive decrease in energy input per combustion cycle [1]. Not mentioned was the fact that the flame front velocity (at constant mixture strength) is lower for CH_4 than for petrol [4] and decreases further on dilution of CH_4 with CO_2 [5]. This means that, with fixed ignition timing, the peak combustion chamber pressure must become increasingly retarded on the successive replacement of petrol with CH_4 and with biogases of increasing CO_2 content. This was borne out in a subsequent study [3]: peak cycle pressures occurred at 4° after top dead centre (TDC) on petrol fuelling, at 17° after TDC on CH_4 fuelling and in the region of 34 to 42° after TDC on fuelling with a 50% CH_4/50% CO_2 gas. Clearly therefore the power output on gas fuelling can be increased by advancing the ignition timing.

The octane rating of CH_4 is stated to be 'about 130' [6] and its critical (knock-limited) compression ratio (CR) is 13 under conditions where that of iso-octane is 6,5 [7]. Dilution of the CH_4 with CO_2 must (as in the case of recirculation of exhaust gases to petrol-air mixtures [8]) increase these ratings. The power output of biogas-fuelled SI engines may thus, in principle, be considerably increased by raising the CR. Experimental results on this subject have been hinted at [1,2], but no systematic study specific to biogas fuels appears to be available.

In our firstmentioned study [1] the engine ran smoothly on gases containing up to 23% CO_2, somewhat noisily at 31% CO_2 and 'harshly' at 42% CO_2. Harshness appeared to be associated with poor and irregular combustion: cylinder peak pressures were low in amplitude, severely retarded and variable from cycle to cycle [3].

The engine could be hand-started from cold with gases containing up to 31% CO_2. At higher CO_2 contents starting on petrol and 'blending-over' to gas became necessary. During such blending-over the engine ran in the dual fuel manner, that is, simultaneously on petrol and gas. From this stemmed the idea of steady-state dual fuelling as a means of correcting the harsh running and low power

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output associated with biogas fuels of high CO_2 content.

Biogas-liquid dual fuelling of SI engines has been studied by Picken and Soliman [9], the gas having been supplemented by diesel oil and kerosene. The latter two, however, unlike petrol, cannot serve as sole fuels, as required in the event of failure of the gas supply. Picken [2] and Sharma [10] made mention of supplementation with petrol, but presented no experimental results. A detailed study of biogas/petrol dual fuelling was conducted by Jawurek et al [3], but the journal of publication is not freely accessible locally.

Specific aims

In view of the foregoing the specific aims of this study were defined as follows:

(1) To assess the improvement in maximum power that may be obtained by optimising the ignition advance of the engine-alternator set on fuelling with various biogases.

(2) To determine the performance of the gas-fuelled engine-alternator set as a function of compression ratio.

(3) To highlight the effect of petrol/biogas dual fuelling on the poor combustion and low power output that are obtained with gases of high CO_2 content.

In examining the above three points, the air-fuel ratio is to be adjusted for maximum power at each data point, thus eliminating an otherwise confusing variable.

Test facility and procedure

Engine and associated equipment

The engine was the same as that used previously [1], namely, a Briggs and Stratton 195400. This is a single-cylinder, governed, four-stroke, side-valve machine having a swept volume of 314,5 cm³. In standard form the CR was 6,4 and the ignition system was of the magneto type with an essentially fixed timing of 14° before TDC.

Three cylinder heads were skimmed to give increased CR's and a fourth was both skimmed and filled-in by welding. The use of two gaskets on the original cylinder head gave a reduced CR. The CR's that could thus be studied were: 5,5; 6,4; 7,2; 8,1; 9,0 and 10,5.

A modified motor cycle ignition system (of the 'coil, contact breaker and condenser' type) was retrofitted to the engine and the flywheel was marked in degrees of crankangle. By these arrangements the ignition timing could be varied (and measured) between 45° before TDC and 30° after TDC.

A Beam 1120B venturi-type 'gas carburettor' was fitted to the engine's air intake upstream of the petrol carburettor. The unit (previously described [1]) required a gas supply at ambient pressure and could be operated with or without simultaneous use of the liquid carburettor. Gas mixture strength could be set by a needle valve.

Directly coupled to the engine was a revolving field alternator having a 220 V output at a frequency of 50 Hz (3 000 rev/min).

Gas supply system

Simulated biogases were made up of CH_4 and CO_2 (occasionally also H_2) all of 99% or greater purity. The gases were supplied from high pressure cylinders and fed to a gas mixer unit manufactured to our requirements by Witt-Gastechnik, Germany. The unit required merely the entering of the desired gas composition and then automatically metered, mixed and supplied the gas at a pressure of 120 kPa (absolute). The gas was reduced to ambient pressure by two Beam 52B demand regulators arranged in parallel (this ensured sufficient gas capacity [3]) and fed to the gas carburettor.

Measurement of engine input and output

The mass flow rates to the engine of air, gas fuel and petrol were determined as previously described [1]. The output of the alternator was dissipated in a step-switchable resistance box and a slide rheostat; power output was measured by means of a wattmeter, and frequency (speed) by a digital tachometer. Ignition timing was determined by stroboscope to a resolution of about 1° crankangle.

Measurement of combustion chamber pressure

The cylinder heads of the engine were drilled and fitted with a Kistler 9121A piezo-electric pressure transducer enclosed in a water cooling jacket (or a blanking-off plug).

The output of the transducer was amplified and captured by a transient recorder having a 10 bit resolution. Pressure versus time traces of various durations could be obtained by varying the sampling rate of the recorder. The records were displayed on an oscilloscope and copied using an XY recorder, or stored by microcomputer for subsequent processing and plotting.

The transient recorder was triggered with the piston at bottom dead centre (BDC), by means of an optocoupler-interrupter device. BDC signals were recorded in parallel with transient pressure.

Crankangle positions were estimated for the pressure versus time records by linear interpolation between the BDC marks. The accuracy of this procedure is dependent on the steadiness of engine speed during the cycle.

Test procedure

In our previous study [1] the complete power versus speed (frequency) curve was recorded for each fuel. The curves demonstrated the response of the (governed) engine-alternator set to variations in load. Mixture strength was adjusted subjectively. In the present study maximum power only was sought, but with mixture strength optimally adjusted for each fuel. Thus the engine was started, loaded to give full throttle opening, and fine-tuned by alternately varying the load rheostat and the mixture screw, to the point of maximum watt-meter deflection. In those cases where ignition timing was to be optimised the fine-tuning procedure was extended to include the incremental adjustment of the advance-retard mechanism. The settings giving maximum power were generally rather difficult to detect unambiguously and this is reflected in the scatter of the reported data points.

Results and discussion

Three groups of experimental results are presented here, dealing – in turn – with the effects of ignition timing, compression ratio and dual fuelling on engine performance. The data points within each group were gathered in rapid succession and with conditions, equipment and techniques held constant (except for the parameter being studied). This consistency could, however, not be maintained from group to group (including that previously reported [1]). The studies were widely spaced in time, two engines were employed which, though nominally identical, differed in performance when new, and were not necessarily at the same state of wear (or overhaul) when tested. Finally, some minor changes were made to the gas supply system and to its method of operation. Thus the comparison of data points is valid only within each group.

Ignition timing

Figure 1 shows the performance of the engine-alternator set on fuelling with various mixtures of CH_4 and CO_2, firstly, with standard ignition timing of 14° before TDC (open data points), and secondly, with the ignition timing optimised for maximum power (square data points).

Figure 1(a) shows the increases in maximum power resulting from the optimisation of ignition timing to have been 3% for pure CH_4, 11% for a 70% CH_4/30% CO_2 fuel (representative of an average biogas), and some 30% for a 55% CH_4/45% CO_2 mixture (a 'poor' biogas). The ignition timing that brought about this improved performance, shown in Figure 1(b), varied approximately linearly from 24° before TDC for pure CH_4 to 32° before TDC for a gas of 45% CO_2 content.

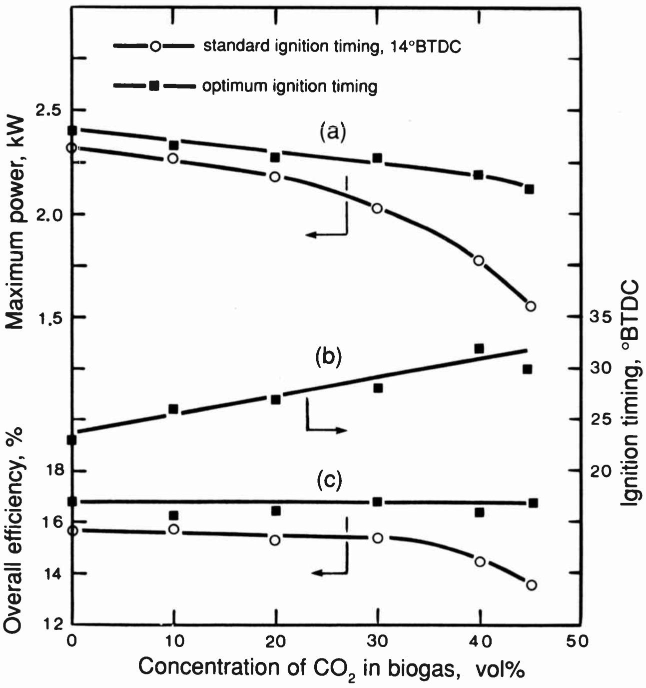

Figure 1. Effect of ignition timing on engine-alternator performance on fuelling with various biogases.

Figure 1(c) shows the overall efficiency of the unit (electrical output divided by calorific input) to have been consistently higher at power-optimised ignition timing than at the fixed value of 14° before TDC.

Figure 2 shows combustion chamber pressure versus time (crankangle) for a 70% CH_4/30% CO_2 mixture with ignition at 14° before TDC (dotted line) and at 30° before TDC (solid line). (The pressure traces were recorded separately from the main tests discussed here hence the discrepancy between the value 30° before TDC and that of 29° shown in Figure 1(b)). The benefits of advancement of the ignition timing are evident: the peak cycle pressure occurred at a more favourable crankangle (14° rather than 28° after TDC), its amplitude was increased (by some 50%), and the mean effective pressure was higher.

Figure 3(a) shows the frequency (speed) of the unit to have remained essentially constant at 47,5 Hz, irrespective of ignition timing, up to a concentration of 40% CO_2 in the fuel. At 45% CO_2 however, the frequency is seen to have dropped sharply with standard ignition timing, but to have remained at its previous value with advanced timing. This fall-off in speed, at CO_2 concentrations above 40%, has been noted previously (with standard spark timing) and was found to have been associated with a reduction in fuel gas flow rate to the engine [3]. It was further found that at such high concentrations of CO_2 the mixture strength could no longer be optimally adjusted, maximum power being achieved with the mixture screw fully open. The engine was thus starved of fuel. It was shown that this problem could be alleviated to some extent by the bypassing of the demand regulators and by slight pressurisation of the gas supply. (The pressurisation can be extended to no more than 10 to 20 kPa gauge; further increases in pressure lead to sudden cutting-out of the engine, owing to displacement of air and the subsequent aspiration of an unignitably rich mixture.) The tests reported in Figures 1 and 3 were conducted with such pressurisation of the gas supply.

Figure 3(b) shows the resulting gas fuel flow rate to the engine (CH_4 plus CO_2, normalised to 20 °C, 101 kPa). Clearly the fall-off in frequency shown in Figure 3(a) (dotted line) cannot be attributed to a fuel starvation problem.

Figure 3(c), however, shows the air flow to the engine to have dropped sharply in te region under discussion (dotted line). Insufficiently advanced (standard) ignition timing thus leads to a reduction in the **total** (fuel plus air) breathing capacity of the engine at high concentrations of CO_2.

Figure 2 offers a possible explanation for this behaviour. The combustion chamber pressure in the latter stages of the power stroke and in the first third of the exhaust stroke is seen to be higher on standard spark timing than with correctly advanced timing. Some such difference in pressures, even if greatly attenuated, must be expected to persist throughout the exhaust stroke and thus to the start of the period of valve overlap. Hence the reverse flow of residual 'exhaust' gases into the inlet manifold – and thus the displacement of potentially aspirable combustibles – is greater with standard (late) ignition timing than with advanced timing. It would seem that this effect significantly reduces the total flow rate of gases to the engine when the mixture becomes sufficiently

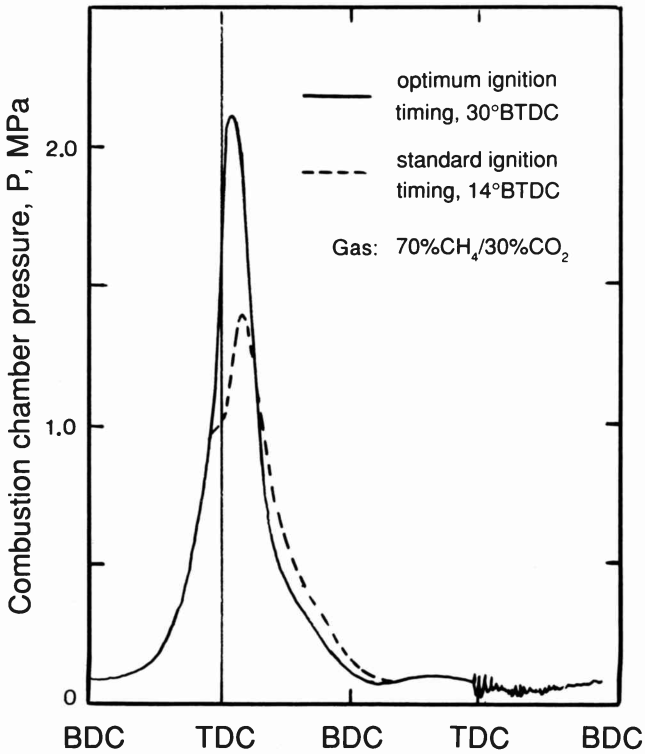

Figure 2. Effect of ignition timing on combustion chamber pressure, P. (BDC and TDC signify bottom dead centre and top dead centre, respectively.)

slow-burning, in the present case at a CO_2 content of 40%.

Finally, Figure 3(d) shows the variation of equivalence ratio, ϕ, with CO_2 content of the biogas for the two cases

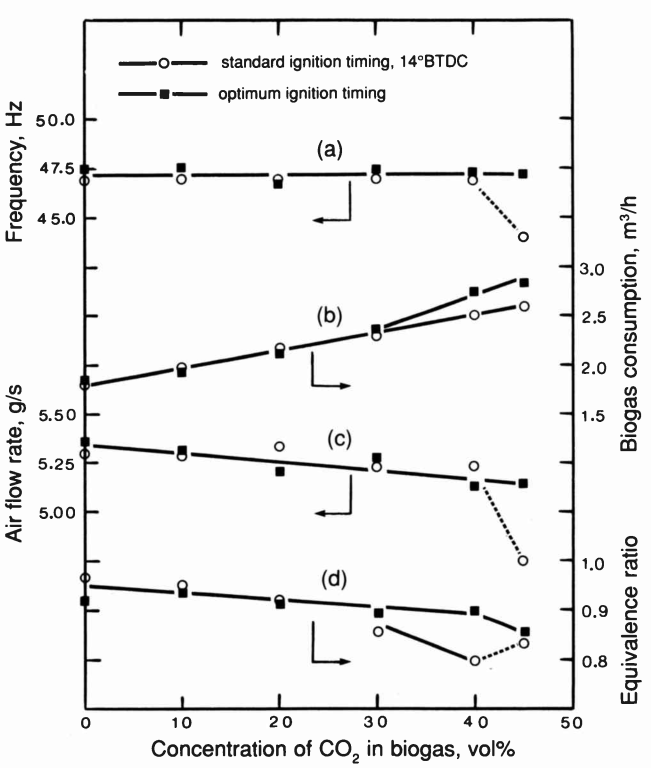

Figure 3. Effect of ignition timing on speed, biogas consumption, air flow rate to engine, and mixture strength, with various biogases.

of ignition timing. The equivalence ratio ϕ is the ratio of stoichiometric to actual air-fuel ratio, where fuel refers to combustibles only (here CH_4). The ratio ϕ is thus a measure of mixture strength with $\phi = 1$ signifying a mixture that is stoichiometric, $\phi < 1$ one that is lean, and $\phi > 1$ one that is rich. It will be noted that the values of ϕ were in the lean region throughout and fell with increasing CO_2 content of the gas. This is contrary to normal (liquid fuel) experience, according to which maximum power occurs with rich mixtures (typically $\phi \doteq 1{,}2$).

The occurrence of maximum power with lean mixtures (on gas fuelling) has been previously studied in some detail, but remains incompletely understood [3]. It was shown, however, that with CO_2 concentrations of 0 to 30% the lean mixtures were truly optimal (richer mixtures giving reduced power outputs). At CO_2 levels of 40% and greater, on the other hand, the engine was starved of fuel (maximum power corresponding to the maximum rate of fuel flow that could be sustained). The latter problem was at least partially overcome in the present group of tests by feeding the fuel under slight positive pressure. Thus the fall-off in equivalence ratio at high CO_2 concentrations was less severe than that obtained previously [3].

Figure 3(d) further shows the equivalence ratio at CO_2 concentrations beyond 30% to have been higher with optimal ignition timing than with standard timing. This was due to the higher fuel flow rate in the former case (Figure 3(b)), which in turn was due to the slightly higher gas supply pressure that could be maintained. (The reasons for this behaviour might well lie with the reverse flow phenomenon, discussed earlier.) The sudden increase in equivalence ratio between 40 and 45% CO_2 with standard ignition timing is clearly due to the reduction in air flow rate in this region (dotted line, Figure 3(c)), also previously discussed.

It must be stressed that the feeding of biogas under positive pressure was undertaken here purely for purposes of studying the fuel starvation problem. The technique is not recommended for practical implementation. An engine thus operated runs on the brink of cutting-out (as explained earlier) and such an event if undetected, results in the liberation of biogas to the surroundings and the formation of a potentially explosive mixture. In general, demand regulators should be retained. These shut off the gas supply in the absence of reduced pressure ('vacuum') in the gas carburettor.

While the benefits of optimisation of ignition timing of biogas-fuelled SI engines are evident from the foregoing, the feasibility of its implementation in developing technology situations is dependent on engine type. For small engines of non-variable ignition timing it would probably be best to effect a once-off modification to a fixed advance of some 27° before TDC. Casual tests on our engine with such timing gave powers close to maximum for 'reasonable' biogases (25 to 40% CO_2). Rough running, however, persisted at CO_2 concentration of 40% and greater. The performance on petrol fuelling with such advanced spark timing was also satisfactory.

Compression ratio

In the tests reported in this section the ignition system of the engine was in its original form (fixed spark timing of

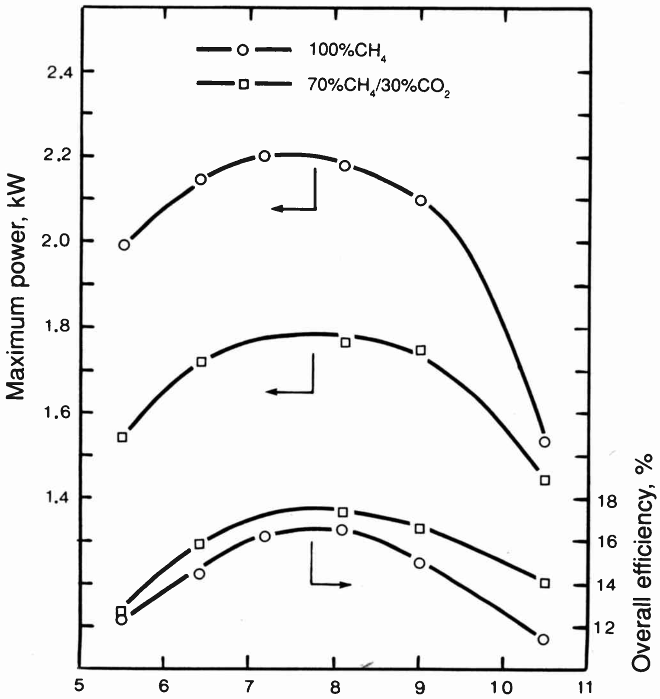

Figure 4. Effect of compression ratio on performance.

14° before TDC), and the gas supply was operated normally (non-pressurised) with the demand regulators installed.

Figure 4 shows the performance of the engine as a function of compression ratio (CR), on fuelling with two fuels, namely, pure CH_4 and a 70% CH_4/30% CO_2 mixture. The increase in maximum power that could be achieved by raising of the CR from its standard value of 6,4 was in both cases a mere 3%, the optimum CR being in the region of 7 to 8.

This optimum was initially thought to be the 'highest useful compression ratio' (HUCR) as defined by Ricardo [11], that is, the CR corresponding to the onset of knock. This notion is almost certainly incorrect. True knock (autoignition of the combustible mixture ahead of the flame front) is ruled out in the present case in view of the autoignition properties of CH_4 and, even more so, CH_4/CO_2 mixtures [6-8] and of the relatively low prevailing CR's. Further, the engine was not **heard** to have knocked, even on CH_4 fuelling at the highest CR. The observed behaviour is more likely to have been associated with non-knock pressure fluctuations in the combustion chamber. Pressure measuring equipment was unavailable in the present tests. Previous measurements on CH_4 fuelling at a CR of 6,4 (Figure 2(b) in [3]), however, showed 'spikey' pressure oscillations to have occurred immediately after the attainment of maximum cycle pressure. It would seem reasonable to expect these oscillations to grow with increasing CR, ultimately to the point where the associated dissipative losses outweigh the benefits of the raised CR. The particularly sharp fall-off in maximum power with CH_4 fuelling at CR's beyond 8, might additionally have been due to non-optimal (in this case, overadvanced) ignition timing.

The overall efficiencies shown in Figure 4 show the same trend with CR as does maximum power. This is consistent with the combustion chamber events suggested above. Higher efficiencies were obtained with CO_2-diluted fuel than with pure CH_4. This behaviour is normal (for CO_2 concentrations of 0 to 50%) with conventional operation of the fuel supply system [3]; the behaviour shown in Figure 1(c) (open data points) was due to the altered method of fuelling.

Dual fuelling

The low powers, over-lean mixtures and rough running that are obtained with biogases of high CO_2 content could all be corrected by simultaneous fuelling with petrol.

Such dual fuel operation was achieved as follows: the engine was started and allowed to warm up on petrol. The needle valve fitted to the petrol carburettor was shut off, the float chamber was allowed to empty, and at first faltering of the engine the gas supply was opened. Petrol was now readmitted (as required) and the gas mixture strength was readjusted for maximum power.

Figure 5 shows results, thus obtained, for a fuel gas of 45% CO_2 and 2% H_2 content. Maximum power is plotted as a function of petrol content of the fuel, the latter given as mass percent of the total combustibles (CH_4, H_2 and petrol). The entire range of fuels extending from pure gas to pure petrol was covered. The admixture of petrol to the biogas resulted in improved maximum power in all cases, but most markedly so at low percentages of petrol. The equivalence ratio increased from 0,74 on gas-only fuelling, to unity at 40% petrol, and to 1,28 on all-petrol fuelling.

The poor combustion of fuel gases of high CO_2 content was alleviated even by small admixtures of petrol. Figure

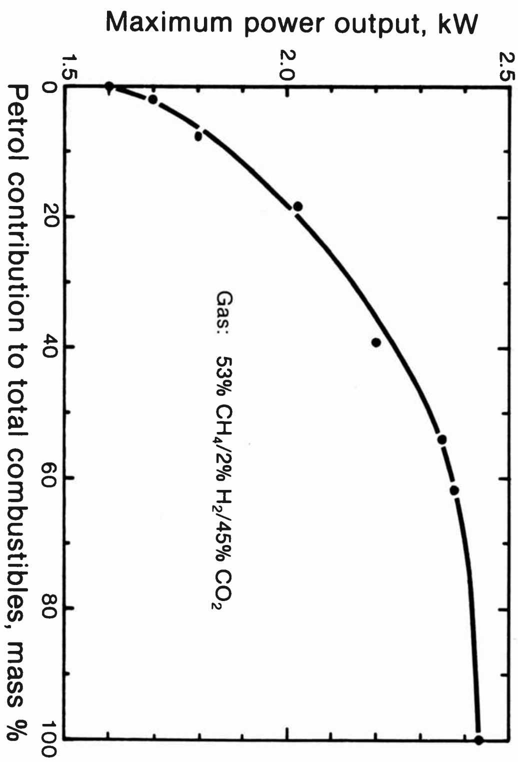

Figure 5. Biogas/petrol dual fuelling.

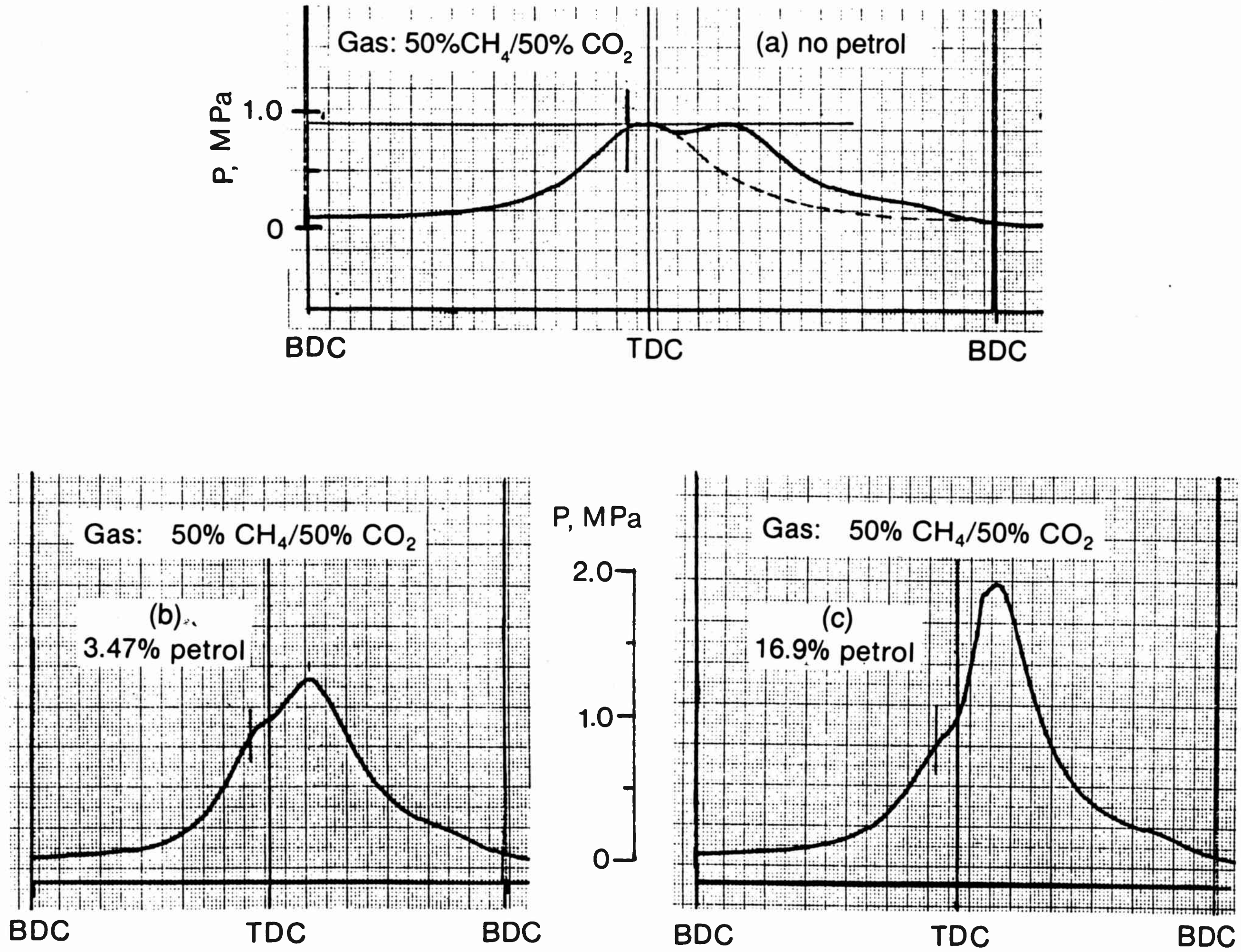

Figure 6. Effect of biogas/petrol dual fuelling on combustion chamber pressure.

6 shows combustion chamber pressure versus time (crankangle) on fuelling with a 50% CH_4/50% CO_2 gas (a) without additional petrol (b) together with 3,47% of petrol, and (c) with 16,9% petrol (mass percent of total combustibles). Supplementation of the gas with petrol clearly led to greatly improved pressure rise and peak pressure. Subjectively, the engine was perceived to have run 'sweetly' on such dual fuelling, with no apparent harshness.

Dual fuelling, however, holds benefits beyond the increase in maximum power and the prevention of rough running. Since operation at any gas/petrol ratio is possible, even small quantities of gas, insufficient for normal gas fuelling, can be utilised in a blend with resultant savings in petrol.

In rural situations the optimisation of mixture strength for maximum power will not generally be possible nor necessary. More likely, engine performance will be checked casually on gas fuelling, and if unsatisfactory, will be improved by a suitable admixture of petrol. This simply entails the opening and adjustment of the petrol needle valve until the engine sounds 'right' at the required load. Such tuning by ear is quickly learned and leads to power outputs differing not greatly from those achieved by quantitative tuning procedures.

Conclusions

These refer to the engine of the present study and, more broadly, to engines of similar size and design, all with the mixture strength adjusted for maximum power for each fuel or fuel mixture.

Substantial increases in maximum power may be obtained by optimising the ignition timing of biogas fuelled SI engines. For a typical biogas fuel of 30% CO_2 content the advancement of the ignition timing from its original (fixed) value of 14° before TDC to the optimal setting of 29° before TDC results in an 11% increase in maximum power. The improvement in maximum power and the optimal spark advance both increase with increasing CO_2 content of the gas, being some 30% and 32° before TDC, respectively, for a gas of 45% CO_2 content.

The flame front velocity (at constant mixture strength) is lower for CH_4 than for petrol and is lower still on dilution of CH_4 with CO_2. The appropriate advancement of the ignition timing (on switching from petrol to biogas fuels) allows the peak combustion chamber pressure to occur at a more favourable crankangle and with increased amplitude. The above improvements in performance are largely due to the corresponding increase in mean effective pressure. Optimisation of the ignition timing also improves the overall breathing capacity of the engine (at least with fuels of high CO_2 content). This appears to be due to the reduction in reverse flow of exhaust gases into the inlet manifold during the period of valve overlap.

Little is to be gained from the raising of the CR on gas fuelling of SI engines of the type here studied. An increase in CR from the original value of 6,4 to some 7 to 8 leads to an increase in maximum power of a mere 3%. Further raising of the CR results in reductions in both maximum power and thermal efficiency. This behaviour appears to be due to the progressive intensification of dissipative losses arising from non-knock pressure fluctuations in the combustion chamber.

The low power output, poor combustion and harsh running that are obtained with biogas fuels of high CO_2 content can be corrected by supplementing the gas with petrol. Such dual fuelling is beneficial for all biogases, but particularly so with low to moderate quantities of petrol and with very poor fuels. For example, for a gas of 45% CO_2 content, the admixture of 15% petrol (mass percent of total combustibles) leads to a 27% increase in maximum power. Thus even minor amounts of gas – insufficient for all-gas fuelling – can be used with resultant savings in petrol.

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