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Mathematical Modeling and Simulation of CNG-Diesel Dual Fuel Engine Cycle Processes

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Abstract: This study presents a detailed mathematical modeling and simulation study of CNG-diesel dual fuel compression ignition engine cycle processes using EES (Engineering Equation Solver) software. By means of a step-by-step simulation approach, the values of pressure, volume and temperature at all the state points in the ideal, air-diesel and CNG-diesel dual fuel cycles are computed. Various performance parameters such as thermal efficiency, power output, mean effective pressure and network output are evaluated for chemically correct ($Y = Y_{CC}$) air fuel mixtures assuming adiabatic combustion and full load operation for diesel and CNG-diesel fuel modes. CNG available in National Capital Territory of Delhi has chemical composition with methane content of 84.5% on Mole basis. In this study, the impact of other two main constituents of CNG, i.e., Ethane (C_2H_6) and Propane (C_3H_8) on the CNG-diesel dual fuel engine parameters is also analyzed. The predicted results of various performance parameters of dual fuel cycle using 15% diesel fuel and 85% CNG fuel with 100% methane (CH_4) in it and with 84.5% methane, 12.5% ethane and 3% propane are computed and presented graphically. Predicted results of cycle simulation indicate that conversion of diesel to CNG-diesel is possible with no change in compression ratio, engine design and engine basic structure.

Key words: Mathematical modeling, dual fuel mode, compressed natural gas, mean effective pressure, adiabatic combustion, India

INTRODUCTION

Future oil supply limit and the high cost of oil refining and liquid fuels production in third world calls for efforts throughout all areas of transportation to use lower cost fuels. Also the changes in the air pollution standards, dictates a shift to move decentralized, diffused and renewable energy sources. The broad spectrum of alternative fuels for vehicular needs is becoming increasingly competitive, particularly with the conventional gasoline and diesel fuelled internal combustion engines.

Compressed natural gas, mainly methane (CH_4) is a pragmatic alternative fuel for several reasons. Firstly, it is cheaper than petrol and diesel fuels and abundantly available. Natural gas reserves are vast and the supply is in a relatively high state of purity. Also methane, the first in the paraffinic series, requires no significant changes in the current design and minimal adaptation to the existing fueling system. However, efforts to design vehicular engines to burn expressly gases fuels are underway and promising (Eghbali, 1984). As compared to the experimental study, computer simulation has become a powerful tool which minimizes time and it is the most economical process. A proposed theory can be analyzed quickly using a computer modeling and simulation and the cost of setting up an experimental apparatus can be postponed until optimization is achieved. In an internal combustion engines, the processes involved are extremely complex. In earlier days, the design of engine relied heavily on previous experience and know-how. As a result in engine design, extensive testing of the prototype is considered necessary. Selecting the best design from such testing is a very difficult task. Because of these difficulties, analysis by means of computer simulation has become quite popular in recent years (Ganesan, 2000). Mathematical modeling is an important tool to simulate all the four strokes (compression, expansion, exhaust and inlet strokes) of direct injection diesel engine. These models can be categorized as thermodynamic (zero dimensional) and fluid dynamic (multidimensional) in nature, depending on whether the equations which give the model its predominant structure are based on energy conservation or on a full analysis of the fluid motion. Thermodynamic models are based on the first law of thermodynamic and are used to analyze the performance

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characteristics of engines. Multi-dimensional models give significant spatial details inside the engine at the cost of large investment in computational time and equipment. Thermodynamics or phenomenological models give reasonable prediction of bulk pressure and temperature with crank angle or in other words with respect to time (Sarkar *et al*., 2008).

The main objective of this research study is to develop computer codes to simulate CI engine and dual fuel cycle processes for the use of conventional and alternate fuels of the form $C_XH_YO_Z$. The model describes the processes occurring within the working fluid of compression ignition and CNG-diesel dual fuel engine and provides an indication of the limits and trends of the performance as function of some system variables. Simulation in this study is done in step by step approach. In the first step, the engine is assumed to work with air as the working medium in an ideal air-standard diesel cycle. This analysis is termed as Ideal Cycle Simulation (ICS). In the second step, ICS is modified by taking diesel fuel into account, incorporating adiabatic flame temperature calculations for heat addition or combustion analysis. This analysis is termed as Fuel-air Cycle Simulation (FCS). Here, the combustion is assumed to take place at constant pressure and under adiabatic conditions. In the next step, FCS is modified to simulate CNG-diesel dual fuel engine cycle processes. This analysis is newly termed as Dual Fuel-air Cycle Simulation (DFCS). So by means of this step-by-step simulation approach, the values of pressure, volume and temperature at all the state points in the ideal, air-diesel and CNG-diesel dual fuel cycles are computed. Various performance parameters such as thermal efficiency, power output, mean effective pressure and network output are evaluated for chemically correct ($Y = Y_{CC}$) air fuel mixture assuming adiabatic combustion and full load operation for diesel and CNG-diesel dual fuel modes. CNG available in National Capital Territory of Delhi has a chemical composition with methane content of 84.50% on mole basis. In this study, the impact of other two main constituents of CNG, i.e., Ethane (C_2H_6) and propane (C_3H_8) on the CNG-diesel dual fuel engine performance parameters is also analyzed. The predicted results for various percentage of methane (CH_4) and percentages of ethane and propane present in CNG supply available in Delhi are computed and presented graphically.

MATERIALS AND METHODS

CNG composition and properties: The exact composition of natural gas depends on whether the gas is sourced from an oil condensate field, i.e., whether it is associated gas or it exists by itself which is referred to as non associated gas. Associated gas may contain significant amounts of heavier hydrocarbons such as ethane, propane and butane together with lighter liquids such as pentane and hexane. In this category, methane percentage can be as low as 50%. Non-associated gas contains a much higher percentage of methane. Table 1 shows the composition of pipeline quality natural gas supplied in India (Maji, 2002).

Table 1: Pipeline quality natural gas in India

	Source 1-3 (Mole %)		
Constituent	Ex HBJ (Delhi)	Ex HBJ (IPCL)	Ex Mumbai
Methane (CH_4)	84.50	88.42	82.55
Ethane (C_2H_6)	07.70	08.79	07.67
Propane (C_3H_8)	02.40	01.59	03.85
I-Butane (I-C_4H_{10})	0.26	0.29	0.64
N-Butane (n-C_4H_{10})	0.32	0.28	0.78
I-Pentane (I-C_5H_{12})	0.18	0.05	0.13
N-Pentane (n-C_5H_{12})	0.19	0.05	0.13
Hexane (C_6H_{14})	0.17	0.04	0.09
Nitrogen (N_2)	0.12	0.20	0.07
Carbon dioxide (CO_2)	4.23	0.27	0.07

Table 2: Properties of CNG and diesel fuels

Properties	CNG	Diesel
State	Gas	Liquid
Boiling point (K at 1 atmos)	147	433-655
Density (kg m^{-3})	128	785-881
Auto-ignition temperature (K)	905	477-533
Flash point (K)	124	325
Research octane number	130	N/A
Flammability limits range	5.0-15	0.7-5
Net energy content (MJ kg^{-1})	49.5	43.9
Combustion energy (KJ m^{-3})	24.6	36
Vaporization energy (MJ m^{-3})	215-276	192

Since, the proportion of methane in CNG is the largest as compared to other gases like propane and ethane, the main characteristics of CNG can be directly related to the characteristics of methane. So, it is very important to know the composition of CNG for this study because different composition has different effects on the combustion process in the diesel engine. Physically, CNG is colorless, tasteless, relatively non-toxic and not a volatile organic compound. CNG is lighter than air with a vapor density of 0.68 relative to air. This is advantageous because if leaking happens, it will not cause explosion but instead it will disperse to the atmosphere. It has high auto-ignition temperature compared to gasoline or diesel which means that it is more difficult to ignite. This property of CNG is very useful for dual fuel engine application in the sense that it will reduce fire hazard and exhibit very high anti-knock ability. Main properties of CNG and diesel are shown in Table 2 (Li, 2004).

Numerical simulation of CI engine with air as the working medium: It is advantageous to devise closed cycles that approximate open diesel cycle for analysis of compression ignition processes. One such approach is Ideal Cycle Simulation (ICS) with air as the working medium. This simulation is based on the following assumptions (Heywood, 1989):

- Working medium is assumed to be an ideal gas
- There are no intake and exhaust processes
- Working fluid throughout the cycle is a fixed mass of air
- Combustion process is replaced by a heat addition process from an external source
- Heat addition takes place at constant pressure
- The cycle is completed by heat rejection to the surrounding at constant volume
- All processes are internally reversible
- No heat transfer to the surrounding
- Friction is neglected
- Working medium has constant specific heats (C_P and C_V)

Based on these assumptions, computer codes are developed in EES software for simulation of ideal diesel cycle and the values of various engine parameters are computed and the predicted results are represented graphically.

Mathematical modelling and simulation of CNG-diesel dual Fuel-air Diesel Cycle (DFCS): Based on the ideal cycle simulation, for diesel-air cycle processes (FCS), computer codes were developed in EES software for $Y = Y_{cc}$ (Chemically correct fuel-air mixture). In FCS simulation, fuel-air mixture is considered as the working medium and the intake process with simplified assumptions is also included. In order to simulate the combustion process, constant pressure adiabatic combustion calculations are incorporated and this simulation is termed as Fuel-air Cycle Simulation (FCS). The main aim of the simulation using adiabatic combustion calculations is to evaluate the power output and thermal efficiency for a given engine speed, ambient air temperature, fuel and fuel-air ratio assuming naturally aspirated conditions. The predicted results of diesel-air cycle (FCS) and Dual Fuel Cycle (DFCS) simulation are presented graphically.

Numerical simulation of CNG-diesel dual fuel cycle processes is achieved by suitably modifying the FCS equations by assuming CNG-air mixture behaves as a perfect gas and it does not undergo any chemical change during the compression process. Dual fuel cycle remain based on the diesel cycle (Semin *et al*., 2008). The processes of diesel cycle, i.e., compression, constant pressure heat addition, expansion, exhaust and intake are modified in order to simulate the dual fuel cycle processes assuming constant pressure adiabatic combustion. Once

Table 3: Details of engine for simulation

Parameters	Values
Stroke length (S)	110.0 (mm)
Cylinder bore (B)	80.0 (mm)
Connecting rod length (L)	230.0 (mm)
Compression ratio (r)	16.5
Displacement volume (V_disp)	552.9 (cc)
Volume at TDC (V_tdc)	35.7 (cc)
Volume at BDC (V_bdc)	588.6 (cc)

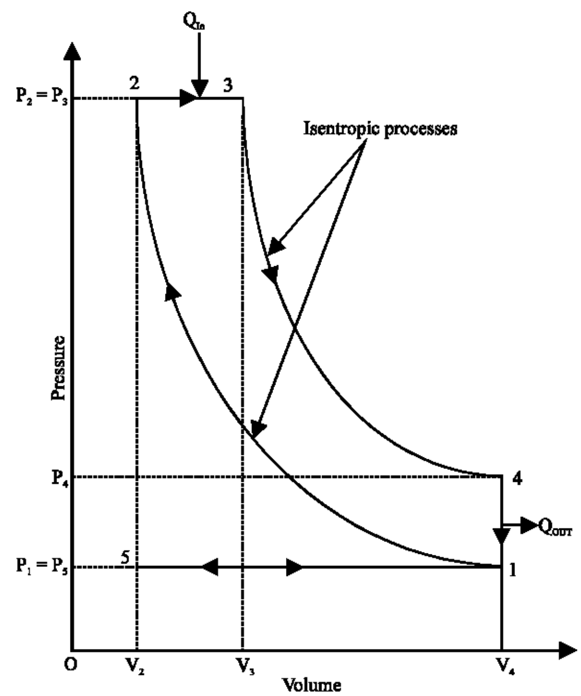

Fig. 1: P-V diagram of CNG-diesel dual fuel cycle for naturally aspirated conditions

again computer codes were developed in EES software for chemically correct air fuel mixture ($Y = Y_{cc}$) for evaluating the performance parameters of the dual fuel cycle engine with 15% diesel fuel and 85% CNG fuel with 100% methane and with 84.5% methane, 12.5% ethane and 3% propane content in it. The predicted results of dual fuel thermodynamic cycle simulation are tabulated and presented graphically for 15% diesel and 85% CNG. The details of the engine under consideration for simulation are shown in Table 3.

For estimation of pressure, volume, temperature at all the state points of CNG-diesel dual fuel cycle and for computing the thermal efficiency, power output, mean effective pressure and network output computer codes are developed in EES software. The initial conditions at state point 1 in the dual fuel cycle are assumed as: P1 = P5 = $P_a = P_{ag}$ = ambient pressure = 101.325 kPa. V1 = V_{bdc}, $T_a = T_m = T_{ag}$ = 298 K. Following equations were formulated for compression, combustion, expansion and exhaust and intake processes in order to compute various CNG-diesel dual fuel engine cycle parameters (Fig. 1).

Isentropic compression process of dual fuel cycle: Initially, let us assume the value of N_{xg} and T_1 (Fig. 1) both depending upon the previous cycle which in turn depends on the cycle before that and so on. To start with let us assume $N_X = 0$, $T_1 = T_a = T_{ag} = 298$ K:

$$N_{ag} = \frac{P_1 \times V_{bdc}}{R_u \times T_{ag}} \tag{1}$$

Where:

N_{ag} = Kilo moles of air and CNG taken in during suction
P_1 = Pressure of air + CNG at state point 1
R_u = Universal gas constant = 8.314 KJ/kg mol. K
T_1 = T_{ag}; absolute temperature at state point 1:

$$Kr_{dual} = \frac{Cpr_{dual}}{Cpr_{dual} - R_u} \tag{2}$$

where, Kr_{dual} is polytropic index of compression for the reactants mixture of dual fuels:

$$Cpr_{dual} = \frac{Cpf_{dual} + Y_{cc} \times CpO_2 + 3.76 Y_{cc} \times CpN_2}{N_{mO}} \tag{3}$$

where, Cpr_{dual} stands for constant pressure heat capacity of reactant mixture of air and CNG:

$$N_{mo} = 1 + 4.76 \times Y_{cc} \tag{4}$$

where, N_{mo} represents kilomoles of air + diesel + CNG vapours in a mixture containing 1 kmol of fuel:

$$T_2 = T_1 \times r^{(Kr_{dual} - 1)} \tag{5}$$

where, T_2 is the temperature at state point 2 in the CNG-diesel dual fuel cycle, Kr_{dual} is ratio of specific heats of reactant mixture of dual fuels:

$$P_2 = P_1 \times r^{Kr_{dual}} \tag{6}$$

where, P_2 is the pressure at state point 2 in the dual cycle:

$$V_2 = V_{tdc} \tag{7}$$

Adiabatic combustion process: Combustion process is assumed to take place adiabatically and at constant pressure. For no heat transfer, the energy equation can be written as (Ferguson, 1989):

$$Hr(T_2) + Hf(T_f) = Hp(T_3) \tag{8}$$

where, the suffixes r, f and p denote the N_a+N_g+N_X moles of reactants, fuel (Diesel+CNG) and the products of combustion. Solution of Eq. 8 will give the value of T_3. Since, the combustion process is at constant pressure, $P_3 = P_2$. Knowing P_3 and T_3, volume at point 3 is given by Eq. 9:

$$V_3 = N_p \times R_u \times \left(\frac{T_3}{P_3}\right) \tag{9}$$

For a constant pressure adiabatic combustion, the energy released when the products are cooled to initial temperature T_1 of the reactant when a unit quantity of fuel burn at constant pressure, Q_p is given by Eq. 10:

$$Q_p = 1\{N_i \times [h_i(T_3) - h_i(T_2)]\} \tag{10}$$

This equation was solved using Newton-Raphson Iteration Technique by knowing the product gas mole numbers, N_i in Eq. 10:

If Y_{cc} represents moles of O_2 mole^{-1} of ($C_{10}H_{22}$ + CNG) fuel for chemically correct mixture then the reactants fuel air mixture for dual fuel combustion is:

$$\begin{aligned} &N_{diesel} \times C_{10}H_{22} + N_{CNG} \times (N_{methane} \times CH_4 + \\ &N_{ethane} \times C_2H_6 + N_{prop} \times C_3H_8) + \\ &Y_{cc} \times O_2 + 3.76 \times Y_{cc} \times N_2 \end{aligned} \tag{11}$$

where, N_{diesel}, N_{CNG}, $N_{methane}$, N_{ethane} and N_{prop} denote percentage number of moles of diesel, CNG, methane, ethane and propane, respectively:

$$Y = Y_{cc} \times RAF \tag{12}$$

where, Y represents actual number of moles of O_2 mole^{-1} of ($C_{10}H_{22}$+CNG) fuels. RAF stands for relative air-fuel ratio:

$$Y_{cc} = N_{mC} + 0.25 \times N_{mH_2} - 0.5 \times N_{mO_2} \tag{13}$$

$$\begin{aligned} N_{\downarrow mC} = {} & N_{\downarrow diese} \times 10 + N_{\downarrow CNG} \times (N_{\downarrow CH_4} + \\ & N_{\downarrow C_2H_6} \times 2 + N_{\downarrow C_3H_8} \times 3) \end{aligned} \tag{14}$$

$$\begin{aligned} N_{mH_2} = {} & N_{diesel} \times 22 + N_{CNG} \times (N_{CH_4} + \\ & N_{ethane} \times 6 + N_{prop} \times 8) \end{aligned} \tag{15}$$

where, N_{mc}, N_{mh2} and N_{mO2} represent the number of moles of carbon, hydrogen and oxygen atoms, respectively in the fuel:

$$Qp_{dual} = (Hrp_{dual} - N_{co} \times 282800) \times \frac{N_m}{N_{mo}} \tag{16}$$

where, Qp_{dual} represents heat energy released at constant pressure when the products are cooled to initial temperature T_1 = 298 K of the reactants consisting of CNG and diesel when a unit quantity of these fuels burns at constant pressure:

$$Hrp_{dual} = N_{diesel} \times Hrp_{diesel} + N_{CNG} \times Hrp_{CNG} \quad (17)$$

$$Hrp_{CNG} = N_{CNG} \times (0.845 \times Hrp_{CH_4} + 0.125 \times Hrp_{C_2H_6} + 0.03 \times Hrp_{C_3H}) \quad (18)$$

where, Hrp_{dual} and Hrp_{CNG} represent heat of reaction for dual and CNG at constant pressure, respectively and is defined as the heat energy added to bring the products of combustion to the initial temperature T_1 = 298 K when a unit quantity of fuel and chemically correct O_2 burn at constant pressure. The adiabatic flame temperature for constant pressure combustion process is given by Eq. 9:

$$Tnew_{dual} = T_{dual} - \frac{(HpT_{dual} - HpTr_{dual} - Qp_{dual})}{Cp_{dual}} \times \frac{N_p}{N_{po}} \quad (19)$$

$$T_3 = Tnew_{dual} \quad (20)$$

where, T_3 is the absolute temperature at state point 3 and T_{dual} is the assumed value of adiabatic flame temperature of the products of combustion for a dual fuel combustion process and $Tnew_{dual}$ is value of AFT obtained after first iteration. The values of HpT_{dual}, Qp_{dual} and Cp_{Tdual} have been calculated using standard JANAF tables.

Isentropic expansion process: After computing the values of T_3, P_3, V_3, corresponding values at point 4 in CNG-diesel dual fuel cycle can be calculated by Eq. 21-23:

$$T_4 = T_3 \times \left(\frac{V_3}{V_{bdc}}\right)^{(Kp_{dual}-1)} \quad (21)$$

$$P_4 = P_3 \times \left(\frac{V_3}{V_{bdc}}\right)^{Kp_{dual}} \quad (22)$$

$$V_4 = V_{bdc} \quad (23)$$

where, T_4, P_4 and V_4 are the values of temperature, pressure and volume at state point 4 of dual fuel cycle, respectively:

$$Kp_{dual} = \frac{Cpp_{dual}}{Cpp_{dual} - R_u} \quad (24)$$

where, Kp_{dual} is specific heat ratio of the products of combustion of dual fuels:

$$Cpp_{dual} = \frac{Cp(T_{3dual})}{N_{po}} \quad (25)$$

where, Cpp_{dual} represents constant pressure specific capacity of products of dual fuel combustion and CpT_{3dual} is the constant pressure specific heat capacity of products at temperature T_{3dual}.

Exhaust and intake processes: At state point 5, pressure drops to ambient pressure and the temperature T_5 can be calculated using Eq. 26:

$$T_5 = T_4 \times \left(\frac{P_1}{P_4}\right)^{\frac{Kp_{dual}-1}{Kp_{dual}}} \quad (26)$$

$$N_{xg1} = \frac{P_1 \times V_{bdc}}{R_u \times T_5} \quad (27)$$

where, N_{xg1} is the number of moles of exhaust gas fraction left in the combustion chamber after the first cycle. The absolute temperature of air, CNG and residual gas fraction at state point1, i.e., at the commencement of next cycle was calculated using Eq. 28. This temperature of air in the combustion chamber in beginning of next cycle will be little higher than ambient air temperature due to the hot gases left from the previous cycle:

$$T_{ag1} = \frac{r \times T_{ag}}{r - 1 + \frac{T_{ag}}{T_5}} \quad (28)$$

where, r stands for compression ratio, T_{ag} and T_5 represent ambient air-CNG temperature and exhaust gas temperature, respectively. Number of kmoles of air and CNG mixture taken in during intake process of the next cycle is given by Eq. 29:

$$N_{ag1} = \frac{P_1 \times V_{tdc}}{R_u \times T_{ag}} - N_{xg} \quad (29)$$

With new values of T_{ag1} (Eq. 28), N_{ag1} (Eq. 29) and N_{xg1} (Eq. 27), new value of Cpr_{dual} is calculated and the computations proceeded through a second cycle commencing with the compression stroke (Eq. 1). This way the third cycle can follow the second cycle and so on. Table 4 shows the predicted results of computations for 6 consecutive CNG-diesel cycle parameters.

Table 4: Predicted results of CNG-diesel Dual Fuel Cycle Simulation (DFCS) with 100% methane in 85% CNG

Parameters	Cycle 1	2	3	4	5	6
T_1 (K)	298.000	313.200	329.000	345.400	362.400	379.900
T_2 (K)	616.400	639.500	663.100	687.200	711.800	736.600
T_3 (K)	2994.000	2994.000	2294.000	2995.000	2995.000	2995.000
T_4 (K)	2228.000	2208.000	2189.000	2171.000	2153.000	2135.000
T_5 (K)	1506.000	1510.000	1514.000	1518.000	1522.000	1526.000
P_2 (kPa)	3458.000	3414.000	3370.000	3326.000	3284.000	3242.000
P_3 (kPa)	3458.000	3414.000	3370.000	3326.000	3284.000	3242.000
P_4 (kPa)	757.500	714.400	674.300	636.800	601.900	569.400
Rc	4.857	4.682	4.516	4.358	4.208	4.066
Power (kW)	23.790	22.670	21.600	20.570	19.580	18.650
Pmep (bar)	16.980	16.180	15.410	14.680	13.980	13.310
Eta_thermal	52.560	52.650	52.690	52.680	52.630	52.530
W_net	951.600	906.900	864.000	822.800	783.400	745.900

Thermal efficiency, work output, mean effective pressure and power output computations for dual fuel cycle: Network output during one cycle was computed using Eq. 30:

$$Wnet_{dual} = Wexp_{dual} + Wcomb_{dual} - Wcomp_{dual} \quad (30)$$

$$Wexp_{dual} = Up(T_{3dual}) - Up(T_{4dual}) \quad (31)$$

where, $Wnet_{dual}$, $Wexp_{dual}$, $Wcomb_{dual}$ and $Wcomp_{dual}$ are the net, expansion, combustion and compression works, respectively for dual fuel cycle:

$$Up(T_{3dual}) = Sf \times Hp(T_{3dual}) - R_u \times N_p \times T_{3dual} \quad (32)$$

$$Up(T_{4dual}) = Sf \times Hp(T_{4dual}) - R_u \times N_p \times T_{4dual} \quad (33)$$

where, Up(T) is internal energy of products of dual fuel combustion at temperature T_{3dual} and T_{4dual}, respectively, N_p is number of moles of gaseous products and Hp(T) stands for enthalpy of products at temperature T_{3dual} and T_{4dual}:

$$Sf = \frac{N_m}{N_{mo}} + \frac{N_{xg}}{N_{po}} \quad (34)$$

Sf denotes the scale factor which reduces the mole numbers to a proper size to fit the engine, once N_{ag} and N_{xg} are known. N_m and N_{po} represents moles of fuel vapour and air in engine during compression stroke, kilomoles of air-fuel vapour in a mixture containing 1 kmol of diesel and CNG fuels and kilomoles of products formed from the combustion of N_{mo}, respectively:

$$Wcomb_{dual} = N_p \times R_u \times (T_3 - T_2) \quad (35)$$

$$Wcomb_{dual} = (N_{ag} + N_{xg}) \times (Cpr_{dual} - R_u) \times (T_2 - T_1) \quad (36)$$

Thermal efficiency of the dual fuel cycle was computed using Eq. 37, mean effective pressure using Eq. 38 and power output using Eq. 39 given as:

$$\eta_{dual} = \frac{Wnet_{dual} \times N_{mo}}{-Hrp_{dual} \times N_{ag}} \quad (37)$$

$$Pmep_{dual} = \frac{Wnet_{dual} \times 4}{\pi \times B^2 \times S \times 101325} \quad (38)$$

$$Poutput_{dual} = \frac{Wnet_{dual} \times RPM}{60 \times 1000} \quad (39)$$

Using Eq. 19, the adiabatic flame temperature for diesel fuel is 2998 and 2994 K for CNG-diesel dual fuel, i.e., diesel adiabatic temperature is more than CNG adiabatic temperature. Therefore, conversion of an existing compression ignition engine to CNG-diesel dual mode can be carried out without any change in engine design and basic engine structure as there will be no effect of heat socks due to replacement of diesel fuel with CNG.

RESULTS AND DISCUSSION

The predicted results of various engine performance parameters for pure diesel Fuel Cycle Simulation (FCS) are represented graphically and CNG-diesel Dual Fuel Cycle Simulation (DFCS) for chemically correct mixture and full load operation at 1500 RPM taking diesel as $C_{10}H_{22}$ and CNG as a mixture of 84.5% CH_4+12.5% C_2H_6+3% C_3H_8 for first case and 100% CH_4 for second case assuming initial temperature and pressure conditions as 298 K and 101.325 kPa, at compression ratio (r) of 16.5 are shown in Table 4 and 5 and represented graphically.

The variation of thermal efficiency of pure diesel, dual fuel with 15% diesel and 85% CNG for six consecutive thermodynamic cycles is shown in Fig. 2. This is very clear from the Fig. 2 that thermal efficiency of dual fuel cycles is more than pure diesel cycle. Thermal efficiency of dual fuel cycle with CNG having 100% methane content is higher than that of dual fuel cycle using CNG with 84.5% methane, 12.5% ethane and 3% propane because higher methane content adds to higher calorific value of CNG. It is also observed from Fig. 2 that as the cycles proceed successively there are not many variations in thermal efficiency values of different cycles. This indicates that combustion of both diesel and CNG fuels is complete and smooth. Its value for dual fuel cycle with CNG having 100% methane content is 8.9% more in cycle 1, 8.94% in cycle 2, 9.38% more in cycle 3, 9.79% in cycle 4, 10.21% in cycle 5 and 10.63% more in cycle 6 than

Table 5: CNG-diesel dual fuel cycle simulation predicted results with 85% CNG having 84.5% CH_4+12.5% C_2H_6+03% C_3H_8

	Cycle					
Parameters	1	2	3	4	5	6
T_1 (K)	298.000	312.000	326.400	341.200	356.400	372.000
T_2 (K)	515.100	533.500	552.200	571.100	590.400	609.900
T_3 (K)	2994.000	2995.000	2995.000	2995.000	2995.000	2995.000
T_4 (K)	2327.000	2307.000	2289.000	2270.000	2252.000	2235.000
T_5 (K)	1140.000	1146.000	1151.000	1156.000	1161.000	1167.000
P_2 (kPa)	2890.000	2859.000	2828.000	2798.000	2769.000	2741.000
P_3 (kPa)	2890.000	2859.000	2828.000	2798.000	2769.000	2741.000
P_4 (kPa)	791.200	749.400	710.400	674.200	640.300	608.700
Rc	5.813	5.613	5.424	5.244	5.074	4.911
Pmep (bar)	16.230	15.570	14.940	14.340	13.750	13.200
Power (kW)	22.740	21.820	20.940	20.090	19.270	18.490
Eta_ther (%)	50.290	50.520	50.710	51.860	50.970	51.040
Wnet (kJ kg^{-1})	909.600	872.800	837.400	803.400	770.800	739.600

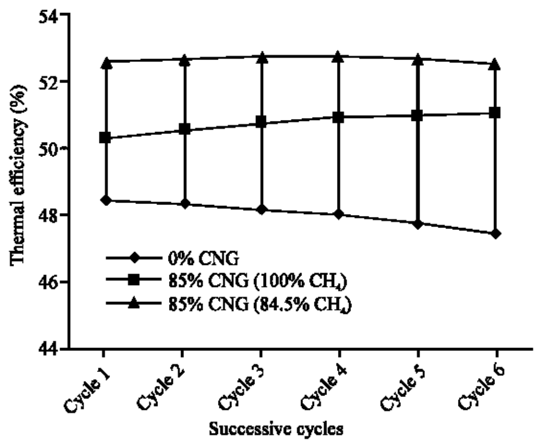

Fig. 2: Variation of thermal efficiency of dual and diesel cycles with successive cycles (Eta_thermal vs. successive cycles (r = 16.5)

that of corresponding cycles of pure diesel. Dual fuel cycle simulation with CNG having ethane and propane in its composition, yields lower thermal efficiency than with 100% methane but more than that of pure diesel cycle. So, the simulation results indicate that methane content in CNG should be maximum and the percentages of other constituents in CNG should be lowest to get higher value of thermal efficiency in dual fuel cycle operation.

Figure 3 shows the variation of mean effective pressure of pure diesel and dual fuel cycle simulation during six successive cycles. Mean effective pressure of pure diesel cycle is more than both dual fuel cycles and as the successive cycles proceed one after the other and its value decreases smoothly as the cycles proceeds.

Figure 4 shows variation of power output of pure diesel, dual fuel with CNG having 100% methane content and dual fuel with 84.5% methane content cycles. Power output follow the same trend as followed by mean effective pressure. Its value for dual fuel cycles is slightly lower than that of pure diesel fuel mode cycles.

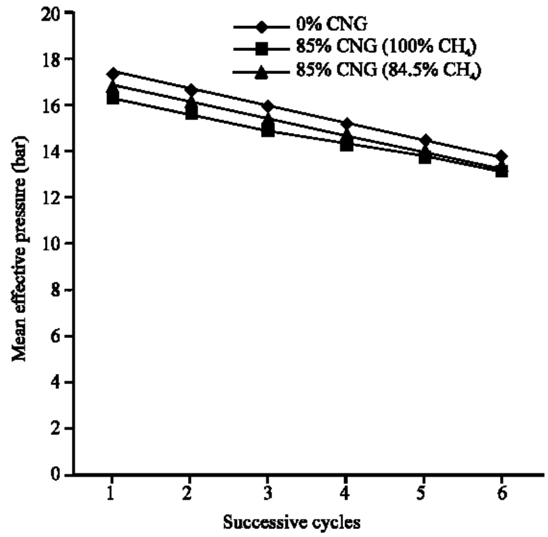

Fig. 3: Variation of mean effective pressure with successive cycles of diesel and duel fuel cycles (P-mep vs. successive cycles (r = 16.5)

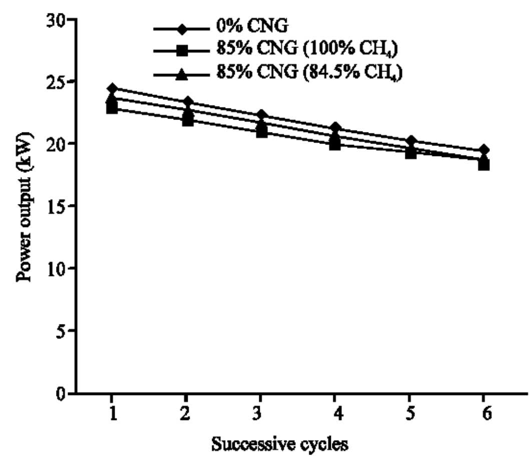

Fig. 4: Variation of power output with successive cycles of diesel and dual fuel cycles (P-output vs. successive cycles (r = 16.5)

Effect of compression ratio on performance parameters of diesel, dual (CNG as 100%CH_4) and dual (CNG as 84.5%CH_4+12.5%C_2H_6+3%C_3H_8) fuel cycles: The effect of varying the compression ratios from 12-22 for pure diesel, dual fuel having CNG 100% methane and CNG having 84.5% methane, 12.5% ethane, 3% propane have been studied for various engine parameters by developing 3 computer programs in EES software and the predicted results are analyzed by representing them graphically. Figure 5-8 show the effect of variation of compression ratio on thermal efficiency, mean effective pressure, power output and net work output during ICS, FCS, dual fuel with CNG having 100% methane and dual fuel with 84.5% methane content cycles simulations. It is very clear from

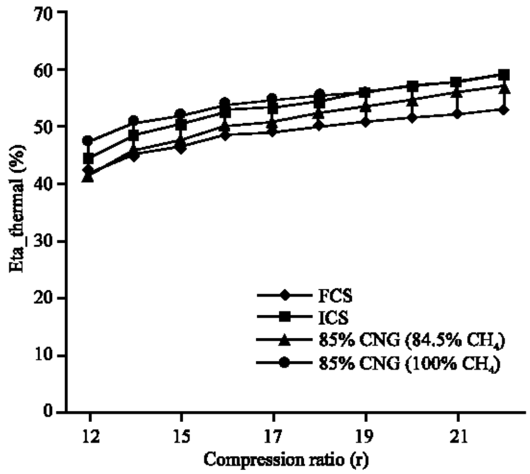

Fig. 5: Effect of compression ratio (r) on the thermal efficiency of CI engine during ICS and FCS and dual fuel cycle modes (Variation of Eta_thermal vs. comp ratio (r))

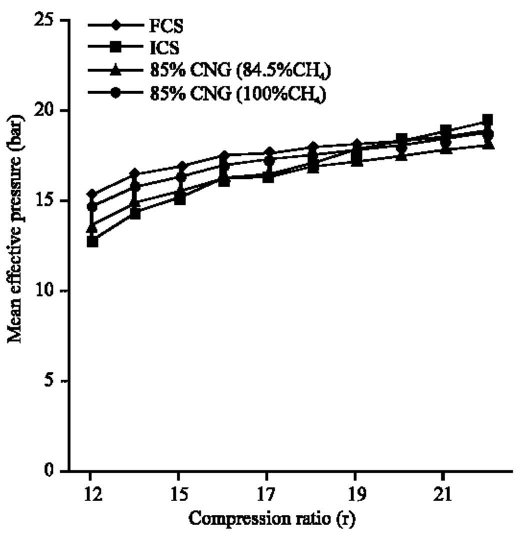

Fig. 6: Variation of mean effective pressure with compression ratio of CI engine during ICS FCS dual fuel cycle modes (r vs. P-mep)

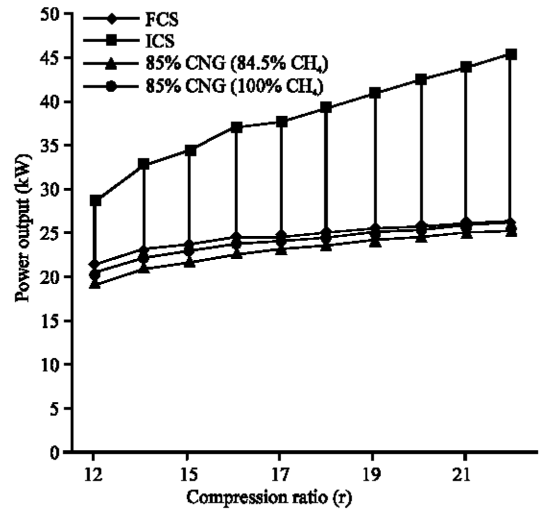

Fig. 7: Variation of power output with compression ratio of CI engine during ICS, FCS and dual fuel cycle modes (r vs. power output)

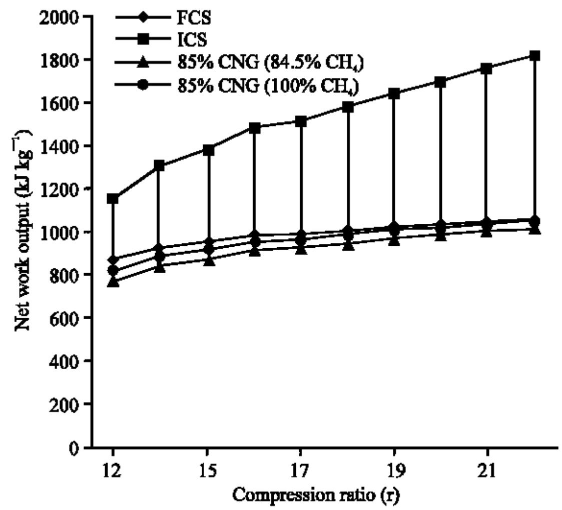

Fig. 8: Variation of network output with compression ratio of CI engine during ICS, FCS and dual fuel cycle modes (r vs network_output

efficiency at all the compression ratios from 12-22. Its value for ideal cycle simulation is highest of these four Fig. 5-8 that with increase in compression ratio from 12-22, all the performance parameters of the four modes considered for cycle simulation as the compression ratio increases, amount of residual exhaust gas in clearance volume decreases, temperature T2 after compression process increases. These two factors combined together produce an increase in thermal efficiency, mean effective pressure, power output and network output of the four cycles. Thermal efficiency of dual fuel cycle simulations of the two modes is more than that of pure diesel mode cycles. For dual fuel cycle with CNG having 100% methane content, thermal efficiency is greater than pure diesel and dual fuel with CNG having 84.5% methane content modes for all the compression ratios. So this simulation study suggest that CNG having higher percentage of methane content should be utilized for dual fuel modes as this type of fuel will result in better performance of CNG-diesel dual fuel engine. Mean effective pressure of dual fuel cycles increase with increase in compression ratio but its value for pure diesel cycle is more than ICS and dual fuel modes up to CR = 20. As the compression ratio is further increased its value for ICS mode becomes more than diesel and dual fuel cycle modes. At this CR = 20 mean effective pressure of ICS, FCS and Dual fuel with CNG having 100% methane content becomes equal and beyond this if CR is further increased, its value for ICS becomes higher than other modes.

Power output and net work output for all the four modes increase with increase in compression ratio. There is a wide gap between their values for ICS mode, diesel mode and both the dual fuel modes under cycle simulation. Their values for ICS mode are much higher than the other modes. In case of diesel and dual fuel modes power output and network output there is no marked difference in their values. For dual fuel cycles for both modes their values is slightly less than that of pure diesel mode.

As can be shown from Fig. 5-8, the lines of thermal efficiency, mean effective pressure, power output and network output for diesel and dual fuel modes almost fall on each other. This means that substitution of diesel fuel with compressed natural gas is possible with no change in compression ratio of the existing diesel engine and in the present study, the ratio of 85% CNG and 15% diesel is suitable and targeted blend ratio for dual fuel cycle engines.

CONCLUSION

The successive cycle simulation predicted results reveal that there are no marked variations in various performance parameters of ICS, FCS and DFCS as the cycles proceed one after the other. This means that conversion of diesel to dual fuel engine is possible without any change in engine design.

Conversion for CNG substitution in diesel engines requires no change in compression ratio of the existing diesel engine. This study also suggest that CNG composition must contain maximum amount of methane content in it as the results with 100% methane content are superior than with 84.5% methane content CNG. In the present study, the ratio of 85% CNG and 15% diesel is optimized as suitable and targeted blend ratio for dual fuel cycle engines.

The adiabatic flame temperature for diesel fuel is computed equal to 2998 and 2994 K for CNG-diesel dual fuel, i.e., diesel adiabatic temperature is more than CNG adiabatic temperature. Therefore, conversion of an existing compression ignition engine to CNG-diesel dual mode can be carried out without any change in engine design and basic engine structure as there will be no effect of heat socks due to replacement of diesel fuel with CNG. From this study, it is concluded that dual fuel cycle simulation is based on diesel fuel cycle which is carried out with chemically correct air-fuel mixture ($Y = Y_{cc}$) and the results indicate that conversion of conventional existing diesel engines to dual fuel engines requires no change in compression ratio. This study is being further extended by the authors to simulation of CNG-diesel dual fuel engine cycle processes for rich and lean mixtures.

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