Influence of nozzle hole diameter on combustion and emission characteristics of diesel engine under pilot injection mode

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Abstract. Present work was divided in two different parts. In first part, an experimental study was carried out to analyze the effect of dwell on diesel engine combustion and emission characteristics. Experimental data showed that shorter dwell results in improved specific fuel consumption (SFC) but with increased smoke emissions penalty. To overcome the disadvantage of higher smoke emissions associated with shorter dwell, an effort was made to reduce smoke emissions under pilot injection mode by changing nozzle hole diameter (NHD) while keeping rest of the injection parameters unchanged. In second part, this was done numerically by using commercial Converge 3-D computational fluid dynamics (CFD) tool. In addition to baseline NHD (0.120 mm), two other NHDs i.e. 0.08 mm and 0.160 mm were considered. Numerical results showed that for given injection parameters, lower NHD resulted in faster evaporation of injected fuel as well as greater homogeneity of the mixture prepared due to better air entrainment inside fuel spray and hence led to overall faster combustion process. This faster combustion process subsequently reduced the time available for spray interactions. Therefore, in spite of getting further shortening of dwell in case of 0.08 mm NHD compared to other two cases, soot emissions were reduced by 87.2% when NHD was reduced from 0.160 mm to 0.08 mm. However, nitric oxide (NO) emission was found to increase by around 43% when NHD was reduced from 0.160 mm to 0.08 mm.

1. Introduction
Use of diesel engines is extensively increasing in heavy duty sector due to their larger power output and low carbon dioxide (CO₂) emissions. However, diesel engines suffer from greater soot and nitrogen oxides (NOx) emissions [1]. There is an urgent necessity of reducing these emissions from diesel engines due to implementation of more and more stringent emission norms. Achieving these stringent emission norms while maintaining higher fuel economy; a prime demand of customers especially in developing countries like India has become a difficult task for diesel engine manufacturers.

Modern commercial diesel engines use pilot injection mode in order to control the formation of NOX emissions inside engine cylinder. Dwell between pilot and main injection is one of the most significant parameter which greatly affects combustion process and hence emissions. Shorter dwell results in improved specific fuel consumption (SFC) as main fuel gets smoothly linked with pilot combusted fuel [2]. A study conducted by Huang et al. [3] reported that shorter dwell can reduce energy loss in diesel engines in terms of friction and also leads to improvement in thermal efficiency. SFC improvement with shorter dwell was also claimed by several other studies [4-7]. Shorter dwell also resulted in lesser NOX emissions due to shorter delay period and subsequent lesser intense pre-mixed combustion of main fuel [8]. Unfortunately, soot emissions were increased with shorter dwell due to longer and more intense diffusion combustion of main fuel [9]. In diesel engines, mostly both pilot as well as main injections are performed by a single fuel injector. So, fuel is injected in the same
region of combustion chamber by both these injections, especially in diesel engine without intake swirl. Therefore, liquid components of main fuel interact with still burning pilot flame which reduces the mixing time of main fuel with air and combustion takes place under locally rich conditions leading to augmented soot formation. Shorter dwell between two injections further aggravates this phenomenon and further augments the soot formation [10]. From the flame luminosity images, Cung et al. [11] confirmed that spray-combustion interactions were more aggressive with shorter dwell time compared to longer dwell time, which were highly responsible for accelerating soot formation. Increased smoke/soot emissions with shorter dwell were claimed by several other studies too [2, 5, 7, 12-14]

From above literature survey, it can be found that shorter dwell offers the advantage of improved fuel consumption but at the same time, shorter dwell affects soot/smoke emissions negatively. Few earlier studies suggested some solutions to reduce smoke/soot emissions under pilot injection mode. Binde et al. [10] suggested to use separate fuel injector for injecting pilot and main injections so that both these injections get spatially separated and soot formation due to spray interactions can be significantly reduced. Sahoo et al. [15] recommended to reduce pilot quantity for reducing soot formation under pilot injection mode. However, too small pilot quantity produces over-lean mixture which becomes difficult to ignite [16]. Park et al. [17] suggested to use higher fuel injection pressure to reduce smoke emissions under pilot injection mode. However, upper values of injection pressure are governed by the limitations of injection system. Another method is to increase swirl rate (SR), which was proposed by Li et al. [18], in which authors reported simultaneous reduction of soot KL factor (a measure of soot particle concentration, where K is absorption co-efficient which is proportional to soot volume fraction and L is the path length of measurement) and SFC. However, generation of swirling intake flow in diesel engine involves complex designing and higher cost of the intake system (intake ports, deflector etc.). In this work, an effort has been made to reduce smoke emissions under pilot injection mode by changing the nozzle hole diameter (NHD) of the fuel injector. Some earlier studies have investigated effect of NHD on diesel combustion process. Vijay kumar et al. [19] investigated effect of variation of NHD on diesel engine combustion characteristics using pure diesel as well as biodiesel fuel (B20). Authors reported that better fuel atomization with smaller NHD improved air-fuel mixing for both the test fuels which led to improvement in SFC and smoke emissions compared to greater NHD case. Experimental results obtained by Bergstrand and Denbratt [20] showed that smaller NHD significantly reduces the delay period. Moreover, improved air-fuel mixing with smaller NHD also resulted in shorter combustion duration which reduced heat loss and improved thermal efficiency along with reduction in soot and hydrocarbon (HC) emissions. Pang et al. [21] numerically investigated effect of NHD using Eulerian Stochastic Field (ESF) model. Results of the study demonstrated that fuel spray with larger NHD produces fuel-richer pre-mixed core region which augmented soot formation compared to smaller NHD case. Zhang et al. [22] investigated effect of NHD on flame lift-off length and soot formation process. Authors reported reduction in soot formation with lower NHD. Optical images confirmed that reduction in soot formation was mainly attributed to increased lift-off length in case of lower NHD. From above literature survey, it can be found that most of the earlier studies have investigated effect of NHD on diesel combustion characteristics under conventional single injection mode and studies related to effect of NHD on combustion characteristics of diesel engine under pilot injection mode are scarce. Based on literature study related to effect of NHD on diesel combustion and emission characteristics, it can be realized that smaller NHD results in improved air-fuel mixing and reduces soot emissions and at the same time, it also results in longer injection duration due to reduced fuel flow rate because of smaller flow area [23]. So, for a given pilot quantity and given pilot as well as main injection timings, smaller NHD results in shorter dwell due to longer injection duration of pilot fuel. Therefore, use of fuel injector with smaller NHD should result in greater smoke emissions due to more aggressive spray interactions. However, it is also possible that smaller NHD may lead to lower smoke emissions due to improved fuel-air mixing under pilot injection mode. Hence, it would be interesting to analyse the effect of NHD on diesel combustion characteristics under pilot injection
mode. Moreover, it is also noticed from literature studied that when effect of NHD is analysed on combustion characteristics of diesel engine, changes of spray angle and spray cone angle are inevitable which are not taken into account by most of the earlier studies. However, spray angle as well as spray cone angle have significant impact on overall combustion process and subsequent engine out emissions [24]. Hence, when effect of NHD is analysed on diesel combustion and emission characteristics, it represents combined effects of NHD, spray cone angle and spray angle. Therefore, spray angle and spray cone angle needs to be maintained constant with changing NHD in order to investigate the sole effect of NHD on diesel combustion characteristics. Considering these backdrops, present work is aimed to investigate the sole effect of NHD on combustion and emission characteristics of diesel engine under pilot injection mode. Initially, an experimental study was conducted in order to explore the effect of dwell on combustion, performance and emissions characteristics of the test engine. Then after, effect of NHD on combustion and emission characteristics of the test engine was numerically investigated using Converge 3-D computational fluid dynamics (CFD) tool. Base nozzle has NHD of 0.120 mm. Two other NHDs under consideration are 0.08 mm and 0.160 mm respectively. Variation of NHD was conducted while maintaining spray angle and spray cone angle constant. Moreover, total mass of injected fuel per cycle, pilot as well as main injection timings, distribution of fuel in pilot and main injection, co-efficient of discharge (C_d) and injection pressure were also maintained constant with changing NHD.

2. Experimental setup and operating conditions

All the experiments for present work were conducted on Kirlosker TV1 single cylinder diesel engine. Table 1 represents the basic engine specifications. A pressure transducer (HSM111A22, PCB) was used to measure the in-cylinder pressure. Combustion pressure was measured for 50 consecutive cycles at 1° crank angle (CA) interval and average value of these 50 cycles was used as pressure data. “ICEEngineSoft”; which is a Lab-view based software, was used to conduct performance and combustion analysis. Software calculates indicated mean effective pressure (IMEP), brake mean effective pressure (BMEP), heat release rate (HRR), brake power (BP) etc. from the obtained pressure data. An electronic fuel injection system was used in order to have flexibility in terms of variation of injection timing, number of injections per cycle and distribution of fuel in each injection. A much more detailed description of test engine, injection system, emission measurement devices, fuel properties and uncertainty of measuring instruments as well as performance parameters can be found in authors’ earlier work [25].

| Table 1. Test engine specifications. |
|--------------------------------------|
| Compression ratio                   | 17.5 |
| Displacement (cc)                    | 661.45 |
| Stroke (mm) × Bore (mm)              | 110 × 87.5 |
| Length of connecting rod (mm)        | 234 |
| Max. power (kW@rpm)                  | 5.2@1500 |

2.1. Experimental operating conditions

In present work, the test engine was operated at rated speed (1500 rpm) and at 4.6 bar BMEP (75% load) for all the test runs. Main injection timing was fixed at -20° after top dead centre (ATDC) throughout the study. This is because retardation of main injection timing beyond -20° ATDC was resulting in reduction of IMEP (indicating power loss) as shown in Figure 1 which cannot be afforded. Moreover, this is the injection timing which resulted in least brake specific fuel consumption (BSFC) as indicated in Figure 1. Based on this combination, main injection timing was selected as -20° ATDC. Start of pilot injection (SOPI) was changed to vary the dwell between two injections. Injection pressure was maintained at 350 bar. The reason why injection pressure was kept limited to 350 bar only is discussed in Section 3.2. Figure 2 shows a representative line diagram of injection profile under pilot injection mode. As the aim of present work is to lower the smoke emissions under pilot
injection mode, pilot quantity was fixed at 40% of total injected fuel per cycle. This is because at given pilot and main injection timings, soot emissions increase with increasing pilot quantity due to more severe spray interactions [10]. Dwell was varied from 8° to 16° in steps of 4° which are represented as run nos. 1, 2 and 3 respectively in Table 2.

![Figure 1](image1.png)

**Figure 1.** Effect of variation of injection timing on BSFC and IMEP.

![Figure 2](image2.png)

**Figure 2.** Representative line diagram of injection profile under pilot injection mode.

| Run No. | Pilot quantity (%) | SOPI (° ATDC) | Main injection timing (° ATDC) | Dwell (° CA) |
|---------|--------------------|---------------|-------------------------------|--------------|
| 1       | 40                 | -35           | -20                           | 8            |
| 2       | 40                 | -39           | -20                           | 12           |
| 3       | 40                 | -43           | -20                           | 16           |

3. Experimental results and discussion

3.1. Effect of dwell on combustion characteristics

In present subsection, effect of dwell on two combustion characteristics i.e. net heat release rate (NHRR) and in-cylinder pressure was analyzed.

Effect of dwell on NHRR traces has been portrayed in Figure 3. Almost all the pilot fuel gets combusted in pre-mixed mode only which led to significant rise in HRR as inferred from Figure 3. This is because temperature and pressure inside engine cylinder are not sufficient to promote auto-ignition of injected pilot fuel at the time of SOPI. Hence, injected pilot fuel passed through a long delay period and burnt instantaneously. Moreover, larger the dwell, higher the pre-mixed peak of HRR of pilot fuel as can be noticed from Figure 3. This is because SOPI is advanced in order to attain larger dwell (refer Table 2). So, in case of larger dwell, pilot fuel is injected in lower ambient in-cylinder conditions at the end of injection. Due to this, delay period of injected pilot fuel increased the further and subsequent higher pre-mixed peak of HRR for pilot fuel was obtained. Lower and earlier pre-mixed peak of main fuel with shorter dwell can be observed from Figure 3. This is because high temperature zone produced by combustion of pilot fuel shifts nearer to main injected fuel and
promotes auto-ignition of main fuel more effectively in case of shorter dwell as compared to longer dwell. Therefore, main fuel passed through a shorter delay and subsequent lower and earlier premixed peak of main fuel is attained. This shorter delay period of main injected fuel with smaller dwell resulted in poor mixing of main fuel with air and larger amount of main fuel got combusted in diffusion mode. This is the reason for getting more heightened as well as earlier diffusion combustion peak of main injected fuel with shorter dwell. Additionally, diffusion combustion duration of main fuel also increases with shorter dwell as expected.

![Figure 3. Effect of dwell on NHRR traces.](image)

The variation of in-cylinder pressure traces with varying dwell is illustrated in Figure 4. Shorter the dwell, higher the peak pressure as noticed from Figure 4. Moreover, peak pressure continues to shift near top dead center (TDC) as dwell is reduced. This is because of greater impact of pilot combustion on main fuel combustion with shorter dwell which caused faster auto-ignition of main fuel. Increased pressure before TDC is representative of greater negative work, while increased pressure after TDC induces positive work on piston. In case of 8° dwell, rise of positive work and negative work (calculated from pressure-volume diagram) were found to be 6.32% and 3.08% respectively compared to 16° dwell case.
3.2. Effect of dwell on emissions and BSFC

Figure 5 represents the effect of varying dwell on emissions and BSFC. Smoke emissions (represented as smoke opacity) reduce with longer dwell as shown in Figure 5. For e.g., smoke emissions reduce by
around 80% when dwell was increased from 8° to 16°. This is because severity of interactions between still burning pilot flame and liquid components of main fuel (spray interactions) reduced with longer dwell. Shorter and less severe diffusion combustion of main fuel (refer Figure 3) also supports this trend of reduced smoke emissions with longer dwell. Smoke emissions approached quite near to zero (~0% opacity) in case of 16° dwell. This is the reason why fuel injection pressure was kept limited to 350 bar in spite of much higher injection pressure offered by the injection system (up to 1200 bar). Due to the same reason, dwell was not increased beyond 16°. Nitric oxide (NO) emission was increased with increasing dwell. When dwell was increased from 8° to 16°, NO emission rose by around 11%. This is because of larger amount of heat released in pre-mixed combustion of pilot fuel as discussed earlier. A much reported NO-smoke trade-off can be noticed from Figure 5. HC emissions were increased with longer dwell as represented in Figure 5. This is mainly due to advanced SOPI associated with longer dwell which caused greater wall wetting as well as loss of fuel in crevice region and squish regions [26]. BSFC was reduced by 7.4% when dwell was reduced from 12° to 8°. This is due to greater impact of pilot combustion on main fuel combustion event which accelerated the combustion of main fuel.

4. Numerical simulation results and discussion

As can be noticed from experimental results, shorter dwell offers the advantage of reduced fuel consumption, but results in greater smoke emissions due to more severe spray interactions. Severity of these interactions can be reduced if dwell can be increased for a given pilot quantity as well as for given pilot and main injection timings. This is done by increasing NHD in present work. Larger NHD results in greater fuel flow rate (per °CA) and subsequent lesser fuel injection duration. Therefore, for a given pilot quantity as well as for given pilot and main injection timings, if fuel injection is performed with an injector having larger NHD, it results in shorter pilot as well as main injection duration and therefore dwell period increases compared to an injector with smaller NHD. This larger dwell might help to achieve lesser smoke emissions for a given pilot quantity as well as for given pilot and main injection timings.

With this background, effect of NHD on combustion and emission characteristics of diesel engine under pilot injection mode was numerically investigated in present work using commercial Converge 3-D CFD tool.

A 60° sector (1/6th sector of engine geometry), as shown in Figure 6 was created for the test engine as the fuel injector used for present work contains 6 nozzle holes as well as due to symmetric combustion chamber. Pilot injection mode with 12° dwell (run 2 in Table 2) was validated against respective experimental results. Closed cycle simulations were performed for the selected pilot injection mode with 1.4 mm base grid size. Grids were further refined in the area surrounding spray, piston and cylinder head by using Fixed Embedding technique available in Converge with embedding scale of 3. Adaptive mesh refinement (AMR) technique; a unique feature offered by Converge, was used to automatically refine the grids based on gradients of velocity and temperature. Maximum number of AMR cells was restricted to 10 lacs. Experimentally obtained combustion pressure and temperature at the start of simulation (CA of intake valve closing) were given as inputs. Basic models used in present work to simulate various combustion phenomena and emissions are listed in Table 3. A much more detailed description of different models as well as chemical mechanism used for simulation can be found in work carried out by Hiren et al. [25].

![Figure 6. A 60° sector for hemispherical combustion chamber.](image-url)
Table 3: Details of models used for simulating combustion process and emissions.

| Process/Emission       | Simulation model                                      |
|------------------------|-------------------------------------------------------|
| Combustion process     | SAGE detailed chemistry model [27]                   |
| Spray break-up         | Kelvin HolmHeltz-Rayleigh Taylor (KH-RT) [28]        |
| Turbulence             | Re-Normalization Group (RNG) k-ε model [29]          |
| Spray droplet collision| Schmidt and Rutland [30]                             |
| Wall heat transfer     | Han and Reitz [31]                                   |
| NO_x                   | Extended Zeldovich [32]                              |
| Soot formation         | Hiroyasu soot model [33]                             |
| Soot oxidation         | Nagle-Stickland-Constable (NSC) model [34]           |

4.1 Code validation

Figure 7 represents comparison of experimental and simulated in-cylinder pressure traces for selected pilot injection mode (run no. 2 in Table 2). An acceptable match between experimental and simulated pressure traces can be noticed from Figure 7. Table 4 represents comparison of experimentally and numerically obtained values of NO emission for selected case. A good comparability between experimentally and numerically obtained values of NO emission can be observed from Table 4. Hence, it can be said that combustion process is modelled successfully.

![Comparison of experimental and simulated pressure traces](image)

**Figure 7.** Comparison of experimental and simulated pressure traces.

Table 4. Comparison of experimental and simulated NO emission data.

| NO (g/kWh) | Deviation (%) |
|------------|---------------|
| Experimental | 14.68          |
| Numerical   | 15.33          |
|             | 4.24           |
4.2 Numerical methodology

Base injector (which is used for experimental work presented in Section 3) has NHD of 0.120 mm. Two other NHDs under consideration are 0.08 mm and 0.160 mm respectively. NHD was varied in Converge from spray modelling tab. Rest of the injection parameters like spray angle, spray cone angle, SOPI, main injection timing, total mass of injected fuel per cycle, distribution of fuel in pilot and main injection, Cd as well as injection pressure were maintained constant with changing NHD. Variation of pilot as well as main injection duration with varying NHD was calculated in Converge by giving injection pressure, fuel mass and Cd as inputs. Based on calculated pilot injection duration, dwell was calculated (since pilot as well as main injection timings were held constant).

Representative line diagrams of injection profiles under pilot injection mode with varying NHD are represented in Figure 8. For selected pilot injection mode, change in dwell period with varying NHD is represented in Table 5. Dwell period increases with increasing NHD as inferred from Figure 8 as well as from Table 5. This is mainly attributed to shorter pilot injection duration in case of larger NHD due to increased fuel flow rate. Table 5 shows the variation of dwell with varying NHD for selected pilot injection mode.

| Sr. No. | NHD (mm) | Pilot quantity (%) | SOPI (° ATDC) | Main injection timing (° ATDC) | Dwell (° CA) |
|---------|----------|--------------------|---------------|-------------------------------|--------------|
| 1       | 0.08     | 40                 | -39           | -20                           | 8.5          |
| 2       | 0.120 (Baseline) | 40           | -39           | -20                           | 12           |
| 3       | 0.160    | 40                 | -39           | -20                           | 15.7         |

4.3 Numerical simulation results for combustion and performance characteristics

Effect of NHD on start of combustion (SOC), end of combustion (EOC) and combustion duration is tabulated in Table 6. In Converge, CA locations where 10% and 90% of the total injected fuel is burned are considered as SOC and EOC respectively. Based on these data, combustion duration was calculated.
Table 6. Effect of varying NHD on combustion parameters.

| Sr. No. | NHD (mm) | SOC (° ATDC) | EOC (° ATDC) | Combustion duration (° CA) |
|---------|----------|-------------|--------------|--------------------------|
| 1       | 0.08     | -11.28      | 14.32        | 25.60                    |
| 2       | 0.120    | -11.29      | 17.24        | 28.53                    |
| 3       | 0.160    | -10.60      | 24.19        | 34.79                    |

SOC is somewhat delayed in case of 0.160 mm NHD compared to other two cases as inferred from Table 6. This is due to longer delay period in case of 0.160 mm NHD due to larger diameter fuel droplets exiting from nozzle holes compared to other two cases. This can be clearly seen in Figure 9 which represents spray distribution inside combustion chamber for three different NHDs under consideration at -29° ATDC. At this CA, pilot fuel spray has sufficiently spread inside combustion chamber after striking on the piston bowl edge even for longest injection duration case (0.08 mm NHD). Larger spread area of spray inside combustion chamber at similar CA for 0.160 mm NHD case compared to other two NHDs also proves shorter injection duration with increased NHD. Moreover, density of larger diameter fuel droplets inside spray as well as on the edge of piston bowl is highest in case of 0.160 mm NHD and lowest in case of 0.080 mm NHD. In other words, larger the NHD, larger will be the fuel droplet diameter and vice-versa. These larger diameter fuel droplets in case of 0.160 mm NHD has smaller surface to volume ratio which results in poor mixing and vaporization of fuel droplets causing elongated physical delay period. However, surface area of spray droplets increased due to finer atomization in case of smaller NHD. Due to this, air entrainment inside fuel spray improved significantly and led to better air-fuel mixing in case of smaller NHD. Although, there is almost no difference in SOC location for 0.08 mm and 0.120 mm NHD cases. This might be due to longer injection duration in case of 0.08 mm NHD compared to 0.120 mm NHD case which surpassed the benefit of improved atomization. However, EOC is much earlier and combustion duration is much shorter in case of 0.08 mm compared to baseline case. Detailed explanation of this trend is given in subsequent discussion.

Simulated mean gas temperature (MGT) traces with changing NHD are represented in Figure 10. Rate of rise in temperature in earlier stages of combustion as well as peak temperature are highest in case of 0.08 mm NHD as inferred from Figure 10. This is mainly attributed to faster progress of combustion process due to improved atomization in case of 0.08 mm NHD. While having similar SOC, rate of temperature rise after initiation of combustion is much more rapid in case of 0.08 mm NHD compared to baseline case as can be seen from Fig 9. This proves that even though there is no noticeable difference in SOC, progress of combustion process is much more rapid in case of 0.08 mm NHD compared to baseline case. Peak temperature is lowest in case of 0.160 mm NHD as expected but a close observation from Figure 10 reveals that temperature in the later stage of expansion process is larger in case of 0.160 mm NHD compared to baseline case. This is clearly shown in Figure 10 (A).

Figure 9. Distribution of fuel droplets for different NHDs at -29° ATDC.
This might be due to longer combustion duration associated with 0.160 mm NHD case (refer Table 6) which persisted higher temperature for longer time in expansion stroke compared to baseline case.

![Diagram](image)

**Figure 10.** Effect of NHD on mean gas temperature traces.

![Diagram](image)

**Figure 11.** Effect of NHD on in-cylinder pressure traces.

In-cylinder pressure traces with varying NHD are represented in Figure 11. Baseline NHD obtained highest peak in-cylinder pressure compared to other two as can be noticed from Figure 11. Difference in peak pressure between baseline NHD and 0.080 mm NHD is quite negligible (0.72%), but noteworthy reduction in peak pressure (2.48%) is noticed with 0.160 mm NHD as compared to 0.120 mm NHD. Almost similar peak pressure in case of 0.08 mm NHD and 0.120 mm NHD is mainly
attributed to longer injection duration in case of 0.08 mm NHD compared to 0.120 mm NHD case which surpassed the benefit of shortened delay period due to finer atomization. Lowest peak pressure in case of 0.160 mm NHD is mainly due to deteriorated fuel evaporation rate (due to larger diameter of fuel droplets) which lowered the rate of pressure rise during initial stages of combustion and led to overall slow progress of combustion process compared to other two cases. Moreover, peak pressure continued to shift away from top dead centre (TDC) with decreasing NHD as inferred from Figure 11. This is due to faster progress of combustion process after initiation of combustion with decreasing NHD as explained earlier. This faster progress of combustion process with lower NHD produced larger force on piston in shorter time so that pressure drop in expansion stroke due to downward motion of piston was partly compensated and pressure continued to increase in expansion stroke for longer duration. This delayed occurrence of peak pressure persisted somewhat higher pressure in expansion stroke in case of lowest NHD (0.08 mm) compared to baseline case as well as largest NHD (0.160 mm) case.

- Figure 12. Effect of NHD on in-cylinder temperature and equivalence ratio distribution at (a) -8° ATDC and (b) 13° ATDC.
Two distinguished regions, i.e. a high temperature region (HTR) and a low temperature region (LTR) are clearly visible in Figure 12(a). Presence of HTR and LTR is attributed to pilot fuel combustion and evaporation of main fuel spray respectively. Greater HTR can be noticed from Figure 12(a) in case of lowest NHD compared to other two cases. This trend is further supported by Figure 10 which represents simulated MGT for all three NHD cases. Most of the HTR in case of 0.08 mm NHD is gathered around piston circumference and top of the piston, whereas HTR in other two cases is much more staggered. HTRs in case of 0.120 mm and 0.160 mm NHD are losing their contact from piston circumference and pointing towards center of combustion chamber and results in stretching of HTR. This effect is more pronounced in case of 0.160 mm NHD. This stretching of HTRs in cases of 0.120 mm NHD and 0.160 mm NHD led to reduced temperature whereas HTR in case of 0.08 mm stays on the piston circumference and maintains higher temperature. Moreover, LTR continues to decrease with decreasing NHD as can be seen in Figure 12(a). This indicates faster rate of main fuel evaporation with reducing NHD. Evaporation rate is much faster in case of 0.08 mm NHD due to finer droplets exiting from nozzle which produced smallest LTR among all the three cases. Slowest evaporation rate associated with largest NHD (0.160 mm) produced largest LTR among all the considered cases. Moreover, as stated earlier, total mass of injected fuel per cycle as well as distribution of fuel in pilot and main injection are maintained constant with varying NHDs. Therefore, greater HTR in case of 0.08 mm NHD (Figure 12(a)) compared to other two cases proves that greater pilot fuel mass has already been combusted. Additionally, main fuel evaporates at faster rate in case of 0.08 mm NHD compared to other two cases (evident by presence of smallest LTR). The combined effect of both these factors (presence of greater HTR and faster evaporation of main fuel) led to faster combustion of main fuel in case of 0.08 mm NHD compared to other two cases. This is the main reason for getting earlier EOC as well as shorter combustion duration in case of 0.08 mm NHD compared to 0.120 mm NHD case in spite of having almost similar SOC (refer Table 6). Due to this combined effect, interactions between still burning pilot flame and liquid components of main fuel continued for shorter duration and hence became less severe in case of 0.08 mm NHD compared to other two cases. This hypothesis is further supported by presence of least rich regions in case of 0.08 mm NHD compared to other two cases in Figure 12. HTR is smallest and LTR is widest in case of 0.160 mm NHD. This smallest HTR indicates that least pilot fuel has been combusted by this time compared to other two cases and widest LTR depicts that evaporation of main fuel is also slowest. Due to both these factors, combustion of main fuel continued for much longer duration which resulted in longest combustion duration in case of 0.160 mm NHD (refer Table 4) compared to other two cases. This will facilitate more time for interactions between still burning pilot flame and liquid components of main fuel and therefore these spray interactions became more severe. Due to this, combustion of main fuel took place under more locally rich condition which is evident by presence of highest rich regions in Figure 12 in case of 0.160 mm NHD compared to other two NHDs under consideration. These results are totally different from what was expected. This is because in case of 0.08 mm NHD, dwell between pilot and main injection is shortest and dwell is largest in case of 0.160 mm NHD (refer Figure 8 and Table 5). Moreover, all the injection parameters are also held constant with changing NHD as discussed earlier. Therefore, most aggressive spray interactions as well as largest rich regions were expected in case of 0.08 mm NHD (due to shorter dwell) and least aggressive spray interactions as well as smallest rich regions were expected in case of 0.160 mm NHD (due to longer dwell) based on results reported by previous studies as well as based on experimental results obtained in present work which showed greater smoke emissions with shorter dwell (refer Figure 5). The prime reason for this obtained trend is the greater homogeneity of mixture prepared in a shorter time due to improved atomization of injected fuel droplets which significantly reduced their physical delay period in case of smaller NHD. Greater homogeneity of the charge is further evident by much more consistent distribution of equivalence ratio as well as presence of least rich regions in case of 0.08 mm NHD compared to other two cases in Figure 12. This highly homogeneous mixture in case of 0.08 mm NHD compared to other two cases caused faster combustion of pilot as well as main injected fuel and drastically reduced the time available for spray interactions compared to other two cases.
Overall faster combustion process is further evident by faster growth of HTR in Figure 12 as well as least rich regions near and inside the squish regions in Figure 12(b) in case of 0.08 mm NHD.

4.4. Numerical simulation results for emission characteristics

Computed net soot emissions (represented by full lines) along with soot formation (represented by dotted lines) and soot oxidation process (represented by dashed lines) are shown in Figure 13. First lower peak of soot formation for all three cases under consideration indicates that almost all the pilot fuel burnt in pre-mixed mode only and prime source of soot emissions is the main injected fuel. Soot formation is highest in case of largest NHD and lowest in case of smallest NHD as inferred from Figure 13. This is because spray interactions continued for longest duration in case of 0.160 mm NHD and for shortest duration in case of 0.08 mm NHD as discussed earlier. This facilitated longest time for soot formation in case of 0.160 mm NHD and least time for soot formation in case of 0.08 mm NHD respectively. Therefore, severity of spray interactions increased drastically in case of 0.160 mm compared to other two cases. This resulted in much higher rate of soot formation in case of 0.160 mm NHD compared to 0.08 mm NHD and baseline NHD cases which can be clearly visualized from Figure 13. Highest and lowest fuel rich regions with of 0.160 mm and 0.08 mm NHDs respectively in Figure 12 also supports this trend of soot emissions. Baseline NHD produced soot emissions which are in between 0.08 mm and 0.160 mm NHD cases. One important thing to be noticed from Figure 13 is that inspite of highest soot formation, net exhausted soot emissions are somewhat lower in case of 0.160 mm NHD compared to baseline NHD case. This is mainly attributed to greater temperature persisted in expansion stroke in case of 0.160 mm NHD compared to baseline NHD case as seen in Figure 10 which helped to enhance the oxidation rate of formed soot in case of 0.160 mm compared to baseline NHD case. This can be seen in Figure 13 also that rate of soot oxidation as well as duration of soot oxidation are higher in case of 0.160 mm NHD compared to baseline NHD case due to which net exhausted soot emissions are lower. When NHD is reduced from 0.160 mm to 0.08 mm, soot emissions reduced by 87.2%. Therefore, it can be said that under pilot injection mode, smaller NHD results in lesser soot formation in spite of further shortening of dwell period for a given pilot mass as
well as given pilot and main injection timings. Moreover, with smaller NHD, SOPI can be further retarded (or dwell can be further reduced) to get additional improvement in BSFC with still lower or similar smoke emissions compared to larger NHD case.

Figure 14. Computed NO emission for different NHDs.

Figure 14 represents computed NO emission for three different NHDs under consideration. NO emission is increased continuously with reduction in NHD as inferred from Figure 14. When NHD is reduced from 0.160 mm to 0.080 mm, NO emission was increased by around 43%. Highest NO emission is exhausted in case of 0.08 mm NHD. This is primarily due to faster combustion process associated with 0.08 mm NHD compared to other two cases which produced highest in-cylinder temperature as indicated in Figure 10 and augmented formation of NO emission.

Figure 15 shows variation of computed HC emissions for varying NHDs. Two different peaks of HC emissions i.e. first one for pilot fuel and second one for main fuel can be seen for all the three NHDs under consideration. HC emissions from pilot fuel are mainly due to wall wetting caused by pilot spray. Pilot injection duration is longest in case of 0.08 mm NHD and shortest in case of 0.160 mm NHD as discussed earlier. Therefore, in case of smaller NHD, pilot injected fuel has greater chance of wall wetting and greater pilot fuel mass sneak into crevice volume and squish regions, which led to higher HC emissions from injected pilot fuel. This is the reason for getting highest and lowest first peak of HC emissions in case of 0.08 mm NHD and 0.160 mm NHD respectively. A second peak of HC emissions is lowest for 0.08 mm NHD and increases continuously with increasing NHD as inferred from Figure 15. Improved mixing due to finer atomization resulted in lesser HC emissions formation from main fuel in case of smaller NHD compared to larger NHD case. Moreover, HC emissions formation from main fuel continued for longer duration in case of smaller NHD which is evident from delayed second peak of HC emissions in case of smaller NHD. This is mainly attributed to longer main injection duration associated with smaller NHD. Overall least HC emissions exhausted in case of 0.08 mm NHD (as seen from Figure 15(A)) is mainly attributed to least HC emissions formation. In spite of overall greater HC emissions formation in case of 0.160 mm NHD compared to baseline case, exhausted HC emissions are larger in case of baseline NHD compared to
0.160 mm NHD as inferred from Figure 15(A). This mainly due to higher temperature persisting in the later stage of expansion process in case of 0.160 mm NHD compared to baseline case (refer Figure 10) which facilitated oxidation of formed HC emissions for longer duration in expansion stroke.

**Figure 15.** Computed HC emissions for different NHDs.

5. Conclusions
In first part of present work, an experimental study was conducted to analyze the effect of dwell on combustion, performance and emission characteristics of diesel engine. Experimental results showed that shorter dwell results in improvement of BSFC but with increasing smoke emissions penalty. In second part of work presented, effect of variation of NHD on combustion and emission parameters of diesel engine was investigated numerically with an aim to mitigate the disadvantage of increased smoke emissions associated with shorter dwell. All the injection parameters were kept unchanged with varying NHD so that sole effect of NHD on combustion and emission characteristics can be investigated. In addition to baseline NHD (0.120 mm), two other NHDs, i.e. 0.08 mm and 0.160 mm were considered. Numerical results showed that fuel injection with smaller NHD resulted in faster evaporation rate of injected fuel droplets and hence overall faster combustion process compared to that with larger NHD case. Due to this, spray interactions continued for shorter duration and hence became less severe in case of smaller NHD compared to larger NHD case. Because of this, least NHD (0.08 mm) resulted in lowest soot formation in spite of having shortest dwell among all the three NHDs under consideration. However, smaller NHD affected NO emission negatively.

Therefore, for a given pilot quantity as well as for given pilot and main injection timings, soot/smoke emissions can be reduced by lowering NHD. Moreover, in case of smaller NHD, dwell can be further reduced in order to achieve additional reduction in BSFC with smoke/soot emissions being still lower or equivalent to larger NHD case.
Acknowledgement
Authors sincerely acknowledge the financial support provided by SVNIT-Surat towards this work under institute annual plan head.

6. References

[1] Brijesh P and Sreedhara S 2013 Int. J. Auto. Technol. 14 195
[2] Lee J, Hong K, Choi S, Yu S, Choi H and Min K 2013 J. Mech. Sci. Technol. 27 1135
[3] Huang H, Huang R, Guo X, Pan M, Teng W, Chen Y and Li Z 2019 Appl. Energy 250 185
[4] Yun H, Choi K and Lee CS 2018 Appl. Therm. Eng. 141 90
[5] Liu B, Cheng X, Liu J and Pu H 2018 Appl. Therm. Eng. 141 90
[6] Huang H, Zhai Z, Zhu J, Lv D, Pan Y, Wei H and Teng W 2019 Appl. Energy 249 377
[7] Okada K, Mori K, Shiino S, Yamada Y and Matsumoto Y 2007 SAE Technical Paper 2007-01-4178
[8] Huang H, Wang Q, Shi C, Liu Q and Zhou C 2016 Appl. Energy 160 581
[9] Zheng Z, Yue L, Liu H, Zhu Y, Zhong X and Yao M 2015 Energy Convers Manag 90 1
[10] Herfatmanesh MR, Lu P, Attar MA and Zhao H 2013 Fuel 109 137
[11] Sahoo D, Miles PC, Trost J, Leipertz A 2013 SAE Int J Engines 6 1716
[12] Qin D, Leick M, Liu Y and Lee CF 2011 Fuel 90 1884
[13] Park C, Kook S and Bae C 2004 SAE Technical Paper 2004-01-0127
[14] Li X, Zhou H, Zhao LM, Su L, Xu H and Liu F 2016 Energy Convers. Manag. 129 180
[15] Vijay Kumar M, Veeresh Babu A and Ravi Kumar P 2018 Renewable Energy 119 388
[16] Bergstrand P and Denbratt I 2001 SAE Technical Paper 2001-01-2010
[17] Pang KM, Jangi M, Bai X, Schramm J and Walther JH 2017 Energy Proc. 142 1028
[18] Zhang W, Tian J and Nishida K 2011 SAE Technical Paper 2011-01-1813
[19] Du W, Lou J and Liu F 2017 SAE Technical Paper 2017-01-2300
[20] Ganesh V, Deshpande S, Sreedhara S 2015 Int. J. Eng. Res. 17 469
[21] Hiren D, Bharatkumar S and Brijesh P 2019 Clean Technol. Environ Policy 21 905
[22] Brijesh P and Sreedhara S Clean Technol Environ Policy 2016 18 2325
[23] Senecal PK, Pomraning E, Richards KJ, Briggs TE, Choi CY, McDavid RM, Patterson MA 2003 SAE Technical Paper 2003-01-1043
[24] Beale JC, Reitz RD 1999 Atomiz. sprays 9 623
[25] Han Z and Reitz RD 1995 Combust. Sci. Technol. 106 267
[26] Schmidt DP and Rutland CJ 2000 J Comput. Phys. 164 62
[27] Han Z and Reitz RD 1997 Int. J. Heat Mass Trans. 40 613
[28] Zeldovich YB, Barenblatt GI, Librovich VB and Makhviladze GM 1986 Consultants Bureau, New York
[29] Hiroyasu H and Kadota T 1976 SAE Technical Paper 760129
[30] Nagle J and Stickland RF 1962 Proc. Fifth Carbon Conf. 1 154