

Estimation of Life Time of Fuel Droplet Evaporation with Various in Cylinder Pressures, Compression Ratios and Exhaust Gas Recirculation

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Abstract: *The current research proposes a novel technique using simulation-analytical method to determine the lifetime of droplet evaporation in compression ignition (CI) engine. Using this technique, investigation was carried out to determine the optimum conditions for minimum droplet lifetime. The variation of droplet lifetime was studied for three parameters namely in cylinder pressure, compression ratio and exhaust gas recirculation (EGR) percentage. The best conditions for low droplet lifetime were obtained.*

Keywords: simulation, analytical, lifetime, in cylinder pressure, compression ratio, exhaust gas recirculation.

Introduction -

Ignition is the initial phase of combustion. It is characterized by rapid increase of species temperature. The study of combustion and understanding of the in cylinder process enable us to design a much more fuel efficient engine. However, the difficulty encountered in combustion study is the cost involved in experimental setup and the complexity involved in understanding the mechanism. Numerical models are developed to understand the combustion process [1]. Commercial thermodynamic tools provide another approach to investigations in combustion.

Effective combustion can occur only when the ignition process is initiated before the piston reaches TDC. Combustion process actually begins when the liquid fuel vaporizes completely to form the mixture. Once the fuel is injected into the combustion chamber through the injector, the liquid spray undergoes various physical processes and interacts dynamically with the turbulent fluid inside the combustor, followed by splitting up of the liquid fuel into droplets which collide with other droplets and coalesce with them. The high momentum of the liquid spray creates a turbulent flow field and gas entrainment. Droplet spray may impinge on the cylinder wall surfaces due to the tight confinement inside the intake manifold or cylinders and form a liquid film which will eventually evaporate. Droplets can thus be considered the basic element which provides the fuel vapor; Understanding of single-droplet evaporation and combustion processes therefore provides important guidance in design of practical burners. [2]

Droplet evaporation in the present research is considered a convective drop. The reason is that, when a liquid is injected in the combustor, there exists a relative velocity between the drop and the surrounding oxidizer (mostly air). It is surrounded by a convective boundary layer. The shape of the drop is assumed to be spherical. The laws of heat transfer and conservation of energy are then applied to this drop and analytical relationships

are developed. Diesel-RK is used for simulation. The parameters used in the analytical relationship have been acquired through simulation. The most important parameter involved for our research is the expression for the lifetime of evaporation of the drop. The lifetime is directly related to how fast a fuel droplet can vaporize. Smaller droplets evaporate faster. The faster the droplet evaporates, the more efficient is the combustion. This in turn reduces the particulate matter emissions. In this research, a method has been developed to calculate the lifetime of an ethanol fuel droplet. The method developed has then been used to analyze the lifetime of the ethanol fuel drop under various design conditions with EGR. An optimum engine design condition is then suggested. Plots are made for the life time of fuel drop versus ICP and comparison is made for different CRs and EGR percentage. Bio-Ethanol has been chosen in this research as ethanol would use as a major fuel in the future as predicted by B. Hahn-Ha Gerdal et al. [4]. The properties of bio ethanol were obtained from ref [10].

II. Methodology-

The analytical expression used to determine the life time of a fuel drop is derived based on the work of Mc Callister et al. [2] as follows. The principle of conservation of energy is applied to the convective fuel drop.

Energy stored in drop = energy convected.

This can be mathematically expressed as follows,

$$-\frac{d}{dt} \left\{ \rho_l \frac{4}{3} \pi \left(\frac{D}{2} \right)^3 \right\} h_{fg} = \pi D^2 \cdot q_s \quad (1)$$

Where,

q_s = heat flux convected from the droplet (W/m²)

h_{fg} = latent heat of vaporization of droplet (J)

D = Diameter of the fuel droplet (m)

ρ_l = Density of the fuel drop (kg/m³)

On solving further we get,

$$\frac{dD^2}{dt} = -2C_1\beta_0 - \beta \quad (2)$$

Where,

$$\beta \equiv \frac{1.6k Re_D^{1/2} Pr^{1/3} (T_a - T_b)}{\rho_l h_{fg}} \quad (3)$$

And

$$\beta_0 = \frac{4k(T_a - T_b)}{\rho_l h_{fg} C_1} \quad (4)$$

The constant C_1 is assumed for simplicity purposes to be 1/2, i.e., the thermal layer is equal to the radius of the droplet. [2]

Assuming an average Reynolds number and integrating equation 3, we have,

$$D^2 = D_0^2 - 2C_1\beta_0 t - \beta t \quad (5)$$

Solving the above differential equation,

$$t_{life} = \frac{D_0^2}{2C_1\beta_0 + \beta} \quad (6)$$

Here, t_{life} is the required expression for calculating the droplet lifetime of evaporation. [2]

Simulation:

DIESEL-RK, a full cycle thermodynamic engine simulation software was chosen here as the results obtained from Diesel-RK was in close proximity to those obtained from experimental study. This was shown by Kuleshov A.S in ref [5,6,7].

The engine considered in the current research was set to the following parameters

Table: 1 Experimental Engine Data

S.No	Parameter	Value
1	Cylinder Bore diameter	150mm
2	Piston stroke length	180mm
3	No. of cylinders	6
4	Engine Speed	1500rpm
5	No. of injector	1
6	Distance between spray centre and cylinder head	1mm
7	Engine Type	Turbocharged

Table: 2 Variable parameters

1	EGR	10%, 20%, 30%, 40%
2	CR	27,28,29
3	Nozzle diameter(mm)	0.195,0.200,0.210,0.225

Table 3: Ethanol Fuel Properties

S.No	Parameter	Value
1	Specific gravity	0.794
2	Lower calorific value MJ/kg	27.2
3	Energy density (net) @ 15°C[MJ/l]	21.6
4	Enthalpy of vapourisation @ b.p [kJ/kg]	845
5	Research octane number	111
6	Motor octane number	89
7	Cetane number	8
8	Stoichiometric air-fuel ratio[kg/kg]	8.98
9	Spontaneous ignition temp ^o C	365
10	Flammability limit in air (air/fuel)mass	2.7-18.5
11	Flammability limit	0.49-3.3
12	boiling point	78.5
13	Composition	C ₂ H ₅ OH

The ICP vs Θ , T vs Θ , injection velocity versus Θ , mass fraction of fuel vaporization vs crank angle and outlet fuel velocity versus crank angle plot were obtained from the simulation. The numerical values pertaining to the plots were obtained using Microsoft Excel 2007. The T and ICP values were noted at a Θ

of 13° before TDC. The simulation also gives the sauter diameter of droplet (d32). The plots obtained for one case is shown in Fig 1,2,3,4,5. The mixture formation and combustion simulation is also shown.

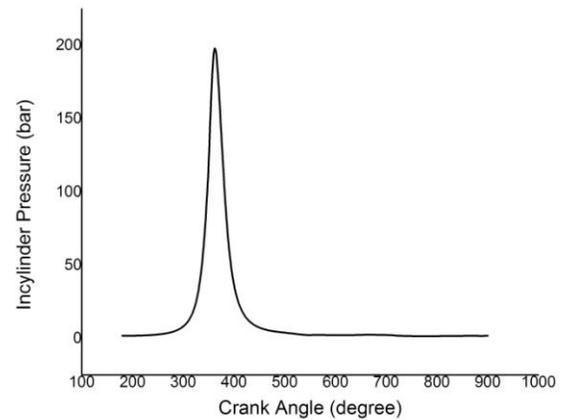


Fig 1. ICP (bar) versus Θ (degree) (CR 28 with 10%EGR)

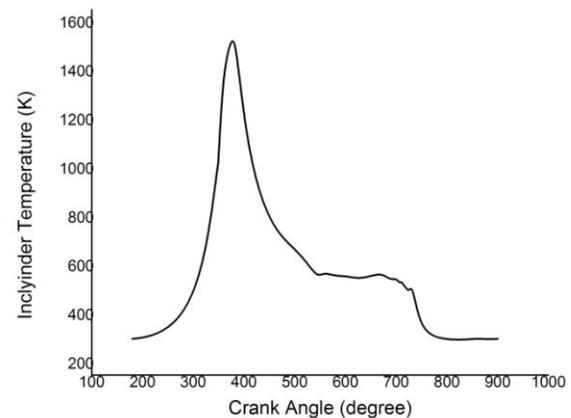


Fig 2. T (K) versus Θ (degree) (CR 28, 10%EGR)

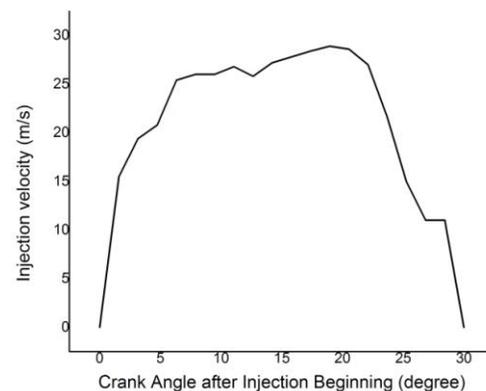


Fig 3. Injection Velocity (m/s) versus Θ (degree) (CR 28, 10%EGR)

Table :4 Combustion Simulation Data(cr 28:1)

No	Parameter	Expansion	Value
1	A/F_eq	Air Fuel Equivalence Ratio in the Cylinder	2.7070
2	F/A_eq	Fuel Air Equivalence Ratio in the Cylinder	0.36941
3	p_max	Maximum Cylinder Pressure, bar	217.84
4	T_max	Maximum Cylinder Temperature, K	1562.4
5	CA_p.max	Angle of Max. Cylinder Pressure, deg. A.TDC	2.0000
6	CA_t.max	Angle of Max. Cylinder Temperature, deg. A.TDC	15.000
7	dp/dTheta	Max. Rate of Pressure Rise, bar/deg	11.510
8	p_inj.max	Max. Injection Pres. (before nozzles), bar	1603.6
9	d_32	Sauter Mean Diameter of Drops, microns	8.9534
10	Theta_i	Injection / Ignition Timing, deg. B.TDC	13.000
11	Phi_inj	Duration of Injection, deg	26.500
12	x_e.id	Fuel Mass Fraction Evaporated during Ignit. Delay	0.05879
13	Rs_tdc	Swirl Ratio in the Combustion Chamber at TDC	0.10000
14	Rs_ivc	Swirl Ratio in the Cylinder at IVC	0.06194

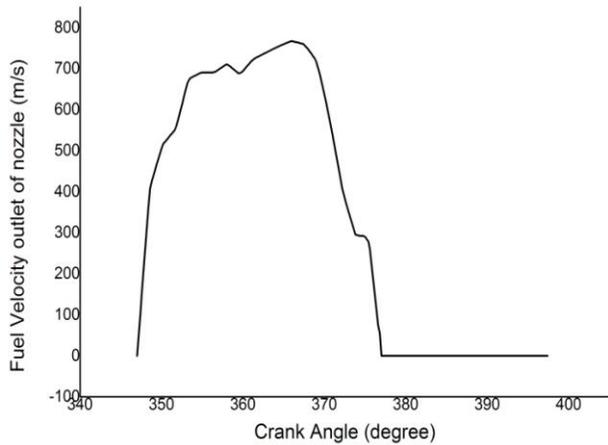


Figure :4 Variation fuel velocity outlet of Nozzle with Crank Angle (rpm 1500, cr 28: 1)

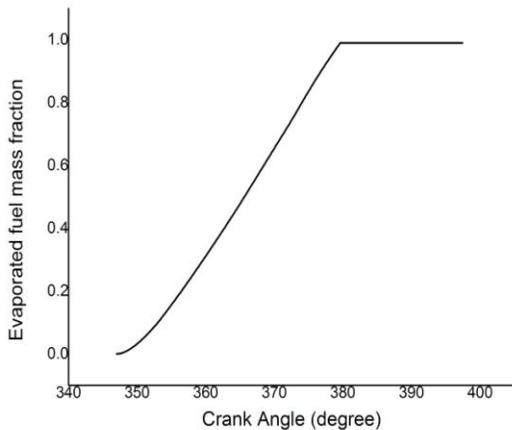


Figure :5 Variation of Evaporated fuel mass fraction with Crank Angle (rpm 1500, cr 28: 1)

Estimating the life time of drop:

The parameters found using the simulation method are then substituted in the following relationships. Average Temperature (T_{avg}): The ambient temperature is assumed to be 303K. The average temperature is hence determined.

$$T_{avg} = \frac{T + T_{ambient}}{2}$$

[12]Average thermal conductivity of air is given by

$$k = (1.5207e^{-11})T^3 - (4.8574e^{-08})T^2 + (1.0184e^{-04})T - 3.9333e^{-04}$$

The viscosity of air is given by [8]

$$\mu = \mu_0 \left(\frac{T}{T_0} \right)^{3/2} \frac{T_0 + S}{T + S}$$

Where, $\mu_0 = (1.716) 10^{-5} \frac{kg}{m-s}$, $T_0 = 273.1K$, $S = 110.56 K$.

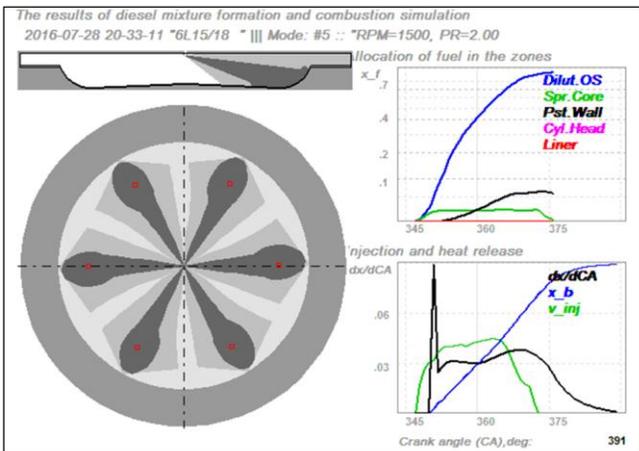


FIG: 6 Mixture formation and Combustion Simulation - Compression ratio 28:1

Air density is calculated by [9] $\rho = \frac{ICP}{R \cdot T}$ Where, ρ is the density(kg/m³), ICP is the In cylinder Pressure (Pa), R (J/Kg-K) is the gas constant and T is the In cylinder temperature(K).

Prandtl Number is given by [3] $Pr = \frac{\mu \cdot c_p}{k}$

Specific heat capacity [11] given by:

$$C_p = (1.9327 e^{-10})T^4 - (7.9999 e^{-07})T^3 + (1.1407 e^{-03})T^2 - (4.4890 e^{-01})T + 1.0575 e^{03}$$

III. Results and Discussion-

The droplet life time corresponding to the various input parameters are tabulated. Graphs have been plotted to show the variation of the droplet lifetime with ICP for various CRs and EGR percentage.

Table: 5 Droplet Life Time Variation with EGR 10%

CR 27		CR 28		CR 29	
ICP (bar)	τ_{life} (s) ($\times 10^{-4}$)	ICP (bar)	τ_{life} (s) ($\times 10^{-4}$)	ICP (bar)	τ_{life} (s) ($\times 10^{-4}$)
94.74	714	98.15	686	101.37	662
94.92	506	98.25	487	101.43	470
94.96	345	98.28	332	101.46	320
94.97	317	98.37	305	101.48	295

Table: 6 Droplet Life Time Variation with EGR 20%

CR 27		CR 28		CR 29	
ICP (bar)	τ_{life} (s) ($\times 10^{-4}$)	ICP (bar)	τ_{life} (s) ($\times 10^{-4}$)	ICP (bar)	τ_{life} (s) ($\times 10^{-4}$)
96.53	693	99.78	668	103.09	644
96.55	492	99.84	474	103.11	458
96.57	335	99.86	323	103.19	312
96.6	308	99.93	297	103.23	287

Table: 7 Droplet Life Time Variation with EGR 30%

CR 27		CR 28		CR 29	
ICP (bar)	τ_{life} (s) ($\times 10^{-4}$)	ICP (bar)	τ_{life} (s) ($\times 10^{-4}$)	ICP (bar)	τ_{life} (s) ($\times 10^{-4}$)
93.19	694	96.19	669	99.62	645
93.28	492	96.52	475	99.67	458
93.3	335	96.54	323	99.68	312
93.31	308	96.57	297	99.71	286

Table: 8 Droplet Life Time Variation with EGR 40%

CR 27		CR 28		CR 29	
ICP (bar)	τ_{life} (s) ($\times 10^{-4}$)	ICP (bar)	τ_{life} (s) ($\times 10^{-4}$)	ICP (bar)	τ_{life} (s) ($\times 10^{-4}$)
89.08	700	92.24	675	95.27	651
89.09	497	92.26	479	95.27	462
89.14	338	92.26	326	95.275	315
89.19	311	92.27	300	95.28	290

The above values are plotted in a graph. They are shown as follows:

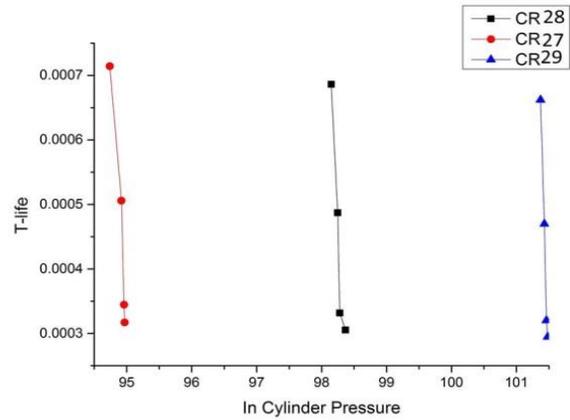


Fig 7 : Droplet Lifetime Vs. ICP (EGR 10%)

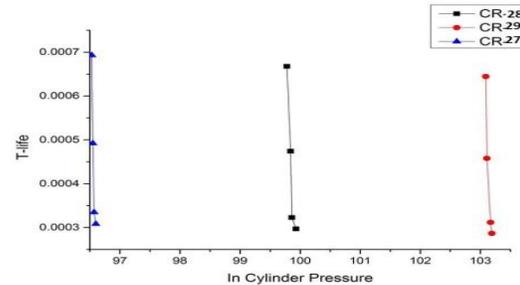


Fig :8 Droplet Lifetime Vs ICP (EGR 20%)

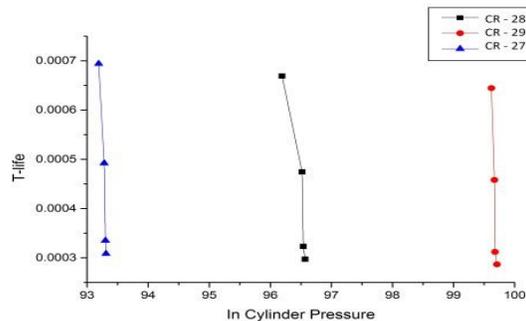


Fig:9 Droplet Lifetime Vs ICP (EGR 30%)

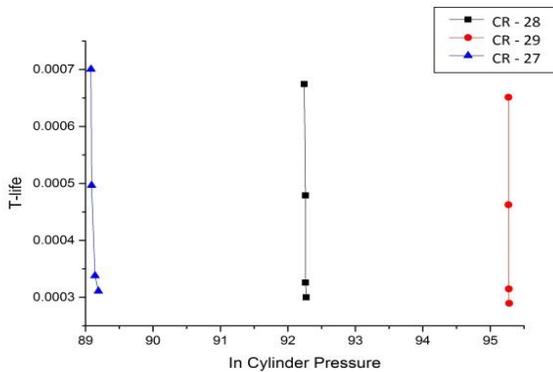


Fig: 10 Droplet Lifetime Vs ICP (EGR 40%)

It can be seen from the graphs that the droplet lifetime decreases with compression ratio and in cylinder pressure. From Table (5) for an EGR of 10% for a compression ratio of 27, the value of droplet lifetime has decreased from 714×10^{-4} to 317×10^{-4} s. This indicates a drop of about 55.6% in the droplet lifetime. Now keeping the ICP constant and observing the change in compression ratio, it is observed that the droplet life time drops by about 6.9% from a CR 27 to CR 29. For an EGR of 20%, the droplet life time has dropped by 55.43% by increasing ICP for CR29 and a 6.81% drop by increasing the CR from 27 to 29 at constant ICP. For an EGR of 30%, there is a decrease of 55.503% in droplet lifetime for CR 29 and decreasing ICP, and when ICP is constant, there is a drop of 0.681% by increasing CR from 27 to 29. With 30% EGR, there is a drop of 55.45% in droplet lifetime for CR29 and increasing ICP, while there is a 6.752% drop when ICP is fixed but CR is increased from 27 to 29. Hence, it can be inferred that as the compression ratio increases and as the in cylinder pressure which indirectly refers to fuel injection pressure increases, the droplet life time decreases. This is because, with higher injection pressure, the fuel is injected at a higher velocity as the pressure energy at the injector nozzle is converted to kinetic energy. Now, when the fuel is injected at such high pressures, the fuel spray will split into droplets of very small diameters; a smaller drop will naturally evaporate faster as shown in fig 7- 16. A low evaporation time indicates a better mixing of the fuel with the oxidizer in the combustion chamber. This results in a higher proportion of the fuel being oxidized giving a better combustion efficiency. Also, this indicates a lower emission of particulate matter.

IV. Conclusion:

The method developed to determine the droplet lifetime was developed from fundamental principles [3] and also validates the claim that combustion characteristics improve by increasing compression ratio [13] and fuel injection pressure [14]. Also, a CI engine with EGR 30% and CR29 and nozzle injector diameter 0.222 mm gives the lowest droplet lifetime of vaporization.

Nomenclature:

T	In Cylinder Temperature
ICP	In Cylinder Pressure
EGR	Exhaust Gas Recirculation
CR	Compression Ratio
T-Life	Droplet Lifetime
θ	Crank Angle

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