Investigation of cyclic variations in air-fuel ratio, cylinder wall temperature, and residual gas fraction of a dual fuel compression ignition engine

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Abstract. Dual fuel operation in compression ignition engines is an effective way to reduce the NOx emission. Within the certain range of fuel premixing ratio (PR), the dual fuel operation in CI-engines results in improved thermal efficiency. The dual fuel CI-engine has relatively higher cyclic variations in comparison to conventional CI-engine which limits the range of fuel premixing in dual fuel CI-engine. The cyclic variations in air-fuel ratio, cylinder wall temperature, and residual gas fraction are the major factors, which govern the variations in combustion parameters. The cyclic variations in combustion need to control for stable engine operation. The present study estimates the cyclic air-fuel ratio, cylinder wall temperature, and residual gas fraction from the measured in-cylinder pressure data of dual-fuel CI-engine. The experiments are performed on a modified single cylinder CI-engine equipped with a separate port fuel injector and its controller to operate an engine in dual fuel mode. In this study, 1500 consecutive engine cycles are recorded, and air-fuel ratio, cylinder wall temperature, and the residual gas fraction is estimated for each cycle. Pressure moment method is used to estimate the cyclic air-fuel ratio. The cyclic cylinder wall temperature is calculated by determining the inversion angle. The cyclic variations are analyzed using statistical methods.

1. Introduction
The physiochemical processes in IC-engine is a complex dynamic process which depends on the several factors such as fuel-air ratio, residual gas fraction, in-cylinder charge motion, fuel injection events, etc. [1]. The variations in these factors directly influencing the combustion behavior (i.e., in-cylinder pressure, the rate of heat release, pressure rise rate, etc.) and emission characteristics of the engine. The investigation of these cyclic variations in the engine combustion cycles is an area of interest for researchers because, excessive variations in the combustion cycles lead to fluctuation in engine power output, excessive engine noise, higher emissions, lower fuel economy and also may cause structural damage to the engine. These cyclic variations should be minimum to operate an engine in stable condition. The CI-engines are relatively more fuel economic and produce higher engine torque and power which makes these engines first choice for heavy-duty applications. However, CI-engines emit regulated and unregulated emissions in higher concentration. The trade-off between the NOx and soot emission is one of the major challenges in the CI-engine. To tackle with this challenge, post-treatment devices are installed in the tailpipe (engine exhaust line) of the engine to meet the stringent legislation limits. These devices are negatively influencing the fuel economy of the engine and aid extra cost to the vehicle. In previous studies, several alternative fuels were proposed for CI-engine; however, they were not commercialized due to their operational limitations. Alcohol fuels
are most investigated alternative fuels for internal combustion engines and are considered as the best alternative fuel for gasoline engines. These alcohol fuels can also be used in CI-engines by preparing a blended fuel of diesel and alcohol with the addition of a small amount of additives to avoid the fuel layer separation. Another method to use high octane fuel such as gasoline, ethanol, etc. in CI-engine is by separate injection of these fuels in the intake manifold and direct injection of diesel fuel in the cylinder. This strategy is known as dual fuel operation in CI-engine. The dual fuel operation in CI-engine has relatively higher cyclic variations in the combustion cycle in comparison to the conventional diesel engine. The premixing of high octane fuel in dual fuel operation significantly influencing the cyclic variation, and increases with an increase in the fuel PR. The higher cyclic variations in the combustion parameters limit the range of fuel PMR [2]. To reduce or control the cyclic variations, the air-fuel ratio (AFR), residual gas fraction (RGF) and in-cylinder wall temperature can be used since these parameters can be controlled externally. The AFR can be controlled by varying the amount of injected fuel in the cycle. The RGF and in-cylinder wall temperature can be controlled through variable valve timing and engine cooling system respectively. Additionally, for advanced automotive engines, online/ real-time estimation of AFR, RGF and in-cylinder temperature can also be used to control or minimize the cyclic variations through the development of closed-loop system. However, availability of the sensor, accuracy, higher cost are the major constraints in direct measurement of these parameters and their online control. By using measured in-cylinder data, the AFR, RGF and cylinder wall temperature can be calculated which requires no additional sensor. The piezo-electric pressure transducer is still being used for measuring the in-cylinder combustion pressure which is further used for analysis of combustion characteristics. By using measured in-cylinder pressure, the algorithm can be developed and used for estimating these parameters and to develop a closed loop control system. There are different methods such as pressure moment method, fast Fourier transform descriptors, heat release model, etc. which were used for the estimation of AFR from the in-cylinder pressure measurement [3-8]. There are various methods such as iterative approach, ideal gas equation, valve overlap backflow model, etc. are used for the estimation of RGF from cylinder pressure in various studies [9-12] for SI and CI-engines. The inversion method is typically used for the estimation of in-cylinder wall temperature [13, 14]. The present study estimated the cyclic variations in AFR, RGF and in-cylinder wall temperature and their effect on the cyclic variations in the combustion parameters of gasoline/diesel dual fuel CI-engine. The dual fuel engine was tested for various fuel premixing ratio at different engine loads.

2. Experimental Setup

The experiments were performed on a single cylinder, 661.45 cc CI-engine. To run a conventional CI-engine in dual fuel mode, the intake manifold of the engine is modified, and a solenoid injector is installed in it after suitable hardware modification. The mass of port injected gasoline in each cycle is controlled through a separate port fuel injector controller. The port fuel injector and the fuel pump were calibrated to govern the amount of injected fuel for a particular pulse width given from the controller. The diesel was injected directly in the cylinder using a mechanical fuel injector. In-cylinder combustion pressure is measured by using a piezoelectric pressure transducer. The crank angle position is determined by using an optical crank angle encoder of 1 CAD resolution. For logging the online combustion pressure, a high-speed data acquisition system is used. The more detail about the experimental setup is illustrated in the previous studies by the authors [2]. The cyclic AFR, RGF and wall temperature is estimated by developing MATLAB program.

The engine was tested at 25%, 50% and 100% engine loads for various fuel premixing ratios. As discussed the gasoline is used as low reactivity fuel and injected in the intake manifold, and diesel is used as a high reactivity fuel and directly injected in the cylinder. Fuel PR illustrates the energy contribution ratio of two fuels in the engine cycle and is determined from the equation given below

\[ \text{Fuel premixing ratio (PR)} = \frac{m_g \times LHV_g}{m_g \times LHV_g + m_d \times LHV_d} \times 100 \]  

(1)
Where \( m_g \) and \( m_d \) represent the mass of gasoline and diesel respectively. \( LHV_g \) and \( LHV_d \) are the lower heating value of gasoline and diesel respectively. The D100 represents the neat diesel operation, PR20 represents the 20% fuel premixing ratio of gasoline/diesel (on an energy basis), PR40 represents the 40% fuel premixing ratio of gasoline/diesel and so on for other fuel premixing ratios.

3. Methodology

3.1. AFR Estimation

For the estimation of AFR, pressure moment method has been used. It is based on the statistical moments to capture the shape of the in-cylinder pressure time history data. Theoretically, an infinite number of moments are required to capture the in-cylinder pressure data entirely, but due to the computational limitations, a limited number of moments has been used to determine the AFR. Cyclic AFR has been estimated by using the following correlation

\[
AFR = a_1 + a_2 \times I_2 + a_3 \times I_3 + a_4 \times N
\]  

(2)

Where \( I_2 \) and \( I_3 \) are the normalized moments. \( a_1, a_2, a_3, \) and \( a_4 \) are the polynomial constants. 'N' corresponds to engine speed. The central moments are used as descriptors of the in-cylinder pressure time history. The \( n \)th central moment is given by equation (3)

\[
M_n = \int_{\theta_{final}}^{\theta_{initial}} (\theta - \bar{\theta})^n P(\theta) d\theta
\]  

(3)

Where, the centroid, \( \bar{\theta} = \frac{M_1}{M_0} \) is used to shift the second and third moments to centroidal moment to compensate for the effect of any arbitrary reference angle. The normalized moments \( I_2 \) and \( I_3 \) are given as

\[
I_2 = \frac{M_2}{M_1}
\]  

(4)

\[
I_3 = \frac{M_3}{M_1}
\]  

(5)

The normalization reduces the influence of low-pressure measurement uncertainty, thus enabling the measurement of pressure curve by a low-cost sensor [8]. The coefficients are determined by the Least-square fit of the experimental data. In this study, \( \theta_{initial} \) has been chosen as the start of injection (SOI). As before SOI, i.e., during compression, the pressure curve is more dependent on the mass of the present gases as compared to the AFR. \( \theta_{final} \) has been chosen in such a way that there is the least amount of error between the estimated average AFR and the experimentally calculated average AFR.

3.2. RGF Estimation

In CI-engine, since there is no throttle valve to control the airflow, so the mass of air intake per cycle can be assumed constant for a given engine load. The masses of gasoline and diesel being very low as compared to that of air. Thus their variations doesn’t impose much error in the RGF estimation. The RGF can be calculated from equation (6)

\[
RGF = \frac{m_{res}}{(m_{IVC})_{next\ cycle}}
\]  

(6)

From ideal gas law, the in-cylinder mass at the EVC can be determined which is equal to the residual mass provided the effect of valve overlap can be neglected.

\[
m_{res} = m_{EVC} = \frac{P_{EVC} V_{EVC}}{T_{EVC} R_{EVC}}
\]  

(7)

The gas constant at EVC is equal to the gas constant of the exhaust gas.

\[
R_{EVC} = R_{res}
\]  

(8)

The total in-cylinder mass for the subsequent cycle is the sum of the Residual Mass from the previous cycles, the mass of the air intake and the mass of the port injected gasoline.

\[
(m_{IVC})_{next\ cycle} = m_{EVC,present\ cycle} + m_{a,\ next\ cycle} + m_{GASOLINE,\ next\ cycle}
\]  

(9)
3.3. In-cylinder wall temperature estimation

In an IC engine, after intake stroke, the temperature of the in-cylinder gases is lower than that of the cylinder which increases during compression. At the particular crank angle, the temperature of the gas becomes equal to that of the cylinder wall, after that it rises above the cylinder wall. The crank angle at which both gas and wall are at the same temperature is known as inversion point. At this point, there is no heat flux between the wall and the gas. If the temperature of the gas is known at that crank angle, then the temperature of the wall can also be determined.

Since both, gas and wall, are at the same temperatures, there is no heat transfer between them. Therefore at inversion point, the adiabatic condition prevalent. At inversion point, the polytropic index (m) during the compression stroke becomes equal to that of the adiabatic index (k). The polytropic index during the compression stroke is given by

\[ m(\theta) = \frac{\log\left(\frac{P(\theta+\Delta\theta)}{P(\theta-\Delta\theta)}\right)}{\log\left(\frac{V(\theta+\Delta\theta)}{V(\theta-\Delta\theta)}\right)} \tag{10} \]

In this study, eq. (10) has been used to determine polytropic index (m) at each crank angle during the compression stroke. The inversion crank angle can be determined at that crank angle where the polytropic index becomes equal to that of the adiabatic constant of the gas. The temperature at the inversion crank angle will be equal to the mean gas temperature and is calculated by equation (11)

\[ T(\theta_{inv}) = \frac{P(\theta_{inv})V(\theta_{inv})}{mass \cdot R} \tag{11} \]

4. Results and discussion

This section firstly presents the estimation of cyclic AFR, RGF and in-cylinder wall temperature from in-cylinder pressure measurement of 1500 consecutive engine cycles. Later on the effect of cyclic variation of AFR, RGF and in-cylinder wall temperature on indicated mean effective pressure (IMEP) and total heat release have been investigated.

4.1. Estimation of cyclic AFR, RGF and in-cylinder wall temperature

The cyclic AFR, RGF and in-cylinder wall temperature has been calculated using correlations discussed in the methodology section. The AFR has been taken as the ratio of the mass of the intake air to the total mass of the fuel injected (gasoline+diesel). For each tested engine load, the different regression equation coefficients were calculated which are shown in Table 1. For each tested engine load the normalized pressure moments is calculated to determine the regression coefficients. The normalized 2nd and 3rd order pressure moments \( I_2 \) and \( I_3 \) with respect to AFR at different engine load is presented in figure 1. Figure 1 depicts that the 2nd order normalized moment shows a good linear correlation with AFR while for 3rd order normalized moment, no significant correlation was found (especially at 25% and 50% engine loads).

| Regression Coefficients | 25% Load | 50% Load | 100% Load |
|-------------------------|---------|---------|----------|
| a1                      | 217.04  | 237.67  | 58.89    |
| a2                      | -162.03 | 9.81    | -29.57   |
| a3                      | -0.11   | -3.15   | -0.857   |
| a4                      | -0.038  | -0.14   | -0.00954 |
Figure 1. The normalized 2nd and 3rd pressure moments with respect to AFR for different loads.

The cyclic AFR is calculated using eq.2. Figure 2 illustrates the cyclic variations in AFR for 1500 consecutive engine cycle of dual fuel CI-engine. The variations in the fuel injection characteristics such as variation in the quantity of fuel (either port or directly injected fuel), vaporization of low/high reactivity fuels, mixing process and aerodynamic variations collectively contributes to variations in the AFR [15]. Figure 2 depicts that on increasing the engine load from 25% to 100%, the cyclic variations in AFR decreases and this observation is similar for neat diesel operation (D100) as well as dual fuel operation. At lower engine load, the amount of injected fuel is very less (overall very lean mixture) which leads to longer ignition delay and slower post ignition energy release. Less amount of fuel is burnt in the combustion leads to reduced mean gas temperature into the combustion chamber which results in poor atomization and mixing of diesel (in case of neat diesel operation) and gasoline/diesel in dual fuel operation. This leads to create non-uniformity in AFR distribution in the chamber during the combustion. Additionally, in dual fuel operation, the valve wetting (due to port injection of gasoline) is another possible reason which varies the amount of gasoline fuel supplied into the combustion chamber. On increasing the engine load, the higher mean gas temperature leads to increase the intake manifold temperature and valve temperature which reduces the valve wetting effect and results in lower cyclic variations in AFR. Figure 2 also demonstrates that on increasing the fuel PR, the variations in AFR increases. Increase in the premixing of low reactivity fuel (gasoline) leads to longer ignition delay [2]. More fraction of gasoline injection in the intake manifold leads to higher wall wetting which results in more fluctuations in AFR distribution. Moreover, there are several non-uniformities in the fuel composition, temperature distribution and velocity during the compression stroke which varies with crank position (time). This results into dynamic change in the distribution of AFR in successive engine cycles at hottest region.
Figure 2. Estimated cyclic AFR for various fuel PR at different engine loads.

Figure 3 shows the comparison of estimated and measured mean AFR for various fuel PR at different engine loads. Figure depicts a good correlation between the measured and estimated mean AFR. It has been found for 25%, 50% and 100% engine load, the root mean square error (RMSE) between measured and estimated mean AFR are 1.33%, 4.04% and 1.14% respectively. Figure depicts mean AFR decreases with engine load as higher amount of fuel is required to inject at higher engine load to maintain the constant engine speed. Additionally, the mean AFR decreases with increase in fuel PR. On increasing the fraction of port injected gasoline, the more fraction of gasoline may trapped into the crevices volume and some fraction of charge will be loss during valve overlap, thus more fraction of diesel need to be injected to maintain the constant engine speed and results in reduced AFR.

Figure 3. Comparison of measured and estimated average AFR for different fuel PR at different engine loads.

The RGF has a significant effect on the combustion characteristics and engine-out emissions from the engine. The advanced automotive diesel engines are equipped with exhaust gas recirculation (EGR) system to reduce the NOx emission. There is a significant difference in the thermodynamic properties of residual gases in the cylinder and the fresh air/fuel mixture. The differences in the thermodynamic
properties may cause temporal variations in mean gas temperature which further affects the heat transfer during and after the compression stroke. Thus, the variations in RGF directly influencing the cyclic variations in the combustion process which needs to be investigated. The methodology used for the estimation of RGF is discussed in the previous section. The tested engine has a positive valve overlap that means, for some crank angle duration both the intake and exhaust valve remains open simultaneously. This causes to escape the residual gases from the cylinder due to the thrust from the fresh incoming charge from the intake valve. Additionally, the tested engine has no EGR facility. All these factors contribute to less RGF in the combustion chamber after each cycle for the tested engine. The time series of estimated RGF for 1500 successive engine cycles for various fuel PR at different loads is presented in figure 4. It has been observed from figure 4 that the variation in RGF is nearly constant throughout the engine cycles except for highest fuel PR. This reveals that RGF is less affected by the low and medium fuel premixing. At higher fuel premixing operation the unburned HC emission are very high as compared to lower or medium fuel premixing ratios. The possible reasons for higher HC emissions were discussed in the previous study by the authors [2]. The comparatively higher RGF at higher fuel PR ratio operation is might be due to the higher unburnt HC emission due relatively more fraction of fuel (gasoline) may trapped in the crevice volume.

Figure 4. Estimated cyclic RGF for various fuel PR at different engine loads.

Figure 5 depicts the estimated mean RGF rapidly increases for higher fuel PR (especially at 25% and 50% engine loads). Figure 5 also demonstrate a rapid increase in COV of RGF for higher fuel PR at 25% and 50% loads. On increasing the fuel PR, the ignition delay increases due to lower overall reactivity of the charge. The fluctuations tend to be more for lower loads due to higher ignition delay and subsequently more amount of fuel burn during the expansion stroke which results in higher cyclic variations. An increase in fuel PR leads to lower mean gas temperature, which results in higher residual gas density (figure 6). The higher residual gas density results into a higher residual fraction for higher fuel PR. Figure 6 indicates that at 100% engine load, there are no significant variations in the mean gas temperature at EVO, and the decrease in piston speed should be occurring due to decreasing maximum pressure with increasing PR. This decreased piston speed provides higher time for residual gases to escape as the valve duration is constant in terms of CAD. Thus resulting in a decrease of residual gas mass for 100% load.
The estimated in-cylinder wall temperature at inversion point is calculated using equations discussed in the previous section and the time series of estimated in-cylinder wall temperature for 1500 consecutive engine cycles is for low and high engine loads is presented in figure 7. With an increase in the engine load, higher fraction of fuel burnt into the combustion chamber which leads to higher mean gas temperature (figure 6), correspondingly results in higher in-cylinder wall temperature (figure 7). Figure 7 reveals that there is no significant cyclic variation in cylinder wall temperature with fuel PR (time series seems similar). The in-cylinder wall temperature is higher for neat diesel operation and decreases with an increase in the fuel PR. The gasoline has a higher heat of vaporization, injection of gasoline in the air leads to decrease in the temperature of charge before the start of ignition thus, results in lower cylinder wall temperature. The cyclic in-cylinder wall temperature shows not much variations with the fuel PR (coefficient of variation (COV) in figure 7) as the fluctuations in cyclic wall temperature is not much seems sensitive to fuel PR. The in-cylinder wall temperature strongly relies on the mean gas temperature, and the mean gas temperature depends on the combustion phasing. Typically, with an increase in the fuel PR, the peak pressure and rate of heat release decrease with retard combustion phasing. Combustion with retard phasing results in lower mean gas temperature. Since the experiments were performed for a constant supply of fuel energy at particular engine load condition, thus the amount of fuel energy inducted into the cylinder is constant. Therefore the reduction in mean gas temperature is not severe up to certain fuel PR (80% fuel PR) at fixed engine load (figure 6). Additionally, the wall temperature in itself has higher inertia to cyclic changes.
It can also be observed from figure 7 that the effect of load on the cyclic variation of in-cylinder wall temperature is not significant. There is a very slight decrease in the COV at high loads. The outer wall is well maintained at a nearly constant temperature by engine coolant, which also results in lower fluctuations in cyclic in-cylinder wall temperature. Even though an engine running with high cyclic variations in AFR at higher fuel premixing ratios and low load condition, the variations in wall temperature is not much significant.

4.2. Influence on IMEP

Effect of fuel PR on mean Coherence of IMEP with AFR, wall-temperature, and RGF, respectively at 25% load along with their COV is illustrated in figure 8. Figure 8(a) shows that up to 40% fuel PR (low premixing) the COV of IMEP gradually increases as the influence (Coherence) of AFR increases along with the gradual rise in the COV of AFR. While from 40% to 80% fuel PR, the variations in IMEP seems to be more influenced by the RGF (figure 8b). For higher fuel PR, the residual gas fraction mainly contains a higher amount of unburned hydrocarbon, which makes the combustion more sensitive to the RGF. For fuel premixing above 80%, the variations in IMEP is influenced by both AFR and RGF as well. The higher variations of IMEP for higher fuel PR is possibly due to the combined effect of higher variations in AFR and RGF. Figure 8(c) reveals that the coherence and COV of wall temperature seem lower and nearly constant throughout the fuel premixing. This indicates that wall temperature has a lower effect on the IMEP variations. Figure 16 shows the effect of fuel PR on mean Coherence of IMEP with AFR, Wall-Temperature, and RGF, respectively at 50% load along with their COV. Figure 9 depicts that the COV of IMEP is relatively less (less than 5%) at lower fuel PRs due to lesser influence of AFR, RGF and Wall Temperature on the IMEP. With increasing the fuel PR, the COV of IMEP increases along with the increasing COV of AFR (figure 9).

Additionally, the figure depicts the rapid increase in COV of IMEP for higher fuel PR which is mainly due to the rise in the COV of AFR and higher influence of RGF whose COV is also high in that region. Figure 10 depicts the effect of fuel PR on mean Coherence of IMEP with AFR, wall-temperature, and RGF at 100% load along with their COV. Figure 10 shows that the influences as well as the variations in all the parameters as nearly constant. The low COV of IMEP along with low COV of AFR indicates more complete combustion because of higher mean gas temperature at higher engine
load. The influence of AFR increases for higher fuel PR which results in an increase in variations in IMEP along with increasing variations in AFR. Still, constant but significant variations in the Wall Temperature can be observed. This is possibly due to the variations in RGF. Figure 8, 9 and 10 also depicts the influence, as well as the COV of wall temperature, has been nearly constant during the whole range for all the loads. The energy released to run the engine at constant load and RPM is almost same resulting in minor variations in Wall Temperature which in turn reflects the lesser influence on IMEP.

5. Conclusion
This study estimated the cyclic AFR, RGR and in-cylinder wall temperature from in-cylinder pressure measurement of a dual fuel CI-engine. The engine was tested for various fuel PRs at different engine loads. Additionally, the influence of cyclic AFR, RGF and wall temperature on IMEP is also investigated in this study. It is found that the estimated mean AFR follows the trend of measured mean AFR with good accuracy. The estimated AFR shows a decrease in mean AFR with increasing premixing and engine load. Additionally, the COV of AFR increases with fuel PR. The mean RGF increases with fuel PR. For higher fuel PR (>80%), steeper increase in mean RGF has been observed. The in-cylinder wall temperature shows a steeper decrease with fuel PR for 25% and 50% load as compared to a higher load (100%). Additionally, cyclic variations of wall temperature have been nearly constant over the entire range of fuel premixing. The coherence results are summarized in the table given below.
Table 2. Influence of cyclic AFR, RGF and In-Cylinder Wall Temperature on IMEP variations.

| Load    | Low (0% to 40% fuel PR) | Medium (40% to 80% fuel PR) | High (>80% fuel PR) |
|---------|-------------------------|-----------------------------|---------------------|
|         | AFR         | RGF       | Wall Temp | AFR         | RGF       | Wall Temp | AFR         | RGF       | Wall Temp | AFR         | RGF       | Wall Temp |
| 25%     | Medium      | Low       | Low       | Low        | Medium     | Low       | High      | High      | Low       | High      | High      | Low       |
| 50%     | Medium      | Low       | Low       | High       | Low        | Low       | High      | High      | Low       | High      | High      | Low       |
| 100%    | Low         | Low       | Low       | High       | Medium     | Low       | Medium   | Medium   | Medium   |

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