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# Performance Evaluation by Heat Release Method for a Diesel Engine on Producer Gas

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## ABSTRACT

*A three-cylinder, diesel engine was tested at a constant speed of 1550 rpm on dual fuel mode of operation. The conventional diesel oil was substituted by producer gas from a downdraft charcoal gasifier. The engine's performance for this fuel was analysed by a heat release evaluation program and the results are presented. A maximum of 62% of the total heat input was replaced by the producer gas at an indicated power output of 20.6 kW. When 47% of the total heat input was supplied by the producer gas, the indicated specific fuel consumption of the engine went up by a maximum value of 43%, at an indicated power output of 14.02 kW. The indicated thermal efficiency values of the engine were generally lower and the total heat input to the engine was always higher on the producer gas operations. At the indicated power output of 20.6 kW, the maximum values of the average temperature of the diesel and producer gas mixtures, at different flow rates of the producer gas, were lower than that in the pure diesel operation. At higher flow rates of producer gas, the ignition delay periods increased considerably and the values of the fraction of the total chemical energy of the combustible mixture that was converted into useful heat were lower. The results are discussed in detail.*

## INTRODUCTION

The producer gas is obtained by converting the carbon, present in solid fuels, into a gaseous form of carbon monoxide, by partial combustion. The sensible heat liberated during the gasification process is used to dissociate the water vapor present in the fuel into hydrogen and oxygen. Since the invention of gasifiers, producer gas is being used to drive internal combustion engines for producing mechanical/electrical power. Gasifier powered automobiles were in use during the second world war, but due to the bulkiness of the gasifier and the lower cost of conventional fuels, their use dwindled. However the use of gasifiers is becoming popular in developing countries to fuel stationary I.C. engines coupled to small scale power generators or water pumping units. This is because of the increasing burden of the oil import bill, availability of the cheap biomass locally and the alarming rate of depletion of conventional fuels.

The demands are greater on the gasifier for engine operation with respect to the combustion properties and cleanliness of the gas and therefore an analysis of the combustion characteristics of the producer gas inside the combustion chamber of the I.C. engine which will lead to a better understanding of the mutual acceptance of these two systems will be extremely useful.

This article describes the combustion pattern of the mixture of producer gas and diesel in a diesel engine. A downdraft charcoal gasifier was used to run the diesel engine on dual-fuel mode of operation, i.e. a small quantity of diesel oil was used to start the ignition process. This work was carried out at the Energy Technology Division, Asian Institute of Technology, Bangkok.

## EXPERIMENTAL SET-UP

A schematic diagram of the experimental set-up is shown in Fig. 1 and the functions of the different components of this set-up are presented below:

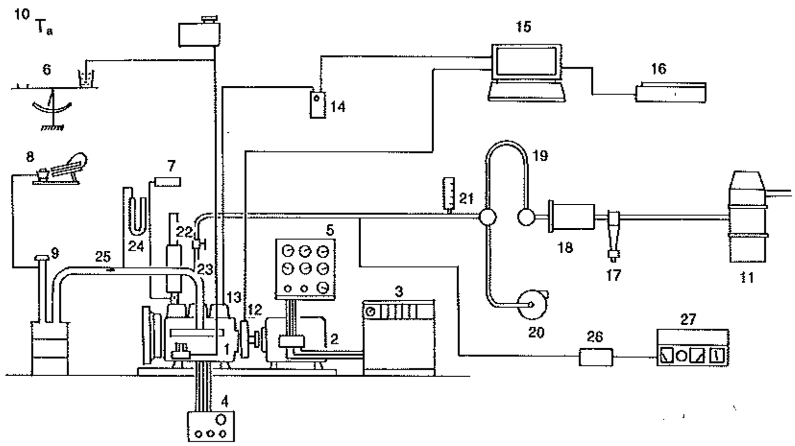


Fig. 1. Schematic diagram of the experimental set-up.

| | | |
|---|---|---|
| 1. Diesel engine | 2. AC generator | 3. Resistor bank |
| 4. Engine instrumentation | 5. Generator instrumentation panel | 6. Balance |
| 7. Digital temperature read-out | 8. Inclined tube manometer | 9. Venturi |
| 10. Ambient temperature | 11. Gasifier | 12. Opto-coupler sensor |
| 13. Piezoelectric transducer | 14. Charge amplifier | 15. aScope-Apple II computer |
| 16. Plotter (or) printer | 17. Cyclone | 18. Cloth filter |
| 19. Heat exchanger | 20. Starting blower | 21. Gas flow-meter |
| 22. Gas control valve | 23. Air intake elbow | 24. U-tube manometer |
| 25. Throttle valve | 26. Gas filter and | 27. BINOS - Gas analyser |

### Gasifier

A downdraft, charcoal gasifier was used to fuel the diesel engine. Based on the recommended maximum and minimum permissible gasification rates 1.0-0.3 standard $m^3/cm^2$ (Kaupp et al., 1984), the throat diameter of the gasifier should be between 9.2 to 16.8 cm. However, the throat diameter of the gasifier used in the experiment was 8 cm. Because of this size limitation, the engine's full load gas requirement could not be met by the gasifier. A smaller throat diameter would also result in a poor quality producer gas due to charcoal dusting in the combustion zone of the gasifier. This necessitated a good gas cleaning system. A blower provided the necessary suction for starting the gasifier. The producer gas was admitted into the engine by throttling the engine. The gasifier is shown in Fig. 2.

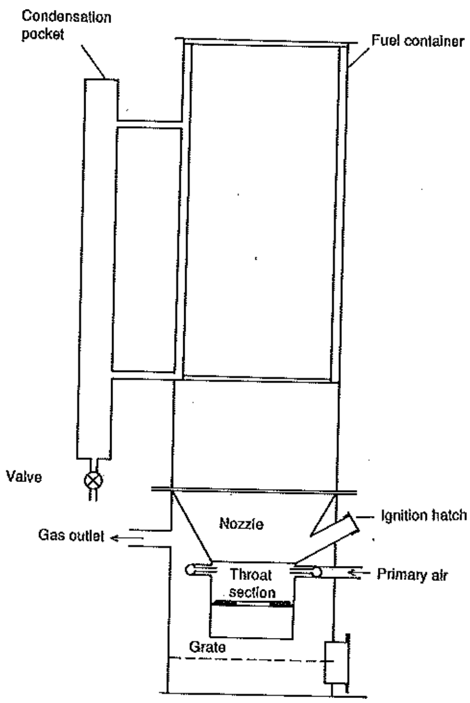


Fig. 2. Downdraft charcoal gasifier.

## Producer Gas Cleaning and Cooling

To effectively remove dust in the producer gas, the gas was first passed through a cyclone. A cloth filter removed the fine dust from the gas. Since the engine is a constant volume machine, the gas should be cooled to offset the loss in the volumetric efficiency of the engine. Because the producer gas is a low calorific value fuel, on average, it occupies half the volume of the air/gas mixture drawn in by the engine. An air cooled heat exchanger was used to reduce the temperature of the gas before it entered the engine.

## Engine and Loading

The diesel engine which was used for the experiments had the following specifications:

| | | |
|---|---|---|
| Type | : | Open chamber |
| No. of cylinders | : | Three |
| Type of cooling | : | Water cooling |
| No. of strokes | : | Four |
| Max. speed | : | 3000 rpm |
| Max. power | : | 25 kW |
| Bore size | : | 110 mm |
| Stroke | : | 110 mm |

| | | |
|---|---|---|
| Compression ratio | : | 17 |
| Total displacement volume | : | 3.316 litres |
| Injection timing | : | 34 degree CA before top dead centre |
| Make | : | Kirloskar (India), RBV-3 |

The diesel engine was connected to a single phase/three phase 230 V/380 V AC generator. The engine speed was maintained at a desired level by means of a mechanical speed governor, which acted on the control rack of the diesel injection pump.

### Instrumentation

The pressure drop in the engine's air intake manifold, which was induced by throttling the engine, was measured by a manometer. The flow rate of the producer gas was measured by a flow cell. Different constituents of the producer gas and the engine's exhaust gas were analysed continuously by a BINOS on-line gas analyser. The engine's air-intake was calculated from the measured pressure drop across a venturi. The pilot diesel fuel consumption rate was arrived at, by noting the time taken for the consumption of a known quantity of diesel fuel. The different temperatures of intake air, producer gas, exhaust gas and cooling water were displayed on a digital read-out. An engine-generator control panel housed the necessary displays to measure the various operating parameters of the experimental set-up. The resistor load bank had provisions to load the engine in steps.

## INDICATOR DIAGRAMS

When testing an I.C. engine on any fuel, the chemical energy input and the mechanical energy output can be measured easily. But, these do not give a clear picture of the combustion pattern of the fuel inside the combustion chamber of the engine. Once the fuel's combustion pattern is known, ways to reduce the losses that occur during the combustion can be discovered. The combustion characteristics can be obtained by analyzing the pressure change inside the combustion chamber. By knowing the pressure course of the combustion chamber, the complete heat release characteristics of the fuel can be evaluated. In this present study the complications involved in the heat release evaluation procedure which was developed by Hohenberg et al. (1982) were reduced and a simple procedure was proposed. This procedure can be used for any type of alternative or conventional fuels used in the I.C. engines. For all these anlyses the basic data required is the pressure versus crank angle diagrams, i.e. the indicator diagrams of the engine.

### Method and Instruments in Obtaining Indicator Diagrams

A piezo electric transducer (Kistler Inst., 1985) was used to pick up the pressure signals from the combustion chamber. A charge amplifier gave an output voltage proportional to the electrostatic charge produced by the pressure transducer. Calibration was necessary to determine the relationship of pressure input to voltage output and this was discussed in detail by Lancaster et al. (1975). The output voltage from the charge amplifier was fed into an oscilloscope cum personal computer (Instruments Systems, 1983). This "Model 85 aScope" stored the indicator diagrams on a computer diskette. Top Dead Center (TDC) position of the piston was marked by an optocoupler and this signals were also fed into one of the two channels of the "aScope". A simple computer program retrieved the data stored on the diskette and converted them into pressure versus crank angle values. These values were corrected for the zero thermodynamic line (Hohenberg et al., 1982). These corrected values formed the base data for finding the heat release pattern of the fuel. The methodology of heat release evaluation and the computer program have been discussed in detail by Asokan (1986) and Kreepong (1986).

## PRESSURE VS. CRANK ANGLE DIAGRAMS WITH AND WITHOUT COMBUSTION

A sample of the indicator diagrams with and without combustion is shown in Fig. 3. It can be seen that the pressure course of "with combustion" deviated from the compression course of the diesel engine "without combustion". The starting point of deviation was an indication of the starting of combustion. Both the pressure courses followed the same path at the beginning of the compression and at the end of exhaust strokes. The beginning of the optocoupler mark was two degree crank angle before TDC. The square wave form shown in the figure is useful for finding the crank angle position corresponding to the pressure values, when the speed of the engine is known.

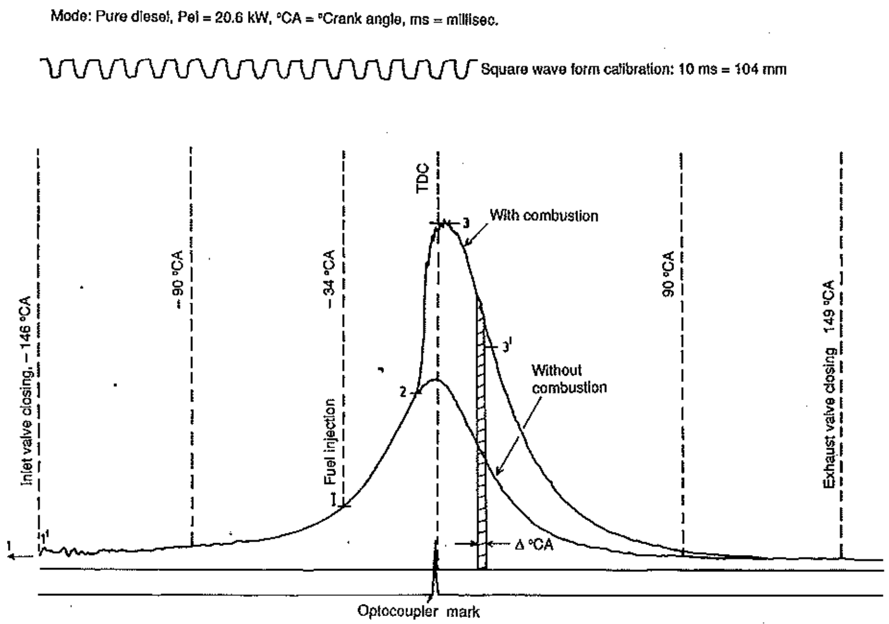


Fig. 3. Indicator diagram of a diesel engine with and without combustion.

| | |
|---|---|
| 1 = Start of compression | 2 = Beginning of heat release |
| 1' = Inlet valve closing position | 3 = Peak pressure |
| I = Beginning of fuel injection | 3' = End of useful combustion |

## EXPERIMENTAL PROCEDURE

Charcoal was charged into the gasifier, upto the top of the throat section. Keeping the lid of the gasifier partially closed, the starting blower was switched on and a torch was introduced. After a few minutes, when the charcoal was burning evenly, the torch was withdrawn and the gasifier was filled to the top with charcoal. With the engine already running on pure diesel mode, the starting blower was switched off and the gas intake valve of the engine was opened. When the engine was throttled, it started consuming the producer gas. The quantity of the producer gas drawn in by the engine was controlled by adjusting the gas intake valve and the throttle valve.

The engine was run at a constant speed of 1550 rpm at different loads, for different quantities of producer gas. The diesel oil injection timing was kept constant at 34 degrees before TDC (Kaupp et al., 1982). Before taking any readings, the engine was allowed to reach steady state, at that particular load and gas flow rate. All the operating parameters like indicator diagrams, various temperatures, percentage of gas constituents, etc., were recorded. On the same personal computer, indicator diagrams and other necessary details were used as the basic data for the heat release evaluation-computer program to analyse the performance of the system.

## RESULTS AND DISCUSSION

Carbon monoxide (CO), hydrogen ($H_2$) and methane ($CH_4$) are the main combustible constituents of the producer gas. When the gasifier is operated on charcoal, due to charcoal's relatively consistent composition, the variation in gas quality during the engine operation could be expected to be small. The producer gas obtained from charcoal consists of 26 to 29% CO, 3 to 4% $H_2$, 1 to 3% gaseous hydrocarbons and the remaining quantity is nitrogen (Naksittle et al., 1982). To start a spark ignition engine on the producer gas, 20% of CO and 3 to 5% of $H_2$ in the producer gas is sufficient (Reines, 1987). Table 1 presents the results of the tests reported in this section.

### Engine Performance using Producer Gas

The engine performance characteristics using producer gas, are presented in Figs. 4 and 5. On pure diesel mode of operation, generally the engine is not throttled, instead it is controlled by controlling the quantity of diesel injected into the engine. The amount of air drawn in by the engine will be in excess on pure diesel mode. Whereas in throttling operation, the pressure drop in the intake manifold increases and the amount of air supplied to the engine goes down. So the air fuel ratio decreases with the increasing quantity of producer gas as shown in Fig. 4. The total heat input to the engine increases with an increasing quantity of producer gas. This is due to the increasing losses, because of the delayed combustion in the expansion stroke, with the increased addition of producer gas.

Generally, the indicated specific energy consumption (fi) was higher and the indicated thermal efficiency (ni) was lower on the dual fuel mode of operation. Increase in "fi" was much higher at lower electrical loads than at higher loads. Similarly, decrease in "ni" was higher at lower electrical loads. This trend is shown in Fig. 5. When the heat input from the producer gas is in the range of 36% to 47%, the decreasing trend in "ni" and the increasing trend in "fi" are reversed. Increase in "fi" is due to the lower flame propagation velocity of the producer gas

**Table 1. Summarised data for a diesel engine running on producer gas.**

| S. No. | Vgas $m^3$/hr | % Heat added by prod. gas | F kg/hr | $Q_{diesel}$ MJ/hr | $Q_T$ MJ/hr | Ma kg/hr | Phi | fi MJ/kW – hr | ni % | fe MJ/hr | ne % | nv % | Te °K |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 |
| Run No. 1 | Pi = 14.02 kW, Pe = 8.9 kW, Pel = 6.9 kW, Pfr = 5.12 kW N = 1550 rpm nm = 63.48%, Pi = 3.415 bar and Pe = 2.168 bar. | | | | | | | | | | | | |
| 1. | 0.0 | 0.00 | 3.211 | 134.85 | 134.85 | 153.09 | 3.29 | 9.618 | 37.43 | 15.152 | 23.76 | 93.84 | 538 |
| 2. | 10.0 | 18.51 | 3.146 | 132.12 | 162.12 | 155.17 | 2.91 | 11.563 | 31.13 | 18.215 | 19.76 | 95.12 | 540 |
| 3. | 16.0 | 27.55 | 3.006 | 126.24 | 174.24 | 152.26 | 2.73 | 12.428 | 28.97 | 19.578 | 18.39 | 93.52 | 543 |
| 4. | 20.0 | 33.00 | 2.900 | 121.82 | 181.82 | 151.56 | 2.64 | 12.963 | 27.76 | 20.429 | 17.62 | 92.91 | 546 |
| 5. | 25.0 | 39.92 | 2.688 | 112.88 | 187.88 | 146.59 | 2.52 | 13.401 | 26.86 | 21.110 | 17.05 | 89.86 | 551 |
| 6. | 30.0 | 46.77 | 2.439 | 102.42 | 192.42 | 138.38 | 2.36 | 13.725 | 26.23 | 21.620 | 16.65 | 84.83 | 570 |
| 7. | 34.0 | 55.64 | 1.936 | 81.33 | 183.33 | 119.37 | 2.20 | 13.076 | 27.53 | 20.599 | 17.48 | 73.17 | 589 |
| Run No. 2 | Pi = 16.12 kW, Pe = 11 kW, Pel = 7.8 kW, Pfr = 5.12 kW N = 1550 rpm nm = 68.24%, Pi = 3.926 bar and Pe = 2.679 bar. | | | | | | | | | | | | |
| 1. | 0.0 | 0.00 | 3.680 | 154.55 | 154.55 | 148.77 | 2.79 | 9.587 | 37.55 | 14.050 | 25.62 | 91.19 | 582 |
| 2. | 10.0 | 16.50 | 3.615 | 151.82 | 181.82 | 148.55 | 2.47 | 11.279 | 31.92 | 16.529 | 21.78 | 91.06 | 585 |
| 3. | 20.0 | 30.00 | 3.333 | 140.00 | 200.00 | 142.99 | 2.24 | 12.407 | 29.02 | 18.182 | 19.80 | 87.66 | 589 |
| 4. | 26.0 | 38.13 | 3.013 | 126.55 | 204.55 | 131.43 | 2.06 | 12.689 | 28.37 | 18.595 | 19.36 | 80.57 | 596 |
| 5. | 30.0 | 43.68 | 2.763 | 116.06 | 206.06 | 124.57 | 1.97 | 12.783 | 28.16 | 18.733 | 19.22 | 76.86 | 614 |
| 6. | 39.0 | 58.50 | 1.976 | 83.00 | 200.00 | 101.50 | 1.73 | 12.407 | 29.02 | 18.182 | 19.80 | 62.22 | 668 |
| Run No. 3 | Pi = 20.6 kW, Pe = 15.48 kW, Pel = 11 kW, Pfr = 5.12 kW, N = 1550 rpm nm = 75.15 %, Pi = 5.02 bar and Pe = 3.77 bar. | | | | | | | | | | | | |
| 1. | 0.0 | 0.00 | 4.521 | 189.39 | 189.39 | 145.99 | 2.23 | 9.194 | 39.16 | 12.235 | 29.42 | 89.49 | 655 |
| 2. | 10.0 | 13.75 | 4.481 | 188.18 | 218.18 | 145.40 | 2.00 | 10.591 | 33.99 | 14.094 | 25.54 | 89.13 | 669 |
| 3. | 21.0 | 27.18 | 4.019 | 168.82 | 231.82 | 137.74 | 1.85 | 11.253 | 31.99 | 14.975 | 24.04 | 84.43 | 682 |
| 4. | 29.0 | 36.81 | 3.556 | 149.36 | 236.36 | 119.89 | 1.62 | 11.474 | 31.38 | 15.269 | 23.58 | 73.49 | 714 |
| 5. | 43.0 | 57.76 | 2.340 | 98.27 | 227.27 | 88.49 | 1.32 | 11.033 | 32.63 | 14.682 | 24.52 | 54.24 | 770 |
| 6. | 48.0 | 62.03 | 1.658 | 69.64 | 232.15 | 70.89 | 1.18 | 11.269 | 31.95 | 14.947 | 24.01 | 43.46 | 829 |

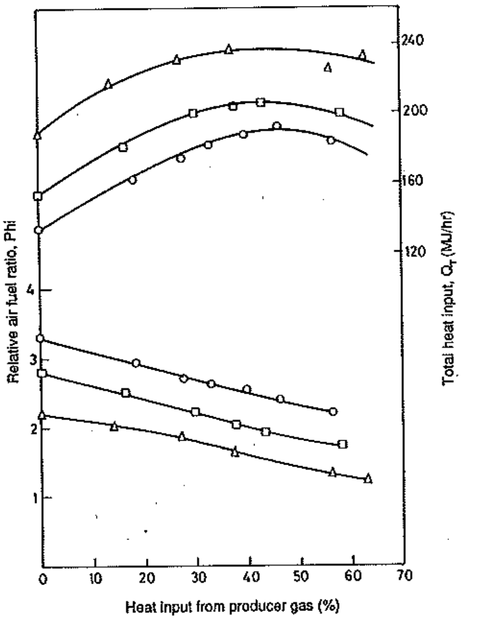


Fig. 4. Effect of producer gas addition on total heat input and relative air fuel ratio.

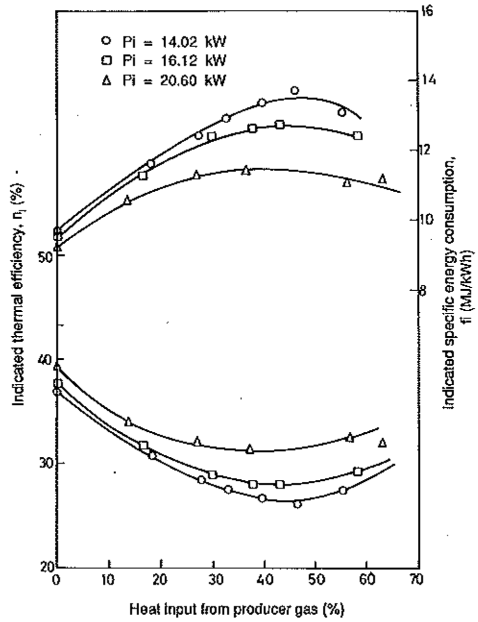


Fig. 5. Effect of producer gas addition on indicated specific energy consumption and indicated thermal efficiency.

compared to the conventional hydrocarbon mixtures. The presence of a higher percentage of carbon monoxide and nitrogen in the producer gas reduces the reaction velocity. Because of this, the heat delivery rate is also delayed in the expansion stroke. "ni" also goes down with this delayed heat delivery rate. At higher flow rates of producer gas, both the decreasing trend in "ni" and the increasing trend in "fi" are reversed.

## Heat Release Characteristics using Producer Gas

The heat release course is a measure of energy conversion efficiency, which has a direct influence on the performance of the engine. The results that were obtained from the heat release evaluation program, for a single test point (indicated power output = 20.6 kW and gas flow rate = 43 $m^3$/hr) are presented in Figs. 6, 7 and 8. Figure 6 shows the variations of the temperature of burnt (Tb) and unburnt (Tu) portions of diesel plus producer gas mixtures and the average temperature of the total fuel mixture (Tm) with respect to the crank angle position of the piston. These temperature variations are shown in the useful combustion range. The cumulative useful heat released by the fuel as a fraction of the average useful heat released by the fuel per cycle (X, termed as burnt charge fraction) and the cumulative useful heat released by the fuel as a fraction of the average heat input in the form of fuel per cycle (Z, termed as heat release factor) are calculated.

The variations of "X" and "Z" over crank angle position, both starting at the beginning of combustion and ending at the completion of useful combustion, are shown in Fig. 7. The starting position of the heat release can be easily identified from this. The change in pressure inside the

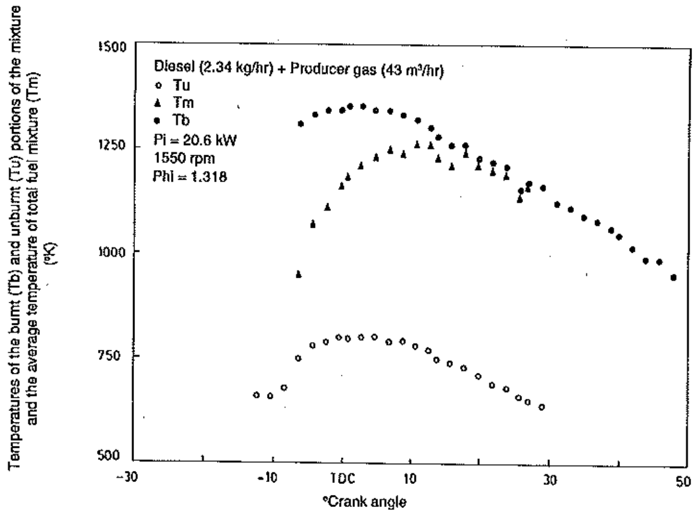


Fig. 6. Variation of the temperatures of burnt (Tb) and unburnt (Tu) portions of the fuel mixture and the average temperature of the total fuel mixture (Tm) with respect to the crank angle position.

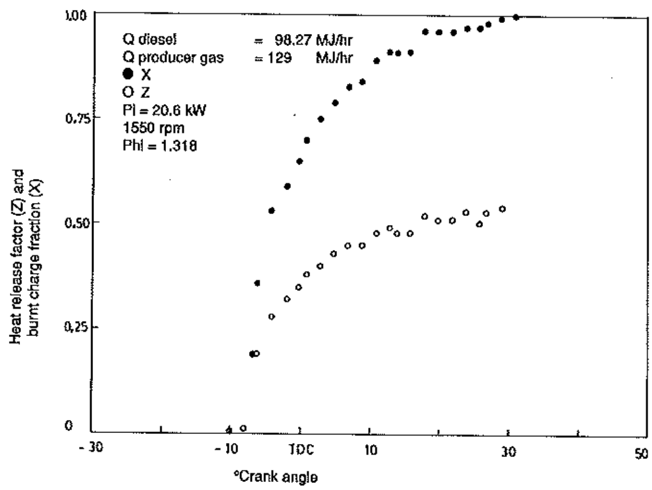


Fig. 7. Variation of the heat release factor (Z) and the burnt charge fraction (X) with respect to crank angle position.

combustion chamber and the quantity of heat released over a unit change in crank angle are presented in Fig. 8. This gives the positions of the piston, at which the maximum values of pressure change and heat release occur. The quantity of heat released at the tail end of the exhaust stroke does not contribute anything to improve the engine's efficiency. Similar analyses were carried out at each test point and the results are presented in the following sections.

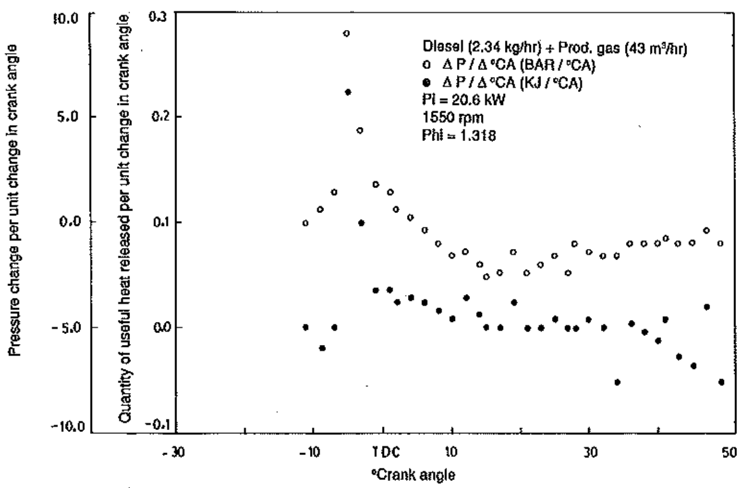


Fig. 8. Change in pressure inside the combustion chamber and heat release over unit change in crank angle position.

When the engine's operating parameters like ignition timing, load, etc., are within comparable limits, the heat release course becomes the property of the fuel and the heat release courses of different fuels could be compared to analyse the suitability of the fuel to the engine.

Due to the heterogeneous nature of the fuel mixture in a diesel engine, the flame front concept of a gasoline engine is not suitable for a diesel engine. So an average temperature of the fuel mixture, which includes both the burnt and the unburnt portions of the fuel, is obtained from the heat release program and presented in Fig. 9. Figure 9 shows the variation of "Tm" with

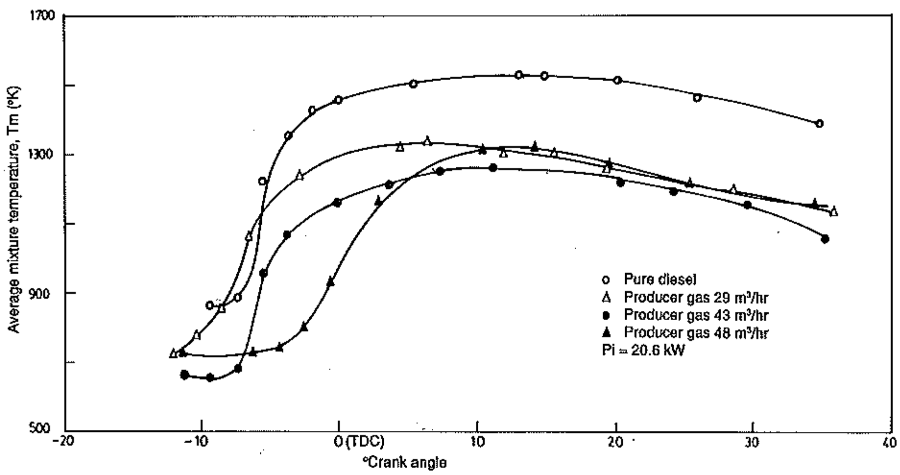


Fig. 9. Variation of the average temperatures of fuel mixture with respect to crank angle position.

respect to the change in crank angle position for pure diesel operation and for three different test points on dual fuel mode, at a constant indicated power output of 20.6 kW. On dual fuel mode of operation the maximum values of "Tm" are much lower than the maximum value of "Tm" on pure diesel operation. This is due to the lower heating value of the producer gas. At a lower gas flow rate (20 $m^3$/hr) the "Tm" value starts rising earlier when compared with pure diesel operation. Whereas, for higher producer gas flow rates (43 $m^3$/hr and 48 $m^3$/hr) the temperature rise is delayed. This can be explained in the following manner. When the producer gas is used with the diesel oil on the dual fuel mode of operation there are two opposite effects on the ignition delay period. Due to throttling, the induction pressure in the combustion chamber goes down. The producer gas which is entering the engine after the heat exchanger is at a slightly higher temperature than the ambient temperature. Both the lower pressure and the higher temperature evaporate the diesel oil at a faster rate, which results in a decreased ignition delay period. So, at the lower flow rate of producer gas, the temperature rise is earlier. As explained before, at higher gas flow rates, the higher presence of carbon monoxide and inert nitrogen in the producer gas increases the ignition delay period. So the temperature rise is also delayed.

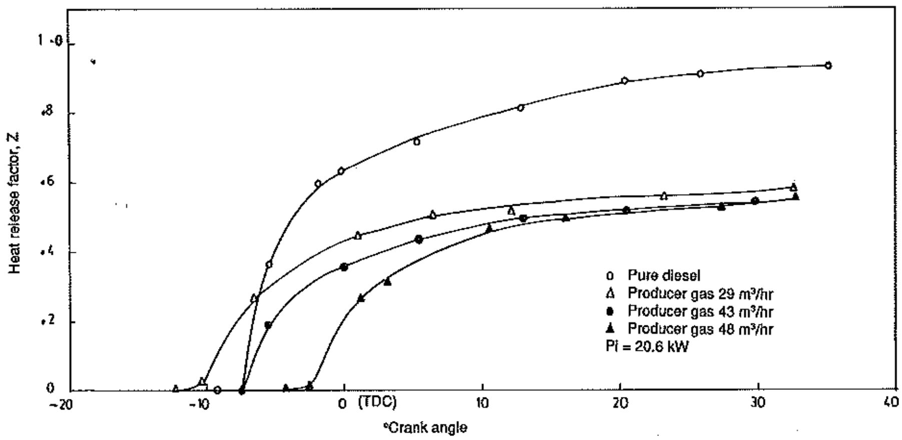


Fig. 10. Variation of the heat release factor (Z) with respect to crank angle position.

The fraction of the total fuel mixture that is used effectively in the combustion process is given by the heat release factor "Z". Figure 10 gives the variation of "Z" with respect to the crank angle position of the piston. The maximum value of Z (Zmax) is much higher for the pure diesel mode than for the dual fuel mode of operation. All these tests were carried out at constant engine operating conditions for better comparison of fuel performance. To get optimum performance, at higher flow rates of producer gas, the pilot diesel injection timing should have been advanced to compensate for the increase in the ignition delay period. Because of the unchanged injection timing, combustion of carbon monoxide takes place at the later part of the expansion stroke. This was clearly indicated by the higher exhaust gas temperatures on dual fuel mode of operation (Table 1). Here the chemical energy of the fuel is lost as sensible heat in the exhaust gas. So the net value of heat input increases to maintain the load of the engine and the maximum value of Z comes down.

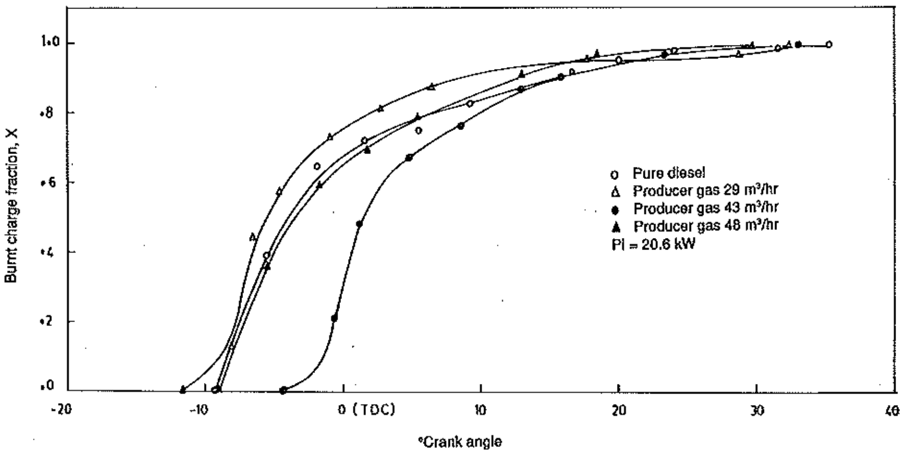


Fig. 11. Variation of the burnt charge fraction (X) with respect to crank angle position.

The fraction of the useful heat released with respect to the crank angle position is given by burnt charge fraction, "X". The variation of "X" with respect to the change in crank angle position of the piston is shown in Fig. 11. At higher gas flow rates, the heat release is slower because the constituents of the carbon monoxide have a lower flame propagation velocity. This results in the reduced combustion period and the increased ignition delay period. At lower gas flow rate, the heat release is earlier because of the earlier evaporation of diesel oil. Figure 12 summarises the important results of Figs. 9, 10 and 11.

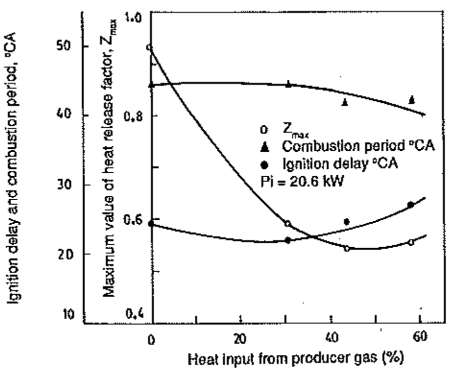


Fig. 12. Effect of producer gas addition on the maximum value of heat release factor ($Z_{max}$), combustion period and ignition delay period.

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## NOMENCLATURE

$A_{min}$ = Stoichiometric air requirement of diesel fuel
= 14.5 kg of air/kg of diesel (Stahl, 1986)

$A_{ming}$ = Stoichiometric air requirement of producer gas (std. $m^3$ of air/std. $m^3$ of producer gas)

CA = Crank angle in degrees

Da = Density of air at std. conditions ($kg/m^3$)
fe = Brake specific fuel consumption (MJ/kW-hr)
fi = Indicated specific fuel consumption (MJ/kW-hr)
F = Mass flow rate of diesel (kg/hr)
Ma = Mass flow rate of air (kg/hr)
ne = Brake thermal efficiency (%)
ni = Indicated thermal efficiency (%)
nv = Volumetric efficiency (%)
Pfr = Friction power of the engine (kW)
Pe = Brake power output of the engine (kW)
Pel = Electrical power output of the engine (kW)
Pi = Indicated power output of the engine (kW)
= Pe + Pfr
Phi = Relative air fuel ratio
= $Ma/[(F)(A_{min}) + (Vgas)(A_{ming})(Da)]$
$Q_{diesel}$ = Heat added by diesel (MJ/hr)
$Q_T$ = Total heat added by diesel plus producer gas (MJ/hr)
Tb = Temperature of the burnt portion of the fuel mixture (°K)
Te = Exhaust gas temperature of the engine (°K)
Tm = Average temperature of the fuel mixture (°K)
Tu = Temperature of the unburnt portion of the fuel mixture (°K)
Vgas = Volume flow rate of producer gas (std. $m^3$/hr)

$$X = \frac{\text{(Cumulative useful heat released by the fuel)}}{\text{(Average useful heat released by the fuel per cycle)}}$$

= Burnt charge fraction (dimensionless number)

$$Z = \frac{\text{(Cumulative useful heat released by the fuel)}}{\text{(Average heat input in the form of fuel per cycle)}}$$

= Heat release factor (dimensionless number)