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Thermodynamic Potential Of GHG (CO_2) Emission Reduction By Using A Combination Of Methane 90% And Diesel 10% As An Alternative Fuel To Diesel In CI Engines

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ABSTRACT

This paper presents the results of the computational research investigations on the possible reduction of GHG emissions by using methane 90% and diesel 10% as an alternative dual fuel in the conventional diesel engine. The results are simulated in the professional thermodynamic internal combustion engines simulation software AVL BOOST. The software involves the use of the conservation laws of mass and energy. Further the software uses the 1-D Navier Stokes equation for the computation of the velocity component of the working medium. The software further has the facility of using alternative thermodynamic models for combustion, engine heat transfer, engine frictional power etc. The equations involved in the thermodynamic flowchart based algorithm are solved by the application of the numerical finite volume method in order to compute the values of various thermodynamic and gas dynamic properties in different engine components. First the modelling was done for a 6 cylinder 4 stroke TCI diesel engine. The modelled engine was run in its conventional diesel mode and results were generated to map the engine for its GHG emissions along with its other performance parameters. Next the modelled engine was modified for dual fuel mode operation and run with methane 90% and diesel 10% as its dual fuel. The software gave satisfactory results in both the cases. The results show that the minimum GHG emissions (CO_2) reduction with methane 90% and diesel 10% dual fuel mode was 26.86% when compared with the neat diesel mode of the engine. The power and torque developed by the engine were reduced in the methane 90% and diesel 10% dual fuel mode operation. The fuel consumption per unit energy output of the engine in its methane 90% and diesel 10% dual fuel mode operation was drastically reduced when compared to the fuel consumption with neat diesel.

Keywords: *GHG, CO_2, Global Warming, Performance, Compression Ignition, Engine, Diesel, Methane.*

INTRODUCTION

The increasing number of engines used in automotive applications has given rise to the pollutants like CO, HC, NOx and Soot emissions. In addition to this the CO_2 or GHG emission has also been increasing simultaneously. There is a practical evidence of global warming caused by the GHG or CO_2 produced from internal combustion engines in addition from other sources. This has generated the need of developing complete engine technologies suitable for the reduction of the GHG being produced from such prime movers. One of the methods for reducing the GHG or CO_2 emissions from the conventional diesel engines is the use of methane or CNG and conventional diesel in the dual fuel mode. The possible highest percentage substitution of the main fuel methane or CNG being ignited by a small fraction of diesel supplied in the form of a pilot injection for initiation of combustion in the same high compression ratio based diesel engine will help in reducing the GHG as is evident from the following chemical equations.

Representing the commercial diesel by cetane and the commercial CNG by methane we have the following three chemical equations for their combustion stoichiometry in the neat and the dual fuel modes.

$C_{16}H_{34} + 24.5(O_2 + 3.773N_2) = 16CO_2 + 17H_2O + 92.4385N_2$

$CH_4 + 2(O_2 + 3.773N_2) = CO_2 + 2H_2O + 7.546N_2$

$0.9\ CH_4 + 0.1\ C_8H_{18} + 4.35(O_2 + 3.773N_2) = 2.5CO_2 + 3.5H_2O + 77.157N_2$

It is seen from these stoichiometric combustion related equations that one molecule of cetane produces 16 molecules of GHG or CO_2 as compared to 2.5 molecules of the GHG or CO_2 produced in the dual fuel mode of methane 90% and diesel 10%.

In Complete combustion, the hydrocarbons in the fuel's chemical bonds are transformed to carbon dioxide (CO_2). Its proportion is also dependent on the operating point. Here again, the proportion depends on the engine operating conditions. The amount of converted carbon dioxide in the exhaust gas is directly proportional to fuel consumption. Thus the only way to reduce carbon-dioxide emissions when using standard fuels is to reduce the fuel consumption. Carbon dioxide is a natural component of atmospheric air, and the CO_2 contained in automotive exhaust gases is not classified as a pollutant. However, it is one of the substances responsible for the greenhouse effect and the global climate change that this causes. In the period since 1920, atmospheric CO_2 has risen continuously, from roughly 300 ppm to approx. 450 ppm in the year 2001. This renders the effort to reduce carbon-dioxide emissions and fuel consumption more important than ever.[1]

F. Mundroff et. al., conducted research investigations on direct injection type of internal combustion engines for both gasoline and diesel fuels. It was concluded that direct injection technology reduces the

CO_2 emissions by reducing the fuel consumption of the engine.[2]

Adlercreutz, L., et. al., conducted theoretical research investigations on optimizing the combustion concept for CNG combustion on a single cylinder research engine. The overall goal was to reduce the CO_2 emissions in g/Kwh by 50% compared to modern gasoline engine, while trying to maintain the performance and characteristics of the engine. It was concluded that downsizing the engine and increasing its specific performance reduced the CO_2 emissions by 45% at full load and 25-34% on part load.[3]

Johnson, T., wrote a review on CO_2 emissions and fuel consumption reductions and the technologies involved from vehicles in the road transportation sector. Regarding the CO2 emissions reduction he writes that a plan was proposed by the United Nations for upwards of 80% CO_2 reductions from the 1990 levels by 2050. Engine technology trends are indicating nominally 15% reductions using advancements in currently utilized technologies. Many of the reductions will come from the use of direct injection technology in gasoline engines and downsizing diesel and gasoline engines for more specific power.[4]

Okmoto, K., et. al., conducted experimental investigations on spark ignition engines used in vehicles for reducing the CO_2 emissions as per Japanese 10-15 mode test. A new hybrid fuel technique was adopted in which a fuel additive was used with gasoline to act as a friction modifier. It was found that the fuel economy was improved by 2.4% with a 2.4% reduction in carbon dioxide emissions from vehicles. Further the improvements in engine power and vehicle acceleration were also observed.[5]

Bach, C., estimated that the 20% of all CO_2 emissions originate from road traffic. He further conducted studies on the natural gas hybrid passenger cars in order to reduce the CO_2 emissions. It was concluded that the use of Hybrid NGVs reduces the CO_2 emissions partly because of lower carbon content of the natural gas.[6]

Mueller, W., et. al., conducted experimental investigations with a new family of three way catalyst (TWC) in order to reduce the CO_2, HC, CO and NOx emissions from gasoline engines. It was concluded that the modifications in the catalyst design and its tuning with the engine performance was able to reduce the CO_2 emissions from the experimental engine.[7]

Warey, A., et. al., conducted analytical investigations on the possible reduction of CO_2 emissions by replacing the state of the art four stroke diesel engines by the two stroke engines for light duty vehicle applications. They developed 1-D models for a loop scavenged two stroke diesel engine and an opposed piston two stroke diesel engine. based on these models and the in-house vehicle models, projections were made for the CO_2 emissions for a representative light duty vehicle over the New European Driving Cycle and the Worldwide Harmonized Light Vehicles Test Procedure. It was concluded that the loop scavenged two stroke diesel engine had about 5-6% lower CO_2 emissions, while the opposed piston diesel engine had about 13-15% potential benefit.[8]

Chiara Guido, et. al., conducted experimental investigations on a 2.0 L Euro 5 compliant diesel engine in NG/Diesel based dual fuel mode. The effect of engine control parameters like CNG substitution ratio, diesel pilot injection strategies, EGR, boost pressure was investigated for possible reduction of GHG emission along with the reduction of other pollutants. It was concluded that there was more than 10% reduction of CO_2 emissions besides a significant reduction in THC (Total Hydrocarbon) emissions.[9]

Mufaddel Dahodwala, et.al., conducted computational investigations on advanced after treatment system and two advanced engine models to in order to develop technical knowhow to meet the phase 2 GHG and ultra-low NOx emission standards for heavy duty diesel engines. The advanced after treatment system included DOC, an SDPF, a standalone SCR and an ammonia slip catalyst calibrated against the experimental data. The engine models comprised the advanced technologies like downsizing, down-speeding, variable compression ratio, cylinder deactivation and turbo-compounding. The results indicated that with appropriate selection of engine and after treatment technology packages the 2027 Phase 2 GHG emission standards and the proposed 2024 ultra-low NOx emission standards could be achieved simultaneously.[10]

THEORETICAL BASIS[11]

THE CYLINDER , HIGH PRESSURE CYCLE, BASIC EQUATION.

The calculation of the high pressure cycle of an internal combustion engine is based on the first law of thermodynamics:

$$\frac{d(m_c.u)}{d\alpha} = -\frac{p_c.dV}{d\alpha} + \frac{dQ_F}{d\alpha} - \sum \frac{dQ_w}{d\alpha} - \frac{h_{BB}.dm_{BB}}{d\alpha}$$ ---(Eq.1)

where

$\frac{d(m_c.u)}{d\alpha}$ = change of the internal energy

in the cylinder.

$-\frac{p_c.dV}{d\alpha}$ = piston work.

$\frac{dQ_F}{d\alpha}$ = fuel heat input.

$\sum \frac{dQ_w}{d\alpha}$ = wall heat losses

$\frac{h_{BB}.dm_{BB}}{d\alpha}$ = enthalpy flow due to blow-by

$\frac{dm_{BB}}{d\alpha}$ = blow-by mass flow

The first law of thermodynamics for high pressure cycle states that the change of internal energy in the cylinder is equal to the sum of piston work, fuel heat input, wall heat losses and the enthalpy flow due to blow-by.

In order to solve this equation, models for the combustion process and the wall heat transfer, as well as the gas properties as a function of pressure, temperature, and gas composition are required.

Together with the gas equation

$p_c = \frac{1}{V}.m_c.R_o.T_c$ ---(Eq.2)

Establishing the relation between pressure, temperature and density, Eq. 2 for in-cylinder temperature can be solved using a Runge-Kutta method. Once the cylinder gas temperature is known, the cylinder gas pressure can be obtained from the gas equation.

COMBUSTION MODEL

Air Requirement And Heating Value modeling is given below.

STOICHIOMETRIC AIR-FUEL MIXTURE

The following equation for the stoichiometric air requirement specifies how much air is required for a complete combustion of 1 kg fuel:

$L_{st}=137.85(\frac{c}{12.01}+\frac{h}{4.032}+\frac{s}{32.06}-\frac{o}{32.0}$ [kgAir/kgFuel] --------------------------------(Eq.3)

LEAN MIXTURE

For lean combustion, the total heat supplied during the cycle can be calculated from the amount of fuel in the cylinder and the lower heating value of the fuel.

RICH MIXTURE

In rich air fuel mixture combustion, the total heat supplied during the cycle is limited by the amount of air in the cylinder. The fuel is totally converted to combustion products even if the amount of air available is less than the amount of stoichiometric air.

HEATING VALUE

The lower heating value is a fuel property and can be calculated from the following formula:

H_u = 34835 . c +93870 . h +6280 . n +10465 . s -10800 . o -2440 . w [kj/kg] --------(Eq.4)

HEAT RELEASE APPROACH.

VIBE TWO ZONE

The rate of heat release and mass fraction burned is specified by the Vibe function given by equation No.5 below.

The first law of thermodynamics is applied separately to the burned and unburned mixture while assuming that the temperatures of these two mixtures is different.

$$\frac{dx}{d\alpha} = \frac{a}{\Delta\alpha_c}(m+1)\,.y^{m}\,.\,e^{-a.y(m+1)} \quad \text{------(Eq.5)}$$

$$dx = \frac{dQ}{Q} \quad \text{------(Eq.6)}$$

$$y = \alpha - \frac{\alpha_0}{\Delta\alpha_c} \quad \text{------=(Eq.7)}$$

The integral of the vibe function gives the fraction of the fuel mass which was burned since the start of combustion:

$$x = \int\left(\frac{dx}{d\alpha}.d\alpha\right) = 1-e^{-a.y(m+1)} \quad \text{------(Eq.8)}$$

GAS EXCHANGE PROCESS

BASIC EQUATION

The equation for the simulation of the gas exchange process is also the first law of thermodynamics:

$$\frac{d(m_c.u)}{d\alpha} = -\frac{p_c.dV}{d\alpha} - \sum\frac{dQ_w}{d\alpha} + \sum\frac{dm_i}{d\alpha.h_i} - \sum\frac{dm_e}{d\alpha.h_e} \text{-----------------------------------(Eq.9)}$$

The variation of the mass in the cylinder can be calculated from the sum of the in-flowing and out-flowing masses:

$$\frac{dm_c}{d\alpha} = \sum\frac{dm_i}{d\alpha} - \sum\frac{dm_e}{d\alpha} \text{--(Eq.10)}$$

PISTON MOTION

Piston motion applies to both the high pressure cycle and the gas exchange process.

For a standard crank train the piston motion as a function of the crank angle α can be written as:

$$s=(r+l).\cos\psi-r.\sqrt{1-\{\frac{r}{l}.\sin(\psi+\alpha)-\frac{e}{l}\}^2} \text{---(Eq.11)}$$

$$\psi = \arcsin(\frac{e}{r+l}) \text{ --(Eq.12)}$$

HEAT TRANSFER

The heat transfer to the walls of the combustion chamber, i.e. the cylinder head, the piston, and the cylinder liner, is calculated from:

$Q_{wi} = Ai\ .\alpha_w\ .\ (T_c\text{-}T_{wi})$ --(Eq.13)

In the case of the liner wall temperature, the axial temperature variation between the piston TDC and BDC position is taken into account:

$$T_L = T_{L,TDC}\ .\ \frac{1-e^{-cx}}{x.c} \text{ ---(Eq.14)}$$

$$c = \ln\{\frac{T_{L,TDC}}{T_{L,BDC}}\} \text{ --(Eq.15)}$$

For the calculation of the heat transfer coefficient, the Woschni 1978 heat transfer model is used.

WOSCHNI MODEL

The woschni model published in 1978 for the high pressure cycle is summarized as follows:

$$\alpha_w = 130.D^{-0.2}.p_c^{\,0.8}.T_c^{\,-0.53}.\left[C_1.c_m + C_2.\frac{V_D.T_{c,1}}{p_{c,1}.V_{c,1}}.(p_c - p_{c,o})\right]^{0.8}$$ --------------------------(Eq.16)

C1 = 2.28+0.308.cu/cm

C2 = 0.00324 for DI engines

For the gas exchange process, the heat transfer coefficient is given by following equation:

$$\alpha_w = 130.D^{-0.2}.p_c^{\,0.8}.T_c^{\,-0.53}.(C_3.c_m)^{0.8}$$ ---(Eq.17)

$C_3 = 6.18+0.417.c_u/c_m$

FUEL INJECTOR

The fuel injector model is based on the calculation algorithm of the flow restriction. This means that the air flow rate in the fuel injector depends on the pressure difference across the injector and is calculated using the specified flow coefficients.

For the injector model, a measuring point must be specified at the location of the air flow meter. In this case the mean air flow at the air flow meter location during the last complete cycle is used to determine the amount of fuel. As is the case for continuous fuel injection, the fuelling rate is constant over crank angle.

PIPE FLOW

The one dimensional gas dynamics in a pipe are described by the continuity equation

$$\frac{\partial \rho}{\partial t} = -\frac{\partial(\rho.u)}{\partial x} - \rho.u.\frac{1}{A}.\frac{dA}{dx},$$ --(Eq.18)

the equation for the conservation of momentum

$$\frac{\partial(\rho.u)}{\partial t} = -\frac{\partial(\rho.u^2 + p)}{\partial x} - \rho.u^2.\frac{1}{A}.\frac{\partial A}{\partial x} - \frac{F_R}{V},$$ -------------------------------------(Eq.19)

and by the energy equation

$$\frac{\partial E}{\partial t} = -\frac{\partial[u.(E+p)]}{\partial x} - u.(E+p).\frac{1}{A}.\frac{dA}{dx} + \frac{q_w}{V}.$$ ------------------------------------(Eq.20)

The wall friction force can be determined from the wall friction factor λ_f :

$$\frac{FR}{V} = \frac{\lambda f}{2.D}.\rho.u.|u|$$ --(Eq.21)

Using the Reynold's analogy, the wall heat flow in the pipe can be calculated from the friction force and the difference between wall temperature and gas temperature:

$$\frac{q_w}{V} = \frac{\lambda_f}{2.D}.\rho.|u|.c_p.(Tw - T) \text{ --(Eq.22)}$$

During the course of numerical integration of the conservation laws defined in the Eq.20, Eq.21 and Eq.22, special attention should be focused on the control of the time step. In order to achieve a stable solution, the CFL criterion (stability criterion defined by Courant, Friedrichs and Lewy) must be met:

$$\Delta t \leq \frac{\Delta x}{u + a} \text{ ---(Eq.23)}$$

This means that a certain relation between the time step and the lengths of the cells must be met. The time step to cell size relation is determined at the beginning of the calculation on the basis of the specified initial conditions in the pipes. However, the CFL criterion is checked every time step during the calculation. If the criterion is not met because of significantly changed flow conditions in the pipes, the time step is reduced automatically.

An ENO scheme is used for the solution of the set of non-linear differential equations discussed above. The ENO scheme is based on a finite volume approach. This means that the solution at the end of the time step is obtained from the value at the beginning of the time step and from the fluxes over the cell borders

RESULTS AND DISCUSSIONS

EFFECT OF SPEED ON THE GHG (CO_2) EMISSIONS PRODUCED BY THE ENGINE IN DIESEL AND METHANE 90% and DIESEL 10% DUAL FUEL MODES

The fig.1 below shows the effect of speed on the GHG emissions produced by the engine in the diesel and methane 90% and diesel 10% dual fuel modes. It is seen from the figure that the GHG emission increases by increasing the speed of the engine with both with neat diesel mode as well the methane 90% and diesel 10% dual fuel modes. This is because the engine executes more number of power cycles per unit time at higher engine speeds. Further the engine produces higher GHG emissions with neat diesel as compared to the methane 90% and diesel 10% dual fuel mode operation. The reason is that the number of carbon atoms per molecule are reduced drastically with methane 90% component of the methane 90% and diesel 10% dual fuel as compared to neat diesel. The drop in the GHG emissions with methane 90% and diesel 10% dual fuel varies between 28.57% at the lowest speed of 1000rpm to 26.86% at 4000rpm.

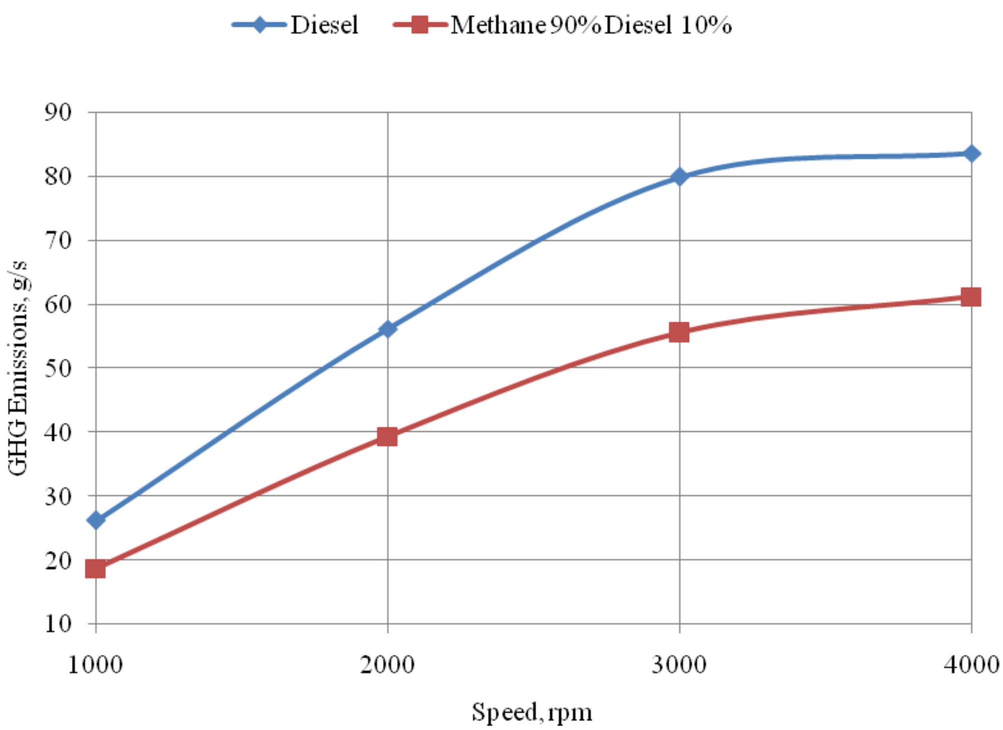


Fig.1 Effect of Speed on The GHG (CO_2) Emissions Produced by The Engine With Diesel and Methane 90% and Diesel 10% Dual Fuel Mode

EFFECT OF SPEED ON THE MASS FRACTION OF GHG (CO_2) EMISSIONS IN THE EXHAUST GAS PRODUCED BY THE ENGINE IN DIESEL AND METHANE 90% AND DIESEL 10% DUAL FUEL MODES

The fig.2 below shows the effect of speed on the mass fraction of the GHG emissions in the exhaust gas produced by the engine with neat diesel and methane 90% and diesel 10% dual fuel modes. It is clear from the graph that on the basis of the mass fraction of the exhaust gas of the engine, the GHG emissions are reduced by 5.36% with methane 90% and diesel 10% dual fuel mode as compared to its conventional diesel engine mode. This is due to lowest number of carbon atoms in the methane component of the methane 90% and diesel 10% dual fuel as compared to neat diesel fuel. Further it is seen from the graph that on percentage basis the mass fraction of GHG in the exhaust gas of the engine remains the same over the entire range of the speed of the engine with either of the two fuels under consideration under constant air-fuel ratio or constant load operation.

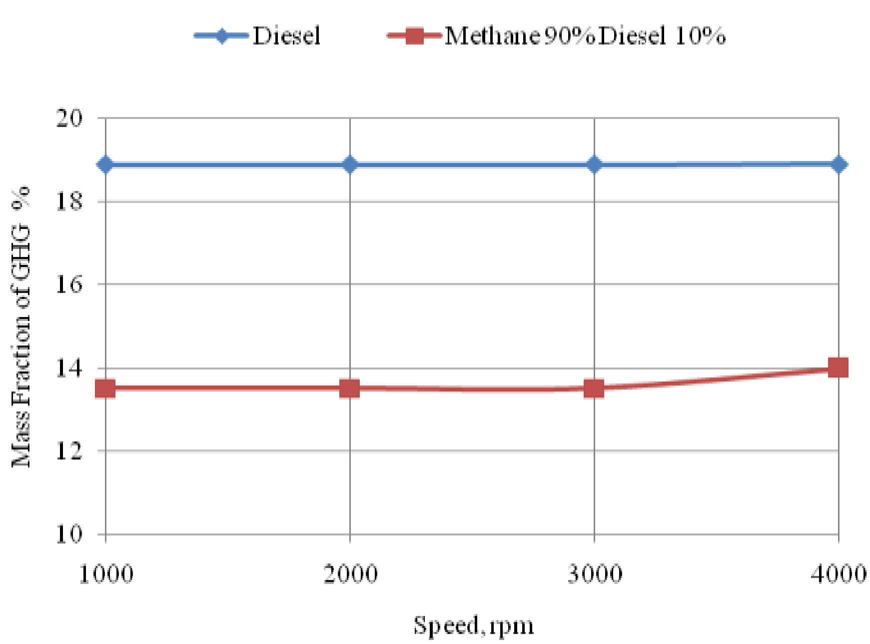
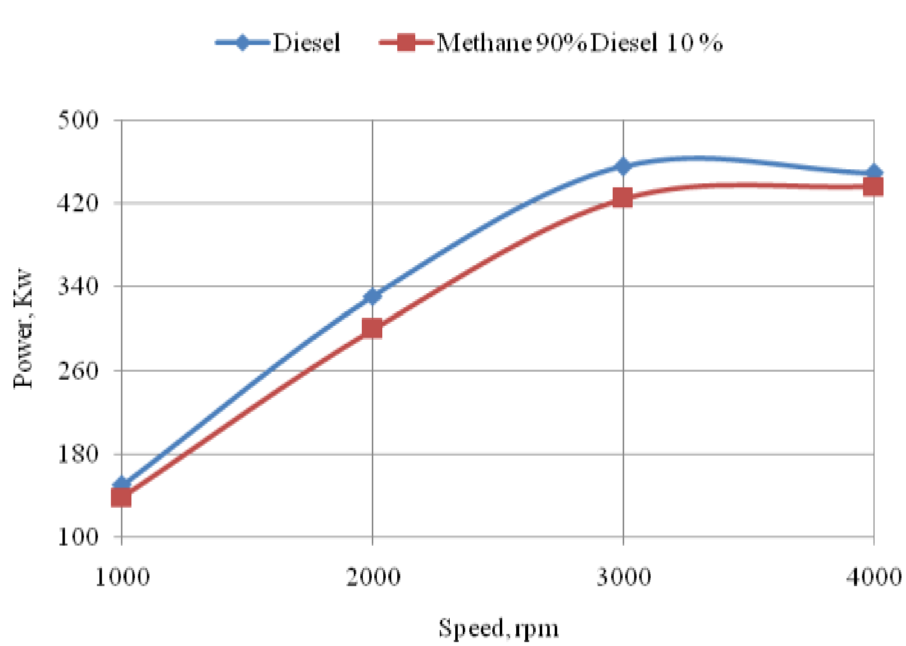


Fig.2 Effect of Speed on The Mass Fraction of GHG (CO_2) in The Exhaust Gas of The Engine in The Diesel and Methane 90% and Diesel 10% Dual Fuel Modes

EFFECT OF SPEED ON THE POWER DEVELOPED BY THE ENGINE IN DIESEL AND METHANE 90% AND DIESEL 10% DUAL FUEL MODES

The Fig.3 below shows the effect of speed on the power developed by the engine with neat diesel and methane 90% and diesel 10% dual fuel operation modes. It is seen from the figure that the power developed by the engine increases by increasing the speed of the engine with both neat diesel fuel and methane 90% and diesel 10% dual fuel modes. This is because the engine executes more number of power cycles per unit time at higher speeds. Further the engine develops higher power with neat diesel as compared to the methane 90% and diesel 10% dual fuel mode. This is due to lower volumetric efficiency with gaseous component of methane in the dual fuel mode of methane 90% and diesel 10%..



Effect of Speed on The Power Developed by The Engine with Diesel and Methane 90% and Diesel 10% Dual Fuel Mode

EFFECT OF SPEED ON THE TORQUE DEVELOPED BY THE ENGINE IN THE DIESEL AND METHANE 90% AND DIESEL 10% DUAL FUEL MODES

The Fig.4 below shows the effect of engine speed on the torque developed by the engine with neat diesel and methane 90% and diesel 10% dual fuel mode operations. It is seen that the torque developed by the engine varies with respect to speed with both the modes of fuel. The engines develops highest torque at the crank shaft rotational speed of 2000rpm with both modes of the fuels. This is because the combustion characteristics of the engine are the best at the speed of 2000rpm under the design and operating conditions under consideration. Further the engine develops higher torque with neat diesel fuel as compared to methane 90% and diesel 10% dual fuel. This is because the volumetric efficiency of the engine gets reduced by introducing a major component of the dual fuel in the gaseous phase.

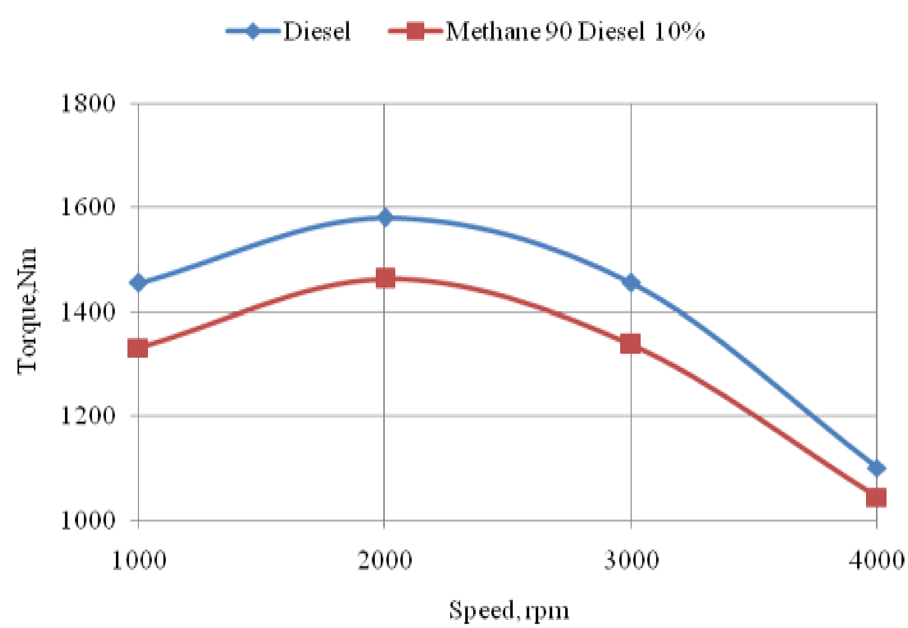


Fig.4 Effect of Speed on The torque Developed by The Engine with Diesel and Methane 90% and Diesel 10% Dual Fuel Modes

EFFECT OF SPEED ON THE BRAKE SPECIFIC FUEL CONSUMPTION OF THE ENGINE IN THE DIESEL AND METHANE 90% AND DIESEL 10% DUAL FUEL MODES

The Fig.5 below shows the effect of speed on the brake specific fuel consumption of the engine in the neat diesel mode and the dual fuel mode of methane 90% and diesel 10%. It is seen from the figure that the fuel consumption per unit energy output of the engine varies with speed for both the fuels neat diesel and methane 90% and diesel 10% dual fuel. The fuel consumption of the engine per unit of energy output of the engine is lowest at the engine speed of 2000rpm. This is because this is the optimum speed of the engine for best possible combustion process. With this operating speed the engine consumes lesser fuel in case of both the modes of fuel operation. Further the brake specific fuel consumption of the engine in the dual fuel mode of methane 90% and diesel 10% is lower than that in the neat diesel mode. This is because the drop in the fuel consumption rate in methane 90% and diesel 10% mode in more than the corresponding drop in power of the engine in the same mode of the fuel as compared to neat diesel fuel..

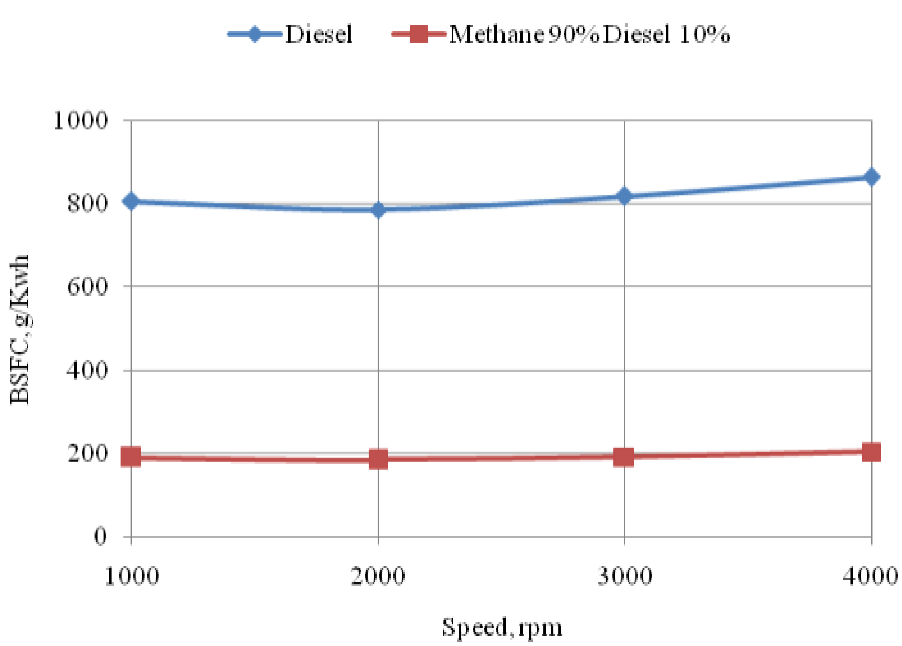


Fig.5 Effect of speed on The Brake specific Fuel Consumption of The Engine With Diesel and Methane 90% and Diesel 10% Dual Fuel Modes

CONCLUSIONS

1. The GHG emissions produced by the engine with methane 90% and diesel 10% dual fuel mode is lower than that produced by the engine in its neat diesel version.
2. The GHG emissions reduction varies between 28.57% at the lowest speed of 1000rpm to 26.86% at 4000rpm, the rated speed of the engine.
3. On the basis of the mass fraction of the exhaust gas of the engine the GHG emissions are reduced by 5.36% with methane 90% and diesel 10% dual fuel mode as compared to its conventional diesel engine mode.
4. The power and torque developed by the engine with methane 90% and diesel 10% dual fuel mode is satisfactory as compared with the conventional diesel engine.
5. The drop in power with methane 90% and diesel 10% dual fuel mode varies between 8% at the speed of 1000rpm to 2.88% at the speed of 4000rpm when compared with neat diesel engine.
6. The drop in torque with methane 90% and diesel 10% dual fuel operation varies between 8.58% at the engine speed of 1000rpm to 5.09% at the engine speed of 4000rpm.
7. The brake specific fuel consumption of the engine is much lower with methane 90% and diesel 10% dual fuel mode as compared to that of the conventional diesel engine. The reduction in fuel consumption with methane 90% and diesel 10% dual fuel mode operation varies between 76.41% at the engine speed of 1000rpm to 76.34% at the maximum rated engine speed of 4000rpm.
8. In order to make the above concluded GHG emissions reduction a reality the conventional diesel engine needs to be modified and supplied with methane as a major fuel by port fuel injection system. The combustion will be initiated by the pilot injection of diesel fuel directly into the engine cylinder towards the end of the compression process. Both the fuel supply systems can be controlled by means of the embedded electronic engine management system involving the use of a micro-controller.

ACKNOWLEDGEMENT

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APPENDIX-A

NOMENCLATURE

a = speed of sound, m/sec
A = pipe cross-section, m^2
A_{eff} = effective flow area, m^2
A_i = surface area (cylinder head, piston, liner), m2
AF_{CP} = air fuel ratio of combustion products
A_{geo} = geometrical flow area, m^2
c = mass fraction of carbon in the fuel
c_V = specific heat at constant volume, J/Kg.K
c_p = specific heat at constant pressure, J/Kg.K
Cm = mean piston speed, m/sec
Cu = circumferential velocity, m/sec
c_u = circumferential velocity, m/sec
D = cylinder bore, m
dm_i = mass element flowing into the cylinder, kg
dm_e = mass element flowing out of the cylinder, kg
d_{vi} = inner valve seat diameter (reference diameter), m
$\frac{dm_{BB}}{d\alpha}$ = blow-by mass flow, kg/degree of crank angle
e = piston pin offset, m
E = energy content of the gas (=ρ. $\bar{c_V}.T + \frac{1}{2}.\rho.u^2$) J
f = fraction of evaporation heat from the cylinder charge
F_R = wall friction force, N
h = mass fraction of hydrogen in the fuel
h_{BB} = enthalpy of blow-by, J/Kg
h_i = enthalpy of in-flowing mass, J/Kg
h_e = enthalpy of the mass leaving the cylinder
H_u = lower heating value, J/Kg
k = ratio of specific heats
l = con-rod length, m
m = shape factor
$\dot{m}$ = mass flow rate, kg/sec
m_c = mass in the cylinder, kg
m_{ev} = evaporating fuel, kg
m_{pl} = mass in the plenum, kg
n = mass fraction of nitrogen in the fuel
o = mass fraction of oxygen in the fuel
p = static pressure, bar
P_{01} = upstream stagnation pressure, bar
Pc,o = cylinder pressure of the motored engine ,bar
Pc,1 = pressure in the cylinder at IVC, bar
p_{pl} = pressure in the plenum, bar

p_c = cylinder pressure, bar
p_2 = downstream static pressure, bar
q_{ev} = evaporation heat of the fuel, J/kg
q_w = wall heat flow, J
Q = total fuel heat input, J
Q_F = fuel energy, J
Q_{wi}= wall heat flow (cylinder head, piston, liner), J
r = crank radius, m
R_0 = gas constant, J/mol.K
s = piston distance from TDC, m
t = time , sec
T = temperature, K
Tc,1= temperature in the cylinder at intake valve closing (IVC), K
T_c = gas temperature in the cylinder, K
T_{wi} = wall temperature (cylinder head, piston, liner), K
T_L = liner temperature, K
$T_{L,TDC}$ = liner temperature at TDC position, K
$T_{L,BDC}$ = liner temperature at BDC position, K
T_w = pipe wall temperature, K
T_{01}= upstream stagnation temperature, K
u = specific internal energy, J/Kg
u = flow velocity, m/sec
V = cylinder volume, m^3
V = cell volume (A.dx), m^3
VD = displacement per cylinder, m^3
w = mass fraction of water in the fuel
x = relative stroke (actual piston position related to full stroke)
α = crank angle, degrees
α_o= start of combustion, crank angle degrees
$\Delta\alpha_c$= combustion duration, crank angle degrees
α_w = heat transfer coefficient, $J/m^2.K$
ρ = density, kg/m^3
μσ = flow coefficient of the port
ψ = crank angle between vertical crank position and piston TDC position, degrees
λf = wall friction coefficient
Δt = time step, sec
Δx= cell length, m

APPENDIX-B

Engine Specifications	
Engine Type	Four Stroke
Method Of Ignition	Compression Ignition With Pilot Injection of Diesel
Displacement, cc	6125
Compression Ratio	18
Number Of Cylinders	6
Rated Speed, rpm	4000

APPENDIX-C

PHYSICAL AND CHEMICAL PROPERTIES OF DIESEL AND METHANE[12]

Fuel Property	Diesel	Methane
Formula	C8 TO C25	CH4
Density, g/m_3	820	725
Lower heating value, MJ/Kg	42.5	45
Stoichiometric air-fuel ratio, weight	14.5	17.24
Octane No.	-	120
Cetane No	51	-

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