EXPERIMENTAL ANALYSIS ON THE EFFECT OF NOZZLE HOLE NUMBER ON SINGLE CYLINDER FOUR STROKE DI DIESEL ENGINE WITH OLIVE OIL AND DIESEL BLENDS

K.Veeranjaneyulu[1], Dr T.Venkateswar Rao[2]
[1]Research Scholar in KLEF & Assistant Professor in Anurag Engineering College, Ananthagiri (V&M), Suryapet Dt,TS
[2]Department of Mechanical Engineering, Professor in DBS Institute of Technology, Kavali, SPSR Nellore Dt, AP

Abstract

The performance and emission characteristics of diesel engine depend on many parameters. One of the critical factors for controlling the rate of mixing of fuel and air is the number of injector holes in a diesel engine. The fuel injector orientation plays a vital role in fuel-air mixing. To achieve, fast and perfect air-diesel mixing is the most important requirements in engine research. Design of the diesel fuel injector is critical to the performance and emissions of modern diesel engines. These features not only affect the combustion characteristics of the diesel engine, they can also affect the stability of the emissions and performance over the lifetime of the engine and the mechanical durability of the injector. we propose additive of diesel & olive oil with number of holes for superior emissions and engine performance in diesel engine. The diesel engine test were conducted on a 4-stroke tangential vertical(TV1) single cylinder water cooled direct injection diesel engine. The effects of parameters of a fuel injector and combustion chamber geometries on nitrogen oxide (NO_x), soot and hydro carbons were investigated by means of a parametric study. One of the critical factors for controlling the rate of mixing of fuel and air is the number of injector holes in a diesel engine. The results show that increasing and decreasing the size of holes with additive of olive oil significantly influences evaporation, atomization, and combustion. However, when the size of holes and additive exceeds a certain threshold, there is an adverse effect on combustion and emissions due to a lack of the air entrainment required for the achievement of a stoichiometric mixture. The present investigation showed an improvement in brake thermal efficiency and reduction of mass fuel consumption and emission level.

I. INTRODUCTION

Among different types of thermal engines, diesel engine is regarded as a highly efficient combustion engine that plays an important factor in transportation, agriculture, and industry. Because of its efficient fuel economy and relatively lower engine raw emissions, direct-injection (DI) mode has become the main injection mode for diesel engines. The diesel engine is a type of internal combustion engine; more specifically, it is a compression ignition engine, in which the fuel ignited solely by the high temperature created by compression of the air-fuel mixture. The engine operates using the diesel cycle. The fuel injection system in a direct injection diesel engine is to achieve a high degree of atomization in order to enable sufficient evaporation in a very short time and to achieve sufficient spray penetration in order to utilize the full air charge. The fuel injection system must be able to meter the desired amount of fuel, depending on engine speed and load, and to inject that fuel at the correct time and with the desired rater Further on, depending on the particular combustion chamber, the appropriate spray shape and structure must be produced. Emissions regulations have become increasingly stringent in order to control these harmful emissions, which are detrimental to the environment as well as human beings.

For years the world has been in search of alternative methods to produce clean, renewable energy. From both an economic and environmental standpoint, alternative fuels are a necessity. Since 2001, the national average price of gasoline has risen by as much as $2.20 per gallon. Similarly, since 1979 carbon dioxide emissions have risen 70% . With the rising gas and oil prices and a lack of commercially available alternative energy, the search for alternative fuels has entered the home. In recent years, the concept of olive oil conversions for diesel cars has become more prevalent among consumers. Over three billion gallons of waste oil are drained from fryers in restaurants across the world each year. While this is not nearly enough oil to support the world's growing energy need, it has the ability to offset the foreign oil dependency of the United States by as much as 3,700,000 gallons per year or 1% (Ramadhas, 2004). Although this does not make it an attractive candidate for big businesses, it has found its way to the individual consumer, encouraging change on an individual level. Bio fuels are in the news as an answer to reducing our dependence on foreign oil and petroleum based fuels. Bio fuels are fuels that are derived from plant material such as soy oil or animal fats. The sources of these fuels are varied. They can come from plant oils derived from many different crops, and fats from slaughter houses and restaurant wastes. Bio fuels represent an ecologically friendly source of energy that is replaceable, unlike petroleum based fuels which are not. In many cases these fuels represent the saving of energy from waste material that would be thrown away and discarded onto a waste dump. Biodiesel is a non-fossil fuel alternative to petro diesel produced from biomass. It can be produced from vegetable oil or animal fats. Although biodiesel contains very different compounds than petroleum derived diesel, it has combustion properties very similar to the former including the energy content and cetane ratings (Ability to start cold) it is comparable to diesel. Biodiesel is composed of a mixture of "ESTERS" that are the products of the reaction of fatty acids (FFAs) and triglycerides with an alcohol. The alcohol usually used is methanol (CH3OH) with a catalyst, usually sodium hydroxide (NaOH) lye. Fatty acids are organic acids derived from natural fats and vegetable oils. They are naturally produced by a biosynthesis process involving a coenzyme acetyl-CoA. These acids when combined with glycerin become triglycerides which make up most animal fats and vegetable oils. Biodiesel can be blended with petrodiesel in any amount. Biodiesel can gel at a higher temperature than petrodiesel, blending in cold weather can solve this problem.

OLIVE OIL AS ADDITIVE: Olive oil is a vegetable oil obtained from the olive (Olea europaea), a traditional tree crop of the Mediterranean Basin. It is used in cooking, cosmetics, soaps, and as a fuel for traditional oil lamps. Olive oil is regarded as a healthy dietary oil because of its high content of monounsaturated fat (mainly oleic acid) and polyphenols.Olive oil is a liquid fat obtained from olives (the fruit of *Olea europaea*; family Oleaceae), a traditional tree crop of the Mediterranean Basin. The oil is produced by pressing whole olives. It is commonly used in cooking, whether for frying or as a salad dressing. It is also used in cosmetics, pharmaceuticals, and soaps, and as a fuel for traditional oil lamps, and has additional uses in some religions. There is limited evidence of its possible health benefits. The olive is one of three core food plants in Mediterranean cuisine; the other two are wheat and grapes.

PROPERTIES OF OLIVE OIL:

Melting point	6C^o
Boiling point	300C^o
Smoke point	190C^o
Specific gravity	0.9150-0.9180
Viscosity	8.5 cP
Iodine value	75-94
Saponification value	184-196
Cetane number	33
Refractive index	1.467

FUEL INJECTOR NOZZLES:

Nozzle is the part of an injector through which the fuel is injected into the combustion chamber. Design of nozzle should be such that the liquid fuel leaving the nozzle is atomised which helps in proper mixing of fuel and air. Diesel injector nozzles are spring-loaded closed valves that spray fuel directly into the combustion chamber or pre-combustion chamber when the injector is opened. Injector nozzles are threaded or clamped into the cylinder head, one for each cylinder, and are replaceable as an assembly.The tip of the injector nozzle has many holes to deliver an atomized spray of diesel fuel into the cylinder of the Engine. Parts of a diesel injector nozzle include:

- Heat shield. This is the outer shell of the injector nozzle and may have external threads where it seals in the cylinder head.
- Injector body. This is the inner part of the nozzle and contains the injector needle valve and spring, and threads into the outer heat shield.
- Diesel injector needle valve. This precision machined valve and the tip of the needle seal against the injector body when it is closed. When the valve is open, diesel fuel is sprayed into the combustion chamber. This passage is controlled by a computer-controlled solenoid on diesel engines equipped with computer-controlled injection.

Fig 1: General appearance of a Nozzle

FUEL INJECTION CHARACTERISTICS:

The most important injection characteristics:

- Injection pressure
- Injection duration
- Injection timing
- Injection rate history

Injection pressure:

Injection pressure is the fuel pressure just before the injector holes.It depends on the type of injection system, on engine speed and load, and on fuel properties .Diesel engines with divide combustion chambers operate with high air speed in th pre-chamber or whirl chamber and combustion chamber.

Injection timing:

Injection timing is the time span between the start of injection and the TDCof the engine piston .Injection timing has a strong influence on injection pressure, combustion process, and practically all engine emissions.

Injection duration:

Injection duration is the time span from the beginning to the end of injection. In general, injection duration should be as short as possible. At higher engine speeds, injection duration should become longer an the mean injection pressure should be reduced.

Injection rate:

Injection rate represents the quantity of fuel injected per unit of time in to combustion chamber. Variation of injection rate history influence mixture preparation, combustion process, and harmful emissions.

II. EXPERIMENTAL SETUP

EXPERIMENTAL SETUP:

A four stroke single-cylinder direct injection diesel engine system was used to perform the experiments in this work. The main specifications of the diesel engine are given in Table 1.

Bore*Stroke 80*110

Displacement (L)	1.85
Geometric compression ratio	17.3:1
Swirl ratio	1.85
Piston bowl shape	180
Intake valve open timing	337
Intake valve close timing	-171
Exhaust valve opening timing	133
Exhaust valve closing timing	-339

Table 1: Engine Specifications

Fig 2: Experimental Setup

III. RESULTS & DISCUSSIONS

CALCULATIONS:

3 HOLES (8:2)
(DIESEL&OLIVE OIL)

1) **BRAKE POWER:**
$=\frac{2\times3.14\times N\times T}{60000} KW$

=0.98KW

2) **MASS FUEL CONSUMPTION:**
$=\frac{10\times0.82\times3600}{1000\times T}$**kg/hr**

=0.49kg/hr

3) **SPECIFIC FUEL CONSUMPTION:**
$=\frac{SFC}{B.P}$ =0.5kg/ kwhr

4) **ACTUAL VOLUME OF AIR SUCKED IN TO THE CYLINDER:**
=cd$\times a\times\sqrt{2\times g\times h}\times3600$

=0.62 3.14 10^4 $\times\sqrt{2\times9.81}$
3600
=6.44m^3/hr

5) **SWEPT VOLUME:**
$=\frac{\pi d^2}{4}\times L\times\frac{N}{2}\times60$

=24.88m^3/hr

6) **VOLUMETRIC EFFICIENCY:**
$=\frac{Va}{Vs}\times100$

=25.88%

7) **BRAKE THERMAL EFFICIENCY:**
$=\frac{B.P\times3600}{M.F.C\times C.V}$

=15.28%

4 HOLES :

1) **BRAKE POWER:**
$=\frac{2\times3.14\times N\times T}{60000} kW$

=0.98kW

2) **MASS FUEL CONSUMPTION:**
$=\frac{10\times0.82\times3600}{1000\times T}$**kg/hr**

=0.51kg/hr

3) **SPECIFIC FUEL CONSUMPTION:**
$=\frac{SFC}{B.P}$

=0.5kg/kwh

4) **ACTUAL VOLUME OF AIR SUCKED IN TO THE CYLINDER:**
=cd$\times a\times\sqrt{2\times g\times h}\times3600$

=0.62 3.14
10^4 $\times\sqrt{2\times9.81\times\frac{8}{10}}$ 3600
=6.44m^3/hr

5) **SWEPT VOLUME:**
$=\frac{\pi d^2}{4}\times L\times\frac{N}{2}\times60$

=24.88m^3/hr

6) **VOLUMETRIC EFFICIENCY:**
$=\frac{Va}{Vs}\times100$

=25.72%

7) **BRAKE THERMAL EFFICIENCY:**
$=\frac{B.P\times3600}{M.F.C\times C.V}$

=15.2%

5 HOLES :

1) **BRAKE POWER:**
$=\frac{2\times3.14\times N\times T}{60000} KW$

=0.98kW

2) **MASS FUEL CONSUMPTION:**

$= \frac{10 \times 0.82 \times 3600}{1000 \times t}$ **kg/hr**

=0.49kg/hr

3) **SPECIFIC FUEL CONSUMPTION:**

$= \frac{M.F.C}{B.P}$

=0.53kg/kwhr

4) **ACTUAL VOLUME OF AIR SUCKED IN TO THE CYLINDER:**

$= 0.62 \times 3.14 \times 10^{\wedge}4 \times \sqrt{2 \times 9.81 \times \frac{6.4}{1000} \times \frac{1000}{1.9}} \times 3600$

$= 0.62 \quad 3.14 \quad 10^{\wedge}4 \quad \times \sqrt{2 \times 9.81} \; 3600$

=6.64m^3/hr

5) **SWEPT VOLUME:**

$= \frac{\pi d^2}{4} \times L \times \frac{N}{2} \times 60$

=24.88m^3/hr

6) **VOLUMETRIC EFFICIENCY:**

$= \frac{V_a}{V_s} \times 100$

=25.88%

7) **BRAKE THERMAL EFFICIENCY:**

$= \frac{B.P \times 3600}{M.F.C \times C.V}$

=14.11%

3 H0LES (9:1)
(DIESEL&OLIVE OIL)

1) BRAKE POWER:

B.P=0.98kw

2) MASS FUEL CONSUMPTION:

MFC= 0.488kg/hr

3) SPECIFIC FUEL CONSUMPTION:

SFC==0.49kg/kw hr

4) ACTUAL VOLUME OFAIR SUCKED INTO THE CYLINDER

$V_A = C_d \; A \quad 3600$

$= 0.62 \times 3.14 \times 10^{-4} \times \sqrt{2 \times 9.81 \times \frac{8.7}{1000} \times \frac{1000}{1.19}} \times 3($

$= 6.64 m^3/hr$

5) SWEPT VOLUME :

$V_s = D^2$

$= 24.88 m^3/hr$

6) VOLUMETRIC EFFICIENCY

=26.68

7) BRAKE THERMAL EFFICIENCY

Bth=16.18 %

<u>4 HOLES:</u>

1) **BRAKE POWER**

B.P= $\frac{2\pi \times 1500 \times 4 \times 0.16 \times 9.81}{60000}$

=0.98KW

2) **MASS FUEL CONSUMPTION :**

MFC= $\frac{10 \times 0.82 \times 3600}{1000 \times 62.8}$

=0.47kg/hr

3) **SPECIFIC FUEL CONSUMPTION:**

SFC= $\frac{mfc}{b.p}$ =0.47kg/kw hr

4) **ACTUAL VOLUME OF AIR SUCKED IN TO THE CYLINDER:**

$V_A = 0.62 \times 3.14 \times 10^{-4} \times \sqrt{2 \times 9.81 \times \frac{8.7}{1000} \times \frac{1000}{1.9}} \times$ 3600

$= \quad 6.64 m^3 /hr$

5) **SWEPT VOLUME:**

$V_s = D^2 = 24.88 m^3/hr$

6) **VOLUMETRIC EFFICIENCY:**

$V\% = \frac{V_a}{V_s} \times 100$

$V\% = \frac{6.64}{24.88} \times 100$

=26.68%

7) **BRAKE THERMAL EFFICIENCY**

Bth= $\frac{0.98 \times 3600}{0.47 \times 42.50}$

=15.18%

<u>5 HOLES:</u>

1. **BRAKE POWER:**

B.P= $\frac{2\pi \times 1500 \times 4 \times 0.16 \times 9.81}{60000}$

=0.98KW

2. **MASS FUEL CONSUMPTION:**

MFC= $\frac{10 \times 0.82 \times 3600}{1000 \times 62.8}$

=0.49kg/hr

3. **SPECIFIC FUEL CONSUMPTION:**

SFC= $\frac{mfc}{b.p}$

=0.5kg/kw hr

4. **ACTUAL VOLUME SUCKED IN TO THE CYLINDER:**

$V_A = 0.62 \times 3.14 \times 10^{-4} \times \sqrt{2 \times 9.81 \times \frac{8.7}{1000} \times \frac{1000}{1.9}} \times$

3600

$= 6.68 m^3 /hr$

5. **SWEPT VOLUME :**

$V_s = D^2 \qquad = 24.88 m^3/hr$

6. **VOLUMETRIC EFFICIENCY:**

$$V\% = \frac{V_a}{V_T} \times 100$$

$$V\% = \frac{6.64}{24.88} \times 100$$

$$= 26.84\%$$

7. **BRAKE THERMAL EFFICIENCY :**

$$Bth = \frac{[illegible] \times 3600}{0.49 \times 43.[illegible]}$$

$$= 15.82\%$$

By the present study by adding olive oil with diesel we calculated the performance of three different nozzles. In this comparison was made among the three different injector holes to analyze the additive and performance that includes emissions.

RESULTS FOR (8:2) DIESEL&OLIVE OIL

S. No	No.of holes	Mass fuel consumption kg/hr	Brake thermal efficiency %	Emissions nox,soot mg/l	Actual emissions mg/l
1.	3 Holes	0.49	15.82	1.15	1.94
2.	4 Holes	0.51	15.2	1.14	1.92
3.	5 Holes	0.52	14.11	1.11	1.91

RESULTS FOR (9:1) DIESEL&OLIVE OIL

S. No	No.of holes	Mass fuel consumption kg/hr	Brake thermal efficiency %	Emissions nox,soot mg/l	Actual emissions mg/l
1.	3 Holes	0.47	16.18	1.56	1.94
2.	4 Holes	0.48	15.18	1.55	1.92
3.	5 Holes	0.49	15.82	1.53	1.91

Fig 3: Emissions Result

IV. CONCULSION

In the process of determining the best number of injector holes for single cylinder 4-stroke DI diesel engine. The Experimental analysis is done with additive of olive oil and diesel. The values obtained for 3-holes, 4-holes and 5-holes by Experimental analysis are made comparison among (8:2) and (9:1) ratios of olive and diesel. Minimum NOx emissions are found for 5-holes=1.14 and the mass fuel consumption is comparatively high and for 3-holes brake thermal efficiency =16.18 for (9:1) ratio is comparatively high.

References

[1] Dec J. A conceptual model of DI diesel combustion based on laser-sheet imaging. SAE Technical Paper 970873;1997. http://dx.doi.org/10.4271/970873.
[2] Hwang W, Dec J, Sjöberg M. Fuel Stratification for low-load HCCI Combustion: performance & fuel-PLIF measurements. SAE Technical Paper 2007-01-4130; 2007. http://dx.doi.org/10.4271/2007-01-4130.
[3] Musculus Mark PB, Miles PC, Pickett LM. Conceptual models for partially premixed low-temperature diesel combustion. Prog Energy Combust Sci 2013;39(2–3):246–83. [4] Hashimoto K. Effect of ethanol on the HCCI combustion. SAE Technical Paper 2007- 01-2038; 2007. http://dx.doi.org/10.4271/2007-01-2038

[5] Koci C, Ra Y, Krieger R, Andrie M, et al. Detailed unburned hydrocarbon investigations in a highly-dilute diesel low temperature combustion regime. SAE Int J Engines 2009;2(1):858–79. http://dx.doi.org/10.4271/2009-01-0928.

[6] Saxena Samveg, Bedoya ID. Fundamental phenomena affecting low temperature combustion and HCCI engines, high load limits and strategies for extending these limits. Prog Energy Combust Sci 2013;39(5):457–88.

[7] Inagaki K, Fuyuto T, Nishikawa K, Nakakita K. et al. Dual-fuel PCI combustion controlled by in-cylinder stratification of ignitability. SAE Technical Paper 2006-01-0028; 2006. http://dx.doi.org/10.4271/2006-01-0028.

[8] Reitz RD, Duraisamy G. Review of high efficiency and clean reactivity controlled compression ignition (RCCI) combustion in internal combustion engines. Prog Energy Combust Sci 2015;46:12–71. http://dx.doi.org/10.1016/j.pecs.2014.05. 003.

[9] Chao Y, et al. Comparative study on gasoline homogeneous charge induced ignition (HCII) fuels(GDBF) combustion. Fuel 2013;106:470–7.

[10] Chao Y, et al. Comparative study on gasoline homogeneous charge induced ignition (HCII) fuels(GDBF) combustion. Fuel 2013;106:470–7.

[11] Yang B, Yao M, Cheng WK, et al. Experimental and numerical study on different