IOSR Journal of Engineering (IOSRJEN) **www.iosrjen.org**
ISSN (e): 2250-3021, ISSN (p): 2278-8719
PP 16-21

A Review on Performance and Emission Characteristics of a Dual Fuel Engine

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Abstract: *Abatement of emissions from transport sector is one of the major concerns throughout the globe. One of the technical challenges in transportation sector is to reduce emissions from diesel engine and simultaneously to meet the satisfactory performance; hence Dual fuel engines are most attractive due to its lower emission levels in comparison with the conventional diesel engine particularly at full loads. However based on my studies of distinct conclusion of authors, it is required to study the dual fuel combustion process with more information at part loads due to the poor performance and high CO and UHC emissions at these conditions. It is found that at part loads, areas that are influenced by diesel diffusion flames are ignited and premixed charge flame could not propagate properly as the combustion process is slow and leads the heat release to be drawn more towards the expansion stroke which causes incomplete combustion and consequently high amount of UHC and CO will be emitted. Also, at part load due to low gas temperature in the environment of diesel spray and low diesel/ biodiesel fuel temperature, diesel liquid droplets evaporate lately which are far from injector nozzle. But the diesel diffusion flames and premixed charge flame could propagate suitably at full load due to high gas temperature. The poor performance of dual fuel engine at part load can be enhanced by certain operating parameters like increasing diesel fuel quantity, enhancing injection pressure, advancing injection timing, multiple injections, increasing number of diesel injectors, optimizing intake charge temperature and pressure.*

Keywords: *Dual fuel engine, Part load, Full load, premixed charge flame, Diffusion Flames*

I. Introduction

Dual fuel engines is an alternate way of conventional compression ignition engine in which it operates with high volatile liquid fuel or gaseous fuel with high octane number is inducted along with air trough the intake manifold as primary fuel. The resulting homogenous mixture is compressed. A pilot fuel with high cetane number is injected through the conventional injection system as secondary fuel. This self ignites and initiates the combustion in the primary fuel air mixture. The dual fuel engine has many advantages over the conventional diesel engine in both the technical and environmental aspects. As the dual fuel name indicates, it can be used as blending, fumigation and Emulsion technique, but the exact meaning of dual fuel engine is that fumigation method.

Blending:In this method bio fuel and diesel are premixed with desired proportion and kept in a single tank for injecting fuel in combustion chamber. This type of engine doesn’t needs any modification.

Fumigation: Fumigation is the most common method of dual fuel concept for using alcohol as primary fuel in CI engine. In this method bio fuel is injected into the intake manifold to mix with fresh air by spraying or carburetting. This method requires additional components such as carburettor, Vaporiser or injector along with separate fuel tank line and control is required to inject bio fuel into intake manifold.

Emulsion: It is the Process of mixing two immiscible substances (in which two different phases are mixed together)

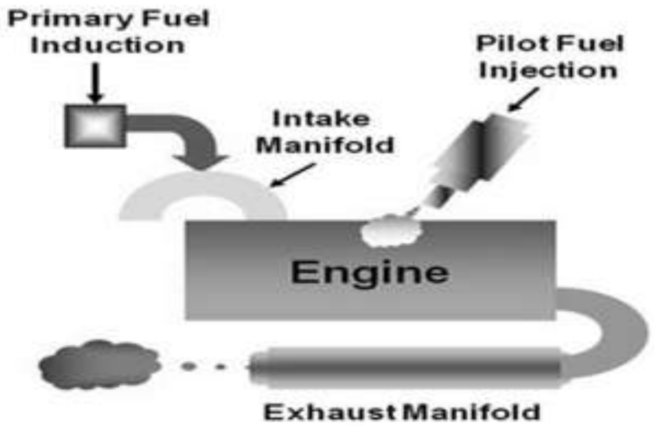

Fig.1: Dual Fuel Concept

II. Literature Survey

2.1 Seyed Mohammed Mousavi et.al In this study, medium duty turbocharged OM-355 direct injection diesel engine converted to dual fuel mode is used as a dual fuel engine for a numerical investigation on combustion and emission characteristics of dual fuel engine at part load conditions and found that there are good agreements are made between experimental results and numerical results especially at ignition and peak pressure times at all operating conditions. At part loads, it is seen that there is some difference between the measured and predicted pressure data during the combustion and expansion phases. The combustion process at full load starts earlier than part load owing to high in- cylinder temperature and pressure. Slow progress of combustion process at part load leads heat release to be drawn more towards the lower temperature expansion stroke. To overcome the part load disadvantages, any plan to enlarge the size of diesel combustion region or increase the flammability limit of premixed natural gas could be beneficial. Increasing diesel fuel quantity, multiple injection , advancing diesel fuel injection, multiple injection, restricting intake air quantity, increasing the diesel fuel injector nozzle and optimizing the intake charge temperature and pressure. However it shown the effect of increasing the diesel fuel quantity on improving the methane combustion at part load condition, the diffusion flame penetration of diesel fuel is increased and hence very lean mixture that are influenced by the diesel diffudion flame could be ignited prolperly and propagated rapidly [1].

2.2 M.J. Abedin et.al In this paper. Comparative engine performance and emission characteristics of different techniques involved in diesel engine as duel fuel operation were studied.The blend mode on diesel engine increases the brake specific fuel consumption (BSFC) at every case due to lower calorific value, high density and viscosity of alcohol and biodiesel than diesel. It decreses the Brake thermal efficiency (BTE) due to poor atomization , vapourisation and higher heat of vapourisation of alcohol , hence it increases the fuel consumption. NOx emission decreases due to low heating value (LHV). Fumigation mode increases the BSFC due to higher heat of evaporation of biofuel which abates the incylinder gas temperature during the combustion process, hence to support the complete combustion and to generate the required amount of power, more fuel is delivered to engine resulting higher BSFC. At low engine loads, due to presence of large amount of air,the intake air and the fumigated biofuel form a poor mixture which reduces the brake thermal efficiency. Increase in UHC and CO emission is observed in fumigation mode. Unburned fumigated biofuel forms its quench layer inside the cylinder, due to collong effect of biofuel, combustion temperature is poor during expansion.Due to alcohol lower calorific value, a smaller amount of heat emitted throughout the combustion process ans as a result combustion temperature is reduced which results in reduced NOx formation.PM reduces significantly in fumigation mode. In emulsion method, less amount of diesel is substituted by the same amount of water which reduces the average temperature of combustion. As a result, more fuel is required to produce work which increases the BSFC but the BTE is also increases since the combustion efficiency is is improved due to better atomization and evaporation of fuel. Micro emulsion causes the formation of fine spray and more vapourization of fuel. The comtinous braking of water droplets in emulsion process increases the evaporation surface of droplet and ensures the accurate mixing of burning fuel in air. As a result, combustion reaction and burning efficiency are improved. It reduces the NOx emission compared to neat biodiesel. Smoke opacity and PM emission significantly decrease in emulsion mode [2].

2.3 Hongyuan Wei et.al A diesel engine with 6-cylinder, turbocharged intercooling and common rail system was modified to operate on diesel/methanol dual fuel (DMDF). They have showed the effect of heat release rate (HRR), Incylinder Pressure, mass fraction burnt (MFB) for different Proportions of methanol fumigation. In general, the peak in-cylinder pressure and mean temperature before main combustion of single injection cases at different MSR is lower than that of operating points with pilot injection. When compared with pure diesel mode, the fumigation of

methanol increases regulated and unregulated emissions tested in this study except NOx and CO2. HC and CO emissions increase but NOx emissions decrease with rising MSR. Most of unregulated emissions increase with the rise of MSR but a reduction in unburned methanol emissions is observed when MSR changes from 30% to 50%. The application of pilot injection increases the in-cylinder pressure and temperature before main combustion compared with single injection conditions, thus avoids misfire combustion, and improves the combustion stability, unburned hydrocarbon and incomplete combustion products emissions of DMDF engine at high MSR under low load condition. Most regulated and unregulated gaseous emissions tested in this study reduce with the application of pilot injection except NOx, CO2 on M0 and M10 mode and toluene on M50 mode. Additionally, with larger amount of pilot quantity and earlier pilot injection timing, the beq on M50 mode reduces remarkably when compared with single injection case. With the increase of pilot quantity, the peak in-cylinder temperature and pressure increase gradually for a given MSR. The change of pilot injection quantity has little influence on almost all regulated and unregulated gaseous emissions on M0 and M10 mode. Regulated HC and CO emissions decrease but NOx emissions increase with rising pilot quantity. When MSR is 30%, the increasing pilot quantity could reduce all unregulated emissions tested in this study; but at higher MSR (M50 mode), unburned methanol, 1,3-butadiene and benzene emissions increase gradually with rising quantity of pilot injection fuel[3].

2.4 Zhongshu Wang et.al Discussed about the impact of pilot diesel ignition mode on combustion and emission characteristics of a diesel and natural gas duel fuel heavy duty engine. To understand it, detailed study concerned injection timing was conducted at light loads. The testing engine was operated on a 6-cylinder turbocharged intercooler diesel/natural gas dual fuel heavy duty engine at light loads and diesel injection was controlled over the wide range. The investigated results shows that diesel injection timing (T_{inj}) has an obvious affect on pilot diesel ignition mode.A significant edvancing injection timing leads to pilot diesel ignition mode differs from traditional diesel engine compression ignition mode in the sense that it does not occur at a specific place in the spray which is a two stage autoignition mode. With advancing T_{inj} , engine combustion and emission characteristics including cylinder pressure and temperature , heat release rate, start of combustion (SOC), ignition delay, combustion delay, combustion duration, crank angle of 50% heat release rate, nitrogen oxides (NOx) and total hydrocarbon (THC), show completely different variation trends in different ignition modes. Overall, higher thermal efficiency and lower amission can be achieved simultaneously in two stage combustion /autoignition mode and concluded that the satisfactory results is obtained with higher brake thermal efficiency of 35% , lower NOx (60 ppm) and THC (0.4%) at T_{inj} is 42.5⁰CA BTDC.

2.5 Qiang Zhang et al An experimental investigation was carried out to study the effect oj injection timing, direct injection pressure, substitution rate of natural gas as well as engine load on the combustion and emission characteristics of an electronically controlled common rail diesel/natural gas dual fuel engine. The test was conducted at a constant speed of 1200rpm, while other parameters are varied in reference to different study purpose. The combustion process of in-cylinder pressure could be identified to take place in three stages. The first stage is the premixed combustion phase of diesel, due to the high activation energy and the resulting high auto-ignition temperature of natural gas, the combustion of premixed diesel formed during the pilot ignition delay is the main contributor of this stage, during which, a rapid pressure rise occurs resulting from the initiation of diesel combustion. The second stage involves the diffusion combustion of pilot diesel along with the initiation and flame propagation of natural gas combustion; in this stage, the combustion of pilot diesel is mixing controlled and premixed natural gas flame propagates from the ignition kernels. The third stage is the late combustion phase of natural gas in the vicinity of liner wall or piston wall. When the injection timing of pilot diesel is advanced, the heat release before top dead centre increases, the peak pressure raises accordingly. Therefore, the maximum in-cylinder pressure shows a consistent increasing trend with the advancing injection timing at both substitution rates. In their study, minor difference between peak pressures of different injection timings was revealed at similar operating condition. At the substitution rate of 60%, diesel takes a bigger proportion, leading to longer injection duration and subsequent increased spray penetration. Thus the ignition power is enhanced and the burning rate of natural gas is raised. Effect of diesel injection pressure is concluded from the effect of injection pressure on in-cylinder pressure that faster pressure rise, higher maximum in cylinder pressure as well as earlier detachment from compression line with raised diesel rail pressure (DRP) are revealed by the in-cylinder pressure traces. This change is more distinguishable when DRP is increased from 800 bar to 1100 bar due to the significantly improved atomization and air entrainment of diesel spray along with the correspondingly advanced combustion. When DRP is raised above 1100 bar, the formation of premixed mixture and the subsequent combustion process are less affected by DRP, hence the difference between in cylinder pressure curves is difficult to recognize in the range of 1100 bare1400 bar. Additionally, the maximum in cylinder pressure appears earlier with higher DRP and the peak values at 70% substitution rate are higher and occurs earlier than that of 80% substitution rate. Effect of natural gas substitution rate shows, the maximum in-cylinder pressure decreases with the increase of substitution rate of natural gas, albeit occurs at approximately the same crank angle. The peak value of in-cylinder pressure is little affected by substitution rate when it is varied from 30% to 60%; whilst the increasing magnitude is larger at substitute rate higher than 70%. Additionally, the rate of pressure rise is also strongly influenced by substitution rate of natural gas; the maximum value increases and the slope of the rising section becomes steeper with reducing substation rate, the peak value of pressure rise rate for 30% substitution rate is twice of that for 80%, indicating significantly deteriorated combustion noise and durability. the effect of engine load on in-cylinder pressure and rate of pressure rise respectively. It can be found that the curve of in-cylinder pressure rises with the increase of engine load owing to increased fuel mass and raised boost ratio. The rate of pressure rise and contributions to the pressure rise rate resulted from combustion of natural gas increases with increasing engine load and profiles of two-peak appear at higher loads.

2.6 Yangyang Li et.al The diesel–methanol dual-fuel (DMDF) combustion mode is conducted on a six-cylinder, turbo-charged, inter-cooled diesel engine. In DMDF mode, methanol is injected into the intake pipe to premix methanol and air, and then ignited by the direct-injected diesel in cylinder. This study is aimed to investigate the effects of rapid burning characteristics on the vibration performance of the DMDF engine. The experimental results show that the combustion process of the DMDF engine can be divided into two phases. The rapid burning characteristics, the centroid angle of rapid burning (a) and the rapid burning fraction (b) have important influence on the vibration characteristics of the DMDF engine. Effect of CCR shows the heat release rate at different CCRs, the rapid burning characteristics (a and b) at different CCRs, which are calculated based on the data. It is shown that a increases slightly (about 1 _CA) with the increase of CCR. The reasons can be shown in the following two points. The first is that the quantity of premixed fuel increases with the increase of CCR before ignition, which may promote the production of active radicals and advance the ignition timing. The second is that the increased quantity of premixed fuel enlarges the rapid burning duration. It is also shown that b increases dramatically with the increase of CCR, and when CCR is 50.73%, 40.96%, 21.25%, and 0%, b values are 45.58%, 40.61%, 25.91% and 19.60%, respectively. Sometimes, b is smaller than the corresponding CCR, which means that the premixed methanol–air mixture burns not only at the rapid burning phase but also at slow burning phase. Effect of engine load shows the heat release rate at different loads. A first increases and then decreases and the variety range is small (about 4 _CA). b decreases sharply first and then increases slightly, ams and dp/dumax decrease first and then increase at the same time. The turning point is 60% load, which means that the DMDF engine operates most smoothly at moderate load. This is mainly because the increasing in-cylinder temperature leads to decreasing ignition delay period with the increase of load. Effect of diesel injection timing shows the heat release rate at different diesel injection timings. It is shown that with the decrease of injection advance angle, the second peak of the heat release rate becomes more and more unapparent. When injection timing is 0 _CA BTDC, the second peak disappears, and it is considered that the combustion is the complete rapid burning. Effect of diesel injection pressure shows the heat release rate at different diesel injection pressures. It is shown that with the increase of injection pressure, a and b change slightly, but ams and dp/dumax increase obviously. The reason is that when the fuel injection quantity keeps the same, fuel injection duration may be shortened with the increase of injection pressure, which leads to the reduction of combustion duration and increase of heat release rate. The increase of heat release rate and reduction of combustion duration may enlarge the rise rate of in-cylinder pressure and lead to severe vibration of cylinder wall.

2.7 Silvana Di Iorio et.al The use of gas fuels in compression ignition engines through dual fuel technology represents a promising way to reach a good compromise among sustainable development, energy conservation and environmental preservation. Methane is particularly attractive because of its potential to reduce particulate emissions that are likely the most critical issue of diesel engine exhaust. The present work deals with an experimental activity carried out on a compression ignition engine modified to run in diesel/methane dual fuel mode. The engine is a three-cylinder, 1.0 L of displacement, equipped with a common rail injection system. Experiments were carried out at different engine speeds and loads. When diesel/methane DF operation is investigated, it is necessary to take into account that methane and diesel fuel have different physicochemical properties resulting in different combustion development. In the present research activity, pressure signals were detected in the three cylinders; negligible differences were observed in the pressure traces of the three cylinders in diesel and diesel/ methane fuelling. The rate of heat release (ROHR) and the bulk temperature were calculated by pressure data through the first law of thermodynamics and the perfect gas law and found indicated data of the second cylinder are plotted for conventional diesel and DF operation at all the investigated conditions. In conventional diesel operation, pressure histories exhibit two peaks due to the combustion of pilot and main fuel. When the engine runs in DF mode two peaks are still distinguishable because two diesel injections occur providing two different ignition sources for the air/methane charge.

At all investigated conditions, DF combustion is characterized by slightly lower pressure during the compression stroke. This trend could be due to the higher specific heat capacity ratio of methane with respect to the air resulting in lower charge temperature and, hence, lower in-cylinder pressure. Because of the longer ignition delay, DF combustion process occurs late in the piston down stroke thus showing lower pressure rise and lower pressure peaks with respect to conventional diesel fuelling. Since the whole combustion process is shifted towards the expansion stroke, DF operation is characterized by pressure traces of higher values, in this phase, with respect to diesel mode. A better understanding of combustion evolution is provided by ROHR evolution. In the operating conditions at 1600 rpm, diesel operation shows two peaks due to the pilot and main combustion. Both peaks are visible also in DF mode even if their behaviour is quite different because of the methane presence. DF mode has retarded start of combustion (SOC) of the pilot with respect to diesel operation; in particular, the combustion delay increase from 1.4 up to 2.4 cad when the load increase from 30% up to 70%. This trend could be ascribed to the increased methane concentration. The gas fuel, in fact, lowers the temperature in the chamber thus inhibiting the diesel ignition. Moreover, the longer diesel ignition delay is correlated to chemical factors due to the gas fuel presence. The methane mixed with air, in fact, alters the properties of the charge, the oxygen concentration and the pre-ignition reaction activities during compression. After the pilot peak, DF main combustion starts before the end of pilot event as evidenced by the fact that it starts at higher ROHR values than conventional diesel mode. This behaviour is due to the flame propagation of the methane charge, ignited by the pilot diesel fuel, and then further promoted by main diesel fuel. Concerning the main combustion of diesel operation, it shows a first premixed peak followed by a diffusive phase.

As result, more fuel is burnt in the last phase of combustion leading to higher temperature during expansion and exhaust stroke. Similar considerations regarding the differences in combustion evolution between diesel and DF operation can be done also at 2800 rpm. DF combustion, in fact, has retarded pilot combustion. Main combustion starts before the end of pilot event and it is characterized by lower rate with respect to diesel operation because of the slower burning velocity of the gaseous fuel. Moreover, DF main combustion is more strengthened with respect to diesel operation. How DF combustion diverges from the diesel one is particularly evident at 2800 rpm–35 Nm where both premixed and diffusive phase are clearly distinguishable for diesel operation while DF combustion seems to occur essentially in premixed mode. At 3200 rpm at all investigated loads , there is no evidence of the pilot combustion for both diesel and DF operations maybe because of the lower amount of fuel injected during the pilot event at high engine speeds. The absence of the pilot combustion makes clearer the effect of methane on main combustion. In these operating conditions, in fact, the slower combustion rate of the gaseous fuel results in delayed and longer combustion with respect to diesel fuel. [15].

III. Conclusion

The combustion process increment at full load starts earlier than part load condition owing to sufficient methane concentration and the rapid growth of flame. Hence ignition delay and combustion duration time at part load is longer than full load condition.

It is found that at part load because of low gas temperature in surrounding of diesel spray and low diesel fuel temperature, diesel liquid droplets evaporate lately and far from injector nozzles. Therefore, it causes diesel diffusion flame from spray of each injector nozzles to be developed, distinctly. This mentioned fact is the cause of flame structure differences at both conditions.

To overcome part load disadvantages, increasing diesel fuel quantity was investigated. At part load condition, by increasing the amount of diesel fuel quantity, with the assumption of constant input energy, diffusion flame penetration of diesel fuel is increased. Hence very lean methane/air mixture could be ignited suitably and propagated quickly due to enlarge the size of diesel combustion region.

The difference of pressure cylinder-to-cylinder decreases with the increase of engine speed, and causes the UED for various MSP decreases simultaneously. And the increases of engine speed can reduce the difference of combustion phase for different MSP. It can be considered raising MSP at high engine speed in order to gain better engine economy.

Blend, fumigation and emulsion, each of them increase the BSFC. BSFC is increased about 10% in blend mode, 7–12% in fumigation mode and 17% in emulsion mode. The increase of BSFC is the corollary of lower caloric value.

When compared with pure diesel mode, the fumigation of methanol increases regulated and unregulated emissions tested in this study except NOx and CO_2. HC and CO emissions increase but NOx emissions decrease with rising MSR. Most of unregulated emissions increase with the rise of MSR but a reduction in unburned methanol emissions is observed when MSR changes from 30% to 50%.

Advanced pilot injection timing increases the in-cylinder temperature and pressure before main combustion. The emissions of HC, CO, nitrous oxide, formaldehyde, formic acid and toluene always decrease and NOx always increase with the advance of pilot injection timing at different MSR. However, increased emissions in unburned methanol, 1, 3-butadiene and benzene are observed on M50 mode when advances pilot injection timing.

In two-stage autoignition mode, the brake thermal efficiency and THC emissions almost keep unchanged, and NOx emission drops dramatically with advancing diesel injection timing. But, too early injection certainly will lead to diesel impingement and poor atomization quality for lower incylinder temperature. Optimized result in twostage autoignition mode can be achieved with higher brake thermal efficiency (35%), lower NOx (60 ppm) and THC (0.4%) emissions, when diesel injection timing is fixed at 42.5^{o}CA BTDC.

IV. Nomenclature

UHC	-	unburned hydro carbon	THC	-	Total Hydrocarbon
BTDC	-	Before Top Dead Centre	CA	-	Crank Angle
BSFC	-	Brake Specific Fuel Consumption	T_{inj}	-	Injection Temperature
SOI	-	Start of Injection	SOC	-	Start of combustion
UED	-	Unevenness Degree	HRR	-	Heat Release Rate
MSP	-	Methanol substitution Percent			

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