Numerical investigation of emission reduction techniques applied on methanol blended diesel engine

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Abstract The present investigation works out a two-stage strategy to achieve higher level of emission reduction to meet more stringent emission norms. In the first stage, an optimum blend of diesel methanol fuel has been determined using numerical simulation to give maximum possible NOx and soot reduction. In the next stage, numerical simulation has been performed by three different methods of emission reduction namely through variation of swirl ratio, variation in quantity of recirculation of exhaust gases in Exhaust Gas Recirculation (EGR) technique and finally by means of adding water in various proportions to the same optimum diesel methanol blended fuel to obtain further reduction of emission. The numerical simulation has been performed on a single cylinder Kirloskar diesel engine (model TV1) using commercially available CFD software AVL FIRE. Simulation starting with the optimum diesel–methanol blend as the base fuel, effects of swirl ratio; 1.0, 1.3, 1.6 and 2, percentage EGR varied between 10% and 20% and addition of water to the base fuel in the ratio of 5%, 10% and 15% by volume on emission are analyzed. Results indicate that water blend method tends to reduce NOx emission by 95% and soot by 14% with respect to emissions of base fuel.

1. Introduction

Growing awareness of the ill effects of global warming has forced governments across the globe to formulate and enforce stringent emission legislation to cut greenhouse-emission significantly. Thus, paving a way for extensive research focusses on emission reduction. Diesel engines which are widely used in surface transport due to their high thermal efficiency are also considered as principal source of environmental pollution. Carbon-monoxide, hydrocarbon, particulate matter (soot) and nitric oxide are primarily identified as major constituents of the emission emerging from diesel engines. Out of these pollutants, NOx and soot emission are considered as most hazardous to the mankind, therefore, receiving greater attention worldwide. Moreover, it is also a well established fact that it is a formidable task to reduce both NOx and soot emission simultaneously. Several technologies, have so far been developed to reduce engine emission and tend to increase soot.
while reducing NOx emission and vice-versa. Methanol, potentially known to reduce NOx and soot emission [1–5], is used as an alternative fuel for diesel vehicles due to its economic and environmental friendly nature. Methanol has a high latent heat of vaporization (1178 kJ/kg), high oxygen content (49.93%), low carbon content (37.50%), sulfur free and high burning speed [6], whereas, ethanol has low heat of vaporization (840 kJ/kg), low oxygen content (34.73%) and higher carbon content (52.2%) than methanol [7].

Methanol is typically injected into the combustion chamber either by blending with diesel or through fumigation. In diesel–methanol blend, an additive should be used to maintain stability of the mixture because of the poor miscibility of diesel–methanol blend [8,9]. On the other hand, a low pressure fuel injector is used to inject methanol in the combustion chamber while diesel is injected through the high pressure injector in the fumigation method [10–12]. Cheng et al. [13] carried out an experimental analysis by using both blending and the fumigation method to add methanol to the diesel–biodiesel blend. In both the methods, 10% methanol used in diesel biodiesel blend and results indicated that CO and HC emission increased in fumigation method as compared to that obtained with diesel, whereas diesel–methanol blend method showed almost similar results as diesel fuel. It was also concluded that diesel–biodiesel blend could reduce soot emission effectively as compared to diesel alone, while methanol addition, can reduce emission to even higher level. Further, it was concluded that the blend method was more efficient in terms of emission reduction than fumigation method.

Preparation of Blend is well known method to reduce emissions from I C engines [14–16]. The blend of biodiesel with urea injection [17] and another blend with non petroleum fuel at different injection timings [18] promise to reduce emissions up to certain mark. In added, emissions can reduce by experiments with the fuel and physical parameters of the engine. An et al. [19] carried out numerical modeling of diesel–biodiesel blends with methanol (5–15%) with varying load condition; it was found that CO and soot emission decreased, whereas NOx emission remained constant at full load but increased at partial load. Ciniviz et al. [20] used diesel–methanol blend from 0% to 15% by volume and results indicated that, NO emission increased, whereas CO and HC emission decreased with respect to pure diesel. All the components of emission first increased and then decreased when engine speed varied from 800 to 2800 rpm. Yao et al. [21] carried out experimental analysis with air–methanol mixture ignited by diesel fuel and concluded that diesel with an air–methanol mixture decreased NO and soot emissions while HC and CO emission increased with varying load and speed. Recently, Li et al. [22] investigated the effect of methanol addition in a reactivity controlled compression ignition engine (RCCI) using numerical simulation and concluded that at moderate methanol addition and advanced start of injection HC and soot emission decreased but no significant reduction was observed in case of NOx and CO emission. Qi et al. [23] carried out experimental analysis on diesel–biodiesel fueled engine with methanol as an additive. It was reported that NO, HC and CO remained unchanged for all blend ratios and speed. Cheung et al. [24] reported that methanol–biodiesel blend with methanol could reduce soot and NOx emission simultaneously.

Zhu et al. [25] compared the emission from a methanol–biodiesel blend with that of ethanol–biodiesel blend. The methanol blended fuel resulted in less NOx and soot emission as compared to the ethanol blends; however, HC and CO emission was found in higher proportions.

It is evident from the above discussion that emission-reduction capability of methanol can be used to explore the possibility of higher reduction especially in NOx and soot emission. Recently, attempts such as increasing fuel injection pressure [26] and advanced injection strategies such as high pressure multiple injections [27] have been employed to reduce both soot and NOx simultaneously. Pierpont et al. [28] have used a combination of EGR and multiple injections to reduce both soot and NOx. The multiple-injection techniques can so effectively control the production of soot and NOx during combustion process due to retardation of injection timing which reduces NOx yet containing soot at low level. Such techniques, however, may not be cost effective as well as difficult to implement. In the present investigation the issue of higher reduction of both NOx and soot emission is explored by employing much simpler, cost-effective techniques, by controlling initial swirl and exhaust gas recirculation and by introducing water in the combustion chamber in the form of blend. Present investigation will be carried out using numerical simulation with AVL FIRE on a Kirloskar (TV1 model) single cylinder diesel engine using diesel–methanol blend as base fuel.

Extensive research is in progress using initial swirl to reduce emissions from diesel engines. Solaimuthu et al. [35] deduced that in-cylinder air flow in diesel engines is generally characterized by swirl, squish and turbulence which had strong influence on air–fuel mixing and burning rate. The higher swirl can be generated by improving design of the inlet port, thus, improving the burning process. Wei et al. [29] performed numerical simulation and concluded that small swirl in a reentrant type of combustion chamber led to reduction in NOx and soot mass fraction. Jafarmadar and Khanbarazadeh [30] stated that the reentrant piston geometry with high depth reduces formation of NOx and soot due to the effect of intense swirl and tumble. Prasad et al. [31] simulated different geometries to investigate the effect of swirl at varying injection timings and found that the NOx and soot mass fraction reduced up to 27% and 85% respectively at an injection timing of 8.6° CA BTDC. Further, Gafoor and Gupta [32] examined the effect of combined initial and piston generated swirl and concluded that soot and CO decreased and NOX increased. Swirl can, therefore, be used for emission-reduction.

NOx emission has also been reduced by means of EGR technique either independently or in conjunction with other emission-reduction technologies. EGR tends to reduce oxygen concentration and flame temperature leading to reduction in NOx emissions. However, long term use of EGR technique increases engine wear as well as soot emission due to high carbon deposits and degradation of lubrication oil [33,34]. Solaimuthu et al. [35] studied the effect of selective catalytic reduction (SCR) and hot/cold EGR technique on a single cylinder diesel engine fueled with diesel–biodiesel blends and result showed that SCR technique is more effective to reduce NO emission compared to other techniques of emission reduction used with biodiesel. Abdelaal and Hegab
carried out experiments on a dual fuel single cylinder diesel engine by employing variable amount of EGR, fueled with diesel and natural gas. Comparative study showed reduction in NO emission with EGR when compared to other mode of operations, whereas less reduction in HC and CO emission by using EGR but their values are still higher than those of conventional diesel fuel. Park and Bae [38] considered the outcome of low pressure EGR and high pressure EGR on a passenger car diesel engine and find out that NO emission gets reduced as the portion of low pressure EGR increased due to lower intake temperature. Saravanan [39] investigated the effect of EGR and advanced injection timings on combustion characteristics of a diesel locomotive and reported that the delay period, peak pressure and combustion duration increased as the EGR introduced at advanced injection timing. Bhaskar et al. [40] studied the effect of EGR on diesel and fish oil methyl ester blends and concluded that the EGR technique could reduce NO emission up to a considerable level with 20% diesel–fish oil methyl ester blend. In general, EGR method directly affects the flame temperature of the combustion process, which serves to reduce NO emission.

It is well known that water when blended with diesel in the form of blend leads to reduction of both NOx and soot emission. This may be attributed to the micro-explosion of water droplets in the presence of elevated temperatures leading to internal gasification of water vapor, subsequently, causing temperature reduction and concentration increase of OH radicals in the fuel-rich region of the flame, which tend to reduce the formation of NOx and soot emission [41]. Literature is replete with numerous analyses on diesel engines using water–diesel blended. Subramanian [42] and Zhang et al. [43] reported significant reduction in NOx and smoke emissions. Lif and Holmberg [44] reported reduction of NOx emissions by 30% and PM by 60% using diesel–water blend with water contents varying from 5% to 45%. Chiosa et al. [45] reported improvement in spray optimization due to the micro-explosion of water droplets which resulted in faster combustion and reduction in the formation of emission. Bedford et al. [46] and Wagner et al. [47] performed CFD analysis on DI diesel engine and reported reduction in NOx emission. Although water blend has been used extensively with pure diesel for emission reduction, it can also be added to the diesel–methanol blend; this will perhaps control formation of NOx and soot emissions more effectively owing to the presence of OH radicals in the methanol blended fuel.

In the present paper, two-stage simulation is carried out. In the first stage, simulation is performed to determine the optimum blend from diesel–methanol blend in terms of emission reduction. Three different combinations of diesel–methanol are taken with varying proportion of methanol added to diesel. Simulation is performed to predict emission characteristics of these combinations and the combination that gives minimum level of emission for NOx and soot is taken as optimum fuel for the second stage simulation. In the next stage, three methods of emission reduction, namely by changing swirl ratio, percentage EGR and water blend method; have been numerically simulated to achieve greater degree of emission reduction from that obtained with optimum blend.

### 2. Numerical simulation

#### 2.1. Model description

In the present research, numerical simulation is performed on a single cylinder, DI diesel engine having a hemispherical bowl shaped piston. The specifications of the single cylinder diesel engine are given in Table 1 and initial and boundary conditions are listed in Table 2. Moreover, for the entire set of simulation, compression ratio, bowl volume, mass injected and speed of the engine remain constant. To reduce computational effort, only one-third segment of the piston geometry is used and a mesh is created by using AVL FIRE ESE diesel module with a hexahedral element; 40,868 cells are taken as shown in Fig. 1. To verify whether the mesh chosen is independent of grid size, a grid independency test is conducted; three different

<table>
<thead>
<tr>
<th>Table 1</th>
<th>Engine specifications.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Make</td>
<td>Kirloskar engine, TV-1</td>
</tr>
<tr>
<td>Number of cylinders</td>
<td>1</td>
</tr>
<tr>
<td>Bore × stroke</td>
<td>87.5 mm × 110 mm</td>
</tr>
<tr>
<td>Swept volume</td>
<td>661 cc</td>
</tr>
<tr>
<td>No. of nozzle holes</td>
<td>3</td>
</tr>
<tr>
<td>Connecting rod length</td>
<td>234 mm</td>
</tr>
<tr>
<td>Rated output</td>
<td>5.2 kW</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>17.5</td>
</tr>
<tr>
<td>Type of combustion chamber</td>
<td>Hemispherical open</td>
</tr>
<tr>
<td>(Piston bowl shape)</td>
<td>combustion chamber</td>
</tr>
<tr>
<td>Rated speed</td>
<td>1500 rpm</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 2</th>
<th>Initial/boundary conditions.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial pressure</td>
<td>0.65 MPa</td>
</tr>
<tr>
<td>Initial temperature</td>
<td>300 k</td>
</tr>
<tr>
<td>Piston temperature</td>
<td>550</td>
</tr>
<tr>
<td>Liner temperature</td>
<td>425</td>
</tr>
<tr>
<td>Head temperature</td>
<td>475</td>
</tr>
<tr>
<td>Fuel injection timing</td>
<td>23° CA BTDC</td>
</tr>
<tr>
<td>Fuel spray angle</td>
<td>120°</td>
</tr>
<tr>
<td>Injection type</td>
<td>Single injection</td>
</tr>
<tr>
<td>Mass of fuel injected</td>
<td>1.6e-05 kg</td>
</tr>
</tbody>
</table>

![Figure 1](image-url) Computational grid at TDC.
In the present work, commercial CFD software AVL FIRE is used, which is specially developed for IC engine applications. This software is based on a finite volume approach by applying the Cartesian coordinate system. This software is validated by Tatschl et al. [48]. Simulation performed by using several meshes are treated for test with 29,516, 40,868 and 50,208 cells as shown in Table 3 and it can also be seen that in Fig. 2, predictions are well within acceptable limits and simulation time is approximately 6 h. Thus, the simulation is carried out with the mesh of 40,868 cells for further simulation (see Table 3).

### 2.2. Applied models and validation

In the present work, commercial CFD software AVL FIRE is used, which is specially developed for IC engine applications. This software is based on a finite volume approach by applying the Cartesian coordinate system. This software is validated by Tatschl et al. [48]. Simulation performed by using several improved physical and chemistry models, is listed in Table 4 [48–57]. The k-ω turbulence model used in the present study was developed by Hanjalic et al. [49], in 2004. This model is used to improve numerical stability by solving transport equation. A coherent flamelet model is used for simulation of both premixed and non-premixed. This model is based on the laminar flamelet concept. Fig. 4 indicates that, flamlet models are based on the layer separation concept. It assumes that the chemical reaction taking place inside the fragile layer splits the unburned gas from fully burned gas. In the present study, the 1850 gas phase reaction, 186 species and 100 heterogeneous reactions with the participation of micro-heterogeneous particles of different types [53]. In this study, both Dukowicz model [57] and multi-component evaporation model are used subject to type of fuel involved in the combustion process. Extended coherent flamlet model – three zone (ECFM-3Z) is used to couple combustion – spray module to describe direct injection combustion phenomena [50–52]. The ECFM-3Z model can also be used for multi-component fuel [52]. The wave breakup model is used for spray modeling, which depends upon the physical and dynamic parameters of the injected fuel and the domain fuel; however, it depends mainly on the wavelength of the speed of the droplets. This model is used for diesel fuel spray simulation [53]. Spray wall interaction model is used for accounting the effect of non-atomized or non-evaporated fuel particles striking the wall of the combustion chamber. A spray wall interaction model known as ‘wall-jet’ is used [54]. In the case of walljet1, the droplets get a rebound or slide over the wall by the formation of vapor cushion under the droplets. The extended zeldovich mechanism is used to calculate the NO emissions and this mechanism considers the effect of hydrocarbon radicals, nitrogen and oxygen on NO formation. NO formation of nitrogen, oxygen and hydrocarbon radicals very much depends on combustion temperature [34,55]. This model can be coupled with ECFM-3Z combustion model based on equilibrium approach. The kinetic soot model is used in the present simulation as it can be used for different fuel classes to describe the behavior of soot formation and oxidation [56]. The kinetic soot model can solve the 1850 gas phase reaction, 186 species and 100 heterogeneous reactions with the participation of micro-heterogeneous particles of different types [53]. In this study, both Dukowicz model [57] and multi-component evaporation model are used subject to type of fuel involved in the combustion process.

![Figure 2](gridindependencytest.png)

**Figure 2** Grid independency test.

### Table 3 Grid independency test.

<table>
<thead>
<tr>
<th>No of cells at TDC in the domain</th>
<th>Maximum pressure (bar)</th>
<th>% Deviation</th>
</tr>
</thead>
<tbody>
<tr>
<td>29,516</td>
<td>44.46</td>
<td>–</td>
</tr>
<tr>
<td>40,868</td>
<td>43.24</td>
<td>2.75</td>
</tr>
<tr>
<td>50,208</td>
<td>42.99</td>
<td>0.58</td>
</tr>
</tbody>
</table>

### Table 4 Models in use [48–57].

<table>
<thead>
<tr>
<th>Turbulence model</th>
<th>K-zeta-f model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Combustion model</td>
<td>ECFM-3Z</td>
</tr>
<tr>
<td>Spray model</td>
<td>Wave breakup model</td>
</tr>
<tr>
<td>Wall interaction model</td>
<td>Walljet1</td>
</tr>
<tr>
<td>Evaporation model (for diesel only)</td>
<td>Dukowicz model</td>
</tr>
<tr>
<td>Evaporation model (for dual fuel)</td>
<td>Multi component evaporation model</td>
</tr>
<tr>
<td>No emission model</td>
<td>Extended zeldovich model</td>
</tr>
<tr>
<td>Soot emission model</td>
<td>Kennedy model</td>
</tr>
</tbody>
</table>

### Table 5 Details of operating conditions of emission-reduction techniques.

<table>
<thead>
<tr>
<th>Operating conditions</th>
<th>Swirl ratio</th>
<th>Water blends (% by volume)</th>
<th>EGR fraction</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1</td>
<td>5%</td>
<td>10%</td>
</tr>
<tr>
<td></td>
<td>1.3</td>
<td>10%</td>
<td>20%</td>
</tr>
<tr>
<td></td>
<td>1.6</td>
<td>15%</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Figure 3  Comparison of experimental and simulated results of (a) pressure of diesel at partial load, (b) HRR of diesel at partial load, (c) pressure of diesel (70%) and methanol (30%) at full load, (d) HRR of diesel (70%) and methanol (30%) at full load and (e) NO mass fraction of all diesel methanol blends.
3. Result and discussion

To obtain an optimum fuel, diesel was blended with methanol in three different proportions: D + M10, D + M20, and D + M30 and simulation was carried out. The base fuel will then be chosen based on the predicted emission characteristics of three blends of diesel–methanol. Once the diesel–methanol blend which gives optimum values of emission is chosen, it can be used as the base fuel for next stage of investigation. In this stage, simulation will be conducted using three techniques of emission–reduction (see Table 5). The first method in which swirl-ratio is varied as 1, 1.3, 1.6, and 2 has the engine speed constant at 1500 rpm. In EGR method, the percentage of exhaust gas recirculated in the engine is taken as 0%, 10%, and 20%. Finally, reduction of emission is attempted by means of water introduced in the combustion chamber blended with the base fuel, diesel–methanol blend, in the ratio of 5%, 10%, and 15% by volume.

3.1. Influence of diesel–methanol blends on emissions

Methanol possesses some dominating properties over conventional diesel fuel in view of emissions reduction. Higher latent heat of vaporization of methanol reduces temperature of the combustion chamber. Methanol takes heat from the combustion chamber to evaporate and then combines with air. Additionally, Methanol is a higher oxygenated biodiesel, which participates with extra amount of oxygen in combustion reaction of methanol–diesel blend. At a low total equivalence ratio, i.e. below the stoichiometric ratio (lean mixture), high heat capacity of methanol reducing combustion temperatures leads to reduction of NO formation [57]. Fig. 5 shows total equivalence ratio increasing with increasing proportion of methanol in successive blends, indicating that all the blends are lean mixtures. With a lean mixture, combustion temperature decreases due to the low flame temperature of methanol. Formation of NO directly depends on combustion temperature. Thus, there is reduction in NO formation due to low flame temperatures. Fig. 6 shows NO mass fraction plotted for diesel and various blends of diesel–methanol. The NO mass fraction appears to be monotonically decreasing with increasing methanol percentage in the mixture, therefore, preserving the expected trend. Maximum reduction of NO emission is reported with 30% diesel–methanol blend, which is 65% of diesel fuel.

Fig. 7 shows increase in soot emission due to continuous reduction in combustion temperature at different blend ratios. Low combustion temperature results in high soot formation because of incomplete combustion, thus, leaving unburned carbon particles. These unburned carbon particles are moving through exhaust gas and remain in the same state in the environment.

Fig. 8 shows reduction of CO emission with increase in methanol in the diesel–methanol fuel; hence, a reduction of CO by 68% as compare to diesel fuel alone is achieved.
Significant reduction of CO emission can be explained by the fact that methanol being the highly oxygenated fuel providing sufficient oxygen for complete combustion whereas the rich fuel zones inside the combustion chamber lead CO formation due to deficiency of oxygen molecules in the burning process. Unburned hydrocarbons (HC) are caused by incomplete combustion of air fuel mixture besides that other sources are engine lubricants and cylinder geometry. From fig. 8 it can be noted that, HC emission reduced as methanol percentage increases in diesel-methanol blends. The likely reason for HC reduction is the reduction in the quantity of hydrogen and carbon in methanol. According to Ciniviz et al. [20], quantity of hydrogen and carbon is 7 times and 14 times less than diesel fuel respectively, which contribute reduction in HC emission with increasing methanol in successive blends. In addition to that, methanol can provide larger quantity of oxygen required for complete combustion, consequently, leading to reduction in HC emission.

3.2. Influence of initial swirl on emissions

3.2.1. Influence of swirl ratio on air/fuel equivalence ratio ($\lambda$)

Fig. 9 shows the relation between air/fuel equivalence ratios with different crank angle at varying swirl ratios. It shows that spray distribution of SR1.3 is more uniform inside the cylinder. It can be observed that, when the piston is at TDC, mixing zone of fuel and air is diminished; however, some amount of fuel rich zones can be discovered in the near piston wall. As the crank angle increased from TDC to 5° ATDC, accordingly equivalence ratio shows obvious changes and equivalence ratio covers more piston boundary as swirl increased. There is formation of fuel rich zones in most of the cases of swirl ratio, as the piston moves to 10° ATDC. It can also be observed that, the equivalence ratio concentration zone is gradually shrinking toward the bottom of the piston and covers the less circumferential area at 10° ATDC, at the same time SR1.3 covers a more circumferential area than others. The fuel rich zone of SR1.3 is reduced more rapidly at 20° ATDC than for other swirl ratios. The value of equivalence ratio is reduced from 6.5 to 5.4 for SR1.3; whereas the equivalence ratio of SR1.6 and SR2 is

<table>
<thead>
<tr>
<th>SR0</th>
<th>SR1</th>
<th>SR1.3</th>
<th>SR1.6</th>
<th>SR2</th>
</tr>
</thead>
<tbody>
<tr>
<td>At</td>
<td>TD</td>
<td>C</td>
<td>5° ATDC</td>
<td>10° ATDC</td>
</tr>
</tbody>
</table>

Figure 9 Comparison of air–fuel equivalence ratio ($\lambda$) at different swirl ratio.
reduced to 5.8 and 6.4 respectively. This trend is same as the trend observed by Wei et al. [29] and they stated that equivalence ratio increases with increasing swirl ratio.

3.2.2. Influence of swirl ratio on turbulent kinetic energy

Fig. 10 shows that, air motion is affected by squish and piston movement during upward movement of the piston whereas the maximum value of TKE is reducing. The high TKE zones are formed here with peak values of 120 m²/s² and 113 m²/s² at 5° BTDC and 9° BTDC respectively. It can be understood that the high TKE zone is brought down and lies toward the chamber wall as the swirl ratio increases from 0 to 2. After TDC, high TKE zones are mainly influenced by induced swirl and reverse squish, and during this period fuel atomization and combustion process take place. It can be clearly observed that there are two TKE centers appeared for SR ≤ 1.3 whereas for SR > 1.6 only one TKE center can be seen at 10° ATDC. It can be deduced that low swirl ratio is favorable for better air-fuel mixing as also corroborated by Wei et al. [29]. The area of high TKE zone is decreasing as the swirl ratio increases. In addition to this, increasing swirl ratio minimizes the possibility of fuel rich zones inside the combustion chamber. However, the high swirl ratio increases air motion, and consecutively the front spray bending angle is leading toward downward direction. Hence, this spray is separated out with the adjacent spray and mixture will lead to incomplete combustion [29].

3.2.3. Influence of swirl ratio on emissions

Fig. 11 shows the NO mass fraction at different swirl ratios; the variation of NO mass fraction is found to be quite scattered. However if NO mass fraction at SR0 and SR1 is ignored, the mass fraction appears to be increasing gradually for SR1.3 to SR2. It can be seen that, the soot mass fraction increasing rapidly up to SR1.3 in soot mass fraction slows down with further increase in the swirl ratio. At SR1.3, NO and soot mass fraction has the best trade off as compared to their values at other swirl ratios, when the result was recorded at 80°ATDC. Qualitative trend of NO and soot emission as observed in Fig. 11 is similar to the trend seen in Wei et al. [29]. Fig. 12 shows continuous increase in CO and HC mass fraction and the peak is observed at SR2. The peak value of
local fuel rich zones of SR2 is higher than that of other swirl ratios. The probable reason for the increase of CO and HC in the fuel rich zones is due to incompleteness of combustion.

### 3.3. Influence of exhaust gas recirculation (EGR) on emissions

The addition of EGR technique to diesel engines has significant impact on emission reduction, which has been proved by many researchers [30–35]. The air fuel mixture is non-premixed just before the compression stroke, while the engine is operating with conventional diesel fuel in direct injection diesel engines. The peak pressure and temperature are higher with diesel fuel due to the shorter delay period before the combustion process. Unlike non premixed combustion, pre mixed combustion with EGR method is operating at the leaner side of the air–fuel ratio. Lean mixture is causing a negative effect on combustion efficiency, which results in low cylinder pressure. In addition to that, the whole combustion process is shifted toward expansion stroke due to longer delay period. As a result, reduction of tip pressure and increasing volume can be observed while piston is moving from TDC to BDC during the expansion stroke. Incidentally, this is also corroborated in Abdelaal and Hegab [36]. Further, they concluded that the mixing of EGR with intake air leads to conversion of O₂ to CO₂ and H₂O, which in turn suppresses the combustion process and consequently reduces the peak pressure and temperature. Fig. 13 shows reduction of NO formation as percentage of EGR increased. Dual fuel mode has a longer ignition delay which reduces peak pressure, whereas, conversion of O₂ to CO₂ and H₂O reduces peak temperature which leads to lower NO formation [36]. Fig. 13 shows a reducing trend of soot emission. The addition of EGR at a rate of 0–10% reduces soot because of reduction in combustion temperature. A similar effect was also observed by Lujan et al. [37], using EGR with 4% and 8% rate. Further, they observed increasing soot emission while using EGR with 8% and 14%. However, it could take place due to merging of low cylinder temperature and decreased oxygen concentration. In the same way, soot emission trend is noted in Fig. 14 by using the increased EGR rate from 10% to 20%. Introduction of EGR dilutes the intake air, therefore, causing reduction of flame temperature, and incomplete combustion and instability inside cylinder [22]. The other possible reason for incomplete combustion may be due to the poor fuel utilization in case of dual fuel mode [36]. Fig. 14 shows high CO formation due to low oxygen concentration at lean mixture at high loads [36]. Poor combustion quality resulted in high CO emission as EGR increases.

Fig. 14 shows the HC emissions at different EGR rates. It can be seen that HC emissions are continuously increasing with increase in EGR rates from 0% to 20%. High HC emissions are observed at high EGR rates just because of the longer combustion duration and low combustion temperature [37].

### 3.4. Influence of water addition on emissions

Methanol–water injection is widely used in high performance automotive applications. Methanol–water injection increases...
power and decreases combustion temperature in diesel engines. Reaction of methanol with water produces carbon dioxide and hydrogen at high temperature as shown in Eqs. (1):

\[
\text{CH}_3\text{OH} + \text{H}_2\text{O} \iff \text{CO}_2 + 3\text{H}_2 \tag{1}
\]

The formation of carbon-dioxide leads to complete combustion and reduces CO emissions. One of the prime advantages of above reaction is the hydrogen formation and its effect is very well explained by many researchers by using diesel hydrogen blends. Diesel and water combination has significant effect on performance and emissions of CI engines. Combustion characteristics of diesel–water blend depend on the difference in boiling points of both the constituents. The boiling point of any constituent is directly related to the evaporation rate of the molecules. Evaporation of water particles is resulted in dispersion of tiny droplets inside the cylinder which is known as micro explosion. Micro explosion is also delineated as the secondary atomization which leads to fast evaporation and improved air fuel mixing. Besides that, micro combustion is also regulated by the heat utilization of water molecules to change into steam. Hence, reduced cylinder heat resulted in low combustion temperature. Fig. 15 shows decrease in NO mass fraction as the water quantity increases from 0% to 15%. This is primarily due to the low combustion temperature inside the chamber. Water particles vaporize inside the cylinder by taking the heat of combustion and convert to finer droplets i.e. atomization, which leads to better air fuel mixing; subsequently CO and HC mass fraction is also decreasing as shown in Fig. 16. Further, the soot mass fraction is also found to decrease. This may be attributed to adequate air fuel mixing leading to complete combustion. The same trend was also observed by many researchers in recent studies [41–47].

3.5. Influence of optimum blends on performance parameters

Fig. 17 shows an increasing trend of BSFC as the quantity of methanol added to diesel fuel and it is further treated with water. For instance, the value of BSFC for Diesel, D+M30 and D+M30+W15 is 242, 315 and 547 g/kWh respectively. The results showed that the addition of methanol and water increases BSFC, which is due to LHV of the blend. LHV of the blend continuously decreases, as methanol and water quantity increase in the blend. Therefore, more amount of the fuel is injected into the cylinder. The maximum brake thermal
efficiency and engine power are shown in Figs. 18 and 19 respectively. The results are recorded with diesel fuel and optimum blends. BTE shows the adaptability of the combustion system at varying blends, whereas, the reduction in engine power with blends may be due to variation in physical properties of the blends such as lower calorific value and density.

Table 6 shows the combination of prime cases from every method and their comparison with an optimum blend (D+M30). All emissions of water blend methods are reduced, which trend is not obtained from any other method. Then, finally, it can be concluded that water blend method has tendency to reduce emission up to minimum levels from diesel methanol blends.

### 4. Conclusion

The present work focused on investigating the further possible reduction of emissions from optimum fuel of diesel–methanol blend by applying swirl, EGR and water blend methods. This strategy is supposed to be unique and one of the possible ways of implementation is through numerical simulation. Numerical investigation is less time consuming, cost effective and no manufacturing risk involved with it. In the detailed simulation of further treatment of optimum blend, swirl ratio is changed from 1 to 2, EGR is varied from 10% to 20% and optimum fuel is blended with 5–15% water by volume. The predictions are promising to reduce emissions up to lowest value. The principal findings from this investigation are as follows:

1. As the percentage of methanol increases in diesel from 10% to 30%, significant reduction has achieved 65%, 68% and 56% in NO, CO and HC emission respectively with respect to diesel alone. Therefore, D+M30 blend may be considered as optimum blend in terms of emission reduction. Soot mass fraction may be noted to be an exception at this stage.
2. The water addition method reduces NO emission up to 95%, which is lower than initial swirl method (2.5%) and EGR method (36%). This much amount of reduction in NO emission (95%) is achieved by using 15% water with D+M30 blend.
3. The Soot mass fraction is reduced up to 14% in water addition method, whereas it increased up to 118% by using initial swirl method. It can be seen from Table 6 that, EGR method (20% EGR) gives same result of soot formation as D+M30 blend.
4. The introduction of EGR method (20% EGR) increased CO emission up to 6%. However, initial swirl and water addition method decrease CO emission. Comparatively, lowest reduction of CO emission (29%) from D+M30 blend is achieved by water addition method only.
5. Further, in case of HC emission, 36% reduction is achieved by water addition method. It seems to be a single method to reduce HC emissions from D+M30 blend, because other methods increase HC emissions in the environment.

### References


[57] T. Powell, Methanol and ethanol as a major fuel substitute, in: SAE Paper NO-750124.