Numerical analysis of effect of fuel injection timings on performance characteristics of direct injection diesel engine using isobutane as fuel

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Abstract – *This paper investigates the effect of isobutane fuels on the combustion, and emission reduction characteristics in a diesel engine. In diesel engine, fuel is injected into the engine cylinder near the end of the compression stroke. Injection timing is the time at which injection of fuel into the combustion chamber begins. It is usually expressed in crank angle degrees (CAD) relative to TDC of the compression stroke. An advanced CFD simulation (STAR-CD) has been carried out in order to study the effect of change in start of fuel injection timing i.e. (12^0 BTDC, 9^0 BTDC, 6^0 BTDC, 3^0 BTDC, AT TDC, 3^0 ATDC). An improved version of the ECFM combustion model has been applied coupled with advanced models for NOx and soot formation.*

***Key Words*: Diesel Engine, Injection Timing, Isobutane, NOx, STAR-CD**

1. INTRODUCTION

One day, our sources for traditional fuels including petroleum would be depleted. Owing to the fact that these fuels are typically not renewable, a lot of people are worried that a day would come when the demand for these fuels would be more than the supply, triggering a considerable world crisis. The NOx is produced at a great extent, due to the high local temperatures found in Diesel engines which are highly dependent on the initial rise of heat release. In addition, soot production and oxidation are both dependent on the mixing rate and local flame temperatures [1]. It is well known that injection strategies including the injection timing and pressure play the most important role in determining engine performance, especially in pollutant emissions. [2] Among the different possibilities in a fuel injection system, such as level and control of injection pressure, injection rate, and timing control, rate shaping mechanisms which produce boot- type injections have been recognized as the most effective ways to reduce NOx without a huge detriment in PM and fuel consumption [3]. In recent years, computer codes have been used for simulating three-dimensional combustion in internal combustion engines [4]. This paper studies the numerical effects of fuel injection timing on performance and pollutants in the four-cylinder direct injection diesel engine using fuel as isobutane.

2.1 GOVERNING EQUATION

Governing equations including continuity, momentum and energy are modified based on Reynolds average and according to Reynolds-averaged Navier-Stokes (RANS) equations based on semi-implicit method for pressure-linked equations (SIMPLE) algorithm and k-ε standard turbulence model for numerical simulation of flow, inside the combustion chamber is used.

2.2 COMPUTATIONAL PROCEDURE

The modelling for CFD simulation start drawing the piston bowl geometry. Re-Entrant piston bowl geometry was done using SolidWorks 2010. A sector of 72^0 was drawn. The dimensions of the geometry [5] is given in the fig: 1.The geometry is meshed by using ES_ICE (Expert Systems in IC Engine) prosurf software. Meshed geometry is shown in fig: 2. The meshing of the in-cylinder fluid domain is done using ES_ICE grid generation tool. 72^0 sector is taken for the analysis due to the symmetrical location of the five-hole injector at the centre of the combustion chamber. It is important to study the in-cylinder fluid dynamics during the latter part of combustion and initial part of expansion strokes in DI diesel engines. Analysis is carried out from 40^0 before TDC (BTDC) to 80^0after TDC (ATDC), as fuel injection, combustion and pollutant formations are taken place during this period.

The computational Grid when the piston is at Top Dead Centre (TDC) is shown in fig: 3.A 3D sector mesh is modelled in ES_ICE to produce volume of combustion in the cylinder. Computational grids at 100 degree before TDC is given in fig: 4. After this, the sector grid is used as a part of STAR-CONTROLS. ECFM-3Z (Extended Coherent Flame Model-3Z) combustion model is used to characterize ignition and combustion. Also the analysis continued for applying initial conditions, boundary conditions like beginning temperature, initial pressure and cylinder crown temperature, wall temperature and so on. A fully automatic process is used to minimize mesh generation time and allow the user easy control over mesh size.

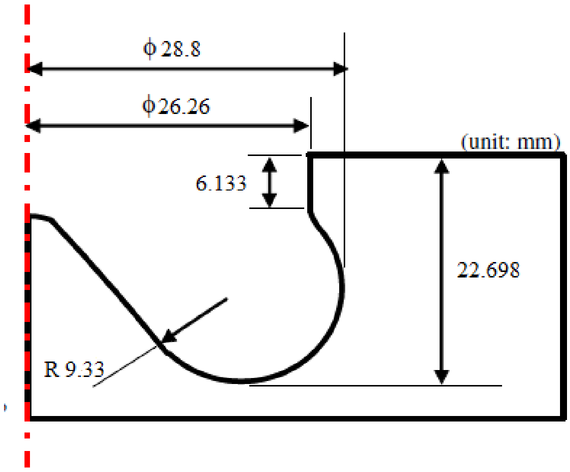

Fig.1 Dimensions of the Re-Entrant geometry

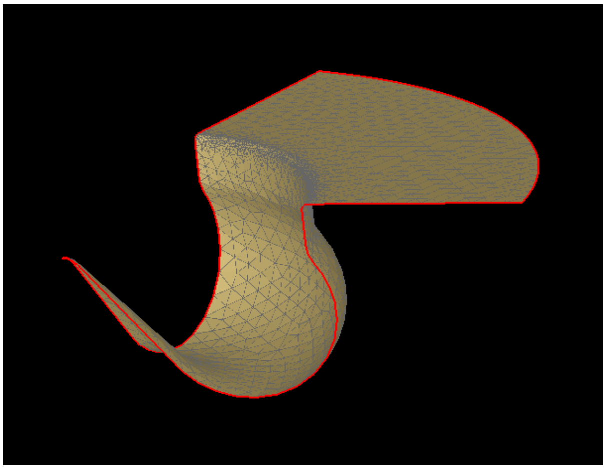

Fig.2 Surface geometry meshed in prosurf

3. ENGINE PARAMETERS & DETAILS

The diesel engine used for the model validation is a single-cylinder version of MT4.44 agricultural engine[6]

Table -1: Engine Specifications

Engine	MT4.44 Agricultural Engine
Bore	100 mm
Stroke	127 mm
Bowl Type	Re-Entrant
Compression Ratio	17.5 : 1
Connecting Rod Length	240 mm
Squish Clearance	4.5 mm
Injection Duration	13^0
Injection Hole Diameter	0.000176 m
Number of Holes	5
Injection Pressure	650 bar
Rate of Injection	0.006736 kg/sec
Engine speed	2000 rpm

4. RESULTS AND DISCUSSIONS

4.1 VALIDATION

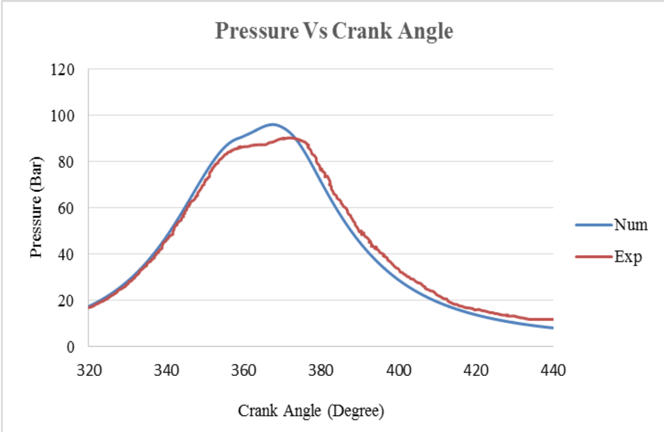

Fig. 3.Validation between numerical and experimental graphs

A comparison between experiments and simulation is presented, in order to assess the accuracy of the subsequent predictions. The injection pressure is 275 bar and the injection timing at 3^0 ATDC. The results shows the numerical value pressure has a marginal increase over experimental value. The peak pressure discrepancies between experimental and computation are 7.09% and the trend predicted by the model is reasonably close to experimental results, although there are still some differences.

4.2 IN-CYLINDER PRESSURE

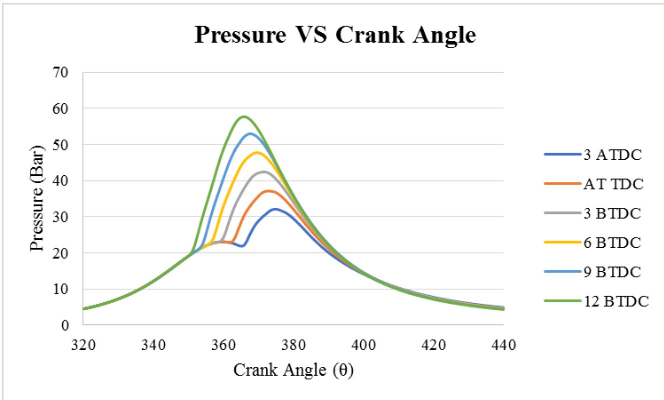

Fig.4 Cylinder pressure vs crank angle of isobutane at different injection timing.

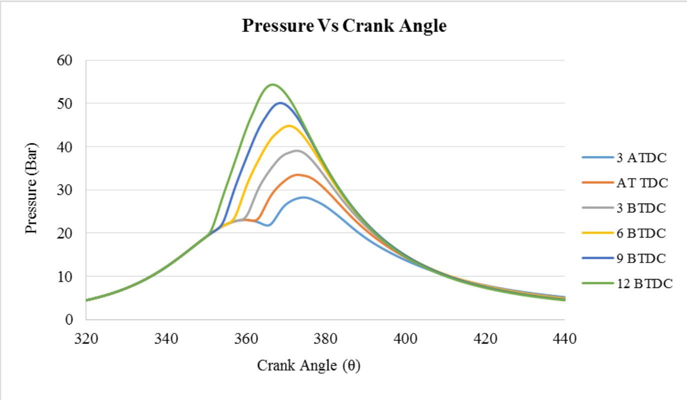

Fig. 5 Cylinder pressure vs Crank angle of diesel at different injection timing.

Fig.4 shows variation of in-cylinder pressure of isobutane and maximum pressure is obtained in 12^0 BTDC and it is 57 bar Pa and 3^0 BTDC get better pressure for combustion and it is 42 bar. In cylinder pressure increases with decreaseing injection timing.Fig.5 show diesel in-cylinder pressure and it is less than isobutane. The maximum pressure obtain in diesel is 54 bar. For a fixed total equivalence ratio, advancing the injection timing means that the combustion occurs earlier in the cycle, and this can increase the peak cylinder pressure because more fuel is burned before TDC, and the peak pressure moves closer to TDC, where the cylinder volume is smaller. Retarding the injection timing decreases the peak cylinder pressure because more of the fuel burns after TDC. So the in cylinder pressure for isobutane is higher.

4.3 IN-CYLINDER TEMPERATURE

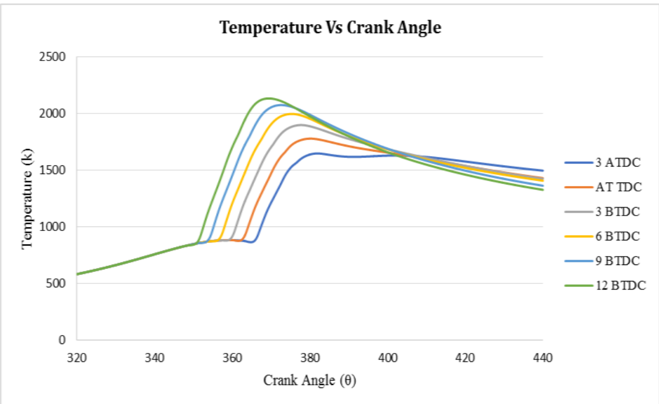

Fig.6 Temperature vs Crank angle of isobutene at different injection timing

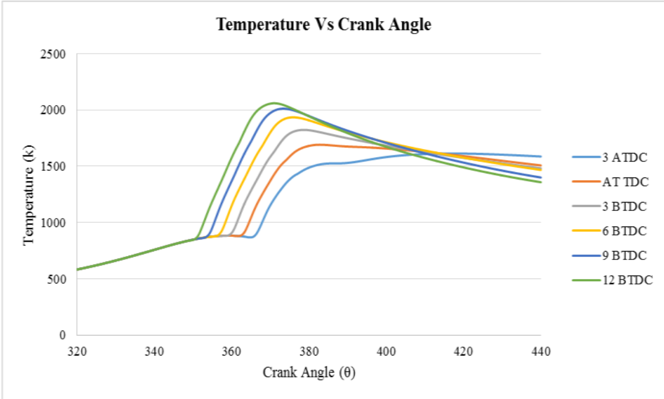

Fig.7 Temperature vs Crank angle of diesel at different injection timing

In the Fig.6 temperature of different injection timing of isobutane is shown. The maximum temperature obtain is at -12^0 BTDC is 2230 k because of maximum pressure at this point. The temperature also increase with decreasing injection timing. And Fig.7 show the temperature of diesel at different injection timing. As the start of fuel injection timing is advanced by 3^0CA from 3^0BTDC (original start of injection timing) 12^0BTDC and 3^0ATDC and AT TDC. The peak in-cylinder temperature increases and retarding the injection timing by 3^0CA from 12^0BTDC to 3^0BTDC and AT TDC and 3^0ATDC decreases peak in-cylinder temperature. Higher peak cylinder pressures result in higher peak charge temperatures and vice-versa. The maximum in cylinder pressure at 12^0BTDC for diesel is 2060 k.

4.4 HEAT RELEASE RATE

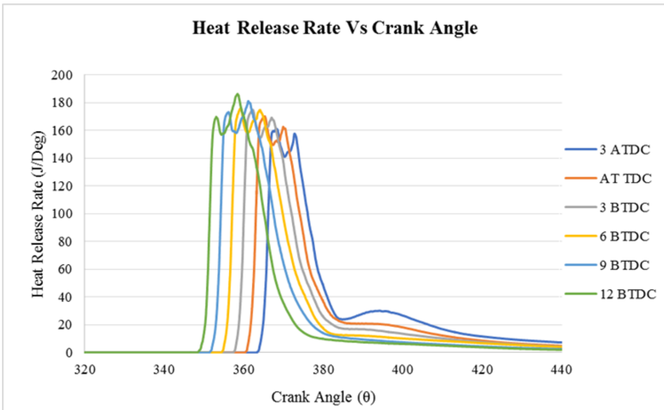

Fig.8 Heat release rate vs Crank angle of isobutene at different injection timing.

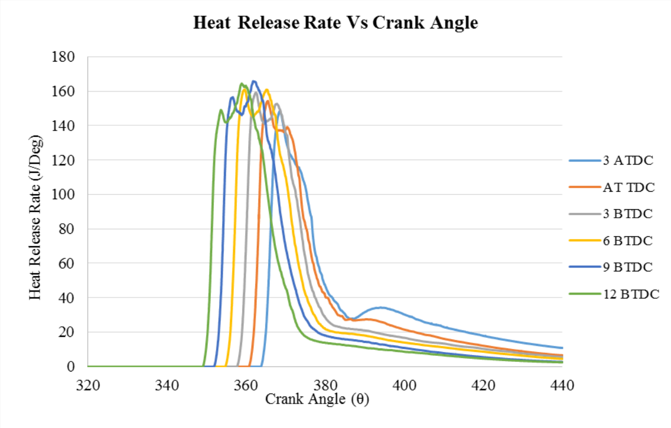

Fig.9 Heat release rate vs Crank angle of diesel at different injection timing.

Fig.8 and Fig.9 shows heat release rate variations versus crank angle at different start of injection timings of isobutane and diesel. As the start of fuel injection timing is advanced by 3^0CA from 3^0BTDC (original start of injection timing) to 12^0BTDC and 3^0BTDC peak heat release rate increases and retarding the injection timing by 3^0CA from 12^0BTDC to 9^0BTDC and 6^0TDC to 3^0ATDC decreases peak heat release rate. In the case of advanced injection timing, a large amount of evaporated fuel is accumulated resulting in longer ignition delay. The longer ignition delay leads to rapid burning rate and the pressure and temperature inside the cylinder rises suddenly. Hence, most of the fuel burns in premixed mode causes maximum peak heat release rate and shorter combustion duration. In the case of retarded injection timing, the accumulation of evaporated fuel is relatively less resulting in shorter ignition delay. The shorter ignition delay leads to slow burning rate and slow rise in pressure and temperature. Hence, most of the fuel burns in diffusion mode rather than premixed mode resulting in lower peak heat release rate and longer combustion duration.

5. CONCLUSION

- In first part of this study, an advanced CFD simulation was performed to demonstrate the effects of start of fuel injection timing on a DI diesel engine using isobutane as fuel and the results were compared with that of diesel.
- It is observed that on advancing the injection timing from 12^0 BTDC to 3^0 ATDC pressure, temperature increased.
- Heat release rate is higher in isobutane because of the low cetane number.
- Compare with diesel, isobutane has more advantage.
- The best injection timing for the good performance and less emission is at 3^0 BTDC.

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