Nano World Journal

https://doi.org/10.17756/nwj.2023-s3-009

Research Article

Open Access

Analysis of Heat Transfer Control in CI Engines Fueled with Diesel/Methanol/1-pentanol Blends with Nanoparticles: Enhancing Performance and Lowering Emissions

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Received: July 26, 2023
Accepted: September 26, 2023
Published: September 29, 2023

Citation: Shahee S, Gautam PS, Singh AP, Kamaraj L. 2023. Analysis of Heat Transfer Control in CI Engines Fueled with Diesel/Methanol/1-pentanol Blends with Nanoparticles: Enhancing Performance and Lowering Emissions. *NanoWorld J* 9(S3): S46-S51.

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Abstract

Diesel engines' performance and emissions characteristics may be greatly enhanced by effective heat transfer control. In this work, the heat transfer losses in a CI engine running on PM5 and PM10 blends ternary fuel combinations made of diesel, methanol, and 1-pentanol, are examined. The goal is to assess these blends' potential for lowering heat losses and improving engine performance by adding nanoparticles. The convective and radiative heat transfer coefficients, blowby volume flow rates, and total blowby losses for PM5, PM10, and diesel under various engine loading situations were examined through experimental research. The comparisons and trends between the fuel mixes and diesel were clearly seen in the data. In terms of convective heat transfer, PM5 exhibited slightly higher coefficients than diesel at low load conditions, but lower coefficients at higher loads. Moreover, PM10 consistently demonstrated lower convective heat transfer coefficients than diesel across all loading conditions. Except at 75% load, where it showed higher coefficients, PM5 typically showed lower radiative heat transfer coefficients than diesel. At all loading circumstances, PM10 consistently shown lower radiative heat transfer coefficients than diesel. Following the trend of cylinder temperature, blowby volume flow rates for PM5 and PM10 were typically lower than those for diesel. However, except at 75% load, PM5 demonstrated greater overall blowby losses per cycle than diesel. Diesel exhibited higher overall blowby losses than PM10. According to the results, both PM5 and PM10 blends might decrease heat transfer losses in CI engines, which could boost thermal efficiency. The examination of losses due to heat transfer in a CI engine running on a combination of PM5 and PM10 fuel, in summary, sheds light on the potential for these fuel mixes to enhance engine performance. The results lay a foundation for further investigation and the creation of innovative fuel mixes for improved engine performance and lower emissions. They also contribute to our understanding of the features of heat transfer in CI engines.

Keywords

Diesel engine, Heat losses, Methanol, Emission, Combustion characteristics

Introduction

For many emerging nations throughout the world, the depletion of oil sources and the ensuing environmental contamination have become major problems. The internal combustion engine has been recognized as the major contributor to environmental pollution during the previous 20 years. Diesel consumption rates have been rising quickly in tandem with increased transportation and agricultural demand, which has resulted in an increase in diesel-related environmental pollution. An immediate requirement is required to create a sustainable alternative fuel to address these issues that can meet the high energy demands while reducing environmental impact. Numerous authors have explored alternative fuels which

includes hydrogen, biodiesel and alcohols aiming to reduce emissions hydrocarbon, CO (carbon monoxide), and smoke [1]. However, the commercialization of biodiesel has faced obstacles due to issues such as higher viscosity and oxidation stability [2]. Although alcohols, particularly methanol, have the potential to reduce emissions due to their oxygen content, stabilizing diesel-methanol blends has proven challenging because of strong polarity of hydroxyl group [3, 4].

In order to address the problem of phase separation in blended fuels, some scientists have employed mixers or magnetic stirrers within fuel tanks during their experimental investigations. The miscibility of methanol/diesel blends by mixing in a fuel tank and conducted the performance and emission metrices of a CI engine [5]. Their findings resulted in elevated BSFC (Brake specific fuel consumption) and NO_x (nitrogen oxides) emissions, accompanied by a decrease in BTE (Brake thermal efficiency), smoke and CO. A stirrer to prevent phase separation of methanol-diesel blend and analyzed the combustion, performance, and emissions [6]. They discovered higher CP, HRR, and BTE as well as reduced BSFC in contrast to diesel. An electromagnetic mill in a different study to keep blends of diesel and methanol stable. According to the findings, adding methanol had no appreciable impact on emissions [7]. While NO_x emissions exhibited an increase. A mechanical blender to mix diesel with methanol at different percentages (0 - 15% by volume), and they found that the higher the methanol content, the worse the brakes performed with consideration to BTE, peak CP, smoke, UHC, and CO emissions, as well as the worse the brakes performed in terms of fuel consumption, BSEC, CO, and NO_x emissions [8].

Some studies have looked into using higher alcohols and biodiesel to address the issues of miscibility. Tests at different speeds to investigate the impact of and methanol/ethanol/diesel blends on engine performance and emissions [9]. The study revealed that the blends led to increased BSFC and NO_x emissions. However, it was reported an improved BTE, reduced smoke opacity, and lower emissions of CO and total hydrocarbons. Scrutinized the influence of delayed-injection in a diesel/methanol dual-fuel engine on combustion and emissions. Higher methanol substitution ratios reduce NO_x and PM emission [10]. Small late-injection quantities have minimal effect on pressure and heat release rate. Increasing late-injection quantity decreases NO_x and accumulation mode PM. Delayed late-injection reduces NO_x emissions. The utilization of sustainable methanol as a partial alternative for diesel fuel in dual-fuel combustion engines [11]. The focus is on the reactivity and nanostructure of soot during oxidation and its impact on particulate filter regeneration. The results indicate that higher methanol substitution ratios improve soot oxidation reactivity and reduce activation energies. The implementation of a dual-fuel DI configuration to enhance combustion control in dual-fuel engines [12]. The experiments revealed that by increasing the quantity of methanol injection, the maximum energy substitution ratio reached 65.1%. The optimized configuration achieved a 52.4% energy substitution ratio and the highest thermal efficiency. Methanol addition improved combustion stability at low loads and increased combustion rate while reducing NO_x emissions. A mixture of biodiesel obtained from water hyacinth and hydroxy was tested in a CI engine for heat loss and combustion metrices [13]. The BD40 blend also exhibited higher peak HRR and CP. Overall, this study highlights the potential of combining hydroxy and biodiesel for improved engine performance with the exception of NO_x. A reliable computational fluid dynamics model for analyzing heat transfer in internal combustion engines [14]. By incorporating the effects of heat loss through solid walls and validating the model with experimental data, the researchers investigate the potential advantages of using thermal barrier coatings to enhance engine efficiency. This work establishes a robust framework for accurately studying heat transfer in reciprocating engines.

With consideration to the literature review, the predominant focus of authors has been on methanol/diesel blends in combination with either biodiesel or dual-fuel engines. With only a limited number exploring the potential of higher alcohols as additives to stabilize diesel/methanol blends. On the other hand, heat loss computation using alternative fuels are limited specifically with thermodynamic heat transfer models. Therefore, further research is necessary to thoroughly investigate the potential benefits and drawbacks of higher alcohol blends in compression-ignition engines and analyze the heat losses from the engine.

In this study, we aim to stabilize diesel-methanol blends using equal percentages of methanol and 1-pentanol. The test fuels are denoted as PM5 (“5% methanol, 5% 1-pentanol, and 90% diesel by volume”) and PM10 (“10% methanol, 10% 1-pentanol, and 80% diesel by volume”). It will evaluate the heat losses of combustion characteristics based on pressure data and employ a single-zone thermodynamic heat transfer model.

Materials and Method

Blend preparation

In this investigation, introduce an innovative strategy to enhance the compatibility of methanol/methanol blends by integrating 1-pentanol as an auxiliary solvent. 1-pentanol, alternatively known as n-pentanol or n-amyl alcohol, is a five-carbon alcohol featuring the molecular formula $C_5H_{12}O$. To characterize 1-pentanol as a fuel, diverse chemical properties can be found in the table 1 to gain a comprehensive understanding. Methanol/diesel blends often face challenges in achieving complete and stable mixing due to the differences in their chemical properties. By introducing 1-pentanol, we aim to enhance the compatibility and stability of the blend. Figure 1 showcases the results of our investigation, providing visual evidence of the stability achieved in the diesel-methanol-1-pentanol blends. This finding highlights the potential of using 1-pentanol as a promising co-solvent to overcome miscibility issues and improve the overall performance of these fuel blends.

Experimental setup

In this study, we conducted experimental investigations using a 1-cylinder diesel engine, as shown in (Figure 2). The detailed technical particulars of the engine are available in (Table 1). To ensure precise crank position measurement, a 1°

Table 1: Chemical properties of the blends.

Fuel	ASTM Method	Diesel	Pentanol	Methanol	PM5	PM10
Density (g/cm^3)	ASTM D4052	0.832	0.815	0.791	0.829	0.826
Calorific Value (MJ/kg)	ASTM D240	44.99	35.15	20.05	43.31	41.63
Cetane No.	ASTM D613	50	20	5	46.25	42.5
Viscosity (mm^2/s)	ASTM D445	2.76	2.88	0.58	2.65	2.55
Oxygen content (%)	-	0	18.2	50	3.41	6.82

CA resolution optical encoder was employed. A piezoelectric transducer was used to monitor the pressure within the cylinder, and a Lab VIEW-based programme and high-speed data gathering system were used to record the results [15]. These measurements were used to calculate important combustion parameters such as the CP, HRR, and MGT, ID, etc. Analysing in-cylinder pressure data in relation to crank position provides valuable insights into the complex combustion process [16, 17].



Figure 1: Stable blends.

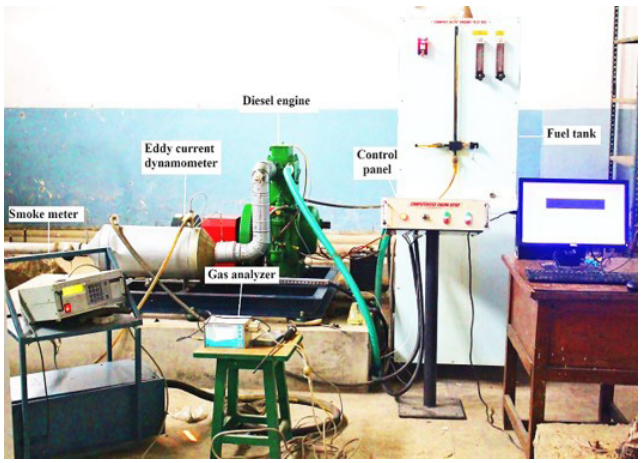


Figure 2: Diesel engine test rig.

Heat release rate

According to equation 1, the HRR is determined by the combined effect of two components: the rate of heat input into the system and the rate of heat loss from the system [18, 19]. These components contribute to the overall HRR and influence the performance and efficiency of the engine [20].

$$\frac{dQ}{d\theta} = \frac{dQ_{in}}{d\theta} + \frac{dQ_{loss}}{d\theta} \quad (1)$$

$$\frac{dQ}{d\theta} = \left(\eta_c \text{ x } m_f \text{ x } LHV \text{ x } \frac{d\chi}{d\theta}\right) + \frac{dQ_{loss}}{d\theta} \quad (2)$$

Heat loss or heat transfer model

The model integrates multiple internal processes occurring within the cylinder to facilitate precise calculations of heat transfer. These processes encompass elements such as airflow velocity, fuel spray dynamics, mixing, interactions between the spray and cylinder walls, turbulence, droplet evaporation, ID etc. The heat transfer correlation, as developed by Annand and Ma [19-21] offers a framework for estimating heat loss within the cylinder.

$$\frac{dQ_{loss}}{d\theta} = \left(h_c(\theta) + h_r(\theta)\right) A(\theta)\left(T_\theta - T_w\right)\left(\frac{1}{\omega}\right) \quad (3)$$

$$\omega = \frac{60}{360 \text{ x } RPM}$$

$h_c(\theta)$ was determined by the equation 4.

$$hc(\theta) = \frac{k_g Nu}{D} \quad (4)$$

To calculate the thermal conductivity of the gas (k_g), equation 5 was utilized, taking into account the gas temperature (T_θ). This equation incorporates coefficients that account for the temperature dependence of thermal conductivity.

$$k_g = 6.1944 \text{ x } 10^{-3} + 7.3814 \text{ x } 10^{-5} T_\theta - 1.2491 \text{ x } 10^{-8} T_\theta^2 \quad (5)$$

The Nusselt number was computed based on the cylinder bore diameter (D) and a power-law relationship with the Reynolds number (Re), as given by equation 6.

$$Nu = aRe^{0.7} \quad (6)$$

Finally, the radiation heat transfer coefficient ($h_r(\theta)$) was computed as:

$$h_r\left(\frac{w}{m^2 * K}\right) = 4.25 \text{ x } 10^{-9}\left(\frac{T_\theta^4 - T_w^4}{T_\theta - T_w}\right) \quad (7)$$

Furthermore, blowby, which involves the transfer of mass through the piston crevices, significantly impacts engine performance and emissions, particularly hydrocarbon emissions [22-25]. To estimate blowby flow velocity (Equation 8) and mass flow rate (Equation 9) [21], a compressible flow model is applied.

$$Vbb = \sqrt{2.\frac{\gamma}{\gamma+1}.\frac{R}{M}.T} \quad (for\ choked\ flow) \qquad (8)$$

Where γ, R, M, and T are the specific heat ratio, universal gas constant, molecular mass, and cylinder temperature.

$$dm = V_{bb}.A_{bb}.d\theta.N^{-1} \qquad (9)$$

Where A_{bb} and N are the "equivalent flow area and engine speed (rad/s)". In V_{bb} can be computed using the following equation 10.

$$V(\theta) = V_c + \frac{\pi B^2}{4}\left[L + R - R\cos\theta - \sqrt{\left(L^2 - \theta\right)}\right] \qquad (10)$$

Results and Discussion

Heat loss rate due to cylinder wall and cumulative heat loss

Figure 3, figure 4, figure 5, and figure 6 show the fluctuation in the rate of heat loss caused by the cylinder wall and cumulative heat loss per cycle for PM5, PM10, and diesel under different loading conditions. The maximum heat loss rate from the cylinder wall (in J/deg CA) for diesel, PM5, and PM10 was determined at each load condition. At 25% load, the maximum heat loss rate for diesel, PM5, and PM10 was 2.052, 1.787, and 1.613, respectively as shown in figure 3. At 50% load, these values were 2.502, 2.458, and 2.043, while at 75% load, those were 3.04, 3.104, and 2.503, and at 100% load, they were 3.51, 2.998, and 2.998, respectively (Figure 4). The PM10 blend showed lower heat loss rate at 25 and 50% load than PM5 and diesel due to reduced calorific value and higher latent of heat vaporization.

Upon comparison, it is evident that the maximum rate of heat loss for PM5 exhibited a reduction of 12.91%, 1.75%, and 14.58% in comparison to diesel at 25%, 50%, and 100% load, respectively. However, at 75% load, it was observed to be 2.10% higher than that of diesel. This trend can be ascribed to the disparity in peak pressure, with PM5 showing higher peak pressure at 75% load and lower peak pressure at other load conditions (Figure 5). Likewise, the maximum heat loss rate for PM10 demonstrated a reduction of 21.39%, 18.34%, 17.66%, and 14.58% compared to diesel at 25%, 50%, 75%, and 100% load, respectively. This indicates that the heat loss resulting from the cylinder wall exhibits a parallel trend to the cylinder pressure.

Furthermore, the maximum cumulative heat loss per cycle (in J/cycle) for diesel, PM5, and PM10 was determined at each load condition. At 25% load, the maximum cumulative heat loss was 150.1, 118.1, and 106.6 for diesel, PM5, and PM10, respectively. At 50% load, these values were 188.8, 183, and 147.8, while at 75% load, they were 214.9, 233.1, and 200.1, and at 100% load, they were 277.3, 240.9, and 226, respectively. Upon closer examination, it can be observed that the maximum aggregate heat dissipation for PM5 exhibited decreases of 21.31%, 3.07%, and 13.12% in comparison to diesel at 25%, 50%, and 100% load, respectively. However, at

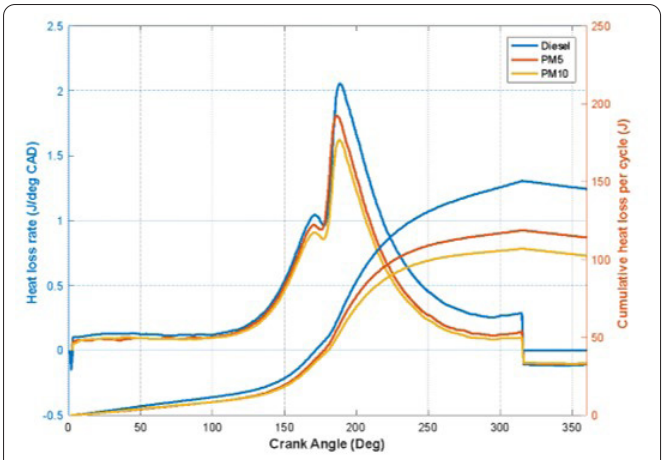


Figure 3: Heat loss from the wall and cumulative heat loss per cycle at 25% load.

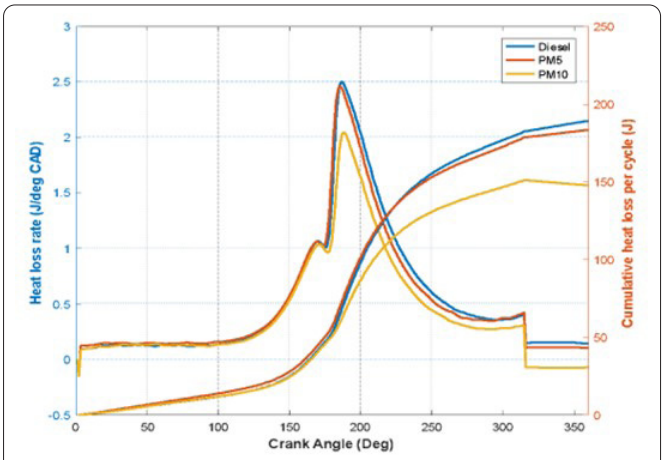


Figure 4: Heat loss from the wall and cumulative heat loss per cycle at 50% load.

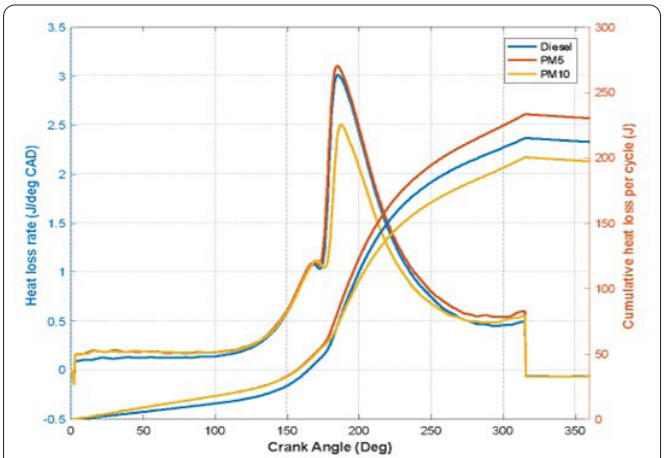


Figure 5: Heat loss from the wall and cumulative heat loss per cycle at 75% load.

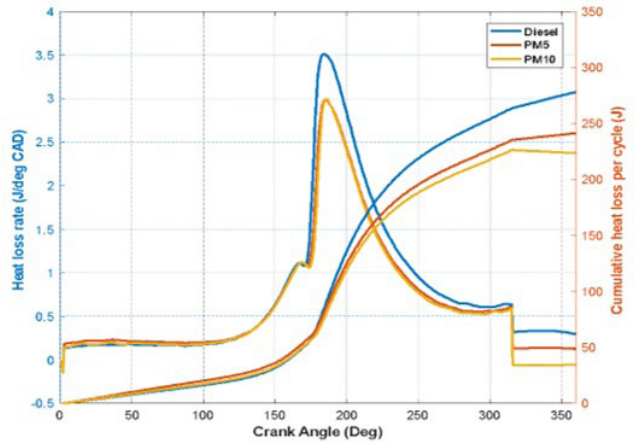


Figure 6: Heat loss from the wall and cumulative heat loss per cycle at 100% load.

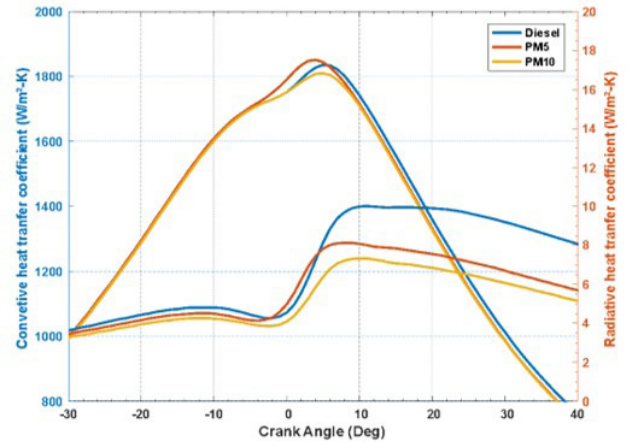


Figure 7: Convective and radiative heat transfer coefficient variation at 25% load.

75% load, it was identified to be 8.46% higher than that of diesel. Similarly, the maximum cumulative heat loss for PM10 demonstrated declines of 28.98%, 21.71%, 6.88%, and 18.49% in comparison to diesel at 25%, 50%, 75%, and 100% load, respectively. Therefore, it can be deduced that both the heat dissipation associated with the cylinder wall and the overall cumulative heat loss follow analogous patterns as the cylinder pressure under diverse loading conditions.

Variation of heat transfer coefficient

Figure 7, figure 8, figure 9, and figure 10 provides valuable insights into the heat transfer coefficients (convective and radiative) for PM5, PM10, and diesel fuels under different loading conditions. The data reveals interesting trends: At 25% engine load, the convective heat transfer coefficient of PM5 is slightly superior to diesel by 1.20%. However, for PM10, it is lower than diesel by 1.03% (Figure 7). At 50% engine load, both PM5 and diesel exhibit comparable convective heat transfer coefficients. However, PM10 demonstrates a 3.51% lower convective heat transfer coefficient compared to diesel (Figure 8). When the engine load is increased to 75%, the convective heat transfer coefficient for PM5 is 0.7% lower than that of diesel, while PM10 shows a significantly lower value of 4.86% compared to diesel (Figure 9). At maximum engine load (100%), both PM5 and PM10 demonstrate reduced convective heat transfer coefficients compared to diesel. PM5 is lower by 3.04% and PM10 by 2.99% (Figure 10).

These findings suggest that PM5 has a slight advantage over diesel in terms of convective heat transfer at lower loads, but diesel outperforms both PM5 and PM10 at higher engine loads. Understanding these heat transfer characteristics is crucial for optimizing engine performance and developing effective strategies to manage heat-related challenges.

At 25%, 50%, and 100% load, the maximum radiative heat transfer coefficient for PM5 exhibited reductions of 18.52%, 7.20%, and 20.91% compared to diesel, respectively. However, at 75% load, it was found to be 5.10% higher than diesel. For PM10, this value was lower than diesel by 26.62%, 24.25%, 19.92%, and 22.95% under 25%, 50%, 75%, and 100% load, respectively. It is evident from figure 6 that the "convective and

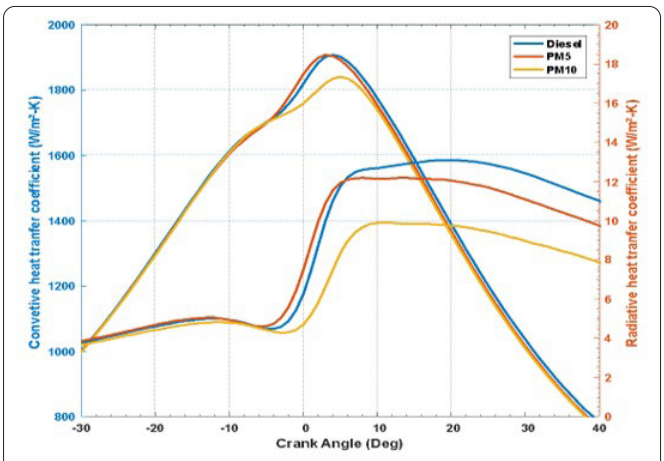


Figure 8: Convective and radiative heat transfer coefficient variation at 50% load.

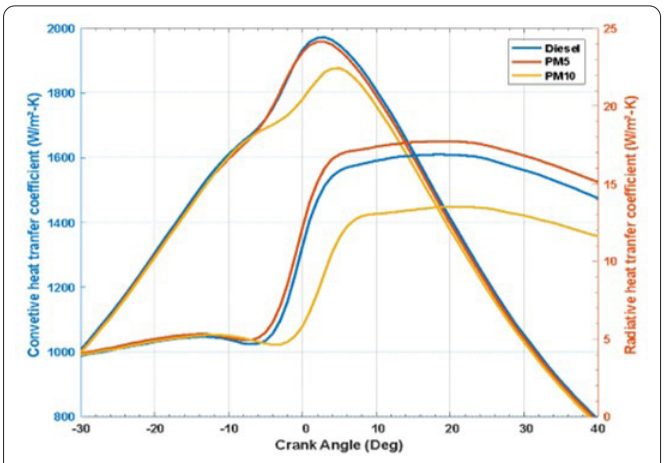


Figure 9: Convective and radiative heat transfer coefficient variation at 75% load.

radiative heat transfer coefficients" follow a similar trend as the cylinder pressure and temperature, respectively.

Blowby volume flow rate and total blowby loss

Figure 11, figure 12, figure 13, and figure 14 depict the blowby volume flow rate and total blowby loss per cycle for

PM5, PM10, and diesel under different loading conditions. The blowby volume flow rate is influenced by cylinder pressure and temperature, thus following the same trend as the cylinder temperature. From the data depicted in figure 11, it can be observed that the maximum blowby volume flow rate for PM5 displayed consistent decreases in comparison to diesel across various load conditions. Specifically, PM5 exhibited reductions of 4.09%, 1.35%, and 4.34% relative to diesel at 25%, 50%, and 100% load, respectively. However, at 75% load, it was found to be 0.96% higher than diesel. For PM10, the blowby volume flow rate was lower than diesel by 26.62%, 24.25%, 19.92%, and 22.95% under 25%, 50%, 75%, and 100% load, respectively. Figure 15 reveals that the total blowby loss per cycle for diesel was higher than both the ternary fuel blends at all loading conditions except 75% load. At 75% load, the total blowby loss for PM5 was higher than diesel. However, the to-

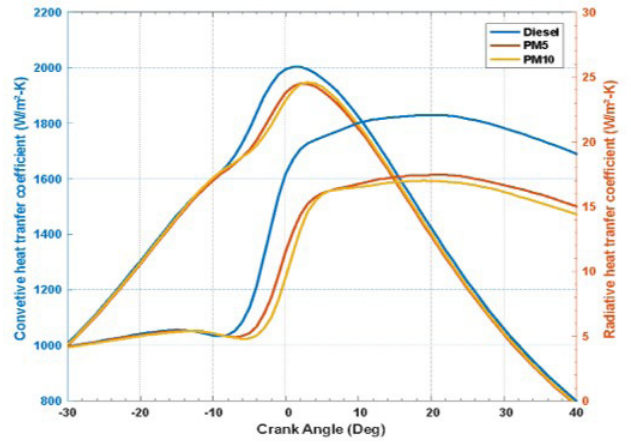


Figure 10: Convective and radiative heat transfer coefficient variation at 75% load.

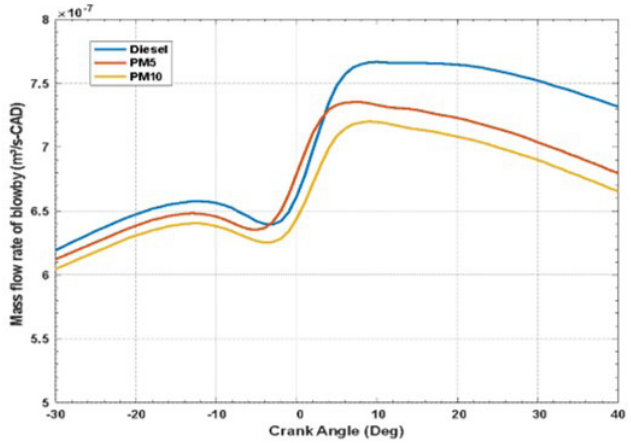


Figure 11: Blowby mass flow rate for PM5, PM10, and diesel at 25% load.

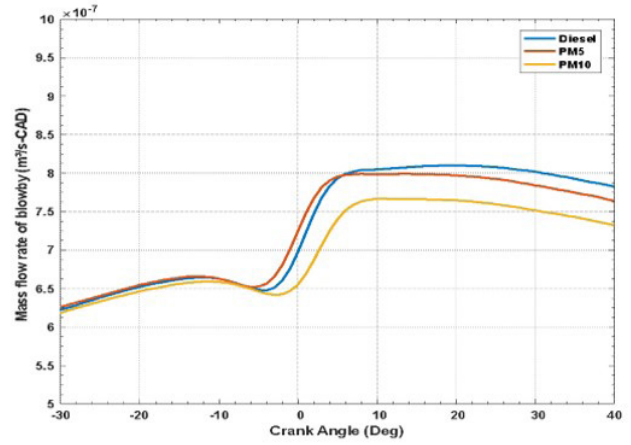


Figure 12: Blowby mass flow rate for PM5, PM10, and diesel at 50% load.

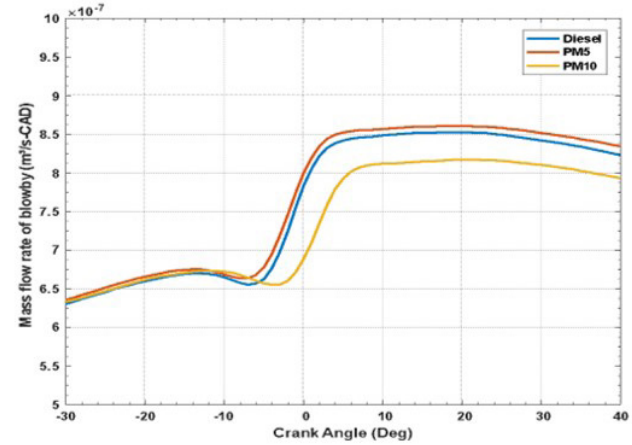


Figure 13: Blowby mass flow rate for PM5, PM10, and diesel at 75% load.

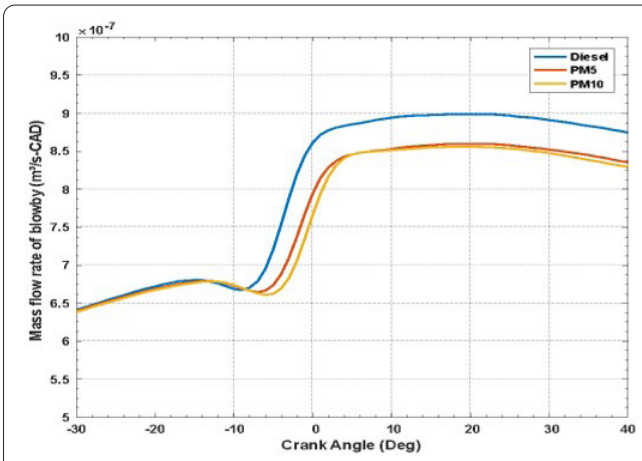


Figure 14: Blowby mass flow rate for PM5, PM10, and diesel at 100% load.

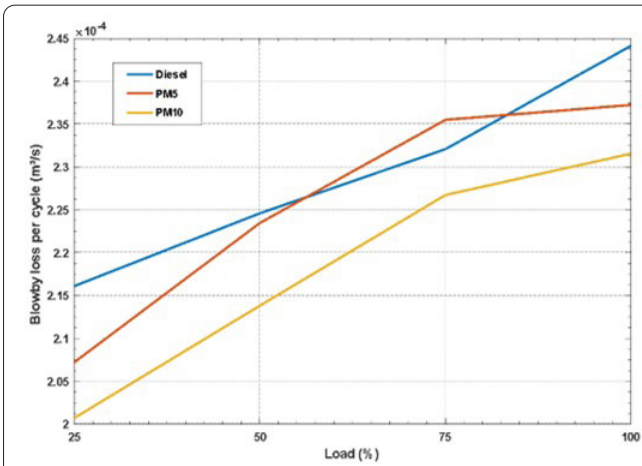


Figure 15: Blowby loss per cycle for PM5, PM10, and diesel at different loading conditions.

tal blowby loss for PM10 was lower than diesel.

Conclusions

To summarize, the experimental inquiry sought to evaluate and contrast the performance metrics of PM5 (a blend of diesel, methanol, and 1-pentanol) and PM10 (a blend of diesel and methanol) with conventional diesel fuel under different loading conditions. Several important findings were obtained from the study.

- PM5 and PM10 fuel blends exhibited lower maximum heat loss rates from the cylinder wall compared to diesel at different load conditions.
- PM5 showed reductions of 12.91% to 14.58% in maximum heat loss rate compared to diesel, while PM10 exhibited decreases of 17.66% to 21.39%.
- The convective heat transfer coefficients for PM5 were slightly elevated compared to diesel at 25% load but decreased at higher loads. In contrast, PM10 consistently demonstrated lower coefficients in comparison to diesel.
- Radiative heat transfer coefficients of PM5 were lower than diesel at most loads, except for 75% load. PM10 consistently had lower coefficients than diesel.
- Blowby volume flow rates for PM5 and PM10 were generally lower than diesel, following the trend of cylinder temperature.
- PM5 had higher total blow by loss per cycle than diesel, except at 75% load. PM10 had lower total blow by loss than diesel.

The experimental results suggest that the PM5 and PM10 fuel blends have potential advantages in terms of reduced heat loss, altered heat transfer characteristics, and improved blow by characteristics compared to conventional diesel fuel. The investigation into the impact of heat transfer in a CI engine using blends of PM5 and PM10, composed of diesel, methanol, and 1-pentanol, has yielded significant findings that can enhance engine performance and reduce emissions. Through experimental research, notable patterns and comparisons between these fuel blends and traditional diesel have emerged Additional research and optimization endeavours are imperative to comprehend the performance and combustion metrics of these fuel blends, as well as their influence on engine efficiency and emissions.

Acknowledgements

None.

Conflict of Interest

None.

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