INVESTIGATION OF DUAL–FUEL DIESEL ENGINE WITH PARTICULAR REFERENCE TO ENGINE CYCLE MODEL.

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Abstract In order to use gaseous fuels in Diesel Engines, Dual-Fuel Diesel Engine (D. F. D. E) the pilot injection approach is chosen. To predict its performance, an engine cycle model, based on limited-pressure Diesel cycle, is constructed. The model predicts D. F. D. E performance with LPG, and CNG gases. Comparing with pure Diesel engine, by increasing gas proportion in dual-fuel, indicated power and hence indicated mean effective pressure are increased. Effect on thermal efficiency is not so appreciable, but indicated specific fuel consumption is quite considerable, and hence fuel economy of D. F. D. E is superior.

چکیده : برای استفاده از سوختهای گازی در موتورهای دیزلی ، روش "موتور دیزل با سوخت دوگانه" (D. F. D. E.) با پاشش آتش‌زا ، انتخاب گردید . جهت پیش بینی عملکرد آن ، یک "مدل سیکل موتور" ساخته شد که بر پایه سیکل دیزل فشار محدود قرار گرفته است . این مدل عملکرد ایده‌آل موتورهای دیزل با سوخت دوگانه را با گازهای LPG و CNG پیش‌بینی مینماید . در مقایسه با موتورهای دیزلی خالص ، با افزایش سهم گاز در سوخت دوگانه ، قدرت اندیکه و در نتیجه فشار موثر متوسط اندیکه ازدیاد میابند ، تاثیر آن در بازده حرارتی چندان چشم‌گیر نبوده بلکه در مصرف سوخت ویژه اندیکه قابل ملاحظه است . بدین ترتیب موتورهای D. F. D. E. نسبت به موتورهای دیزلی خالص، دارای اقتصاد سوخت برتری هستند .

INTRODUCTION

The prospect of future energy shortages has drawn researchers' attention to alternative fuels and new energy sources. One of these alternative fuels is gaseous fuel. The proved gas reserves offer a large energy supply for the foreseeable future. In the I-C Engines different types of gaseous fuels have been used successfully, and large bore Spark-Ignition (S-I) engines has been used for many years. However the engines which are the subject of this paper are Compression-Ignition (C-I), Diesel-gas engines, which do not easily lend themselves for gas operation. Also, compared with conventional liquid fuels, the gaseous fuels, namely natural gases (NG) and liquid petroleum gases (LPG), from a combustion and emission point of view, have better quality and can be supplied very cheaply.

Conventional methods to use gaseous fuels in C-I Engines.

In order to use gaseous fuels in C-I engines, usually two main methods are used, [1] namely:

1. Converting C-I engine to S-I gas engine (full gas operation)

In this method the following alterations must be carried out in the C-I engine design:

a) To prevent auto-ignition of gas and air mixture, the compression ratio of the C-I engine must be reduced to lower compression ratios in S-I engine range.

b) Since the octane number of gaseous fuels is higher than ordinary petrol (i.e-Octane number of LPG is greater than 100), a high energy spark system will be required.

c) High precision fuel injection system must be removed from C-I engine.

To conduct the above alterations in C-I engine, considerable changes in the engine detail design and its ignition system must be done, and at last we will have a S-I gas engine with its inherent low thermal efficiency Otto cycle.

2. Dual-Fuel Diesel Engine with Pilot Injection. (Dieselgas operation).

To burn gas in C-I engines, Dual-Fuel Diesel Engine (D. F. D. E) with pilot injection is the suitable concept. D. F. D. Engines are those engines which burn both gaseous and Diesel fuels at the same time, or they can be worked as straight C-I engine if only Diesel fuel is available. In these engines during the suction stroke, a lean mixture of the gas and air is drawn into the cylinder, and near the end of the compression stroke, a small quantity of liquid Diesel oil, as a pilot fuel, injected into the hot gas-air mixture and ignited it almost simultaneously, probably with propagation of local flame fronts. Although the amount of pilot fuel used is only 5-10% of the full load quantity of Diesel fuel, energy released from it, is considerally greater than conventional spark system and consequently ignition of the lean mixture of gas-air is more reliably obtained. Therefore in this method, C-I engine design will not be altered and as a result engine working cycle will be remained as highly efficient as dual Diesel cycle. As already mentioned in this paper D. F. D. E. (Diesel-gas) concept will be discussed.

D. F. D. E Advantages

Compared with conventional C-I engines, D. F. D. E. offers the following advantages:

a) With gaseous fuels, smoke-limited maximum power output of the engine can be extented [2].

b) With addition of gaseous fuels, the amount of smoke and oxides of nitrogen in combustion products will be reduced [1, 2].

c) By using gaseous fuels, peak cylinder pressure and corresponding rate of pressure rise will be lower than the pure Diesel operation, hence the engine runs smoother and "Diesel Knock" problem may be eliminated. [2]

d) Since gaseous fuels are clean burning fuels, cylinder deposit will be lower than pure Diesel engine, and hence engine service intervals will be extended.

Literature Review

The literature shows that, previous works in D. F. D. E are limited to some experimental investigations (test bed type work), in order to compare their performance with corresponding pure Diesel operation [2-7]. However it seems that on the theoretical side, the least work has been done on the development of thermodynamic models to predict D.F.D.E. performance. This will considerably save test bench time. In this regard a number of engine cycle models are developed at TNO, Holland [8,9] for their own engine development and optimization purposes. Here we started to develop our own engine cycle models. Initially ideal air-fuel cycle for D.F.D.E. has been constructed and later on by introducing suitable "heat release pattern" and "heat transfer correlation", the real cycle of these engines will be constructed.

Proposed cycle for D. F. D. E

Here we are not considering in detail the type of combustion occuring in D. F. D. E. but it has been shown that [3] two types of combustion occur simultaneously in these engines, one the compression-ignition type and the other similar to that of the spark-ignition engines. Therefore it seems that D. F. D. E cycle may lie between (C-I) and (S-I) engines cycles. But because of higher compression ratio of the D. F. D. E., constant-volume cycle will overestimate the thermal efficiency. For this reason limited-pressure fuel-air cycle [10] is chosen and for D. F. D. E. operation is modified. Figure 1 shows (P-V) diagram of the modified fuel-air cycle for D. F. D. E. in question.

Along with cycle calculations, necessary stoichiometric and thermochemical calculations for working fluid before and atter combustion are also carried out.

Assumptions made for cycle construction

To construct ideal fuel-air cycle for D. F. D. E., the following assumptions are considered:

a) All processes are adiabatic, that is, no heat transfered through the cylinder walls.

b) Adiabatic combustion processes are instantaneous, and with excess air in fuel-air mixture, hence dissociation in combustion products is ignored.

c) Calculation has been conducted with variable specific heat in terms of charge temperature and composition.

d) Gaseous fuel introduced to dry air during

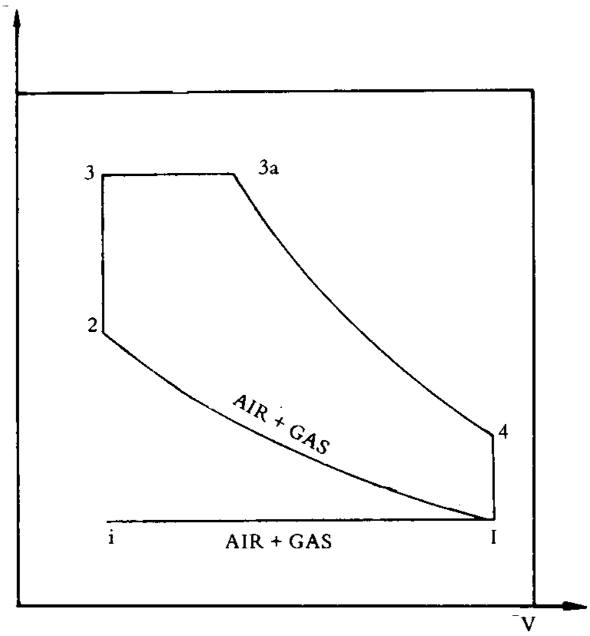

Figure 1. (P-V) Diagram of ideal FUEL-AIR cycle for D.F.D.ENGINES.

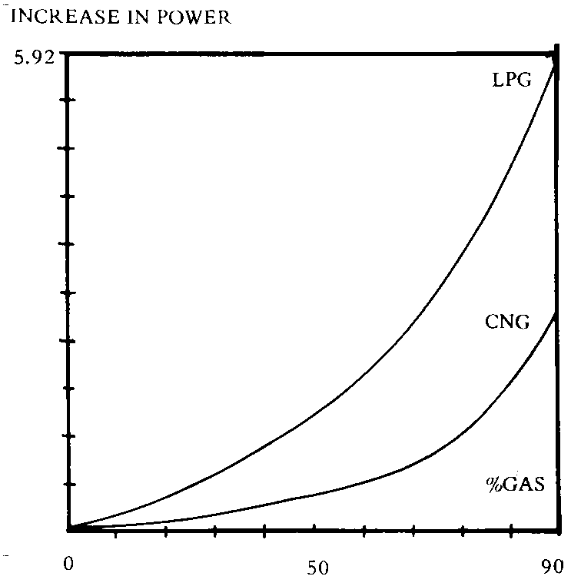

Figure 2. Curves of percentages of increase in power Vs different proportion of Gas.

suction stroke, compressed up to point (2), and at this point pilot fuel injection starts.

Cycle Calculations

For computation purposes, the whole cycle has been divided into small volume steps (ΔV) and the first law of thermodynamics has been written in its general form for each step. With known initial condition at the start of the step, condition of the step has been found by solving the energy equation using Newton-Raphson Iteration method. Computation algorithm is given in Appendix (1). The computer programme has been written in Fortran IV, and executed for I. D. E. M-OM-330 Diesel Engine with different percents of gases replaced with Diesel oil. Engine Specifications are given in Appendix (2).

Discussion of Results

Figures (2-6) shows curves of performance of D. F. D. E. with different percents of LPG and CNG (zero to 90%). These curves have been obtained with the following constant conditions:

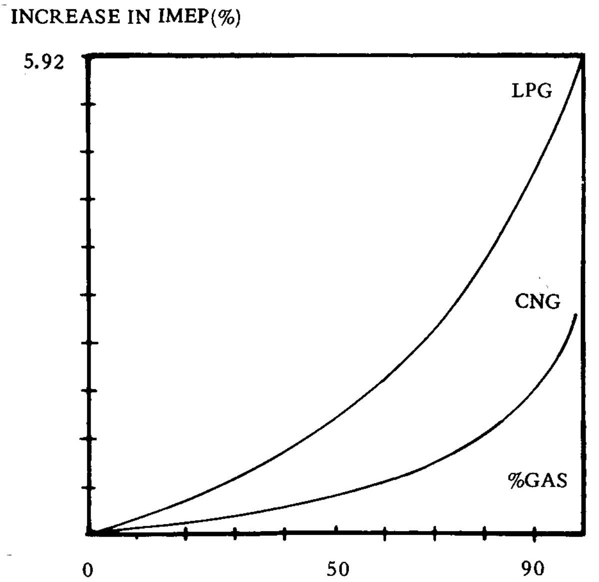

Figure 3. Curves of percentages of increase in IMEP vs different proportions of gas.

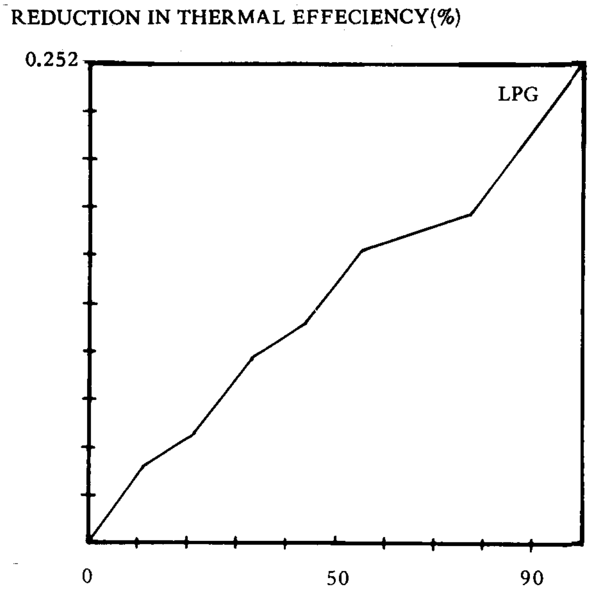

Figure 4. Curve of percentage of reduction in thermal effeciency Vs different proportions of LPG.

– Ratio of maximum pressure to pressure at point (1)	70
– Relative fuel air ratio	0.77
– Engine speed	2000 rpm
– Trapped condition	101.325 Bar, 288°K
– Diesel fuel	$C_{16}H_{30}$
– LPG	30% C_3H_8, 70% C_4H_{10}
– CNG	Typical Natural Gas

Figure 2 shows curves of percentage of increase in indicated power (Pi) versus different proportion of gases. These curves indicate that, with increasing percentages of gaseous fuels (LPG or CNG), engine power compared with pure Diesel engine is increased. This is due to increase in net work done at whole cycle, which is caused from the larger area of the corresponding (P-V) diagram of D. F. D. Engines.

Figure 3 shows curves of percentage of increase in indicated mean effective pressure (Imep) versus different proportions of LPG and CNG. With constant piston displacement volume, Imep will be increased with increasing net total work done in the whole cycle.

Figure 4 indicates percentage of drop in indicated thermal efficiency (η_{th}) versus different proportions of LPG. As indicated, the rate of decrease of the η_{th} is nearly constant, and comparing with pure Diesel engine, the variation is very small. The reason is clear from thermal efficiency relationship. Since dual-fuel flow rate $\dot{m}_f$ is fixed, both P_i and lower calorific value Q_c are increased. But the rate of increase of Q_c is larger than P_1, hence η_{th} will be decreased with constant slope. However, comparing with pure Diesel engine, its variation is not appreciable.

Figure 5 shows percentage of increase in indicated thermal efficiency versus different percents of the CNG. The reason for increasing η_{th} is due to higher rate of increase of P_i relative to Q_c (since Q_c for CNG is less than LPG), therefore η_{th} increases with gas proportions. However, here again, maximum increase for 90% gas is less than 3%.

Figure 6 shows percentage of decrease in indicated specific fuel consumption (Isfc) of the D. F. D. Engines for different proportions of LPG and CNG. This figure indicates that, Isfc decreases very rapidly, when gas

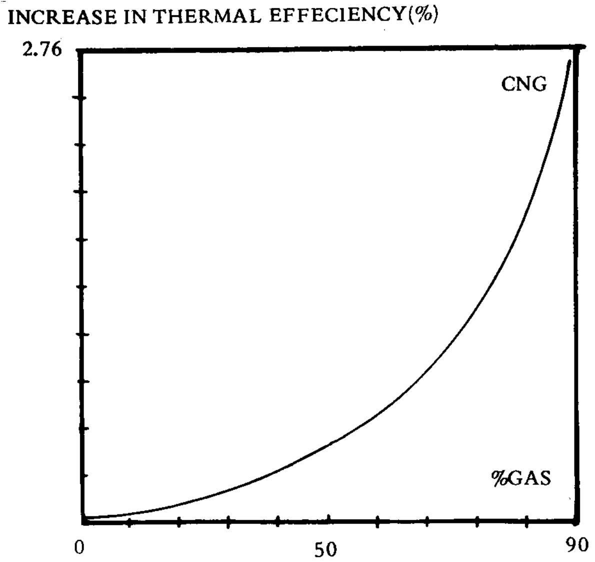

Figure 5. Curve of percentage of increase in thermal effeciency us different proportion of CNG.

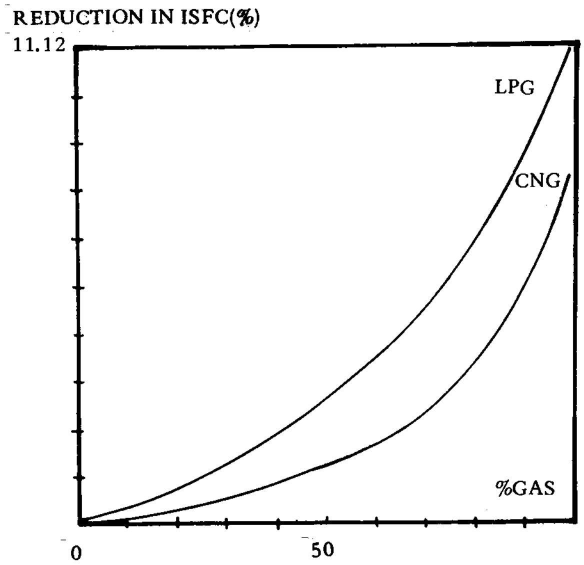

Figure 6. Curves of percentages of reduction in ISFC us different proportions of Gases.

proportions are increased. This is due to increase of P_i. Therefore Isfc of the D. F. D.E. will be always less than pure Diesel engine.

CONCLUSIONS

Examination of the results above shows that the present thermodynamic model leads to the following important conclusions:

a) With increasing gas proportion in D. F. D. E. both indicated power and indicated mean effective pressure are increased. This effect is considerable in the case of LPG, because of its higher calorific value.

b) Comparing with pure Diesel engine, indicated thermal efficiency of D. F. D. E. with LPG is not so affected, but with CNG, its increase is relatively small.

c) Indicated specific fuel consumption of D. F. D. E. is always less than pure Diesel engine. With very high Diesel fuel substitution rates, its rate of decrease is quite considerable.

d) Existing thermodynamic model can predict performance of any types of D. F. D. Engines with LPG and CNG and with other types of gaseous fuels as well. The model is available as a tool for D. F. D. E. conversion and development work. This will reduce test bench time and costs.

APPENDIX (1)

1) As imput data, specifications of the engine, gaseous fuel, and trapped conditions at point (1) are given to the model. Specification of point (1) is obtained from necessary stoichiometric calculations.

2) Isentropic compression stroke has been divided in to small volume steps ($V_2=V_1 - \Delta V$) and for each step, first law of thermodynamics written (dual-fuel mass burnt is zero), and with known initial condition at the start of the step, end condition of step has been found by solving the evergy equation. This condition will be initial condition to the next volume step. With repeating this type of calculations for other steps, and with known compression process path, the characteristics of the point (2) will be obtained.

3) With known limited-pressure P_3, and condition at point (2), fraction of dual-fuel burnt during adiabatic combustion process at constant-volume (2-3) has been obtained, and temperature at point (3) will be known.
4) For remaining of dual-fuel, adiabatic constant-pressure (3-3a) combustion, has been considered, and cycle maximum temperature (T_{3a}) calculated. At point (3a), all of the dual-fuel has been burnt.
5) Similar to the compression stroke, isentropic expansion stroke of the combustion products from point (3a), has been divided into small volume steps ($V_2=V_1+\Delta V$), and energy equation has been solved with dual-fuel mass burnt equal to zero, until condition at point (4) is obtained.
6) In each volume step, the work done has been calculated (dW) and integrated for whole cycle events to get total net work.

$$W_{net} = \Sigma dW$$

7) Then performance of the D. F. D. E. calculated from following relations:

– Indicated power

$$P_i = W_{net} \; N/2 \cdot n$$

where N is engine speed and n is engine cylinder number

– Indicated mean effective pressure

$$Imep = \frac{W_{net}}{V_1 - V_2}$$

– Indicated thermal efficiency

$$\eta_{thi} = \frac{P_i}{\text{[illegible]}}$$

where $\dot{m}_f = m_l + m_g$, subscripts l, g refer to liquid and gas respectively.

– Indicated specific fuel consumption

$$Isfc = \frac{1}{\eta_{thi} \cdot Q_c} = \frac{\dot{m}_f}{P_i}$$

APPENDIX (2)

Engine Specifications

Engine type	4-stroke, Direct injection
cylinder Number	6
Bore	115 mm
Stroke	140 mm
Displacement Volume	8.72 lit
Compression ratio	16.8
Power at 2300 ppm (Continuous)	107 KW

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