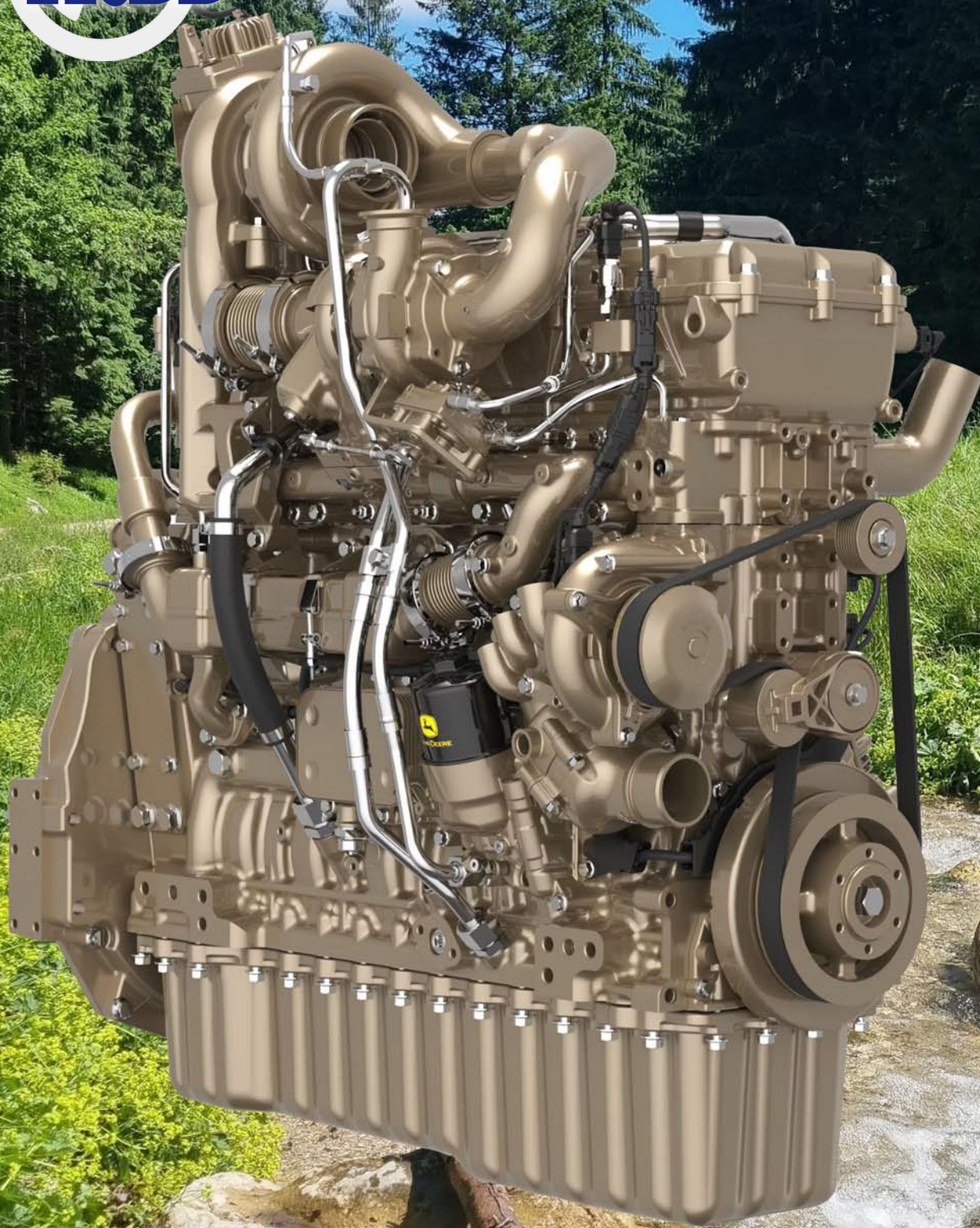




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Comparison of CO₂ emissions and costs of combustion and electric light commercial vehicles

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This paper compares CO₂ emissions, energy use, operating costs, and external climate costs of light commercial vehicles (N1-I–N1-III) with spark-ignition, compression-ignition, and battery-electric propulsion, operating under the Polish electricity mix. Fuel and electricity consumption, as well as CO₂ emissions, for ICE vehicles were obtained from the COPERT 5 model, while BEV emissions were calculated using national KOBIZE emission factors. Annual mileage was set at 30,000 km. The results show that BEVs reduce CO₂ emissions by approximately 25–50% and energy costs by 35–55% compared with combustion vehicles, with the most significant benefits in the heaviest N1-III class. External costs of CO₂ are also substantially lower for BEVs. However, the high carbon intensity of the Polish power sector limits achievable reductions, underscoring that fleet electrification must be accompanied by power system decarbonization to fully exploit its climate benefits.

Key words: *light commercial vehicles, battery electric vehicles, CO₂ emissions, operating costs, external costs*

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1. Introduction

Transforming transport systems towards low emissions is a key element in achieving the EU's climate goals. Light commercial vehicles (LCVs) play a significant role in this context, as they are heavily utilised in urban and regional logistics and account for a substantial share of carbon dioxide (CO₂) emissions from road transport. In recent years, battery-electric vehicles (BEVs) have garnered increasing attention as an alternative to internal combustion engine vehicles (ICEVs), in both the passenger car and commercial vehicle segments.

The literature emphasises that a fair comparison of BEVs and ICEVs requires a life-cycle analysis (LCA) that encompasses vehicle production, operation, and disposal. Most studies indicate that BEVs generate lower total greenhouse gas emissions than ICEVs, with the scale of this advantage depending on the energy mix, battery technology, and regional conditions [12, 18, 24, 32, 43, 50]. On the other hand, in regions with a high share of fossil fuels in electricity generation, the climate benefits of transport electrification may be limited or even reversed [13, 54].

This is particularly important for countries with a highly emission-intensive electricity sector, such as Poland. Numerous studies indicate that the dominance of coal in the national energy mix limits the potential climate benefits of electric vehicle adoption, and in specific scenarios, BEVs may have comparable or higher CO₂ emissions than modern combustion-engine vehicles [1, 28, 46, 54, 55]. At the same time, the electrification of city fleets is seen as an important tool for improving air quality and public health [30].

In Europe, the COPERT model is a commonly used tool for estimating road transport emissions, serving as the basis for national emission inventories and transport policy scenarios. Numerous analyses have also demonstrated the sensitivity of COPERT results to assumptions regarding speed, traffic structure, and fleet composition, as well as the need for local calibration [16, 37].

Against this backdrop, this study aims to compare CO₂ emissions and the associated operating and external costs for light commercial vehicles with combustion engines (Diesel, Euro 6) and electric vehicles operated within the Polish energy mix. The analysis is based on fuel and energy consumption data, CO₂ emissions from the COPERT model, and electricity emission indicators published by KOBIZE, with reference to research on the impact of the energy mix on BEV climate efficiency.

2. Literature review

Literature reviews comparing BEV and ICEV life-cycle emissions consistently indicate that electric vehicles can reduce greenhouse gas emissions by about 40–70%, and under favourable conditions up to 97%, relative to combustion vehicles [12, 14, 33, 42, 52]. The most significant advantage of BEVs is evident in operation, thanks to the higher efficiency of the propulsion system and the absence of tailpipe emissions [12, 53].

At the same time, it is emphasised that the production of electric vehicles – especially traction batteries – is associated with higher emissions during manufacturing, which may partially offset the benefits obtained during use, especially in regions with high electricity emission intensity [12]. The literature also highlights other categories of environmental impacts, such as metal depletion and human toxicity, in which BEVs often fare less favorably due to the consumption of raw materials and the production of batteries [12].

The vast majority of studies emphasise that the climate-enhancing effect of BEVs depends strongly on the emission intensity of the energy mix supplying charging energy. In regions with a high share of renewable energy or nuclear power, electric vehicles show a significant advantage over ICEVs in GHG emissions, whereas in countries with a mix of coal or other fossil fuels, the advantage may decrease or even reverse [2, 30].

Many studies analyse scenarios for transitioning from a coal-based to a low-emission mix, showing that the cli-

mate benefits of BEVs increase with the share of renewable energy sources, making transport electrification an increasingly effective tool for reducing emissions [29, 30]. Attention is also paid to the temporal and spatial variability of emissions – the carbon footprint of electric vehicles may vary not only between countries but also with time of day, grid load profile, and local generation structure [2, 29].

Analyses of EV indirect emissions from the power sector lead to similar conclusions: in fossil-fuel-dominated systems, BEV indirect emissions can be similar to or exceed those of efficient combustion vehicles [35, 49], while in low-emission energy countries, EVs achieve significant GHG reductions [5, 49].

On a European scale, well-to-wheel studies indicate that EVs generally reduce GHG emissions compared to ICEVs, but the scale of the reduction depends on the mix of individual countries and the pace of their decarbonisation [6]. In countries with a clean mix (such as France, Norway, Austria, and Switzerland), the benefits are the greatest. In contrast, in countries with a high share of coal (such as Poland and Estonia), a parallel acceleration of energy sector decarbonisation is necessary for transport electrification to achieve full climate benefits [15].

In Poland, where the energy mix remains heavily reliant on coal, the literature highlights the limited effectiveness of BEVs as a short-term tool for reducing CO₂ emissions. It is indicated that, with a dominant share of coal-fired power plants, electric vehicles can generate CO₂ emissions comparable to or higher than those of combustion vehicles, especially in higher vehicle segments and with high mileage [1, 46, 54, 55]. Simulations of scenarios for increasing the share of renewable energy sources (e.g. 25–75% in the mix) show a gradual reduction in BEV emissions relative to ICEV, reaching several dozen per cent, with a clear advantage achieved only with a high share of renewable sources [19, 21, 46, 55].

Electricity emission indicators published by KOBiZE play a crucial role in emission analyses, serving as a standard in national carbon footprint calculations. It has been shown that electricity emission intensity in Poland is among the highest in the EU, with values of ~1.06 kg CO₂-eq/kWh resulting from the dominance of hard coal and lignite in the generation structure [4, 20, 46]. These studies emphasise that improving the BEV emission balance in Poland is closely related to the decarbonisation of the electricity sector [22, 40, 44, 53].

LCVs are a significant source of CO₂ and air pollutant emissions, particularly in urban environments. The literature highlights that, despite tightening Euro emission standards, real-world NO_x and CO₂ emissions from LCVs – especially those with diesel engines – often significantly exceed the approval values [38]. A gap persists between approval emissions and measurements in real-world driving conditions, which are sensitive to driving style, traffic conditions, and ambient temperature [7].

Studies comparing BEVs and diesel LCVs in urban applications indicate that electric vehicles can reduce CO₂ emissions by 15–98% and air pollutant emissions by 7–99%, depending on the age of the vehicles being replaced, the intensity of use, and the energy mix [27, 48]. The most

significant benefits are observed when replacing older diesel vehicles and when powered by clean energy sources, while GHG emission reductions are limited in coal-based systems [27, 34].

More broadly, studies on the electrification of urban fleets (buses, delivery vans, taxis) show that switching to electric drives can reduce GHG emissions by 33–92% over a decade and bring significant health and social benefits, provided that the appropriate energy mix and infrastructure support are in place [31, 39].

COPERT is the standard tool for estimating road transport emissions in Europe and forms the basis for national GHG and air pollutant emission inventories. This methodology is based on emission factors dependent on vehicle class, technology, speed, and traffic conditions, and its use is recommended by the European Commission and the IPCC [11, 16]. For diesel LCVs, COPERT provides CO₂ emission and energy demand factors, widely used in comparative analyses of ICE vs. BEVs.

At the same time, numerous studies indicate a significant sensitivity of COPERT results to the quality of input data (fleet composition, speed structure, trip lengths) and to underestimation or overestimation of emissions under certain conditions, especially in urban traffic and at low speeds [16, 36]. Hence, it is recommended to calibrate the model using local data and use bottom-up approaches where possible [8, 25].

Comparisons of the operating costs of combustion and electric vehicles indicate that, despite the higher purchase price of BEVs, lower energy and maintenance costs may lead to a levelling or superior total cost of ownership (TCO), especially with high annual mileage and a longer operational period [3, 41]. For light commercial vehicles in urban applications, it has been shown that BEVs can achieve TCOs comparable to or lower than those of diesel LCVs, provided sufficient mileage and charging infrastructure are available [3].

In turn, the literature on the external costs of transport shows that transport emissions generate significant health and climate costs, amounting to hundreds of billions of euros per year in Europe [10]. The standard valuation methodology is the “Handbook on External Costs of Transport” and the Impact Path Approach Pathway Approach), developed, among others, in the ExternE projects [23]. Valuation of climate damage related to CO₂ emissions is crucial for comparing drive variants and serves as the basis for applying the “polluter pays” principle [26].

The contribution of this work is to combine the COPERT model emission and energy data with national energy emission indicators (KOBiZE), allowing for a clear comparison of CO₂ emissions, operating costs, and external costs for N1 light-duty vehicles under the Polish energy mix. Unlike previous analyses, actual energy demand and emissions for three LCV classes (N1-I, N1-II, N1-III) were used, rather than catalogue values, enabling a realistic assessment of the cost-effectiveness and environmental impact of BEVs and combustion vehicles.

3. Methodology – analysis of CO₂ emissions and total costs

3.1. Input data and scope of analysis

The analysis compared light commercial vehicles (LCVs) of class N1 in three subclasses: N1-I, N1-II, and N1-III, encompassing three drive types: internal combustion engine SI (Petrol), CI (Diesel), and electric (BEV). Fuel consumption for internal combustion vehicles and electricity consumption for BEVs were obtained from the COPERT 5 model, which determines a vehicle's specific energy demand based on the assumed traffic pattern and annual mileage.

The analysis was limited to carbon dioxide (CO₂) emissions because they are a key climate indicator and their values are stable, comparable, and directly available in both emission models and the literature. CO₂ emissions for BEVs were calculated using the national electricity emission index KOBIZE (kg CO₂/MWh), based on the annual energy consumption determined by COPERT.

Annual vehicle mileage was assumed to be 30,000 km, reflecting typical fleet use of light commercial vehicles in urban and regional transport, in line with values commonly adopted in European studies on LCV operation and total cost analyses. Based on this, annual fuel consumption (ICE) and electricity consumption (BEV) were calculated, followed by annual CO₂ emissions and annual energy costs.

The environmental costs of CO₂ emissions were determined according to the unit external costs recommended by the European Environment Agency (EEA), using a reference value of €25/tonne CO₂ corresponding to the “average” scenario recommended in the EEA Handbook on the External Costs of Transport. Conversion to Polish zloty was based on current exchange rates. The procedure used is consistent with the EEA's recommended methodology, ensuring comparability of results with European analyses.

3.2. Driving pattern structure

The share of journeys in road classes was adopted in accordance with the official data of the national greenhouse gas emission inventory used in the COPERT model calculations:

- 35% urban traffic
- 50% extra-urban traffic
- 15% motorways.

This structure reflects the average vehicle activity distribution used for national emissions reporting. This distribution is significant because specific fuel consumption and emissions are strongly dependent on traffic conditions – in particular, the higher energy intensity of urban driving and the more stable emission profile on rural roads. The adopted proportions are consistent with the literature on the Polish road traffic profile [53] and the operational characteristics of light commercial vehicle fleets.

3.3. CO₂ emissions of combustion engine vehicles (ICE)

CO₂ emissions for combustion engine vehicles were determined based on the results of the COPERT 5 model, which gives:

- specific CO₂ emissions [g/km] for individual road categories

- total emissions after considering the annual mileage and use structure.

The COPERT model calculates CO₂ emissions indirectly, based on fuel consumption and fuel physicochemical properties, using formulas consistent with the EMEP/EEA and IPCC methodologies. This approach ensures complete consistency with national emission inventories, enabling the comparability of results with official national and European reports.

3.4. Electric vehicle (BEV) CO₂ equivalent emissions

Since the COPERT model does not calculate emissions for electric vehicles, the equivalent CO₂ emissions for BEVs were determined using the approach described in [27]. CO₂ emissions were calculated based on specific energy consumption and the electricity emission index according to the following formula:

$$E_{CO_2} = N \cdot EF_{CO_2} \cdot M \quad (1)$$

where: N – specific electricity consumption [Wh/km], EF_{CO_2} – emission factors for electricity [g/Wh], M – annual mileage [km].

The value of the EF_{CO_2} was adopted based on the current “CO₂, SO₂, NO_x, CO, and total dust emission indices for electricity” [52] published by KOBIZE/IOŚ-PIB for the national energy mix. This indicator was converted from kg/MWh to g/Wh, enabling its direct use in the above formula.

The use of the national emission factor reflects the actual CO₂ intensity of the Polish electricity sector. It is a key element in assessing the environmental impact of BEVs in national conditions.

3.5. Energy-related operating costs

Due to the limited availability of reliable data on vehicle maintenance costs, the analysis considered only fuel costs (ICE) or electricity costs (BEV). The annual cost was determined as:

$$C_{exp} = Q \cdot C \quad (2)$$

where: Q – annual fuel or energy consumption (from COPERT), C – average price of fuel or electricity in Poland in 2024, used as annual mean values to maintain consistency with the yearly aggregation of mileage, energy consumption, and emissions.

The operating cost, defined in this way, reflects the main component of commercial vehicle fleets' operating expenses and, at the same time, ensures comparability across different drive types.

3.6. External costs of CO₂ emissions

The external cost of CO₂ emissions was calculated according to the formula:

$$C_{ext} = E_{CO_2, year} \cdot C_{CO_2} \quad (3)$$

where: $E_{CO_2, year}$ – annual CO₂ emissions [t], C_{CO_2} – unit climate cost [PLN/t].

The unit cost of CO₂ emissions was adopted in accordance with the recommendations of the Handbook on the External Costs of Transport (EEA), using the average scenario (€25/t CO₂). Values were converted to Polish zloty

using the exchange rate and conversion method used in the Central Statistical Office (GUS) reports on environmental damage costs, ensuring consistency with national economic analyses [17].

This approach enables the comparability of results with both European studies (EEA, OECD) and national studies using the environmental damage valuation method.

4. Results

4.1. Annual fuel and energy consumption

This chapter presents the results of calculations regarding energy consumption, CO₂ emissions, operating costs, and external costs for N1 light commercial vehicles. The analysis covers three vehicle subclasses (N1-I, N1-II, N1-III) and three drive types (SI, CI, and electric), in accordance with the methodology's assumptions. The results are presented in bar charts, allowing easy comparison of relationships between segments and identification of differences between combustion and electric vehicles. The following subchapters discuss energy consumption, CO₂ emissions, annual costs, and external costs of emissions.

A summary of the annual energy consumption for all N1 classes and drive types is shown in Fig. 1.

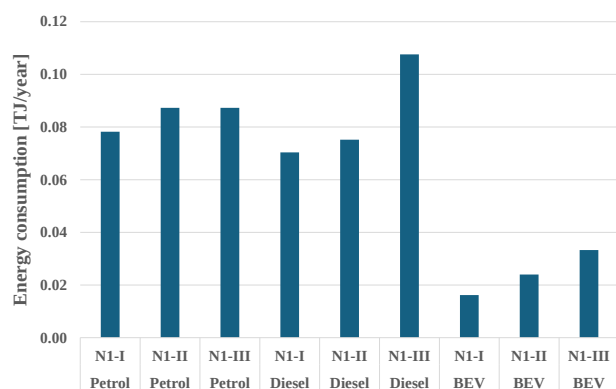


Fig. 1. Annual energy consumption of N1 light commercial vehicles with ICE SI and CI and BEV

Annual energy consumption shows a clear differentiation between drive types and N1 segments (Fig. 1). For gasoline vehicles, annual energy consumption is approximately 0.078 TJ/year (N1-I) and 0.087–0.088 TJ/year (N1-II and N1-III), indicating an increase in demand within the N1 segments.

For diesel vehicles, even more pronounced differences are observed: from approximately 0.071 TJ/year (N1-I) to 0.108 TJ/year (N1-III). The extreme increase between N1-II and N1-III is due to the greater curb weight and operational characteristics of this vehicle class.

For BEVs, annual energy consumption ranges from 0.019 to 0.034 TJ/year, increasing in proportion to the segment. However, the growth rate is significantly slower than for combustion vehicles. This is due to the higher efficiency of the electric drive and the lower dependence of energy consumption on vehicle weight.

4.2. Annual CO₂ emissions

A comparison of the annual CO₂ emissions for all vehicles analysed is shown in Fig. 2.

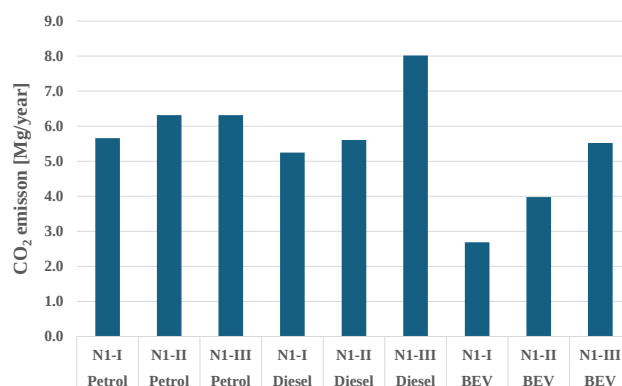


Fig. 2. Annual CO₂ emissions of N1 ICE SI and CI light commercial vehicles and BEV

Annual CO₂ emissions from combustion vehicles show an apparent increase with the transition to higher N1 segments (Fig. 2). For vehicles with a spark-ignition engine, these values are about 5.7 Mg/year (N1-I) and 6.3 Mg/year for classes N1-II and N1-III. For vehicles with a CI engine, CO₂ emissions are slightly lower for segments N1-I and N1-II (approx. 5.2–5.6 Mg/year), while in class N1-III an apparent increase to about 8 Mg/year is observed.

This means that the differences between segments are significantly larger in diesel vehicles than in petrol vehicles, due to higher fuel consumption in heavier, higher-payload vehicles.

Electric vehicle CO₂-equivalent emissions are significantly lower than those for ICE vehicles in all segments. For BEVs, they are approximately 2.7 Mg/year (N1-I), 4.0 Mg/year (N1-II), and 5.5 Mg/year (N1-III). Emissions growth between segments is due to higher electricity consumption; however, the rate of growth is slower than that of combustion vehicles.

4.3. Annual operating costs

The annual fuel and electricity costs corresponding to the analysed energy consumption are illustrated in Fig. 3.

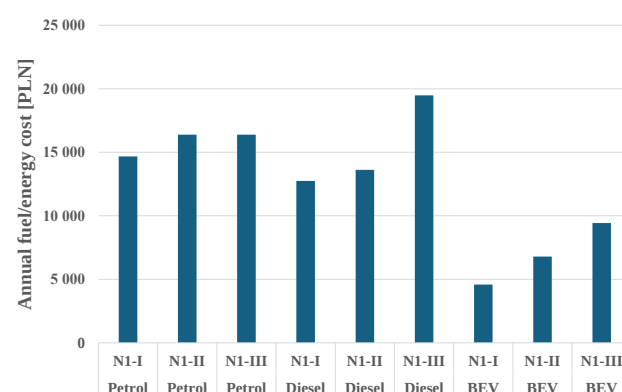


Fig. 3. Annual fuel (ICE) and electricity (BEV) costs for light commercial vehicles

Annual fuel costs for gasoline vehicles range from PLN 14,700 to PLN 16,500 per year (Fig. 3). The cost increase between segments N1-I and N1-II is significant, while the values for classes N1-II and N1-III are very similar, which results from similar fuel consumption determined by COPERT.

For diesel vehicles, operating costs are lower than those of petrol versions in the N1-I and N1-II segments (approximately PLN 12.8–13.7 thousand/year). However, in the N1-III class, they increase significantly to approximately PLN 19,500 per year. This means the segment's impact on fuel costs is much greater in diesel vehicles than in petrol units.

Annual electricity costs for BEVs are significantly lower than fuel costs for ICEs, amounting to approximately PLN 4,500 (N1-I), PLN 6,600 (N1-II), and PLN 9,500 (N1-III). The increase in energy costs across segments is proportional to the increase in energy consumption with greater vehicle weight.

4.4. External costs of CO₂ emissions

Figure 4 shows a comparison of the annual external costs of CO₂ emissions for all vehicle variants.

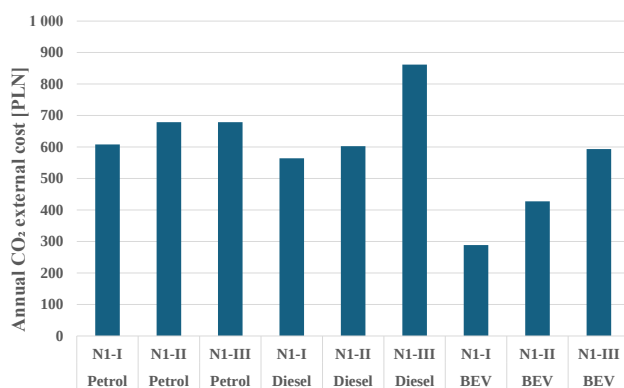


Fig. 4. Annual external costs of CO₂ emissions of light commercial vehicles N1

The external costs of CO₂ emissions for combustion vehicles increase with the N1 segment, with the strongest relationship for diesel vehicles (Fig. 4). In the petrol segment, these values are approximately PLN 600/year (N1-I) and approximately PLN 680/year for classes N1-II and N1-III, reflecting minor differences in their annual CO₂ emissions.

For diesel vehicles, climate costs are lower than for gasoline vehicles in the N1-I and N1-II segments (approximately PLN 560–595/year), but in the N1-III class, they increase significantly, reaching approximately PLN 860/year. This means the segment's impact on CO₂ emissions is particularly strong for the heaviest diesel vehicles.

External costs for BEVs are significantly lower than those for ICE vehicles in all segments, ranging from PLN 285 to PLN 595/year. The cost increase for N1-I to N1-III results from growing electricity consumption but remains significantly lower than that for combustion vehicles.

The analysis revealed apparent differences between internal combustion engines (ICEs) and electric vehicles across the three N1 segments. ICE vehicles are characterised by higher energy consumption, higher CO₂ emissions, and higher operating and external costs, with the scale of these differences increasing with vehicle weight. The increase in fuel consumption, emissions, and costs is particularly noticeable in the N1-III segment, especially for diesel vehicles.

Electric vehicles are characterised by significantly lower energy consumption and CO₂-equivalent emissions, as well as the lowest operating costs among the drive types analysed. The increase in energy and cost burdens, particularly in the N1 segment, is significantly less pronounced for BEVs than for ICE vehicles, confirming the greater energy efficiency of electric drive.

The results indicate that even with Poland's current emissions-intensive energy mix, BEVs remain more advantageous in terms of CO₂ emissions, operating costs, and environmental impact. These differences are most pronounced in the N1-III segment, highlighting the significant potential for electrification, especially in heavier vehicle classes and those with intensive fleet use.

5. Summary and discussion

The results of this analysis are consistent with observations from recent literature comparing internal combustion and electric vehicles during their operational phases. They confirm that BEVs are characterised by significantly lower energy consumption and lower CO₂-equivalent emissions than ICE vehicles, even in highly carbon-intensive energy mix conditions, such as those in Poland. According to numerous studies [46, 47, 55], the advantage of BEVs in the operational phase stems primarily from the high efficiency of the electric drive and the absence of tailpipe emissions.

Although the present analysis focuses on operational energy costs and CO₂ emissions, it should be noted that battery-electric vehicles typically have higher purchase costs than conventional light commercial vehicles. At the same time, BEVs generally require less maintenance due to the absence of components such as internal combustion engines, oil systems, and exhaust aftertreatment, which reduces both operating expenses and the environmental burden associated with vehicle servicing. These aspects are relevant for a full total cost of ownership and life-cycle assessment; however, they were intentionally excluded from the present study to maintain a clear focus on the operational phase and ensure comparability of emission and cost indicators.

However, the scale of CO₂ emission reduction in BEVs in Poland is still limited by the high share of coal in electricity production, as previously noted in KOBiZE analyses and numerous studies on the Polish power system [4, 20, 44]. Our results are consistent with the conclusions of [53, 54], who showed that BEVs in Poland outperform ICEs in terms of CO₂ emissions only to a moderate extent, and a clear advantage becomes apparent only after significant decarbonisation of the mix.

In the analysis of this paper, the BEV emission reduction relative to ICE was approximately 25–50% and was the highest in the N1-III segment, which confirms that the electrification of heavier delivery vehicles has the most significant reduction potential, similarly to what was indicated in studies on city and logistics fleets [27, 51].

The substantial increase in energy consumption and emissions for diesel vehicles in the N1-III segment is entirely consistent with previous observations on real-world LCV emissions [9, 38]. These studies indicate that the higher the vehicle mass, the greater the fuel consumption in

real-world driving conditions, especially in delivery applications with high dynamic loads.

In the case of BEVs, this dependence is milder, resulting from the higher efficiency of the drive system and the lower sensitivity of the electric system to load and driving conditions – phenomena confirmed in many comparative analyses [12].

The results on annual operating costs confirm the literature's findings that BEVs can be cheaper to operate than diesel and gasoline vehicles, especially with higher mileage and intensive use [3]. In this analysis, cost differences were significant across all segments, with BEVs generating 35–55% lower energy costs than ICEs, which is consistent with the observations by [3] that the price of electricity, even in an environment of rising energy prices, remains a relatively stable and predictable component of TCO.

The external costs of CO₂ emissions were also significantly lower for BEVs. This is a direct result of lower CO₂-equivalent emissions and of the linear climate damage pricing formula, based on EEA and OECD recommendations. The results are consistent with the literature on the impact of transport emissions on health and climate costs [35].

The analysis confirmed the usefulness of the COPERT model as a reference tool for analysing LCV emissions. At the same time, as noted by Orynych [37], Smit [45], and Dey [16], it is important to be aware that the model results are sensitive to traffic structure, speeds, and fleet parameters.

Therefore, the use of national data and Polish energy emission indicators (KOBIZE) provides significant added value to this work.

6. Conclusions

1. BEVs have lower CO₂ emissions than combustion vehicles in all N1 segments, despite the high emissions intensity of the Polish energy mix.
2. The reduction ranges from 25 to 50% and increases with the vehicle segment.
3. Energy consumption and CO₂ emissions in combustion vehicles increase strongly with vehicle weight, especially in the case of Diesel in the N1-III segment, which confirms literature observations on real LCV emissions.
4. BEVs are significantly cheaper to run – energy costs are 35–55% lower than ICE fuel costs, even at 2024 energy prices.
5. The external costs of CO₂ emissions are significantly lower in BEVs, confirming their environmental advantage and compliance with the polluter pays principle.
6. The most significant potential for emission and cost reductions is in the N1-III segment, i.e. the heaviest LCVs used in fleets – their electrification brings the most significant benefits.
7. A key factor for further reducing BEV emissions in Poland is the decarbonization of the electricity sector, as emphasised by numerous studies [44, 47, 55, 56].

Nomenclature

BEV	battery electric vehicle	ICE	internal combustion engine
CI	compression ignition	IPCC	Intergovernmental Panel on Climate Change
COPERT	Computer Programme to calculate Emissions from Road Transport	LCA	life cycle assessment
EMEP/EEA	European Monitoring and Evaluation Programme/European Environment Agency	SI	spark ignition
		TCO	total cost of ownership

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Evaluation of the applicability of low-cost 3D scanning methods for reverse engineering: a case study of a motorcycle engine chrome valve cover

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The chrome-plated valve cover of a motorcycle engine features a complex shape and a shiny surface, making it a difficult element to reverse-engineer. This paper explores the use of low-cost scanning methods and compares their accuracy. Scans were performed using a low-cost 3D scanner, as well as LIDAR and TrueDepth sensors in iPhones. The paper also discusses the method of object preparation and assesses its applicability to reconstructing parts of vehicles, such as the valve cover. The results of testing the 3D Revopoint MetroX scanner and the TrueDepth sensor on the iPhone 12 Pro were promising. Point cloud data were obtained for both an object coated with a scanning spray and a chromed one.

Key words: reverse engineering, TrueDepth, iPhone LIDAR, 3D scanning, engine component

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1. Introduction

The 3D scanning technologies began to emerge in the 1960s. They were continuously developed, but a breakthrough occurred in the 1990s. Several companies launch 3D scanners in forms similar to those we know today [10].

Due to its cost, 3D scanners were initially available only to a narrow group of companies or researchers. The further development of the technology and its increasing adoption across various industries have contributed to the widespread adoption of 3D scanning. At the current stage of technological development, a significant democratization of 3D scanning technology is observable. Manual scanners such as the Revopoint MetroX or the Creality Raptor are now available to individual customers. As manufacturers state, these scanners enable data acquisition with metrological accuracy while remaining affordable [9, 17].

Moreover, the development of mobile technologies in recent years has brought a revolutionary change to the field of 3D scanning. It has enabled the integration of sensors and technologies into smartphones.

In 2017, Apple introduced the TrueDepth facial recognition technology in iPhone X. The principle of operation is that the dot projector emits over 30,000 dots onto the surface in front of the device. The infrared camera captures reflections from these light points, enabling the creation of a surface model from the resulting pattern. To ensure accurate detection even under low-light conditions, the system is equipped with an infrared radiator that also illuminates the area in front of the camera [3, 6].

In 2020, Apple introduced a LiDAR (Light Detection and Ranging) sensor into its mobile devices, starting with the iPad Pro 2020 and then the iPhone 12 Pro and newer models. The principle of operation is that LiDAR uses Time-of-Flight (ToF) measurements, which determine the time it takes a light signal to travel a given distance. Therefore, LiDAR emits a pulse or modulated light signal and measures the time difference between the emitted signal and the returning wavefront. This innovation has opened new possibilities in mobile 3D scanning, particularly in research applications and

in industrial settings, where traditional measurement methods require expensive equipment and specialized personnel training [1, 12, 23].

The 3D scanning is widely used across various industrial sectors. It ranges from scanning large geographic areas for map creation, to applications in medicine, reverse engineering and the design of production workstations [8, 13, 19].

A wide range of research is available on the application of iPhone sensors as an alternative to traditional scanners. The iPhone LIDAR has been widely used for reconstructing buildings, detecting cracks and architectural details, and documenting cultural heritage [2, 4, 7, 25].

There are also highly specialized applications of this technology for inspection and measurement during the sinking of minshafts or for monitoring underground pipe installation sites [15, 18].

An interesting application is its use in reconstructing accidents for virtual recreation of the accident site, as well as in vibration measurement and modal analysis [21, 22].

Regarding TrueDepth technology, research primarily focuses on typical medical applications. Examples include studies on facial registration in individuals with facial nerve palsy, orthodontics, or ear reconstruction [5, 8, 14, 20].

A review reveals a very limited number of publications addressing smartphone-based scanning of regular shapes common in mechanical engineering, and there is no reference to such applications in the context of the potential application of iPhone sensors in mechanical engineering for reconstructing engine components and other vehicle parts. In the study on hydrogen injection simulation, reverse engineering methods were used to map the engine's geometry [24]. The exact scanning technique is not described, however this represents a potential application area for iPhone sensors.

The research objective was to verify the possibility of using augmented reality scanners originally employed in iPhones to reconstruct a representative engine component in a virtual environment. The studies focused on whether obtaining an accurate scan of shiny surfaces is challenging

and whether this depends on the scanning technology. The potential of using iPhone sensors for scanning an engine component approximately $230 \times 160 \times 70$ mm in size was also validated. Based on the reviewed literature, it was expected to achieve greater accuracy with the TrueDepth sensor compared to LiDAR. These studies also aimed to verify whether the acquired data were sufficiently accurate for applications in mechanical engineering.

2. Materials and methods

2.1. Research approach description

To address the formulated hypotheses, a dedicated research setup was prepared. It was established from the ground up by the authors of this study. It consisted of the following elements: a scanning device, a Mac Mini computer with an M4 processor, a portable external monitor, input devices in the form of a mouse and keyboard, a rotating table with markers applied and a device for recording illumination intensity. During scanning, a set of formers with markers was also used to support tracking.

In the context of scanning devices, the Revopoint MetroX scanner was used as the reference device. This model was chosen because it is highly suitable for scientific research, combining two scanning technologies in a single device: structured light and laser scanning. According to the manufacturer, the single-frame accuracy is as low as 0.02 mm, and the volumetric accuracy is $0.025 \text{ mm} + 0.05 \text{ mm} \times L \text{ (m)}$, where L is the measurement length in meters [17].

In this study, three modes of the Revopoint MetroX scanner were used. The first mode is structured-light-based marker tracking, and its variant relies on tracking characteristic features of the scanned objects. The other two scanning modes are the 14 crosslines mode and the 7 parallel lines mode. Both are based on laser technology and use marker-based tracking.

The research setup is presented in Fig. 1. The Revopoint MetroX 3D scanner is located on the rotating table. The Revopoint MetroX is a scanning device used as a benchmark for iPhone sensors.

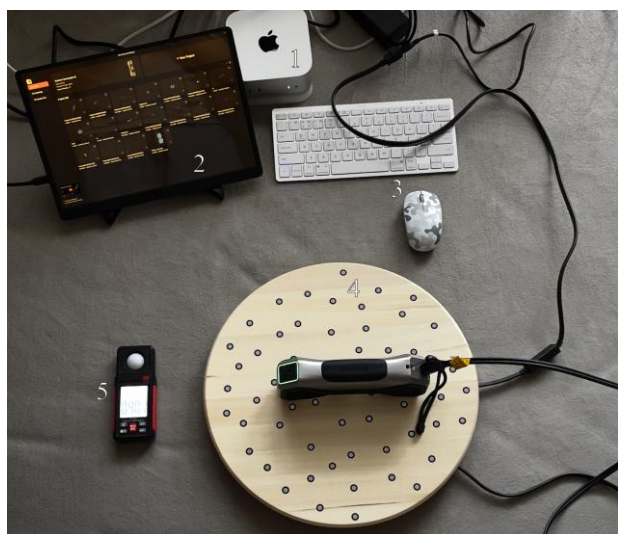


Fig. 1. Research setup equipped with a Mac mini (1), a portable external monitor (2), input devices (3), a rotating table with markers applied (4) and an environment meter Habotest HT 603 (5)

The validated devices used for research were the TrueDepth and LIDAR sensors in the iPhone 12 Pro. The iPhone 12 Pro was selected because it is the first device available to include both TrueDepth and LIDAR [3].

The environmental conditions were maintained at constant levels and recorded using the Habotest HT 603 device, as shown in Fig. 1. Throughout the study, the temperature was 21°C and the humidity was 60%. The controlled parameter rarely considered in other studies was illumination intensity, set at 40 lx.

2.2. Object

The object scanned was the valve cover of a BMW motorcycle engine. This valve cover was selected because it has a challenging surface form from a reverse-engineering point of view. A significant characteristic of this valve cover was its chrome plating. To verify the impact of the chrome-plated surface on scan quality, a scan was also performed on the object coated with AESUB scanning spray. This procedure resulted in a matte, white coating surface, whose thickness may be negligible in the analysis [11, 16].

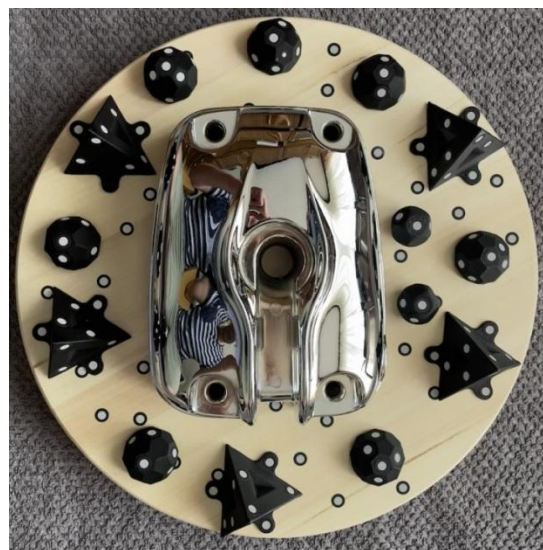


Fig. 2. Chromed valve cover lying on a rotating table



Fig. 3. Spray-coated valve cover lying on a rotating table

Figure 2 shows the valve cover in its natural state, i.e. chromed. The cover was placed on a rotating table, and around it, shape guides equipped with markers were positioned. Figure 3 presents a similar setup, but the valve cover was coated with a scanning spray. The pyramid- and dome-shaped templates, covered with markers, shown in the figures are extremely important in the scanning process, as they allow the scanner to determine the spatial positions of objects.

2.3. Methodology

The object scanned was the valve cover of a BMW motorcycle engine. This valve cover was selected for its numerous slots, holes, and grooves. A significant feature of this valve cover was its complex surface geometry. The Revo Scan 5 MetroX software version V5.6.9 was used to aggregate data from the reference scanner. This software was provided by the manufacturer together with the scanner. The scanning was performed manually, meaning the scanner was moved slowly through space without imposing any mechanical constraints. The software controlled the distance between the scanner and the object through a quality scale display. The distance between the scanner and the object always remained within the “good” or “excellent” range. As the number of frames increased, the number of acquired cloud points also increased, and the points changed color from red to orange to green. Achieving all points in the cloud with a green color indicated a successful scan of the object. All software settings were configured for maximum accuracy at the expense of performance. The goal was to achieve the highest possible data-acquisition quality under the given conditions.



Fig. 4. Scanning of the valve cover using True Depth technology. The iPhone was held in hand with the display facing the object being scanned, while the portable external monitor displayed image from the RGB camera and the TrueDepth sensor

For data acquisition from TrueDepth and LIDAR sensors, the 3D Scanner App version 2.3.6 was used. In this case, the scanning was also performed manually with the phone being moved through space without imposing any specific mechanical constraints. The application parameters were also set to maximum point cloud quality, at the expense of performance. It should be noted that the 3D Scanner App records only 15 seconds of data from the TrueDepth sensor. A photograph of the TrueDepth technology scanning process on a smartphone is presented in Fig. 4.

The four scans of the external surface of the object were performed using the Revopoint scanner (in the crosslines mode, parallel lines mode, fullfield mode – marker and feature tracking mode). The same scanning procedure was applied to the object coated with a scanning spray.

Data acquisition was also carried out using the TrueDepth and LIDAR sensors for both the naturally chromed and spray-coated surfaces for each technology.

It should be noted that during scanning, points are not only recorded from the object itself but also from the background. This situation occurs both when using a 3D scanner and when using iPhone sensors. To eliminate unwanted background points, all point clouds were processed in Revo Scan 5 MetroX. Figure 5 and Fig. 7 show examples of scans before and after the removal of background elements.

Comparisons of the point clouds were conducted using CloudCompare version 2.14 Alpha. All point clouds were compared against a reference cloud generated using the Revopoint scanner in crosslines mode, with the object coated with spray for scanning. CloudCompare is a program equipped with specialized functions enabling such comparisons and is available under an open-source license. Relative positioning of the point clouds was performed using the built-in function in CloudCompare, which minimized the positional error between points of the two compared point clouds. Automatic, software-based scale correction was also possible. It was observed that scans from the iPhone, using both the TrueDepth and LIDAR sensors, are approximately 1000 times smaller than those from MetroX. The scans from Revopoint MetroX do not require scale modification.

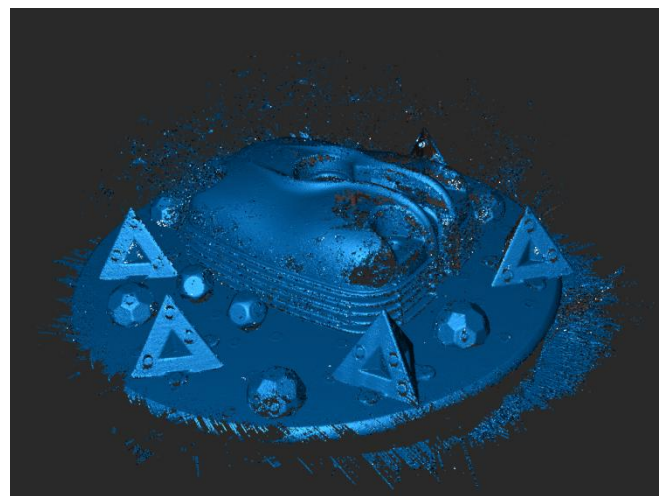


Fig. 5. Example of raw data obtained using the Revopoint MetroX scanner

3. Results

A total of 12 scans were performed. Point cloud data were successfully obtained in only 9 cases. The LiDAR mode on the iPhone and the full-field mode on the Revopoint scanner failed to capture point clouds of the chrome coating on valve components. A test was also conducted using the iPhone's sensors to scan an object rotated on a turntable. During this test, the smartphone was held stationary. The result of this test showed a fused point cloud that bore no resemblance to the original object. This suggests that tracking on the iPhone uses the device's gyroscopes. It should be noted that a similar issue was not observed in scans performed using a 3D scanner. Example scanning results are presented in the following figures.

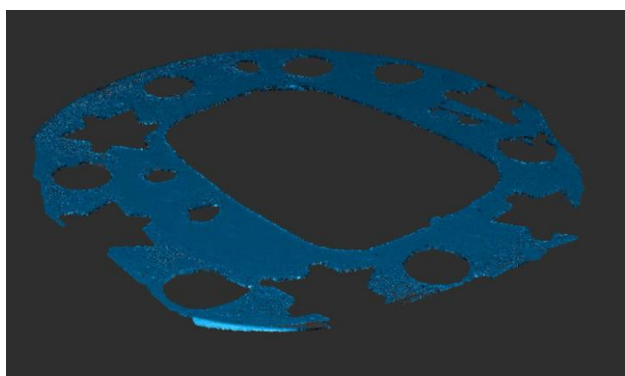


Fig. 6. The result of aggregation data using the RevoPoint MetroX scanner in full-field mode for the chromed part

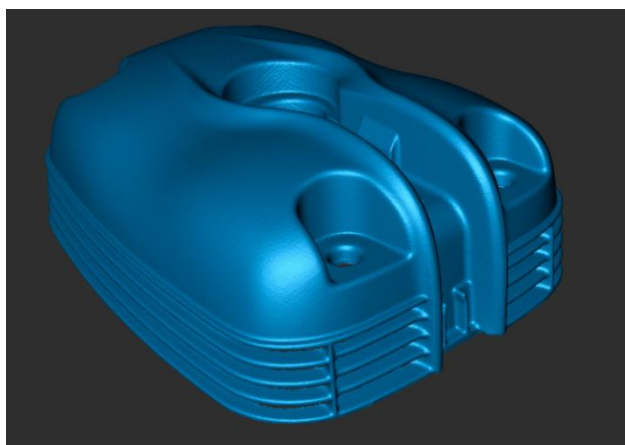


Fig. 7. The result of aggregation data using the RevoPoint MetroX scanner in crosslines mode for the spray-coated part. The point cloud shown in the figure has already been cleaned

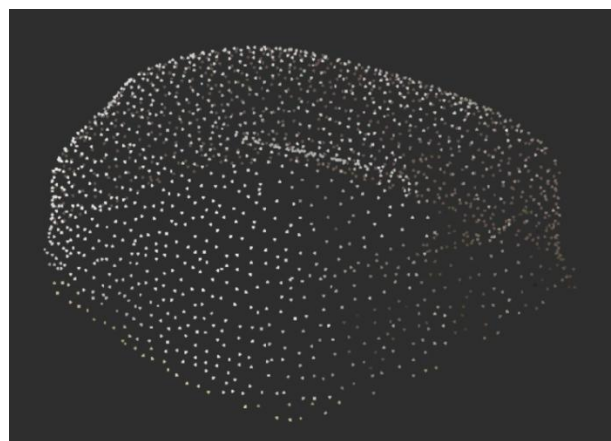


Fig. 8. The result of aggregating data using the iPhone LIDAR for the spray-coated part



Fig. 9. The result of aggregating data using the iPhone TrueDepth for the spray-coated part

Table 1. Detailed results of research for Revopoint MetroX

	CL mode (spray)	CL mode (chrome)	PL mode (chrome)	PL mode (spray)	Fullfield mode mark. trac.	Fullfield mode feat. trac.
Frames	11602	18793	20359	13878	732	766
Points	4592885	4715804	2170745	4275369	10581396	6126736
Points for cover	2297315	1659049	600244	2618915	4759835	3684869
C2C distance < 0.2 mm	N/A	92%	6%	86%	51%	80%
C2C distance < 0.5 mm	N/A	98%	14%	98%	74%	97%
C2C distance < 1 mm	N/A	99%	32%	98%	95%	98%
C2C distance < 2 mm	N/A	100%	68%	99%	100%	99%
C2C distance < 5 mm	N/A	100%	98%	99%	100%	100%

Table 2. Detailed results of research for iPhone sensors

	Crosslines (CL) mode (spray)	iPhone TrueDepth (sparay)	iPhone TrueDepth (chrome)	iPhone LIDAR (spray)
Frames	11602	—	—	—
Points	4592885	4350681	3378698	8920
Points for cover	2297315	140772	169048	1964
C2C distance < 0.2 mm	N/A	9%	9%	5%
C2C distance < 0.5 mm	N/A	22%	25%	9%
C2C distance < 1 mm	N/A	49%	50%	23%
C2C distance < 2 mm	N/A	79%	70%	40%
C2C distance < 5 mm	N/A	97%	88%	82%

The first row of the table presents the number of frames in each mode. The second row shows the number of points after performing a fusion operation, i.e. the removal of duplicate points in space. The third row displays the number of points related only to the valve cover, meaning after eliminating background point cloud data.

A key pattern observed here is that approximately 38% more frames are required to achieve sufficient scanning results for objects with shiny, reflective surfaces. Additionally, the point cloud density for the shiny surface version is about 31% less dense. This can be explained by the fact that a chromed surface scatters light, making it difficult for the 3D scanner to capture the reflected signal.

Importantly, the number of points obtained using TrueDepth technology is comparable to that of laser-based 3D scanner modes. However, after removing the background, the final number of points representing the actual geometry is several orders of magnitude smaller. This indicates that the TrueDepth sensor has a wide field of view and captures significantly more background points than a 3D laser scanner.

Furthermore, the point cloud generated by the iPhone LiDAR sensor is 3 orders of magnitude sparser than the reference point cloud, indicating that the iPhone LiDAR has significantly lower resolution.

Rows four through six of Table 1 show the percentage of point-to-point measurements between two point clouds that fall within specified error margins – 0.2 mm, 0.5 mm, and 1 mm. In Table 2, the LiDAR and TrueDepth sensor results are presented.

Additionally, the results from Tables 1 and 2 regarding the percentage of point-to-point measurements between two point clouds that fall within a specified error margin were presented in Fig. 10 for better readability.

It should be noted that only the crosslines(CL) mode of the Revopoint scanner enabled a successful scan of both shiny and matte surfaces with comparable quality. This means that the number of data gaps or “holes” in the model are acceptable and nearly 100% of point-to-point measurements fall within 1 mm. Importantly, in all tested cases, the surface coverage achieved by scanning spray improved overall scan quality.

Figures 11 and 12 show the point clouds obtained in TrueDepth and LIDAR modes compared to the reference cloud. The point clouds are coloured according to the legend on the right side of the figures. Legend presents the nearest points between two point clouds (C2C). Value is in mm.

The analysis of the values obtained from measurements using the LIDAR and TrueDepth sensors indicates that approximately 80% of the measurements fall within 2 mm for the TrueDepth system and within 5 mm for the LIDAR sensor. This means that, assuming 80% of the points are sufficient to reconstruct the object (with missing data approximated digitally), a smartphone with TrueDepth technology could reconstruct it with an uncertainty of 2 mm. A similar conclusion can be drawn for the LIDAR sensor, with the difference being that the uncertainty would be 5 mm.

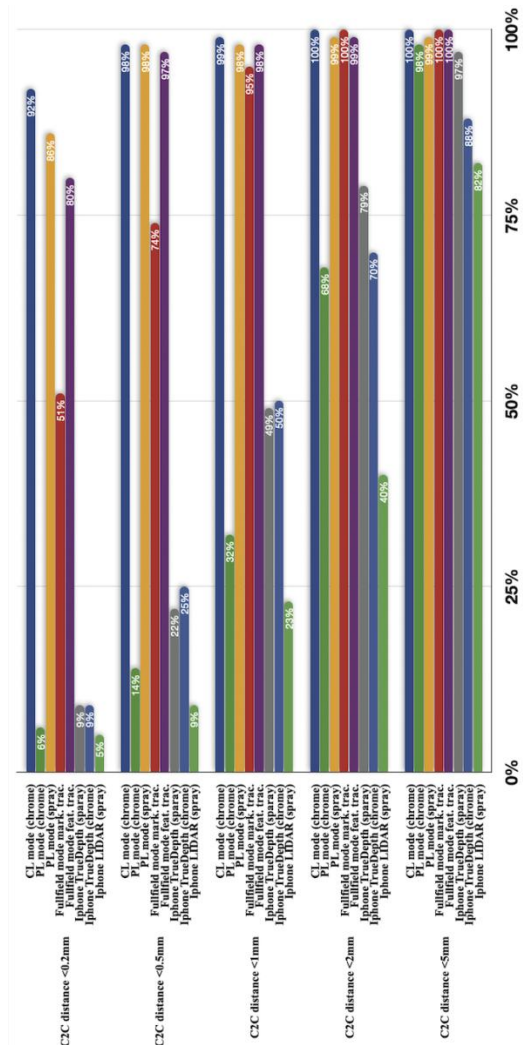


Fig. 10. The research results show cloud-to-cloud distances for each mode

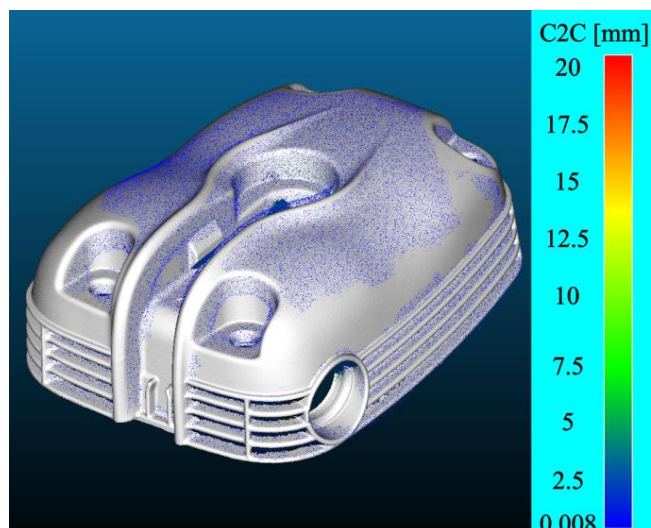


Fig. 11. Comparison of the results of scanning a sprayed-coated valve cover using the iPhone TrueDepth sensor against the reference scan obtained with the Revopoint MetroX scanner in crosslines mode

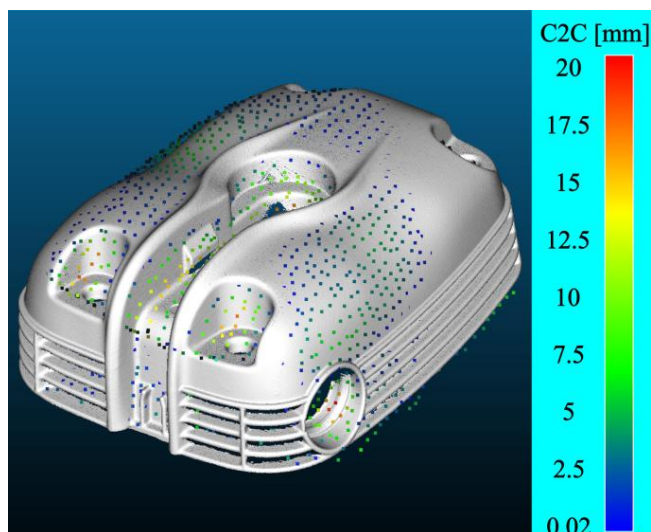


Fig. 12. Comparison of the results of scanning a sprayed-coated valve cover using the iPhone LIDAR sensor against the reference scan obtained with the Revopoint MetroX scanner in crosslines mode

However, this conclusion is not fully valid due to the low point cloud density, which will be used to mesh the model. In such a case, important features of the object may be lost – for example, the texturing on the side wall may be merged into a flat surface.

4. Conclusions

Summarizing the conducted research on the iPhone LIDAR scanner, the authors observe problems with data acquisition of small objects due to the low point cloud resolution. This limitation, combined with higher uncertainty than other methods, may contribute to the restricted applicability of this technique for reverse engineering engine parts.

Spraying during scanning of highly reflective objects significantly improves results and scan quality. The iPhone TrueDepth sensor technology enables data acquisition of both standard matte and shiny surfaces.

The data obtained from the TrueDepth sensor appears promising. Both the resolution and the accuracy of the scans seem sufficient for specific engineering applications. It is also worth noting that position tracking in the iPhone is based on its built-in gyroscope, rather than on features of the scanned object or markers. Here, it should be noted that building a device to track the iPhone's movement in space may prove essential for further research.

The authors emphasize the need for further studies. For instance, validation of the TrueDepth sensor as a tool for scanning objects used in machine construction, particularly those with sharp edges, could be valuable.

Regarding future research directions, it would be worthwhile to investigate whether combining point clouds from multiple scans performed with the TrueDepth sensor yields more accurate results, higher point cloud density, and eliminates the limitation of a 15-second data acquisition period. Another area worth exploring is the application of polarizing lenses during the scanning of highly reflective objects.

The authors believe that the topic of scanning objects using iPhones deserves further development. Advancing this topic will help mitigate the limitations encountered and enable the application of this method to a broader range of objects in the automotive industry.

Nomenclature

CL crosslines (mode in Revopoint MetroX)
PL pallarellines (mode in Revopoint MetroX)

C2C cloud to cloud (distance)

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Comparison of dust emissions from brake systems and tires of electric and conventional cars

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Environmental protection is currently receiving significant attention, particularly in the context of transport's impact on the environment. Limited fossil resources, climate change, and global warming are driving the automotive industry toward more efficient and sustainable solutions. These challenges are driving car manufacturers to adopt new technologies and alternative drive systems. Examples of such vehicles include electric vehicles (EVs) and hybrid vehicles (HEVs or PHEVs). The impact of operating these modes of transport on pollutant emissions other than exhaust gases is crucial. Examples of such emissions include particulate matter generated by brake and tire wear. All vehicles, whether conventionally or alternatively powered, generate such emissions during operation, regardless of their drive type. These particulate matter enter the air and can pose a threat to the environment and human health. While electric cars may emit less particulate matter from their braking systems due to frequent recuperation, tires remain a significant source of emissions. The article included measurements of dust emissions during the operation of an electric vehicle and a conventionally driven vehicle, as well as studies of the elemental composition of particles using scanning electron microscopy, analyzing dust collected from the vehicle's surroundings, braking system and tires.

Key words: dust emissions, brake dust, tire dust, electric cars, Euro 7

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1. Introduction

One of the most significant air pollution hazards posed by road transport operations is dust emissions. The main sources of dust during vehicle operation are braking systems, tires, and clutch discs. Dust generated during vehicle operation is typically divided into two types: fine-grained and coarse-grained dust. Fine-grained dust is defined as small solid particles with a diameter below 50 μm . Coarse-grained dust is defined as solid particles with a diameter above 50 μm . Furthermore, the limiting size does not clearly indicate whether dust is released. Sometimes, larger-diameter dust is released into the air, while smaller-diameter dust falls to the ground. This is influenced by various factors, such as precipitation, the chemical composition, and the physical properties of the dust. Both coarse and fine-grained dust are generated during vehicle operation. Most of these are fine-grained particulate matter, namely PM_{2.5} and PM₁₀ [4, 16, 17, 24, 25].

The term PM_{2.5} refers to particulate matter with a particle diameter below 2.5 μm . Soot, among other things, has this diameter. PM₁₀ particulate matter is particulate matter with a particle diameter below 10 μm . By definition, it includes PM_{2.5} particulate matter, as well as larger particles with diameters ranging from 2.5 to 10 μm . These larger particles are largely composed of mineral dust. PM₁₀ and PM_{2.5} particulate matter are not a homogeneous group of harmful substances. They can include dust particles, sand, and even pollen. Particulate matter generated by motor vehicles primarily consists of soot, ash, and heavy metals emitted from the engine. Furthermore, tire wear, brake discs and pads, and road surfaces also contribute significantly to particulate matter formation. Dust generated by these pollutants accumulates on the road surface and is then thrown into the air by passing vehicles and wind. This is called secondary dust. The dust produced depends on many fac-

tors, including the vehicle's braking force (defined by the brake pad's contact force with the disc), the vehicle's speed, and the materials from which the disc and brake pad are made. These factors correspond to different road conditions, i.e., urban or highway [1, 14, 15, 22, 23].

2. Non-exhaust particulate emissions in road transport

Currently, many companies on the market produce brake components and tires. This explains the differences in the materials used, and systematic research and experimentation is leading to the introduction of new materials, including various composite materials. However, this makes it difficult to identify brake dust from the overall road dust and thus determine the impact of operational factors on pollution levels. Identification difficulties are further compounded by differences in the materials used in individual vehicle types, such as passenger cars and trucks. However, to accurately determine this, a chemical analysis of the dust must be performed to assess the content of elements characteristic of the friction materials in vehicle brakes. Studies suggest that particulate matter emissions from braking systems can range from several to several dozen mg/km. It has been reported that average particulate matter emissions from brakes are around 4-6 mg/km, but in intensive use or in heavy vehicles, this value can be significantly higher. Studies show that the wear of a single 250/40 R18 summer tire over 15,000 km is 130 grams per 1000 km, and a winter tire 139 g/1000 km (averaged ADAC test results). From installation to replacement, average tire wear reaches 2 kilograms. Eighty percent of tire wear products are between 100 and 500 μm in size [2, 7, 9, 13, 24, 27].

The Euro 7 regulation establishes regulations for exhaust emissions from road vehicles, as well as other types of emissions, for example, those produced by tire abrasion

and braking. For passenger cars and vans, the regulation maintains the existing Euro 6 emission limits but introduces stricter requirements for PM10 particulate matter. The limits are 3 mg/km for pure electric vehicles and 7 mg/km for other-drive vehicles (internal combustion and hybrid). The new standard was initially expected to be introduced between 2025 and 2026, but the plan was postponed to 2029 [6, 18, 20, 22].

3. Dust from the use of vehicles

Particulate matter emissions largely depend on driving style, speed, vehicle weight, and the type and composition of tires and friction components in the braking system. Tire wear occurs during tire-road interaction, meaning it's constant while driving. However, greater wear occurs during starting, accelerating, and braking than during constant-speed driving. During braking with a traditional friction braking system, kinetic energy is converted into heat resulting from friction. This process causes wear of the interacting components, leading to the formation of friction products [11, 12, 20, 28].

When rolling freely on a road, a tire experiences elastic deformation. The coefficient of friction associated with rolling a tire on the road is approximately $\mu=0.5$. If a tire rolls freely on a smooth surface, wear is not significant. When a vehicle accelerates, decelerates, or changes direction, slight slippage occurs between the tire and the road. Braking is one of the processes in which high friction is beneficial. The coefficient of friction between brake pads and discs is $\mu = 0.4$ for the front axle (engine) and $\mu = 0.3$ for the rear axle. Brake pads consist of abrasive materials, binders, fillers, and reinforcing components. A large number of chemical compounds are used to produce each of these materials, making their characterization difficult. Based on the components used, brake pads are classified as [2, 12, 20, 23]:

- non-asbestos organic (NAO), consisting of organic compounds, mineral fibers, and graphite
- low-metallic (LM), consisting of metallic and organic compounds
- semi-metallic (SM), containing steel and iron.

The main elements in brake pad composition, according to the literature, are iron (Fe), barium (Ba), antimony (Sb), tin (Sn), and copper (Cu). In recent years, a decrease in the content of antimony and copper has been observed. European law prohibits the use of asbestos in the production of brake pads. Furthermore, the use of copper in the production of brake pads is prohibited in the US due to the risk of kidney and liver damage, anemia, allergic alveolitis, and even cancer [2, 12, 18, 23].

4. Dust from the use of electric vehicles

The torque of an electric motor is available from zero rpm, which can increase tire wear and, in turn, the amount of dust generated by operating this type of vehicle. Vehicle weight is also a significant factor in increasing dust emissions. Due to the weight of the batteries, electric vehicles can emit more tire dust than their combustion-powered counterparts, which meet current emission standards. This, in turn, can translate into higher total dust emissions from electric vehicles in some cases. However, it is claimed that

electric and hybrid vehicles, compared to combustion vehicles, emit less dust from worn brake linings thanks to the RBS (Regenerative Braking System). In this system, regenerative braking recovers energy during braking and converts kinetic energy into electrical energy without friction. The electric motors then act as generators. The generated electricity is stored in batteries. This system extends the life of the disc brake system by allowing its components to wear out more slowly. According to The Platform for Electro-Mobility, RBS reduces brake wear by 25–50% [11, 13, 20, 23].

According to a 2020 OECD (Organization for Economic Cooperation and Development) report, a survey was conducted regarding non-exhaust emissions. Respondents assumed that the increased weight of electric vehicles leads to increased tire wear. A comparison of mid-range electric vehicles showed that their weight is 22% greater than that of combustion vehicles in the same class. The same applies to the opinion that the torque in electric vehicles, which is greater and available from zero engine speed, leads to frequent traction loss during take-off and acceleration, and therefore to higher tire particulate emissions [11, 14, 20, 23].

4. Research object

The research object is three passenger cars. A conventional drive car of the Dacia Sandero III brand with a 1.0 TCe petrol engine with a turbocharger with an output of 67 kW. An electric car of the Dacia Spring brand with a synchronous motor with a permanent magnet with an output of 33 kW. A hybrid drive car of the Toyota C-HR brand with a 1.8 internal combustion engine with an output of 72 kW and an electric motor with an output of 53 kW. The total power of the hybrid system is 90 kW. All vehicles were manufactured in 2021 and had Crossover bodies. The cheapest Dacia vehicles were selected for the research. However, this brand does not offer a hybrid car, which is why it was decided to compare it with a Toyota hybrid.

a) Conventional drive car



Fig. 1. Conventional drive car Dacia Sandero

Table 1. Technical data of the Dacia Sandero [6]

Model	Dacia Sandero III Hatchback
Drive	Conventional
Engine	1.0 TCe petrol turbocharged
Displacement	999 cm ³
Engine power	67 kW, 90 HP for 4600 rpm
Torque	160 Nm for 2100–3750 rpm
Drive	Front axle
Range	962 km
Current weight	1152 kg
Year of production	2021

b) Electric car



Fig. 2. Electric drive car Dacia Spring

Table 2. Technical data of the Dacia Spring [5]

Model	Dacia Spring Crossover Electric
Drive	Electric
Engine	Permanent magnet synchronous
Engine power	33 kW, 45 HP for 3000–8200 rpm
Torque	125 Nm for 500–2500 rpm
Drive	Front axle
Battery	lithium-ion
Gross battery capacity	27 kWh
Charging time socket/fast	13 h/1 h
Range	230 km
Own weight	920 kg
Year of production	2021

c) Hybrid car



Fig. 3. Toyota C-HR hybrid car – view under the engine cover

Table 3. Technical data of the Toyota C-HR [26]

Model	Toyota C-HR Crossover SUV – 5d
Drive	Hybrid (HEV)
Engine	1.8 Hybrid
Engine system power	90 kW, 122 HP @ 5200 rpm
Petrol engine power	72 kW, 98 HP @ 3600 rpm
Petrol engine torque	142 Nm @ 3600 rpm
Electric engine power	53 kW, 72 HP
Electric engine torque	163 Nm
Battery	lithium-ion
Gross battery capacity	1 kWh
Gearbox type	Automatic e-CVT
Fuel consumption	4.8 dm ³ /100 km
Fuel tank capacity	43 dm ³
Range	895 km
Current weight	1380 kg
Year of production	2021

5. Research

5.1. Wear of brake pads and discs

In this study, information on brake pad and disc thickness, as well as tread depth, was used to compare the amount of particulate matter generated by electric and conventionally powered vehicles. Data were obtained from authorized service stations for the selected vehicle brands

tested. The comparison included both electric, conventional, and hybrid vehicles. Front axle brake wear was accounted for, as the rear axle brakes were of different designs (disc or drum). The vehicles analyzed were manufactured in 2021, and the tests were conducted during a scheduled maintenance service at the end of their second year of use. To accurately analyze the results, calculations were performed to account for the different vehicle mileages used in testing. A mileage of 15,000 km was selected as a representative example.

The test results regarding the wear of braking system components are shown in Fig. 4 and 5.

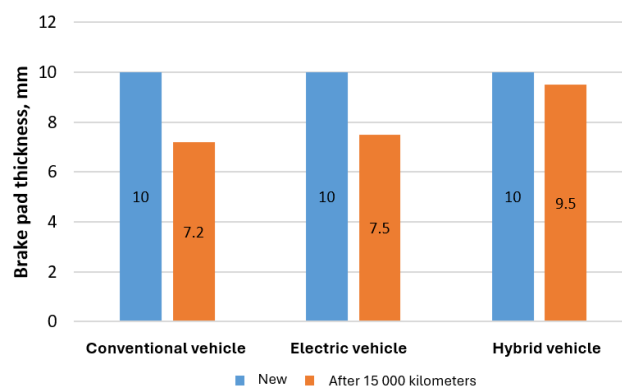


Fig. 4. Brake pad wear after 15,000 kilometers

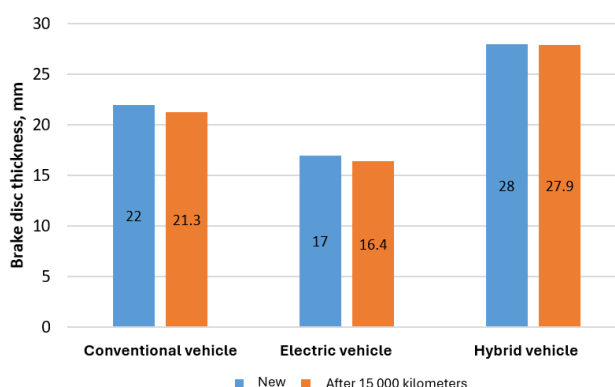


Fig. 5. Brake disc wear after 15,000 kilometers

In a conventionally powered car, after 15,000 kilometers, the front brake pads had worn 2.8 mm, and the front brake discs had worn 0.7 mm. In an electric car, these wear rates were 2.5 mm and 0.6 mm, respectively. In a hybrid vehicle, the front brake pads had worn 0.5 mm, and the front brake discs had worn 0.1 mm.

Based on this data, the volume of dust generated by wear was calculated. The density of the friction material used in brake pads is typically around 2.3–3.2 g/cm³, depending on the mix (resins, mineral fibers, metals). Many manufacturers' materials have a density of around 2.7 g/cm³ [8]. The density of the material used for the brake discs (cast iron) was determined to be 7.3 g/cm³ (gray cast iron) [18, 28]. The mass of dust generated by vehicle operation over 15,000 kilometers was calculated [18].

For 1 kilometer of distance traveled, the dust emissions generated by the operation of various means of transport in

mixed conditions (city and highway traffic), expressed in milligrams, are presented in Fig. 6.

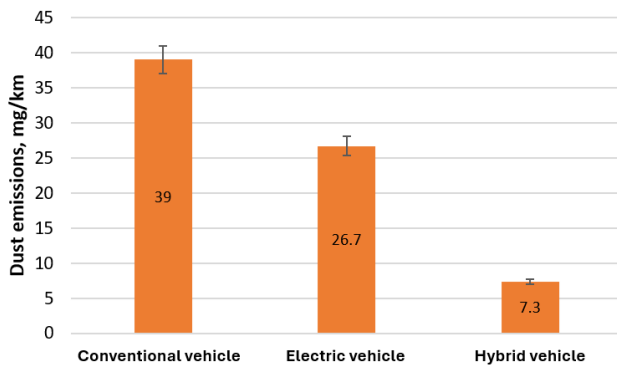


Fig. 6. Dust emission from brake pads and discs per 1 km

Total particulate matter emissions from the braking system (including brake pads and discs) were as follows: 39 mg/km for the conventional vehicle, 26.7 mg/km for the electric vehicle, and 7.3 mg/km for the hybrid vehicle. In the context of particulate matter emissions, it is worth noting that the Euro 7 standard is planned, which will set maximum particulate matter emissions from vehicle braking systems at 0.7 mg/km for conventional and hybrid vehicles and 0.3 mg/km for electric vehicles. Taking the above calculations into account, particulate matter emissions from the vehicles selected for testing exceed those specified by the Euro 7 standard, even when the front axle braking system is included [6, 18, 23].

5.2. Tire wear

Next, tire tread depth was measured on the test vehicles. Tire tread wear after 15,000 kilometers is shown in Fig. 7.

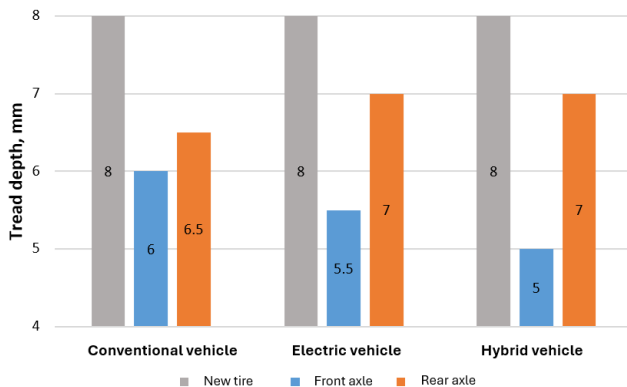


Fig. 7. Tire wear after 15,000 kilometers

The new tires have a tread depth of approximately 8 mm. Tire wear was measured after 15,000 km. The vehicles showed increased brake wear on the front (drive) axle. In the conventionally powered vehicle, the tire tread on both axles wore down by 3.5 mm. In the electric vehicle, wear was also 3.5 mm, but more intense on the front axle. In the hybrid vehicle, total tread wear reached 4 mm.

Due to the tread structure, 80% of the calculated values were used. To calculate the mass of dust generated from tire abrasion, an average tire material density of 1.3 g/cm³ was assumed. Figure 8 shows tire dust emissions over 1 km [18].

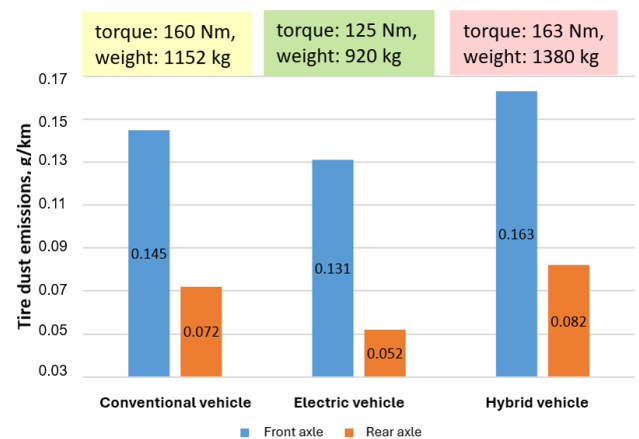


Fig. 8. Dust emission from tires per 1 km

Particulate matter emissions from tire wear on the front axle for the tested vehicles were: 0.163 g/km for the hybrid vehicle, 0.145 g/km for the conventional vehicle, and 0.131 g/km for the electric vehicle. For the rear axle, they were: 0.082 g/km for the hybrid vehicle, 0.072 g/km for the conventional vehicle, and 0.052 g/km for the electric vehicle.

It is worth noting that tire wear is a normal phenomenon and occurs during vehicle use. However, the intensity of this wear varies and depends on factors such as: the torque generated by the drive unit (e.g., engine type – electric, combustion), vehicle weight (tire-road interaction; braking), and traffic conditions (urban and rural). Tire wear is proportional to the torque and weight of the vehicle. The greater the torque and weight of the vehicle, the greater the tire wear [23].

5.3. Chemical composition of dust from vehicle

The next stage of the study involved collecting dust samples from air filters and comparing their chemical composition with wear products from the braking system and tires. The aim of this study was to determine the proportion of dust from worn brakes and tires among the total dust in the environment. Dust samples were analyzed with a Hitachi S-4200 scanning electron microscope equipped with a secondary electron (SE) detector. Chemical composition was determined using a Thermo Scientific Nova 600 energy-dispersive X-ray spectrometer (EDS) at an electron beam acceleration voltage of 15 kV. This spectrometer is coupled to a Hitachi S-4200 microscope.

Dust samples for testing were collected from the air filters of various vehicles operating in both urban and highway traffic. This method of obtaining the test material was chosen because air filters made of cellulose (paper), plastic nonwoven fabric, or a combination of these materials retain almost all particulate matter present in the air. They prevent up to 99.98% of harmful dust and soot particles, as well as tire and brake wear particles. The most precise filters retain almost all particles of environmental pollutants larger than 3 μm. Currently, an air filter permeability of up to two micrometers is considered optimal for passenger cars. It should also be noted that very fine dust is the most harmful. Particles of such a small diameter can be transported considerable distances and are also harmful to the human respiratory system. Therefore, it is important to collect test

material with the smallest possible diameter. To compare the chemical composition of the elements found in the collected dust, it was also decided to collect samples from the braking system and tires. The braking system samples were taken from holes drilled into the brake disc. Collecting dust from this location primarily contains wear products from the brake pad and disc. Tire fragments were also collected for comparison. Figure 9 shows the test samples.



Fig. 9. Dust samples: a), b) passenger car tire sample, c) dust sample collected from the air filter, d) dust sample from the braking system

The study's material was collected from four samples. The dust samples were placed on adhesive-backed supports. The results of the microanalysis of the chemical composition of the samples, along with a quantitative table of elemental composition, are presented below (Fig. 10–12).

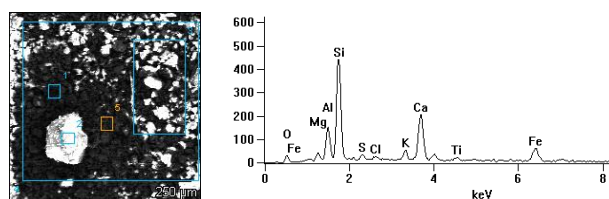


Fig. 10. X-ray microanalysis of the chemical composition of the EDS sample of dust collected from air filters a) with marked areas, b) radiation spectrum

The chemical composition results obtained from dust collected with the air filter indicate the highest concentrations of elements, including 30.5% silicon (Si), 19.5% calcium (Ca), 16.2% iron (Fe), and 9.8% aluminum (Al). Magnesium (Mg) was also detected at 2.3%, and sulfur (S) at 2%.

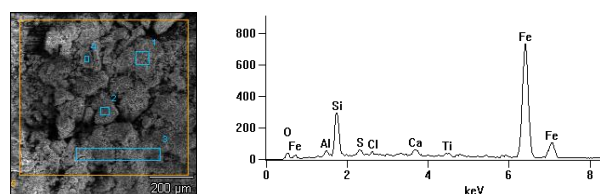


Fig. 11. X-ray microanalysis of the chemical composition of the EDS dust sample collected from the braking system, a) with marked areas, b) radiation spectrum

EDS X-ray microanalysis of the chemical composition of a dust sample collected from the braking system revealed that the dominant element throughout the sample area was iron (Fe) at a level of 84.4%. Silicon (Si) was also present at 5.7%, along with trace amounts of other elements, such as calcium (Ca) at 2.4%, titanium (Ti) at 1.8%, and aluminum (Al) at 1.1%. Brake discs are typically composed of pearlitic gray cast iron, and in some cases, steel, as confirmed by chemical composition analysis.

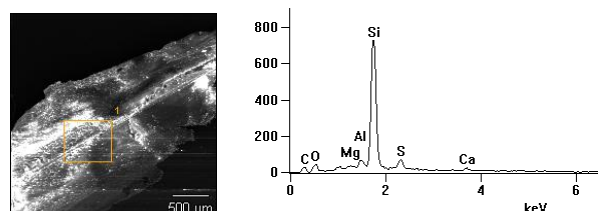


Fig. 12. EDS X-ray microanalysis of the chemical composition of a tire sample, a) with marked areas, b) radiation spectrum

A study of tire wear products revealed that the main elements constituting the tire are silicon (Si) – 39.3%, carbon (C) – 29.3% as carbon black, and sulfur (S) – 4.2%. Aluminum (Al), calcium (Ca), and magnesium (Mg) were also detected. Tire wear particles consist of plasticizers, oils, polymers, carbon black, and minerals.

6. Conclusions

Brake and tire wear is a complex process influenced by both vehicle design features, such as the drive type and materials used, and operational factors, including driving style and road conditions. In the context of the planned Euro 7 standard, particulate matter emissions from brake and tire wear are particularly important, exceeding the expected limits in many of the vehicles tested [18, 23].

The analyses confirmed that regenerative braking significantly reduces wear on the braking system's friction components, particularly in hybrid and electric vehicles. Despite this, particulate matter emissions from the braking system exceeded the limits stipulated by the future Euro 7 standard in all the vehicles tested [27]. The quality of the components used plays a significant role here – a Toyota hybrid vehicle equipped with premium braking components and operated under uniform conditions by a single driver came closest to meeting the standard. In contrast, Dacia vehicles using lower-quality components and used by different drivers in varying conditions exhibited higher wear and higher particulate matter emissions.

Tire wear analysis revealed that particulate matter emissions from this source are strongly dependent on vehicle weight and torque, tire size and class, road surface type, weather conditions, and topography. Electric and hybrid vehicles, due to their high torque available from zero speed and the typically greater mass resulting from battery packs, may exhibit more severe tire wear than conventional vehicles. However, it should be emphasized that in the case of the analyzed electric vehicle, its mass was lower than that of the conventional vehicle, which was a deviation from the general trend.

X-ray EDS microanalysis of the particulate matter chemical composition confirmed its source. In the brake dust samples, the dominant element was iron (Fe), indicating wear of the brake discs and pads. Tire wear products were characterized by the presence of silicon (Si), carbon (C) in the form of soot, and sulfur (S). Dust samples collected from air filters showed a high silicon content, which may indicate the presence of natural silica (e.g., mineral dust) and particles derived from tire wear. A significant finding is the lack of detection of heavy metals, including copper, lead, cadmium, and nickel, which are considered particularly harmful to health.

Summarizing the results of the literature analysis and our own research, it can be concluded that total particulate matter emissions from electric and hybrid vehicles may be higher than from conventionally powered vehicles, primarily due to more intense tire wear. At the same time, the level of these emissions depends largely on vehicle weight, torque, the quality of the materials used, and operating conditions, which indicates the need for a comprehensive

approach to assessing the environmental impact of different drivetrain types.

Research results also indicate that a significant factor influencing the intensity of friction element wear is the vehicle's class and the quality of its components. High-end vehicles, equipped with more advanced braking systems and materials with better tribological properties, may exhibit lower wear rates and lower particulate emissions compared to budget vehicles. These differences result not only from the quality of the components themselves but also from different brake system designs, longer-lasting friction materials, and more advanced braking support systems.

In the analyzed cases, the high-end vehicle achieved results closer to meeting future emission limits, while the budget vehicles exhibited significantly higher wear of wear components. This indicates that comparisons of particulate emissions across different drive types should also account for the vehicle segment, as design and quality differences can significantly affect final test results.

Nomenclature

EDS energy dispersive spectroscopy
SE secondary electrons

PM particulate matter
TRWP tire and road wear particles

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Analysis of fuel economy indicators of a wheeled tractor during transportation of partially filled tankers

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This study examines the fuel performance characteristics of a wheeled tractor transporting a trailer tank under varying load conditions within standardized driving cycles (ELR, EPA, and NRTC). A mathematical model of the tractor-trailer system was developed in MATLAB/Simulink, incorporating the dynamics of the FPT NEF 67 internal combustion engine, the transmission and wheel drives, as well as the effects of fluid redistribution in the tank. The model enables evaluation of instantaneous fuel consumption, specific fuel consumption, effective power, and engine efficiency under dynamic operating modes. Particular attention is given to the impact of fluid sloshing on energy expenditure, which is most pronounced at a tank fill height of 1.3 m (out of 1.6 m), corresponding to resonance conditions. The results demonstrate that the dynamic characteristics of the driving cycles have a significant impact on additional fuel consumption: NRTC yields the highest relative increase (9.22%), EPA the highest absolute increase (6.76%), and ELR the lowest (4.72%). These findings provide a basis for optimizing tractor operation during liquid cargo transport and for assessing the potential benefits of hybridizing tractor transmission systems in transport applications.

Key words: tractor, engine, transport work, tank, free surface of liquid, fuel consumption, fuel efficiency, driving cycles, simulation

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1. Introduction

Transport operations constitute an important component of the operational load of wheeled tractors in the agro-industrial complex. Within the overall cycle of agricultural activities, a tractor performs not only traction-based soil cultivation and operation with tillage implements, but also a significant volume of intra-farm and inter-farm transportation tasks. These include the collection and transportation of grain, root crops, and green fodder; the delivery of fertilizers, spare parts, and tools; and the organization of logistical operations during the harvesting period. According to field studies and technical surveys conducted across various countries, the share of transport operations in tractors' annual operating time can be substantial and varies by region, farm type, and machinery structure. In a number of analyses, transport operations account for approximately 10–60% of annual engine operating hours for certain tractor categories (low-power and general-purpose), especially in regions with fragmented fields and intensive intra-farm logistics. The use of agricultural tractors for road transport may result in significantly higher fuel consumption than conventional transport vehicles, indicating the need to analyze their fuel efficiency in transport operations [13].

According to data reported in [1], agricultural regions of Europe characterized by small and medium-sized farms allocate about 30–60% of annual tractor operating hours to transport operations. This is explained by the fact that many European studies of tractor duty cycles indicate a high proportion of travel, idling, and short-distance transport between fields in daily operation, particularly in areas with fragmented land use [6]. In regions with smallholder farming systems in China, the share of transport operations is approximately 20–50%; in particular, in southern and eastern provinces, this share is higher due to frequent transpor-

tation between small fields and local storage facilities, whereas in large agro-industrial complexes of northern and northeastern China, the proportion of transport cycles is generally lower. A review of mechanization and tractor electrification trends in China [20] confirms significant variability in tractor usage profiles. In the United States, transport operations account for approximately 10–30% of total tractor operating hours [17]. This situation is explained by the fact that many transport functions – such as long-distance logistics and transportation between fields – are performed by specialized trucks and truck combinations, while tractors are more frequently used for field operations with prolonged continuous loads (tillage, planting, harvesting). As a result, the share of purely transport-related operating hours is typically lower than in small European farms.

Transport operations performed by wheeled tractors include all activities [11] related to the movement of cargo, raw materials, supplies, machinery, or auxiliary equipment within a farm or beyond its boundaries. These operations involve delivering seeds to seeders, transporting mineral and organic fertilizers, plant protection products, and water for spraying, and supplying feed in livestock complexes. In addition, tractors are used to transport harvested crops – grain, root crops, and green biomass – from the harvester to the field edge or temporary storage, and subsequently to elevators, warehouses, or processing facilities. Transport operations also include technological transport within tractor-trailer combinations [1, 8, 12, 18], such as towing fertilizer spreader trailers, feed distribution wagons, or grain transfer carts during harvesting. Furthermore, tractors perform auxiliary logistical functions, including the transportation of fuel, technical fluids, spare parts, and tools, the delivery of maintenance crews, or the recovery of disabled machinery. With the increasing volume of intra-farm trans-

portation, the intensification of agricultural production, the expansion of cultivated areas, and the spatial dispersion of land plots, the role of transport operations in the overall technical load structure of tractors continues to grow.

2. Literature review

One of the most common types of transport operations performed by wheeled tractors in agriculture is the transportation of liquid loads in trailed or semi-trailed tanks [2, 10], including liquid organic fertilizers, manure effluents, irrigation water, working solutions of pesticides, and other agrochemicals. Unlike the transportation of bulk or unit (solid) cargo, the transportation of liquid loads is characterized by the presence of internal dynamic disturbances caused by the motion of the liquid inside a cylindrical tank [15]. Such oscillations lead to variations in loads transmitted through the drawbar coupling to the tractor axle and the power unit.

The disturbing forces generated by oscillations of the liquid mass significantly affect the dynamics of the tractor unit: they increase the amplitude of vertical and angular vibrations, alter the distribution of normal reactions on the supporting wheels, and reduce stability during braking and maneuvering. In the case of uneven field roads, slopes, or increased transport speed, the liquid load may generate resonant wave processes, increasing the risk of tank rollover or loss of tractor controllability. This is particularly critical for large-capacity machines (10–18 m³ and more) and for tractor-tank combinations with a total mass exceeding 15 t [12].

An additional challenge of modern agricultural production is the tendency toward increasing volumes of liquid fertilizer transportation within the framework of precision farming technologies and environmentally sound fertilization practices. This leads to higher transport speeds and increased tractor loads. At the same time, from a safety standpoint, tractor-tank aggregates (the “tractor-tank” system) remain significantly less studied than road vehicles, despite substantial accident rates and equipment downtime. Study [19] demonstrates that at high speeds and high tank filling levels, the mass center position shifts, leading to a noticeable increase in lateral and torsional moments. It was established that tank filling levels between 70–90% have the most adverse effect on braking performance and dynamic stability. Furthermore, [9] shows that the greatest liquid sloshing forces occur in tractor tanks filled at 71–92%, a hazardous level that affects not only productivity but also traffic safety.

The evaluation of the fuel characteristics of an internal combustion engine based on its load characteristics is driven by its dynamic operation and the diversity of operating conditions. Fuel economy depends not only on the engine design but also on its response to changes in load and operating modes. Load characteristics reflect real operating conditions and enable the assessment of variations in specific fuel consumption and effective efficiency, as well as the determination of the engines best fuel-economy range [7].

During numerical simulation of wheeled tractor motion, an important aspect is selecting an appropriate driving test cycle that reproduces the characteristic operating modes of agricultural machinery. Modern practice [14, 16] employs

several approaches to forming such cycles. In particular, the international regulatory framework (European Load Response [5], EPA Non-road Duty Cycles [3]) defines test cycles for various types of mobile agricultural and construction equipment, including wheeled tractors. These cycles are based on experimental data describing variations in the angular speed and effective torque of the internal combustion engine crankshaft, which closely reflect real operating conditions.

According to [1, 4], the NRTC cycle is appropriate for off-road specialized machinery, including agricultural tractors and construction equipment. It allows the simulation of load characteristics across a wide range of operating conditions, typical of both field work and operations involving frequent stops, accelerations, and periodic torque changes. Unlike steady-state characteristics, the NRTC includes a full sequence of realistic dynamic effects that occur during machine acceleration, idling, transitions to partial loads, or operation at short-term peak power levels. This structure enables accurate reproduction of transient modes, which dominate during the real operation of internal combustion engines under variable load conditions, when the powertrain continuously adapts to varying road conditions, changing soil conditions, or cyclical operation of mounted and towed implements.

The aim of this study is to provide a theoretical basis for determining changes in the fuel economy of a wheeled tractor during the transportation of liquid cargo with a free liquid surface by simulating standardized driving cycles.

3. Description of the ICE crankshaft motion

When describing changes in the operational performance of an internal combustion engine, a fairly common practice is to neglect the working processes by which the chemical energy of the fuel is converted into mechanical work. In addition, when describing the mechanical operation of the internal combustion engine, an assumption is introduced that the harmonic components of the torque are neglected. During engine operation, the torque on the crankshaft is a non-constant quantity that varies within each working cycle due to the alternation of strokes, changes in cylinder pressure, inertial forces of the valve train, and other factors. These fluctuations are periodic in nature and can be represented as the sum of an average (steady) torque and harmonic (pulsating) components.

Thus, the stated assumptions make it possible to use the equation of dynamic equilibrium of crankshaft motion:

$$J_e \cdot \frac{d\omega_e}{dt} = M_e + M_D \quad (1)$$

where: J_e – moment of inertia of the masses reduced to the crankshaft; ω_e – angular speed of the crankshaft; M_e – actual (effective) torque on the crankshaft; M_D – resistive torque accounting for the load from the transmission, auxiliary units, and friction losses.

Analyzing the equation reveals that it describes the dynamic equilibrium of a rotational system and generalizes Newton’s second law to rotational motion. Its physical meaning lies in reflecting the relationship between the acting torques and the changes in the internal combustion engine’s kinetic state. The left-hand side of the equation –

the product of the moment of inertia J_e and the angular acceleration $d\omega_e/dt$ – characterizes the torque required to change the angular speed of the crankshaft. In this case, the moment of inertia is a measure of the rotational system's inertial properties that resist changes in rotational speed. It includes not only the crankshaft itself, but also the fly-wheel, clutch components, and other parts kinematically connected to the engine.

The actual (effective) torque reflects the internal combustion engine's ability to generate rotational energy from fuel combustion in the cylinders. Its magnitude depends on the fuel supply mode, rotational speed, and other factors. The resistive torque determines the total opposition to rotation and includes mechanical losses, loads from the transmission and auxiliary units, and resistance arising from wheel interaction with the soil or road surface. Thus, the right-hand side of the equation represents the balance of torques acting on the crankshaft.

As shown, a major difficulty in modeling crankshaft motion is the representation of the actual torque component. To address this, it is proposed to introduce the parameter ε_r , which describes changes in the current position of the fuel supply control element. This approach simplifies the mathematical description, as neglecting the automatic control system significantly reduces the number of equations. Also note that the coefficient ε_r allows us to account for partial load in the internal combustion engine frequency and torque, since these can vary, $\varepsilon_r \in [0.46; 1]$.

$$\varepsilon_r = \frac{\omega_e}{\omega_{nom}} - \left(M_{normz} - \frac{M_e}{M_{nom}} \right) \cdot (k_x - 1) \quad (2)$$

where: ω_{nom} – nominal angular speed of the crankshaft; M_{normz} – normalized torque; M_{nom} – nominal torque on the crankshaft; k_x – the coefficient defining the slope of the governor branch of the external speed characteristic.

Such terms of the equation as ω_{nom} and M_{nom} are tabulated values provided by the internal combustion engine manufacturer. The coefficient k_x is taken to be within the range of 1.07–1.086. The unknown term is the normalized torque, which depends on the external speed characteristic of the internal combustion engine, reduced to the nominal torque M_{nom} . Thus, under nominal operating conditions, when $M_e = M_{nom}$, the normalized torque equals $M_{normz} = 1$.

Next, a number of requirements are introduced that must be satisfied:

– the parameter ε_r is always smaller than the engine load coefficient with respect to rotational speed; they are equal only when:

$$M_{normz} = \frac{M_e}{M_{nom}} \quad (3)$$

– the parameter $\varepsilon_r = \omega_e/\omega_{nom}$, when the maximum torque loading M_e/M_{nom} is equal to the normalized torque M_{normz} ;
– in cases where $\varepsilon_r > \varepsilon_{\omega}$, this condition is unacceptable; therefore, $\varepsilon_r = \omega_e/\omega_{nom}$ is adopted. In this case, the ICE torque does not depend on the position of the fuel supply control element.

Considering the above, the determination of the variation of the actual (effective) torque on the crankshaft is reduced to the calculation using the following formula:

$$M_e = M_{normz} \cdot M_{nom} + \frac{\varepsilon_r - \frac{\omega_e}{\omega_{nom}}}{k_x - 1} \cdot M_{nom} \quad (4)$$

Taking this equation into account, we transform the general equation

$$J_e \cdot \frac{d\omega_e}{dt} = \left(M_{normz} + \frac{\varepsilon_r - \frac{\omega_e}{\omega_{nom}}}{k_x - 1} \right) \cdot M_{nom} + M_D \quad (5)$$

It should be noted that the variation of the angular speed ω_e lies within the range $[\omega_{min}; k_x \cdot \omega_{nom}]$, while the variation of the fuel-supply control parameter ε_r lies within the range $[(\omega_{min} - M_{normz} \cdot \omega_{min} \cdot (k_x - 1))/\omega_{nom}; 1]$.

Thus, a mathematical description of the crankshaft's equation of motion in an internal combustion engine is presented, based on the dynamic equilibrium of a rotational system and generalizing Newton's second law to rotational motion.

To construct the fuel economy characteristics of the FPT NEF 67 internal combustion engine, the manufacturer's reference data are used, specifically the external speed characteristic consisting of the dependencies $M_e(\omega_1)$ and $N_e(\omega_1)$. The results are presented in Fig. 1.

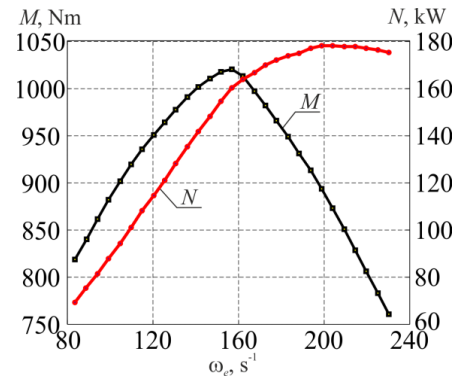


Fig. 1. External speed characteristic of the FPT NEF 67

The fuel consumption characteristics of any internal combustion engine depend on variations in the specific fuel consumption g_e , which is calculated using the equation:

$$g_e = g_{nom} \cdot k_{\omega} \cdot k_N \quad (6)$$

where: g_{nom} – the nominal fuel consumption when the ICE operates in the nominal mode; k_{ω} – the coefficient characterizing the engine load with respect to the angular speed of the crankshaft; k_N – the coefficient characterizing the engine load with respect to power.

The following quadratic functions describe the dependencies of the engine load coefficients k_{ω} and k_N

$$k_{\omega} = a_{\omega} + b_{\omega} \cdot \frac{\omega_e}{\omega_{nom}} + c_{\omega} \cdot \left(\frac{\omega_e}{\omega_{nom}} \right)^2 \quad (7)$$

$$k_N = a_N + b_N \cdot \frac{N_e}{N_{e_{ESC}}} + c_N \cdot \left(\frac{N_e}{N_{e_{ESC}}} \right)^2 \quad (8)$$

where: a_{ω} , b_{ω} , c_{ω} , a_N , b_N , c_N – constant coefficients; $N_{e_{ESC}}$ – the effective engine power on the external speed characteristic at the same operating speed regime as N_e .

It should be noted that equations are valid under the following condition:

$$\begin{cases} a_{\omega} + b_{\omega} + c_{\omega} = 1; \\ a_N + b_N + c_N = 1. \end{cases} \quad (9)$$

After determining the specific fuel consumption, the instantaneous fuel consumption can be calculated.

$$G_M = 0.278 \cdot 10^{-6} \cdot g_e \cdot N_e \quad (10)$$

If it is necessary to determine the instantaneous fuel consumption over a certain time interval (from t_1 to t_2), the equation is transformed as follows:

$$G_M = 0.278 \cdot 10^{-6} \cdot \int_{t_1}^{t_2} g_e \cdot N_e dt \quad (11)$$

In addition to determining fuel consumption through the specific fuel consumption, it is also possible to calculate the effective efficiency of the internal combustion engine η_e , which is determined by the following equation,

$$\eta_e = \frac{3.6 \cdot 10^6}{g_e \cdot H_u} \quad (12)$$

where H_u is the lower heating value of the fuel.

Three-dimensional plots of the dependence of the effective power N_e and the specific fuel consumption g_e of the FPT NEF 67 internal combustion engine on the effective torque and the crankshaft angular speed are constructed. The results are shown in Fig. 2.

Analyzing the results shown in Fig. 2, it is noted that the maximum effective efficiency of the FPT NEF 67 ICE is 0.48, achieved at an effective torque $M_e = 750$ Nm and a crankshaft angular speed $\omega_e = 178$ s⁻¹ (approximately 1700 rpm). Under these conditions, the specific fuel consumption g_e is 176 g/kWh, and the effective power N_e equals 124.75 kW.

The next step in determining the fuel characteristics of the internal combustion engine is to evaluate them using load characteristics.

4. Driving test cycles for wheeled tractors

One of the most important reasons for applying driving cycles is the impossibility of conducting a full-factorial experiment that would cover all possible combinations of rotational speed, load, temperature conditions, injection parameters, boosting modes, and control algorithms of the electronic control system. A full-factorial plan for an engine of such complexity as the NEF 67 would require thousands,

or even tens of thousands, of experimental points, which is practically infeasible due to the enormous time and fuel consumption, engine wear, and the high requirements for measurement equipment. In addition, an experimental setup is not always capable of accurately reproducing the conditions under which a wheeled tractor, construction machine, or vehicle actually operates, especially with respect to abrupt load changes, road conditions, micro-transient processes, and the response of the boosting system.

Therefore, instead of full-factorial experiments, which are extremely complex, time-consuming, and practically unattainable for engines with wide load and rotational speed ranges, it is advisable to use standardized driving test cycles. For modeling the transport operation of wheeled tractors and other self-propelled machinery, the ELR (European Load Response [5]), EPA Nonregulatory Non-road Duty Cycles (Environmental Protection Agency [3]), and NRTC (Non-Road Transient Cycle [4]) cycles are widely applied. Each of these cycles is designed to reproduce different aspects of real engine operation, including variable loads, vehicle dynamics, and characteristic transient processes.

Since no single standardized transport cycle is specified for wheeled tractors, and their operation is often associated with movement in suburban conditions, the ELR cycle is appropriate for modeling the corresponding load modes. This cycle defines the response of the internal combustion engine to stepwise changes in crankshaft rotational speed and load (Fig. 3). This allows assessment of the dynamic characteristics of fuel supply during the transportation of trailed implements.

The essence of the ELR cycle is reduced to modeling the motion of a wheeled tractor at three speed operating modes (A, B, C) [5], which are defined as follows:

$$A = \omega_{\min LO} + 0.25 \cdot (\omega_{\max HI} - \omega_{\min LO}) \quad (13)$$

$$B = \omega_{\min LO} + 0.5 \cdot (\omega_{\max HI} - \omega_{\min LO}) \quad (14)$$

$$C = \omega_{\min LO} + 0.75 \cdot (\omega_{\max HI} - \omega_{\min LO}) \quad (15)$$

where $\omega_{\min LO}$ is the minimum crankshaft angular speed of the internal combustion engine at 50% of the maximum engine power; $\omega_{\max HI}$ is the maximum crankshaft angular speed of the internal combustion engine at 70% of the maximum engine power.

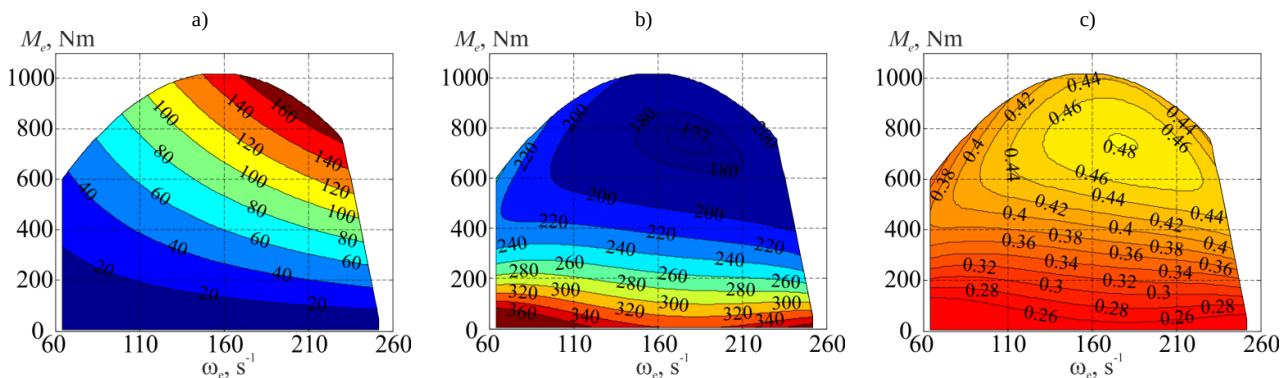


Fig. 2. Variation of the operating parameters of the FPT NEF 67 as functions of the effective torque M_e and the crankshaft angular speed ω_e : a) effective power N_e (kW); b) specific fuel consumption g_e (g/kWh); c) effective efficiency η_e

The EPA cycle (Fig. 3) is used to simulate the transport operating modes of a wheeled tractor or another self-propelled machine that operates on public roads or moves between production sites. It includes characteristic speed and load profiles typical of motion with steady-state sections, repeated accelerations, partial-load operation, and short transient processes. This makes the EPA cycle particularly valuable for evaluating fuel economy when a wheeled tractor operates as a transport vehicle rather than as a traction or working unit. In contrast to steady-state characteristics, the NRTC (Fig. 3) includes a complete sequence of realistic dynamic effects that occur during vehicle acceleration, transition to idle, shifts to partial loads, or operation at short-term peak power levels.

It should be noted that each of the above cycles has its own speed and load profile. In this research, only the speed characteristics of the cycle profiles are used, as the speed profile provides sufficient information, allowing modeling of changes in acceleration, deceleration and constant-speed motion, as well as the overall dynamic character of the transport operation. It is precisely the speed profile of the cycles that determines the duration of motion phases, the required tractive forces, and the subsequent energy consumption.

At the same time, the load profiles of the cycles (engine torque of the ICE) are less appropriate for studying the transport process with a towed tank, as they are derived from typical working operations involving variable technological modes and the operation of mounted implements. Such profiles describe the specific loading characteristics of the internal combustion engine during intensive, power-demanding operations, which do not correspond to the conditions of transport cargo movement, where tractive demand is determined by motion resistances, the unit's mass, and road conditions. In the case of a wheeled tractor with a towed tank, these loads can be calculated using a mathematical model of resistances and mass-inertia parameters rather than imposed by an external load cycle.

5. Investigation of changes in instantaneous fuel consumption

Before analyzing any research indicators, it is necessary to develop a general simulation model of a wheeled tractor towing a tank. For this purpose, we will use the MATLAB dynamic modeling system with the Simulink subsystem. The overall layout of the developed MATLAB/Simulink model is shown in Fig. 4. The general mathematical model of a wheeled tractor towing a tank is described by a system of differential equations. The calculation of these equations is based on time-component integration, which requires the use of a numerical method. Since the mathematical model is implemented in MATLAB, several numerical methods are available, including Euler (ode1), Heun (ode2), Bogacki-Shampine (ode3), Runge-Kutta (ode4), Dormand-Prince (ode5), and Dormand-Prince (ode8). We will select the classical Runge-Kutta method (ode4), which offers high accuracy at a moderate computational cost. The transported MGT-16 tank will reach a maximum speed of up to 20 km/h. Figure 5 shows the distribution of operating points of the change in the ICE's effective power during the simulation of the ELR, EPA, and NRTC cycles.

Analyzing the results in Fig. 5, it can be noted that for the ELR cycle, the operating points are generally localized around stabilized load conditions, without dynamic fluctuations or transient processes. The tractor essentially moves without load variation. In the EPA cycle, the operating points are concentrated over a wider range, closer to realistic transport conditions with stepwise changes in effective torque. Due to the absence of sharp load peaks, the “engine – transmission – wheels” system operates more stably and delivers traction more uniformly. In the NRTC cycle, the ICE's operating points are distributed much more broadly, covering both low and high values of torque and angular velocity. A particularly large number of points fall into the zone of transient loads and peak powers, where the engine produces abrupt, uneven traction values. This directly enhances the dynamic behavior of the “engine–transmission–wheels” system and, consequently, limits its ability to adapt to rapid load changes quickly.

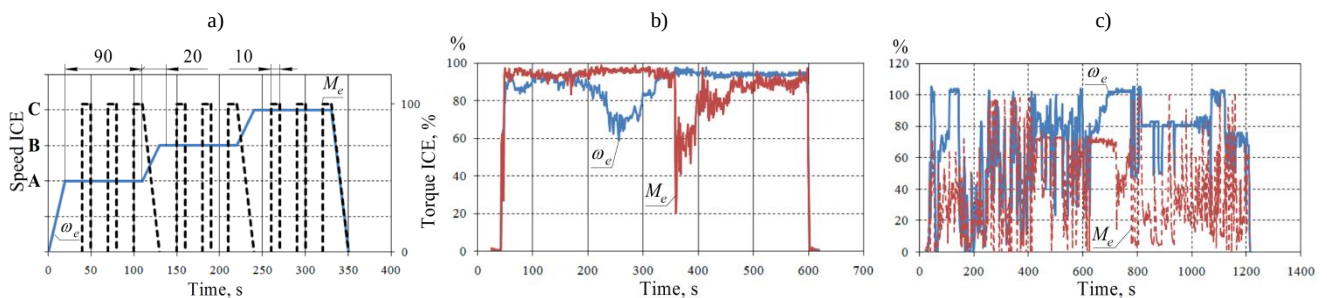


Fig. 3. Investigated driving test cycles of wheeled tractors: a) ELR; b) EPA; c) NRTC

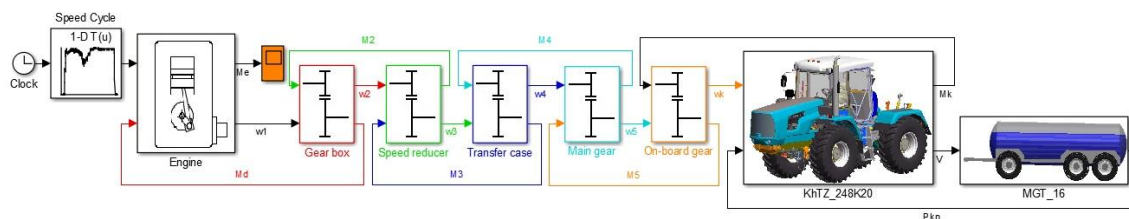


Fig. 4. Implementation of the KhTZ-248K.20 wheeled tractor and MGT-16 towed tank model in MATLAB/Simulink

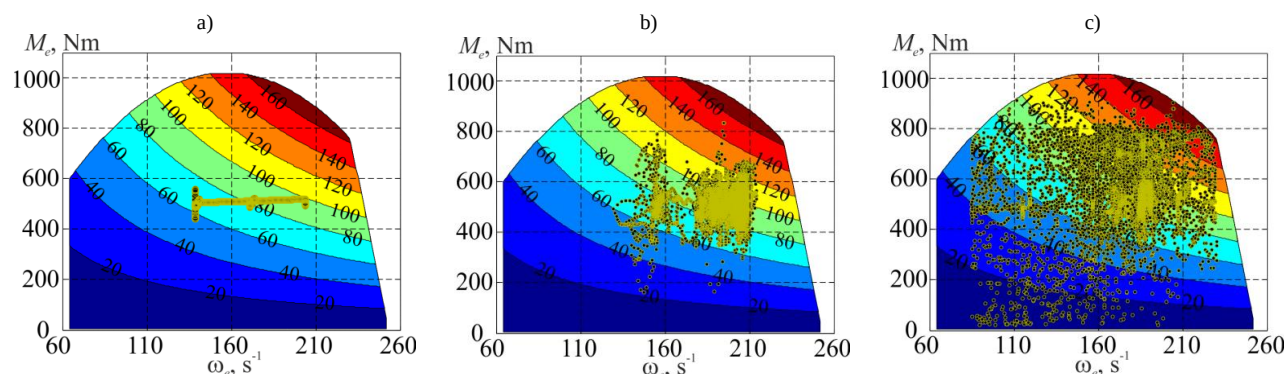


Fig. 5. Change in the distribution of operating points of the ICE's effective power during the simulation of driving cycles: a) ELR; b) EPA; c) NRTC

Considering the above, it becomes possible to determine the instantaneous fuel consumption of the ICE during the simulation of the ELR, EPA, and NRTC cycles. Analyzing the results from Fig. 6, it can be noted that during the simulation of the ELR cycle for the KhTZ-248K.20 wheeled tractor with a fully filled MGT-16 tank ($H = 1.6$ m or $m_p = 16$ t), the instantaneous fuel consumption is $G_M = 1631$ g ≈ 1.6 kg. With an empty tank ($H = 0$ m or $m_{\text{liquid}} = 0$ kg), $G_M = 568.5$ g ≈ 0.6 kg.

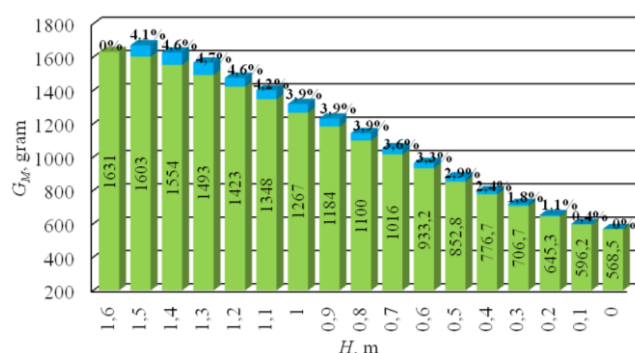


Fig. 6. Change in instantaneous fuel consumption G_M during ELR cycle simulation depending on tank fill level H

Figure 6 shows the change in instantaneous fuel consumption when transporting the tank under the ELR cycle without considering the liquid's sloshing effects, as well as the percentage increase in fuel consumption when sloshing is accounted for. At extreme fill levels, no liquid motion occurs, whereas at intermediate levels, it affects the instantaneous fuel consumption. The greatest influence of liquid sloshing is observed when the tank is filled to $H = 1.3$ m, where $G_M = 1567$ g, which is 4.72% higher than the fuel consumption without considering sloshing. In Fig. 7, as in Fig. 6, the change in instantaneous fuel consumption is presented without considering liquid sloshing, and its increase when sloshing is included during the EPA cycle simulation. With a fully filled MGT-16 tank ($H = 1.6$ m), $G_M = 3119$ g. With an empty tank ($H = 0$ m), $G_M = 1138$ g. The greatest effect of liquid sloshing occurs at $H = 1.3$ m (the same trend as in the ELR cycle), where $G_M = 3068$ g, which is 6.76% higher than the fuel consumption without considering sloshing.

The increase in instantaneous fuel consumption at $H = 1.3$ m in the EPA cycle by 6.76%, compared to 4.72% in the ELR cycle, is due to the different dynamic structures of

these test cycles. At $H = 1.3$ m, the liquid in the tank enters the most intensive longitudinal oscillation regime, as the fill level is close to resonance. This maximizes the amplitude of the liquid's center of mass motion, increasing inertial loads on the tractor during changes in engine speed and torque.

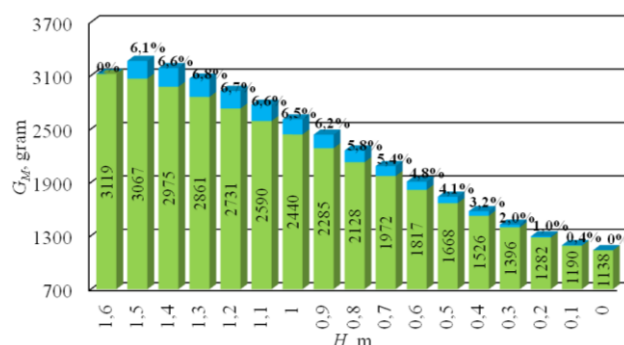


Fig. 7. Change in instantaneous fuel consumption G_M during EPA cycle simulation depending on tank fill level H

In the ELR cycle, the intensity of load and speed changes is moderate and smoother. Stepwise load increases without sharp accelerations or deep drops in instantaneous traction characterize this cycle. Therefore, the inertial forces generated by the liquid have a less pronounced effect on the engine, and the increase in fuel consumption is only 4.72%. In contrast, the EPA cycle contains a significantly larger number of rapid load changes, short-term peak regimes, and frequent transitions between acceleration, stabilization, and deceleration. Under these conditions, the sloshing mass of the liquid generates additional inertial moments that the tractor must compensate for, resulting in higher fuel consumption. Consequently, the fuel consumption increase reaches 6.76%.

In Fig. 8, as in Fig. 6 and Fig. 7, the change in instantaneous fuel consumption is shown without considering sloshing effects, and its increase when sloshing is taken into account during the NRTC cycle simulation. With a fully filled MKT-16 tank ($H = 1.6$ m), the instantaneous fuel consumption is $G_M = 6508$ g. With an empty tank ($H = 0$ m), $G_M = 2600$ g. The greatest influence of liquid sloshing occurs at a fill level of $H = 1.3$ m (the same trend observed in the ELR and EPA cycle simulations), where $G_M = 6557$ g, which is 9.22% higher than the fuel consumption without considering sloshing.

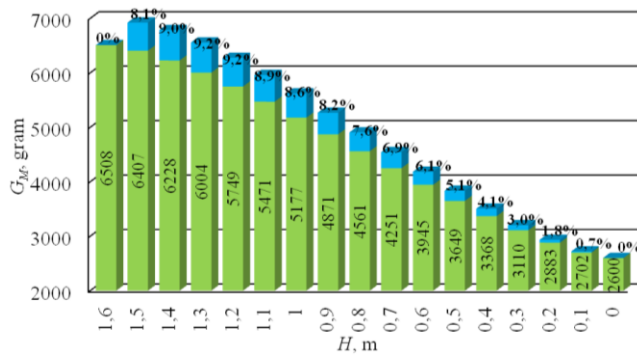


Fig. 8. Change in instantaneous fuel consumption G_M during NRTC cycle simulation depending on tank fill level H

The NRTC cycle exhibits high absolute fuel consumption, which is explained by its high energy intensity, high average power, and long duration. Consequently, the relative percentage increase is the largest because this cycle contains frequent, abrupt transient events whose spectrum closely matches the liquid's natural frequencies, leading to additional fuel consumption. In the EPA cycle, the relative percentage increase is smaller than in NRTC, because energy is expended to compensate for liquid accelerations, which constitute only a small portion of the already high baseline consumption, resulting in a 6.76% increase. The ELR cycle is the smoothest, so the liquid is minimally agitated, and both absolute and relative increases are the smallest.

The mechanism by which sloshing is converted into fuel consumption is that, during each acceleration or deceleration, the liquid must change its speed and position, performing inertial work that is powered by the ICE through increased fuel supply. The fill level $H = 1.3$ m is critical, as it corresponds to near-resonance conditions for the tank geometry, producing maximum liquid displacement amplitude and maximum additional inertial work.

The NRTC cycle yields the highest relative increase (9.22%), EPA the highest absolute but moderate relative increase (6.76%), and ELR the smallest (4.72%). Practically, this indicates that the fuel efficiency of transporting a filled tank depends not only on the fill level but also on the dynamics of motion, highlighting the importance of analyzing the transport of a partially filled tank with a tractor equipped with a hybrid transmission.

6. Conclusions

1. A mathematical representation of the fuel economy indicators of an internal combustion engine is presented,

including instantaneous fuel consumption, volumetric fuel consumption, as well as the calculation of the effective efficiency factor, which is based on the determination of specific fuel consumption.

2. It is substantiated that determining the fuel characteristics of an internal combustion engine based on load characteristics makes it possible to evaluate its efficiency under real operating conditions, where rotational speed and torque continuously vary. This approach accounts for operating dynamics and transient processes, in contrast to fixed operating modes, and is practically necessary due to the complexity of conducting full-factorial experiments. Standardized driving cycles ELR, EPA, and NRTC make it possible to reproduce real operating conditions and assess fuel economy.

3. It is determined that the maximum increase in instantaneous fuel consumption G_M during the transportation of the MGT-16 tanker is observed at a filling height of $H = 1.3$ m, which is caused by the inertial behavior of the liquid. For the ELR cycle, $G_M (H = 1.3 \text{ m}) = 1567 \text{ g}$, with a relative increase of 4.7% and an absolute increase of 74 g. For the EPA cycle, $G_M (H = 1.3 \text{ m}) = 3068 \text{ g}$, with a relative increase of 6.8% and an absolute increase of 207 g. For the NRTC cycle, $G_M (H = 1.3 \text{ m}) = 6557 \text{ g}$, with a relative increase of 9.2% and an absolute increase of 553 g. The largest relative increase occurs in the EPA cycle due to abrupt transient operating modes, the largest absolute increase occurs in the NRTC cycle due to its high energy intensity, and the smallest increase is observed in the ELR cycle due to smoother operating conditions.

Further research

In future work, experimental studies are planned under real road conditions, with traffic parameters recorded. This approach will allow verification of the adequacy of the mathematical model and simulation data, as well as the assessment of the impact of tank filling level on the tractor's fuel economy under variable traffic flow conditions. Additionally, the analysis is planned to be extended to different soil and road surface types, providing a more comprehensive understanding of the efficiency of transport operations in agricultural practice.

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Nomenclature

ICE internal combustion engine
MTA machine-tractor aggregate
ESC external speed characteristic

ELR European Load Response
EPA Environmental Protection Agency
NRTC non-road transient cycle

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Analysis of emissions from a heavy vehicle designed for transporting concrete

ARTICLE INFO

In the context of global efforts to reduce greenhouse gas emissions and air pollution, increasing importance is being attached to the accurate monitoring and analysis of emissions from transport, including commercial vehicles used in specialized applications. One of the key groups of vehicles whose emissions remain insufficiently studied is trucks equipped with pump-type superstructures (concrete mixers with concrete pumps), widely used in the construction sector. This article presents the results of research conducted on the emissions of harmful compounds from concrete mixers in real traffic conditions. The measurements were performed using the Axion R/S+PM portable emission measurement system from Global MRV, which enabled the mass measurement of pollutant emissions, including CO, CO₂, NO_x, and HC, under real truck driving conditions. Based on the collected data, the time-density characteristics of the tested compounds were determined as functions of crankshaft speed and engine load, and their emission intensity was determined as a function of vehicle speed. On this basis, the impact of changes in load weight on the emission intensities of the tested pollutants was demonstrated. The analysis of pollutant emissions from a heavy concrete transport vehicle enabled the determination of the impact of various road conditions and engine operation on the amount and type of compounds emitted. By monitoring operating parameters such as load and engine crankshaft speed, it is possible to gain a more accurate understanding of the mechanisms of exhaust emissions under actual vehicle operating conditions. In this way, it is possible to more effectively identify situations with particularly high pollutant emissions and determine the optimal operating conditions to reduce them.

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1. Introduction

Nowadays, transport is an integral part of the global economy and plays a significant role in the daily functioning of society. With the continuous increase in the number of means of transport intended for land traffic, it has become possible to transport various types of goods on a large scale, mainly thanks to the use of heavy vehicles. However, despite the benefits of this sector's development, growing public awareness has led to measures to protect the environment from its negative effects. An area of particular concern is road transport, especially heavy goods vehicles, whose activities generate significant amounts of pollutants such as carbon monoxide (CO), carbon dioxide (CO₂), nitrogen oxides (NO_x), hydrocarbons (HC), and particulate matter (PM) and particulate number (PN). Their emissions negatively impact air quality and pose a serious threat to the natural environment [8, 16, 23, 27].

Road transport has many aspects that make it a key factor in the functioning of the global economy. It ensures mobility, supports market integration and access to goods and services. Its development contributes to economic growth by facilitating the movement of people and goods, thereby promoting trade, production, and distribution, both nationally and internationally. The road network for road transport is one of the most extensive and coherent compared to other forms of transport. The dense infrastructure is perfectly suited to serving consumer, production and commercial areas, enabling efficient and effective logistics. In addition, road transport vehicles are adaptable to various types of transport, enabling them to handle even the most specific goods. Compared to other modes of transport, the costs of road transport are more favorable in terms of

transport time to cost, making it more attractive economically. An important aspect of road transport is the possibility of direct access to the destination, which distinguishes it from other modes of transport and enables direct access to the destination [29]. The foundation of road transport is primarily heavy vehicles. Chart 1 shows the number of heavy goods vehicles registered in the European Union and in Poland in 2022 [7]. Their importance to the economy is particularly significant because they enable the transport of larger quantities of goods over long distances, delivering raw materials, products and materials necessary for the functioning of various industries. In addition, heavy goods vehicles are an integral part of global supply chains and enable connections between manufacturers, suppliers, and customers worldwide [9].

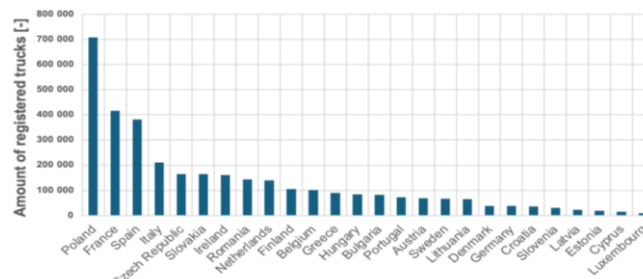


Fig. 1. Number of heavy goods vehicles intended for transport within the EU in 2022 [7]

The impact of road transport on the natural environment is significant and multifaceted. Vehicles not only generate noise during operation but also emit harmful and toxic compounds. Pollutant emissions from this sector are in-

creasing mainly due to rising freight transport demand [22]. The distribution of pollutants emitted into the environment is subject to constant change, with traffic intensity and vehicle technical condition as the main determinants. Exceedances of the permissible emissions limit are often caused by insufficient road infrastructure. Particularly in large urban agglomerations, where traffic is heavy, traffic jams occur frequently. This process negatively impacts traffic flow, reducing vehicle speed and leading to frequent acceleration and braking, which in turn increases pollutant emissions [3]. Heavy-duty vehicles contribute significantly to pollutant emissions, shaping air quality and posing a major challenge for ecology. Their engines, especially those powered by diesel fuel, emit a variety of pollutants, including carbon oxides, nitrogen oxides, hydrocarbons, and particulate matter. The combustion of fuel (especially diesel fuel in CI engines) is the main source of these emissions. In recent years, the Polish transport sector has made significant efforts to reduce emissions, comparing favorably with its Western European competitors, thanks to investments in a new fleet. Furthermore, in 2011, it emitted 1.1 thousand tons of greenhouse gases per 1000 euros of added value generated, exceeding the average for all EU countries by 130%. However, by 2021, these emissions had fallen to 0.6 thousand tons, which was 80% of the European average [6]. Notably, in recent years, the number of electric trucks operating in the European Union has increased significantly. In 2021, it amounted to 1300 units, while in 2022 – 3800, a significant jump from 2019, when only 600 units were recorded. In Poland, although an upward trend is evident, the pace of electric vehicle adoption is much slower. Data from the European Alternative Fuels Observatory show that in 2022, only 12 electric trucks were in operation in Polish companies, which indicates the limited spread of this technology on the domestic transport market. This is mainly due to the long battery charging time and low load capacity, as well as limited range [6]. Currently, the transport sector in Poland emits approximately 8 million tons of greenhouse gases per year, but, according to plans, this figure will gradually decrease to zero within the next three decades. Reducing emissions by approximately 3 million tons per decade, while maintaining continuous economic growth, will require significant changes and innovations in the transport sector [5].

In an era of growing environmental awareness and stricter emission standards, increasing attention is being paid to analyzing the impact of transport on the environment – not only in passenger transport, but also in the freight sector. One of the less-researched but important areas is specialized vehicles, such as concrete pump trucks, commonly used in the construction industry to transport and pump concrete. The specific nature of their work – frequent stops, low speeds, loads resulting from concrete's weight – means that the emissions they generate can differ significantly from the declared values measured in laboratory or test conditions.

Therefore, researchers focus on obtaining as much information as possible about emissions in real traffic condi-

tions. Such tests are carried out on passenger cars [1, 4, 14, 25, 33], heavy goods vehicles [11, 12, 17, 20] and other commercial vehicles [15, 26, 34]. Currently, there is a shortage of studies on the emissions of harmful compounds from heavy concrete transport vehicles. This topic is particularly important given the nature of these vehicles' work, as they often operate on construction sites and directly affect workers. Construction personnel are particularly exposed to suspended dust. According to the authors in [28], this exposure is particularly associated with dust containing free crystalline silica, which is generated, among other processes, during the processing of building materials. Importantly, in real construction conditions, workers are often surrounded by vehicles powered by combustion engines (e.g. excavators, loaders, concrete pumps), which emit much finer particulate matter (PM1 and PM2.5) [8]. These ultrafine fractions can penetrate deeply into the respiratory system, significantly increasing health risks, especially with prolonged exposure. This is confirmed by the studies presented in [30]. The authors [21, 24] also confirmed that particulate matter emitted by compression-ignition engines is carcinogenic and responsible for cardiovascular and respiratory diseases. An analysis conducted in Canada confirmed that particulate matter emitted from vehicles (including off-road machinery) is responsible for over 700 deaths [13].

An important aspect to consider in assessing emissions from heavy-duty vehicles is that they are operated under conditions that differ significantly from those assumed in type-approval procedures. The authors of this publication have identified a research gap concerning HDVs, whose operating model is based on their use also in off-road conditions [2]. The laboratory conditions in which standard emission tests are conducted do not reflect the actual operating profiles of heavy-duty vehicles, particularly those used in construction and forestry. As noted in the literature [14, 17, 27], vehicles in this category very often operate in environments with high topographical variability, unstable ground and variable loads, which affects the distribution of engine operating parameters and, consequently, the level of actual emissions. This group of vehicles also includes the object analysed in this study – a concrete pump truck – which is used not only on roads but also on construction sites and on unpaved, temporary access roads. The variability of operating conditions, combined with operating in load modes typical of transporting heavy materials, makes it reasonable to measure emissions in the vehicle's actual working environment.

2. Research methodology

2.1. Research objects

The research object was a Volvo truck designed for transporting concrete (concrete mixer), model VTR3R/VP, manufactured in 2022 (Fig. 2). The vehicle was equipped with a 323 kW combustion engine that complies with the Euro VI emission standard. The technical parameters of the vehicle are presented in Table 1.

Table 1. Specification of research objects

Parameter	Value
Purpose	Concrete mixer
Model	VTR3R/VP
Year of manufacture	2022
Nett weight	14430 kg
Permissible load capacity	19570 kg
Engine power	323 kW
Engine displacement	10837 cm ³
Emissions standard	Euro VI



Fig. 2. A truck designed for transporting concrete, serving as a research object

2.2. Measuring instruments

The research used a portable Axion R/S+PM IV generation emission measurement system from the American company Global MRV (Fig. 3a, 3b). It enabled mass measurement of emissions of pollutants such as CO, CO₂, NO_x, HC and PM in real driving conditions of a truck used as a research object. Table 2 presents the parameters of the Axion R/S+PM device corresponding to the types of pollutants tested. CO, CO₂ and HC were measured using an NDIR analyser, while NO_x was measured using an electrochemical analyser. In addition, PM particulate matter was measured using the laser light scattering method [10]. The Axion R/S+PM system offers high measurement accuracy and reliability, even in changing conditions. Its compact, lightweight design enables easy installation on various trucks without structural changes. The device provides real-time exhaust emission monitoring

Table 2. Emission measurement parameters by Axion R/S+PM [10]

	Measurement range	Measurement accuracy	Measurement repeatability
CO	0–10%	±3%	±2%
CO ₂	0–16%	±3%	±2%
HC	0–2000 ppm	±4%	±2%
NO _x	0–5000 ppm	±1%	±1%
PM	0–250 g/m ³	±2%	±0,2%

a)



b)



Fig. 3. a) Axion R/S+PM portable emission measurement system, b) PEMS probes installed on the truck's exhaust system.

2.3. Research cycle

The emission analysis was conducted on a 36 km measurement section. It began and ended at the cement plant located in Rogówko, Kujawsko-Pomorskie Province, Poland. The route's destination was a property in the same province, in Mirakowo. Due to restrictions on the vehicle's total permissible weight, the measurement route included national and provincial roads, as well as unpaved access roads. Its course is presented in Fig. 4. The first pollutant emission measurements were taken during loading. The vehicle then set off for the planned unloading location along the established test route. Pollutant emissions continued to be monitored during the subsequent stages of the route, i.e., during the journey to the unloading location, the unloading itself and the return to the starting point.



Fig. 4. Measurement route used to measure pollutant emissions

3. Results of the study

3.1. Analysis of the operating parameters of the engine

To accurately examine pollutant emissions from a heavy concrete transport vehicle, the test was conducted at different total weights. These differences resulted from variations in load quantity at different stages of the test route. At the starting point of the measurement route, the vehicle was loaded. Approximately 9 m³ of concrete was used to reach its maximum possible load (1 m³ of the mixture used weighs approximately 2,000 kg, so the vehicle was loaded with approximately 18 tons of concrete). Then, in accordance with the vehicle's operating model, the load was unloaded, and the vehicle returned to the starting point without additional load.

Both during loading (Fig. 5a) and unloading (Fig. 5b), the most frequently maintained crankshaft speed range was 600–800 rpm and, to a slightly lesser extent, 400–600 rpm. The load on the drive unit during unloading extends very slightly to the range of 400–800 Nm at a crankshaft speed of 400–800 rpm. This is a direct result of the pump delivering the load to a considerable height of approx. 30 metres for unloading at a specific construction site. The journey to the unloading site (Fig. 5c) took place with the vehicle at maximum load. As in the previous case, the engine crankshaft speed ranged mostly from 600–800 rpm, with occasional values up to 2200–2400 rpm. The engine load usually fluctuated between 0 and 400 Nm, sometimes reaching 1200–1600 Nm. During the journey to the base (Fig. 5d), the vehicle's weight was comparatively lower due to the lack of load. This stage was characterized by the widest range of engine crankshaft speeds, mostly maintained at 1200–1400 rpm and not exceeding 2200–2400 rpm. As in

the case of the journey to the unloading site, the engine load usually fluctuated between 0 and 400 Nm and did not exceed 1600 Nm.

3.2. Analysis of the exhaust gas emission

The study also analysed the temporal density of pollutant emissions as a function of crankshaft speed and engine load. An indispensable part of the operation model of HDVs adapted for transporting concrete mix is their loading and unloading, during which the vehicle's combustion engine drives the pumps that pump the concrete mix during both filling and emptying of the system. It should be noted that standard concrete pump models can deliver concrete to heights of 20–30 m, corresponding to 6–10 floors of a building. Therefore, the authors subjected the unloading and loading processes to a similar analysis, which is presented in Fig. 6 and 7.

For CO, CO₂, THC and NO_x, the time density characteristics for the emissions of individual exhaust gas compounds as a function of crankshaft speed and engine load during loading and unloading are summarized in Fig. 6a–d. During loading, the areas of peak emissions occurred at speeds of 400–800 rpm and engine loads of 0–400 Nm. The situation was different during unloading (Fig. 7a–d), for which the emission areas of these compounds oscillated in the speed range of 0–1400 rpm, reaching the highest level at an engine load of 400–800 Nm for CO, CO₂ and THC, and 0–400 Nm for NO_x. Both during loading and unloading, the highest pollutant emissions therefore occurred at maximum engine load. The exception was NO_x emissions (Fig. 7d), which reached a maximum at the average engine load.

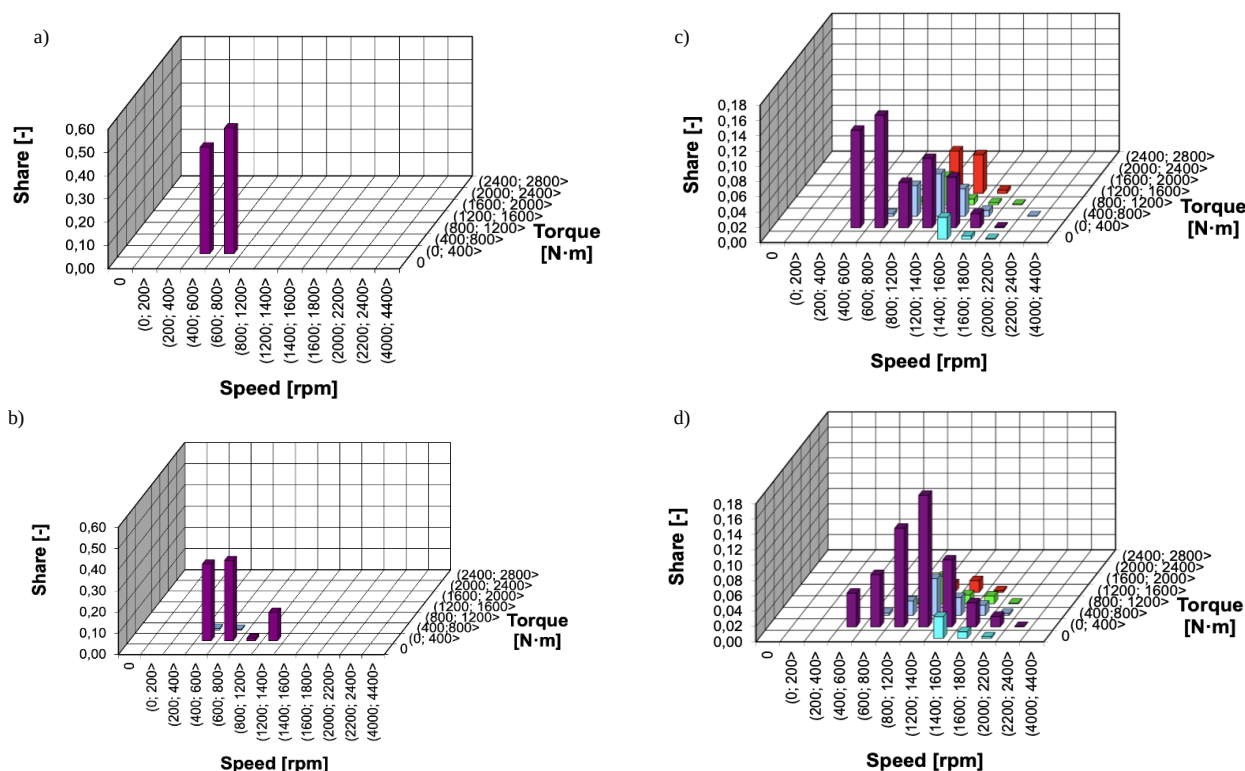


Fig. 5. Characteristics of working time as a function of engine speed and engine load a) during loading b) during unloading c) during travel to destination with load d) during return from destination without load

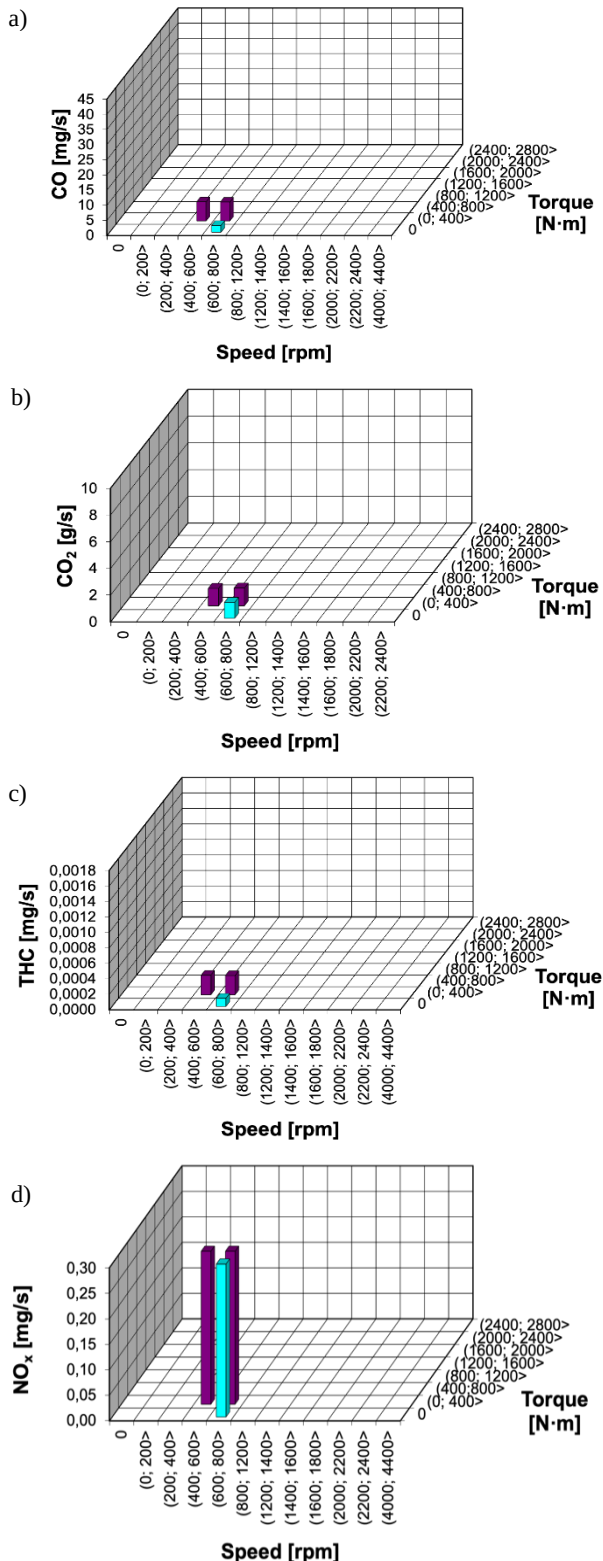


Fig. 6. Characteristics of time density of emissions as a function of crankshaft speed and engine load during loading: a) CO, b) CO₂, c) THC, d) NO_x

It should be noted that during loading, carbon monoxide emissions remain at lower levels of drive unit load (Fig. 6c). During concrete unloading (Fig. 7c), however, a significant increase in CO emissions can be observed at higher torque and rotational speed values. Due to the higher load,

unloading generates higher emissions of the analysed compound, a direct result of the unit's more intensive operation (including the use of a pump). This relationship can be observed for carbon dioxide, which, as a product of complete combustion, increases in proportion to fuel consumption. During unloading, energy demand increases, leading to a clear increase in CO₂ emissions at higher unit operating parameters. Emissions of unburned hydrocarbons also increase during unloading (at higher crankshaft speeds). This may be due to irregular engine operating conditions during concrete emptying.

Nitrogen oxide emissions are closely related to combustion temperature, which increases with vehicle load. However, when analysing the graph, the opposite is observed. During the loading phase, significantly higher nitrogen oxide emissions were observed – in the operating range described by the crankshaft speed of 400–800 rpm and a load of 0–400 Nm. During the unloading phase, the operating range over which emissions of the compound were recorded is wider (400–1400 rpm; 0–800 Nm), but the number of emissions is marginal. During concrete unloading, the engine load is higher, but the higher exhaust gas temperature makes the SCR system more effective, significantly reducing NO_x emissions. During loading – the initial phase of the test – the exhaust gases do not reach a sufficient temperature for the SCR system to work effectively.

Therefore, emissions of the compound are significantly higher in the first stage of the test. When driving with a load (Fig. 8a–d), numerous sudden and short-term increases in the value of carbon monoxide CO emissions were observed, reaching up to 1900 mg/s. In contrast, when driving to the base (Fig. 9a–d), the density of peaks for this compound was slightly lower, as was the peak emission value, which was around 1400 mg/s. The results suggest that CO emissions increase with vehicle weight. Based on the characteristics of second emissions as a function of drive-unit load and crankshaft speed, very high carbon monoxide emission values were observed at medium and high torque (1200–2400 Nm) and at medium speeds during driving to the unloading site. When analyzing the graph, a clear area with very high emissions of the analyzed compound can be identified: the range of medium loads from 800 Nm to 1600 Nm at 1200–2200 rpm. This corresponds to typical driving conditions with a load on mixed roads (extra-urban and urban driving, varied speed profile). Returning to the loading site results in lower CO emissions, a direct consequence of the absence of concrete loading. The maximum values are up to 350 mg/s, but across almost every range, emissions are up to 40% lower than with a load. Higher values when driving without a load are due to dynamic route conditions that require sudden acceleration or hill climbing. Overall, the results obtained clearly show the complexity of truck emission behavior under real driving conditions. The differences observed between the loading and unloading phases confirm that emission levels depend not only on engine load, but also on the efficiency of the exhaust aftertreatment system and driving dynamics. Analysis of emission maps provides valuable insights into the impact of various operating parameters on the formation of harmful compounds.

For carbon dioxide emissions, both with and without a load, they are concentrated in the same engine operating ranges (1200–2000 rpm, 1200–2000 Nm). The CO₂ emission characteristics show that emissions of this compound increase almost linearly with increasing engine load (Fig. 8b and Fig. 9b), clearly reflecting the increase in fuel-air mixture combustion during concrete transport. Returning vehicles without a load results in significantly lower CO₂ emissions (approx. 10–20% difference). However, the distribution of characteristics is very similar to that obtained for concrete transport. The highest emissions occur at torques of 1200–2000 Nm and speeds of 1400–2000 rpm, which means that even without a load, the vehicle emits a lot when accelerating or driving uphill. The highest emissions occur during typical engine operation, i.e., at medium speeds and medium or high torques, which is the case when driving outside built-up areas.

THC (total hydrocarbons) emissions remained at a relatively low level (Fig. 8c, Fig. 9c). The characteristics show an increase in hydrocarbon emissions with increasing speed, but these values are not significant. Maximum emissions are 0.02 mg/s for driving with a load and 0.025 mg/s for measurements taken during the return to base. This confirms the effectiveness of the exhaust gas purification systems. It is characteristic that the spread of THC emissions is wider for driving without a load – the characteristics show significantly more emission peaks for different load values and crankshaft speeds.

When driving to the unloading site, the highest nitrogen oxide emissions occur in the speed range of 800–1600 rpm and in the load range of 800–2000 Nm (Fig. 8d). Significant emissions occur at the highest torque values and at around 800–1200 rpm. A significant load on the drive unit occurs at relatively low crankshaft speeds when starting or climbing hills while transporting a load. When driving without a load (Fig. 9d), the recorded NO_x emissions are significantly lower. Particularly high emissions of this compound also occurred at 800–1200 rpm and 1600–2000 Nm torque. Sudden peaks in NO_x emissions when driving without a load may have been caused by sudden and short-term operating conditions of the drive unit (including sudden dynamic acceleration) during which the SCR system was unable to effectively reduce the nitrogen oxides produced.

An analysis of emission characteristics shows that peak pollutant formation occurs during periods of increased torque demand, regardless of vehicle load. While advanced exhaust aftertreatment systems can significantly reduce total emissions, operating conditions such as acceleration, hill climbing and variable loads continue to pose a significant challenge for effective emission control. These results highlight the need to optimize both driving strategies and emission system calibration to effectively reduce environmental impact under varying load conditions. In addition, accounting for weather conditions and route specifics can further improve the effectiveness of measures to reduce harmful emissions, which is important for protecting air quality in urban and non-urban environments. It also enables better planning of transport operations and optimisation of vehicle performance in real driving conditions.

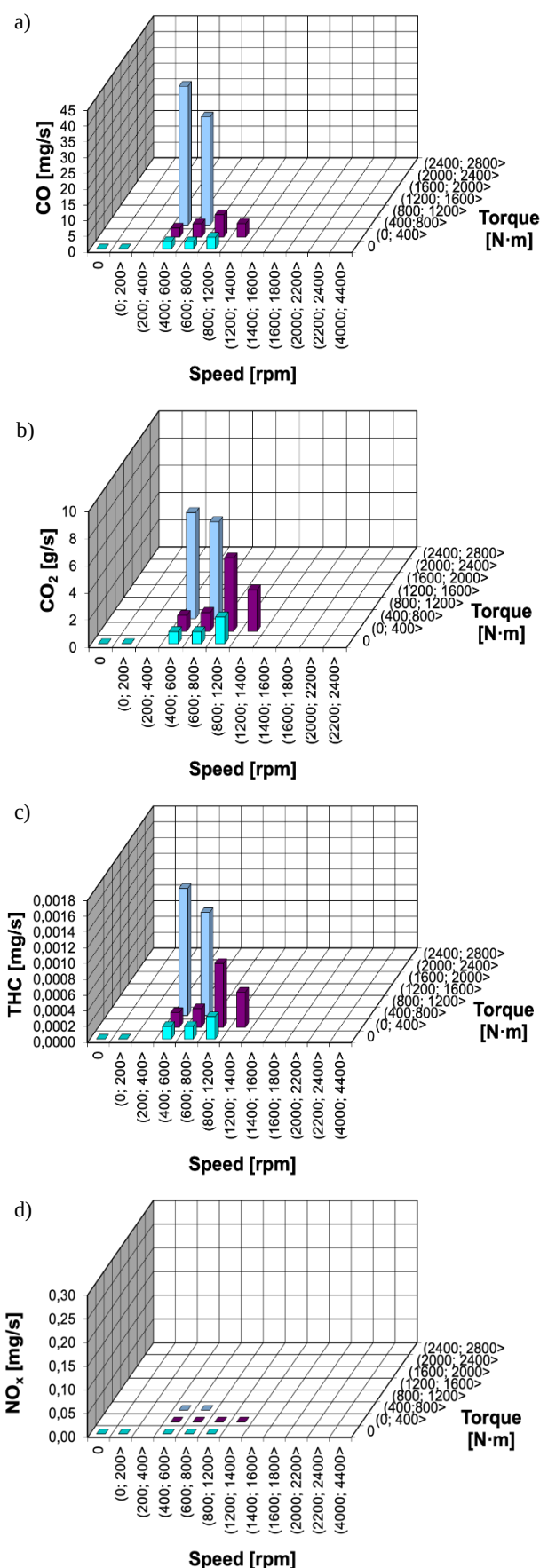


Fig. 7. Characteristics of time density of emissions as a function of engine speed and engine load during unloading: a) CO, b) CO₂, c) THC, d) NO_x

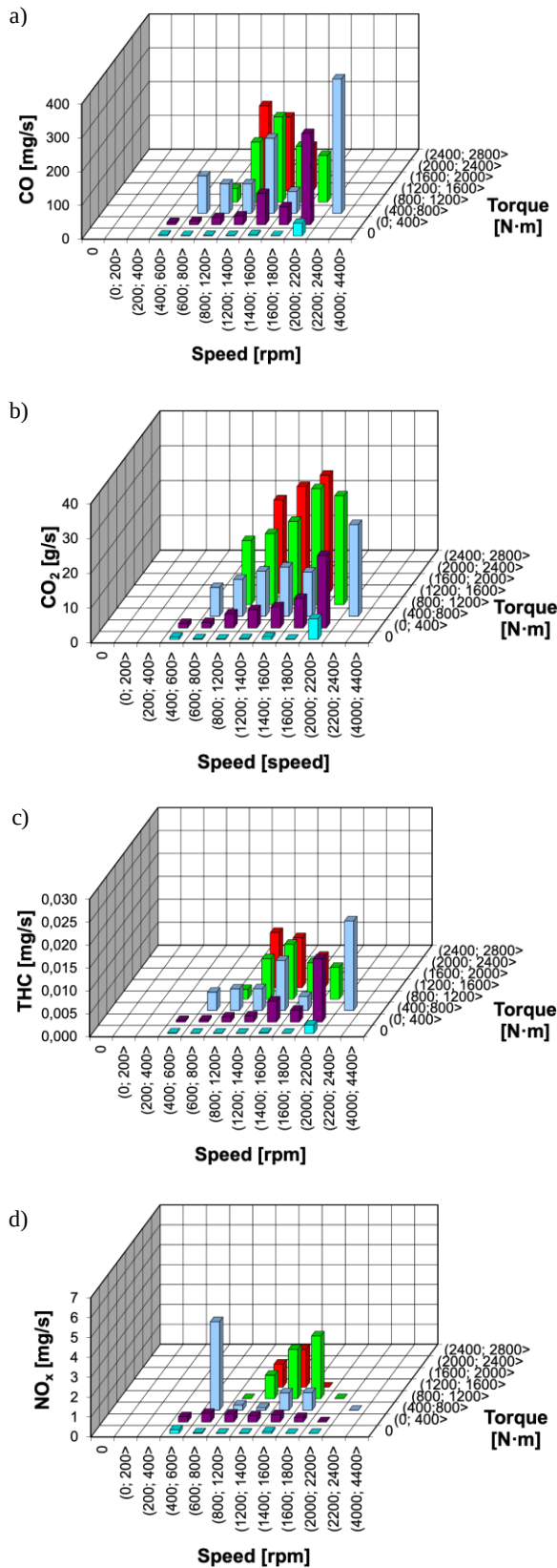


Fig. 8. Characteristics of time density of emissions as a function of crankshaft speed and engine load during travel to the destination (with load): a) CO, b) CO₂, c) THC, d) NO_x

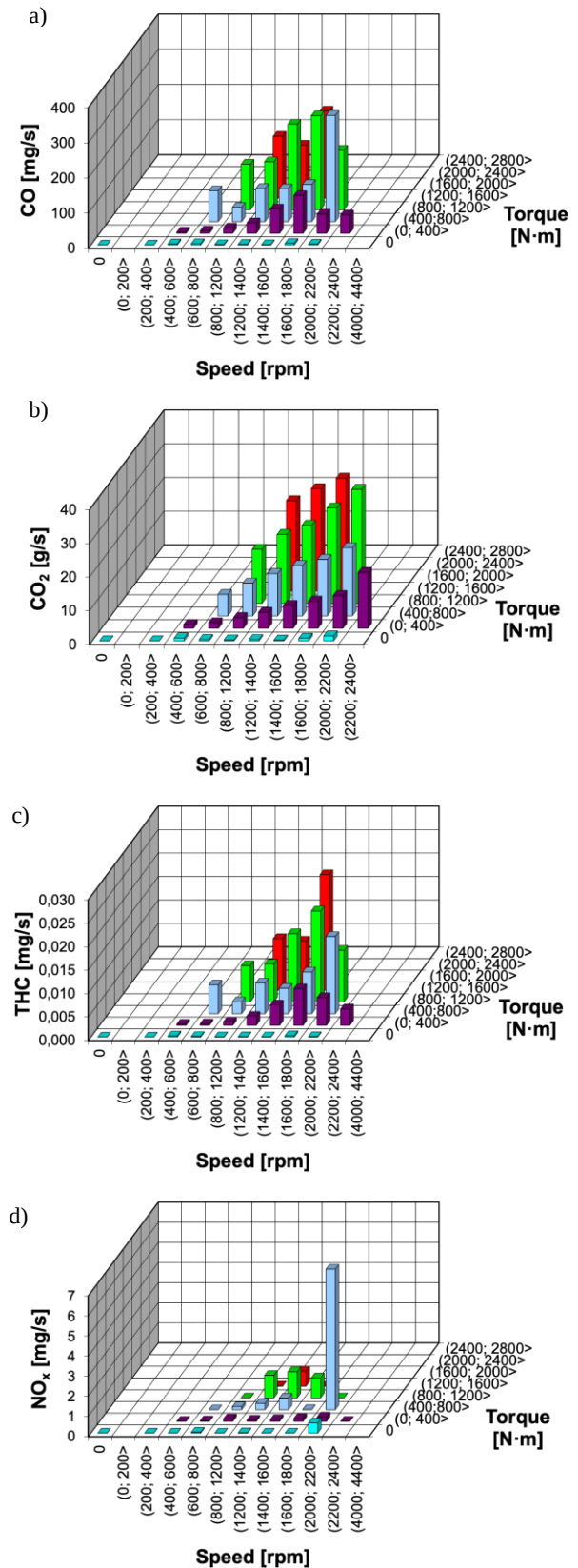


Fig. 9. Characteristics of time density of emissions as a function of crankshaft speed and engine load during travel to base (no load) a) CO b) CO₂ c) THC d) NO_x

Comparative graphs of road emissions of toxic exhaust gases for load transport and return without load were prepared. In both cases, the vehicle travelled just over 23 km (Fig. 10). The measured values are relatively low, with carbon monoxide as the dominant harmful emission. The percentage analysis showed that concrete transport increased road emissions by approximately 80% for CO, 35% for THC, and 150% for NO_x. Higher nitrogen oxide emissions during concrete transport result from higher unit loads and increased combustion temperatures. However, it is worth emphasizing that emissions depend primarily on the engine's thermal load and the SCR system's efficiency, which is designed to reduce nitrogen oxides. In fact, emissions are influenced not only by the weight of the load being transported, but also by the driver's driving style and transitional conditions (e.g., rapid acceleration). THC emissions are low in both cases, confirming the effectiveness of HC purification systems. A similar analysis was performed in [32], which presented studies of heavy vehicle emissions in real traffic conditions. Their studies showed increases of 18–41% in NO_x emissions, 6–67% in CO emissions, and 37–125% in THC emissions. It is important to note that the study was conducted in three variants – empty, half-loaded and fully loaded vehicles in relation to the permissible total mass. The test was based on driving on paved urban, extra-urban and motorway roads. In the tests presented in this paper, the vehicle performs work in accordance with its operating model, travelling on both unpaved and paved roads. In article [16], the authors analysed nitrogen oxide emissions from over 200 heavy goods vehicles. Their research also showed that NO_x emissions depend on the SCR system's efficiency and the driver's driving style, including the number of accelerations and sudden braking events. The results clearly indicate that the emission profile of a heavy-duty truck is closely correlated with operating conditions and the specific nature of the transport task. Despite relatively low total emissions, the noticeable differences between driving with and without a load emphasise the importance of optimising vehicle use to minimise environmental impact. The results show that even slight changes in driving dynamics or route characteristics can lead to significant fluctuations in emissions. Therefore, continuous monitoring and further calibration of emission control systems can contribute to a measurable reduction in exhaust emissions. The observations presented not only confirm the trends noted in earlier studies but also emphasize the importance of measuring actual driving conditions for a reliable assessment of the environmental performance of heavy goods vehicles in specific transport applications.

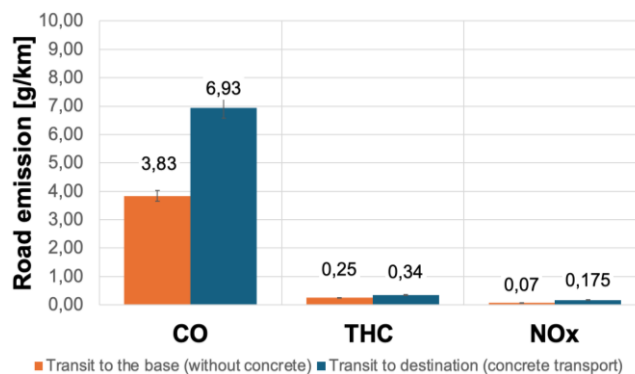


Fig. 10. Road emissions of CO, THC and NO_x

4. Summary

The conducted research on pollutant emissions from a heavy-duty concrete transport vehicle provided important conclusions on the impact of various factors on emission levels and the nature of engine operation under different operating conditions. The results indicate that despite meeting the Euro VI standard, emissions under RDE conditions remain significant under high transport loads. Driving with a full load of concrete is associated with increased fuel consumption and CO emissions, especially during acceleration and uphill driving. Returning without a load results in lower engine load and reduced emissions, though periodic increases are still observed during higher-power demand.

Furthermore, analysis of the engine's performance characteristics showed that changes in crankshaft speed and engine load significantly affect pollutant emissions. High emissions typically occurred at maximum engine load, highlighting the need to monitor and optimise engine operating conditions in order to reduce emissions. The journey to the base, when the vehicle was unloaded, was characterised by lower load values and a wider range of engine speeds. However, even in this case, pollutant emissions remained a significant issue, especially at higher engine speeds.

In addition, significant differences in pollutant emissions were observed across different stages of the route, suggesting that driving dynamics and specific vehicle operating conditions can significantly affect the number of compounds emitted, particularly for HDVs, whose operating models force them to operate in off-road conditions. In such cases, road conditions are often variable and difficult to model, which significantly affects emissions from these vehicles and makes their estimation difficult.

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Nomenclature

CO carbon monoxide
 CO₂ carbon dioxide
 CI compression ignition
 FID flame ionization detector
 GPS global positioning system

HDV heavy duty vehicle
 NDIR non-dispersive infrared
 NO_x nitrogen oxides
 NDUV non-dispersive ultra violet spectroscopy
 NRMM non-road mobile machinery

PEMS portable emissions measurement system
 PM particulate matter
 PN particulate number

THC hydrocarbons
 SCR selective catalytic reduction

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Estimation of the EES parameter value – a case study of the Toyota Yaris Hybrid

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Motor vehicles undergo plastic deformation when they collide with each other or with another obstacle, leading to a loss of the system's kinetic energy. The work of permanent deformation of a vehicle involved in a collision is equal to the loss of kinetic energy and is often represented as the value EES, i.e., Energy Equivalent Speed. This article describes three well-known and commonly used methods for estimating the value of deformation work: comparative, analytical, and graphical methods. The later section presents the calculation of the EES parameter using a comparative approach for a hybrid passenger vehicle struck by a motorcycle. Possible differences in estimating the EES parameter value have been identified using this method for both standard and hybrid vehicles.

Key words: EES parameter, comparative method, hybrid vehicles, car deformation, case study

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1. Introduction

The constantly increasing number of cars on the roads results in a high daily number of accidents. According to the 2024 Annual Police Report [21] on road accidents, the number of passenger vehicles increased by more than 2 million between 2020 and 2023. Although the number of cars fell by 16% between 2023 and 2024, the number of road accidents increased by almost 3% during this period. Moreover, the share of vehicle-to-vehicle collisions remains approximately 53% of total accidents. To enhance safety, new obligations are being imposed on car manufacturers, among other measures. One of these is the obligation to equip new vehicles with so-called black boxes, or EDRs (Event Data Recorders), which are mandatory in the European Union from 7 July 2024. The purpose of this device is to record the vehicle's speed before and during a collision.

However, for older vehicles without a black box installed, calculating collision speed is subject to certain limitations. In the absence of skid marks or evidence of wheel lock-up, which would allow the calculation of speed at the time of the accident, it is possible to estimate the speed based on the extent of vehicle deformation.

As the literature review shows, many authors address the problem of estimating vehicle collision speed using the Energy Equivalent Speed (EES) parameter, presenting both the theoretical foundations and practical applications of selected methods [2, 17]. In [2], a comparative analysis of EES estimation methods for vehicles with different structural materials was performed, demonstrating a significant influence of material characteristics and impact type on the obtained deformation energy values. In turn, [17] presents the results of experimental studies aimed at assessing the applicability and limitations of selected EES methods in the context of changes in vehicle stiffness resulting from technological development.

The practical application of the comparative method is also presented in [4], where collision speed was estimated on the basis of post-accident photographic documentation and damage data for a collision between a Fiat Punto

and a Smart Roadster, classified as vehicles with atypical mass and structural parameters.

Another group of publications comprises studies employing simulation tools based on the finite element method. In [29], a finite element model available in the PC-Crash software was used to determine the EES parameter of a passenger car damaged as a result of a collision with an atypical obstacle, namely the rear of a wheel loader. The authors demonstrated a significant influence of the adopted structural vehicle model on the deformation pattern and the resulting EES value. A similar approach is presented in [16], where the authors emphasize the high sensitivity of simulation results to input parameters and the need to verify simulation outcomes using alternative EES estimation methods.

In many studies, the authors do not limit their analyses to a single computational method. Examples include publications [1, 7, 17], in which the collision speed was estimated using the CRASH3 method and subsequently verified using the comparative method or crash test data. Such an approach enables an assessment of result reliability and facilitates the identification of limitations inherent to individual methods.

The literature also reports alternative approaches to EES estimation, including the application of the Monte Carlo method [26] for uncertainty analysis, the triangle method [25] for simplified energy loss estimation based on deformation geometry, and the total differential method [15] for quantifying uncertainty in EES calculations. An interesting complement to classical approaches is provided in [28], where the possibility of estimating EES solely on the basis of photographic documentation was assessed; however, the authors demonstrated that this method can serve only as an auxiliary tool.

An important source of reference data in EES analyses is provided by crash test reports, particularly studies based on the NHTSA (National Highway Traffic Safety Administration) test databases [15, 20], which enable the evaluation of the influence of vehicle design and age on defor-

mation characteristics and energy absorption during a collision.

A different, more methodological perspective on the problem of EES determination is presented in [18], where the authors focus on analyzing the influence of the deformation profile description on calculation results obtained using the CRASH3 algorithm. In contrast to studies comparing reconstruction methods or computational tools, this publication addresses the selection of an appropriate number of measurement points for describing vehicle deformation.

This study constitutes a complementary contribution to existing research on the EES parameter, shifting the emphasis from the computational model itself to the quality and representativeness of input data, which is particularly important for improving the repeatability and comparability of accident reconstruction results.

The conducted literature review indicates that existing studies on EES determination primarily compare computational methods, simulation tools, and selected input parameters, such as structural material characteristics, vehicle age, and applied technological solutions. Numerous studies have demonstrated that these factors may significantly influence deformation behavior and the amount of energy absorbed during a collision. At the same time, vehicles are typically treated as a homogeneous group, without differentiation by powertrain type.

Meanwhile, hybrid vehicles, compared to vehicles equipped exclusively with internal combustion engines, feature a different structural layout due to additional components, such as traction batteries, electric motors, and structural reinforcements required for their installation. These differences may affect mass distribution, local stiffness, and energy-absorption mechanisms during a collision, and consequently the determined EES value. The lack of consideration of these aspects in previous studies justifies further research focused on assessing the influence of hybrid powertrains on the EES estimation process.

2. Methods of estimating vehicle deformation work

2.1. General proceedings

According to the laws of physics, every moving body has its own velocity and, consequently, kinetic energy [23]. It can be calculated using formula (1):

$$E_k = \frac{1}{2} * m * v^2 \text{ [J]} \quad (1)$$

where: E_k – kinetic energy [J], m – vehicle mass [kg], v – vehicle velocity [m/s].

During collisions with elastic-plastic bodies, kinetic energy is lost within the system. The loss is equal to the total deformation work (W_D) of both vehicles, which can be expressed as (2):

$$\Delta E \cong W_D \quad (2)$$

where: ΔE – a loss of kinetic energy [J], W_D – deformation work [J].

In practice, the work of permanent deformation is usually represented as equivalent to kinetic energy, using the parameter EES (Energy Equivalent Speed) [27]. This parameter is expressed in units of m/s and corresponds to the

speed at which an identical vehicle, when collided with a rigid, non-deformable barrier, would undergo identical plastic deformations [24]. The kinetic energy lost by the vehicle during a collision is entirely absorbed by its deformation (3):

$$E_d = W_D = \frac{m * EES^2}{2} \quad (3)$$

where: E_d – kinetic energy of deformation [J], EES – Energy Equivalent Speed [km/h], W_D and m – like before.

Hence, the value of the EES parameter can be determined from equation (4):

$$EES = \sqrt{\frac{2 * W_D}{m}} \quad (4)$$

where: EES, W_D and m – like before.

To estimate the EES parameter, the following methods can be applied:

1. comparative, using available catalogues containing actual cases of vehicle damage with known W_D
2. graphically, using rasters, i.e., distribution grids of W_D
3. analytical, using appropriate mathematical formulas to calculate W_D depending on the depth of vehicle body deformation [27].

Research carried out [2] indicates that different materials used in vehicle construction have a substantial impact on the magnitude of residual deformation, and therefore on the correct estimation of the EES parameter using the methods discussed. Furthermore, when comparing identical vehicle models from different production years that differ in construction materials, including rigidity, it has been shown that newer vehicles experience less deformation [15]. Therefore, in light of the growing number of "new generation" vehicles on the roads, it is appropriate to review the methods used so far for assessing traffic theory issues related to collisions.

2.2. Comparative method

The comparative method relies heavily on the database to identify the vehicle model that matches the one for which the EES parameter value is sought. The identified vehicle must be identical not only in the make and model but also in the extent of its damage. If an appropriate vehicle cannot be found, the one with the most similar construction and a comparable deformation pattern should be chosen [7]. In this scenario, it is essential to consider the difference in the masses of the cars being compared using the appropriate formula (5) [17]:

$$EES_{\text{vehicle}} = \sqrt{\frac{m_{\text{etalon}}}{m_{\text{vehicle}}}} * EES_{\text{etalon}} \quad (5)$$

where: EES_{vehicle} – EES parameter value sought, m_{etalon} – mass of the vehicle selected from the catalogue [kg], m_{vehicle} – mass of the vehicle for which we are seeking the EES value [kg], EES_{etalon} – EES parameter value for the vehicle selected from the catalogue.

Among the available catalogues, the database in the PC-Crash program [22], Dr. Melegh's database [6], or the online databases [8–10, 19] can be mentioned, which, in addition to the Melegh catalogue data, also contain other cases. This method is considered the simplest and is there-

fore the most commonly used. However, its drawback is the limited range of vehicles available in the databases, mainly the absence of the latest models.

2.3. Graphic method

To apply the graphic method, it is essential to use deformation work distribution grids of the vehicle's body. Each section of the grid shows the work required to deform that part of the vehicle [27]. In 1974, Campbell [3] proposed a grid model assuming an even distribution of deformation energy across the vehicle's width and height. Subsequently, Röhlich developed this model based on crash-test results, indicating that the deformation energy does not distribute evenly [13]. He developed deformation work distribution grids for the following areas of the vehicle: front, rear, sides, front corners, and rear corners. Currently, the use of these grids is limited, as they are suitable only for vehicles weighing 950 kg with rear-wheel drive and a front-mounted engine [27]. Only A-segment vehicles, including the Toyota Aygo, Fiat 500, and Peugeot 107, currently comply with the mass criterion. It is also necessary to consider the structural and technological differences between current vehicles and designs from the 1970s and 1980s.

2.4. Analytical method

Analytical methods constitute the largest group of methods for estimating the EES parameter. Various authors suggest applying different mathematical models or combining certain methods, such as comparative and analytical methods.

An example is [4], where the author calculated the EES parameter for the unique sports vehicle, the Smart Roadster. This car collided with a Fiat Punto, for which the EES parameter was estimated by comparing it with three other similar vehicles. Knowing that the maximum force acting on both vehicles during the collision is the same for each [11], the author first calculated the plastic component of the deformation energy for the Fiat vehicle and, based on this, determined the EES parameter for the Smart vehicle.

Meanwhile, Wach [26] presents the Monte Carlo method, which involves repeatedly performing calculations using the same mathematical model. Calculation data are randomly selected from specified uncertainty ranges. This method is used, for example, in the PC-Crash program to obtain optimal pre-collision movement parameters of the vehicles involved in the incident. This program performs 100 iterations, allowing it to avoid situations where other algorithms might get stuck in a local minimum [5].

Vangi describes a method known as the "triangle method" [25], which combines a visual assessment of vehicle deformation with methods for measuring residual deformation to evaluate energy loss in any vehicle deformation profile. In this method, a stiffness curve is determined for a reference vehicle, the EES parameter of the tested vehicle is then established, and the deformation energy is calculated.

The vehicle's deformation energy can also be calculated using the CRASH3 algorithm, based on NHTSA's published results [5]. This is a computer program from CALSPAN Corporation, whose name is an acronym for Calspan Reconstruction of Accident Speeds on the High-

way. It allows analysis of vehicle collisions, determining their speed, energy, and direction of movement using analytical techniques [14]. Many factors influence the determination of EES using the CRASH3 computational algorithm, particularly stiffness, availability of studied or comparable vehicles, and the limited range of crash tests performed [18]. As demonstrated in [12], the uncertainty in estimating the EES parameter using the described method is the lowest among commonly applied approaches, such as the simplified method or the method implemented in the PC-CRASH software.

3. Case study for a hybrid vehicle

The results of calculating the EES parameter for the 2018 Toyota Yaris Hybrid vehicle, which was struck by a Honda CBR600F motorcycle, are summarized below. The vehicles involved in the incident are shown in Fig. 1 and Fig. 2, respectively. Before the collision, the motorcycle was traveling on a main road in a built-up area with a 50 km/h speed limit. The Toyota Yaris Hybrid was stopped on a secondary road, right next to the intersection with the main road. The driver initiated a left turn onto the main road. While at the intersection, he noticed a motorcycle approaching. The driver decided to stop the vehicle within the motorcycle's lane of travel. The motorcycle driver failed to avoid the vehicle, resulting in a head-on collision. The motorcycle struck the center of the Toyota's front bumper.

In the analysis of damage to the Honda CBR600F motorcycle, an empirical formula (6) cited in the literature [24] was applied, which allows estimation of the motorcycle's collision speed based on the reduction in the wheelbase length between its front and rear wheels:

$$v_k = 1.38 \cdot d + 16.58 \quad (6)$$

where: v_k – motorcycle's collision speed, d – reduction of the wheelbase [cm].

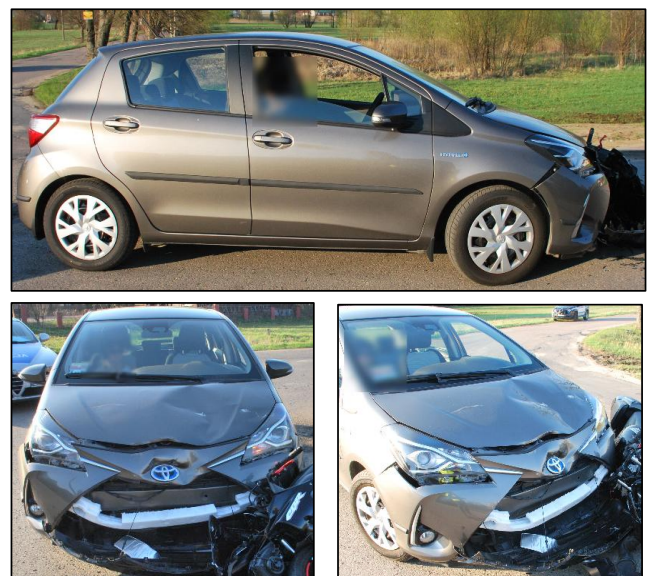


Fig. 1. Illustration of the deformation of the Toyota Yaris Hybrid vehicle



Fig. 2. Illustration of the deformation of the Honda CBR600F motorcycle

The wheelbase reduction for this motorcycle was 11 cm $\pm$ 1 cm. The motorcycle's collision speed was determined to be 30 to 33 km/h.

Measuring the deformation depth of the Toyota Yaris Hybrid was impossible because only photos of the damaged vehicle were available. Moreover, the available photographs were insufficient to generate a point cloud, which would have allowed an approximate measurement of these depths using computer software. Deformation meshes (rasters) were unavailable for this vehicle, so it was impossible to determine the resulting deformation work (W_D). The only option in this case was to use a comparative method.

The same vehicle model, i.e. the Toyota Yaris Hybrid, was not found in the available EES databases. To implement the comparative method, several similar vehicles were selected from the DSD ReconData database [22] and the Meleggha database [6]. The selected vehicles had similar damage – deformed front ends, including the front bumper and engine cover. Vehicles with a weight within $\pm 20\%$ of the target vehicle's weight were selected for comparison. The accepted mass deviation was due to data availability – only 5 similar vehicles were available within the $\pm 10\%$ mass range, which was deemed an insufficient sample. On the other hand, accepting a deviation beyond 20% signified taking into account vehicles that were too structurally dissimilar, which went against the aim of the comparison.

As mentioned in the theoretical introduction, a drawback of these databases is the absence of the latest vehicles. It should therefore come as no surprise that all the vehicles used were significantly older than the vehicle under study – the selected vehicles were from 1998 to 2010. Furthermore, none of the vehicles selected for comparison were hybrid vehicles. The list of selected vehicles for comparison is shown in Fig. 3 (from the DSD ReconData database) and in Fig. 4 (from the Meleggha database).

The EES parameter for the Toyota Yaris Hybrid vehicle was then calculated using formula (5). Due to the lack of precise mass data for this vehicle, a vehicle mass of 1100 kg was assumed for calculations, based on the manufacturer-provided average mass for this vehicle model. The values of the m_{etalon} and EES_{etalon} parameters were taken from the appropriate databases. The results of the calculations are presented in Table 1.

The calculations presented indicate that the EES parameter for the Toyota Yaris Hybrid was within the range of 15–29 km/h, with an average of 20 km/h. Since the Toyota vehicle in the study was stationary at the moment of impact, the results should be compared with the Honda motorcy-

cle's collision speed. Thus, it can be stated that the average speed at which damage occurred to the Toyota vehicle, estimated using the comparative method, is approximately 33–39% lower than the calculated collision speed of the motorcycle (30–33 km/h).

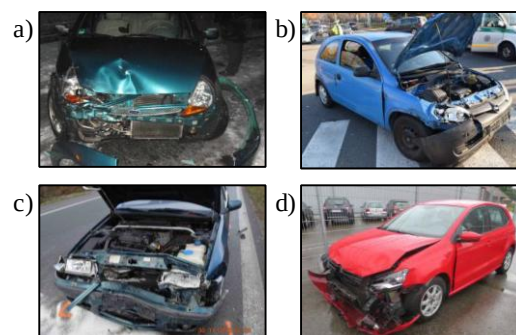


Fig. 3. Selected cases of deformed vehicles from the DSD ReconData database: a) Ford Ka, b) Opel Corsa, c) Skoda Felicia, d) VW Polo

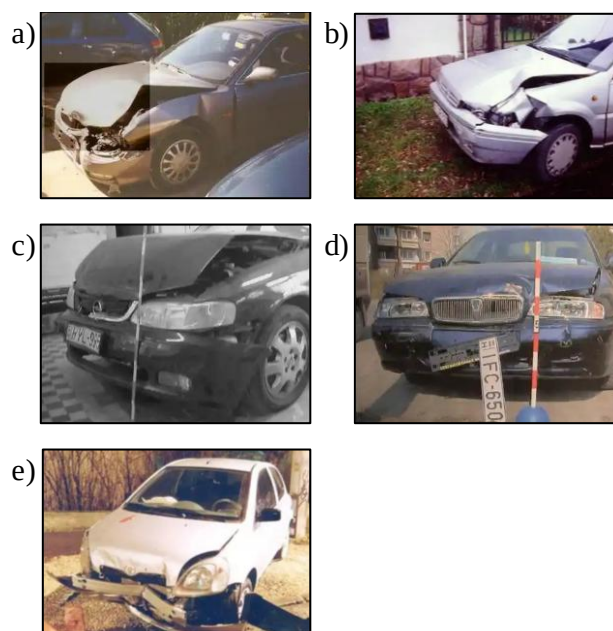


Fig. 4. Selected cases of deformed vehicles from Dr. Meleggh's database: a) Nissan Almera, b) Nissan Almera, c) Opel Vectra, d) Rover 400, e) Toyota Yaris

Table 1. Calculation results of the EES parameter for the Toyota Yaris Hybrid vehicle

Brand & model	Year of production	m_{etalon} [m]	EES_{etalon} [km/h]	$EES_{vehicle}$ [km/h]
Ford Ka	2001	946	20	19
Opel Corsa	2001	905	17	15
Skoda Felicia	1998	985	23	22
VW Polo	2010	1077	20	20
Mazda Xedos	1995	1145	21	21
Nissan Almera	1997	1060	16	16
Opel Vectra	1998	1270	27	29
Rover 400	2001	1120	16	16
Toyota Yaris	2002	1000	24	23

The vehicles selected for comparison, despite having similar parameters (external design, mass), sustained simi-

lar damage at lower collision speeds. This shows that the Toyota Yaris Hybrid is more resistant to deformation than older vehicles, and consequently, a comparable damage pattern required greater deformation work of the body structure. Construction materials used in older vehicles were much more prone to deformation.

The most similar result ($EES_{\text{vehicle}} = 29 \text{ km/h}$) was obtained from the comparison of the Opel Vectra B. It can be concluded that, structurally, this vehicle most closely resembles the Toyota Yaris Hybrid.

The results obtained are therefore subject to an error exceeding 10%, hence their use in the analysed case is not appropriate. It should be noted that the well-known methods for estimating the EES parameter for newer vehicle designs, including hybrid vehicles, may not always be appropriate. There is, therefore, a need to conduct a more comprehensive analysis of the influence of the construction parameters of newer-generation vehicles on the range of their deformation following a collision.

4. Final conclusions

Among the available methods for estimating the EES parameter, the best results are obtained using analytical methods. However, their use requires measuring the actual depth of vehicle deformation. Without this capability, the chance of correctly determining the EES parameter decreases. An alternative is to generate a point cloud and measure the approximate depth of deformation using appropriate computer software. However, this requires appropriate photographic material, i.e., a sufficient number of photos of the damaged vehicle, taken from various angles.

If the available input data is insufficient, the comparative method remains. Still, it is important to consider that a suitable vehicle may not be found in the available databases. This is especially true when estimating the EES

parameter for newer generation vehicles, including hybrid or electric vehicles.

It is advisable to search for more relevant cases in the catalogues. The more similar the cases are, the fewer errors the results should have. Nevertheless, it should not be forgotten that vehicles similar in shape, mass, and extent of damage differ in their internal construction and materials used, which affects the value of the EES parameter.

In the analyzed case, where a Toyota Yaris Hybrid collided with a Honda CBR600F motorcycle, due to insufficient photographic material and the lack of precise deformation depth, the only option was to use the comparative method. No identical vehicles were found in the available databases, either by model or by design (hybrid vehicle). The vehicles most similar in appearance, weight, and extent of damage were older ones, aged 8–20 years.

The results showed that the reference material used was inadequate. The obtained EES values differed by over 30% from the collision speed calculated for the motorcycle. This discrepancy in results may indicate differences in body stiffness between the tested vehicle and the comparison vehicles.

The conducted analysis indicates that the currently used methods for estimating the EES parameter do not explicitly account for structural differences arising from vehicle drive systems. Hybrid vehicles, compared to vehicles equipped exclusively with internal combustion engines, exhibit a different mass distribution and additional structural components, such as traction batteries and electric motors, which may affect deformation behavior and the amount of energy absorbed during a collision. The results confirm that neglecting these differences in selecting reference vehicles or comparative data may lead to significant discrepancies in the determined EES values, underscoring the need for further research on vehicles equipped with alternative drive systems.

Nomenclature

EDR event data recorder
EES energy equivalent speed

NHTSA National Highway Traffic Safety Administration

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Diesel engine performance prediction using fuel blends with a support vector machine

ARTICLE INFO

The urgent need for environmentally friendly alternative fuels arises from concerns over fossil fuel depletion and the harmful effects of exhaust gas emissions. One promising solution involves blending biodiesel with natural additives, such as essential oils, to improve combustion efficiency in diesel engines. However, conventional engine performance testing is often time-consuming and repetitive, requiring a more efficient approach such as predictive modeling using Machine Learning (ML). This study investigates the effect of biodiesel and essential oil blend ratios on engine performance and develops predictive models to estimate engine power and torque. A total of 1045 experimental data points were collected from tests using varying compositions of biosolar, Dexlite, and essential oil additives at different RPMs. The research applied a quantitative method, utilizing Linear Regression and Support Vector Machine (SVM) algorithms implemented in RStudio. Model accuracy was evaluated using MAE, RMSE, and R^2 metrics. The results indicate that variations in biodiesel blend ratios and essential oil additives had no statistically significant effect on engine performance. However, ML-based predictive modeling proved highly effective. The SVM model achieved superior accuracy, with R^2 values of 0.993 for power and 0.973 for torque. In contrast, the Linear Regression model yielded much lower R^2 values – 0.671 for power and 0.093 for torque. These findings demonstrate the potential of Machine Learning, particularly SVM, as a reliable tool for predicting diesel engine performance with biodiesel and additive blends, offering a faster, more accurate alternative to conventional testing methods.

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1. Introduction

Fossil fuels, such as coal, crude oil, and natural gas, have historically been a critical component of the global energy infrastructure, accounting for 81% of the world's primary energy supply in 2019, according to the International Energy Agency [22]. In Indonesia, fuel oil consumption increased by 3.19% in the third quarter of 2021, reaching $48.59 \times 10^6 \text{ m}^3$ compared to the same period the previous year [28]. While these fuels have driven economic growth and urban development, their use has led to significant environmental issues, primarily the rise in carbon dioxide emissions that fuels climate change [27]. This urgency, coupled with the need for energy security, has spurred discussions about the future of energy sources. Renewable energies like solar, wind, hydro, geothermal, and biomass are playing a vital role in this transition, representing a shift toward sustainable energy production and a balanced relationship with the environment [5, 24, 29].

As industries remain dependent on power generation and society on transportation and fossil fuel-based products, the search for alternative energy sources continues. Biodiesel, known as fatty acid methyl ester (FAME), has garnered attention as a viable alternative to traditional fuels, particularly for transportation, due to its potential to reduce greenhouse gas emissions [7]. Biodiesel can reduce carbon monoxide (CO) emissions by 8–41% [9]. This renewable fuel can be produced from various biological materials, including used cooking oil, vegetable seed oils, animal fats, and microalgal oil [11]. Importantly, modern diesel engines can operate on biodiesel without requiring modifications [1]. Researchers recommend using biodiesel as a full or

partial substitute for petroleum diesel, as it effectively reduces exhaust emissions [6, 12].

Challenges in the transportation sector include higher nitrogen oxide emissions compared to diesel, reduced engine performance in cold weather, sediment deposition that can clog storage tanks and fuel lines, and lower fuel density, which necessitates greater fuel consumption for equivalent power output [9]. The physicochemical properties of biodiesel can vary significantly depending on the feedstock and production method, thereby directly affecting its performance in diesel engines [7]. Key properties such as viscosity, density, calorific value, and oxidation stability affect combustion efficiency, engine start-up performance, fuel consumption, emission characteristics, and long-term reliability [6].

In a diesel engine, the combustion process of biodiesel influences thermal efficiency, power output, and exhaust gas emissions. The higher oxygen content in biodiesel promotes more complete combustion, leading to increased Brake Thermal Efficiency (BTE) and effectively reducing emissions from four-stroke compression engines, particularly unburned hydrocarbons, carbon monoxide, and nitrogen oxides [6]. Therefore, it is essential to consider both the physicochemical properties and combustion characteristics of biodiesel during engine optimization, and to establish accurate predictive models to evaluate its overall performance [13]. Determining optimal biodiesel composition through experimental methods is time-consuming and labor-intensive. Advances in computational approaches, particularly computational fluid dynamics (CFD) and Machine Learning (ML), enable faster and more reliable prediction of biodiesel properties [1, 7]. These methods pro-

vide a robust basis for optimizing biodiesel formulations, improving efficiency and engine performance [9, 17, 21, 32].

The development of predictive models plays a key role in improving production efficiency, product quality, scalability, cost, and sustainability in modern industrial applications [6]. However, their effectiveness depends strongly on the selected algorithm and data characteristics. Linear regression offers simplicity and interpretability but is limited to linear relationships. Support Vector Machines (SVMs) perform well on small to medium datasets and can handle non-linear patterns via kernel functions, though they require careful parameter tuning. In contrast, Artificial Neural Networks (ANNs) are better suited for capturing complex non-linear relationships between biodiesel properties and engine performance, but they require large datasets and higher computational costs [2, 8]. Therefore, the choice of model depends on criteria such as data size, nonlinearity, interpretability, and computational resources. By selecting appropriate methods, Machine Learning models can accurately predict key fuel properties, including calorific value, viscosity, and oxidation stability, enabling more efficient optimization of biodiesel production [1].

2. Experimental

2.1. Data source and experimental setup

This study utilized a quantitative approach based on a dataset derived from primary experiments. The data was sourced from engine performance tests conducted at the Engine Performance Laboratory, Department of Mechanical Engineering, Universitas Negeri Semarang, Indonesia. The engine specifications are shown in Table 1. In this study, the diesel engine is a vehicle engine. The engine's fuel system is not naturally aspirated (NA), but rather uses a common-rail system, a modern fuel-injection system in which diesel fuel is stored in a single high-pressure pipe (rail) and distributed electronically to each injector. This system allows for more precise injection control, increasing efficiency and power while reducing emissions.

Table 1. Diesel engine specifications in vehicles

Engine model	Toyota Diesel 2 KD-FTV
Engine type	4 stroke, in-line type, 16 valve DOHC
Cylinder number	4-cylinder
Injection system	Common Rail
Displacement	2494 cm ³
Bore and stroke	92 mm × 93.8 mm
Compression ratio	18.5:1

The population for this study consisted of all experimental data on diesel engine performance using biodiesel (biosolar and Dexlite) fuel blends and an essential oil additive. The fuel was mixed in a stationary-bed reactor at 300 rpm and 60°C for 30 minutes. The fuel mixture is used in the vehicle and then tested for performance, namely, torque and power, on a chassis dynamometer.

A total of 1045 data samples were selected for the analysis using purposive sampling. The selection criteria required the data to include parameters for rpm-engine power and rpm-torque. Data analysis using R Studio. The research variables were defined as follows:

- Independent Variables: The primary independent variables were fuel process, consisting of ten biodiesel–Dexlite blend ratios with and without essential oil additives. The 10 fuel blend variations used in the experiments are detailed in Table 2. The table shows the fuel used to collect engine performance data. A total of 1,045 data points were collected from performance tests, including rpm and engine power. Density testing uses the ASTM D1298 standard, and viscosity testing uses the ASTM D445 standard
- Dependent Variables: The dependent variables were the key engine performance metrics: torque [Nm] and power [kW]
- Control Variables: To ensure consistency across experiments, the test engine, engine speed [rpm], and engine load were held constant.

Table 2. Composition of experimental fuel blends

Blend No.	Biodiesel [%]	Dexlite [%]	Essential oil additive [%]	Density [kg/m ³]	Viscosity [cSt]
1	90	10	0.015	922.7	5.035
2	80	20	0.015	930.7	4.934
3	70	30	0.015	925.3	4.755
4	60	40	0.015	928	4.425
5	50	50	0.015	926.7	4.336
6	90	10	0	934.7	5.328
7	80	20	0	932.7	5.341
8	70	30	0	930.7	5.328
9	60	40	0	928	5.226
10	50	50	0	926.7	5.022

2.2. Machine learning approach

A predictive modeling workflow was designed to estimate engine performance based on the fuel composition. The procedure involved data preprocessing, model training, and evaluation. To provide a clearer overview of the research workflow, a flowchart is presented in Fig. 1. The raw experimental dataset underwent several preprocessing steps to prepare it for Machine Learning analysis. This included data cleaning to handle any incomplete or irrelevant entries and normalization to scale the numerical features to a uniform range. The dataset was then partitioned into 75% for training and 25% for testing and validation.

To guarantee that the visualization accurately and distortion-free depicts the experimental results, the labels and data representation were kept clear. Repeated measurements were used to ensure the dataset's reliability; each test was performed three times under carefully controlled operational conditions to ensure consistency. The alignment of the observed trends with recognized engine performance metrics further supports the data's veracity. The non-linear relationship between input parameters (fuel mix ratio, engine speed, and additive condition) and output performance (power and torque) was modeled using support vector regression (SVR). The model's efficacy in precisely estimating engine performance was confirmed by evaluating its prediction performance using RMSE, MAE, and R².

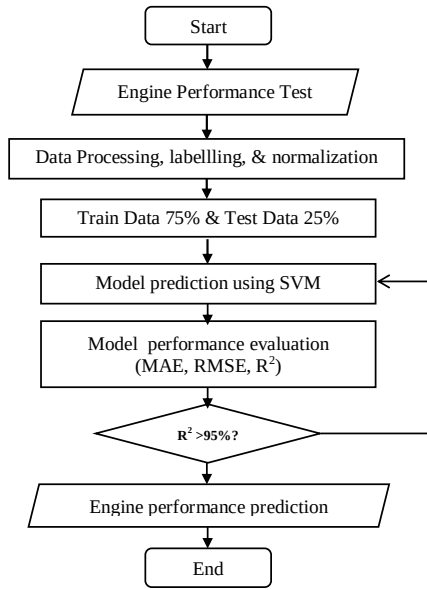


Fig. 1. Machine learning workflow for engine performance prediction

2.3. Support vector regression (SVR) model

A supervised learning approach called Support Vector Regression (SVR) applies Support Vector Machines to regression issues. Finding a function $f(x)$ that minimizes model complexity while approximating the target values within a given tolerance level – defined by the ε -insensitive loss function – is the goal of SVR. By defining a tolerance margin in which errors are not penalized, the parameter ε enables the model to ignore minor deviations and thereby improve generalization.

The fundamental goal is to find a regression function that is as flat as possible. For non-linear data, this function takes the form:

$$f(x) = w^T x + b \quad (1)$$

Here, w^T represents the weight vector, b is the bias term, and x is a non-linear function that maps the input features, x , into a higher-dimensional space where the relationship may become linear. Instead of solving for w directly in the high-dimensional space, SVR solves a more manageable convex optimization problem known as the dual problem. This is achieved using Lagrange multipliers, where the goal is to maximize the following objective function, L , with respect to the Lagrange multipliers α_i and α_i^* :

$$L = -\frac{1}{2} \sum_{i=1}^N \sum_{j=1}^N (\alpha_i - \alpha_i^*)(\alpha_j - \alpha_j^*) K(x_i, x_j) - \varepsilon \sum_{i=1}^N (\alpha_i + \alpha_i^*) + \sum_{i=1}^N y_i (\alpha_i - \alpha_i^*) \quad (2)$$

This maximization is performed under the following constraints:

$$\sum_{i=1}^N (\alpha_i - \alpha_i^*) = 0 \quad (3)$$

where, $0 \leq \alpha_i, \alpha_i^* \leq C$, and for $i = 1, \dots, N$. In this formulation, C is a user-defined hyperparameter that controls the penalty for data points that lie outside the ε -tube, balancing model complexity against prediction error.

The term $K(x_i, x_j)$ represents the Kernel function, a crucial component that calculates the similarity or inner product between two data points, x_i and x_j , in the higher-

dimensional space without explicitly computing the transformation x . This study utilized the Radial Basis Function (RBF) kernel, which is defined as:

$$K(x_i, x_j) = \exp(-\gamma \|x_i - x_j\|^2) \quad (4)$$

Here, γ is a kernel hyperparameter that defines the influence of a single training example. After solving the optimization problem, the data points for which the corresponding Lagrange multipliers (α_i or α_i^*) are non-zero and are designated as Support Vectors (SV). These are the only data points that contribute to the final model and define the boundaries of the regression function.

Once the non-zero Lagrange multipliers are found, the bias term, b , can be determined. This is achieved by selecting a Support Vector, x_k , that lies on the boundary of the ε -tube (i.e., its corresponding multiplier satisfies $0 < \alpha_k < C$) and solving for b using the following relationship:

$$b = y_k - \sum_{i \in SVs} (\alpha_i - \alpha_i^*) K(x_i, x_k) \pm \quad (5)$$

The choice of using $+\varepsilon$ or $-\varepsilon$ depends on whether the support vector lies on the upper or lower boundary of the margin.

With the Support Vectors, their corresponding weights ($\alpha_i - \alpha_i^*$), and the bias term (b) is determined, the model is complete. The prediction for a new, unseen data point, x_{new} , is then calculated using the final SVR function:

$$f(x_{new}) = \sum_{i \in SVs} (\alpha_i - \alpha_i^*) K(x_i, x_{new}) + b \quad (6)$$

This function generates a prediction by computing a weighted sum of the similarities (as defined by the kernel function) between the new data point and each established Support Vector.

2.4. Linear regression model

Linear regression is a fundamental statistical and machine learning algorithm used to model the relationship between a dependent variable and one or more independent variables. In this study, Multiple Linear Regression (MLR) was employed as a baseline model to predict engine performance. The core assumption of MLR is that there is a linear relationship between the input features and the output variable. The following equation represents the mathematical model for Multiple Linear Regression:

$$Y = \beta_0 + \beta_1 X_1 + \beta_2 X_2 + \dots + \beta_p X_p + \varepsilon \quad (7)$$

where: Y is the dependent (target) variable, representing either engine power or torque; $X_1, X_2, \dots, X_p$ are the independent (predictor) variables, which include fuel composition and engine speed (RPM); β_0 is the intercept, representing the predicted value of Y when all predictor variables are zero; $\beta_1, \beta_2, \dots, \beta_p$ are the regression coefficients, each coefficient β_j represents the expected change in Y for a one-unit change in the corresponding predictor X_j , while holding all other predictors constant; ε is the model's error term, which captures the random variation in Y that cannot be explained by the linear relationship with the predictors.

The primary goal during the model training phase is to estimate the optimal values for the coefficients

$(\beta_0, \beta_1, \dots, \beta_p)$. This is typically achieved using the Ordinary Least Squares method. OLS works by finding the coefficients that minimize the sum of the squared differences between the observed actual values (y_i) and the values predicted by the model ($\hat{y}_i$). This is also known as minimizing the Residual Sum of Squares (RSS):

$$\text{Minimize RSS} = \sum_{i=1}^n (y_i - \hat{y}_i)^2 = \sum_{i=1}^n \left(y_i - (\beta_0 + \beta_1 x_{i1} + \dots + \beta_p x_{ip}) \right)^2 \quad (8)$$

Once the optimal coefficients $(\hat{\beta}_0, \hat{\beta}_1, \dots, \hat{\beta}_p)$ have been estimated from the training data, the model can be used to predict new data. The prediction function is simply the regression equation without the error term:

$$\hat{Y} = \hat{\beta}_0 + \hat{\beta}_1 X_1 + \hat{\beta}_2 X_2 + \dots + \hat{\beta}_p X_p \quad (9)$$

While computationally simple and highly interpretable, Linear Regression's performance is fundamentally limited by its core assumption of linearity. It is best suited for problems where the underlying relationship between inputs and outputs is indeed linear.

2.5. Model evaluation

The performance of the trained models was assessed using K-Fold cross-validation to ensure generalization and prevent overfitting. The predictive accuracy of the Linear Regression and SVM models was quantitatively compared using the following standard regression metrics:

- Mean Absolute Error (MAE): Measures the average absolute difference between the predicted and actual values

$$\text{MAE} = \frac{1}{N} \sum_{i=1}^N |y_i - \hat{y}_i| \quad (10)$$

- Root Mean Squared Error (RMSE): Represents the square root of the average of squared differences, providing an error measure in the same units as the target variable

$$\text{RMSE} = \sqrt{\frac{1}{n} \sum_{i=1}^n (y_i - \hat{y}_i)^2} \quad (11)$$

- R-Squared (R^2): Indicates the proportion of the variance in the dependent variable that is predictable from the independent variables

$$R^2 = 1 - \frac{\sum (y_i - \hat{y}_i)^2}{\sum (y_i - \bar{y})^2} \quad (12)$$

3. Results

This section presents the performance evaluation of the linear regression and support vector machine (SVM) models for predicting engine power and torque. The analysis includes a quantitative comparison using standard metrics, a visual inspection of predictive accuracy, and a discussion of the results in the context of existing literature. The results of the engine performance test are shown in Fig. 2. The graphs and data show that engine power increases with increasing revs from 2000 to approximately 5000 rpm, then decreases at higher revs (≥ 5200 rpm) for all fuel variations.

Lowest power occurs at 2000 rpm, while maximum power is in the range of 66 to 70 kW, with variations 6 to 10 generally producing higher power than variations 1 to 5 across most of the rev range.

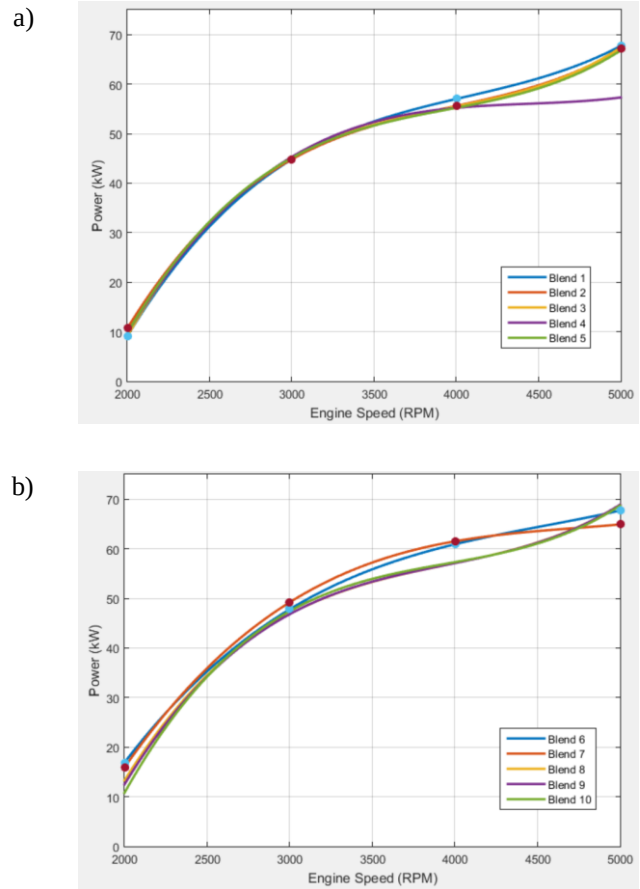


Fig. 2. Engine power trends with rising engine speed for various fuel blends: (a) blends 1–5; (b) blends 6–10

3.1. Statistical analysis of fuel blend effects

To assess the impact of fuel composition on engine performance, a series of nonparametric statistical tests was conducted. Assumption tests, including the Shapiro-Wilk test for normality and Levene's test for homogeneity of variances, were performed first to determine the appropriate inferential tests. The data for both power and torque were found not to be normally distributed ($p > 0.001$), necessitating the use of nonparametric methods. A summary of the key statistical test results is provided in Table 3.

The Wilcoxon rank-sum test confirmed these observations, yielding p -values of 0.732 for power and 0.840 for torque, both well above the 0.05 significance threshold. Furthermore, the calculated effect sizes were negligible for both power (Cohen's $d = -0.016$) and torque (Cohen's $d = 0.002$), indicating that the additive's practical impact was insignificant. The boxplot visualizations in Fig. 3 and Fig. 4 further illustrate the similarity in data distribution between the two groups, supporting the statistical findings.

Table 3. Summary of statistical test results

Analysis	Variable	Test	p-value	Finding
Additive Effect	Power (BHP)	Wilcoxon	0.732	Not Significant
	Torque (Nm)	Wilcoxon	0.840	Not Significant
Blend Ratio Effect	Power (BHP)	Kruskal-Wallis	0.782	Not Significant
	Torque (Nm)	Kruskal-Wallis	0.0504	Not Significant

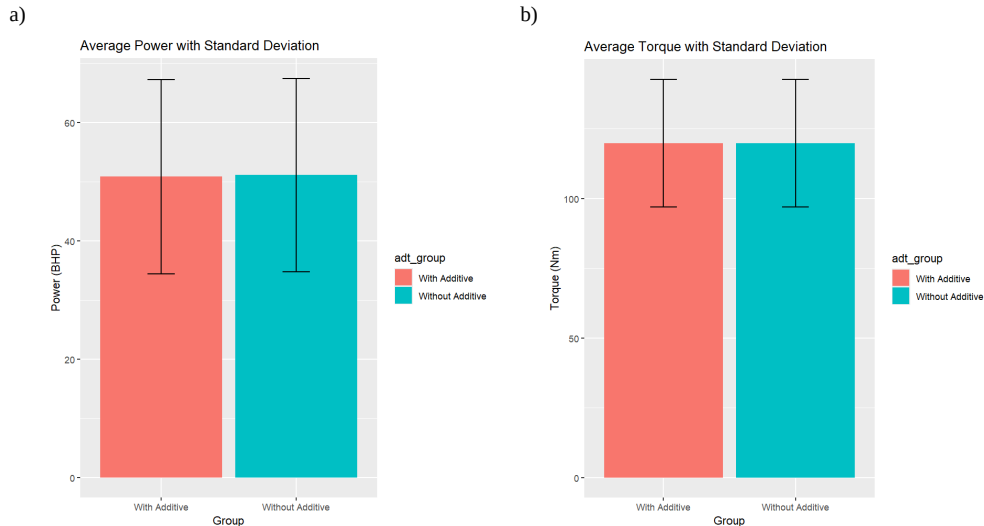


Fig. 3. Statistical comparison of engine performance parameters, showing mean power (a) and torque (b) with standard deviation for fuel blends under additive and non-additive conditions

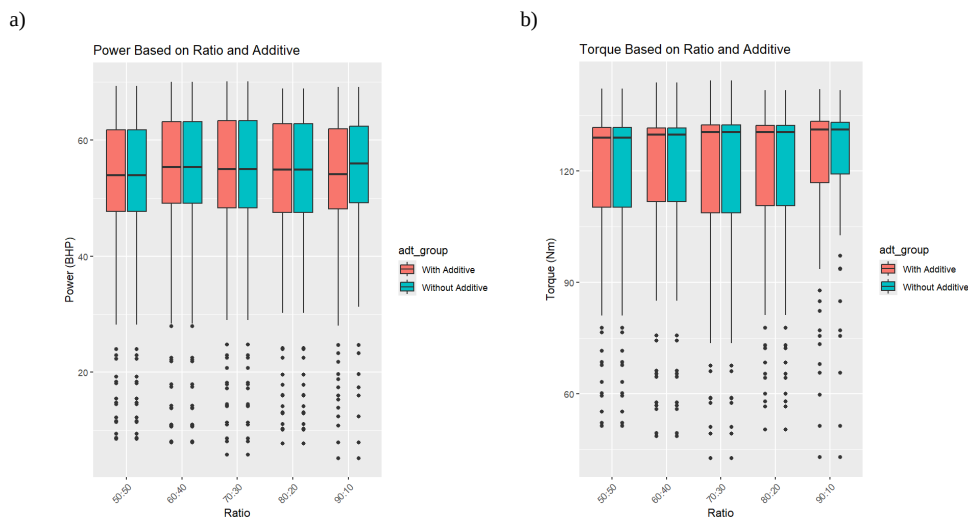


Fig. 4. Statistical distribution of engine performance parameters, showing boxplot comparisons of power (a) and torque (b) across fuel blend ratios under additive and non-additive conditions

This lack of a significant effect aligns with previous research. For example, [3] noted that the addition of essential oils to biodiesel does not consistently improve diesel engine performance across all conditions. Similarly, [4, 19, 20], stated that the effectiveness of essential oil bio-additives is highly dependent on the dosage and their compatibility with the base fuel. The findings are also consistent with [14, 25], who argued that many additives have a more pronounced effect on thermal efficiency and emissivity than on primary performance metrics such as power and torque.

The influence of varying the biodiesel-dexlite blend ratio was also found to be not statistically significant. The Kruskal-Wallis test yielded a p-value of 0.782, indicating no significant differences among the five blend ratios. For torque,

the test yielded a p-value of 0.0504, which is marginally significant. While this borderline result might suggest a subtle trend, it is not statistically strong enough to reject the null hypothesis at a 95% confidence level. The boxplots in Fig. 4 and Fig. 5, which detail performance by ratio, visually support these findings. No clear trend or significant difference in the distribution of power or torque is observable across the different blend ratios. These results are consistent with prior studies [18, 26]. Both found that variations in biodiesel blend ratios (from B20 to B100) often lead to minor, inconsistent fluctuations in power that are not statistically significant. Likewise [15] observed no significant power trend when testing various corn biodiesel blends.

Across various fuel blend percentages, the boxplots show non-linear trends, variability, and outliers. Support Vector Regression (SVR), which is ideal for handling data variability and modeling non-linear relationships, is justified by these features. The significance of adding an additive as a crucial input parameter in the predictive model is further shown by the substantial difference between additive and non-additive situations. In this situation, the figure is a crucial exploratory data analysis tool for identifying influential variables, analyzing data distributions, and assessing whether SVR is appropriate for modeling. It supports the creation of a reliable SVR model for precise diesel engine performance prediction by providing a basic understanding of the relationships between input variables (fuel mix ratio and additive) and output responses (torque and power).

3.2. Model performance evaluation

The predictive accuracy of each model was evaluated on the test dataset using Mean Absolute Error (MAE), Root Mean Squared Error (RMSE), and the coefficient of determination (R^2). To predict the two distinct engine performance parameters, separate models were trained for each target output. A summary of the evaluation metrics is presented in Table 4.

The results clearly indicate that the SVM model significantly outperformed the Linear Regression model for both prediction tasks. The SVM model demonstrated superior performance, with an MAE of 1.119 and an RMSE of 1.490, approximately 7–8 times lower than the errors from the Linear Regression model. Critically, the SVM achieved an R^2 of 0.993, indicating that it can explain 99.3% of the variance in the engine power data. In contrast, the Linear Regression model's R^2 of 0.671 suggests a much weaker fit, explaining only 67.1% of the variance. For Torque Prediction: The performance gap between the models was even

more pronounced. The SVM model yielded an R^2 of 0.974, whereas the Linear Regression model yielded an R^2 of 0.093. This extremely low R^2 value indicates that the linear model almost completely failed to capture the underlying patterns in the torque data.

3.3. Visual analysis of predictive accuracy

To further interpret the models' performance, the predicted values were plotted against the actual experimental values. In these plots, the dashed diagonal line ($y = x$) represents a perfect prediction. The visual results, shown in Fig. 5 and Fig. 6, corroborate the findings from the evaluation metrics. The SVM predictions for both power and torque show data points tightly clustered around the identity line, confirming high accuracy across the entire range of values. Conversely, the Linear Regression plots exhibit significant deviations. For power prediction, the points form a curved pattern that deviates from the diagonal, indicating the model's inability to capture the nonlinearity. For torque prediction (Fig. 9), the model fails, with predictions scattered far from the ideal line, consistent with its near-zero R^2 value.

The significant outperformance of the SVM model can be attributed to its ability to handle complex, non-linear relationships via the Radial Basis Function (RBF) kernel. Diesel engine performance is governed by intricate interactions among fuel composition, engine speed, and combustion dynamics, resulting in non-linear data patterns. A simple linear model is fundamentally incapable of effectively capturing this complexity, leading to its poor performance, especially for the torque parameter. SVM, by mapping the data into a higher-dimensional feature space, successfully identified the complex regression function that accurately fits the data.

Table 4. Summary of model evaluation results

Parameter	Model	MAE	RMSE	R^2
Power	Linear Regression	8.5454	10.2548	0.6712
	SVM	1.1187	1.4900	0.9930
Torque	Linear Regression	18.1674	22.6074	0.09286
	SVM	2.3506	3.8527	0.9736

Note: MAE and RMSE are in the original units of the parameters (BHP and Nm). R^2 is on a scale of 0–1.

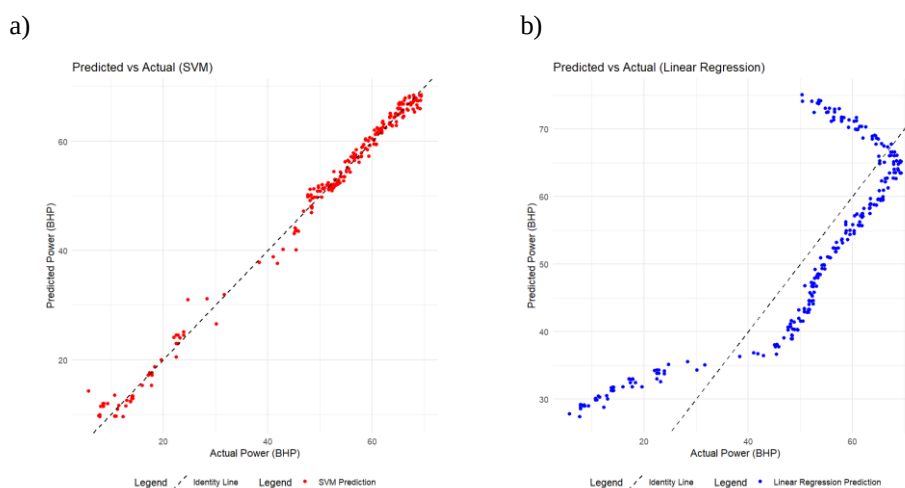


Fig. 5. (a) Predicted vs. actual values for power prediction with SVM; (b) predicted vs. actual values for power prediction with linear regression

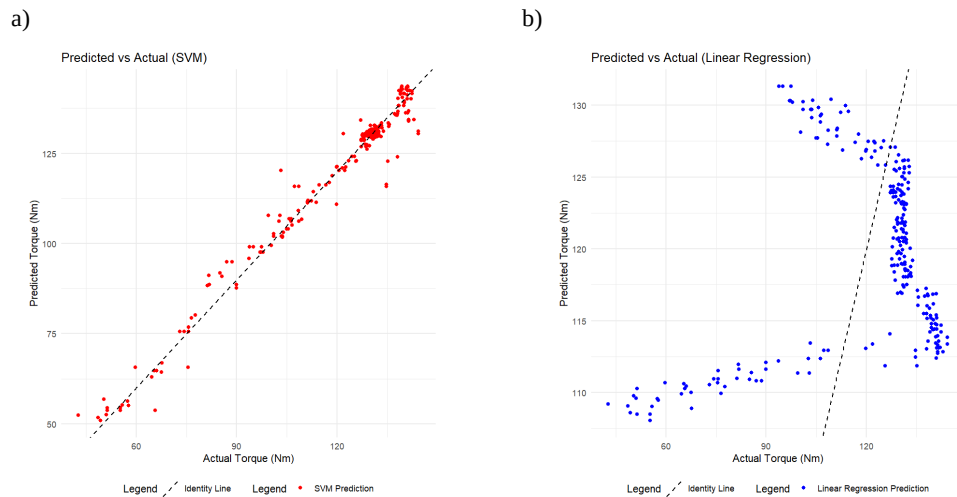


Fig. 6. (a) Predicted vs. actual values for torque prediction with SVM; (b) predicted vs. actual values for torque prediction with linear regression

The high accuracy achieved by the SVM model in this study ($R^2 = 0.993$ and 0.973 for torque) is highly competitive compared with recent literature in the field. In this model, both are treated as different outputs, even with the same input (in power learning, torque is not included as a variable). High accuracy occurs because the SVM parameters are tuned (to maximize accuracy across the model and dataset), and the data used is actual experimental data. This makes the relationship between variables clearer and more consistent for the model to learn. These results are consistent with, and in some cases exceed, the findings of other studies that used SVMs for engine parameter prediction. For instance [31] used an SVR model to predict the performance and emissions of a spark-ignition engine, achieving a validation R^2 of 0.9944 for fuel consumption rate and 0.9889 for NO_x emissions, which are comparable to our model's accuracy. Similarly [23] reported strong results predicting diesel engine performance with limited samples, achieving the best R^2 values of 0.9887 for maximum in-cylinder pressure and 0.9773 for engine efficiency, which align closely with our torque prediction R^2 of 0.974. A study [30] of predicting fuel consumption in marine diesel engines reported a test R^2 of 0.961 for their SVR model, a high value slightly surpassed by the R^2 values for both power and torque in our study. Even when predicting different parameters, such as exhaust gas temperature, [16] found that SVR was a strong predictor, with a validation R^2 of 0.82. Furthermore, a study by [10] on predicting biodiesel yield reported a Pearson Correlation Coefficient of approximately 0.75, underscoring the exceptional performance of our SVM model for this task.

While the high R^2 values indicate an excellent model fit for the given dataset, it is important to acknowledge the limitations. High performance on a test set partitioned from the same source does not guarantee perfect generalization to different engine types, substantially different operating conditions, or fuel variations not represented in the training data. Nonetheless, the results strongly validate that for the specific dataset of biodiesel-essential oil blends used, SVM is a highly effective and efficient modeling tool. This success highlights the significant potential of machine learning to accelerate research and development in alternative fuels

by providing rapid and accurate performance predictions, thereby reducing the reliance on time-consuming and expensive physical experiments.

4. Conclusions

Based on the test results, it can be concluded that engine power tends to increase with rpm until it reaches an optimum range of 4800 to 5000 rpm, then decreases at higher rpm. The observed peak in power at approximately 5000 rpm, followed by a decline at higher speeds, is mainly due to reduced volumetric efficiency, shorter combustion duration, and increased frictional losses. These factors, acting together, reduce torque more rapidly than the increase in engine speed, leading to a decrease in overall engine performance. All fuel variations show a consistent trend, but variations 6 to 10 have produced relatively higher power because they have higher density and viscosity.

This study investigated the impact of an essential oil additive and varying biodiesel blend ratios on diesel engine performance, using both statistical analysis and machine learning. The statistical analysis revealed that neither the additive nor the blend ratio had a statistically significant main effect on engine power or torque ($p > 0.05$). Despite the lack of statistically significant group differences, the machine learning approach yielded clear results. A Support Vector Machine (SVM) model, developed alongside a Linear Regression model, demonstrated exceptional predictive capability. The SVM model achieved outstanding accuracy, with an R^2 of 0.993 for engine performance.

Further research using the SVM model can focus on the following developments. First, expand the input variables beyond RPM and fuel composition to include other operational parameters, such as torque, fuel flow rate, exhaust gas temperature, and injection pressure, so the model can better represent combustion conditions. Second, optimizing the SVM hyperparameters (C , γ , and ϵ) using metaheuristic or grid search methods more systematically is necessary to improve prediction accuracy, especially in high-speed regions that exhibit power degradation. Third, the application and comparison of various kernels (linear, polynomial, and RBF) are recommended to evaluate the model's ability to capture the non-linear relationship between input variables

and output power. Finally, model validation using a broader dataset and a variety of fuels, as well as testing on new experimental data, will improve the generalizability and

reliability of the SVM model in diesel engine performance prediction applications.

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Combustion and emissions of a hydrogen-ammonia mixture engine with passive and active turbulent jet ignition

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The search for fossil fuel substitutes has led to the adoption of carbon-free alternatives, such as hydrogen and ammonia. It is also possible to use mixtures of both these fuels. Due to their different physical and chemical properties, the combustion processes of hydrogen and ammonia differ. The paper describes research on hydrogen and hydrogen-ammonia mixture combustion in the AVL 5804 engine. The engine is equipped with a two-stage combustion system with a passive and active prechamber. During the study, an indicator analysis of the combustion process in the prechamber and main chamber was performed. The study also included an evaluation of changes in hydrogen and nitrogen oxides concentrations, as well as changes in the number of particulate matter particles in the exhaust gases. By comparing the combustion of hydrogen and hydrogen-ammonia mixtures, it was observed that the thermodynamic parameters of the combustion process changed, along with emissions of unburned hydrogen and nitrogen oxides. However, there were insignificant differences in the number of particulate matter.

Key words: hydrogen and ammonia combustion, Turbulent Jet Ignition system, hydrogen emission, PN emission

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1. Introduction

The development of internal combustion engines is focused on reducing carbon dioxide emissions to reduce the greenhouse effect. Currently, the possibility of implementing carbon-free fuels to partially replace conventional fuels is being explored. The main solution in this case is hydrogen, but its low volumetric energy density has led to research on ammonia.

To fully leverage the potential of these fuels, efforts are underway to improve combustion efficiency. Since these are gas fuels, it is advantageous to burn lean mixtures with a relatively high compression ratio [17]. Hydrogen-powered engines commonly operate at compression ratios of $\varepsilon = 11$ –15 [22, 23], whereas ammonia-powered engines require higher compression ratios of $\varepsilon = 20$ and above [15, 16]. The combustion of lean air-fuel mixtures with hydrogen occurs at significantly higher excess air ratios than with conventional gaseous fuels. To avoid knock combustion, hydrogen engines operate at an excess air ratio above 2 [8–10, 24]. In addition to increasing the mixture's dilutions, adding ammonia can reduce knock combustion by limiting charge reactivity [8].

2. Characteristics of hydrogen and ammonia combustion conditions

The combustion of hydrogen enables replacing conventional fossil fuels, provided certain requirements are met. Hydrogen as a fuel requires, among other things, a modernized dose supply system (usually to the intake manifold) or direct injection [31]. The use of such solutions enables precise control of the mixture composition in a wide range of excess air ratios. Using the TJI system, which involves indirect hydrogen injection into the intake manifold and additional direct injection into the prechamber, it is possible to significantly extend the limits of stable combustion towards lean mixtures. The use of APC (active prechamber) or PPC

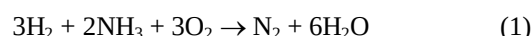
(passive prechamber) results in stable hydrogen combustion at $\lambda = 3$ [27]. Hydrogen and ammonia combustion can be carried out in single-fuel [3, 17, 19], dual-fuel systems [26] or using mixtures of these fuels [26, 28] (Table 1).

The laminar combustion velocity increases with the mixture temperature, due to faster chemical reaction kinetics and higher diffusion rates of active radicals. Research conducted by Guibert et al. [7] indicates significant differences in the combustion velocity of hydrogen and ammonia: at a temperature of 500 K, this velocity is approximately 2 m/s (for hydrogen) and 0.07 m/s for pure ammonia. A mixture of $H_2:NH_3$ in the 20:80 ratio has a laminar combustion velocity of 0.1 m/s.

The impact of EGR on the combustion process in a hydrogen-powered engine was analyzed by Pandey et al. [20]. During research, a single-cylinder engine with a displacement of 0.661 dm³ and a compression ratio of 15 was used. The use of a 15% EGR additive results in:

- a slight decrease in thermal efficiency (from 30.5% to 29%)
- a delay in the start of combustion by approximately 2 degrees
- an extension of the combustion time by approximately 2 degrees
- a reduction in nitrogen oxide emissions from 10 to 8 g/kWh. Increasing the EGR ratio to 25% limits these emissions to 2 g/kWh.

The reaction of the $H_2 + NH_3$ combustion proceeds according to the equation:

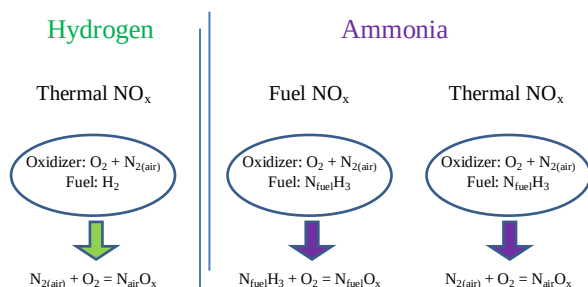


Research [15] indicates that under stoichiometric conditions, an optimal ammonia-to-fuel mass ratio of 0.6 balances emissions and fuel consumption. Empirical studies also indicate that increasing hydrogen's energy share relative to ammonia may initially improve thermal efficiency.

Table 1. Physical and chemical properties of fuels [4, 10, 19]

Fuel	H ₂	NH ₃	H ₂ + NH ₃ (95% + 5%) m/m
Weight of H ₂ [%]	100	17.7	–
Auto-ignition temp. [°C]	773	923	–
Boiling point [°C]	–253.0	–33.4	–
LHV [MJ/kg]	120	18.6	114.93
LHV [MJ/m ³]	2.90	2.83	–
Flammability limit [%]	4–75	15–28	–
Laminar flame speed [m/s]	1.6	0.07	–
Minimum ignition energy [mJ]	0.02	8	–
Liquid density [g/cm ³]	0.08 (–253°C)	0.68 (–33.4°C)	–

The combustion of ammonia leads to an increase in the concentration of nitrogen oxides due to their formation from both fuel and air – Zeldovich mechanism (Fuel NO_x and Thermal NO_x) [9, 16]. In contrast, hydrogen combustion only forms NO_x via the Thermal-NO_x mechanism (Fig. 1).


 Fig. 1. The mechanism of NO_x formation as a result of hydrogen and ammonia combustion (based on [3, 9, 16])

Due to the dual nature of NO_x formation during ammonia combustion, it can be concluded that a significant portion of NO_x is formed as thermal-NO_x, approximately 70% (at an excess air ratio of $\lambda = 1$) [16]. Increasing the leanness of the mixtures increases the proportion of thermal-NO_x; for example, the NO_x concentration in mixtures with $\lambda = 1.6$ is about 7 times higher.

Research conducted by Pyrc et al. [24] indicates that reducing the volume fraction of hydrogen to 7% in a mixture with ammonia enables the highest indicated efficiency of 27.6% (at $\varepsilon = 10$) to be achieved in an engine at $n = 600$ rpm and $\lambda = 1.06$. The high hydrogen fraction in the mixture (70%) resulted in a slightly lower indicated efficiency of 24.8%. However, nitrogen oxide emissions were reduced by a factor of 4 (from 26.7 to 7.7 g/kWh) during combustion of a fuel with a low hydrogen fraction.

In studies conducted by Mounaïm-Rousselle et al. [18], it was found that adding only 10% hydrogen to ammonia reduces nitrogen oxides emissions by 40% (in a standard combustion system). A smaller addition of H₂ – 2.2% at $\lambda = 1$ (standard combustion system) causes an increase in NO_x concentration from 10 to 80% (as a result of advancing or delaying the ignition angle) [11]. This may be because the ignition temperature and pressure are high near TDC, which consumes OH radicals too quickly and inhibits NO_x formation, whereas when the ignition timing is far from TDC, NO_x emissions increase.

Most of the studies mentioned above concern mixtures containing a significant proportion of ammonia and hydrogen. Research on mixtures of hydrogen with a small proportion of ammonia has been conducted by, among others, Xin et al. [33]. The tests were conducted with 5%, 8% and 10% of ammonia added, at an excess air ratio of $\lambda = 1$. As the ammonia proportion increased, the rate of pressure rise after ignition decreased by 15%. The extension of the initial combustion period of the mixture (CA0–10) was also around 15%. However, an increase in the IMEP value of around 5% was observed, regardless of the ignition angle and a few percent reduction in the ITE. The change in ammonia volume fraction had a minor effect on the final nitrogen oxide concentration values, but an increase was observed when the ignition timing was delayed.

Further research by Xin et al. [34] investigated the effects of ammonia concentrations of 0% and 2.2% under lean-burn conditions ($\lambda = 1.2$). A slightly leaner mixture reduced the rate of pressure rise by around 10%, regardless of ignition timing. Similar values were obtained for the increase in the first combustion phase (CA0–10), although these were strongly dependent on the ignition angle. The addition of ammonia significantly increased NO_x concentrations, but this increase depended slightly on the ignition angle.

An overview of the overall TJI process is summarised in Fig. 2, including the key parameters and physical process. In general, the jet ignition process is divided into three phases: prechamber combustion, cold-jet and jet ignition, and lean combustion [35].

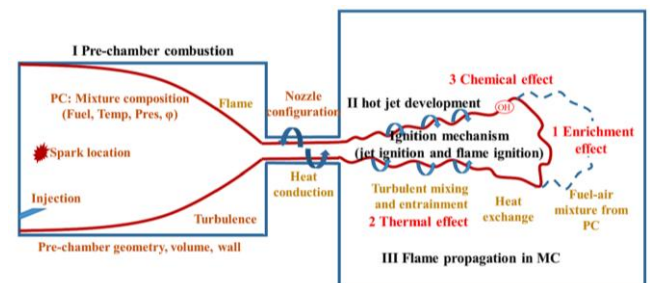


Fig. 2. Overview of TJI process. The injection only symbolises the act of fuel injection, and it does not indicate the actual position and directions of injection [36]

3. Purpose and scope of research

Modern combustion systems with a prechamber are being intensively researched for the use of hydrogen and ammonia fuels. This paper describes the use of a Turbulent Jet Ignition (TJI) system with a passive (PPC) and active prechamber (APC), enabling stable combustion of globally lean mixtures.

The aim of the work was to analyze the combustion process of hydrogen and hydrogen-ammonia fuels in the TJI engine with PPC and APC configurations. The test included analysis of pressure profiles in the cylinder and emissions of unburned H₂, NO_x, and particulate matter. The research was conducted under the following conditions:

- PC (pre-chamber): supplied with hydrogen
- MC (main-chamber): supplied with hydrogen or a mixture of hydrogen (95% m/m) and ammonia (5% m/m)

- $\lambda > 2.0$ for hydrogen and $\lambda > 1.5$ for a mixture of $H_2 + NH_3$
- variable fuel supply to the prechamber ($t_{PC} = 0, 4, 8$ ms).

The use of a 95%/5% H_2/NH_3 mixture is based on a comprehensive analysis of the authors previous research, which focused exclusively on fuelling the TJI engine (active and passive prechamber) with hydrogen. Further work involves adding ammonia (this article refers only to small proportions). The aim for future research is to increase the proportion of ammonia to around 50%. This approach will enable an analysis of the impact of ammonia on the combustion process, including an examination of the exhaust emissions from such systems.

4. Methodology

During the study, a single-cylinder research engine, AVL5804, was used; its technical data are presented in Table 2. The engine was equipped with a two-stage combustion system with an H_2 or $H_2 + NH_3$ injection system (95%/5%) into the intake manifold and direct injection into the prechamber (H_2).

The tests were conducted at a constant 1400 rpm engine speed using an independently electrically driven supercharger, allowing for adjustment over a wide range. The AVL5804 engine (Fig. 3) was powered by two independent fuel systems, each equipped with a thermal flow meter. Fuel was injected into the intake manifold at 7 bar and into the prechamber at 3 bar.

An AVL IndiSmart device was used to record rapidly changing pressure waves in the cylinder, and data from mass flow meters (hydrogen, H_2/NH_3 mixture, and air) were recorded. Detailed technical data of the equipment and the configuration of the measurement systems are presented in Table 3. In addition, the throttle position and spark timing parameters were analysed.

Table 2. AVL5804 engine specifications

Parameter	Unit	Value
Engine	–	1-cyl., 4-valve, SI, TJI
Displacement	dm ³	0.5107
Bore × stroke	mm	85 × 90
Compression ratio	–	15.5
Combustion system		Pre-chamber with 2.29 cm ³ volume and 6×1.7 mm nozzles configuration
Fueling		PFI (EM injectors); passive/active pre-chamber
Air system		External supercharger
Dyno		AMKASYN AVL AMK DW13-170

Two thermal flow meters were used to analyze hydrogen consumption and the hydrogen-ammonia mixture. Mass measurements from both flow meters enable the calculation of the excess air ratio (with simultaneous measurement of mass flow of air).

The combustion of hydrogen and a hydrogen-ammonia mixture was tested using the same injection system settings. The injection time into the main chamber was 6.5 ms, while the injection time into the prechamber was varied from 0 to 8 ms. Using identical control system settings for both fuels meant that the air excess ratio λ was a result rather than an input. The initial combustion conditions for both fuels are summarised in Table 4.

The recorded pressure profiles in the main chamber and prechamber were analysed. The indicated mean effective pressure was determined depending on the change in λ .

The assessment of the combustion system was also based on determining the indicated efficiency of each analysed point of engine operation, by the equations:

$$\eta_i = \frac{1}{g_i \cdot L_{HV}} \quad (2)$$

where L_{HV} is the lower heat value, g_i is a specific fuel consumption, calculated as:

$$g_i = \frac{G_{MC}}{N_i} \quad (3)$$

where G_{MC} is fuel consumption in the main chamber, and N_i is the indicated power:

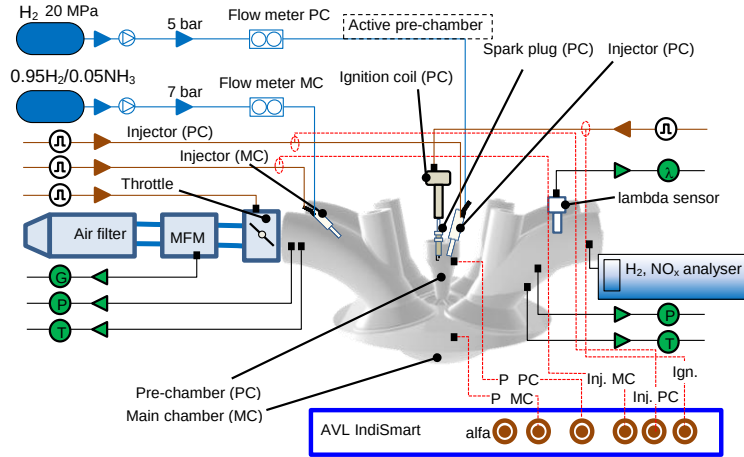
$$N_i = \frac{V_s \cdot p_i \cdot n}{\tau} \quad (4)$$

where V_s is the displacement of the cylinder, n is the engine speed, and $\tau = 2$ (4-stroke engine).

Table 3. Control and measurement equipment

System	Name	Value
Indicating system	AVL IndiSmart	8-canal, 0.1 deg CA; crank angle AVL 365C
Oil and water conditioning	AVL 577	0–150°C
PC pressure	Kistler M3.5 6081 AQ22	0–25 MPa
MC pressure	AVL GH14D	0–25 MPa
Throttle	Bosch ETB 32 mm	±0.5 deg
Ignition	ECU Master EMU Black	0.1 deg
MC injection timing	Mechatronika	0–20 ms; ±0.1 ms
PC injection timing	Mechatronika	0–20 ms; ±0.1 ms
PC fuel flowmeter	Bronkhorst 111AC-70K	0.4–100 nl/h; accuracy ±0.1% FS
MC fuel flowmeter	Bronkhorst F-112AC	0.02–1 kg/h; accuracy ±0.1% FS
Air flow meter	ABB SensyFlow	0–720 kg/h; error < ±0.8%
H_2 , NO_x concentration	E-com D	±10 ppm; ±5%
TSI EEPS 3090	Particle Number	5.6–560 nm 16 channels per decade (5.6–56 nm and 56–560); 32 total

a)



b)

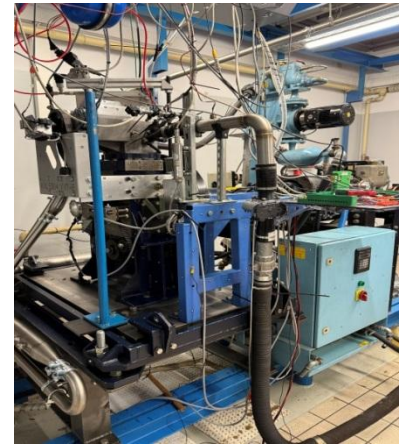


Fig. 3. The test bench: a) diagram with measurement equipment; b) view of the engine

 Table 4. Injection system settings and initial λ obtained at PPC

Fuel PC/MC	H ₂ /H ₂	H ₂ /(H ₂ + NH ₃)
Injection time: MC	6.5 ms	7.5 ms
Injection time: PC	0/4/8 ms	0/4/8 ms
Base lambda-value (passive PC)	2.15	1.91
	2.79	2.51

The injection time measurement into the prechamber shown in Table 4 does not account for the fuel dose. For this reason, Fig. 4 shows the conversion of the prechamber injection time into the mass fraction of the total fuel dose delivered to the cylinder.

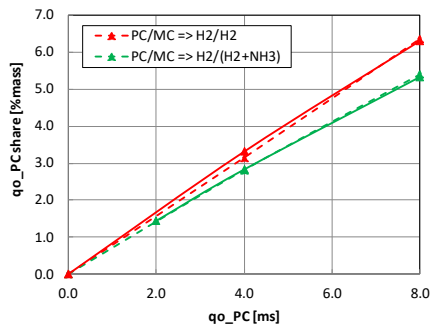


Fig. 4. Conversion of injection duration into the proportion of fuel injected into the prechamber

The plan for research on the combustion of hydrogen and a mixture of hydrogen-ammonia is shown in Table 5.

Table 5. Research plan for the combustion of hydrogen and ammonia

No.	Fuel (MC/PC)	λ -value	PC [g/h]	MC [g/h]
1	H ₂ /H ₂	2.14	0	0.182
2	H ₂ /H ₂	2.06	5.92	0.182
3	H ₂ /H ₂	1.96	12.37	0.182
4	H ₂ /H ₂	2.79	0	0.183
5	H ₂ /H ₂	2.69	6.27	0.183
6	H ₂ /H ₂	2.57	12.32	0.183
7	H ₂ /NH ₃	1.91	0	0.214
8	H ₂ /NH ₃	1.88	6.2	0.214
9	H ₂ /NH ₃	1.83	12.22	0.214
10	H ₂ /NH ₃	2.51	3.14	0.215
11	H ₂ /NH ₃	2.44	6.26	0.215
12	H ₂ /NH ₃	2.43	12.11	0.215

5. Comparison of hydrogen and hydrogen with ammonia combustion

5.1. Evaluation of indicated pressure changes

The analysis of the static test conditions ($n = 1400$ rpm; IMEP = 3.77 bar – a value calculated on the base of 100 subsequent engine cycles; $P_{max} = 42.7$ bar) included an evaluation of the combustion pressure profiles for the use of passive and active prechamber. The IMEP and CoV(IMEP) values are shown in Fig. 5.

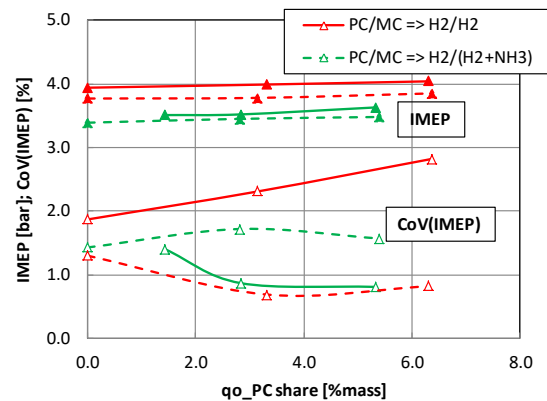


Fig. 5. The IMEP value and engine irregularity

An analysis of the uneven engine operation shows CoV(IMEP) values below 3%. This is acceptable for a single-cylinder engine. Based on the results, it can also be concluded that the use of an H₂ + NH₃ mixture reduces engine running irregularities as the fuel dose into the prechamber is increased.

The combustion of hydrogen and its mixture with ammonia using a passive prechamber was characterized by the lowest maximum combustion pressure (Fig. 6). Due to the initiation of combustion in the prechamber, a characteristic pressure peak is observed in it. Increasing the fuel dose to the prechamber (4 and 8 ms) increased combustion pressure in both the prechamber and the main chamber, with a simultaneous increase in the prechamber pressure peak amplitude.

The center of combustion (CA50) is currently an indicator of combustion efficiency. Many researchers define the optimal CA50 value as being between 6°CA and 10°CA aTDC [5, 22, 28]. During the tests, constant combustion center 9°CA aTDC (angle of 50% heat release) was maintained. Combustion of the hydrogen-ammonia mixture resulted in lower cylinder pressure, due to the lower total calorific value of the fuel and a different global excess air ratio. Despite a slightly higher fuel mass dose and a lower λ value at constant injection time settings, the combustion process deteriorated, resulting in lower maximum combustion pressures in both chambers.

Combustion at an excess air ratio $\lambda \approx 2$ resulted in significant pressure changes in the cylinder as the fuel dose to the prechamber increased, regardless of the fuel type. At higher λ values, pressure changes during hydrogen combustion remained significant, whereas in the $H_2 + NH_3$ mixture,

they were lower in both the prechamber and the main chamber. This phenomenon is partly due to the low laminar combustion velocity of ammonia.

5.2. Analysis of combustion parameters

Engine performance indicators are shown in Fig. 7. Analysis of indicated pressure for different air excess ratios reveals different engine operating conditions for different fuels and fuel doses to the PC. Equal injection timing settings resulted in lower air excess ratios during combustion of $H_2 + NH_3$ (Fig. 7a), mainly due to changes in the injected fuel mass.

The analysis showed an increase in IMEP values due to higher fuel dose supplied to the prechamber. In each case analyzed, regardless of fuel type, an increase in IMEP of approximately 2% was observed (Fig. 7b).

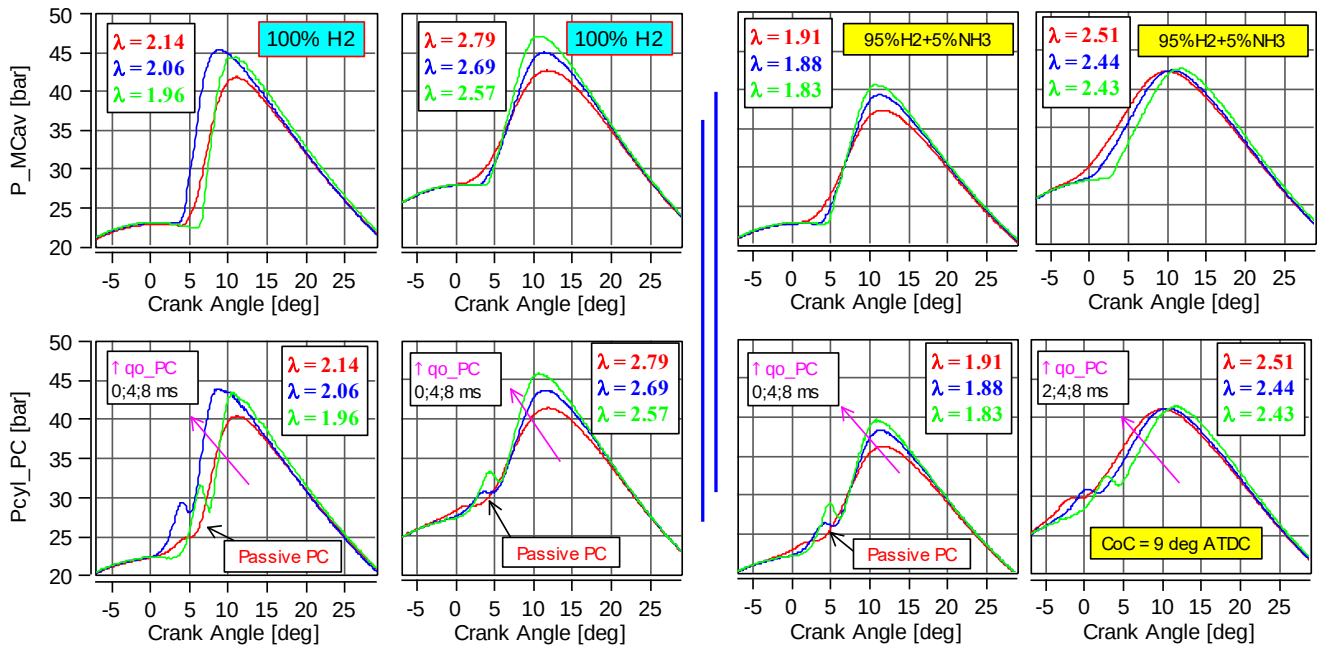


Fig. 6. Engine test results: indicator tests of the combustion process of hydrogen (left) and a hydrogen-ammonia mixture in a ratio of 95%/5% (right) at different excess air ratios

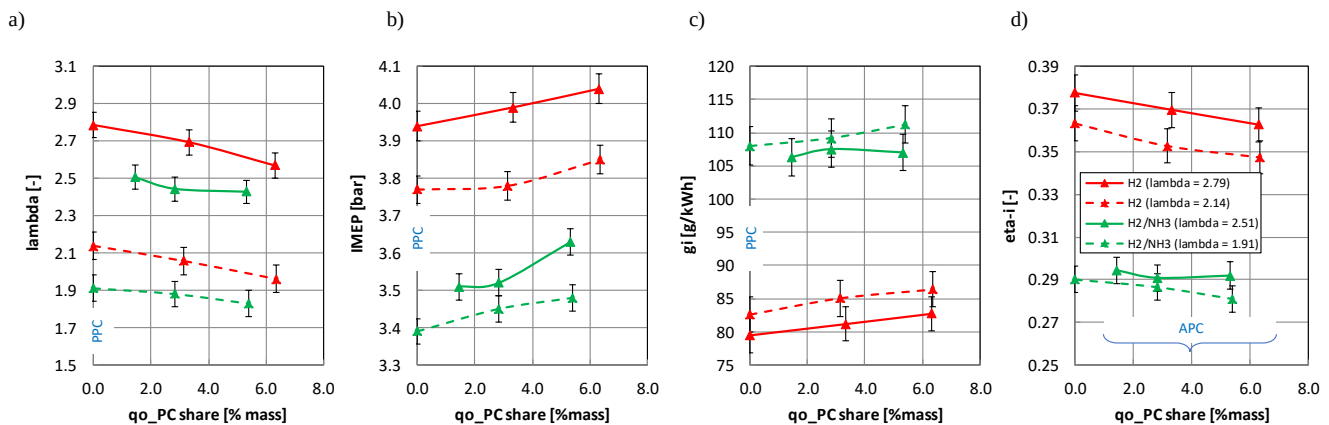


Fig. 7. Performance indicators for engines fuelled with hydrogen and hydrogen-ammonia: a) λ -value; b) IMEP; c) specific fuel consumption; d) indicating efficiency

Increasing the fuel dose to the prechamber also increased specific fuel consumption (Fig. 7c). Although the ammonia content in the mixture was only 5% by mass, the increase in specific fuel consumption reached approximately 4% compared to the passive chamber configuration. The difference in specific fuel consumption between hydrogen and an $H_2 + NH_3$ mixture exceeded 30%.

The analysis of the indicated engine efficiency showed significantly higher values with hydrogen (approximately 38%) than with a hydrogen-ammonia mixture (approximately 29%). This corresponds to a relative efficiency difference of 22% in favor of hydrogen (Fig. 7d). The results indicate that even a small mass addition of ammonia results in a significant deterioration in the performance of an engine with a two-stage combustion system.

6. Analysis of exhaust gas compositions

The combustion of hydrogen and its mixture with ammonia leads to significantly different concentrations of gaseous components in the exhaust gases (Fig. 8a). The combustion of pure hydrogen was characterized by relatively low concentrations of nitrogen oxides, ranging from about 30 ppm to over 300 ppm at higher excess air ratios, which is typical for the combustion of lean mixtures. The use of a 5% mass addition of ammonia resulted in a more than tenfold increase in NO_x concentration in the exhaust gases, due to nitrogen in the fuel and the intensification of the Fuel NO_x mechanism, with the simultaneous participation of the thermal (Zeldovich) mechanism.

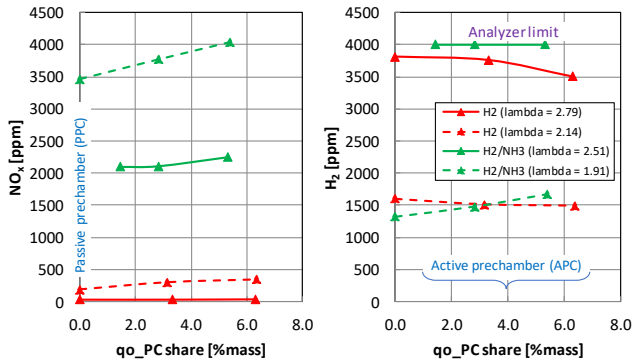


Fig. 8. The effect of variable fuel dose on the concentration of: a) nitrogen oxides; b) unburned hydrogen

Analysis of the unburned hydrogen concentration (Fig. 8b) shows an increase at high excess air ratios. Under the same conditions, using hydrogen or an $H_2 + NH_3$ mixture did not result in significant differences in unburned hydrogen emissions.

7. Analysis of particulate matter emissions

Analysis of particulate emissions in a hydrogen-fuelled engine indicates that their main source is lubricating oil entering the combustion chamber. Under low-load conditions, increased vacuum behind the throttle valve increases oil evaporation, which is then transported to the cylinder, leading to increased particulate emissions. In addition, at lower operating temperatures, the piston-ring-cylinder components undergo less thermal deformation, which increases oil leakage into the combustion chamber. As the load increases, improved sealing of the piston-ring-cylinder system reduces oil consumption, thereby lowering emissions and particulate matter [1, 2, 29].

Analysis of the particulate matter distribution (Fig. 9) indicates that the combustion of different carbon-free fuels has little effect on the shape of the particulate matter number. Regardless of the fuel type, the highest particle count was observed with a passive prechamber, with the distribution peaking at approximately 100 nm. This phenomenon can be associated with lower combustion efficiency in this configuration. The use of an active prechamber led to improved combustion quality and reduced particulate matter emissions.

In hydrogen combustion, at low air excess ratios, prechamber activation reduced particulate matter by approximately 35% (Fig. 9a).

During the combustion of lean hydrogen mixtures ($\lambda \approx 2.8$), the number of particles decreased from approximately $6 \times 10^5 \text{ cm}^{-3}$ with a diameter of 100 nm to approximately $5 \times 10^5 \text{ cm}^{-3}$ (Fig. 9b). Similar relationships were also observed during the combustion of a hydrogen-ammonia mixture (Fig. 9c and 9d). Increasing the fuel dose fed into the prechamber reduced the maximum number of particles. In addition, an increase in the excess air ratio caused a shift in the particle number distributions towards smaller diameters.

Research carried out by Apicella et al. [1] on a single cylinder 255 cm^3 hydrogen fuelled engine ($\lambda = 1.6$; $n = 2000 \text{ rpm}$) showed similar values of particulate matter in the order of $1.0 \times 10^6 \text{ cm}^{-3}$ with an average diameter of

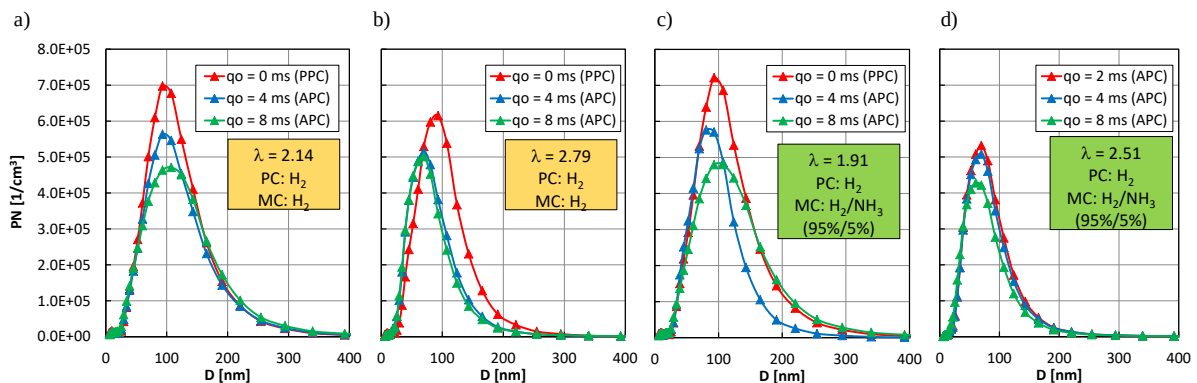


Fig. 9. Analysis of the number of particulate matter during combustion of hydrogen and a mixture of hydrogen and ammonia: a) $\lambda = 2.15$, hydrogen; b) $\lambda = 2.79$, hydrogen; c) $\lambda = 1.65$, hydrogen + ammonia; d) $\lambda = 2.56$, hydrogen + ammonia

approximately 39 nm under low engine load conditions (IMEP = 4.1 bar). At higher loads (IMEP = 5.5 bar; $\lambda = 1.5$), a further reduction in the number of particulate matter to approximately $1.0 \times 10^4 \text{ cm}^{-3}$ was observed, with a diameter of 10 nm.

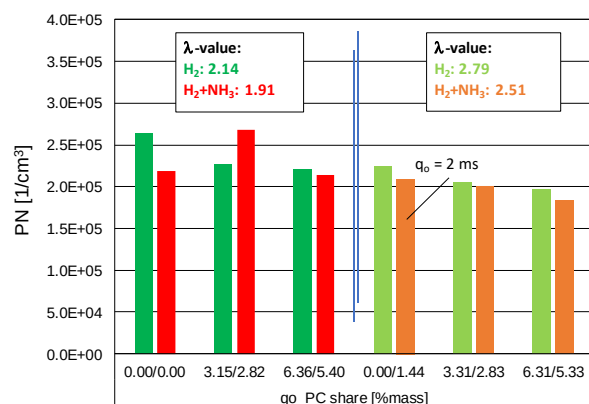


Fig. 10. Analysis of the number of particulate matter

Shi et al. [29], in their research on a single cylinder engine with a capacity of 604 cm³ ($n = 3000 \text{ rpm}$) with direct hydrogen injection ($\lambda = 2.1$), obtained a number of particles in the range from 1×10^6 to $5 \times 10^6 \text{ cm}^{-3}$, depending on the engine operating conditions of 3 and 1 MPa. These results differ from those obtained in the current studies mainly in the reported maximum diameters (7–10 nm). This may mainly be due to differences in engine speed.

Figure 10 shows the values of the number of particulate matter obtained under engine operating conditions without the use of exhaust gas treatment systems. Analysis of combustion with hydrogen ($\lambda = 2.14$) and H₂ + NH₃ mixture ($\lambda = 1.91$) fuel indicates slightly higher values of the number of particles, ranging from $(2.0\text{--}2.5) \times 10^5 \text{ 1/cm}^3$.

At higher excess air ratio values, the number of particulate matter stabilized at around $2.0 \times 10^5 \text{ 1/cm}^3$. The results show that, under the conditions analyzed, the type of carbon-free fuel has little effect on the number of particulate matter emitted.

Conclusions

The combustion of carbon-free fuels is an effective substitute for fossil fuels, especially in two-stage combustion systems in high excess air ratio conditions. Under the investigated operating conditions, hydrogen combustion provided the highest indicated efficiency (38% in current studies – Fig. 7d), while ammonia co-combustion has an efficiency of around 28%. These values were obtained at low engine loads, which indicates the significant potential of these fuels.

The detailed conclusions regarding combustion are as follows:

- The addition of ammonia reduces combustion pressure, which is due to the significantly lower energy content of the mixture
- Adding 5% ammonia to hydrogen reduces IMEP by approximately 10%, regardless of the load
- The addition of ammonia reduces the calorific value of the load, resulting in an increase in the specific fuel consumption of such a mixture by approximately 25%
- A small addition of ammonia causes a sharp increase in the concentration of NO_x in the exhaust gas: from 30 ppm (hydrogen at various values of λ) to over 2000 ppm when burning a mixture containing ammonia.

Analysis of particulate matter indicates that the largest source is engine oil, as changing the fuel has only a minor effect on their levels. Furthermore, it was found that increasing the lean ratio of the fuel-air mixture (hydrogen or a hydrogen-ammonia mixture) has little effect on particulate matter levels.

Nomenclature

APC active prechamber
 g_i specific fuel consumption
 IMEP indicated mean effective pressure
 MC main chamber
 NH₃ ammonia
 N_i indicated power
 PC prechamber

PN particle number
 PPC passive prechamber
 q_o fuel dose
 SI spark ignition
 TJI turbulent jet ignition
 λ lambda value
 η_i indicated efficiency

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Internal combustion modelling for aerospace applications

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A model of an internal combustion engine (ICE) was created in Simulink and could be used as part of the drive component of the propulsion system during the development stage of an unmanned aerial vehicle (UAV)/Aircraft. The model can predict the engine power, engine torque, engine brake-specific fuel consumption, and engine fuel flow of an ICE for a given engine throttle, atmospheric pressure, rotational speed, temperature, and altitude at which the UAV is flying. Results from the model were compared with those of previous researchers reported in the literature. The engine power, torque, fuel flow, and brake-specific fuel consumption curves from simulations were found to be in good agreement with those obtained experimentally by previous researchers.

Key words: *unmanned aerial vehicle, internal combustion engine, brake specific fuel consumption, fuel flow, manifold air pressure*

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1. Introduction

An Internal Combustion Engine (ICE) is a crucial driving component of a UAV as it provides the thrust and power that can be used to propel a UAV in flight. In an ICE, the air from the combustion chamber is mixed with fuel, which is ignited to produce the power and torque that can be used to drive the propeller in flight. To estimate the power and torque produced by a UAV, a working model is very important during the early development stages of an aircraft, as it can help the engineer predict the power and thrust requirements of the ICE to drive the UAV's propeller in flight. One of the main challenges in designing an ICE for a UAV application is creating a model that can simulate the ICE at the various temperatures and altitudes at which the UAV is flying. Previous researchers have attempted to develop an engine model for simulation, which was experimentally validated. The models developed by the researchers were used as drive components for Hybrid Electric Vehicles [1, 19], Hybrid track vehicles [18], and ship hull propulsion for marine applications [21]. Stewart et al. [19] presented an engine simulation model of an ICE for a series hybrid electric vehicle. The engine performance curves obtained from the model were validated with experimental results of a V6 Ford Duratec gasoline engine. A similar study was also conducted by Al-Atabi et al. [1], in which experiments were performed to determine the optimal performance of an ICE for a hybrid power unit of a series-hybrid electric vehicle. The engine performance curves obtained from the experiment showed that the ICE performed at its optimum at an engine speed between 2000 – 2500 rpm and a charging load of 1 kW. Another study was conducted by Shanmuganathan et al. [18], who developed a dynamic model of an ICE as a drive component for a hybrid track vehicle in Simulink. The author generated torque and power curves, which were validated using test data for a 1100 hp, 800 kW generator. Also, Magdas et al. [14] developed an engine simulation model of an ICE using the Loftus Engine simulation software to predict its behavior. Using the simulation software, the author generated torque and brake power curves for the modeled ICE. In

addition, Tadros et al. [21] carried out the numerical simulation of a 2-stroke marine diesel engine as a drive component for the propulsion system of a ship hull. The model results were then compared with experimental data from a real engine, yielding a good match. A lot of related work was also carried out by previous researchers, where they developed engine models for heavy-duty diesel engine trucks [4], automobile vehicle applications [16], and motorsports cars [8]. Another challenge posed by previous researchers was to create an ICE model that could be matched with a turbocharger to increase boost power. Many researchers have attempted to resolve this issue by developing an engine model that can be matched with a turbocharger to increase boost power [3, 5, 6, 9]. For example, Chen et al. [5] developed a simulation model to increase the power output of a 9-L Tier IV turbocharged diesel engine using GT-Power. The authors showed that their engine simulations accurately predict actual engine performance curves, resulting in a 1.5% reduction in BSFC and a 0.4% increase in torque. Hameur et al. [9] developed an engine simulation model to match a turbocharger with an ICE. Through simulations, the authors generated engine performance curves which were validated with results from Richard wave simulations. Chen et al. [6] developed a simulation model of a gasoline engine to perform a turbocharger match to boost its power and torque using GT-Power software. Through simulation, the author obtained torque, power, and BSFC curves for the gasoline engine, resulting in about a 4% reduction in fuel consumption. Ammad Ud Din et al. [3] developed an ICE engine simulation model to match a turbocharger and validated it using engine test data in GT-POWER. The authors obtain engine torque and BSFC curves, resulting in a 5.26% increase in engine torque and a 4.31% decrease in BSFC. Also, the engine performance curves obtained from the model were found to be in close agreement with experimental data by 5%. Very little research was carried out on modeling studies related to UAV Combustion engines. Some contributions are highlighted in this paper. Kravchenko et al. [13] developed a fuel injection system to boost power and reduce fuel con-

sumption in a 2-stroke engine used in a small UAV. Through simulations, the author increased the boost power and UAV flight duration by using an electronic fuel injection system. Kang et al. [12] developed a turbocharger system to provide pressure boost for a High-Altitude Long Endurance UAV powered by an ICE, thereby reducing the engine's sensitivity to altitude. Yixuan et al. [23] proposed an air-fueled Ratio Controlled strategy to boost the power of a fuel-powered UAV. Through simulation, he boosted the UAV engines power from 9% to 33%. Wróblewski et al. [22] designed an experimental test stand for thrust matching across various propeller sizes and for detecting anti-wear coatings in 2-stroke piston engines of small aircraft. However, no attempt to create an ICE model for a UAV has been reported in the literature surveyed. This paper thus provides a unique modelling approach for an ICE that could be used as a drive component of a UAV by estimating the power thrust, BSFC, and Fuel flow of the ICE at various temperatures, pressures, and altitudes during the flight of the UAV.

2. Methodology

The ICE was modelled in the MATLAB-Simulink environment using engine data from [11]. The developed ICE model was applied to UAV combustion, such as the Aer-sonde UAV, which uses a 22 ccm Avgas 1 kW single-cylinder port fuel injection combustion engine [20]. The input parameters to the model are the engine throttle, engine atmospheric pressure, engine rotational speed, and temperature at the UAV's reference altitude, as shown in the boxes in Fig. 1. The model determines the ICE power, ICE Torque, Brake Specific Fuel Consumption, and Fuel Flow, as shown in Fig. 1.

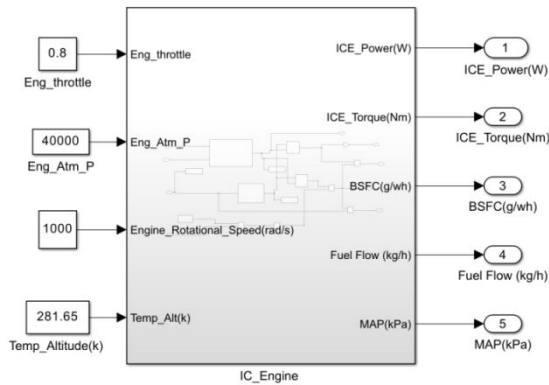


Fig. 1. ICE Simulink model with inputs and outputs

Figure 2 shows the inside view of the Simulink model. The engine Manifold Air Pressure (MAP) is calculated from the model as:

$$P_{ma} = P_{min} + \left(\frac{P_{atm}}{1000} \right) \times \eta_{th} \quad (1)$$

where: P_{atm} is the engine atmospheric pressure (Pa), P_{min} is the minimum atmospheric pressure (Pa) η_{th} is the engine throttle.

The engine Fuel Flow (FF) and ICE power (P_{ICE}) are determined from the engine rotational speed and Manifold Air Pressure (MAP) by interpolation with a 2-D lookup

Simulink using ICE data. The ICE data used to create the lookup is shown in Tables 1 and 2, respectively. An altitude-correction factor has also been included in the model to determine the ICE power at various altitudes during flight. It determines the ICE power at any given altitude from the temperature at sea level, the temperature when the UAV is in flight, and the ICE Power from the lookup table as [11]:

$$P_{ICE,Alt} = P_{ICE} \sqrt{\frac{T_{alt}}{T_{sea_level}}} \quad (2)$$

where: T_{sea_level} is the temperature at sea-level in Kelvin (K), T_{alt} is the temperature at a given altitude in Kelvin (K), P_{ICE} is the ICE power in Watts (W), $P_{ICE,Alt}$ is the ICE power at a given altitude in Watts (W), FF is the engine fuel flow in (kg/h).

An atmosphere model was also created using a 2-Look-up table based on the International Standard Atmosphere (ISA), from which the temperature at any given altitude could be determined from the UAV's altitude. This is shown in Fig. 3. The engine Brake Specific Fuel Consumption (BSFC) is obtained from the engine ICE power and Fuel flow as:

$$BSFC = \frac{FF}{P_{ICE,Alt}} \quad (3)$$

The engine torque is determined from the ICE power and engine rotational speed as:

$$T_{ICE} = \frac{P_{ICE,Alt}}{\omega} \quad (4)$$

where T_{ICE} is the engine torque in N, ω is the engine rotational speed in rad/s.

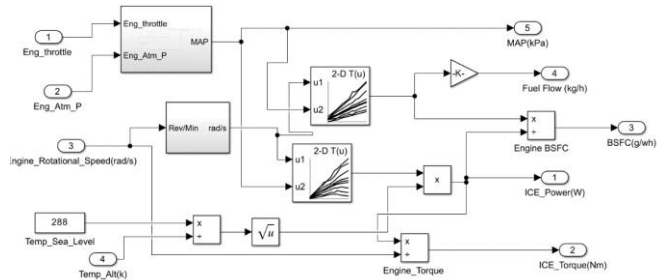


Fig. 2. Structure of the ICE Simulink model with input and output signals

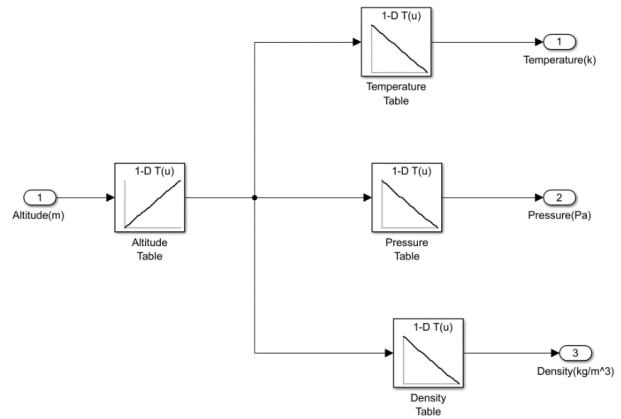


Fig. 3. Atmospheric model based on altitude input

Table 1. ICE power [W] at various engine speeds and MAP values

n [rpm]	MAP								
	60	70	80	90	92	94	96	98	100
1500	18.85	47.12	65.97	67.54	69.12	67.54	67.54	69.12	86.39
2100	59.38	98.96	127.55	149.54	151.74	160.54	178.13	200.12	224.31
2800	93.83	149.54	187.66	237.50	249.23	255.10	307.88	366.52	398.77
3500	109.96	161.27	245.57	307.88	326.20	351.86	421.50	591.14	531.45
4500	164.93	245.04	339.29	438.25	447.68	494.80	565.49	673.84	772.83
5100	181.58	245.67	389.87	496.69	528.73	571.46	662.25	822.47	993.37
5500	184.31	293.74	403.17	535.64	570.20	622.04	748.75	956.09	1059.80
6000	163.36	276.46	420.97	565.49	609.47	691.15	860.80	1131.00	1193.80
7000	124.62	249.23	417.83	586.43	645.07	762.36	996.93	1246.20	1429.40

Table 2. Fuel flow rate [g/h] at various engine speeds and MAP values

n [rpm]	SMAP								
	60	70	80	90	92	94	96	98	100
1500	31	32	46	53	55	57	65	73	82
2100	40	44	54	69	74	80	92	103	111
2800	50	63	69	74	95	98	126	145	153
3500	66	75	87	110	117	127	150	175	190
4500	83	98	115	143	148	162	191	232	246
5100	93	102	130	159	167	182	208	260	310
5500	100	118	137	169	178	190	232	287	313
6000	104	126	151	184	191	206	253	326	337
7000	123	144	174	210	217	244	321	400	408

3. Results and discussion

3.1. Model output results at various engine throttle positions

This section discusses the results obtained from the model, with the validation of the model output results obtained by previous authors. This is explained in the relevant sub-sections below.

The model was run to evaluate the effect of increasing the engine throttle on the various outputs of the model by setting the engine throttle to zero when the engine is idle to full throttle when the engine draws maximum air fuel into the combustion chamber of the engine, as shown in Fig. 4–7. The results of each output are explained in the relevant subsections below.

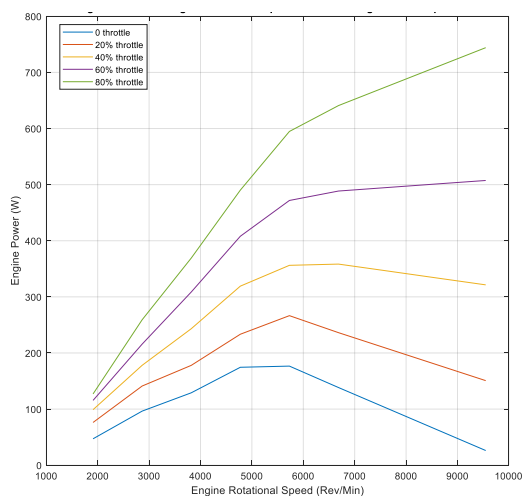


Fig. 4. Engine power against engine rotational speed at various engine throttle positions

Engine power at various throttle positions

The engine power curves increase as the throttle is increased from 0% to 80%, with the lowest power at 0% and the highest at 80%, as shown in Fig. 4.

Engine torque at various throttle positions

A similar trend is also observed in the engine torque curves, which increase as the throttle is increased from 0% to 80%, as shown in Fig. 5.

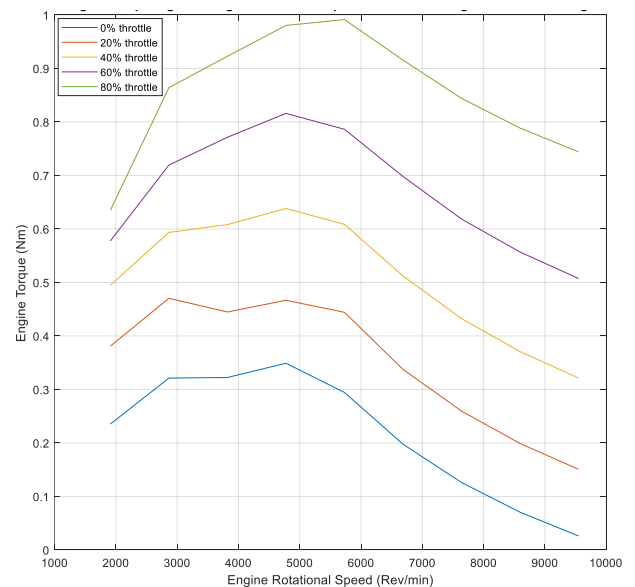


Fig. 5. Engine torque against engine rotational speed at various engine throttle positions

Engine fuel flow at various throttle positions

The engine fuel flow curves rise as the engine throttle is increased from 0% to 80%, with the lowest fuel flow at 0% throttle and the highest at 80% throttle, as shown in Fig. 6.

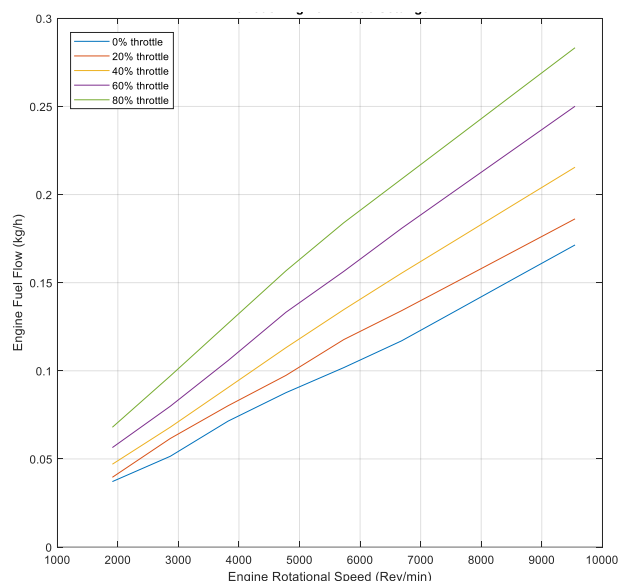


Fig. 6. Engine fuel flow against engine rotational speed at various throttle positions

Engine brake specific fuel consumption at various throttle positions

The engine brake-specific fuel consumption curves, on the other hand, decrease as more throttle is added to the engine i.e. as the engine throttle increases from 20% to 80%, with the highest brake-specific fuel consumption curve at 20% throttle and the lowest brake-specific fuel consumption curve at 80%, as shown in Fig. 7.

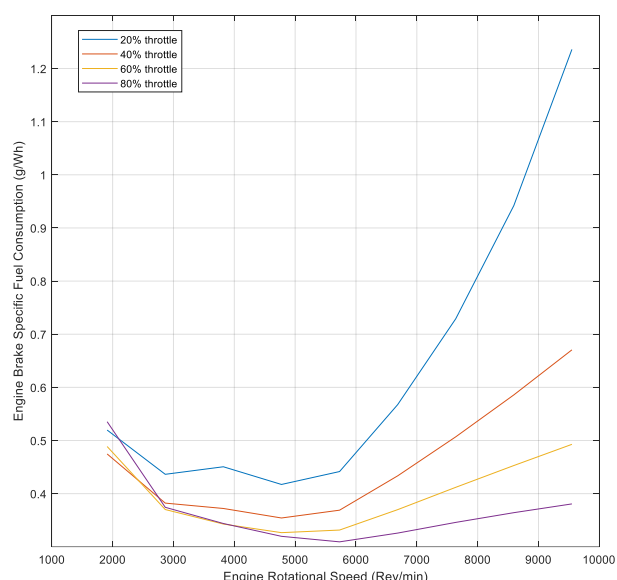


Fig. 7. Engine brake specific fuel consumption against engine rotational speed at various throttle positions

3.2. Model results and comparison

A comparison between model results and those of previous researchers experiments was conducted, as real en-

gine data from a manufacturer could not be obtained for the developed model. 8 papers from the literature were used for comparison. Figures 8–11 show the modeled results, which were compared with those of previous researchers. This is explained in the relevant sub-sections below.

Engine power comparison

Figure 8 shows the engine power curves obtained from the model, which were compared with results from previous authors' experiments on ICEs [2, 7, 17]. It can be observed that the simulated engine power curves closely predict the power curves obtained from experiments since they both follow the same increasing trend pattern, where the engine brake power increases with increased engine rotational speed, with maximum power attained at a specific rpm value (8200 rpm, 693 W for the simulated power curve). A discrepancy is observed at various engine rotational speeds. The main reason for the discrepancy is that the modeled ICE was fitted with power curves obtained experimentally by previous researchers from the literature for different-sized ICE (between 30 kW and 40 kW). In contrast, the modeled ICE has a maximum power of only 700 W. Proper validation could only be achieved if the author could obtain engine performance curves from the manufacturer for the modeled ICE. However, a deviation always exists between the modeled ICE and the experimentally obtained ICE. The deviation is mainly due to the fact that the actual combustion process is a complex phenomenon, and simulations from a developed model are not sufficiently sophisticated to account for the factors that occur during the actual combustion process [7].

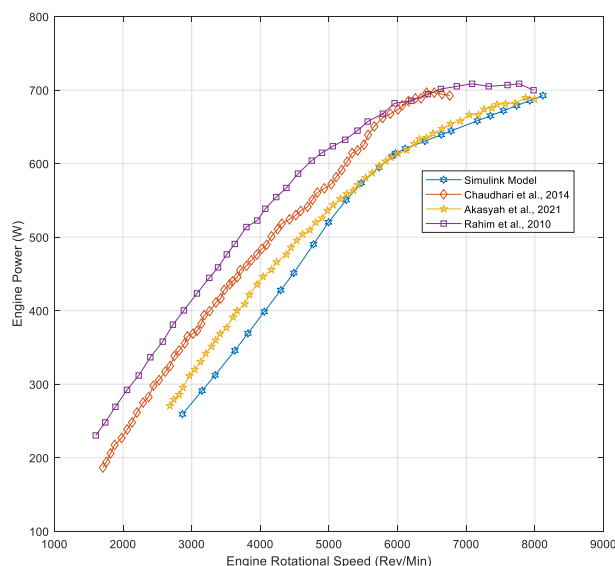


Fig. 8. Engine power vs engine rotational speed validation

Engine torque comparison

Figure 9 shows the engine torque results from the model, which were compared with engine torque curves obtained by previous authors experimentally. The simulated engine torque curves increases with increased rotational speed to a maximum value (0.35 Nm, 4800 rpm), which represents the maximum operating torque of the engine, and then decrease again as the engine speed is further increased.

The experimental engine torque curves obtained by previous authors [7, 15, 17] also show the same predictions, where it increases to a maximum operating point and decreases when the engine rotational speed is further increased, as shown in Fig. 9.

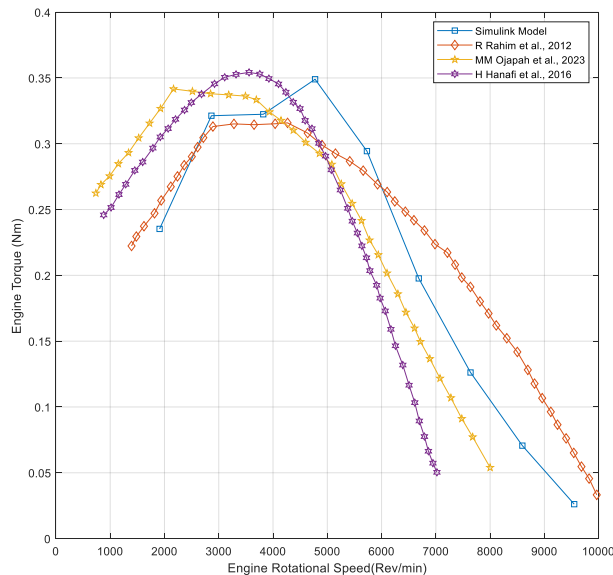


Fig. 9. Engine torque vs engine rotational speed validation

A discrepancy is also observed between the modelled engine torque curves and experimental torque curves obtained by researchers due to the same reason stated in previous section.

Brake specific fuel consumption comparison

Figure 10 shows the simulated BSFC obtained from the developed model, along with experimental BSFC curves from previous authors [2, 10, 17].

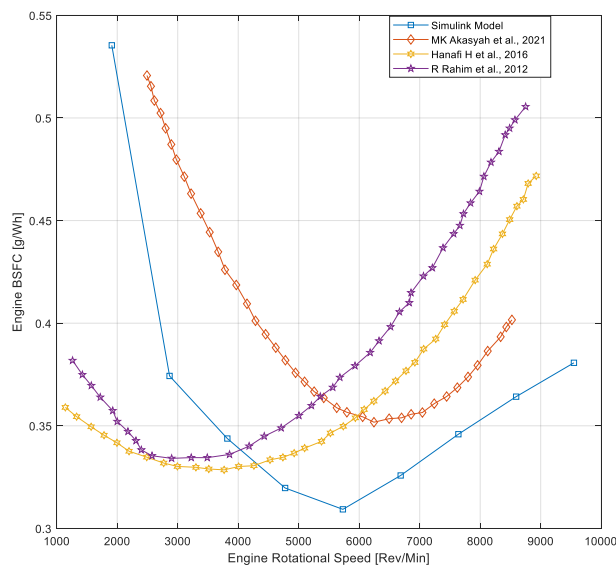


Fig. 10. Engine brake specific fuel consumption vs engine rotational speed validation

Both the simulated curve and the experimental curves follow the same trend: the engine BSFC decreases with

increasing rotational speed to a minimum value, which represents the maximum operating point of the engine, and then increases again as the rotational speed is further increased. A discrepancy is also observed between the modelled BSFC curves and the experimental BSFC curves reported by previous researchers, for the same reasons stated in the previous section.

Engine fuel flow comparison

Figure 11 shows the model output for engine fuel flow, compared with experimental curves for engine fuel flow obtained by previous authors for ICEs [21]. The simulated fuel flow curves closely approximate the experimental curves, as they all show an increase in fuel flow with increasing rotational speed. A deviation is also obtained between the engine fuel flow curves obtained from simulation and those obtained experimentally due to the same reasons stated in the previous section.

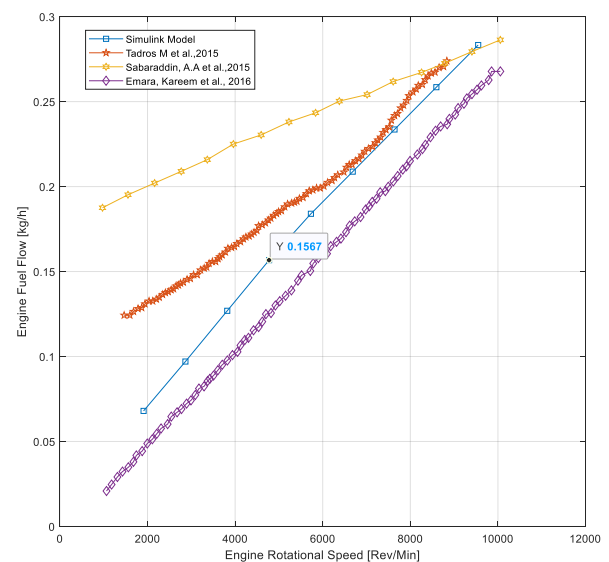


Fig. 11. Engine fuel flow vs engine rotational speed validation

3.3. Engine contour maps

An engine contour showing the engine's relative operating point was obtained from the model, as shown in Fig. 12. At this time, the contour map is static. More research would be conducted on how to make the map dynamic during simulation.

4. Conclusion

This paper provides a unique modelling approach of an ICE for a UAV given some input parameters to the model. The model results were compared with those of previous researchers' experimental results. Both the model results and the experimental results agreed, as they showed the same curve. The Novelty in the modelling approach used in this article can be summarized as follows:

- The ICE model has been developed so it can be used as a drive component for UAV propulsion.
- The developed model allows the ICE to estimate power, torque, Fuel Flow, and BSFC at various temperatures, pressures, and altitudes during the flight of the UAV, unlike ground vehicles, where there are no pressure or altitude variations.

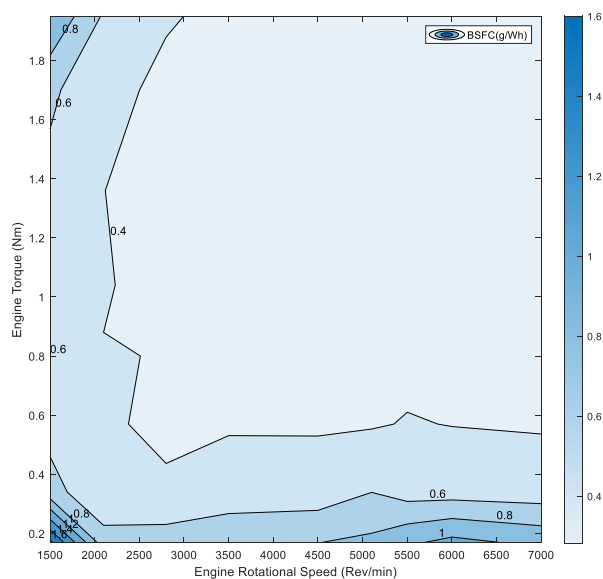


Fig. 12. Engine fuel map with BSFC contours using ICE data

Nomenclature

BSFC	brake specific fuel consumption
FF	engine fuel flow
ICE	internal combustion engine
ISA	international standard atmosphere
MAP	manifold air pressure
P_{atm}	engine atmospheric pressure
P_{ICE}	engine power
$P_{ICE,Alt}$	ICE power at a given altitude

P_{ma}	engine manifold air pressure
P_{min}	minimum engine atmospheric pressure
T_{alt}	temperature at a given altitude
T_{ICE}	engine torque
T_{sea_level}	temperature at sea level
UAV	unmanned aerial vehicle
ω	engine rotational speed
η_{th}	engine throttle

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It should be noted that the model has the following limitations:

- The model is limited to Combustion engines suitable for small UAVs, as it can only produce a maximum power of 700 W
- The model is only operational between 1000 rev/min and 10000 rev/min.

The following research directions could be explored from the present study:

- Integrating the developed model into a UAV powertrain to evaluate its functionality in simulation
- Performing an experimental setup to validate the developed model with experimental data of UAV engines from a manufacturer
- Developing scaling laws suitable for UAV combustion engines that could boost the engine power beyond 700 W.

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Synergistic effects of hydrogen enrichment and EGR on the performance and emissions of a diesel dual fuel engine: a statistical investigation

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Balancing the demand for high thermal efficiency with strict NO_x emission limits remains a persistent dilemma in modern diesel engine development. This study explores a potential solution by investigating how hydrogen enrichment interacts with EGR in a diesel dual-fuel engine under constant load. Instead of treating these variables in isolation, we employed a randomized full factorial design to map their simultaneous effects, testing hydrogen flow rates up to 7.5 LPM and EGR levels up to 15%. Through rigorous statistical analysis (ANOVA and Tukey's HSD), we uncovered a distinct "sweet spot" in engine operation. While hydrogen injection successfully boosted thermal efficiency and reduced fuel consumption due to its rapid combustion, it was expected to trigger a spike in NO_x emissions. However, our analysis revealed a crucial interaction: adding 10% EGR effectively neutralized this emission penalty without severely crippling the engine power. Interestingly, we also identified a saturation point at 7.5 LPM of hydrogen, beyond which the engine ceased to improve in efficiency due to volumetric losses. Ultimately, the data suggests that pairing 5.0 LPM of hydrogen with 10% EGR offers the most balanced configuration restoring power to baseline levels while keeping emissions in check.

Key words: ANOVA, biodiesel, diesel dual-fuel, hydrogen-EGR interaction, NO_x trade-off

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1. Introduction

The global energy crisis and environmental degradation caused by fossil fuel combustion have become key catalysts in the fundamental transformation of the transportation and power generation sectors [12]. At the national level, Indonesia commitment to achieving NZE by 2060 has been outlined in a strategic roadmap that prioritizes the energy transition [10]. CI engines remain the cornerstone in industrial and heavy transportation sectors due to their high thermal efficiency, robust durability, and superior fuel economy [33]. However, the sustainability of CI engine use is increasingly threatened by stricter emissions regulations, such as the Euro 6 standards and national decarbonization targets that mandate progressive reductions in greenhouse gas emissions [5].

Although CI engines offer high efficiency, their application faces complex environmental challenges [7]. In general, diesel dual fuel engines are recognized as major contributors to NO_x and PM/soot emissions, stemming from their inherent combustion trade-offs [24]. Yet, emission issues are not limited to these two pollutants. Byproducts of incomplete combustion, such as CO and HC, along with CO₂ greenhouse gas emissions, pose significant concerns [6]. The accumulation of these emissions not only presents serious carcinogenic risks to human health but also significantly contributes to global warming, thereby underscoring the urgent need for comprehensive mitigation strategies [2].

One of the most promising approaches to address these challenges is the use of renewable alternative fuels, such as biodiesel. However, biodiesel utilization is often constrained by its lower calorific value compared to pure diesel combustion, which negatively impacts engine performance [25]. To compensate for this limitation, dual-fuel technology with hydrogen (H₂) supplementation has gained considerable attention as a future energy solution [30]. Nationally,

hydrogen development has been designated as a key pillar in Indonesia Hydrogen and Ammonia Roadmap, where hydrogen is projected not only as an industrial commodity but also as a clean energy vector for decarbonizing the transportation sector [21]. Hydrogen has unique combustion characteristics, including high laminar flame velocity and substantial energy content per unit mass. Its addition has been shown to increase thermal efficiency beyond that of single-fuel diesel engines [8]; however, its rapid combustion characteristics inevitably lead to sharp increases in in-cylinder temperature, thereby drastically intensifying nitrogen-based emissions, particularly NO_x [16].

To mitigate NO_x emissions in hydrogen-fueled engines, EGR is often used as a control strategy. EGR operates by recirculating a portion of exhaust gases back into the combustion chamber, thereby reducing oxygen concentration and peak combustion temperature [13]. Nevertheless, the primary challenge of EGR implementation lies in the potential penalties on combustion efficiency and operational stability if its ratio is not precisely optimized relative to the main fuel supply.

Recent studies have extensively explored hydrogen-enrichment strategies across various flow rates. Sathyamurthy et al. investigated low-to-medium flow ranges (0–10 LPM). They reported that hydrogen induction consistently improved thermal efficiency beyond levels achieved with biodiesel, albeit accompanied by sharp increases in NO_x emissions [27, 32]. These findings were reinforced by Elnajjar et al., who extended the operating range up to 18.4 LPM, achieving substantial performance gains but with increasingly unavoidable NO_x risks [3]. To counteract this escalation, EGR techniques have been applied in several dual-fuel studies. Kannappan et al. tested a constant hydrogen flow (6 LPM) combined with varying EGR levels (10% and 20%) in a Syzygium cumini biodiesel engine, recording

NO_x reductions of up to 25% compared to non-EGR operation. However, performance declined slightly relative to diesel single fuel (B100 and D100) at full load [13]. Similarly, Zhang et al. optimized a broader EGR range (0–30%), concluding that while EGR effectively suppressed NO_x levels above 20% tended to worsen soot emissions [34]. Meanwhile, Marikatti et al. highlighted that EGR addition (10–20%) successfully mitigated hydrogen explosive combustion behavior, though stability at low loads remained challenging compared to the steadiness of mono-fuel diesel flames [19].

Recent research has begun using advanced statistical methods and fundamental analysis of low-carbon fuels to optimize the diesel dual-fuel engines. Bora et al. demonstrated that analysis of variance (ANOVA) effectively maps the nonlinear relationship between engine load and injection pressure, achieving optimal thermal efficiency (BTE) in a hydrogen-powered engine [11]. Meanwhile, a comparative study by Grab-Rogaliński et al. found that hydrogen enrichment improves thermal efficiency but doubles NO_x emissions via thermal mechanisms, driven by higher combustion temperatures [8]. These studies lay a solid foundation for optimizing injection parameters and understanding emission mechanisms. However, a gap remains in describing how hydrogen flow and EGR mitigation strategies interact simultaneously.

To address this gap, this study offers a more comprehensive approach. Its originality lies in testing specific combinations of hydrogen flow and EGR levels while quantitatively describing their interaction in one integrated experiment. The study aims to use statistical analysis (ANOVA and Tukey's HSD) to map the synergistic effect between hydrogen enrichment and EGR application and to identify precise 'compensation points'. The focus is on quantifying the trade-off between energy efficiency and pollutant reduction, so hydrogen-induced NO_x emissions can be reduced without compromising engine thermal performance at specific operating conditions.

2. Material and method

This section details the experimental apparatus and methodological framework used to evaluate the combined effects of hydrogen induction and EGR on a CI engine fueled with biodiesel. The investigation was designed to systematically quantify the trade-offs between performance parameters (BP, BSFC, BTE) and NO_x emissions. The specifications of the test fuels, standard engine configuration, instrumentation accuracy, and the statistical design of the experiment are described in the following subsections.

2.1. Fuel properties and preparation

This experimental investigation is focused on the exclusive use of renewable fuels. The primary liquid fuel employed is pure Palm Oil Methyl Ester (POME) or B100 biodiesel, while hydrogen gas (H₂) is introduced as an energy supplement through the intake system. The B100 biodiesel used in this study was produced from crude palm oil (CPO) as the raw material.

In this work, engine operation with pure B100 (without hydrogen) is established as the baseline condition to evaluate the effectiveness of hydrogen supplementation and EGR

intervention. The physicochemical properties of B100 and hydrogen utilized in the tests are summarized in Table 1.

Table 1. Fuel properties

Properties	Unit	POME CPO (B100)	Hydrogen (H ₂)
Density 15°C	kg/m ³	870	0.089
Kinematic viscosity 40°C	mm ² /s (cSt)	4.5	0
Lower heating value	MJ/kg	39.910	119.810
Cetane number	—	55	—
Flash point	°C	170	—
Auto ignition temperature	°C	> 101	585
Stoichiometric air-fuel ratio	—	12.5	34.3

2.2. Experimental setup

The central experimental unit consists of a single-cylinder, four-stroke, water-cooled Diamond DI 800 H diesel dual fuel engine with direct injection. The engine is coupled with an Eddy Current Dynamometer to vary the engine load during testing. In line with the study focus on practical applicability, the engine was maintained in its original factory configuration. No mechanical modifications were made to the compression ratio, fuel injection timing, or combustion chamber geometry. The engine technical specifications are presented in Table 2.

Table 2. Diamond DI 800 H engine specifications

Parameter	Specifications
Brand & model	Diamond DI 800 H
Engine type	1-cylinder, 4-stroke, horizontal, water cooled
Combustion system	Direct injection
Bore x stroke	82 mm x 78 mm
Cylinder volume (displacement)	411 cc
Compression ratio	18 : 1
Maximum power (max output)	5.96 kW at 2400 rpm
Rated power (rated output)	5.22 kW at 2200 rpm
Max torque	25.5 Nm at 1900 rpm
Cooling system	Hopper type
Lubrication system	Trochoid type
Pilot injection timing	13 BTDC
Rotation direction	Counter-clockwise

Modifications were made to the air intake system to accommodate dual-fuel operation and EGR. A schematic of the experimental setup is shown in Fig. 1.

Hydrogen gas is supplied from a high-pressure cylinder (200 bar) equipped with a two-stage pressure regulator. To ensure precise flow control, hydrogen is directed through a Digital Mass Flow Controller (MFC) with a measurement range of 0–20 LPM (accuracy: ±1% FS). The hydrogen is then injected into the intake manifold using the fumigation technique (continuous port injection). For safety purposes, a flashback arrestor is installed along the hydrogen supply line to prevent flashback from the combustion chamber.

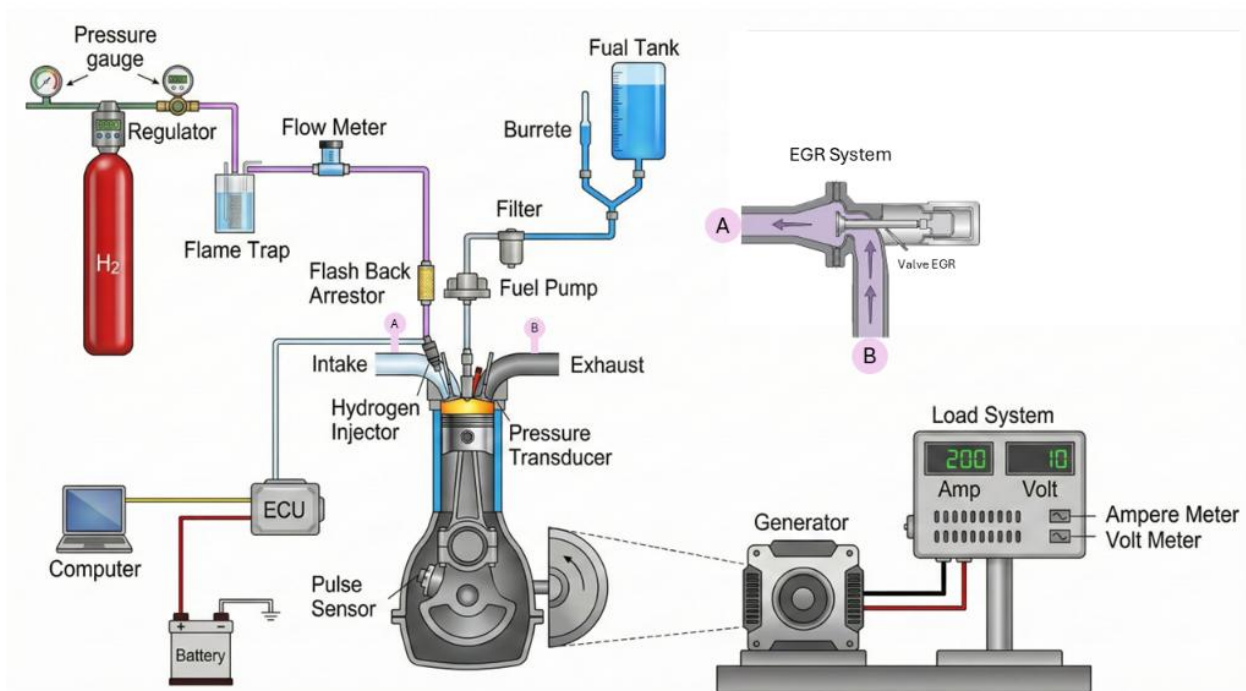


Fig. 1. Experimental setup

Table 3. Test instrument specifications

Generator electrical dynamometer		Gas mass flowmeter		Gas analyzer		Burette	
Type	ST-6	Type	MF5712 (air flow meter)	Type	Gas analyzer (EPSC5)	Type	Pyrex
Volt-age/current	230 V / 26.1 A (AC)	Flow range	0–200 SLPM	Gas measurement	CO, HC, CO ₂ , O ₂ , NO _x , Lamda	Measure range	0–100 mL
Max. AC output	6 kW	Accuracy	±(2.0% + 0.5% FS)	NO _x measurement range	0–5000 ppm	Accuracy	0.05 mL
Frequency	50 Hz	Repeatability	±0.5% FS	Accuracy	1 ppm		
Loading system	Electric bulb system	Turn-down ratio	30:1	Response time	10 s		
Electric control	Volt meter, amperemeter, switch	Response time	≤ 2 s	Power consumption	50 W at 110–220 V		

The EGR system employed is of the external low-pressure loop type. A bypass pipeline connects the exhaust manifold back to the intake manifold, positioned immediately after the air filter but before the hydrogen injection point. The EGR flow rate is manually regulated using a precision gate valve.

2.3. Instrumentation and data acquisition

The technical specifications of the leading equipment used for engine loading, fuel flow rate measurement, and exhaust gas emission analysis are summarized in Table 3.

2.4. Design of experiment & test procedure

The experimental design (DoE) in this study is focused on investigating the influence of two main independent variables, namely hydrogen flow rate and EGR level, on the characteristics of a diesel dual fuel engine. All tests were conducted under a constant load of 2 kW to ensure consistent comparison of thermal performance and emissions. Hydrogen flow rates were varied at 2.5, 5.0, and 7.5 LPM, while EGR levels were set at 5%, 10%, and 15%. Operational data from biodiesel mode (single-fuel) without EGR were first recorded as baseline data to evaluate the effec-

tiveness of energy substitution and emission-reduction strategies.

The testing procedure began with a thorough inspection of measurement instruments and calibration of the gas analyzer under ambient air conditions (zeroing). The engine was then warmed up for 15–20 minutes, until the coolant temperature reached a stable operating range (70–80°C). Once thermal stability was achieved, a 2 kW electrical load was gradually applied through the dynamometer generator. During data acquisition, hydrogen gas was injected into the intake manifold at specified flow rates (2.5, 5.0, and 7.5 LPM) using a Mass Flow Meter. After the engine speed stabilized, the EGR valve was manually opened to achieve the target EGR levels (5%, 10%, and 15%). Fuel consumption, electrical parameters, and exhaust emissions were recorded only when the engine had reached steady-state conditions for at least 3 minutes at each test point combination.

The procedure concluded by shutting off the hydrogen supply, closing the EGR valve, releasing the engine load, and allowing the engine to run at idle for cooling before being switched off.

2.5. Data processing and statistical analysis

The experimental data obtained from engine testing (engine performance: BP, BSFC, BTE and emissions: NO_x) were analyzed using statistical methods to evaluate the influence of the independent variables (hydrogen flow and EGR rate) on the dependent variables. All statistical analyses were conducted at a 95% confidence level ($\alpha = 0.05$) using Minitab statistical software. The data analysis procedure was carried out in three main stages: (1) verification of assumption checks, (2) analysis of variance (ANOVA), and (3) Post-hoc analysis [22].

The following hypotheses were formulated for this analysis:

- Normality
 - H_0 : The residuals are normally distributed
 - H_1 : The residuals are not normally distributed
- Constant variance
 - H_0 : $\sigma_{11}^2 = \sigma_{12}^2 = \sigma_{13}^2 = \sigma_{21}^2 = \sigma_{22}^2 = \sigma_{23}^2$
 - H_1 : At least one σ^2 is different
- ANOVA
 - Hydrogen Flow
 - H_0 : $\tau_1, \tau_2, \tau_3 = 0$
 - H_1 : At least one $\tau_1, \tau_2, \tau_3 \neq 0$
 - EGR
 - H_0 : $\beta_1, \beta_2, \beta_3 = 0$
 - H_1 : At least one $\beta_1, \beta_2, \beta_3 \neq 0$
 - Interaction (Hydrogen Flow*EGR)
 - H_0 : $\tau\beta_{ij} = 0$
 - H_1 : $\tau\beta_{ij} \neq 0$
- Post-hoc (comparison)
 - Hydrogen Flow
 - H_0 : $\mu_{i1} = \mu_{i2}$

$$H_1 : \mu_{i1} \neq \mu_{i2}$$

EGR

$$H_0 : \mu_{j1} = \mu_{j2}$$

$$H_1 : \mu_{j1} \neq \mu_{j2}$$

Interaction (Hydrogen Flow*EGR)

$$H_0 : \mu_{ij} = \mu_{ij}$$

$$H_1 : \mu_{ij} \neq \mu_{ij}$$

3. Result and discussion

The quantitative data obtained from the acquisition system were averaged to ensure reliability. The recapitulation of the experimental results for all test configurations is presented in Table 4. This dataset serves as the quantitative foundation for the subsequent statistical analysis.

3.1. Statistical check assumptions

3.1.1. Normality test

The validity of the statistical model was first evaluated by testing the normality of residuals to ensure the appropriateness of applying parametric ANOVA methods. This assessment was conducted using the Normal Probability Plot technique and the Anderson-Darling statistical test [22]. Based on the analysis results in Table 5, the mean residual values for all response parameters were minimal, approaching zero (ranging from 10^{-14} to 10^{-16}). Furthermore, the P-values obtained for each variable were 0.593 for BP, 0.588 for BTE, 0.824 for BSFC, and 0.198 for NO_x emissions. Since all P-values were significantly greater than the predetermined significance level ($\alpha = 0.05$), the null hypothesis (H_0) was accepted. Accordingly, it was concluded that the residual data are typically distributed, thereby fulfilling the fundamental assumption required for variance analysis (ANOVA).

Table 4. Experimental design matrix and average response values

Run order	Hydrogen Flow [l/m]	EGR [%]	BP [kW]	BSFC [g/kWh]	BTE [%]	NO _x [ppm]
1	2.5	15	2.095	339.73	21.43	175.629
2	5	10	2.184	262.87	21.54	280.783
3	5	5	2.2	276.92	20.9	298.768
4	7.5	10	2.191	238.11	19.14	315.345
5	2.5	5	2.18	366	20.31	280.929
6	5	5	2.203	276.03	20.95	295.877
7	5	15	2.178	258.51	21.74	285.928
8	5	15	2.173	259.45	21.68	286.928
9	5	10	2.184	264.53	21.45	279.783
10	7.5	15	2.163	238.49	19	287.232
11	2.5	5	2.183	371.19	20.08	278.726
12	2.5	10	2.126	328.25	22.09	227.687
13	2.5	10	2.132	343.3	21.32	228.935
14	2.5	5	2.175	352.63	20.93	284.937
15	5	15	2.168	262.13	21.52	284.928
16	7.5	10	2.197	236.44	19.24	310.614
17	2.5	10	2.125	335.98	21.68	231.952
18	7.5	15	2.165	237.57	19.05	285.939
19	7.5	10	2.184	239.08	19.07	312.456
20	7.5	5	2.213	245.92	18.92	348.238
21	7.5	15	2.176	235.2	19.19	270.925
22	7.5	5	2.216	243.14	19.05	347.439
23	2.5	15	2.071	337.74	21.48	168.763
24	5	5	2.207	277.44	20.89	292.345
25	5	10	2.182	265.42	21.4	281.783
26	7.5	5	2.22	239.99	19.19	345.938
27	2.5	15	2.089	337.19	21.54	177.136

Table 5. Normality test

Response	Mean	P-value
BP	$-1.315852 \times 10^{-16}$	0.593
BSFC	$-1.779365 \times 10^{-16}$	0.588
BTE	6.227376×10^{-14}	0.824
NO _x	$-9.789701 \times 10^{-14}$	0.198

3.1.2. Constant variance

Subsequently, the homogeneity of variance across treatment groups (Hydrogen Flow Rate and EGR) was verified using Levene's test [22]. Based on the statistical results in Table 6, the P-values obtained for each response were 0.695 (BP), 0.200 (BSFC), 0.234 (BTE), and 0.737 (NO_x emissions). All probability values were well above the $\alpha = 0.05$ significance threshold, indicating that the variances among treatment groups were uniform and not significantly different. Therefore, the null hypothesis (H₀) was accepted, and the assumption of variance homogeneity was confirmed for all tested parameters, thereby validating the application of ANOVA for further comparative analysis.

Table 6. Constant variance test

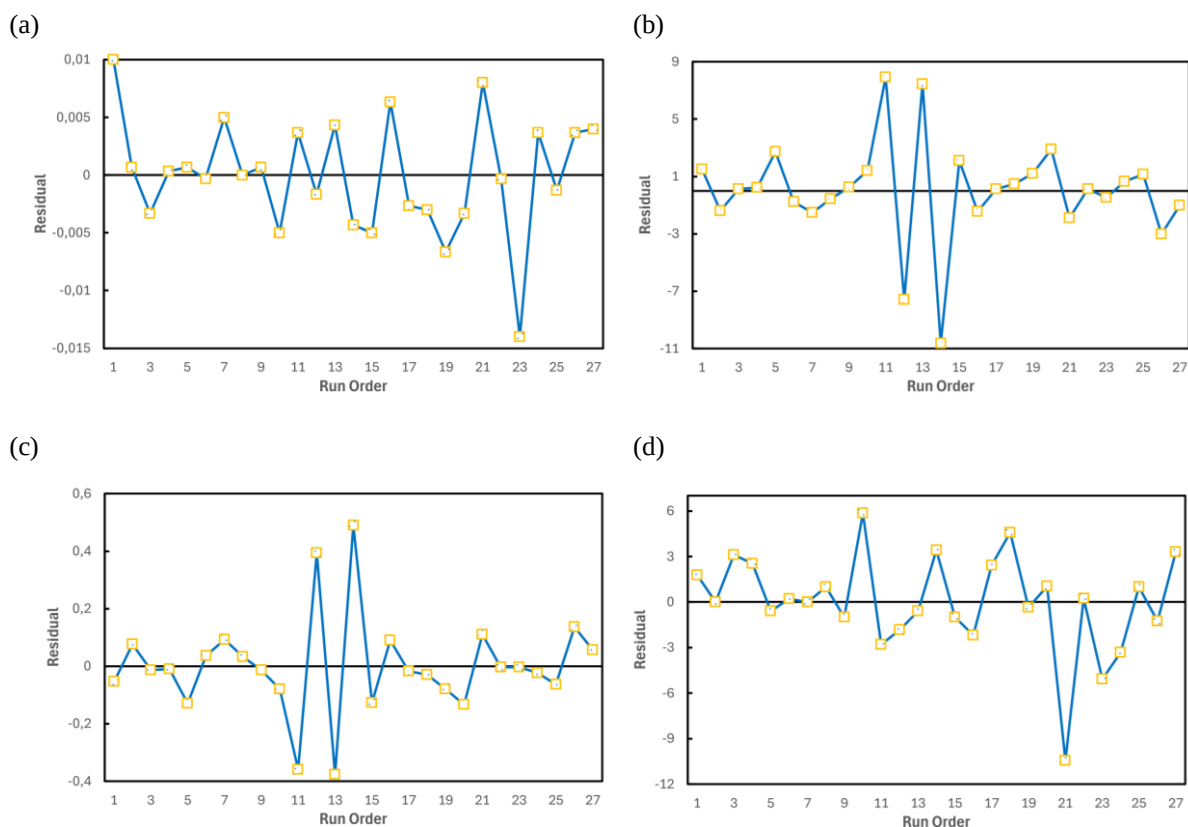
Response	Test Statistic	P-value
BP	0.69	0.695
BSFC	1.58	0.200
BTE	1.48	0.234
NO _x	0.64	0.737

3.1.3. Independent test

The independence of residuals was verified using Residuals versus Observation Order plots to detect potential autocorrelation or time-related bias during the experiment. This assumption requires that errors between observations are uncorrelated, as indicated by random fluctuations without cyclic patterns or temporal trends [22]. Based on visual evaluation of the BP, BTE, BSFC, and NO_x plots, the residuals were observed to scatter stochastically around the zero line throughout the testing sequence. No systematic patterns, consistent upward or downward trends, or indications of long runs suggesting data dependence were detected. The absence of serial correlation confirms that each data acquisition was independent, thereby fully satisfying the ANOVA model independence assumption [4].

3.1.4. Residual versus fitted values

Finally, the Residuals versus Fitted Values plots were analyzed to detect signs of heteroscedasticity and to verify the independence of error variance. Theoretically, a valid model is characterized by residuals that are randomly scattered around the zero line, without forming any specific geometric patterns [22]. Based on visual inspection of the four response parameters (BP, BTE, BSFC, and NO_x), all plots exhibited randomly distributed residuals with no structural patterns. No funnel-shaped or megaphone-like widening trends were observed with increasing fitted values, which are typical of unequal variance. The absence of systematic patterns confirms that the constant variance assumption is fully satisfied, thereby reinforcing the robustness of the ANOVA model for further interpretation [1].

Fig. 2. Independent test result graphs: (a) BP, (b) BSFC, (c) BTE, (d) NO_x

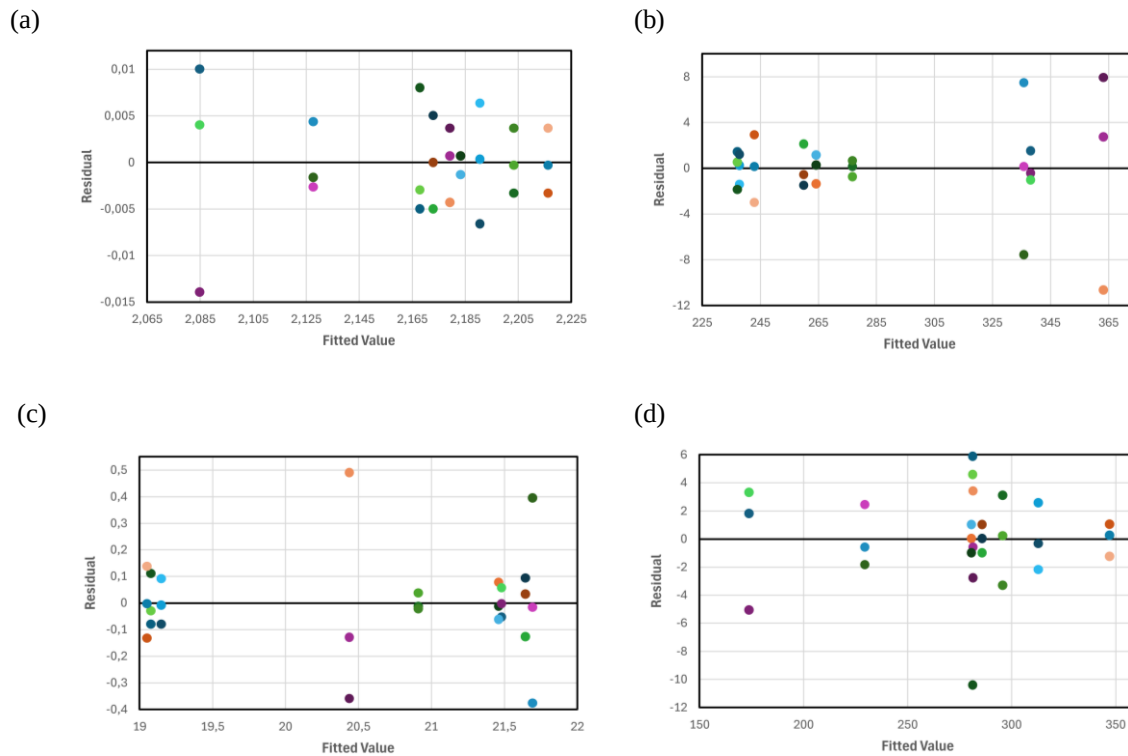
Fig. 3. Plot of residual results versus fitted value: (a) BP, (b) BSFC, (c) BTE, (d) NO_x

Table 7. Summary of ANOVA results for performance and emission parameters

Factor (X)	Analysis of variance							
	BP		BSFC		BTE		NO _x	
Source	F-Value	P-Value	F-Value	P-Value	F-Value	P-Value	F-Value	P-Value
Hydrogen flow	281.93	0.000	1459.3	0.000	322.34	0.000	1139.99	0.000
EGR	205.75	0.000	38.28	0.000	25.88	0.000	556.83	0.000
Hydrogen flow*EGR	22.44	0.000	6.16	0.003	7.37	0.001	122.18	0.000

3.2. Engine performance and emission ANOVA analysis

The analysis of variance (ANOVA) results in Table 7 statistically confirm that hydrogen injection, EGR rate, and their interaction significantly influence all response parameters ($P < 0.05$).

Specifically, hydrogen flow was identified as the most dominant factor affecting BSFC, as indicated by the extreme F-value of 1459.3. This dominance suggests that substituting liquid fuel with hydrogen significantly improves fuel consumption efficiency, a phenomenon consistent with hydrogen's higher specific heat of combustion than biodiesel [29].

Meanwhile, for NO_x emissions, both primary variables demonstrated nearly equivalent levels of influence, although Hydrogen Flow ($F = 1139.99$) remained slightly more dominant than EGR ($F = 556.83$). The magnitude of these F-values confirms the presence of a competing mechanism within the combustion chamber: hydrogen tends to elevate combustion temperature, thereby promoting thermal NO_x formation [28]. At the same time, EGR effectively suppresses this temperature rise through its thermal dilution effect [34].

The most noteworthy finding lies in the significance of the interaction effect (Hydrogen Flow*EGR). With P-values of 0.003 across all responses. The results demon-

strate that the influence of hydrogen supplementation on engine performance and emissions does not operate independently. But it is highly dependent on the level of EGR applied (and vice versa), for instance, in the case of NO_x emissions. The high interaction F-value (122.18) indicates that the effectiveness of EGR in reducing emissions varies nonlinearly with the amount of hydrogen injected. This validates the importance of simultaneous optimization strategies in this study to identify the optimal trade-off point between high performance and low emissions, as highlighted in previous literature [14].

3.3. Post ANOVA (comparison)

3.3.1. Brake power

Multiple-comparison testing using Tukey's Simultaneous Tests was conducted to identify specific mean differences and determine the optimal operating configuration for Brake Power (BP), as summarized in Table 8. Increasing hydrogen flow from 2.5 LPM to 5.0 LPM significantly improved performance ($T = 19.61$, $P < 0.05$), confirming that initial hydrogen injection effectively boosts combustion pressure. However further increasing the flow to 7.5 LPM did not yield a significant difference ($P = 0.200$). This suggests a saturation point where excessive hydrogen displaces intake air, reducing volumetric efficiency and offset-

ting hydrogen's calorific advantage due to limited oxygen availability [9]. Consequently, 5.0 LPM is identified as a more efficient operating point than 7.5 LPM.

Regarding EGR levels, negative T-values consistently indicated significant reductions in BP compared to baseline ($P < 0.05$). The most substantial reduction occurred at 10% EGR ($T = -20.23$), where the dilution effect of exhaust gases suppressed the fuel oxidation rate and the peak cycle temperature, thereby reducing piston work [17].

Interaction analysis revealed critical trade-off phenomena. Notably, the performance of (5.0 H₂, 10% EGR) and

(2.5 H₂, 5% EGR) was statistically identical ($P = 0.995$), demonstrating that increasing hydrogen to 5.0 LPM can compensate for the power loss caused by doubling the EGR rate. Hydrogen successfully restores the combustion reactivity suppressed by inert gases [20]. Conversely, at 15% EGR, increasing hydrogen from 5.0 to 7.5 LPM was ineffective ($P = 0.979$), reinforcing that excessive hydrogen cannot overcome high dilution levels. Thus, 5.0 LPM of hydrogen is recommended as the optimum limit, as higher flow rates do not provide statistically significant gains, especially when combined with EGR.

Table 8. Post ANOVA (comparison) BP response

Tukey Simultaneous Tests for Differences of Means	Source	T-value	P-value
Difference of Hydrogen Flow Level	5.0–2.5	19.61	0.000
	7.5–2.5	21.4	0.000
	7.5–5.0	1.79	0.200
Difference of Hydrogen Flow Level	5	-11.38	0.000
	10	-20.23	0.000
	15	-8.85	0.000
Difference of Hydrogen Flow*EGR Level	(2.5 10)–(2.5 5)	-10.47	0.000
	(2.5 15)–(2.5 5)	-19.11	0.000
	(5.0 5)–(2.5 5)	4.86	0.003
	(5.0 10)–(2.5 5)	0.81	0.995
	(5.0 15)–(2.5 5)	-1.28	0.924
	(7.5 5)–(2.5 5)	7.49	0.000
	(7.5 10)–(2.5 5)	2.3	0.393
	(7.5 15)–(2.5 5)	-2.3	0.393
	(2.5 15)–(2.5 10)	-8.64	0.000
	(5.0 5)–(2.5 10)	15.33	0.000
	(5.0 10)–(2.5 10)	11.28	0.000
	(5.0 15)–(2.5 10)	9.18	0.000
	(7.5 5)–(2.5 10)	17.96	0.000
	(7.5 10)–(2.5 10)	12.76	0.000
	(7.5 15)–(2.5 10)	8.17	0.000
	(5.0 5)–(2.5 15)	23.97	0.000
	(5.0 10)–(2.5 15)	19.92	0.000
	(5.0 15)–(2.5 15)	17.83	0.000
	(7.5 5)–(2.5 15)	26.6	0.000
	(7.5 10)–(2.5 15)	21.4	0.000
	(7.5 15)–(2.5 15)	16.81	0.000
	(5.0 10)–(5.0 5)	-4.05	0.017
	(5.0 15)–(5.0 5)	-6.14	0.000
	(7.5 5)–(5.0 5)	2.63	0.239
	(7.5 10)–(5.0 5)	-2.57	0.266
	(7.5 15)–(5.0 5)	-7.16	0.000
	(5.0 15)–(5.0 10)	-2.09	0.506
	(7.5 5)–(5.0 10)	6.68	0.000
	(7.5 10)–(5.0 10)	1.49	0.848
	(7.5 15)–(5.0 10)	-3.11	0.106
	(7.5 5)–(5.0 15)	8.78	0.000
	(7.5 10)–(5.0 15)	3.58	0.043
	(7.5 15)–(5.0 15)	-1.01	0.979
	(7.5 10)–(7.5 5)	-5.2	0.002
	(7.5 15)–(7.5 5)	-9.79	0.000
	(7.5 15)–(7.5 10)	-4.59	0.005

Table 9. Post ANOVA (comparison) BSFC response

Tukey Simultaneous Tests for Differences of Means	Source	T-value	P-value
Difference of Hydrogen Flow Level	5.0–2.5	-38.51	0.000
	7.5–2.5	-52.07	0.000
	7.5–5.0	-13.55	0.000
Difference of Hydrogen Flow Level	5	-7.35	0.000
	10	-7.78	0.000
	15	-0.43	0.902
Difference of Hydrogen Flow*EGR Level	(2.5 10)–(2.5 5)	-7.75	0.000
	(2.5 15)–(2.5 5)	-7.07	0.000
	(5.0 5)–(2.5 5)	-24.42	0.000
	(5.0 10)–(2.5 5)	-27.96	0.000
	(5.0 15)–(2.5 5)	-29.15	0.000
	(7.5 5)–(2.5 5)	-33.96	0.000
	(7.5 10)–(2.5 5)	-35.41	0.000
	(7.5 15)–(2.5 5)	-35.63	0.000
	(2.5 15)–(2.5 10)	0.67	0.999
	(5.0 5)–(2.5 10)	-16.67	0.000
	(5.0 10)–(2.5 10)	-20.21	0.000
	(5.0 15)–(2.5 10)	-21.41	0.000
	(7.5 5)–(2.5 10)	-26.21	0.000
	(7.5 10)–(2.5 10)	-27.66	0.000
	(7.5 15)–(2.5 10)	-27.89	0.000
	(5.0 5)–(2.5 15)	-17.34	0.000
	(5.0 10)–(2.5 15)	-20.88	0.000
	(5.0 15)–(2.5 15)	-22.08	0.000
	(7.5 5)–(2.5 15)	-26.88	0.000
	(7.5 10)–(2.5 15)	-28.34	0.000
	(7.5 15)–(2.5 15)	-28.56	0.000
	(5.0 10)–(5.0 5)	-3.54	0.047
	(5.0 15)–(5.0 5)	-4.73	0.004
	(7.5 5)–(5.0 5)	-9.54	0.000
	(7.5 10)–(5.0 5)	-10.99	0.000
	(7.5 15)–(5.0 5)	-11.21	0.000
	(5.0 15)–(5.0 10)	-1.2	0.947
	(7.5 5)–(5.0 10)	-6	0.000
	(7.5 10)–(5.0 10)	-7.45	0.000
	(7.5 15)–(5.0 10)	-7.68	0.000
	(7.5 5)–(5.0 15)	-4.8	0.004
	(7.5 10)–(5.0 15)	-6.26	0.000
	(7.5 15)–(5.0 15)	-6.48	0.000
	(7.5 10)–(7.5 5)	-1.45	0.863
	(7.5 15)–(7.5 5)	-1.67	0.754
	(7.5 15)–(7.5 10)	-0.22	1.000

3.3.2 Brake specific fuel consumption

In contrast to the saturation observed in BP, the multiple-comparison analysis for BSFC shows a linear, monotonic improvement trend (Table 9). For instance, transitions from 2.5 LPM to 5.0 LPM, and further to 7.5 LPM, consistently yielded significant reductions in fuel consumption ($P < 0.001$, $T = -38.51$ and -13.55). This sustained improvement is primarily driven by hydrogen's high Lower Heating Value (~ 120 MJ/kg); substituting liquid fuel with a lighter, more energetic gaseous fuel naturally lowers the mass-to-energy ratio (g/kWh) [18]. Furthermore, hydrogen's high diffusivity promotes more complete combustion, ensuring better energy conversion across all tested levels [15].

A notable finding in the interaction analysis is the diminishing influence of EGR at higher hydrogen substitution rates. Specifically, at 7.5 LPM of hydrogen, variations in EGR levels (5%, 10%, and 15%) no longer significantly impacted fuel consumption, evidenced by high P-values ($P = 1.000$ for the 15% vs. 10% comparison). These results suggest that at sufficient concentrations, hydrogen's reactivity dominates the combustion characteristics. In this state, the typical efficiency penalties associated with EGR, such as slower combustion or pumping losses, are effectively masked or mitigated by hydrogen's high thermal efficiency [17].

While the BP analysis suggested an optimal limit of 5.0 LPM due to volumetric efficiency trade-offs, the BSFC results indicate that 7.5 LPM remains the superior choice for fuel economy. However, the engine's sensitivity to EGR becomes more apparent at lower hydrogen flows. For example, at 2.5 LPM, increasing EGR from 5% to 10% significantly altered BSFC ($P < 0.001$). Though further increases to 15% showed no significant change ($P = 0.999$), this indicates that, while a minimal hydrogen supply initially makes the engine susceptible to EGR-induced disturbances, it eventually reaches a tolerance threshold [26]. Overall, high hydrogen concentrations provide a "shielding" effect, allowing for more flexible emission control strategies without compromising fuel economy.

3.3.3 Brake thermal efficiency

Post-hoc analysis in Table 10 reveals a distinct thermal behavior: an increase in the hydrogen energy fraction does not always translate into improved thermal efficiency, unlike the linear improvement observed in BSFC. BTE exhibits a clear operational limit. Tukey test results indicate that BTE reaches its optimum within the low-to-medium injection range. The comparison between 2.5 LPM and 5.0 LPM hydrogen flow yielded a T-value of 1.35 with a P-value of 0.385. This lack of statistical significance suggests that increasing the hydrogen supply within this range does not provide a meaningful boost to energy conversion efficiency.

A more critical shift occurs at 7.5 LPM, where a sharp decline in performance is observed. This is confirmed by large negative T-values (-21.28 and -22.63) and $P < 0.001$ when compared to both the 2.5 and 5.0 LPM levels. This phenomenon indicates that at 7.5 LPM, the engine experiences reduced volumetric efficiency due to the high volume of hydrogen displacing intake air. The air-fuel ratio deviates from the optimal combustion point, resulting in reduced energy conversion despite hydrogen's high calorific value [9].

One of the most compelling findings from this analysis is the interaction behavior under high-hydrogen conditions. At a flow rate of 7.5 LPM, variations in EGR levels (5% to 15%) had no impact on BTE, as evidenced by "perfect" P-values of 1.000 across these comparisons. This suggests that at high concentrations, hydrogen combustion becomes so dominant that it effectively overrides the thermodynamic effects typically associated with EGR.

In contrast, at a lower flow of 2.5 LPM, the engine remains highly sensitive to EGR changes ($P < 0.001$ for the 5% vs. 10% EGR comparison). This implies that when the hydrogen proportion is small, diesel diffusion mechanisms still govern the combustion process, which are inherently susceptible to the presence of inert gases.

In summary, to maximize thermal efficiency while maintaining reasonable fuel consumption, an operating range of 2.5–5.0 LPM is recommended. Using 7.5 LPM of hydrogen should be avoided, as it results in a significant loss in conversion efficiency.

3.3.4 Nitrogen oxide

Post-hoc analysis of NO_x emissions reveals a critical thermal interaction between hydrogen enrichment and EGR. While previous sections focused on performance, this analysis evaluates the strategic mitigation of environmental impacts. Notably, the Tukey test results in Table 10 confirm a strong positive correlation between hydrogen flow and NO_x formation. Comparisons across all hydrogen levels (2.5, 5.0, and 7.5 LPM) yielded significant increases ($P < 0.001$), with T-values reaching 46.62. This trend is a direct consequence of hydrogen's high adiabatic flame temperature, which accelerates the heat release rate and creates localized temperature spikes. These conditions promote thermal NO_x formation via the Zeldovich mechanism in an exponential manner [23]. Without EGR intervention, the high hydrogen flow rate (7.5 LPM) would likely result in emissions exceeding regulatory standards.

Importantly, a fundamental finding of this study is EGR's capacity to neutralize the NO_x increases induced by hydrogen. Specifically, interaction analysis demonstrates a precise compensation pattern; for instance, while NO_x rose significantly at 5.0 LPM of hydrogen under low EGR, increasing the EGR rate to 10% restored emissions to baseline levels. This is evidenced by the comparison between (5.0 H_2 10% EGR) and the baseline (2.5 H_2 5% EGR), which yielded a P-value of 1.000. This statistical equality indicates that the heat-absorbing capacity of 10% EGR effectively offsets the thermal effect of 5.0 LPM hydrogen. Similarly, at maximum hydrogen levels, applying 15% EGR to the 7.5 LPM hydrogen configuration resulted in an emission profile statistically identical to the baseline ($P = 1.000$, $T = -0.05$).

In contrast to the BTE analysis, where EGR lost its influence at high hydrogen concentrations, EGR remains highly effective for NO_x reduction even at 7.5 LPM hydrogen. Specifically, comparisons within the 7.5 LPM group (e.g., 15% vs. 5% EGR) showed substantial reductions ($P < 0.001$, $T = -20.73$). This confirms that the high specific heat capacity of exhaust gas constituents, particularly CO_2 and H_2O , remains capable of absorbing the intense heat of hydrogen combustion. These results align with Vellaiyan

[31]. confirming that EGR technology in biodiesel-hydrogen engines can significantly suppress nitrogen oxide formation. In summary, these results validate an adaptive control strategy: increasing hydrogen flow enhances performance (BP/BSFC). The EGR level must be raised proportionally to maintain NO_x neutrality. Thus, the operating pairs of (5.0 LPM H₂ + 10% EGR) and (7.5 LPM H₂ + 15% EGR) are identified as optimal configurations for achieving high performance without compromising emission standards.

4. Conclusion

This study successfully evaluated the synergistic and antagonistic effects of hydrogen induction and EGR in a biodiesel-based diesel dual-fuel engine using a comprehensive statistical approach. Based on the experimental data analysis and hypothesis testing, several key conclusions can be drawn:

- Hydrogen induction proved to be a highly effective strategy for enhancing energy efficiency. The addition of hydrogen consistently reduced BSFC linearly, due to

Table 10. Post ANOVA (comparison) BTE response

Tukey Simultaneous Tests for Differences of Means	Source	T-value	P-value
Difference of Hydrogen Flow Level	5.0–2.5	1.35	0.385
	7.5–2.5	–21.28	0.000
	7.5–5.0	–22.63	0.000
Difference of Hydrogen Flow Level	5	6.39	0.000
	10	6.06	0.000
	15	–0.34	0.940
Difference of Hydrogen Flow*EGR Level	(2.5 10)–(2.5 5)	7.31	0.000
	(2.5 15)–(2.5 5)	6.07	0.000
	(5.0 5)–(2.5 5)	2.75	0.197
	(5.0 10)–(2.5 5)	5.95	0.000
	(5.0 15)–(2.5 5)	7.02	0.000
	(7.5 5)–(2.5 5)	–8.07	0.000
	(7.5 10)–(2.5 5)	–7.5	0.000
	(7.5 15)–(2.5 5)	–7.91	0.000
	(2.5 15)–(2.5 10)	–1.24	0.936
	(5.0 5)–(2.5 10)	–4.56	0.006
	(5.0 10)–(2.5 10)	–1.36	0.900
	(5.0 15)–(2.5 10)	–0.29	1.000
	(7.5 5)–(2.5 10)	–15.38	0.000
	(7.5 10)–(2.5 10)	–14.81	0.000
	(7.5 15)–(2.5 10)	–15.22	0.000
	(5.0 5)–(2.5 15)	–3.32	0.072
	(5.0 10)–(2.5 15)	–0.12	1.000
	(5.0 15)–(2.5 15)	0.95	0.986
	(7.5 5)–(2.5 15)	–14.13	0.000
	(7.5 10)–(2.5 15)	–13.57	0.000
	(7.5 15)–(2.5 15)	–13.98	0.000
	(5.0 10)–(5.0 5)	3.2	0.090
	(5.0 15)–(5.0 5)	4.27	0.011
	(7.5 5)–(5.0 5)	–10.82	0.000
	(7.5 10)–(5.0 5)	–10.26	0.000
	(7.5 15)–(5.0 5)	–10.66	0.000
	(5.0 15)–(5.0 10)	1.07	0.972
	(7.5 5)–(5.0 10)	–14.02	0.000
	(7.5 10)–(5.0 10)	–13.46	0.000
	(7.5 15)–(5.0 10)	–13.86	0.000
	(7.5 5)–(5.0 15)	–15.08	0.000
	(7.5 10)–(5.0 15)	–14.52	0.000
	(7.5 15)–(5.0 15)	–14.93	0.000
	(7.5 10)–(7.5 5)	0.56	1.000
	(7.5 15)–(7.5 5)	0.16	1.000
	(7.5 15)–(7.5 10)	–0.41	1

Table 11. Post ANOVA (comparison) NO_x response

Tukey Simultaneous Tests for Differences of Means	Source	T-value	P-value
Difference of Hydrogen Flow Level	5.0–2.5	32.26	0.000
	7.5–2.5	46.62	0.000
	7.5–5.0	14.36	0.000
Difference of Hydrogen Flow Level	5	–18.41	0.000
	10	–33.31	0.000
	15	–14.9	0.000
Difference of Hydrogen Flow*EGR Level	(2.5 10)–(2.5 5)	–16.37	0.000
	(2.5 15)–(2.5 5)	–33.9	0.000
	(5.0 5)–(2.5 5)	4.45	0.007
	(5.0 10)–(2.5 5)	–0.24	1.000
	(5.0 15)–(2.5 5)	1.38	0.890
	(7.5 5)–(2.5 5)	20.68	0.000
	(7.5 10)–(2.5 5)	9.85	0.000
	(7.5 15)–(2.5 5)	–0.05	1.000
	(2.5 15)–(2.5 10)	–17.53	0.000
	(5.0 5)–(2.5 10)	20.82	0.000
	(5.0 10)–(2.5 10)	16.14	0.000
	(5.0 15)–(2.5 10)	17.76	0.000
	(7.5 5)–(2.5 10)	37.05	0.000
	(7.5 10)–(2.5 10)	26.22	0.000
	(7.5 15)–(2.5 10)	16.32	0.000
	(5.0 5)–(2.5 15)	38.35	0.000
	(5.0 10)–(2.5 15)	33.67	0.000
	(5.0 15)–(2.5 15)	35.29	0.000
	(7.5 5)–(2.5 15)	54.58	0.000
	(7.5 10)–(2.5 15)	43.75	0.000
	(7.5 15)–(2.5 15)	33.85	0.000
	(5.0 10)–(5.0 5)	–4.68	0.005
	(5.0 15)–(5.0 5)	–3.06	0.115
	(7.5 5)–(5.0 5)	16.23	0.000
	(7.5 10)–(5.0 5)	5.4	0.001
	(7.5 15)–(5.0 5)	–4.5	0.007
	(5.0 15)–(5.0 10)	1.62	0.783
	(7.5 5)–(5.0 10)	20.91	0.000
	(7.5 10)–(5.0 10)	10.08	0.000
	(7.5 15)–(5.0 10)	0.18	1.000
	(7.5 5)–(5.0 15)	19.29	0.000
	(7.5 10)–(5.0 15)	8.46	0.000
	(7.5 15)–(5.0 15)	–1.44	0.869
	(7.5 10)–(7.5 5)	–10.83	0.000
	(7.5 15)–(7.5 5)	–20.73	0.000
	(7.5 15)–(7.5 10)	–9.9	0.000

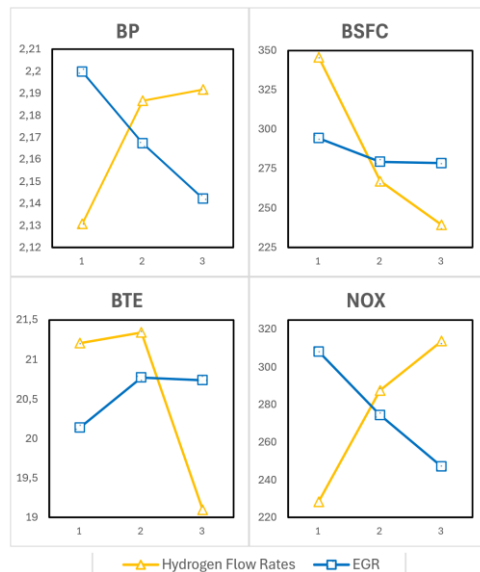


Fig. 4. Main effects plot for BP, BSFC, BTE and NO_x

high calorific value and rapid combustion. However, improvements in BP and BTE exhibited diminishing returns at higher flow rates. The 7.5 LPM configuration did not provide statistically significant power gains compared to 5.0 LPM ($P > 0.05$) and even reduced thermal efficiency due to lower volumetric efficiency. Therefore, the injection range of 2.5–5.0 LPM was identified as the most thermodynamically efficient operating zone.

- A clear trade-off exists between performance and emissions. Hydrogen significantly increased NO_x emissions via the Zeldovich mechanism, driven by elevated combustion temperatures. While EGR effectively suppressed NO_x emissions, but imposed significant penal-

ties on engine power. Variance analysis confirmed that the interaction between these two variables was highly significant ($P \leq 0.003$), indicating that the negative impact of one variable could be mitigated by the other.

- A key statistical finding was the pattern of compensation between hydrogen and EGR. In these test conditions, hydrogen reactivity offset the power loss from EGR, simultaneously. EGR's thermal capacity reduced the NO_x emission spike caused by hydrogen. Specifically, 5.0 LPM hydrogen with 10% EGR and 7.5 LPM hydrogen with 15% EGR restored NO_x emissions to baseline levels ($P = 1.000$) in this engine setup.
- Considering the balance between maximum thermal efficiency fuel economy and emission compliance, the configuration of 5.0 LPM hydrogen combined with 10% EGR is recommended as the optimal operating point (Clean High-Performance). This configuration offers significant performance improvements over mono-fuel B100 mode without compromising the NO_x emission profile.

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Nomenclature

BP brake power
BSFC brake specific fuel consumption
BTE brake thermal efficiency
CI compression ignition
CO carbon monoxide
CO₂ carbon dioxide

EGR exhaust gas recirculation
HC hydrocarbons
NO_x nitrogen oxide
NZE net zero emissions
PM particulate matter

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Analysis of uncertainty in vehicle collision calculation methods

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This article presents an uncertainty analysis of several of the most commonly used methods for calculating motor vehicle collision speeds. Based on a real-world crash test, a simulation-based collision model and an energy-based post-impact vehicle motion were developed in MATLAB. The calculation results were optimized using the Monte Carlo method. A custom-designed script was employed to search the full range of feasible solutions, and the results were compared with those obtained from analytical reconstruction methods and simulations conducted in two commercially available vehicle dynamics software packages. The study demonstrated that analytical calculations are highly sensitive to uncertainties in estimating the base post-impact trajectory angles and, depending on the reconstruction type (I, II, III, IV), to pre-impact approach angles. This sensitivity is not observed in simulation-based calculations, regardless of whether the post-impact motion is modeled using energy methods or vehicle dynamics models integrated into accident reconstruction software. Additionally, the performance of optimization algorithms built into these programs was analyzed. In this test, the uncertainties for the different methods – at measured speeds of 42.1 and 30.0 km/h – did not exceed $\pm 13\%$. In contrast, randomizing the friction coefficient (μ) resulted in uncertainties of up to $\pm 28\%$ and varied across individual vehicles. These values are consistent with the theoretical analyses conducted by other researchers.

Key words: vehicle collision, reconstruction calculations, simulation-based calculations, computational uncertainty, energy equivalent speed (EES)

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1. Introduction and aim of the study

The process of recreating road traffic incidents is commonly referred to as reconstruction [13]; however, from an engineering perspective, a distinction is made between reconstruction calculations and simulation-based calculations. Theoretical reconstruction calculations can be categorized into four fundamental types, where the selection of the appropriate model depends on the available data set, such as information regarding post-impact rotational motion, velocity vector angles, etc. The specific nature of theoretical reconstruction calculations – reflected in the mathematical forms of physical dependencies (e.g., trigonometric functions) – can lead to potentially significant computational uncertainties, sometimes exceeding 100%. These methods are highly sensitive to the angles describing the positions of the vehicles' center-of-mass velocity vectors, both at the moment of impact and immediately thereafter. Post-impact motion poses a particular challenge due to limited data availability, such as the absence of tire marks [14]. Another reconstruction method, though characterized by significantly lower uncertainty, involves calculating collisions with fixed objects (e.g., poles or trees), using empirical mathematical models [16]. Reconstruction calculations do not allow visualization of post-impact trajectories, and a single calculation run can give a misleading sense of accuracy. Therefore, it is advisable to perform multiple iterations with varying approaches and departure angles to assess the uncertainty specific to a given case.

Such uncertainties are not characteristic of computational methods implemented in simulation environments [6, 8]. These environments utilize iterative vehicle dynamics models to visualize post-impact trajectories, as demonstrated in this study using two dedicated accident reconstruction software packages: V-Sim [1, 15] and PC-Crash [10, 11].

However, simulation-based calculations can also be performed without a full vehicle dynamics model by modeling post-impact motion through energy-based relationships (e.g., Marquard, Burg, or McHenry-Marquard [16]). These models incorporate correction factors for non-parallel wheel reaction force vectors (empirical models) that account for rotational motion and sideslip angles [7, 8], although they do not model post-impact trajectories *sensu stricto*. This approach was implemented in the proprietary MATLAB algorithm presented in this paper.

The most advanced methods for vehicle collision calculations are based on the Finite Element Method (FEM) [2, 4, 5, 18]. However, these are primarily used to determine the EES (Equivalent Energy Speed) parameter, which does not represent the actual impact speed, but rather the energy of deformation work expressed in speed units. Consequently, this parameter is subsequently used as an input in some of the calculation types presented in this study. FEM methods are not commonly used by forensic experts to reconstruct impact speeds in road accidents because they are time-consuming and resource-intensive. A significant drawback is also the lack of FEM models for the wide range of vehicles used in road traffic. Simulation software used by experts typically contains only a few vehicle FEM models, mostly of American origin. These methods are primarily used for research purposes and, on rare occasions, to solve collision problems involving atypical obstacles, which are individually modeled based on specific strength test results.

It must be emphasized that the "reliability" of accident reconstruction calculations is inversely proportional to the computational uncertainty, given the unique, non-repeatable nature of a single collision. Consequently, the probability of the occurrence of a single, precise speed value tends toward

zero [17]. This perspective is dictated by legal requirements in judicial proceedings, where the range of uncertainty often determines the range of accident-avoidance possibilities and, thus, the assignment of liability [12].

In light of the above, the aim of this study was to investigate the uncertainty of calculated impact speeds using the aforementioned methods, based on a real-world crash test.

The authors were unable to identify any studies that compare the results of several different methods based on a single real-world crash test. Existing literature focuses on the uncertainty of a specific method used by researchers for a particular test; therefore, a reliable comparison of uncertainties across methods is not feasible, as different crash tests were used in each case. While study [2] presented a comparison of method uncertainties, it was based on theoretical analysis rather than specific crash tests. Their findings for various methods were as follows: limit values $\pm 30.5\%$, total derivative $\pm 31.3\%$, second-order total derivative $\pm 33.0\%$, finite differences $\pm 31.3\%$, Gaussian probabilistic method $\pm 19.0\%$, stochastic process probabilistic method $\pm 18.8\%$, and Monte-Carlo probabilistic method $\pm 29.5\%$.

Consequently, this constitutes the innovative contribution of the present publication. Nevertheless, establishing a comprehensive view of the uncertainties inherent in individual methods would require numerous analogous analyses, which remains a goal for future research.

2. Research object – crash test

2.1. Vehicles

The crash test data was retrieved from the PC-Crash software database. The collision involved two vehicles: a Ford Mondeo and an Opel Vectra, as shown in Fig. 1. The fundamental technical parameters of both vehicles are presented in Table 1.

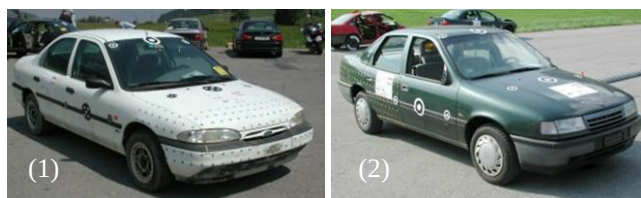


Fig. 1. View of the vehicles selected for the study: Ford (1) with an initial speed of 42.1 km/h and Opel (2) with an initial speed of 38.0 km/h

Table 1. Vehicle parameters

No.	Make, model, type	Ford (1) Mondeo MK I	Opel (2) Vectra Bi
1.	impact speed [km/h]	42.1	38.0
2.	year of manufacture	1995	1991
3.	curb weight [kg]	1306	1170
4.	length [m]	4.48	4.48
5.	width [m]	1.75	1.70
6.	height [m]	1.50	1.42
7.	wheelbase [m]	2.70	2.60
8.	track width [m]	1.50	1.50
9.	distance CoG to front axle [m]	1.03	1.09
10.	CoG height [m]	0.56	0.50

2.2. Modeling of the collision and post-impact motion

The relative positioning of the vehicles at the moment of impact, the distances traveled during post-impact motion, and their final rest positions are presented in Fig. 2.

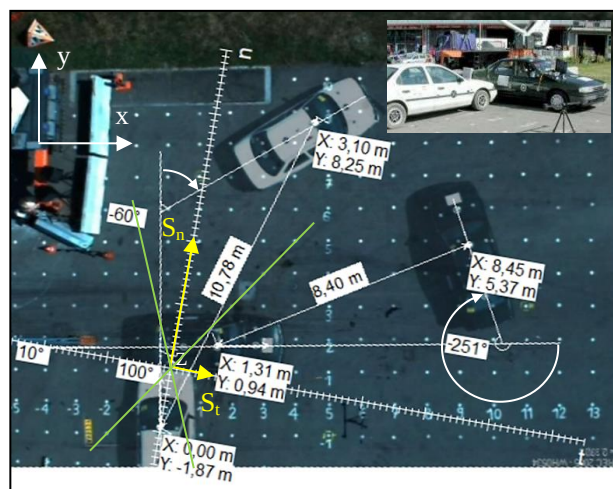


Fig. 2. Impact and final rest positions in the normal (n) and tangential (t) coordinate system relative to the vehicle deformation surface, with the global coordinate system (x, y) overlaid. Shown are the impulses components S_t and S_n (yellow) and the friction cone (green)

In simulation-based calculations, distances and angles are defined in the normal and tangential impact plane coordinate system, with the origin at the point of application of the collision impulse. The ratio of the tangential impulse component S_t to the normal component S_n is defined as the friction at the collision node and is described by the predefined parameter tgp (friction cone). If the S_t/S_n ratio equals or exceeds the given value of tgp (where the collision impulse lies on the surface of the friction cone), a so-called rough collision with sliding occurs; otherwise, it is classified as a rough collision without sliding. Furthermore, simulation-based calculations incorporate a predefined coefficient of restitution, k , that determines the amount of energy dissipated during the impact. This coefficient is nonlinearly and inversely proportional to the relative velocity of the vehicles along the impact normal. Consequently, when modeling the collision process, it is possible to influence the relative pre-impact angles, the location of the impulse application point within the contact area, the friction at the collision node, the orientation of the tangential plane, and the vehicle rebound reflected in the coefficient of restitution [6, 8]. All these parameters affect the post-impact trajectory angles and the overall dynamics of the vehicles.

Conversely, in reconstruction calculations, the reference system is the global coordinate system rather than the normal and tangential planes. These methods do not utilize the parameter or the coefficient tgp of restitution k . Therefore, one can only influence the pre- and post-impact motion angles. However, post-impact angles are difficult to determine with sufficient precision, and even minor variations in them can significantly affect the calculation results [14].

2.3. Energy equivalent speed (EES)

In both types of calculations (simulation-based and reconstruction), the Energy Equivalent Speed (EES) param-

ter is utilized. It should not be equated with the actual vehicle speed, except in one specific scenario: when a vehicle strikes a rigid, non-deformable barrier and no significant post-impact motion occurs. The EES parameter is closely related to the Equivalent Barrier Speed (EBS); however, EBS also accounts for the vehicle body's elastic work. This elastic component can typically be neglected, as it usually does not exceed at very low speeds (e.g. 10 km/h) and becomes negligible at higher speeds, which results from the velocity-dependent characteristics of the coefficient of restitution [16].

In simulation-based calculations, the EES parameter is used as a verification (control) value, whereas in reconstruction calculations it serves as an input value, though not in all such calculations. For the purpose of this article, the EES value (which, in this case, is very similar to the EBS) was calculated according to the CRASH3 standard using the RWD software [20], as schematically shown in Fig. 3.

Fig. 3. Input data and calculation results – example for vehicle (1)

For the second vehicle, the EBS was approximately 13.6 km/h.

3. Methodology – simulation-based calculations

3.1. Limitations of the current objective function

The research workstation consisted of a PC equipped with specialized software: V-Sim, PC-Crash, and RWD. To ensure the comparability of results, the Kudlich-Slibar collision model [14] was employed in both primary programs. Although these software packages include alternative collision models (two in PC-Crash and one in V-Sim), those models offer limited adjustability, and the user's ability to influence their parameters is practically restricted.

The calculations were conducted using two distinct approaches. First, the built-in optimizer of each program was

utilized. Subsequently, the data was cleared from memory, and the collision was calculated in manual mode. This procedure was necessary because built-in optimizers often function incorrectly due to flawed objective functions (quality functions). As demonstrated in [9], the objective function in PC-Crash [10], based on the principle of weighted aggregate criteria, combines parameters that are incomparable (e.g., angles, distances, EES). Consequently, the percentage error (uncertainty) calculated by the software does not correlate with the collision energy balance and thus does not represent the actual percentage uncertainty of the speed calculation. In the case of V-Sim, the authors have not disclosed the specific form of the fitness function, although the input parameters and resulting outputs suggest it is more advanced than the one in PC-Crash. Instead of a percentage error, the user receives a range of calculated speeds that satisfy predefined post-simulation motion criteria.

Nevertheless, situations still occur where the calculation results are physically unrealistic, or it is impossible to achieve post-impact vehicle positions consistent with the test documentation. In PC-Crash, it is sometimes impossible to optimize (find a solution for) a virtual collision – even one previously designed in manual mode within the same program – once the sought-after data has been reset [9]. While the vehicle models in these programs are known approximations of reality, it is understandable that they may struggle to optimize a real-world crash test. However, the algorithm should not encounter such difficulties when solving a virtual test generated within its own environment.

The criterion for the correctness of the simulation-based calculations is the achievement of post-impact (post-simulation) positions by the virtual vehicles. These positions should align as closely as possible with the actual rest positions, as the vehicles' kinetic energy is depleted during both planar and spatial motion.

3.2. Simulation-based calculations in V-Sim

The results obtained using the optimization module are presented in Fig. 4.

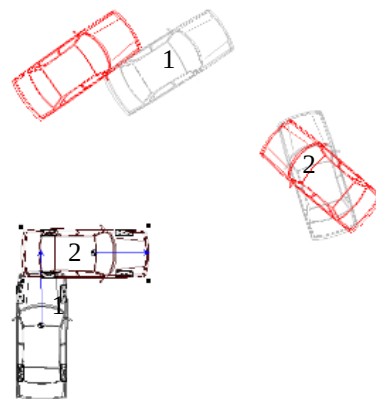


Fig. 4. V-Sim optimizer calculations (matching post-simulation positions to real-world data)

In 'manual' mode (without the use of the software's built-in optimizer), the simulation results were successfully improved, as demonstrated in Fig. 5.

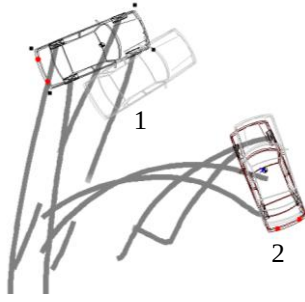


Fig. 5. V-Sim manual calculations (matching post-simulation positions to real-world data)

In both calculation cases, by appropriately selecting the input forces and accounting for the tire-road friction coefficient (μ), the delayed braking of the Ford's wheels (observed in the test video) and the lift-off of the Opel's left wheels were modeled. The calculated speed results will be presented collectively at the end of the paper.

3.3. Simulation-based calculations in PC-Crash

The use of the built-in optimizer yielded the results shown in Fig. 6.

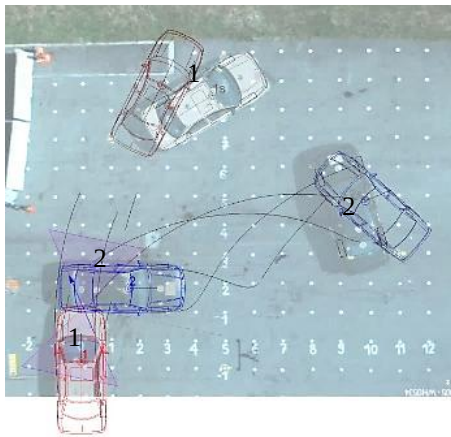


Fig. 6. PC-Crash optimizer calculations (matching post-simulation positions to real-world data)

During the calculations, the software's optimization algorithm tended to converge on local minima, resulting in unrealistic impact speed values. Furthermore, the simulated vehicles came to a rest far from their target positions. In 'manual' mode, the simulation results were significantly improved, achieving a better fit to the final rest positions than in V-Sim, as shown in Fig. 7.

In contrast to V-Sim, manual calculations using the Kudlich-Slibar model are significantly more efficient in PC-Crash, as adjustments to the impulse application point and other parameters can be made without closing the navigation window while maintaining overall stability. This may be because the software was primarily designed for the force-based collision method [15]. However, this force-based model is fully automated and does not allow for user intervention. Consequently, it is impossible to model the observed immediate post-impact motion resulting from, for instance, shifting the impulse application point (due to variable body stiffness at different structural locations) or

varying the friction at the collision node by rotating the tangential plane. Therefore, the Kudlich-Slibar model remains the most practical choice for such analyses.

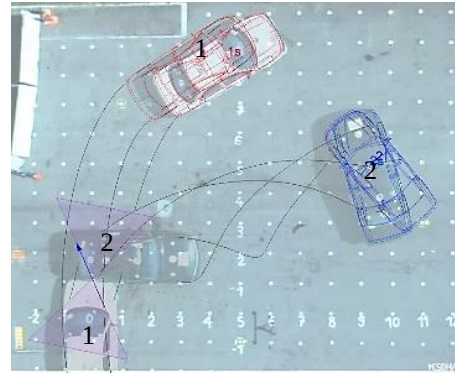


Fig. 7. PC-Crash manual calculations (matching post-simulation positions to real-world data)

4. Methodology – type I analytical reconstruction calculations

Due to the orthogonal directions of the vehicles' pre-impact velocity vectors, Type I analytical reconstruction calculations based on the principle of conservation of momentum were applied. This approach is schematically presented in Fig. 8.

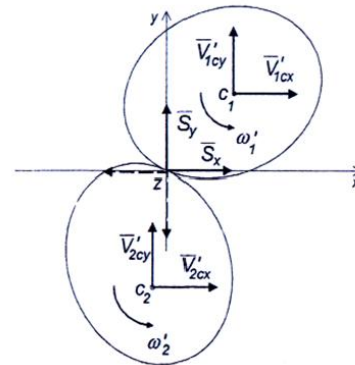


Fig. 8. Parameter designations for reconstruction calculations [14]

After appropriate transformation, the equations take the following form:

$$V_1 = \frac{m_1 \cdot V_1' \cdot \sin(\alpha_2 - \alpha_1') + m_2 \cdot V_2' \cdot \sin(\alpha_2 - \alpha_2')}{m_1 \cdot \sin(\alpha_2 - \alpha_1)} \quad (1)$$

$$V_2 = \frac{m_1 \cdot V_1' \cdot \sin(\alpha_1' - \alpha_1) + m_2 \cdot V_2' \cdot \sin(\alpha_2' - \alpha_1)}{m_2 \cdot \sin(\alpha_2 - \alpha_1)} \quad (2)$$

where: α – the pre-impact velocity vector angle (immediately before impact), α' – the post-impact velocity vector angle (immediately after impact).

Post-impact speeds were calculated using Marquard's energy-based relationships [16] for braking distance, incorporating empirical correction factors:

$$k_s = 0.17 \cdot k^3 - 0.488 \cdot k^2 - 0.03 \cdot k + 1 \quad (3)$$

$$k_\varphi = 0.328 \cdot k^3 - 0.772 \cdot k^2 + 1.072 \cdot k \quad (4)$$

$$k = \left| \frac{\Delta\varphi \cdot l}{2 \cdot s} \right| \quad (5)$$

where k in this relationship denotes the rotation factor rather than the coefficient of restitution, l – vehicle length.

Since the post-impact motion angles were well documented in the crash test, the calculations had low uncertainty. However, it was observed that even minor variations in the departure angles had a drastic impact on the results. In contrast, real-world traffic incidents are typically not as well documented as they are in video footage.

5. Methodology – optimization-based simulation calculations in MATLAB

The Kudlich-Slibar collision model [14] was implemented within the proprietary optimization script, with its fundamental relationships defined as follows:

$$m_1 \cdot (V'_{1ct} - V_{1ct}) = S_t \quad (6)$$

$$m_1 \cdot (V'_{1cn} - V_{1cn}) = S_n \quad (7)$$

$$m_2 \cdot (V'_{2ct} - V_{2ct}) = -S_t \quad (8)$$

$$m_2 \cdot (V'_{2cn} - V_{2cn}) = -S_n \quad (9)$$

$$I_{1c} \cdot (\omega'_1 - \omega_1) = S_t \cdot n_{1c} - S_n \cdot t_{1c} \quad (10)$$

$$I_{2c} \cdot (\omega'_2 - \omega_2) = S_n \cdot t_{2c} - S_t \cdot n_{2c} \quad (11)$$

The quantities $n_{1c}, t_{1c}, n_{2c}, t_{2c}$ (impulse moment arms – see Fig. 2 and Fig. 9) represent the geometric coordinates of the nominal contact point Z relative to the center of mass of each body.

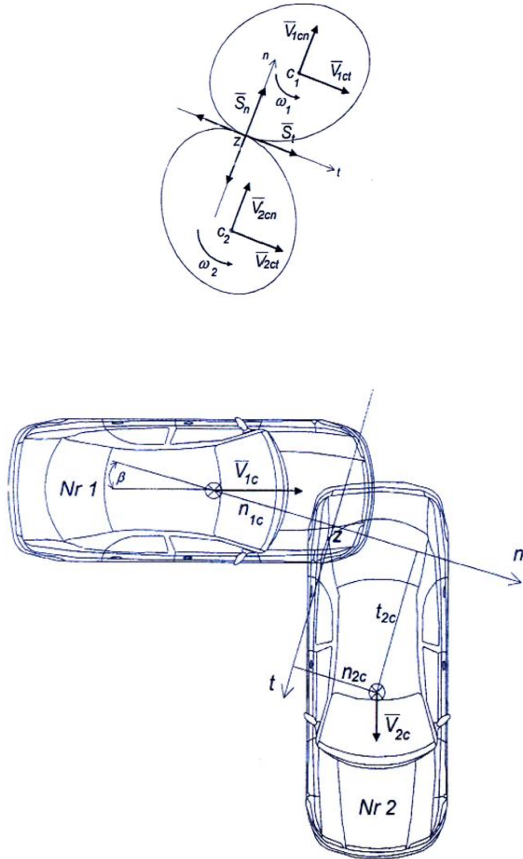


Fig. 9. Parameter designations for simulation-based calculations [14]

The post-impact motion was modeled according to Marquard's method, with the braking distance relationships transformed from a reconstruction (backward) to a simulation (forward) form. Consequently, the input data consisted of the post-impact speeds derived from the collision simulations, along with the known distances and angles of the post-impact motion. The calculation criterion (convergence condition) was satisfied when the speeds at the end of the post-impact phase reached values near zero, indicating that the vehicles' kinetic energy had been sufficiently dissipated. The objective (optimized) values were the pre-impact speeds, which were sampled using a uniform distribution via the Monte Carlo method. The proprietary script continued to iterate until a predefined number of speed pairs satisfying the velocity reduction condition in the post-impact motion were collected. This process can be described by the following objective function, which serves as the equivalent of Marquard's relationship:

$$f(V_k) = \sqrt{V'^2 - 2 \cdot k_s \cdot \mu \cdot g \cdot s} \quad (12)$$

$$0 < V_k < 0.5 \left[\frac{m}{s} \right] \quad (13)$$

The proprietary algorithm also incorporates the capability to randomize (search for) the friction coefficient (μ), reflecting the utilization of braking forces. This approach yielded a three-dimensional relationship between the pre-impact speeds of both vehicles and their post-impact decelerations, as shown in Fig. 10.

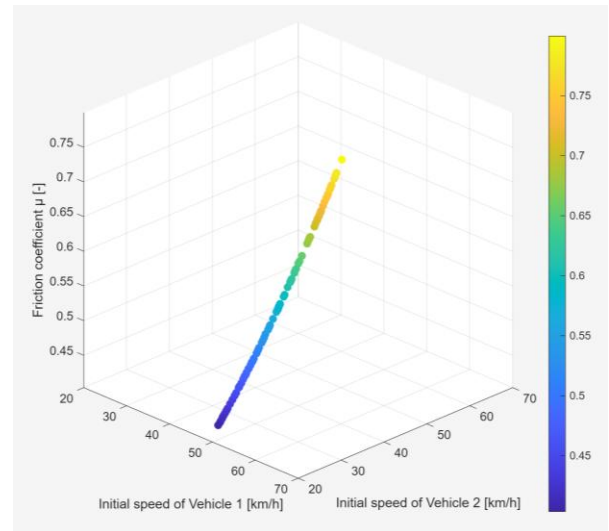


Fig. 10. 3D view of initial vehicle speeds as a function of the friction coefficient (μ) within the range of 0.4 to 0.8

The interdependencies between the calculated impact speeds are presented in Fig. 11.

The relationship, indicating that higher post-impact motion resistance coefficients lead to higher calculated pre-impact speeds, demonstrates the forensic importance of examining the full uncertainty range within physically realistic limits. Post-impact motion can be highly complex; furthermore, it is impossible to precisely reconstruct the actual resistance coefficient at the moment of the incident for the specific vehicles involved. In most cases, further

empirical research is no longer possible, as the vehicles have been destroyed and a significant amount of time has elapsed since the event [12].

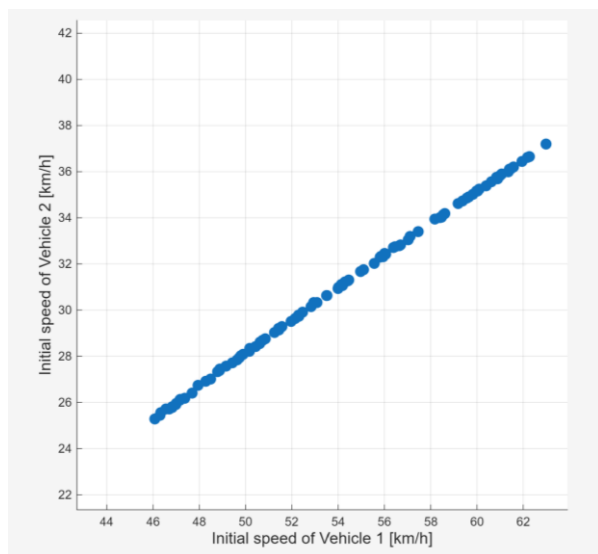


Fig. 11. 2D view of initial vehicle speeds as a function of the friction coefficient (μ) within the range of 0.4 to 0.8

6. Summary of results

During the calculations performed using all methods, it was observed that the speed of vehicle No. 1 (Ford Mondeo) was significantly higher than the value measured during the crash test. However, there is no evidence to suggest that the test measurement was erroneous; therefore, this discrepancy must be attributed to modeling uncertainty and the assumed coefficients – for instance, the tire-road friction utilization, even though the values selected were as low as possible while remaining physically realistic.

The chart in Fig. 12 presents pre-impact speed values calculated using various methods. The horizontal lines represent the mean values; however, calculations from the proprietary MATLAB script were excluded from the overall mean. This is because those results themselves represent the average values derived from 100 solutions that satisfied the criteria. This approach was taken to avoid the "mean of means" bias and to allow for a direct comparison between the averages of individual methods and the mean values generated by the proprietary script. The final two bars of the chart refer to the additional optimization of the friction coefficient (μ), corresponding to the plots shown in Fig. 10 and Fig. 11. The larger deviation of the mean in the case of μ -randomization results from a significant expansion of the speed range satisfying the criteria. Although the μ values were sampled from a range of 0.4 to 0.8, it appears that in this specific case, values closer to the lower bound were more appropriate. Nonetheless, in a real-world traffic incident – lacking a precise video recording – it would be impossible to narrow this range, making all resulting values equally probable.

As mentioned in the introduction, the authors were unable to identify any studies that compare commonly used forensic methods across a single specific crash test or a group of tests. While such comparisons are likely performed during the preparation of expert opinions and foren-

sic reports – as verifying results with a second method is considered best practice (though rarely using all available methods) – these findings are extremely seldom published. An additional complication is the frequent lack of a 'ground truth' reference for actual speed; even when Event Data Recorder (EDR) records are available, they must be verified using independent methods based on other physical evidence. Consequently, in forensic reports, uncertainty is estimated while the absolute true value remains unknown. For instance, if a vehicle is skidding sideways or has locked wheels, the recorded EDR speed may not reflect the actual velocity. The same applies to acceleration with loss of traction.

Since the methods utilized in this study remain standard in forensic accident reconstruction, it is essential to continue research on various crash tests to obtain a broader understanding of uncertainty across different collision types. Furthermore, it is critical to acknowledge that the calculated impact speed is mathematically dependent on post-impact movement parameters, which are typically burdened by high partial uncertainty. Moreover, the pre-impact resistance is often unknown, creating additional difficulties in establishing the initial speed at the onset of danger – the parameter of greatest interest to judicial authorities.

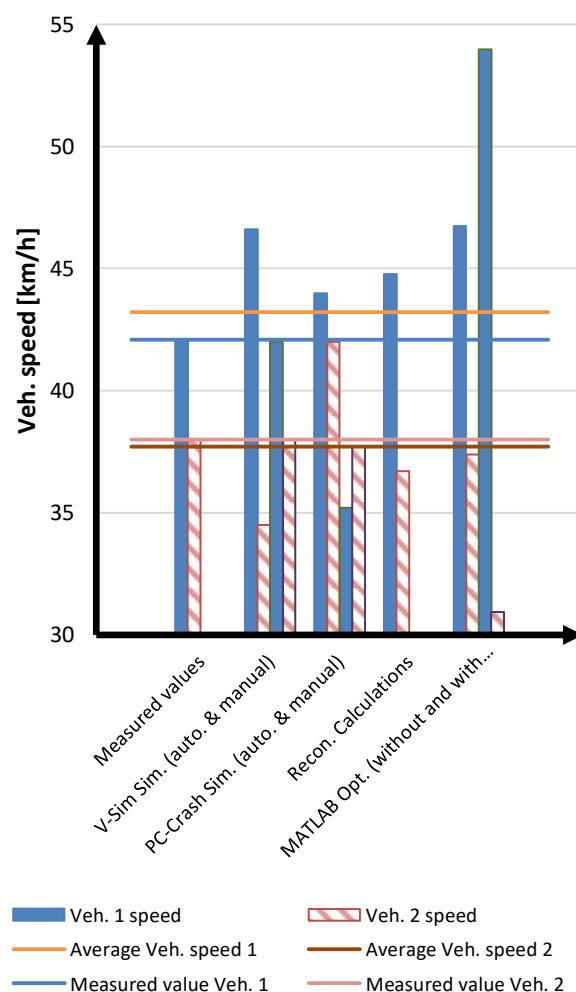


Fig. 12. Impact speeds calculated using various methodologies

In general, for this specific test at measured speeds of 42.1 and 30.0 km/h, the uncertainties for the various methods did not exceed $\pm 13\%$. In contrast, randomizing the friction coefficient (μ) resulted in a maximum uncertainty of $\pm 28\%$, with results varying across vehicles. These values fall within the range of theoretical analyses by other researchers [2], who obtained the following results: limit values $\pm 30.5\%$, total derivative $\pm 31.3\%$, second-order total derivative $\pm 33.0\%$, finite differences $\pm 31.3\%$, Gaussian probabilistic method $\pm 19.0\%$, stochastic processes $\pm 18.8\%$, and Monte-Carlo simulation $\pm 29.5\%$.

7. Conclusions

The computational analysis, conducted using available methodologies and further enriched by calculations performed with a proprietary MATLAB script, has led to the following conclusions.

1. Objective functions in simulation software allow for moderately effective matching of post-impact vehicle positions; however, they provide the capability to randomize input parameters, thus enabling uncertainty analysis in a qualitative sense. The objective function in PC-Crash lacks physical significance in terms of energy balance; consequently, the percentage error reported by the software does not correlate with the actual speed calculation error, although it may be misinterpreted as such. In contrast, V-Sim does not report a percentage error; instead, it provides only the number of results that satisfy the defined constraints. This approach does not mislead the user regarding the actual uncertainty of the results.
2. The collision simulation designed in MATLAB, based on Kudlich-Slibar relationships and Marquard-based post-impact motion, enables rapid assessment of the solution's consistency with observed collision effects and the investigation of uncertainty ranges for various pa-

rameters. However, it lacks visualization capabilities and does not account for post-impact trajectories *sensu stricto*, which in certain cases represents a significant limitation.

3. A comparison of the results obtained through various methodologies—including the analytical reconstruction model, simulation-based calculations in V-Sim and PC-Crash, and the proprietary MATLAB script – demonstrated their mutual consistency. From the perspective of road accident reconstruction, the results showed sufficient alignment with real-world crash test data, thereby validating their practical applicability.
4. Analytical reconstruction calculations demonstrated that even minor deviations in the input data (a few degrees in the approach and departure angles) can drastically affect the resulting speeds (by 100% or more), confirming a well-known shortcoming of this methodology. The computational uncertainty determined in this study remained low only because the precise pre- and post-impact vehicle motion was known – specifically, the trajectory angles immediately following the collision
5. While simulation-based calculations may exhibit sensitivity in certain cases, they are generally significantly less sensitive to uncertainties regarding the aforementioned angles than analytical methods
6. Generally, for this specific test, the uncertainties of the various methods did not exceed a maximum of $\pm 13\%$, whereas the randomization of the friction coefficient (μ) resulted in a maximum uncertainty of $\pm 28\%$, with results varying for each vehicle. These uncertainty levels were consistent with values reported by other authors who have investigated the problem from a theoretical perspective. For other collision configurations and conditions, these values may differ.

Nomenclature

E	energy	m	mass
EES	equivalent energy speed	s	distance
EBS	equivalent barrier speed	tg μ	friction coefficient at the collision node
k	coefficient of restitution, or Marquardt's rotation coefficient	μ	coefficient of friction of the wheels on the road

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Analysis of energy consumption and recovery by the electric motor and generator of a HEV hybrid vehicle

ARTICLE INFO

This article presents an analysis of the operation of the electric motor, generator, and internal combustion engine within the hybrid powertrain system of a Toyota Corolla. To compare energy utilization data from these sources, a specially designed diagnostic scanner was used in our laboratory to capture data from the vehicle's Controller Area Network (CAN).

The tests were conducted on a laboratory chassis dynamometer, taking into account the NEDC (New European Driving Cycle) and WLTP (Worldwide Harmonised Light Duty Vehicle Test Procedure) protocols. Additionally, experiments were performed under real urban traffic conditions. The analysis of the collected data provided detailed insights into the operation of the aforementioned components, which are integral to the powertrain during characteristic driving scenarios. For selected test fragments, an analysis of energy consumption and recovery by the electric motor and generator is presented.

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1. Introduction

Modern hybrid powertrains are complex; optimal energy-flow management is therefore essential for high overall efficiency [1, 4, 6, 13, 14, 21]. There are wide varieties of drive systems, differing in design, operating strategy and the method of power distribution between combustion engines and electric motors [2, 8, 10, 16]. Car manufacturers use their own energy management strategies tailored to their needs, which makes it difficult to assess the efficiency of individual solutions [3, 5, 9, 11, 12, 15, 19]. In the Toyota Corolla 1.8 Hybrid equipped with a 5th generation hybrid system, a drive system consisting of an internal combustion engine and two electric motor-generators was used [24]. The single planetary gear set sums the torques of the individual sources.

The drive system provides stepless gear change (E-CVT), enabling the vehicle to reach a maximum speed of 180 km/h [20, 24].

This article attempts to present the performance of individual drive sources, including a comparison of the behaviours of MG1/MG2 electric machines and the ICE engine under various operating conditions: during standardized tests and in real-world traffic conditions. The analysis was based on selected test fragments: at start, during acceleration, at constant speed, and during deceleration, i.e., under variable load conditions and for different battery charge states.

In order to analyse data on the use of energy from the above-mentioned sources, a specially designed diagnostic scanner was used to capture data from the vehicle's CAN network [17, 18, 23]. The tests of the vehicle's drive system operation were performed on a chassis dynamometer during WLTP and NEDC tests and in real city traffic. The operation of individual electric motors and engines in different

phases of vehicle movement is presented in the following section.

2. Tested vehicle drive system

The object of the research and analysis was a Toyota Corolla 1.8 Hybrid equipped with a 5th generation hybrid drive system. The drive system uses Hybrid Synergy Drive® technology. This is a system that connects two three-phase synchronous AC electric machines (MG – motor generator) with an internal combustion engine (ICE). The drive system parameters of the tested vehicle are presented in Table 1.

The summation of the energy of individual drive sources is carried out by the E-CVT gearbox, using a single planetary gear for this purpose [24].

The combustion engine operating according to the Atkinson cycle is connected to the planetary gear satellite carrier. The MG1 electric motor (reversible machine - motor or generator), working mainly as a generator and acting as a starter for the combustion engine, is connected to the sun gear of the planetary gear. The MG2 electric motor (reversible machine - motor or generator) is connected via a cylindrical gear to the ring wheel of the planetary gear, from which the torque is directed to the final drive of the vehicle's drive system. The E-CVT system works by selecting the appropriate operating conditions of individual drive system components to meet the current energy demand.

Differentiation of the speed at the input of the vehicle's final drive is achieved by comparing the speeds of the MG2 and MG1 electric motors when the combustion engine is not operating, or of the MG2 and the combustion engine when the latter is used to drive the vehicle. At this time, the MG1 engine works as a generator, supplying the MG2 electric motor with electricity and charging the traction battery. At the same time, by loading the combustion engine, it increases its efficiency. This allows for approaching

the optimal operating range of the combustion engine. Figure 1 shows the kinematic diagram of the drive system of the tested Toyota Corolla 5th generation vehicle.

Table 1. Drive system parameters

Internal combustion engine	
Number and arrangement of cylinders	4, in-line, Atkinson cycle
Timing system	4 valves per cylinder, DOHC with VVT-i
Fuel supply system	Direct fuel injection
Displacement	1798 cm ³
Piston diameter x stroke	80.5 × 88.3 mm
Compression ratio	13
Maximum power	72 kW at 5200 rpm
Maximum torque	142 Nm at 3600 rpm
Electric motor MG2	
Maximum voltage	600 V
Maximum power	70 kW at 13500 rpm
Maximum torque	185 Nm
Electric motor MG1	
Maximum voltage	600 V
Maximum power	23 kW
Maximum torque	40 Nm
Battery	
Type	Lithium-ion
Nominal voltage	207.2 V
Capacity (3HR)	0.85 kWh

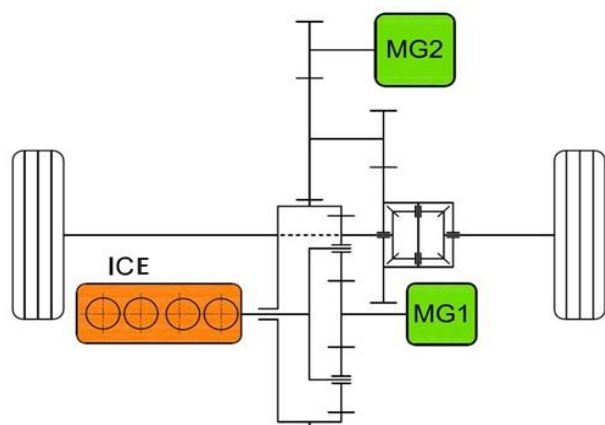


Fig. 1. Kinematic diagram of the HEV Toyota Corolla 5th generation drive system

This system can operate in various modes that automatically adjust based on load and driving dynamics. The hybrid drive system of the Toyota Corolla Hybrid can operate in series and parallel configurations. It can also operate solely in electric mode or use the internal combustion engine and electric motors simultaneously. The vehicle's drive system is a self-charging system. The battery is charged both during regenerative braking and during internal combustion engine operation.

3. Different operating modes of the hybrid drive system

When stationary, both the combustion engine and the electric motor are automatically switched off to reduce fuel consumption and exhaust emissions. In this phase, the system operates in standby mode, monitoring the battery charge level and readiness to restart the drive. This is possible thanks to the Start-Stop system, which minimizes ener-

gy losses by switching off the combustion engine when it is not needed.

When starting from a standstill, the hybrid drive system operates in electric mode (EV), which uses only the energy stored in the traction battery for propulsion. The electric motor, due to its ability to generate maximum torque from the moment of start-up, is suitable for initiating movement and driving at low speeds. At this point, the combustion engine is not running, and the vehicle is not emitting exhaust fumes. When accelerating or reaching higher speeds, the system automatically starts the combustion engine and uses the MG1 electric motor to provide additional power, ensuring greater driving dynamics. The combustion engine also starts up when the battery charge level drops below a certain level, and then charges the battery. During low-speed manoeuvres, such as parking, the drive system uses only the electric motor. Such operations require little power and are short-lived, so an electric drive is sufficient. On the other hand, when driving at a constant speed with moderate loads, the hybrid system works in the most energy-efficient way. At low speeds, the main source of power is the electric motor, while the combustion engine operates in support mode, switching on periodically to maintain optimal battery energy levels. At higher speeds, the combustion engine takes over the dominant role, but is still supported by the electric unit, which allows for optimisation of fuel consumption. During full acceleration, for example, when overtaking, the hybrid drive system engages both power sources to provide the maximum available energy. The combustion engine operates in its optimal speed range, while the electric motor provides additional torque, enabling the desired speed to be reached quickly. This is the operating phase in which fuel consumption and exhaust emissions are at their highest. During long-term high-speed driving, the main source of power is the combustion engine, which provides adequate power to maintain high speeds on expressways. The electric motor, in such conditions, plays a supporting role but does not directly drive the vehicle – it supports energy recuperation and provides assistance when a momentary increase in power is needed.

When decelerating, the hybrid drive system recovers energy. The MG2 motor, operating in generator mode, converts the vehicle's kinetic energy into electrical energy, which is stored in the traction battery. During this process, the combustion engine is switched off, and the drive system switches to charging mode. Anticipating driving situations, such as gradual deceleration, maximizes the efficiency of the recuperation process.

During braking, when deceleration is more intense, the system actively recovers energy, converting part of the vehicle's kinetic energy into electrical energy stored in the traction battery.

In situations where braking is long (downhill), it is possible to use mode B, which increases braking force by using the combustion engine's resistance. In this case, the vehicle's drive wheels drive the MG2 motor, operating in generator mode, to charge the traction battery and supply electrical power to the MG1 motor, which, while maintaining the combustion engine's rotational speed, uses its braking torque. It allows for speed control and reduced brake wear.

The smoother and more gradual the braking, the more effective the recuperation process is [24].

4. Vehicle tests during WLTP, NEDC and city traffic tests

4.1. Measuring equipment

The energy flows were determined from data recorded during chassis dynamometer tests and read from the vehicle's CAN network. The diagnostic connector used in OBD (On-Board Diagnostics) systems was used to read data from the CAN (Controller Area Network). For this purpose, a scanner developed at the Department of Combustion Engines and Vehicles at the University of Bielsko-Biala was used, allowing for the reading of 36 selected vehicle operating parameters from the CAN network. The scanner consists of a motherboard with a microcontroller, a network module, a colour touchscreen panel, and DC/DC converters assembled in a single housing. During chassis dynamometer and road-vehicle tests, data from the scanner were recorded at 5 Hz. In both cases, the operating parameters of the drive system components were saved to a memory card. [17, 22]. Figure 2 shows a diagram of the measurement system used in chassis dynamometer tests.

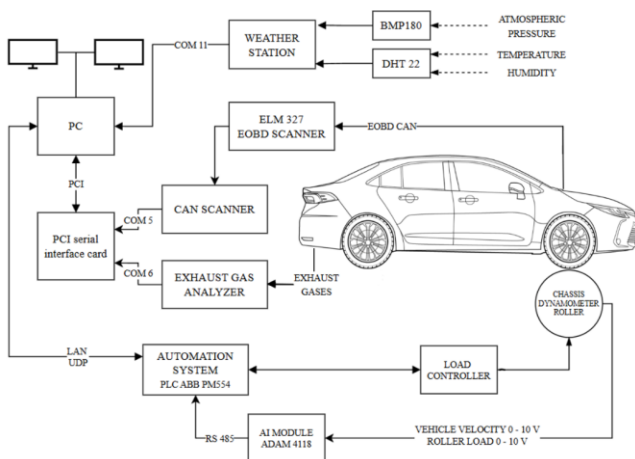


Fig. 2. Diagram of the measurement system used in the energy consumption tests of the Toyota Corolla Hybrid

Due to the significant amount of data to record, only those data ranges were selected that allowed for the assessment of the energy flow in the vehicle during its collection from various sources, i.e., the combustion engine, the traction battery or energy generated during vehicle movement. Additionally, a number of data points were collected to support diagnostics of the drive system of the tested vehicle. This CAN scanner is shown in Fig. 3.

The device allows you to observe the values of 36 selected parameters in real time on the scanner screen. Selected data read from the CAN network were used for the analyses included in the article:

- vehicle speed
- accelerator pedal position
- throttle opening angle
- brake pedal position
- wheel speeds
- engine speed

- electric motors MG1 and MG2 speeds
- engine torque
- electric motors MG1 and MG2 torques
- coolant temperature
- battery charge level (SOC)
- battery voltage
- current intensity of MG1 and MG2 motors
- voltage of MG1 and MG2 motors
- hybrid battery current intensity
- instantaneous fuel consumption.



Fig. 3. ADEL CAN BUS diagnostic scanner

4.2. Vehicle tests

The tests of energy consumption and generation for the Toyota Corolla Hybrid 5th generation were conducted during WLTP and NEDC tests and in real-world city traffic conditions. The tests were performed on a chassis dynamometer at the Department of Combustion Engines and Vehicles at the University of Bielsko-Biala. For technical reasons, the maximum speed during the dynamometer tests was limited to 100 km/h. The real-world drive took place during rush hour in Bielsko-Biala and on a section of the S1 national road. The selected route included driving through streets with numerous intersections controlled by traffic lights, and the final part allowed higher speeds. The selected route traverses mountainous terrain, requiring the need to overcome larger hills and descents.

The tests on the chassis dynamometer and during city driving allowed for the assessment of various driving conditions and thus various operating modes of the hybrid drive system. Figure 4 shows the total energy consumption in the NEDC, WLTP and city traffic tests, with a breakdown of energy drawn from the traction battery, generated by the combustion engine and supplied by the MG1 and MG2 electric motors.

The use of the vehicle's traction battery energy was also analysed. Assessment of the hybrid drive system required selecting fragments from the test results that illustrated the individual operating modes. In the tests carried out, the combustion engine showed the highest energy consumption, respectively: ~5100 kJ in the NEDC test, ~1800 kJ in the WLTP test, and ~5000 kJ in city driving. The MG1 electric motor, which primarily served as a starter for the combustion engine, consumed only a small amount of electricity. The MG2 electric motor used more electricity at start-up, with lower motion resistance and lower driving speeds, whereas the combustion engine required more energy at higher motion resistance and higher driving speeds.

In the individual tests, this consumption was: NEDC ~1800 kJ, WLTP ~3100 and ~2900 kJ in the city driving test.

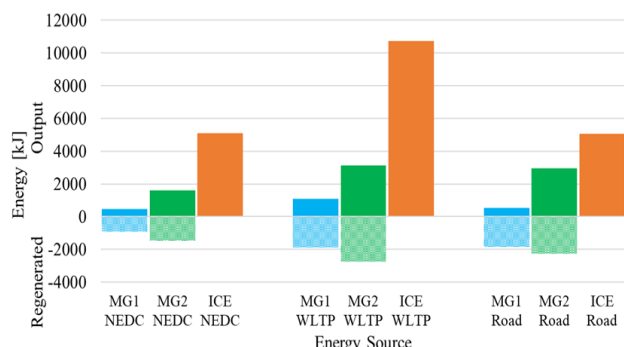


Fig. 4. Total energy consumption during NEDC, WLTP and city driving tests

During vehicle deceleration and braking, the MG2 electric motor, operating in generator mode, recovered part of the vehicle's kinetic energy by converting it to electrical energy. The energy recovered by the MG2 motor in individual tests is similar to the energy required to drive the vehicle. It should be noted that a relatively large amount of electrical energy was generated by the MG1 motor, driven by excess combustion engine power in generator mode.

5. Analysis of energy consumption and recovery during various aspects of vehicle movement

5.1. Starting from a standstill

From the obtained waveforms, fragments were selected to illustrate the hybrid drive system's individual operating modes. The first one concerns starting from a standstill at low speed.

Figure 5 shows the vehicle starting from a standstill with different battery charge levels.

In the case shown in Fig. 5a, the battery of the tested vehicle was charged to 68%. In this case, the vehicle starts moving solely with the MG2 electric traction motor, drawing energy from the battery. At higher speeds (about 35 km/h), the combustion engine powers the vehicle. The MG2 electric motor, to a lesser extent, periodically supports the internal combustion engine.

In turn, the MG1 generator uses the surplus energy of the combustion engine and works in generator mode, charging the battery. Figure 5b shows a case in which the battery was partially discharged (charge level ~30%), leading the combustion engine to serve as the main source of vehicle drive even at low speeds (10 km/h). In this case, the MG1 electric motor, driven by the combustion engine, also generates electricity. The share of the MG2 electric motor in the vehicle drive is insignificant.

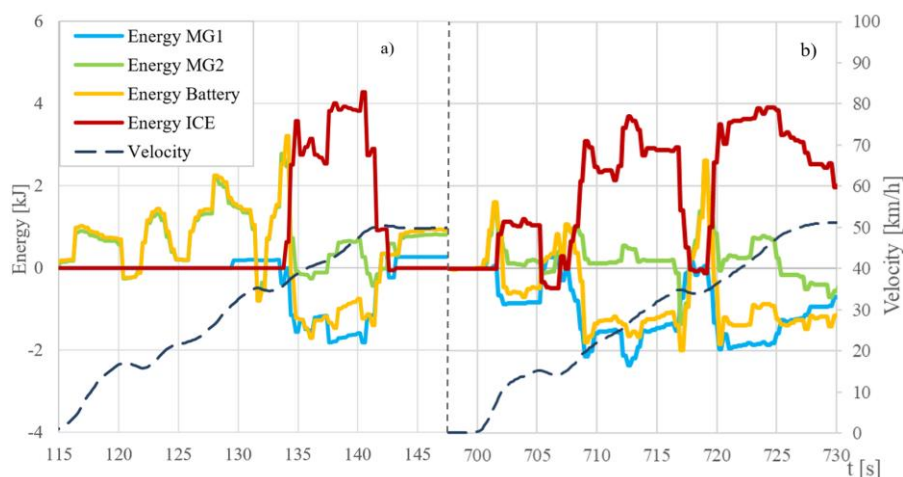


Fig. 5. Accelerating the vehicle at low driving speeds in the NEDC test: a) with a charged battery, b) with a discharged battery

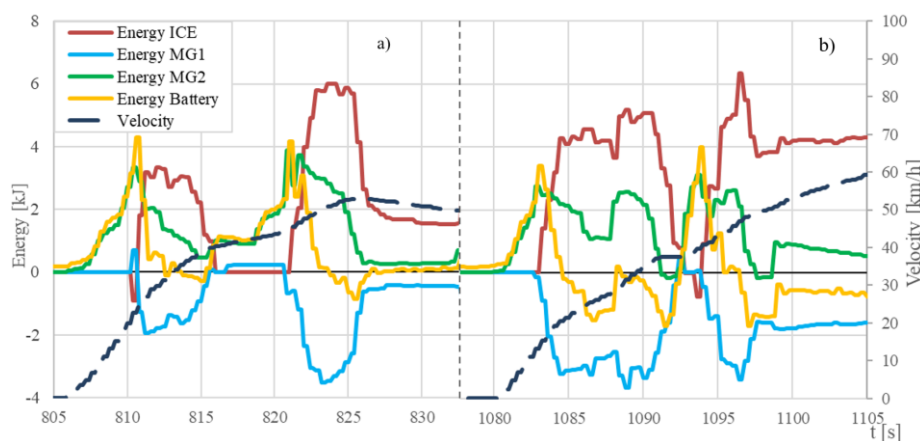


Fig. 6. Vehicle acceleration in a road test: a) with low intensity, b) with high intensity

5.2. Acceleration

Figure 6 shows the vehicle's acceleration at different intensities. In the case shown in Fig. 6a, the vehicle begins to accelerate, initially using only the MG2 electric motor powered by the energy drawn from the traction battery. After the vehicle reaches a driving speed of ~ 25 km/h, the MG1 electric motor starts the combustion engine. From that moment on, the combustion engine is the main source of drive, which generates more than twice as much energy as the MG2 electric motor. Additionally, the combustion engine is "loaded" by the MG1 motor, which generates electricity used to power the MG2 electric motor. A temporary reduction in the acceleration intensity causes the combustion engine to turn off, and the MG2 motor is responsible for the vehicle drive using the energy drawn from the battery. Figure 6b shows a case in which the vehicle accelerates with high intensity. Initially, the vehicle also moves using the MG2 electric motor at the expense of the battery's energy, but in this case, the combustion engine starts operating sooner, at approximately 15 km/h. Due to the high resistance to motion caused by the vehicle's intensive acceleration, the MG2 electric motor further supports the combustion engine by using energy generated by the MG1 motor, which is powered by the combustion engine.

5.3. Driving at a constant speed

Figure 7 shows the energy flow at fixed speeds of 70 km/h and 100 km/h. In the case shown in Fig. 7a, the vehicle moves at a lower speed, at which the resistance to motion is lower. In this case, only the combustion engine drives the vehicle, generating energy at a level of ~ 2.8 – ~ 3.2 kJ. The MG2 machine operates in generator mode at this stage, recuperating small amounts of electrical energy.

The MG2 machine operates in generator mode at this stage, recuperating small amounts of electrical energy, on average at a level of ~ 0.3 kJ. Figure 7b shows a fragment of the test in which the vehicle moves at ~ 100 km/h. In this case, the combustion engine also serves as the main drive source, generating 4 kJ of energy. The MG1 generator uses power from the combustion engine to generate electricity. A small portion is used by the MG2 motor, and the remainder is directed to the traction battery. In the first period of this part of the test, the MG2 motor consumes little electricity and contributes little to supporting the combustion engine. In the final part of the test, the combustion engine reduces the power required to drive the vehicle; this reduction is supplemented by the MG2 electric motor using energy generated by the MG1 electric machine.

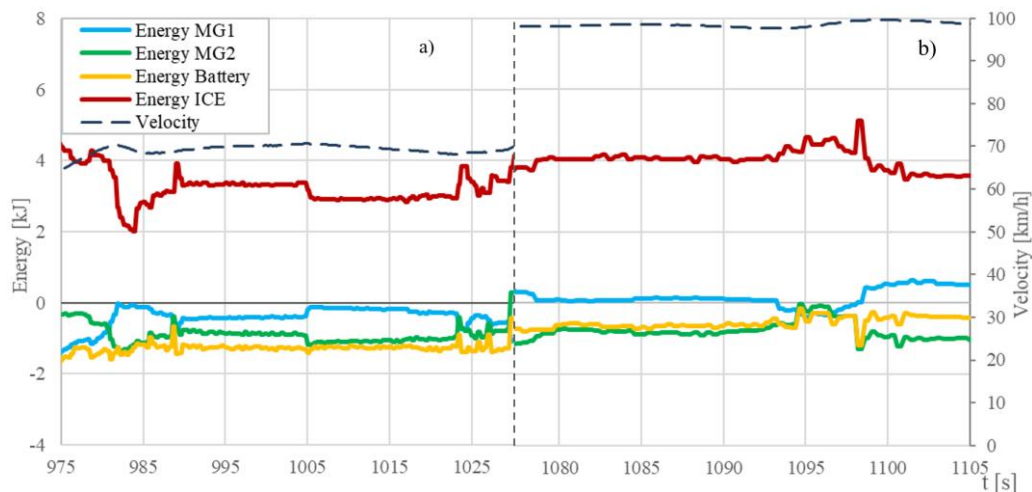


Fig. 7. Driving at a constant speed in the NEDC test: a) driving speed ~ 70 km/h, b) driving speed ~ 100 km/h

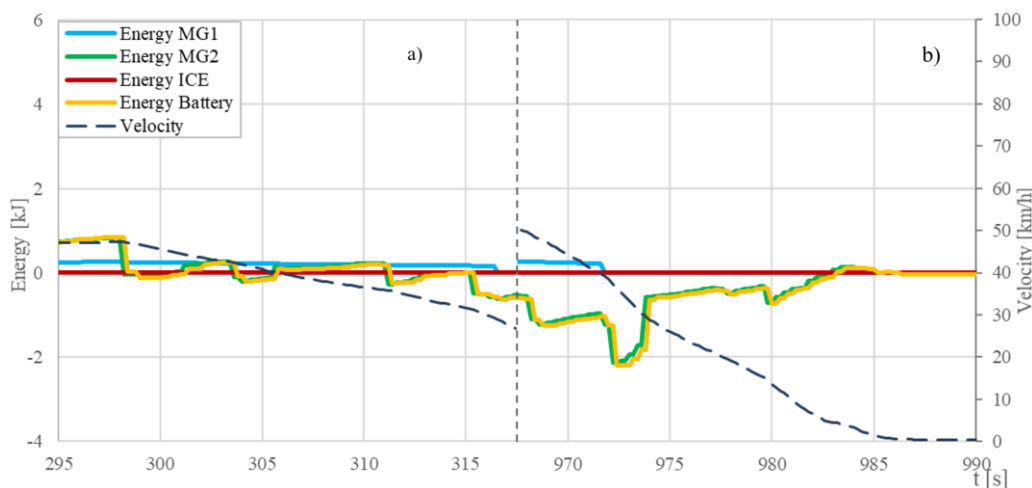


Fig. 8. Deceleration of the vehicle in the WLTP test: a) low intensity, b) high intensity

5.4 Deceleration

Figure 8 shows the energy flow during deceleration. In this phase of the test, shown in Fig. 8a, the vehicle decelerates relatively slowly. In this case, the combustion engine is switched off, and the MG2 motor periodically operates in the drive engine mode, drawing small amounts of energy from the traction battery, then switching to the generator mode to brake the vehicle and recover some of the vehicle's kinetic energy. The MG1 draws negligible electrical power throughout deceleration. Figure 8b shows a phase of the test in which the vehicle decelerates more intensely. In this case, the MG2 motor operates in the generator mode until the vehicle stops. Depending on the braking intensity, it generates varying amounts of electrical energy, which is stored in the traction battery.

6. Summary and conclusions

The study supports the following conclusions:

1. The NEDC and WLTP tests, as well as city driving, were selected for the analyses. The NEDC test includes many ranges with fixed conditions for acceleration, constant speed and vehicle deceleration. The WLTP test is much more dynamic, allowing analysis under rapidly changing vehicle traffic conditions. City driving allows for verification of the obtained results in real conditions.
2. A hybrid drive system based on two electric machines and a combustion engine with a planetary gear designed to sum up the power generated by the system and transfer it to

the main gear and wheels, allows for such a selection of the drive source to optimise fuel consumption and, at the same time, recover the greatest amount of energy during deceleration and braking. This has been presented in graphs in individual phases of vehicle movement.

3. During driving, the MG2 motor-generator draws and returns similar amounts of energy to the batteries, regardless of the conditions in which the vehicle is moving, regardless of whether it is a NEDC or WLTP test, or city driving. Driving at high speeds increases the combustion engine's contribution; at higher vehicle speeds, it supplies a correspondingly larger share of the total energy.

4. During driving, the MG1 motor-generator is used to obtain the required driving speed and during the combustion engine operation to generate energy to power the MG2 engine and batteries. During the tests, the amount of generated energy is significantly greater than the energy drawn.

5. During the tests, the battery charge level was monitored by the controller and after the charge level dropped below 30%, the combustion engine was used to recharge them.

6. For high-dynamic acceleration, the combustion engine was mainly used, supported by the MG2 engine in the initial acceleration phase.

7. During intensive braking, the MG2 engine was used to recover energy stored in the battery.

Nomenclature

CAN	controller area network	MG2	motor-generator 2
e-CVT	electronically-controlled continuously variable transmission	NEDC	New European Driving Cycle
EV	electric vehicle	OBD	on-board diagnostics
ICE	internal combustion engine	SOC	state of charge
MG1	motor-generator 1	WLTP	Worldwide Harmonized Light Vehicles Test Procedure

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Tribological effects of engine oil type on diesel engine efficiency and pollutant formation

ARTICLE INFO

The goal of this research is to determine how different types of engine oil affect how well a four-stroke, single-cylinder, air-cooled, direct-injection diesel engine performs and how much pollution it produces. At 1000, 2000, and 2500 rpm, the engine was linked to a hydraulic dynamometer and driven with a continuous load of 4.0 N·m. Local Baghdad Oil (B1), Gulf Oil (G1), and PRO-TEC Oil (P1) – three lubricating oils – were assessed. Using an AIRREX HG-540 gas analyser, engine performance and emissions were assessed based on brake-specific fuel consumption (BSFC), exhaust gas temperature, and nitrogen oxides (NO_x), carbon monoxide (CO), and carbon dioxide (CO₂) emissions. For all oils, results show that BSFC dropped as engine speed rose. PRO-TEC Oil (P1) had the lowest BSFC of 0.290 kg/kWh at 2500 rpm; Baghdad Oil (B1) had the highest value of 0.351 kg/kWh at 1000 rpm. Exhaust gas temperature increased with speed, reaching a maximum of 243°C for B1 at 2500 rpm; P1 consistently had the lowest exhaust gas temperature, with a minimum of 188°C. As engine speed rose, CO emissions rose; P1 had the highest value at 1000 rpm (0.018%), while B1 had the lowest (0.013%). Rising speed also increased CO₂ and NO_x emissions; B1 had the highest NO_x (300.1 ppm) and CO₂ (2.451%), whereas P1 consistently showed the lowest emissions. The outcomes highlight the advantages of oils with improved viscosity and thermal stability for enhanced performance and decreased environmental impact by showing that engine oil qualities greatly influence diesel engine efficiency and emissions.

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1. Introduction

Internal combustion engines are complex systems in which a significant fraction of the energy produced is consumed within, primarily due to friction and the heat generated between moving parts [9, 30]. Over time, this friction causes wear and deterioration of engine components and raises energy losses [15]. Overcoming friction – which always opposes motion – takes more energy, which eventually turns into heat. This makes wear worse and makes the engine less efficient overall. Lubrication is therefore very important in all engines, as it creates a thin oil film between moving metal surfaces, thereby decreasing direct contact and lowering frictional losses. Given the numerous moving components and frictional surfaces inside an engine, lubrication oil is essential to ensure long life and trouble-free operation. In diesel engines, efficiency depends on how well they are lubricated. Increased friction affects the compression ratio, which can affect engine efficiency. Furthermore, at higher engine speeds, friction forces increase; hence, the importance of proper lubrication to preserve mechanical efficiency and avoid excessive wear. Improved lubrication also helps to improve gas mileage since it lowers frictional losses [4, 5].

Generally speaking, oil viscosity is among the most crucial characteristics that greatly affect its lubricity; usually, high viscosity reduces frictional losses [7, 10]. Oil viscosity, however, is quite susceptible to temperature changes; it drops as temperature rises. The lubricating film may be broken down by high temperatures, which accelerates wear and increases metal-to-metal contact. Though minor increases in oil temperature lower viscosity and friction [18].

The demands on lubricants likewise vary depending on engine load. Increasing load increases cylinder pressures

and friction losses; this is partially offset by the related drop in oil viscosity due to high temperatures [14, 16]. Engine lubrication primarily reduces friction, removes heat, removes contaminants such as wear particles and carbon, and keeps critical parts like piston rings and seats clean. Lubricating oil also reduces engine noise, cushions shocks, and keeps moving parts from touching each other, which helps the engine last longer [3, 4].

Given the high operating temperatures of engines, lubricating oils have to be somewhat thick, stable, and thermally resistant [20]. Heat and mechanical stress degrade oil quality over time; therefore, regular oil changes are essential to keep the engine running in perfect condition [24, 25]. Another key metric is the viscosity index, which shows how temperature affects oil viscosity. A vital attribute that stops engine parts from rusting at high temperatures is oxidation resistance [26].

Although mineral oils perform reasonably well under normal conditions, contemporary engines and challenging operating environments often require improved oil compositions. Detergents, anti-wear agents, anti-rust chemicals, and anti-foam agents are among the additives added to oil to improve its performance, protect engine parts, and keep the oil clean [21, 23]. Notably, studies show a close relationship between engine speed, load, and exhaust emissions, including nitrogen oxides (NO_x), where higher levels are observed at higher operating loads [18, 22]. For these reasons, this study looks at how three different engine oils, Baghdad Oil (B1), PRO-TEC Oil (P1), and Gulf Oil (G1), affect how well a single-cylinder, four-stroke diesel engine runs and how much pollution it puts out. While P1 and G1 are well-known companies that offer a range of additives, including anti-wear agents, oxidation and corrosion inhibi-

tors, detergents, and shear-stability improvers, B1 oil is locally produced with no additives. This study aims to clarify the relationships among oil type, engine performance, and environmental impact by closely examining factors such as brake-specific fuel consumption (BSFC), friction power, exhaust gas temperature, and key emissions (NO_x , CO, and CO_2), thereby guiding the selection of optimal lubricants for diesel engines.

2. Test procedure

This investigation looked at three different types of engine oils: B1, P1, and G1 for their impact on the performance and emissions of a single-cylinder, four-stroke, air-cooled, direct-injection diesel engine. The experimental procedure looked at three speed levels: 1000, 2000, and 2500 rpm. Throughout all the tests, a constant load of 4.0 N·m was maintained.

Before every test, the first step was to drain all remaining oil so the oil samples wouldn't get mixed. Ten minutes of engine running after filling with the preferred oil type, Baghdad Oil (B1), first helps achieve thermal and operational stability. The engine speed was adjusted to 1000 rpm under the fixed load once steady state was reached, and the system was allowed to run for another 10 minutes to ensure steady-state conditions. Each test recorded the following measurements.

Exhaust gas temperature was measured using the test panel display, which has a gas temperature sensor.

Fuel consumption was measured using a graduated fuel tube, with the consumption time tracked via the panel.

Engine power output was calculated from electronic platform readings.

Exhaust emissions by an AIRREX HG-540 exhaust gas analyser with the sampling probe put into the exhaust stream (CO , HC, CO_2 and NO_x) were determined.

The engine speed was raised to 2000 rpm after the 1000 rpm measurements were taken. Following a 10-minute stabilization period, the same set of readings was gathered. The approach was repeated at 2500 rpm.

After the tests with Baghdad Oil were finished, the engine oil was drained and replaced with PRO-TEC Oil (P1). Once more, run for 10 minutes to guarantee stability. The engine was tested at every speed using the full measurement protocol. This method was later applied to the third oil sample, Gulf Oil (G1), to ensure that every oil kind was tested under the same conditions. For subsequent study, all relevant data, including fuel consumption, exhaust gas temperature, engine speed, load, and emission concentrations, were carefully preserved. This rigorous experimental technique ensures reliable, replicable results. It allows a reliable test of the effects of several engine oils on diesel engine performance and emissions under regulated laboratory conditions.

3. Materials and methods

3.1. Test engine

The experiments were conducted using a British-manufactured, single-cylinder, four-stroke, air-injected diesel engine (see Fig. 1). The technical specifications of the engine are detailed in Table 1.



Fig. 1. Testing engine

Table 1. Technical specifications of the engine

Engine manufacturer	TQ company for testing and training
Type of engine	TD 111
Piston displacement	230 cm ³
Stroke	60 mm
Bore	70 mm
Nominal output	3.5 kW at 3600 rev/min
Maximum torque	10.5 Nm at 2200 rev/min

3.2. Test panel and instrumentation

Instruments for monitoring torque, exhaust gas temperature, engine speed, and fuel usage abound on the engine test panel. Over a given period, a graduated tube helped us calculate fuel usage. Using an AIRREX HG-540 exhaust gas analyzer (Korea), real-time CO , CO_2 , and NO_x emissions were measured in ppm. Table 2 lists the specifications of the analyzer; Fig. 2 shows it.

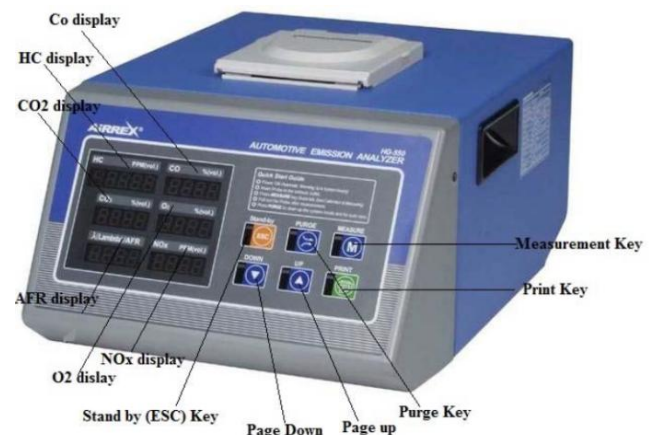


Fig. 2. Exhaust analysis device

Table 2. Technical specifications of the exhaust analyzer

HG-540	Specification			
Measuring range	CO ₂	0–200,000 ppm	CO	0.000–9.999%
Resolution		1 ppm		0.001%
Measuring range	NO _x	0–5000 ppm	AFR	0.5–3.0
Resolution		1 ppm		0.001%
Flow rate	L/min	Power supply	AC90-260 V 50/60 Hz	

3.3. Engine oils

Petroleum labs examined the three oils tested – Baghdad Oil, PRO-TEC Oil, and Gulf Oil – to verify their qualities before and after use; the findings are shown in Table 3.

Table 3. Types of engine oil and its analysis

Lab. Ins. Data	Babil Oil before use	Babil Oil after use	Gulf Oil before use	Gulf Oil after used	PRO-TEC Oil before use	PRO-TEC Oil after use
Vis.c.st @40°C	84.89	70.99	93.1	84.98	94.12	85.99
Vis.c.st @100°C	17.22	16.8	14.82	12.83	14.1	13.78
Vis. index	98	101	161.99	146	156.98	145
T.B.N mg KOH/gm Oil	5.61	4.9	7.45	6.82	6.66	6.42
Ash wt. %	0.64	0.60	0.91	0.82	0.78	0.75

3.4. Performance indicators

The evaluation of engine performance was based on measurements of the following parameters.

- Brake power: It represents the net power output from the engine, and it was calculated using Eq. 1 [12]

$$PW = \frac{2\pi NT}{60000} \quad (1)$$

- BSFC in kg/kW·h: It was determined using Eq. (2) [7]:

$$BSFC = \frac{mf}{B.P} \quad (2)$$

- Friction power: It was calculated according to Eq. (3) [6]:

$$F.B = I.P - B.P \quad (3)$$

4. Results and discussion

Three different engine oils – B1, P1, and G1 – were tested to determine how they affected the performance and emissions of a single-cylinder, four-stroke diesel engine at 1000, 2000, and 2500 rpm under a constant load of 4.0 Nm. Power output, BSFC, exhaust gas temperature, and CO, CO₂, and NO_x emission levels were among the key performance indicators. As shown in Fig. 3, the applied load remained constant across the tested oils, yet there were minor differences in brake performance. Reduced frictional losses cause the lighter, more thermally stable oil (P1) to deliver slightly more brake power across all engine speeds. On the other hand, the higher-viscosity oil (B1) showed somewhat lower power output because it offered greater resistance. G1 showed intermediate behavior.

For all tested oil types (B1, P1, and G1), Fig. 4 shows a decrease in BSFC with increasing engine speed. Usually explained by improved combustion efficiency and lower relative frictional losses at higher speeds, this inverse relationship is a well-documented phenomenon. To be more precise, the engine performs less effectively at the lowest

speed (1000 rpm) due to increased relative friction and insufficient combustion. The highest BSFC values, particularly with B1 (0.351 kg/kWh), point to this. The combustion process is more thorough, and the mechanical losses of the engine represent a smaller proportion of total power output at the highest tested speed (2500 rpm). With a BSFC of 0.290 kg/kWh, P1 here has the lowest BSFC, i.e., the best fuel efficiency.

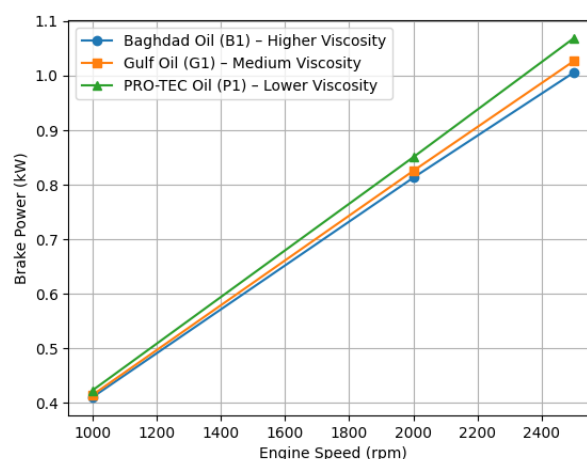


Fig. 3. Effect of oil type on brake power for different engine speeds

Their thermal stability and viscosity indices [2, 6] account for most of the differences in oil performance. The most stable viscosity across the speed and temperature range is found in PRO-TEC Oil (P1). It maintains a protective oil film and reduces friction. Lower BSFC readings result from less energy lost battling internal engine friction. Most likely due to either inadequate viscosity or poor additive package, Baghdad Oil (B1) has better BSFC at all speeds. More friction from this could increase engine wear over time. Gulf Oil's performance ranges from P1 to B1, indicating good viscosity stability and lubricity.

P1's lower BSFC across all speeds shows it is more efficient at lowering frictional losses than B1 and G1. At higher speeds, when oil-film breakdown is a possibility and frictional forces rise, this is particularly important. On the other hand, B1's higher BSFC points to greater internal resistance, perhaps due to low viscosity or insufficient anti-wear chemicals, which not only raise fuel consumption but may also accelerate wear of engine parts. Good lubrication (like that of P1) keeps piston motion more regular and improves sealing, hence enhancing combustion quality and lowering fuel waste [27–30]. On the other hand, insufficient lubrication causes the engine to burn more fuel, incomplete combustion, and increased blow-by [26], as B1 does.

Lower BSFC directly translates into less fuel use per unit of power, thereby benefiting the economy (reduced operating expenses) and the environment (less CO₂ emissions). As a result, the type of oil you use is a crucial operational choice, particularly in high-utilization or large-scale diesel applications, where even minor changes in BSFC can yield high cost and emissions reductions over time. Oils like P1, which retain their thickness well, not only help your car use less gas now but also protect the inside of your

engine. This could help your engine run longer and lower how often you have to change the oil. Modern diesel engines should use premium, thermally stable oils in line with industry advancements toward tougher emission standards and cleaner, more efficient operation. The results reinforce this need. Later plots show that the BSFC trend in Figure 4 backs up the documented decreases in exhaust temperature and emissions (NO_x , CO_2) with P1 oil. This full evolution makes clear how closely related environmental performance, combustion efficiency, and lubrication are.

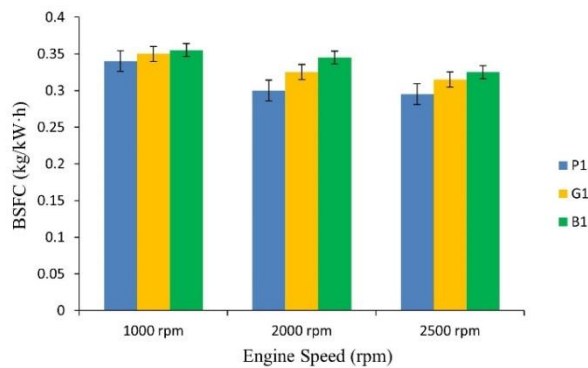


Fig. 4. Effect of oil type on BSFC for different engine speeds

Figure 5 shows a clear, consistent trend across all oil types: exhaust gas temperature (EGT) increases with engine speed. This is natural, since more fuel is required per unit time at higher engine speeds, which generates more heat and results in higher exhaust temperatures. Given the great impact of oil type on EGT, more research is nevertheless justified. Remarkably, Baghdad Oil (B1) has the highest exhaust temperatures across all engine speeds, 243°C at 2500 rpm. On the other hand, PRO-TEC Oil (P1), with a minimum of 188°C , has the lowest exhaust temperatures across all speeds. Usually, between B1 and P1, Gulf Oil (G1) has values in the middle. The various EGT values of the oil kinds may be explained by their viscosity indexes, additives, and thermal stability. P1's superior performance likely results from its high viscosity index and complex additives, which help maintain a stable lubricating film even at high speeds and temperatures. Great viscosity also reduces friction between moving engine parts, and mechanical losses generate less heat. On the contrary, B1's higher EGT points to a lower viscosity index or a less effective additive composition, which increases friction and the heat transferred to the exhaust gases. P1's lower EGT suggests improved engine thermal control [17]. P1 oil's improved frictional heating reduction lowers the thermal burden on engine components and helps prevent oil breakdown and thermal degradation.

Lower exhaust temperatures may signal more efficient energy conversion, since more of the fuel's energy is used for mechanical work rather than being wasted as heat [5]. Although the extraordinarily high EGTs might also encourage the development of nitrogen oxides (NO_x), as these emissions are very temperature-dependent [19]. The lower EGT observed with P1 confirms the link between oil choice, combustion temperature, and pollutant generation,

thereby supporting the study's findings on decreased NO_x emissions.

Therefore, PRO-TEC Oil (P1) not only enhances fuel economy (as seen in Fig. 3) but also helps to better manage temperature, which is very important for engines running under varying or high-load conditions. Although using Baghdad Oil (B1) may lead to higher operating temperatures, increased emissions, and faster engine wear, it is less appropriate for demanding or long-term use.

Lower EGTs have benefits for the environment and the economy by reducing thermal pollution and lowering the risk of heat-related engine failure. Therefore, not only about present engine performance but also about engine durability, emission law compliance, maintenance costs, and other domino effects of oil choice. As will be seen in subsequent pages, the lower EGTs detected with P1 match its overall improved thermal stability, lower NO_x and CO_2 emissions, and lower BSFC profile. This extensive advantage profile makes it obvious the need to pick premium, thermally stable lubricants for modern diesel engines.

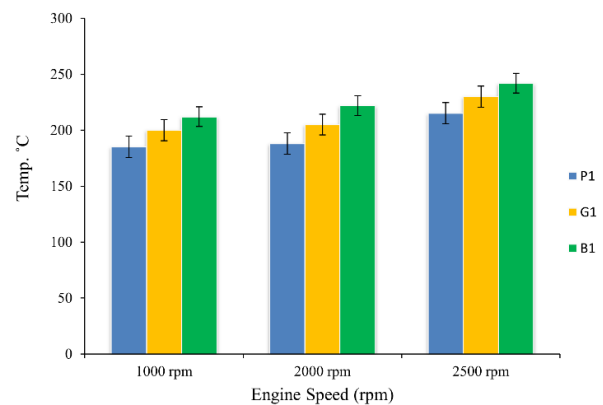


Fig. 5. Effect of oil type on the exhaust gas temperature through different engine speeds

Figure 6 shows that CO emissions for all kinds of oil increase with engine speed. Diesel engines' combustion dynamics match this trend: speed increases; fuel injection and combustion get faster and less complete; hence, carbon monoxide emission rises, a sign of incomplete combustion. Particularly at faster speeds, P1 generated the most CO emission (0.018%); At 1000 rpm, B1 had the least CO emission (0.013%); G1 always showed values between B1 and P1.

Among the oils, variations in CO emissions may be explained by their viscosity and their impact on combustion quality. P1's higher viscosity and thermal stability will help reduce friction and lower engine temperatures, enabling a more sophisticated design. However, at higher speeds, this same feature can create a somewhat thicker oil film that could slow flame propagation or interfere with ideal air-fuel mixing, leading to pockets of incomplete combustion and higher CO emissions. On the other hand, B1's decreased viscosity could enhance atomization and mixing at low speeds, therefore lowering CO emissions; this comes at the cost of higher friction, wear, and other emissions (as shown in Fig. 3, 4, 6, and 7).

P1 oil gives the usual trade-off. As shown by slightly increased CO emissions, its recipe could marginally increase the likelihood of incomplete combustion at high speeds, even though it is very good at reducing friction, heat stress, NO_x, and CO₂. B1 oil is better even if its total emission profile is greater, since its smaller CO production at lower speeds, possibly due to faster combustion and less oil film interference, makes it preferable. Still, this benefit vanishes with more velocity. CO clearly shows combustion efficiency. The somewhat higher CO emissions with P1 at high speeds indicate that although the oil's lubricity is good for engine wear and fuel economy, it can affect the combustion chamber environment in ways that promote partial combustion under certain conditions.

CO emissions are regulated contaminants due to their toxicity and their role in lowering urban air quality. For fleets or high-use engines in particular, even small increases might be quite apparent overall [8]. The data imply that oil selection should align with operational priorities; if lowering CO is important (e.g., in indoor or urban settings), oil choice and engine calibration may need to be adjusted jointly. A slight rise in CO with P1 can be reduced by adjusting injection timing, air-fuel ratio, or using an oxidation catalyst. Future oil mixes could aim to maximize combustion completeness through sophisticated additive packages while still preserving P1's friction-reducing properties.

The CO emission trend in Fig. 6 supports the conclusions from Fig. 3 and 4. P1's better performance in lowering CO₂, NO_x, exhaust temperature, and BSFC comes at a little price in CO emissions at faster speeds. This underlines the complex interaction among lubrication, combustion, and emissions: supporting improvements in one field may occasionally cause problems in another. Lean diesel running shows mostly full combustion, as indicated by low CO emissions (0.013–0.018%). These values are common in part-load situations and support the conclusion that local fuel-rich regions, not global mixture enrichment, determine carbon monoxide production.

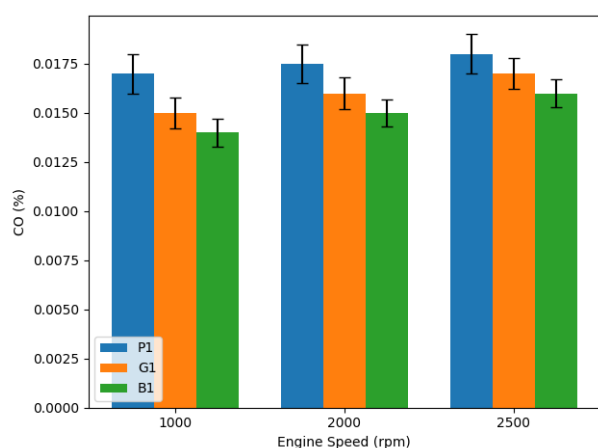


Fig. 6. Effect of oil type on CO emissions for different engine speeds

For every oil tested, Fig. 7 shows that HC emissions increase with increasing engine speed; a trend often observed in diesel engines under load. Unburned fuel components and hydrocarbons in exhaust gases directly indicate that

combustion is incomplete. The variations among oils underscore how lubricant characteristics affect this process. To be more exact, P1 oil produces the least HC emissions at lower speeds (e.g., 18 ppm at 1000 rpm), but shows a steeper rise at higher speeds, reaching 42 ppm at 2500 rpm. At low speeds (25 ppm), B1 type shows higher baseline HC emissions; at higher speeds, though, it shows a steadier increase to a maximum of 38 ppm. Consistent with its performance in past research, the G1 type shows a middle path.

Better combustion chamber sealing is shown by the lower HC emissions of P1 oil at low speeds, hence reducing fuel blow-by into the crankcase. It's a great additive package and constant viscosity, which maintains ideal piston ring lubrication, probably causes this. As engine temperatures increase with P1 oil, less combustion efficiency could account for the higher HC emission at high speeds. Reducing friction at high velocities with P1 oil can also slightly impede flame propagation or air-fuel blending, hence leaving unburned hydrocarbons. In contrast, B1 oil's higher HC emissions suggest poorer low-speed combustion efficiency, most likely due to inadequate lubrication. Still, its lower viscosity at higher speeds may improve mixing and thereby moderate the HC increase.

P1's performance indicates a sensitive equilibrium where its capacity to reduce friction raises fuel efficiency, as seen in Fig. 4, and lowers exhaust temperatures, as shown in Fig. 5. However, at extremely fast speeds, it could somewhat reduce combustion completeness and so produce more HC emissions. Conversely, the opposing trend of B1 stresses its disadvantages in low-speed sealing but some flexibility to fast conditions, albeit with trade-offs in wear, CO₂, and NO_x emissions (Fig. 8, 9). P1's anti-wear additives could reduce oil consumption, thereby controlling HC emissions from burned lubricant. On the other hand, B1's simpler form may raise oil volatility, therefore adding to HC emissions. G1's intermediate results indicate a balance between viscosity stability and additive effectiveness; hence, it performs in the middle of the spectrum across all measures.

Reducing HC emissions is therefore a top compliance priority with emissions standards (e.g., EPA Tier 4, Euro 6) since they help to cause respiratory problems and contribute to pollution. Given that engines often run at lower RPMs, P1's low-speed HC advantage is consistent with urban driving cycles. It should be noted that in applications such as heavy-duty transportation or generators, High-speed HC surges (as seen with P1) may cause issues, underscoring the need to select oil appropriate for the application. The data imply that improving the combustion system (e.g., injector design, swirl ratio) may help to lower P1's high-speed HC increase. Likewise, matching P1 with sophisticated after-treatment systems (such as diesel oxidation catalysts) would further enhance its environmental benefits.

It should also be noted that incomplete combustion yields both HC and CO [13]. P1's higher HC at high speeds should correspond to its higher CO emissions under identical conditions, thereby confirming the link between combustion dynamics and lubrication quality. The better performance of P1 in reducing CO₂, as indicated in Fig. 8, also

comes with a small trade-off in HC/CO at high speeds, underscoring the challenge of reducing emissions.

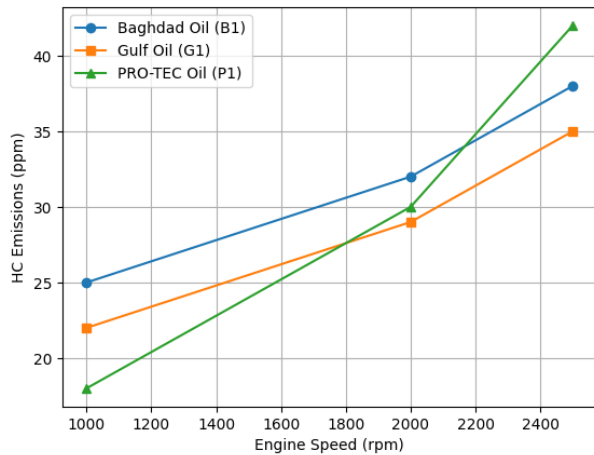


Fig. 7. Effect of oil types on the HC emissions for different engine speed

For every oil type examined, Fig. 8 shows that CO₂ emissions rise with engine speed, thereby highlighting the fundamental link between CO₂ generation and gasoline consumption. As an engine's load and speed increase, more fuel is consumed. More CO₂ results from the oxidation of the fuel's carbon content, hence driving up production. P1 oil has the least CO₂ emissions across all torque and speed ranges; B1 oil emits the most CO₂. G1 form lies between B1 and P1. P1's lower CO₂ emissions suggest better fuel efficiency, consistent with its lower BSFC and exhaust gas temperature, as shown in Fig. 4 and Fig. 5.

P1's increased lubrication lowers engine friction, thereby enabling more of the fuel's chemical energy to be transformed into mechanical work rather than being wasted overcoming friction losses. Lower fuel use from this translates into lower CO₂ emissions. B1's higher CO₂ production points to greater mechanical losses and less effective combustion, likely due to poor oil film stability and increased friction. P1's oil thermal stability and additive systems may also help to keep the combustion chamber in good working order, which stops incomplete combustion and reduces fuel waste.

The main greenhouse gas causing human-induced climate change is CO₂. It makes up most of the greenhouse gas emissions from burning fossil fuels worldwide. The modest CO₂ values (≈ 2.2 – 2.5%) point to high excess air availability throughout combustion, which is typical of diesel engines running at constant low load. Lean operation ($\lambda \geq 1$) and the rise in fuel consumption with engine speed both align with these levels. P1 oil's lowering of CO₂ emissions directly equates to a lesser carbon footprint for diesel engines, hence promoting sustainability objectives and regulatory compliance in ever more demanding emissions norms globally. Lowering CO₂ emissions improves fuel economy, which lowers the cost to operate industrial motors and vehicle fleets. Particularly in applications where they are utilized frequently, great lubricants like P1 can save a lot of fuel over time. Therefore, as shown here, the first stages of reducing emissions are crucial developments at the lubricant and engine levels. The results show the

necessity of well-coordinated plans that employ modern lubrication technology in conjunction with engine design and operating methods to achieve significant reductions in CO₂ emissions.

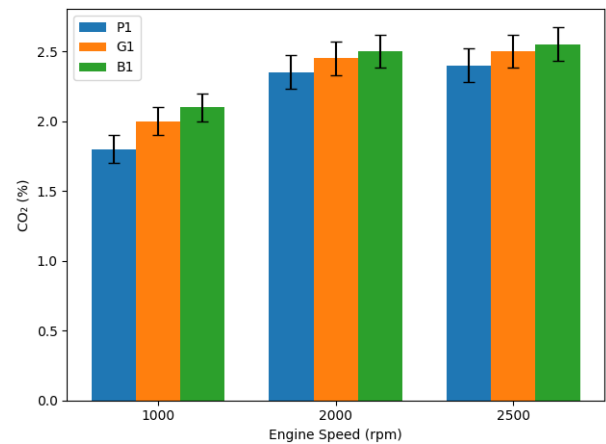


Fig. 8. Effect of oil type on CO₂ emissions for different engine speeds

Given P1's favorable influence on NO_x and hydrocarbon emissions, the CO₂ emission reductions reflect its equal improvement across a range of pollutant categories. Apart from its mechanical performance, the lubricant composition helps with environmental stewardship, as shown in this general emission reduction graph. For all examined engine oils, as shown in Fig. 9, higher combustion temperatures at higher speeds lead to increased NO_x emissions. P1 produces the lowest NO_x emissions among the oils, followed by G1; B1 has the highest values at all speeds. Higher engine speeds bring out the differences more clearly, showing that oils like P1, with better viscosity stability and thermal properties, reduce NO_x generation and frictional heating.

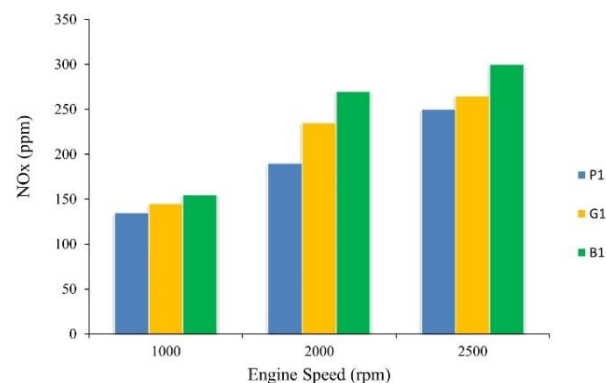


Fig. 9. Effect of oil type on the NO_x through different engine speeds

As shown in Fig. 10a, the friction coefficient for all oils increases with engine speed due to thermal effects and higher sliding velocity. Showing better lubrication and lower mechanical losses, P1 has the least friction; G1 has moderate behavior; B1 has the most friction. For all oils, the wear depth, as seen in Fig. 10b, also increases with engine speed. While P1 shows the lowest wear depth, indicating improved surface protection and increased anti-wear capability, B1 shows the highest wear, and G1 shows intermediate wear.

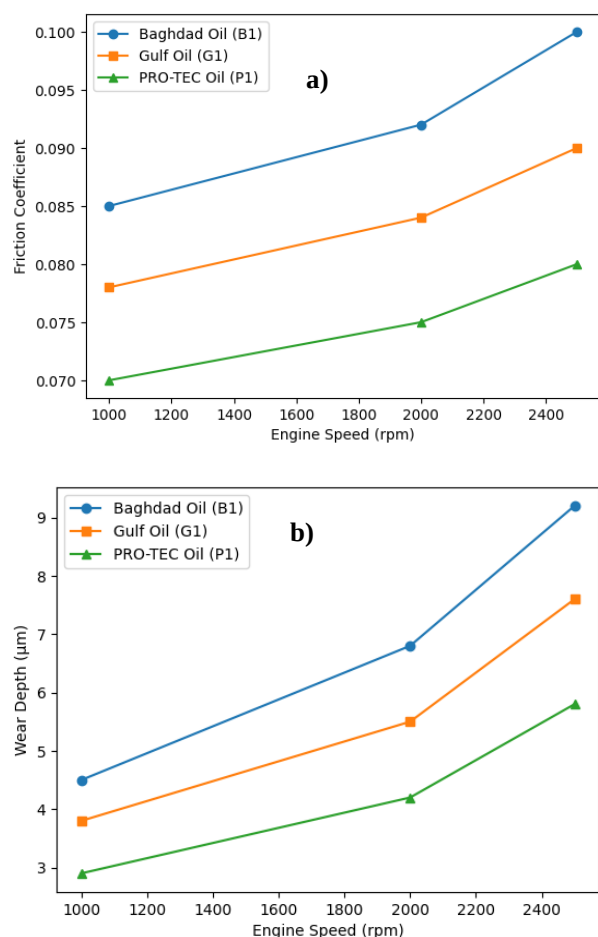


Fig. 10. a) Effect of oil type on the friction coefficient through different engine speeds; b) wear depth

5. Conclusions and future work

In controlled laboratory environments, three different engine oils, B1, P1, and G1, were meticulously evaluated for their impacts on the performance and emission profile of a single-cylinder, four-stroke diesel engine. The results show that oil type significantly affects engine efficiency, exhaust gas temperature, and emissions across a range of engine speeds.

Among the oils investigated, P1 oil showed the lowest BSFC and exhaust gas temperatures, indicating its excellent lubricating properties that help minimize heat stress and friction losses at higher speeds. Higher combustion efficiency and decreased production of harmful substances are also indicated by the lowest NO_x and CO_2 emissions seen with P1 oil use. But with the P1 type at higher engine speeds, a tiny rise in CO emissions was seen. B1, on the other hand, possibly due to its lower viscosity and reduced thermal stability under load, exhibits higher BSFC, higher exhaust gas temperatures, and higher NO_x emissions.

The results of this research emphasize the need for the kind of engine oil applied to lower the environmental footprint and boost engine efficiency. Emphasizing especially the necessity of aligning oil grades with specific engine operating conditions, P1 oil at the lowest examined engine speed (1000 rpm) provided the ideal balance of lower friction, reduced exhaust temperatures, and the least emissions. Future research might look into these avenues to build on these findings:

- To evaluate oil deterioration, wear rates, and maintenance intervals for each oil type under actual operating conditions, long-term engine tests were conducted; therefore, durability testing was expanded.
- Broader oil compositions are researched by observing their behavior to determine how synthetic and semi-synthetic oils, as well as oils with advanced additive packages, influence modern diesel engine technologies.
- Engine performance and emissions under varying loads and transient speed conditions should be evaluated using variable loads and transient conditions to better reflect real-world driving situations.
- Analyzing oil prices, engine maintenance expenses, and possible fuel savings in light of which to direct useful recommendations for fleet managers and single consumers.

By further defining the link between lubricating oil properties, engine lifespan, and environmental performance, these phases help improve the selection of oil for various diesel engine applications.

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Systems for generating thrust force outside the propeller

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This article presents conclusions on the potential to improve the operational indicators of piston aircraft engines by using thrust-generation systems outside the propeller. In this case, energy normally lost as emitted heat and exhaust kinetic energy is utilized through a proprietary flow channel design, which is significantly more efficient than previously used confusor-based solutions. The analysis is based on ongoing design work. To further illustrate the issue, calculations for an existing aircraft engine and its impact on operational indicators are also presented. The article provides a computational procedure as a basis for further research and reports favorable results.

Key words: aircraft piston engines, thrust force, thrust force generation systems, operational indicators

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1. Introduction

Piston aircraft engines have not been significantly developed since the 1950s. The use of modern structural analysis tools and new design solutions enables better development of the propulsion system than in the 1950s [25]. Based on the engine design, two versions were developed: a record-breaking racing engine that would allow for breaking the speed record for piston-powered aeroplane and for aeroplane in which the main source of thrust is a classic, subsonic propeller [27, 48], and a version that could be an alternative to turboprop engines [27].

Based on the designs, solutions are developed to significantly improve the operational indicators of piston aircraft engines, including the power-to-weight and thrust-to-weight ratios. These solutions can be used for various applications, but simply adapting them to currently produced piston aircraft engines can significantly improve the propulsion's operational parameters, and therefore aeroplane's performance. At the same time, while maintaining the same engine power, significantly greater thrust can be achieved with less mass [27].

The article will discuss the concept of a thrust-generating system outside the propeller, along with the calculation procedure. Based on the calculated system parameters, the thrust force will be determined, which, in turn, will enable a comparative analysis.

2. Using propeller propulsion in high-speed flights

Currently, piston engines have given way to turbine propulsion in most aviation applications. They are still used in light aircraft, which usually do not exceed 300–400 km/h [43]. Similar to turboprop engines, the propeller is powered by the engine, which results in similar theoretical speed limitations for both types [22, 27].

The highest officially recognized speed of a propeller-driven aeroplane is held by the Tu-114, at 877 km/h [39]. Unofficially, this record was broken by the experimental Republic XF-84H, which reached 1080 km/h [18, 30] thanks to special propellers designed to work in supersonic conditions. However, their considerable noise made them

impractical for use. Theoretically, they were supposed to allow a plane to break the sound barrier.

Among piston-engine aeroplane, the speed record for a very long time was held by the Rare Bear, a modified Grumman F8F Bearcat, which reached 850 km/h [7]. In 2017, the record was broken by a specially prepared P-51 Mustang "Voodoo", reaching 893 km/h [26].

The main obstacle to increasing the speed of propeller-driven aeroplane remains the decrease in propeller efficiency as it approaches the speed of sound [37] – most classic designs lose significant efficiency at $M > 0.7$ – 0.8 . The propeller blade speed is a component of two velocities: the linear velocity of the propeller blades and the forward velocity of the aeroplane, so in reality it is greater than the speed perpendicular to its radius [37].

One way to limit the problem is to reduce the propeller's rotational speed so that the tips of the blades do not exceed Mach 1. Assuming the speed of sound is 1200 km/h, it can be estimated that classic propellers do not allow for flights faster than 900–950 km/h. For a 4 m propeller, the rotational speed would have to decrease to about 1000 rpm. Further reductions would require more complex gears, increasing the weight and reducing the mechanical efficiency of the entire system [27]:

$$\sqrt{v_p^2 + v_a^2} < 1200 \text{ km/h} \quad (1)$$

where: v_p – linear speed of propeller blade tip, v_a – aeroplane speed.

3. General concept

To develop methods for improving operational indicators, it is necessary to rely on data derived from calculations of the designed engine. Developing its concept, along with determining masses and operational indicators, allows for developing appropriate improvement methods and comparing the advantages of solutions that generate thrust outside the propeller. At the same time, theoretical development of the propulsion system can confirm the thesis that a properly designed two-stroke engine with thrust-generating systems outside the propeller can provide operational indicators comparable to or better than those of turboprop engines [27].

Additionally, a second version of the engine is presented, which may enable setting a speed record for aircraft with a piston engine and a classic propeller. The planned speed is 950 km/h at an altitude of 3000 m. Due to the key requirements, the most important features are the highest possible power and thrust with the lowest possible weight and a simplified design. These are important features compared to turboprop engines. Therefore, a two-stroke cycle [9] is planned, which allows a higher power density than in a four-stroke engine. Unlike the complex Rolls-Royce Crecy or Napier Nomad [23] engine, the planned design will not have a complex valve train to reduce weight [41]. Instead of liquid cooling, air cooling will be used, and lightweight materials will be used in the construction, with limited use of aluminum in favor of titanium and magnesium alloys (Fig. 1 and Fig. 2) [27].

The article will present the advantages of the proposed system for generating thrust outside the propeller, based on the engine design and on sample calculations for the existing propulsion system.

The adopted engine displacement is 32 dm³, and the estimated rotational speed is 7000 rpm. For efficient air cooling, there should be no more than four cylinders in a row [28], allowing several systems to be considered: double- and triple-row radial, H and X systems. Despite the air-cooling system being favorable, radial engines do not reach high rotational speeds [28]. This is due to the large mass concentrated on a single crankpin of the crankshaft and the small number of simultaneous ignitions. The X system with a single shaft is more suitable, but the H system allows for crankcase scavenging (the Schnürle system was chosen) (Fig. 3) [33], which allows the mechanical compressor to be dispensed with and only the turbocharger to be used for supercharging [27]. The H system, in this case two connected engines with an opposed-cylinder arrangement, also allows for better balancing and reduced balancing masses [27].

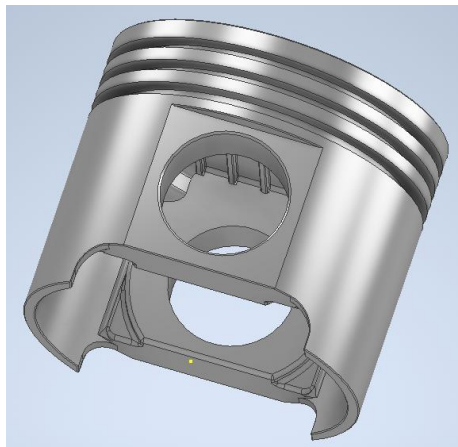


Fig. 1. Engine piston, which can be made of magnesium alloy WE43A or Al-Si casting alloy AK12 AK12

Both crankshafts will be connected by a helical gear (ratio 1:7), which will also serve as an rpm reducer. Due to the dimensions of the large gear, it will be moved above the shaft line [24], forcing the entire engine to be rotated by 90°, with two rows of vertical cylinders and two rows of overhead cylinders [27].

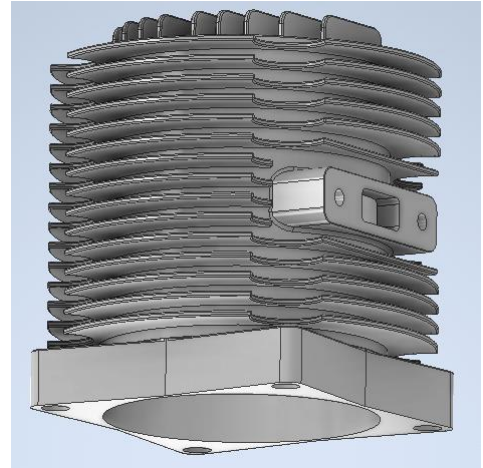


Fig. 2. One of the engine heads made of WE43A magnesium alloy

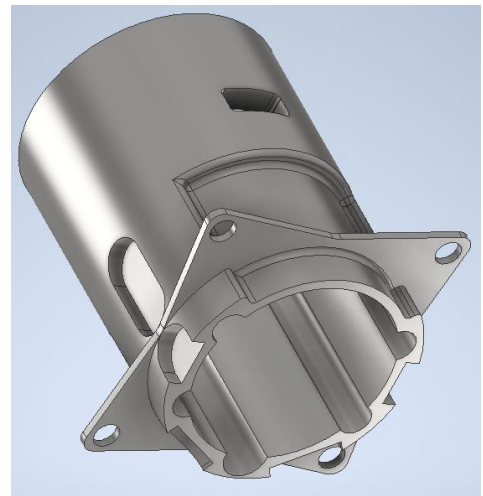


Fig. 3. Cylinder of the designed engine made of 36NiCrMo16 steel; the intake ducts are visible on the inner surface

For the racing version of the engine, a mechanical supercharger with an electric drive is planned, which is activated only when breaking a record. The aircraft's maximum speed would be achieved at an altitude of 3000 m, where the pilot does not need an oxygen system and air resistance is lower than at sea level. The compression ratio of the supercharger should be $\pi = 4.2$. Additionally, the engine will be supported by a turbocompound system, providing 25.4% additional power through a gear connected to the crankshafts [3, 23, 27, 44].

The transport version would require a supercharging system capable of operating throughout the flight. The turbocompound turbine would be smaller, contributing 18% to engine power. It would also drive the second-stage turbocharger. The first stage would be a conventional turbocharger, not connected to the crankshafts. The total pressure ratio would depend on altitude: $\pi = 2.77$ at sea level, $\pi = 4$ at 3000 m, and $\pi = 7.34$ at 7500 m. Compressor pressure would be varied by exhaust gas bleed. This would maintain constant power over a wide altitude range. During operation, the engine would be continuously boosted by both the power turbine and the turbochargers [27].

To increase thrust, the entire propulsion system will be placed in a special channel. The air cooling the engine

would take up some of the heat, increasing its flow velocity, and then it would combine with the exhaust gases, which would increase the kinetic energy in the channel. The outlet directed toward the rear of the aircraft would generate additional thrust force [27].

4. Anzani-Argus connecting rod system

To simplify the crankshaft design, reduce its mass, and improve dynamic balance, a connecting-rod system was used, based on solutions tested in aircraft engines with a set of opposite cylinders developed by Anzani and Argus in the 1930s [36]. Therefore, it is justified to call it the Anzani-Argus system. Thanks to its design, the gas forces in the opposing piston-connecting rod systems (Fig. 4 and Fig. 5) completely balance each other, as demonstrated by the analysis of bending moment distribution in the crank-piston system [40, 47].

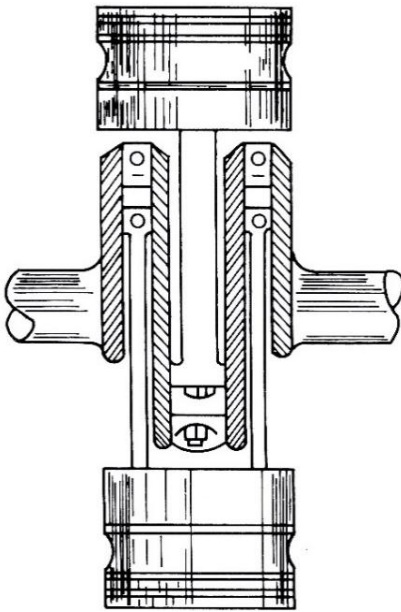


Fig. 4. Anzani-Argus connecting rod system [35]

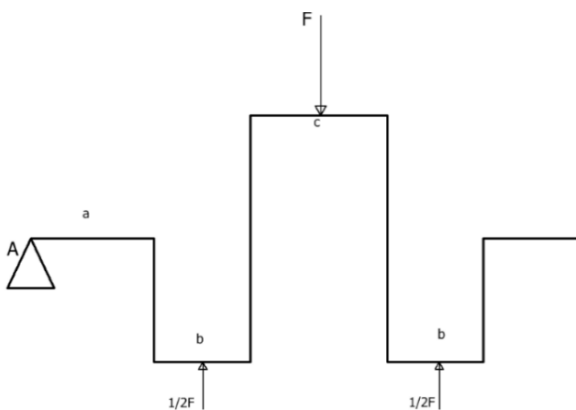


Fig. 5. Distribution of forces in the Anzani-Argus system

$$\sum M_{gA} = 0 \quad (2)$$

$$0 = \frac{1}{2}F\left(a + \frac{b}{2}\right) - F\left(a + b + \frac{c}{2}\right) + \frac{1}{2}F\left(a + \frac{3}{2}b + c\right) \quad (3)$$

$$0 = F\left(\frac{a}{2} + \frac{b}{4} - a - b - \frac{c}{2} + \frac{a}{2} + \frac{3b}{4} + \frac{c}{2}\right) \quad (4)$$

$$0 = F(0a + 0b + 0c) \quad (5)$$

$$0 = 0F \quad (6)$$

It is important to balance the forces resulting from the inertia of moving elements, which was probably the main design challenge during the 1930s experiments conducted by the mentioned companies. For various reasons, the tests were not completed at that time – the Anzani company went bankrupt, while Argus focused on producing power units for Luftwaffe aeroplanes [36, 45]. However, with CAD programs, it is possible to accurately determine the masses on both sides of the crankshaft axis, which greatly facilitates balancing the entire system [27].

The discussed system also has the advantage of allowing the crankshaft to be shortened, and consequently the total length of the engine. This is possible because the opposing connecting rods are not offset relative to each other along the crankshaft. The presented calculations showed that the bending moments of the forces in the piston-crank system acting on the crankshaft are balanced within the piston pair, so the rest of the shaft is not loaded with additional bending moments, but only with the torsional moment and any torques from the speed reducer [27].

Significant forces within the pair must be transferred to the shaft journals and crank arms. This requires their larger dimensions and the use of materials with high bending strength. In this case, 65S2WA alloy steel was selected, with a yield strength of $R_e = 1665 \text{ MPa}$ [51], which ensures adequate bending resistance under demanding operating conditions.

5. Basic operational indicators of a piston engine

The basic operational indicator characterizing a piston engine is its power. This allows for estimating values such as the power-to-weight ratio and the power-to-displacement ratio. Based on Jędrzejowski's publication *Calculations of a Piston Combustion Engine* [13], a simple comparison of engine power as a function of compression ratio was developed, accounting for the cylinder temperature during compression and thus the risk of advanced ignition. Aviation gasoline 115/145, with a self-ignition temperature of 744 K [38], allows for the use of a compression ratio of $\epsilon = 10$ in a record-breaking aircraft engine. However, when designing an engine version intended for passenger aircraft, this fuel proves impractical due to its high cost. For this reason, it is justified to use 100LL gasoline, with an auto-ignition temperature about 30 K lower than 115/145 or cheaper [50], or unleaded UL91 [15], with an auto-ignition temperature about 100 K lower. To avoid major differences between the two engine versions, it was decided to keep the same compression ratio. At the same time, this necessitates the addition of an additional charge air cooler and the extension of the intake system pipes [10].

During the calculations, the efficiencies of the gears and piston-cylinder assemblies were accounted for. When determining power losses due to friction in forty plain bearings and twelve rolling bearings, the friction coefficients given in the literature [16, 49] were used, which allowed for

a more precise estimation of the engine's effective power than in the case of assuming a constant value of mechanical efficiency based on the literature [13]. Based on the calculations, the overall efficiency was determined to be $\eta_o = 29.94\%$ for the racing engine and $\eta_o = 29.66\%$ for the transport aeroplanes engine. It was calculated according to the formula [13]:

$$\eta_o = \frac{P_e \cdot V_s}{W_u} \quad (7)$$

where: P_e – effective pressure (741 750.2 Pa for racing engine and 734893.3 Pa for passenger or transport aircraft engine), V_s – displacement volume needed to burn 1 kg of fuel (17.758 m³/kg), W_u – gasoline calorific value (44 MJ/kg).

To accurately determine the power-to-weight ratio, the actual mass of the entire engine must be known, which is only possible after the design phase is completed. For the comparative analysis, an approximate mass of 550 kg was assumed, based on previous design work and literature data covering many existing designs [1, 8, 11, 14, 19, 21, 27, 28, 42]. Based on the calculations [13], the most important operational indicators were identified. The table presents both the engine version for powering a high-performance aircraft and the engine version intended as a hypothetical power source for commercial aircraft (Table 1). Cruise values are not provided for the racing version, as only maximum power is important for breaking records. The cruise power of the commercial aircraft engine was determined at 90% of the maximum speed, i.e., 6300 rpm [27].

Table 1. Operational indicators of the designed engine

Engine	Racing	Transport
Maximum rotation speed:	7000 rpm	7000 rpm
Max. power	5014.68 hp (3687.27 kW)	4,652.04 hp (3420.61 kW)
Cruising power	–	4186.83 hp (3078.55 kW)
Power-to-weight ratio (estimate)	9.12 hp/kg	8.46 hp/kg
Power to stroke volume ratio	156.71 hp/dm ³	145.38 hp/dm ³
Fuel consumption when operating at maximum power	1007.62 dm ³ /h	943.58 dm ³ /h
Fuel consumption when operating at cruising power	–	849.22 m ³ /h

6. Thrust force generated outside the propeller

To determine the thrust generated outside the propeller, it is important to take into account that calculations and considerations must be conducted separately for both versions of the engine, however, some characteristic features will be common to both versions. The thrust will be generated by a flow channel, inside which the engine will be located, giving off its heat to the air flowing around it, and an exhaust outlet in the shape of a convergent pipe that increases the speed of the exhaust gases. The exhaust gases mix with the air in the flow channel, which would further increase the flow rate by giving up some of their kinetic and thermal energy. An accelerated stream of air mixed with

exhaust gases would generate additional thrust, in addition to the propeller [27].

The method of generating thrust using the kinetic energy of exhaust gases was widely used during World War II. An example is the Rolls-Royce Merlin engine, with a displacement of 26 dm³ [1, 19], in which the exhaust gases were accelerated at the outlet to 580 m/s. This allowed approximately 0.67 kN of thrust to be generated [23]. This is approximately equivalent to 188 hp, assuming that, due to propeller efficiency, 80% of the engine power is converted to thrust. A similarly sized experimental two-stroke Rolls-Royce Crecy engine generated 1.4 kN of thrust in the same way [23]. This also demonstrates the advantage of two-stroke engines in aviation, which not only can achieve higher power but also emit more exhaust gases, the kinetic energy of which can be used to increase the thrust of the powerplant. At the same time, the higher exhaust energy allows for more efficient use of the turbocharger and power turbine. Another example is the Rolls-Royce Crecy, in which the power turbine increased maximum power by 54% [23].

First, the thrust of the racing engine will be determined. To do this, several assumptions must be made (Table 2).

Table 2. Parameters assumed to calculate the mass flow rate in the flow channel for the racing engine version

Fuel consumption	$G_e = 1007.62 \text{ l/h} = 0.2805 \text{ l/s}$
Theoretical amount of air required to burn 1 kg of fuel	$M_t = 15$
Air-fuel ratio	$\lambda = 0.9$
Gasoline density	$\rho = 0.72 \text{ kg/dm}^3$

Based on the data in the table, the exhaust gas mass flow can be determined, taking into account the approximately 3% oil content in the fuel mixture [12, 31].

$$\dot{m}_{pa} = G_e \cdot \rho = 0.2805 \cdot 0.72 = 0.202 \text{ kg/s} \quad (8)$$

$$\dot{m}_p = \dot{m}_{pa} \cdot M_t \cdot \lambda = 0.202 \cdot 15 \cdot 0.9 = 2.727 \text{ kg/s} \quad (9)$$

$$\dot{m}_{sp} = 1.03 \cdot (\dot{m}_{pa} + \dot{m}_p) = 1.03 \cdot (0.202 + 2.727) = 3.017 \text{ kg/s} \quad (10)$$

where: $\dot{m}_{pa}$ – fuel mass flow, ρ – aviation gasoline density, $\dot{m}_p$ – air mass flow rate through the engine, $\dot{m}_{sp}$ – air mass flow.

Assuming that the aircraft is traveling at a speed of 950 km/h and that air enters the flow channel at this speed and the engine releases heat to the environment through direct cooling, then based on the energy balance, its effect on the temperature of the air flowing around it would be approximately $\Delta T = 21 \text{ K}$ [12]. Assuming the air temperature at an altitude of 3000 m is $T_p = 268 \text{ K}$, after heating, $T_1 = 289 \text{ K}$, allowing the flow velocity to increase to $v_p = 285 \text{ m/s}$. Using the data from the table (Table 3), further calculations can be performed. The exhaust gas velocity at the outlet was assumed to be $v_s = 450 \text{ m/s}$. The weighted-average velocity of the mixing exhaust gas and air, based on their mass flow rates, is $v_2 = 293 \text{ m/s}$ (Fig. 6) [2].

Table 3. Assumptions needed to calculate the thrust for the racing version of the engine

Flight speed (at 3000 m altitude)	$v_1 = 950 \text{ km/h} = 264 \text{ m/s}$
Channel inlet area	$A = 0.4 \text{ m}^2$
Air mass flow rate	$\dot{m}_p = 96 \text{ kg/s}$
Air heating by the engine	$\Delta T = 21 \text{ K}$
Air temperature after heating	$T_1 = 289 \text{ K}$
Air velocity after heating	$v_p = 285 \text{ m/s}$
Exhaust mass flow rate	$\dot{m}_s = 3.017 \text{ kg/s}$
Exhaust velocity at the convergent pipe outlet (minimum design value)	$v_s = 450 \text{ m/s}$
Weighted average of air and exhaust velocity	$v_2 = 293 \text{ m/s}$
Exhaust temperature based on calculations	$T_s \approx 1050 \text{ K}$
Air density at 3000 m altitude	$\rho_1 = 0.909 \text{ kg/m}^3$

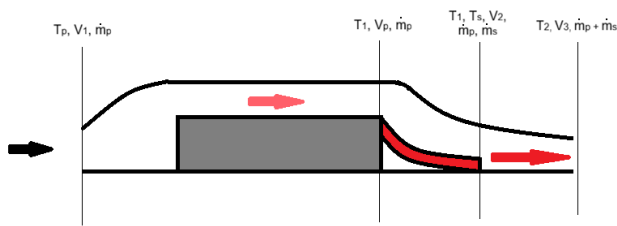


Fig. 6. System generating thrust operation diagram; the first section from the left shows inlet parameters; the second shows parameters directly after the engine (gray); the third shows the moment when exhaust fumes leave the exhaust system (red); and the fourth section shows outlet parameters when the air and exhaust streams have mixed

Based on additional data, the effect of exhaust temperature at the outlet of the piston engine's exhaust system can be determined. Based on the further energy balance, the average air and flue gas temperature in the duct can be determined as $T_2 = 313 \text{ K}$. Based on the continuity of the flow, the continuity equation:

$$\frac{\rho_1}{\rho_2} = \frac{T_2}{T_1} = \frac{313}{289} = 1.083 \quad (11)$$

$$\rho_2 = \frac{\rho_1}{1.083} = 0.839 \text{ kg/m}^3 \quad (12)$$

$$\frac{v_3}{v_2} = \frac{(\dot{m}_p + \dot{m}_s) \cdot \rho_1}{\dot{m}_p \cdot \rho_2} = 1.117 \quad (13)$$

$$v_3 = v_2 \cdot 1.117 = 323 \text{ m/s} \quad (14)$$

where: ρ_2 – density of mixed air and exhaust gases, v_3 – the velocity of the mixed air and exhaust gases at the end of the flow channel.

Based on the determined velocities and mass flow rates, the thrust generated in the flow channel can then be calculated. First, for the exhaust gas mass flow rate itself:

$$F_s = \dot{m}_{sp} \cdot v_3 = 3.017 \cdot 323 = 975.4 \text{ N} \quad (15)$$

The total air and exhaust thrust can be calculated using the momentum change formula, which in this case will have the form [12, 20, 46]:

$$F = (\dot{m}_p \cdot v_3) - (\dot{m}_p \cdot v_1) + F_s = 6667.74 \text{ N} \approx 6.67 \text{ kN} \quad (16)$$

The exhaust gas velocity at the convergent pipe outlet of 450 m/s is not high compared to other engines [23]. When

it is increased to 550 m/s, the weighted average of the air velocity after heating behind the engine and the exhaust gases is $v_2 = 293 \text{ m/s}$. Using the same calculation procedure, the velocity will be $v_3 = 327 \text{ m/s}$. Naturally, the thrust will be higher and equal to:

$$F_s = \dot{m}_{sp} \cdot v_3 = 3.017 \cdot 327 = 985.66 \text{ N} \quad (17)$$

$$F = (\dot{m}_p \cdot v_3) - (\dot{m}_p \cdot v_1) + F_s = 7004.38 \text{ N} \approx 7 \text{ kN} \quad (18)$$

A transport or passenger aeroplane typically achieves lower flight speeds than those assumed for a racing aeroplane. For the purposes of this calculation, a maximum speed for such an aircraft was assumed at 750 km/h. The flight altitude was assumed to be 7500 m. Some assumptions must be established first (Table 4):

Table 4. The adopted parameters necessary to calculate the mass flow in the flow channel for an engine of a transport aeroplane

Fuel consumption	$G_e = 943.58 \text{ l/h} = 0.2621 \text{ l/s}$
Theoretical amount of air required to burn 1 kg of fuel	$M_t = 15$
Air-fuel ratio	$\lambda = 0.9$
Gasoline density	$\rho = 0.72 \text{ kg/dm}^3$

The exhaust gas mass flow calculated according to the same calculation procedure will be:

$$\dot{m}_{pa} = G_e \cdot \rho = 0.2621 \cdot 0.72 = 0.202 \text{ kg/s} \quad (19)$$

$$\dot{m}_p = \dot{m}_{pa} \cdot M_t \cdot \lambda = 0.189 \cdot 15 \cdot 0.9 = 2.552 \text{ kg/s} \quad (20)$$

$$\dot{m}_{sp} = 1.03 \cdot (\dot{m}_{pa} + \dot{m}_p) = 1.03 \cdot (0.189 + 2.552) = 2.823 \text{ kg/s} \quad (21)$$

Based on the energy balance, the effect of the engine on the air temperature around it is approximately $\Delta T = 38 \text{ K}$ [12]. Assuming the air temperature at an altitude of 7500 m is $T_p = 245 \text{ K}$, after heating it will be $T_1 = 283 \text{ K}$. This will allow the flow velocity to increase to $v_p = 240 \text{ m/s}$. The exhaust velocity of the piston engine was set to $v_s = 450 \text{ m/s}$. The weighted-average speed of the mixing exhaust and air, based on their mass flows, is $v_2 = 252 \text{ m/s}$. Here, the necessary parameters for further thrust force calculations will also be presented in Table 5 [2].

The additional thrust generated by heated air flowing through the charge air coolers should also be considered. Due to the inferior properties of UL91 and 100LL gasolines compared to 115/145 gasoline, two intercoolers are required. The engine design does not currently include a detailed supercharger system. However, it is possible to rely on some approximations. The cross-section of both intercoolers can be defined as $A_c = 0.1 \text{ m}^2$. According to the design, after the first stage of the compressor, the air temperature will be 358 K. Assuming that the first intercooler must absorb a temperature of 60 K, the air flowing through it as a cooling agent will increase its temperature by approximately $\Delta T_{c1} = 27 \text{ K}$. Its density after passing through the intercooler can then be determined using the ideal gas equation and the proportion of the velocity behind the cooler. This can be determined as $v_{c1} = 230.5 \text{ m/s}$. The thrust generated by the air flowing through the radiator will be:

$$\dot{m}_c = A_c \cdot v_1 \cdot \rho_1 = 12.27 \frac{\text{kg}}{\text{s}} \quad (22)$$

$$F_{c1} = (\dot{m}_{c1} \cdot v_{c1}) - (\dot{m}_{c1} \cdot v_1) = 276.08 \text{ N} \quad (23)$$

where: $\dot{m}_c$ – air mass flow through the intercooler.

Table 5. Assumptions needed to calculate the thrust for the engine version of the transport aeroplane

Flight speed (at 7500 m altitude)	$v_1 = 750 \text{ km/h} = 208 \text{ m/s}$
Channel inlet area	$A = 0.4 \text{ m}^2$
Air mass flow rate	$\dot{m}_p = 49.1 \text{ kg/s}$
Air heating by the engine	$\Delta T = 38 \text{ K}$
Air temperature after heating	$T_1 = 283 \text{ K}$
Air velocity after heating	$v_p = 240 \text{ m/s}$
Exhaust mass flow rate	$\dot{m}_s = 2.823 \text{ kg/s}$
Exhaust velocity at the convergent pipe outlet (minimum design value)	$v_s = 450 \text{ m/s}$
Weighted average of air and exhaust velocity	$v_2 = 252 \text{ m/s}$
Exhaust temperature based on calculations	$T_s \approx 950 \text{ K}$
Air density at 7500 m altitude	$\rho_1 = 0.59 \text{ kg/m}^3$

The second intercooler cools the air after the second compressor stage from 377 K to 337 K. The temperature change of the air flowing through the condenser will change by $\Delta T_{c2} = 18 \text{ K}$. Based on the same calculation procedure as before, the air velocity after the condenser will be $v_{c2} = 223.3 \text{ m/s}$. The obtained thrust force can thus be calculated:

$$\dot{m}_c = A_c \cdot v_1 \cdot \rho_1 = 12.27 \frac{\text{kg}}{\text{s}} \quad (24)$$

$$F_{c1} = (\dot{m}_{c2} \cdot v_{c2}) - (\dot{m}_{c2} \cdot v_{c2}) = 187.73 \text{ N} \quad (25)$$

In the same way as for the racing version of the engine, the effect of the exhaust gas temperature at the outlet of the exhaust system of a piston engine can be determined. It can be assumed that the exhaust gas temperature at the exhaust system outlet will be approximately 200 K lower. The temperature of the air in the flow channel mixed with the exhaust gas will therefore be approximately $T_2 = 310 \text{ K}$. Based on the flow continuity equation:

$$\frac{\rho_1}{\rho_2} = \frac{T_2}{T_1} = \frac{310}{283} = 1.095 \quad (26)$$

$$\rho_2 = \frac{\rho_1}{1.095} = 0.539 \text{ kg/m}^3 \quad (27)$$

$$\frac{v_3}{v_2} = \frac{(\dot{m}_p + \dot{m}_s) \cdot \rho_1}{\dot{m}_p \cdot \rho_2} = 1.156 \quad (28)$$

$$v_3 = v_2 \cdot 1.156 = 292 \text{ m/s} \quad (29)$$

Based on the determined speeds and mass flows, the thrust generated in the flow channel is calculated. First, for the exhaust mass flow itself:

$$F_s = \dot{m}_{sp} \cdot v_3 = 2.823 \cdot 292 = 825.03 \text{ N} \quad (30)$$

The total air and exhaust gas thrust can be calculated using the momentum change formula, which in this case will have the form [13, 41]:

$$F = (\dot{m}_p \cdot v_3) - (\dot{m}_p \cdot v_1) + F_s + F_{c1} + F_{c2} = 5424.67 \text{ N} \approx 5.42 \text{ kN} \quad (31)$$

By increasing the exhaust gas outlet velocity from the convergent pipe to 550 m/s, the flow velocity in the duct will increase. The weighted average of the air and exhaust gas velocities will be $v_2 = 257 \text{ m/s}$. Using the same calculation procedure, the velocity will be $v_3 = 299 \text{ m/s}$. The thrust force will change as calculated:

$$F_s = \dot{m}_{sp} \cdot v_3 = 2.823 \cdot 299 = 842.85 \text{ N} \quad (32)$$

$$F = (\dot{m}_p \cdot v_3) - (\dot{m}_p \cdot v_1) + F_s + F_{c1} + F_{c2} = 5752.36 \text{ N} \approx 5.75 \text{ kN} \quad (33)$$

7. Comparative analysis with turboprop engines

Assuming an efficiency of 80% for the propeller, the thrust it generates can be determined. For a racing engine, this would be 11.17 kN at a flight speed of 950 km/h, and for the passenger and transport version, 13.16 kN at a flight speed of 750 km/h. The passenger and transport versions will be further compared. The comparative object will be the Pratt & Whitney PW 150A turboprop engine (Fig. 7) [4]. According to the manufacturer, it delivers 5142.16 hp (3781 kW) [4]. Assuming it would move at the same speed as the two-stroke engine of 750 km/h, its propeller would generate a thrust of 14.55 kN. At the same time, thanks to the exhaust energy, it gains an additional 3.75 kN of thrust. A detailed comparison is shown in Table 6.



Fig. 7. Pratt & Whitney PW 150A engine [29]

Table 6. Comparison of the designed two-stroke engine with the PW 150A engine

Engine	Designed a two-stroke engine	PW 150A
Power	4652.04 hp (3420.61 kW)	5142.16 hp (3781 kW)
Mass	Approximately 550 kg	717 kg
Fuel consumption	943.58 dm ³ /h or 669.94 kg/h	740 dm ³ /h or 592 kg/h
Propeller thrust at 750 km/h	13.16 kN	14.55 kN
Thrust obtained from exhaust gases	5.75 kN	3.75 kN
Thrust of the entire power unit	18.91 kN	18.3 kN

Two completely different engines are being compared. Although the PW 150A delivers approximately 11% more power, its thrust is reduced by more than 3%. This is due to the use of a properly shaped channel that captures almost all the energy lost in piston engines, i.e., the energy of exhaust gases and the thermal energy given off to the envi-

ronment, excluding friction losses. The fuel volume consumption is lower for the PW 150A. However, aviation kerosene has a higher density than gasoline, which results in a smaller difference in fuel consumption [27].

To provide a more meaningful comparison of both propulsion systems, specific indicators will be compared, allowing a more effective determination of the hypothetical suitability of two-stroke engines as an alternative to turboprop propulsion, as presented in Table 7. To better illustrate fuel consumption, not only the specific fuel consumption will be compared, but also the specific fuel consumption in relation to thrust, calculated using the following formula:

$$g_c = \frac{G_e}{F} \quad (34)$$

where: g_c – specific fuel consumption in relation to thrust, G_e – hourly fuel consumption, F – thrust of the entire power unit.

Table 7. Comparative characteristics of the specific indicators of the two-stroke engine concept with the PW 150A turboprop engine

Engine	Designed a two-stroke engine	PW 150A
Power-to-weight ratio	8.46 hp/kg	7.17 hp/kg
Specific fuel consumption	0.389 dm ³ /kWh or 0.276 kg/kWh	0.196 dm ³ /kWh or 0.157 kg/kWh
Specific fuel consumption in relation to thrust	0.0499 dm ³ /Nh or 0.0354 kg/Nh	0.0404 dm ³ /Nh or 0.323 kg/Nh
The ratio of the power unit's thrust to the engine's weight	34.38 N/kg	25.52 N/kg
Ratio of power unit thrust to engine power	5.53 N/kW	4.84 N/kW

The specific indicators presented in the table demonstrate the advantage of the two-stroke engine over the turboprop. It is characterized by greater maximum power per unit of mass. The thrust-to-weight ratio is even more favorable. The engine under development also demonstrates improved power utilization in generating thrust, the most important indicator of aircraft propulsion. The main disadvantage is high fuel consumption. This would require in-depth economic analyses, taking into account the specific engine application and the type of operations performed.

If the racing version of the engine were used for comparison, the comparative performance would be even more favorable. For example, the 9.12 hp/kg figure is over 30% higher than the 7.17 hp/kg of the PW 150A engine.

Environmental factors are also important. While CO₂ emissions may be comparable to those of a turbine engine, unleaded UL91 gasoline will generate fewer particulates and nitrogen oxides during combustion [5, 17]. Therefore, using appropriately designed piston engines using UL91 gasoline instead of turboprop engines may be a simple and ecological solution leading to better environmental protection and compliance with increasingly stringent emission regulations.

8. Application of a thrust generation system outside the propeller in a four-stroke engine adapted for this purpose

The developed solution can potentially be easily adapted to existing engines. Most general aviation aircraft are pow-

ered by four-stroke engines [43], which is why this will be the model chosen for comparison in this article. It will be the latest product from Rotax – the 916 iS [52] (Table 8).

Table 8. Rotax 916 iS engine basic operating indicators [32, 52]

Maximum rpm	5800 rpm
Maximum power	160 hp (117 kW)
Maximum cruising power	137 hp (101 kW)
Weight	86 kg
Displacement volume	1.35 dm ³
Power to weight ratio	1.86 hp/kg
Power to displacement ratio	118.52 hp/dm ³
Fuel consumption when operating at maximum power	49 dm ³ /h = 0.0136 dm ³ /s
Specific fuel consumption when operating at maximum power	0.29 kg/kWh

For the Rotax 916 iS, a similar calculation procedure will be performed as for the designed two-stroke engine. It was assumed that the method of generating additional thrust would be the same as in the racing version of the previously analyzed propulsion unit. $\lambda=1$ was assumed. Based on the data from the table, the exhaust mass flow rate can be determined [12, 31].

$$\dot{m}_{pa} = G_e \cdot \rho = 0.0136 \cdot 0.72 = 0.0098 \text{ kg/s} \quad (35)$$

$$\dot{m}_p = \dot{m}_{pa} \cdot M_t \cdot \lambda = 0.0098 \cdot 15 \cdot 1 = 0.147 \text{ kg/s} \quad (36)$$

$$\dot{m}_{sp} = \dot{m}_{pa} + \dot{m}_p = 0.0098 + 0.147 = 0.157 \text{ kg/s} \quad (37)$$

where: $\dot{m}_{pa}$ – fuel mass flow, ρ – density of aviation gasoline, $\dot{m}_p$ – air mass flow through the engine, $\dot{m}_{sp}$ – exhaust mass flow.

Assuming that the aircraft is traveling at a speed of 300 km/h and that air enters the flow channel at this speed and the engine releases heat to the environment through direct cooling, then based on the energy balance, its effect on the temperature of the air flowing around it would be approximately $\Delta T = 3.21 \text{ K}$ [12]. Assuming the air temperature at an altitude of 3000 m is $T_p = 268 \text{ K}$, after heating it will be $T_1 = 272 \text{ K}$, allowing the flow velocity to be increased to $v_p = 84.3 \text{ m/s}$. Further assumptions, as in the case of calculations for a two-stroke engine, are given in Table 9. The exhaust gas velocity at the outlet was assumed to be $v_s = 500 \text{ m/s}$. The weighted-average velocity of the mixing exhaust gas and air, based on their mass flow rates, is $v_2 = 88.6 \text{ m/s}$ [2].

Based on the energy balance, the average air and exhaust gas temperature in the channel is $T_2 = 277 \text{ K}$. According to the flow continuity equation:

$$\frac{\rho_1}{\rho_2} = \frac{T_2}{T_1} = \frac{277}{272} = 1.018 \quad (38)$$

$$\rho_2 = \frac{\rho_1}{1.018} = 0.89 \text{ kg/m}^3 \quad (39)$$

$$\frac{v_3}{v_2} = \frac{(\dot{m}_p + \dot{m}_s) \cdot \rho_1}{\dot{m}_p \cdot \rho_2} = 1.028 \quad (40)$$

$$v_3 = v_2 \cdot 1.028 = 91.1 \text{ m/s} \quad (41)$$

where: ρ_2 – density of mixed air and exhaust gases, v_3 – the velocity of the mixed air and exhaust gases at the end of the flow channel.

Table 9. Assumptions needed for calculations of the Rotax 916 iS engine

Flight speed (at 3000 m altitude)	$v_1 = 300 \text{ km/h} = 83 \text{ m/s}$
Channel inlet area	$A = 0.2 \text{ m}^2$
Air mass flow rate	$\dot{m}_p = 0.147 \text{ kg/s}$
Air heating by the engine	$\Delta T = 3.21 \text{ K}$
Air temperature after heating	$T_1 = 272 \text{ K}$
Air velocity after heating	$v_p = 84.3 \text{ m/s}$
Exhaust mass flow rate	$\dot{m}_s = 0.157 \text{ kg/s}$
Exhaust velocity at the convergent pipe outlet (minimum design value)	$v_s = 500 \text{ m/s}$
Weighted average of air and exhaust velocity	$v_2 = 88.6 \text{ m/s}$
Exhaust temperature based on calculations	$T_s \approx 700 \text{ K}$
Air density at 3000 m altitude	$\rho_1 = 0.909 \text{ kg/m}^3$

Next, the thrust force calculations are performed according to the same procedure:

$$F_s = \dot{m}_{sp} \cdot v_3 = 0.157 \cdot 91.1 = 14.3 \text{ N} \quad (42)$$

The total air and exhaust thrust can be calculated using the momentum change formula, which in this case will have the form [12, 20, 46]:

$$F = (\dot{m}_p \cdot v_3) - (\dot{m}_p \cdot v_1) + F_s = 137 \text{ N} \approx 0.14 \text{ kN} \quad (43)$$

Assuming a propeller efficiency of 80% at a speed of 300 km/h, it would generate 1.13 kN of thrust. With the tested and calculated system, the powerplant thrust would be 1.27 kN, over 12% higher. However, this value is approximate, as it would require a more detailed analysis of the exhaust gas temperature at the end of the exhaust system. The presented example demonstrates the capabilities of the tested system. Although the greatest increase in thrust is achieved in two-stroke engines, it can be applied to four-stroke engines as well. It only requires appropriately shaping the exhaust system and designing a suitable flow channel within the aircraft.

Summary

Thermodynamic calculations of the engine were performed, along with the determination of its operational indicators. Using the obtained data, the thrust generated by the designed flow channel outside the propeller was determined. The solutions being developed allow for a significant increase in the thrust of the propulsion system based on a piston aircraft engine and for improving its operational

indicators. At the same time, they show that two-stroke engines using the proposed solutions can achieve, in most cases, operational indicators comparable to or better than those of turboprops. The presented solutions can be successfully adapted to general aviation aircraft propulsion systems, which mostly use piston engines. The construction of a record-breaking racing aircraft engine is potentially possible.

The use of high-power, high-speed two-stroke engines as an alternative to turboprop propulsion systems is theoretically possible. However, given the higher cost of aviation gasoline compared to aviation kerosene, a detailed economic analysis should be conducted, accounting for the possibility of carrying a larger load due to the lower weight of the propulsion system. The extremely favorable thrust-to-weight ratio of the propulsion system is crucial in this case. Furthermore, the engine from the proposed concept could certainly power transport aircraft not used for commercial flights, e.g., in the military. Similar engines could be successfully used in some aircraft operating from aircraft carriers, where the advantage of a piston engine over a turbine engine in terms of rapid increases in speed and power could significantly improve safety [27].

The design assumes a simple piston engine, unlike other high-power piston engines in aircraft (e.g., Pratt & Whitney R-4360, Rolls-Royce Merlin, or Daimler-Benz DB605 [45]). This would reduce not only its weight but also its price and operating costs.

The use of the proposed solutions in general aviation does not require additional analyses of exhaust emissions or harmful compounds, unlike in the automotive industry [34]. Within the European Union, aircraft performing non-commercial flights are exempt from compliance with emission standards. The same applies to aircraft performing commercial flights, but not exceeding a maximum take-off weight of 5700 kg [6].

Excluding commercial aircraft with a maximum takeoff weight exceeding 5700 kg, it is beneficial to strive to increase the exhaust mass flow, which can drive a turbocharger or power a power turbine, or generate thrust outside the propeller. The proposed concepts and designs for both engine versions serve primarily as examples from which specific solutions can be developed.

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Chiptuning impact on emissions in driving tests

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The article presents the results of engine tests aimed at assessing the impact of modifications to engine control software on emission parameters and fuel consumption. Experiments were conducted on a 1.3 L MultiJet compression-ignition engine compliant with the Euro 6 standard. The electronic control unit's EEPROM was extracted and re-calibrated by a leading chiptuning company. Two operating scenarios-factory calibration and the modified calibration-were executed on an engine test bench equipped with a dynamic dynamometer. To ensure repeatability and comparability, the dynamometer governed engine speed and load in accordance with the New European Driving Cycle (NEDC) homologation procedure. Analysis of the revised calibration maps revealed substantive alterations, chiefly to the boost-pressure characteristic, the maximum-torque map, and the maximum fuel-injection-quantity map as a function of air mass. The resulting emission data demonstrate that chip-tuning exerts a pronounced impact on the engine's emission profile, elevating the release of toxic exhaust constituents, most notably nitrogen oxides.

Key words: chiptuning, combustion engine, emission, ecology

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1. Introduction

The passenger car market has undergone significant changes over the past decade. Automobile companies are struggling with rising production costs, which are caused, among other things, by increasing environmental requirements forcing them to meet increasingly stringent emission standards. The cost of approving power units constitutes a significant part of the total cost of developing and producing a car; manufacturers therefore decide to reduce the number of engine variants available in production vehicles (Table 1). This trend is noticeable, among other things, in the change in the number of available engine variants between the current and previous model generations. For example, for the previous generation of the Fiat Scudo, engines with 3 different displacements and 4 power variants were available, while for the VW Caddy, two engines (1.6 dm³ and 2.0 dm³) with 5 power variants were available.

Table 1. Comparison of available powertrain units of urban vans [1, 2]

	Generation IV				Generation V		
	1.3 MJ	1.6 MJ	2.0 MJ		1.5 BlueHDI		
Fiat Doblo	75 HP	90 HP	105 HP	135 HP	105 HP	131 HP	
VW Caddy	1.6 TDI		2.0 TDI		2.0 TDI		
	75 HP	102 HP	110 HP	140 HP	122 HP	75 HP	102 HP
							122 HP

The latest generation no longer offers a choice of engine capacities, only different power variants. Technically, the units offered do not differ significantly in construction or equipment. Regardless of the variant, they are equipped with identical fuel, supercharging, cooling or exhaust after-treatment systems. The differences are most often in the power transmission system used (different clutch diameters, the presence or absence of a dual-mass flywheel, or the number of gear ratios). Thus, the different power variants result from different engine control programs implemented by the ECU.

Changing engine control strategies and thus providing different power variants is not only the domain of the latest generation of vehicles. Such solutions were already used on a smaller scale in the previous generation, as exemplified by the 1.6 TDI engine used in the VW Caddy or the 1.3 MJ in the Fiat Scudo.

The ability to change parameters in the ECU makes it so that it is not only the manufacturer, at the stage of designing and adapting the engine to meet certain emission standards, who has the ability to influence the setpoints and shape the torque and power output characteristics, but anyone with the appropriate equipment that allows reading and writing the microcontroller's memory [7].

Currently, there are many programmers on the market that allow ECU programming. Among the most popular are Flex – Magic Motorsport, Autotuner, Dfox – DFB Technologies and KESS 3 – Alientech. The listed devices allow programming using non-invasive methods that do not require physical access to the microcontroller, i.e., through the OBD diagnostic connector or directly through the engine controller connector (BENCH), and invasive methods that, depending on the microcontroller chip or hardware architecture used in the engine controller, require physical access to the PCB and microcontroller leads (BOOT).

Almost all engine controllers manufactured in recent years support programming via the OBD connector, as this is how dealers deliver software patches via service diagnostic interfaces [13]. Manufacturers of stand-alone programmers use modified instructions in factory communication protocols that allow them to skip checking the digital signature of the controller's batch file, thereby allowing the controller to be programmed with any modified batch [11].

Simplifying the programming procedure for connecting the appropriate device to the OBD port has made it easier to access the controller's memory, speeding up reverse engineering and allowing experimentation with parameter modifications in the ECU. This has increased knowledge of

engine control algorithms, demonstrated the potential to change engine performance by modifying the control program, and subsequently increased the popularity of software-based engine performance enhancement services called chiptuning.

At present, there is no data to say unequivocally how many cars in operation in Poland have modified software. However, given the dynamics of the chiptuning market and the growing number of workshops offering such services, it can be concluded that the number of vehicles with modified engine controller software may be significant and continue to increase.

Given that cars are homologated with specific engine control software, any change to that software could result in the vehicle losing its homologation and, consequently, being unable to legally operate on public roads. Considering a situation in which the controller is modified to increase torque and maximum power, it can be concluded that this undoubtedly also changes the emissions of harmful compounds generated by the engine, which is one of the standardized parameters. Given the very small number of scientific articles describing this issue, it is worthwhile to better understand the relationship between modifications to vehicle engine controllers and emissions of harmful compounds.

2. Research test benchmark

A test stand consisting of an AVL dynamic engine dynamometer and a Fiat 1.3 MultiJet compression-ignition engine mounted on it was used for the study.

The dynamic dynamometer is a three-phase asynchronous electric brake (AVL DynoRoad type) with a maximum power of 120 kW and braking torque of up to 509 Nm, capable of operating over the full range of motor speeds (0–8000 rpm). The use of an electric machine enables loading the engine (converting its mechanical energy into electrical energy returned to the grid) and conducting tests under both steady-state and dynamic engine loads. The engine mounted on the bench is a four-cylinder turbocharged unit with a displacement of 1.3 dm³, equipped with a common-rail direct injection system and meeting the Euro 6 emission standard (maximum power of about 70 kW, torque up to 200 Nm) [14, 20]. The high-pressure common-rail fuel system is fed directly from an AVL weighing fuel gauge [5]. The bench also has a forced-circulation liquid-cooling system, a radiator and temperature control of the liquid and oil. The engine uses its factory Magneti Marelli MJD9 ECU (Fig. 1), which has been suitably adapted for dynamometer operation (including an interface for remote control of operating parameters and data reading via OBD interface) [3]. The bench has built-in safety features – including guards for moving parts and exhaust extraction systems – and the test software monitors critical parameters (such as oil pressure, maximum rpm or torque) [17], preventing the exceedance of limit values that could threaten engine or dynamometer failure.

Advanced AVL PUMA software is used to control the operation of the bench – it is a dynamometer automation system that allows load setting, test run control and real-time measurement data acquisition. The dynamometer software includes modules for simulating vehicle operation;

this allows mapping on a stationary bench the real conditions of engine operation in a vehicle, including the realization of standard driving tests (such as the European NEDC cycle or other homologation procedures) [4] and freely definable waveforms of dynamic loads. Using built-in algorithms and physical models, it is possible, among other things, to simulate the vehicle's driving resistances (mass, aerodynamics, rolling resistance), the operation of the power transmission system, and even to interfere with the way gears are shifted – thanks to this, the workstation acts as a "vehicle simulator" for the tested engine [8, 12]. The AVL PUMA program interfaces with the engine control system (ECU) via CAN/OBD diagnostic interfaces (among others, the SIRIUS CAN interface is used, which allows reading and writing parameters from the CAN bus online) [6, 19].



Fig. 1. Magneti Marelli MJD9 ECU for 1.3 MultiJet

The communication process of the Autotuner device used in the tests with the MJD9 ECU, which controls the engine under test, varies depending on the selected mode, but some elements remain common. Reading the controller begins with identifying the ECU. Autotuner automatically detects the controller and software version, and checks if a VR file is available in the online database. If the file is not in the database, reading the memory through the OBD connector may take a long time, or even be impossible if the controller is locked. In this case, it is necessary to read the flash memory using BENCH or BOOT mode. In addition, OBD mode reads only the calibration (map) area, not the entire low-level batch – so the file read by OBD is sometimes smaller than the physical memory capacity of the microcontroller. Importantly, when reading OBD, Autotuner does not create a full memory backup, including dflash and iflash memory. In bench or boot modes, the device can perform a full dump of ECU memory. Autotuner writes a so-called backup file (ZIP) containing all memory segments (iFlash/dFlash, etc.) of the controller. Such a full readout allows EEPROM data to be preserved (e.g. immobilizer codes, injector configurations, DTC errors, DPF counters, etc.), which is important especially for safety purposes and possible restoration of the factory state. It is recommended to backup the ECU in BENCH or BOOT mode before modification to protect any unique configuration data.

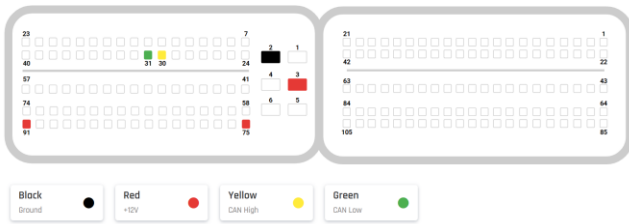


Fig. 2. Autotuner BENCH connection pinout for Magneti Marelli MJD9 ECU

Reading the memory of the microcontroller of the tested engine was performed with the Autotuner equipment using BENCH mode. This equipment allows programming the MJD9 controller through the OBD-II diagnostic connector, which is brought out on the dynamometer bench near the control panel. Reading and writing in this mode is done through the standard CAN diagnostic bus using the UDS protocol. This is the most convenient method: you just need to connect the Autotuner interface to the OBD-II port, and the device will read the microcontroller's memory and let you upload the modified map. The OBD mode is usually used to quickly read/write engine calibration as part of standard modifications, as long as the controller allows it. The MJD9 ECU used in the engine under test is on the list of those supported in OBD mode, but due to the adaptation of the system to the test bench, there were unidentified communication problems and it was not possible to read the memory using this method. For some newer controllers (e.g., Marelli MJD9DF version HW002), Autotuner downloads VR files after the driver identification process, but the file suitable for the tested driver was not on the server. In practice, when tuning the MJD9 for the first time, there is a need to read the ECU in order to unlock it using the BENCH or BOOT mode (the file read by this method can be sent to the Autotuner manufacturer for inclusion in the online database, where it will be available to other users in the future) [15, 16]. After such an operation, OBD can already perform further modifications. In the case in question, since the reading through the OBD connector failed and there was no corresponding VR file on the server, it was necessary to read the contents of the memory in BENCH mode by connecting the appropriate wires directly to the driver's connector. A diagram showing how to connect the appropriate wires to the corresponding pins of the controller is shown in Fig. 2. For the reading, it is necessary to connect the power supply (12V and GND), the two wires corresponding to the CAN-Low and CAN-High lines, and the wire supplying power to the controller to simulate that the ignition is turned on. This reads the entire contents of the microcontroller's memory (full backup), including both the iFlash area (where calibrations are stored) and the dFlash area (where settings and variable parameters are stored).

3. Methods

The read-out file containing the calibration was sent to one of Europe's leading tuning companies, offering remote file modification services. The order involved a typical power increase referred to in the service catalog as STAGE 1. It was decided to outsource the modification to replicate real-world conditions as closely as possible, with the tuning

company performing it on an actual car. The modified file was analyzed in StageX software, which allows comparison of changes in calibration files. The software has mappacks defined for a specific driver type. By mappack is meant a special set of data describing the structure of one-, two- and three-dimensional characteristics stored in the memory of the controller. This data describes the location of a given characteristic in memory, its size-axis description, and appropriately scaled values. The mappack categories used during the calibration analysis are listed in Table 2.

After importing the original and modified ECU files into StageX, the software identifies calibration groups in which differences were detected. These groups are marked in orange in Fig. 3. Note that the engine torque limiter maps, engine desired torque maps, transmission torque limiter, injection system and boost pressure have been modified.

Table 2. StageX software mappack category with description

Category	Description
Air control	Airflow management maps, such as through the EGR throttle or MAF.
Engine torque limiters	Torque limits – e.g., temperature-dependent, gear-dependent, ESP-dependent.
Engine torque request	Torque demand from the gas pedal, cruise control, etc.
Exhaust gas temperature EGT	Monitoring and limiting exhaust gas temperatures – for example, for DPF or turbo protection.
Gearbox torque limiter	Torque limiter due to transmission.
Idle speed RPM	Idle speed – maps that set the resting speed.
Injection system	A wide range of injection maps: times, number of doses, temperature corrections, etc.
Rail pressure	Fuel pressure maps in the fuel rail (common rail).
Smoke limitation	Limits the amount of fuel to not exceed emissions (MAF or MAP sensor).
Start of SOI injection	Injection advance maps - key to performance, combustion and emissions
Turbo boost pressure	Target charge pressure maps
Turbo boost control	Maps controlling the turbo valve/actuator (e.g., N75, VGT, WG).



Fig. 3. Mappack category with difference sign for MJD9DF ECU in StageX software

Engine speed /		RPM							
		750	1000	1450	2100	2500	3200	3600	4000
0		200.0	200.0	200.0	200.0	200.0	200.0	200.0	200.0

Fig. 4. Example of engine torque limiter maps modified in MJD9 ECU

Engine speed /																
RPM	1000	1250	1500	1750	2000	2250	2500	3000	3250	3500	3750	4000	4500	4750	5200	5700
0	135.0	180.0	258.5	260.7	261.5	263.9	263.6	266.2	256.5	243.8	230.0	213.7	159.1	146.2	126.0	0.0

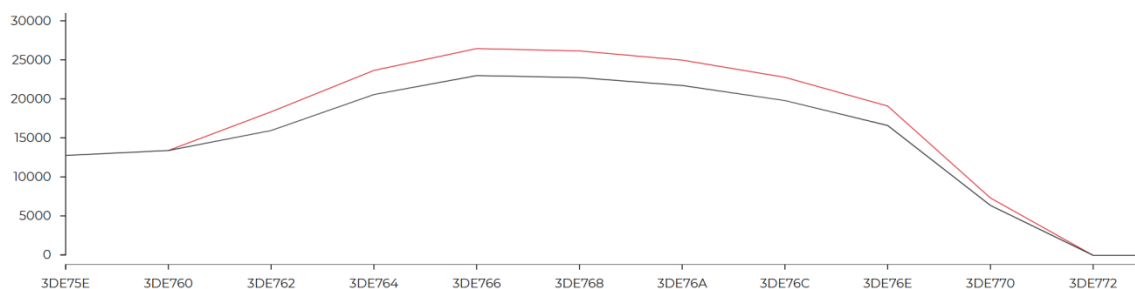


Fig. 5. Main engine torque limiter characteristic modified in MJD9 ECU shown in characteristic table and visualized in raw flash file

Engine speed / Pedal position		RPM								
		600	800	1250	1750	2250	2750	4000	5200	5700
0	%	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
10	%	17.0	16.6	15.1	13.1	11.1	1.3	0.0	0.0	0.0
20	%	45.0	43.3	40.8	36.0	29.9	18.4	3.7	0.0	0.0
30	%	69.8	66.8	64.3	60.8	56.1	45.2	24.8	9.4	0.0
40	%	89.0	88.7	88.1	86.5	81.9	71.9	49.8	29.1	0.0
50	%	100.0	112.5	116.2	117.2	115.0	102.4	74.8	49.8	0.0
70	%	122.4	196.8	197.9	200.5	196.3	180.2	145.6	109.4	0.0
90	%	124.8	208.8	262.1	267.1	265.7	249.4	196.8	163.2	0.0
100	%	126.0	210.0	302.4	312.0	312.0	312.0	225.9	186.4	0.0

Fig. 6. Engine torque request depending on accelerator pedal position and engine speed, modified in the MJD9 ECU

RPM		Engine speed / Internal torque															
		700	800	1000	1250	1500	1750	2000	2250	2500	2750	3000	3500	4000	4500	5000	5500
0.0	Nm	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
20.0		4.88	4.60	4.34	4.04	3.50	3.50	3.52	3.80	4.24	4.26	4.46	4.50	4.52	4.54	4.70	5.00
30.0		7.30	7.06	6.24	5.72	5.08	5.10	5.12	5.46	5.88	5.92	6.30	6.74	6.76	7.06	7.42	8.46
40.0		9.48	9.36	8.30	7.56	6.96	6.88	6.98	7.18	7.46	7.48	7.90	8.44	8.46	8.48	8.50	9.02
50.0		11.60	11.58	10.66	9.56	8.74	8.70	8.78	8.88	9.02	9.10	9.74	10.22	10.24	10.26	10.60	11.44
60.0		13.50	13.42	12.94	11.66	10.78	10.62	10.68	10.72	10.74	10.76	11.46	12.02	12.04	12.06	12.40	13.50
70.0		15.66	15.62	14.72	13.70	12.86	12.68	12.58	12.56	12.64	12.80	13.30	13.52	13.54	13.56	14.34	15.70
80.0		19.10	18.34	17.44	15.92	15.12	14.50	14.02	14.36	14.80	14.92	15.14	15.16	15.18	15.54	16.78	18.32
100.0		28.20	25.40	23.32	20.98	19.72	18.94	18.30	18.78	19.10	19.12	19.22	19.24	20.12	21.32	22.58	23.68
120.0		37.80	34.80	31.86	27.62	26.12	25.08	24.06	24.54	25.00	25.02	25.16	25.16	26.64	28.42	29.62	30.66
140.0		41.08	41.08	41.08	36.64	30.62	29.40	28.80	28.80	29.08	29.08	29.10	29.32	31.50	33.60	36.00	37.70
160.0		47.18	47.18	47.18	41.92	38.62	37.10	34.20	32.76	32.70	31.46	31.60	33.60	36.64	39.80	43.44	45.42
180.0		52.56	52.56	52.56	50.86	43.44	41.38	40.08	39.46	39.24	38.98	39.08	40.06	42.72	47.38	50.30	53.50
200.0		58.14	58.14	58.14	54.26	49.12	47.18	45.92	45.94	46.04	46.04	46.12	48.24	50.50	53.60	59.02	62.64
220.0		67.88	67.88	67.88	58.62	55.02	52.58	50.70	49.40	49.78	51.38	51.40	51.42	53.40	58.22	59.64	62.66
240.0		83.52	83.52	83.52	67.12	58.10	55.12	54.82	54.52	54.72	54.74	57.02	60.74	60.78	61.26	61.56	62.68

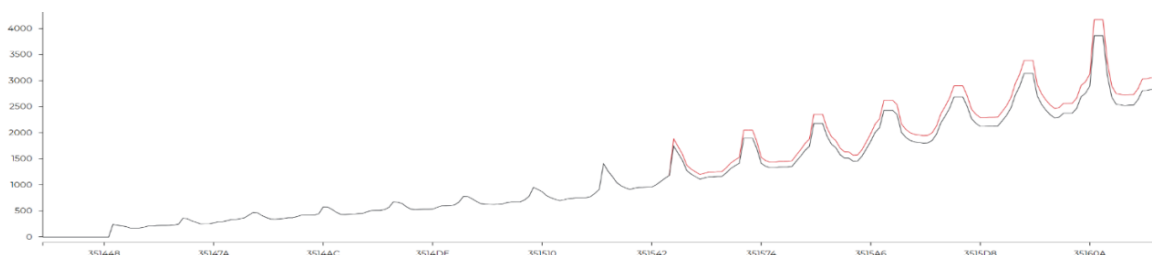


Fig. 7. Example of engine torque to injection quantity conversion maps modified in MJD9 ECU shown in the characteristic table and visualized in raw

One of the modified maps was the torque limiter characteristic. In the modified calibration, the torque limit at 750–1450 rpm was increased to 200 Nm, as shown in Fig. 4.

The main torque limiter, which shapes the engine torque characteristic, was modified over the 1250–4000 rpm range. The modified values of the main torque limiter are presented in Fig. 5.

Calibrations that limit the maximum fuel dosage relative to engine temperature, among other things, allow shaping the course of thermodynamic processes and managing the thermal management of the engine. The modified file also includes changes in the scope of these characteristics. The maximum fuel quantity map was increased by approximately 25% across the entire analyzed range, as shown in Fig. 8. The factory characteristics allow the highest doses in the temperature range from -30°C to 115°C ; above this temperature, the maximum values are reduced to protect the engine from overheating. In the modified file, the values above this temperature have been modified by the same value.

The desired torque map was also modified. The changes mainly affected the values at high accelerator pedal positions, as shown in Fig. 6. The values at the maximum accelerator pedal setting were changed. The maximum desired torque in the range shown was increased by 20%.

The engine calibration modifications carried out included, for the most part, characteristics from the "Injection System" group. This group primarily contains a set of calibrations that specify the maximum fuel dose as a function of speed, a characteristic that converts torque to fuel dose based on speed, and an injection duration characteristic that specifies the injector opening time as a function of fuel pressure and injection dose. In the modified file, this group accounted for the largest number of changes, mainly because the aforementioned characteristics, in a slightly modified form, occur repeatedly, and the algorithm uses different versions for different engine operating states (e.g., depending on DPF filling, DPF regeneration, etc.).

The torque-to-fuel dose conversion maps were also modified. An example of such a modification is presented in Fig. 7. The fuel dose values corresponding to torque values in the range of 120–240 Nm were increased by approximately 8%.

Another characteristic that has been modified within the "Injection system" group is Injection Duration. The map includes an injector flow calibration that determines the injector opening time required to inject a specific volume of fuel, depending on the fuel pressure in the CR rail. The injection duration map was modified for fuel doses from 25 mm^3 to 55 mm^3 and rail pressures from 600 bar to 1800 bar, as shown in Fig. 9. The opening times have been increased by 8%, with the consequence that more fuel is physically injected into the combustion chamber than the parameters processed by the engine controller would suggest.

Changing the parameters primarily responsible for the amount of fuel delivered to the combustion chamber is not the only modification that was made to the calibration file under review. The last set of characteristics in which modifications were made includes a group related to boost pressure control. In the "Turbo boost pressure" tab of the

StageX software, the following maps have been identified that have been modified relative to the original:

- turbo boost pressure request – containing characteristics of the desired boost pressure in different engine modes
- turbo boost pressure limit – containing the limits of boost pressure relative to speed
- turbo boost pressure limit by ATM – containing limits of boost pressure relative to atmospheric pressure.

The tuner's modification to the boost pressure request map increased the boost pressure setpoint over the engine speed range from 1500 to 4500 rpm. This characteristic defines the requested boost pressure as a function of engine speed and load, expressed by the injected fuel quantity. As shown in Fig. 10, the boost pressure setpoint was increased by approximately 6% for loads above about 150 Nm. In addition, the characteristics limiting the value of boost pressure have been modified so that the permissible pressure values therein are higher than the desired pressure values. Similarly, the maps limiting boost pressure relative to atmospheric pressure have been modified. The injection duration and boost pressure request maps discussed above are presented together in Fig. 9 and Fig. 10.

The effect of software modifications on environmental parameters was determined using PEMS equipment, specifically the AXION R/S mobile exhaust gas analyzer. The measurement range and methods used are shown in Table 3.

Table 3. AXION R/S analyzer measurement characteristics [2]

Substance	Measurement range	Relative measurement accuracy	Resolution	Measurement method
HC	0–4000 ppm	$\pm 3\%$	1 ppm	NDIR
CO	0–10%	$\pm 3\%$	0.01 vol.%	NDIR
CO ₂	0–16%	$\pm 4\%$	0.01 vol.%	NDIR
NO	0–4000 ppm	$\pm 3\%$	1 ppm	E-chem
PM	0–300 mg/m ³	$\pm 2\%$	0.01 mg/m ³	Laser Scatter

The emission measurements were performed as a single comparative test series for the factory and modified ECU calibration. Therefore, the repeatability of the complete test procedure could not be determined experimentally, and no standard deviation for repeated test runs was calculated [18]. For this reason, the results should not be interpreted as a full statistical assessment of emission variability, but rather as a comparative case study showing the direction and scale of emission changes resulting from ECU calibration modifications.

The measurement uncertainty was assessed on the basis of the declared accuracy of the AXION R/S analyzer. According to the instrument specification, the relative measurement accuracy was $\pm 3\%$ for HC, CO and NO, $\pm 4\%$ for CO₂ and $\pm 2\%$ for PM. Since both calibrations were tested using the same engine, test bench, analyzer, sampling configuration and NEDC-based operating cycle, the systematic component of the instrumental error affected both measurements similarly [9, 10].

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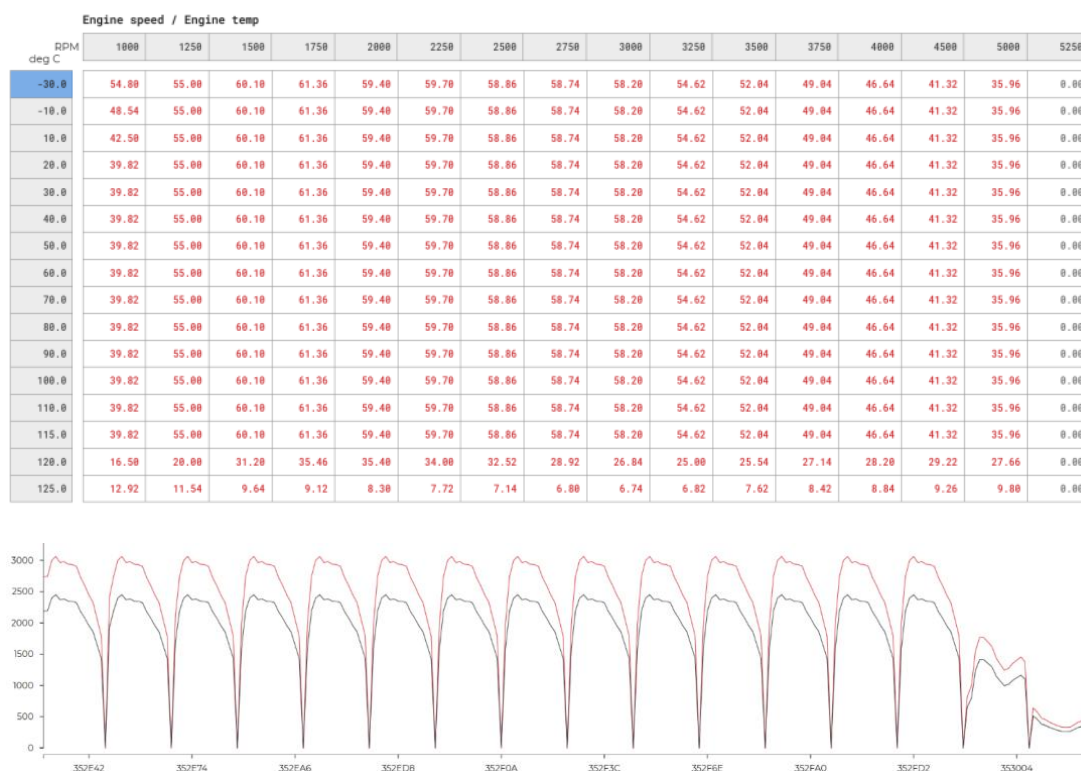


Fig. 8. Example of maximum fuel quantity injected by engine temperature characteristic modified in MJD9 ECU shown in characteristic table and visualized in raw flash file

Consequently, the comparison was based primarily on relative differences between the factory and modified calibration. If the observed change in a given emission component exceeded the declared accuracy of the analyzer, it was interpreted as an indication of a real change in emission behavior. Nevertheless, the lack of repeated test runs is a limitation of the study and should be taken into account when interpreting the results. Future work should include repeated test cycles to determine repeatability, standard deviation and statistical significance of the observed differences.

Using the capabilities of the dynamic dynamometer and AVL PUMA software, the NEDC test was reproduced, including the Urban and Extra-Urban phases. Although WLTP and RDE procedures are currently used for the emission assessment of light-duty vehicles, the NEDC cycle was selected in this study to ensure repeatable and directly comparable engine speed and load conditions on the dynamic engine test bench. The aim of the study was not to reproduce the complete vehicle type-approval procedure, but to isolate the influence of ECU calibration modification under identical operating conditions. For this reason, the results obtained should be interpreted as a comparative assessment of the impact of the Stage 1 ECU modification, rather than as a direct homologation result.

The lack of WLTP- and RDE-based measurements is a limitation of the present study. Future work should include WLTP-based test-bench procedures and PEMS-based RDE measurements to assess the effect of ECU modification under more representative transient and real-driving conditions.

The modified injection duration and boost pressure request maps are shown in Fig. 9 and Fig. 10. These figures are placed after the measurement equipment characteristics due to formatting and readability constraints.

4. Results and discussion

4.1. Emission results for the complete NEDC cycle

Comparative tests of the Fiat 1.3 MultiJet engine on a dynamic engine dynamometer before and after ECU software modification showed clear changes in the emission of toxic exhaust components over the complete NEDC cycle. The specific emission values measured over the complete NEDC cycle are presented in Fig. 11. After the Stage 1 modification, an increase in NO_x , PM and HC emissions was observed in comparison with the factory calibration. This indicates that performance-oriented ECU modification deteriorated the emission characteristics of the tested engine under the analysed operating conditions.

The increase in emissions was directly related to the changes introduced into the calibration file. The fuel dose as a function of desired torque was increased by approximately 8%, and the injection duration was increased by a similar value. As a result, more fuel was delivered to the combustion chamber. This increased the combustion energy and local combustion temperature, which promoted NO_x formation. At the same time, local oxygen deficiency and transient enrichment contributed to higher PM and HC emissions. Although the boost pressure setpoint was also increased by approximately 6% under high-load conditions, this change only partially compensated for the increased fuel quantity.

Fuel rail pressure / Injection quantity																	
mm ³ /inj	bar	0.0	100.0	200.0	250.0	300.0	400.0	500.0	600.0	700.0	800.0	900.0	1000.0	1200.0	1400.0	1600.0	1800.0
	0.00	0.00	170.00	170.00	170.00	170.00	160.00	150.00	130.00	130.00	130.00	130.00	130.00	130.00	130.00	130.00	130.00
0.20	0.00	0.00	559.00	327.00	304.00	292.00	278.00	271.00	267.00	264.00	262.00	261.00	260.00	258.00	257.00	256.00	255.00
0.50	0.00	0.00	728.00	412.00	359.00	333.00	307.00	294.00	287.00	282.00	279.00	276.00	274.00	272.00	270.00	268.00	267.00
1.00	0.00	0.00	859.00	525.00	437.00	393.00	349.00	328.00	316.00	307.00	302.00	297.00	294.00	290.00	286.00	284.00	282.00
1.50	0.00	0.00	990.00	561.00	484.00	438.00	388.00	360.00	343.00	332.00	324.00	317.00	313.00	305.00	301.00	297.00	294.00
2.00	0.00	0.00	1111.00	603.00	525.00	476.00	419.00	387.00	367.00	353.00	342.00	335.00	328.00	319.00	313.00	308.00	305.00
20.00	0.00	0.00	2895.00	1620.00	1366.00	1196.00	984.00	856.00	771.00	711.00	665.00	630.00	602.00	559.00	529.00	506.00	489.00
25.00	0.00	0.00	3296.00	1847.00	1555.00	1359.00	1114.00	967.00	885.00	813.00	759.00	717.00	684.00	633.00	597.00	570.00	548.00
30.00	0.00	0.00	3632.00	2048.00	1727.00	1512.00	1241.00	1078.00	987.00	908.00	890.00	841.00	801.00	742.00	700.00	668.00	643.00
35.00	0.00	0.00	4000.00	2302.00	1919.00	1668.00	1359.00	1176.00	1077.00	987.00	970.00	916.00	875.00	812.00	768.00	735.00	709.00
40.00	0.00	0.00	4000.00	2358.00	2008.00	1769.00	1464.00	1276.00	1171.00	1078.00	1056.00	997.00	950.00	879.00	827.00	789.00	756.00
45.00	0.00	0.00	4000.00	2531.00	2162.00	1908.00	1581.00	1378.00	1264.00	1161.00	1137.00	1073.00	1020.00	941.00	885.00	841.00	808.00
50.00	0.00	0.00	4000.00	2688.00	2311.00	2048.00	1704.00	1487.00	1363.00	1251.00	1221.00	1150.00	1092.00	1003.00	937.00	889.00	850.00
55.00	0.00	0.00	4000.00	2867.00	2470.00	2191.00	1823.00	1590.00	1456.00	1334.00	1301.00	1223.00	1159.00	1061.00	989.00	935.00	892.00

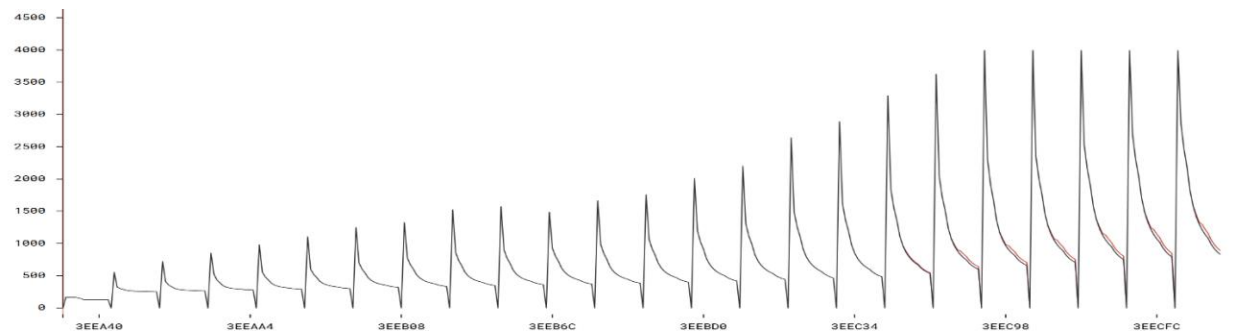


Fig. 9. Injection duration characteristics modified in MJD9 ECU shown in the characteristic table and visualized in the raw flash file

Engine speed / Injection quantity																	
RPM	mg/hub	1000	1250	1500	1750	2000	2250	2500	2750	3000	3250	3500	3750	4000	4250	4500	4750
	5.00	910.0	940.0	960.0	1010.0	1070.0	1110.0	1175.0	1225.0	1265.0	1295.0	1315.0	1335.0	1340.0	1340.0	1315.0	1265.0
8.00	960.0	990.0	990.0	1040.0	1100.0	1165.0	1225.0	1280.0	1315.0	1350.0	1370.0	1390.0	1390.0	1395.0	1395.0	1365.0	1315.0
11.00	1000.0	1020.0	1040.0	1120.0	1170.0	1225.0	1265.0	1325.0	1365.0	1405.0	1435.0	1455.0	1455.0	1455.0	1455.0	1440.0	1395.0
14.00	1060.0	1080.0	1120.0	1180.0	1250.0	1290.0	1325.0	1380.0	1425.0	1465.0	1495.0	1515.0	1515.0	1515.0	1515.0	1495.0	1445.0
17.00	1200.0	1215.0	1220.0	1260.0	1320.0	1370.0	1415.0	1465.0	1505.0	1545.0	1565.0	1585.0	1585.0	1585.0	1575.0	1555.0	1515.0
20.00	1270.0	1290.0	1310.0	1355.0	1405.0	1445.0	1490.0	1545.0	1585.0	1615.0	1645.0	1655.0	1655.0	1655.0	1645.0	1615.0	1560.0
23.00	1360.0	1400.0	1415.0	1475.0	1515.0	1565.0	1615.0	1655.0	1690.0	1720.0	1730.0	1740.0	1720.0	1690.0	1670.0	1620.0	1620.0
26.00	1465.0	1500.0	1525.0	1595.0	1655.0	1710.0	1760.0	1790.0	1810.0	1810.0	1810.0	1810.0	1780.0	1750.0	1715.0	1700.0	1700.0
29.00	1560.0	1600.0	1645.0	1710.0	1770.0	1820.0	1860.0	1880.0	1890.0	1875.0	1875.0	1875.0	1840.0	1810.0	1785.0	1780.0	1780.0
32.00	1600.0	1720.0	1770.0	1820.0	1880.0	1920.0	1960.0	1970.0	1950.0	1935.0	1935.0	1925.0	1895.0	1870.0	1845.0	1820.0	1820.0
35.00	1600.0	1840.0	2006.0	2132.0	2174.0	2205.0	2237.0	2126.0	2100.0	2090.0	2084.0	2069.0	2042.0	2021.0	1969.0	1820.0	1820.0
38.00	1600.0	1900.0	2153.0	2279.0	2321.0	2342.0	2352.0	2168.0	2147.0	2137.0	2132.0	2111.0	2090.0	2032.0	1969.0	1820.0	1820.0
41.00	1600.0	1940.0	2268.0	2363.0	2363.0	2363.0	2363.0	2285.0	2189.0	2179.0	2174.0	2147.0	2090.0	2032.0	1969.0	1820.0	1820.0
43.00	1600.0	2000.0	2363.0	2363.0	2363.0	2363.0	2363.0	2237.0	2226.0	2216.0	2210.0	2147.0	2090.0	2032.0	1969.0	1820.0	1820.0
45.00	1600.0	2040.0	2363.0	2363.0	2363.0	2363.0	2363.0	2263.0	2258.0	2231.0	2210.0	2147.0	2090.0	2032.0	1969.0	1820.0	1820.0
50.00	1600.0	2100.0	2363.0	2363.0	2363.0	2363.0	2363.0	2310.0	2258.0	2231.0	2210.0	2147.0	2090.0	2032.0	1969.0	1820.0	1820.0

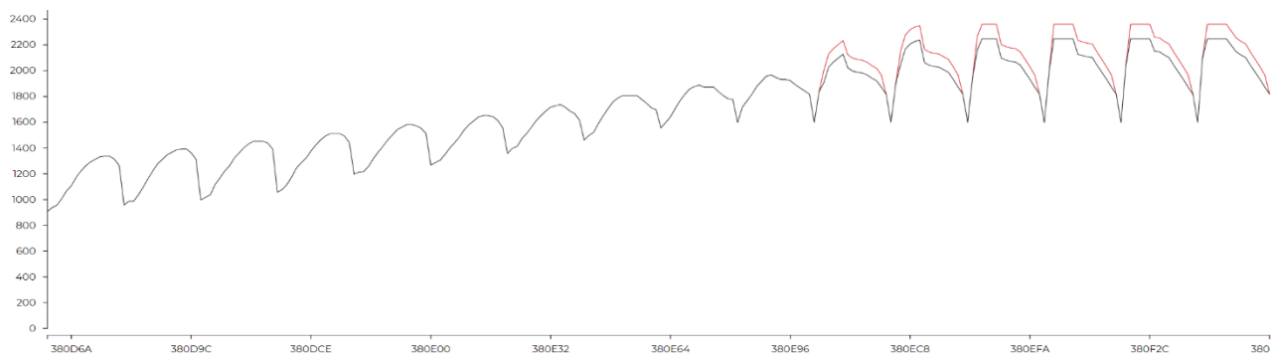


Fig. 10. Request boost pressure characteristics modified in MJD9 ECU shown in the characteristic table and visualized in the raw flash file

4.2. Emission results for the urban phase

The Urban phase of the NEDC cycle is characterized by low vehicle speeds, frequent stops and repeated acceleration events. Under these conditions, an increase in exhaust emissions was also observed after uploading the Stage 1 ECU modification. The specific emission values recorded during the Urban phase are presented in Fig. 12. However, the differences between the factory and modified calibration were less pronounced than in the Extra-Urban phase, because the engine operated less frequently under high-load conditions.

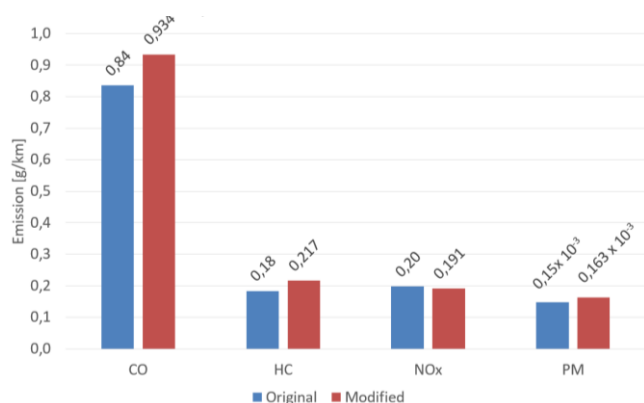


Fig. 11. Specific emission in the complete NEDC cycle for the 1.3 MultiJet Euro 6 engine with factory and modified ECU calibration

The increase in PM and HC emissions in the Urban phase can be associated with transient enrichment during acceleration. Under low-speed and rapidly changing load conditions, the required boost pressure is not always achieved immediately due to the dynamic response of the turbocharger. Consequently, the additional fuel resulting from the modified injection maps may not be fully burned, promoting soot and unburned hydrocarbon formation.

NO_x emissions in the Urban phase also increased, although to a lesser extent than in the high-load part of the cycle. This may be due to the exhaust gas recirculation system remaining active during low-load urban operation, partially limiting NO_x formation. Nevertheless, the increased fuel quantity and higher local combustion temperatures led to higher NO_x emissions than under factory calibration.

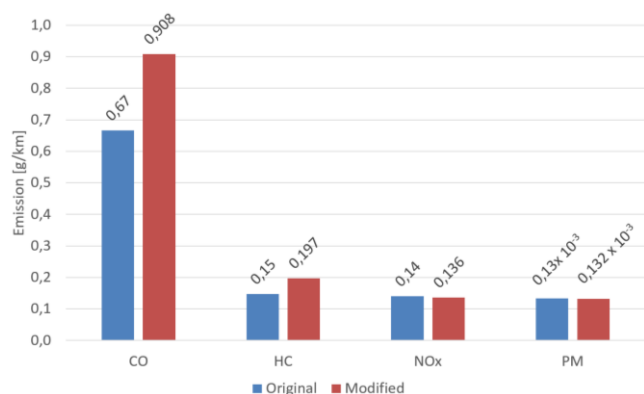


Fig. 12. Specific emission in the Urban phase of the NEDC test for the 1.3 MultiJet Euro 6 engine with factory and modified ECU calibration

4.3. Emission results for the extra-urban phase

The specific emission values obtained for the Extra-Urban phase are presented in Fig. 13. In this part of the NEDC cycle, the differences between the factory and modified calibration were more pronounced due to higher engine speeds and loads. After uploading the modified software, the engine generated significantly higher NO_x and PM emissions during high-speed operation.

The increase in NO_x emissions was mainly associated with higher combustion temperatures resulting from increased fuel delivery and higher boost pressure under high-load conditions. The modified torque limiters and torque-to-fuel quantity maps allowed higher fuel quantities in the engine speed range where the Extra-Urban phase requires higher torque. In addition, the increase in maximum fuel quantity limits over the temperature range reduced the protective effect of factory calibration strategies intended to limit thermal load.

PM emissions in the extra-urban phase also increased due to the extended injection duration and increased fuel quantity. Although the boost pressure setpoint was increased to provide more air for combustion, the relative mixture became richer than in the factory calibration, especially during transient load changes. This promoted soot formation. The DPF present in the Euro 6 engine retains most particulate matter, but increased engine-out soot production may lead to faster filter loading. Since the DPF loading state was not recorded during the tests, this should be considered a limitation of the study.

5. Conclusions

Analysis of the exhaust emission results for the Fiat 1.3 MultiJet engine, comparing the factory calibration with the modified ECU software, showed a marked increase in NO_x, PM and HC emissions. These changes were particularly evident in the extra-urban phase of the NEDC cycle, where the engine operates at higher speeds and loads. The increase in emissions was a direct result of changes to the calibration maps, which included an increase in torque limits, fuel dosage, injection duration and desired boost pressure.

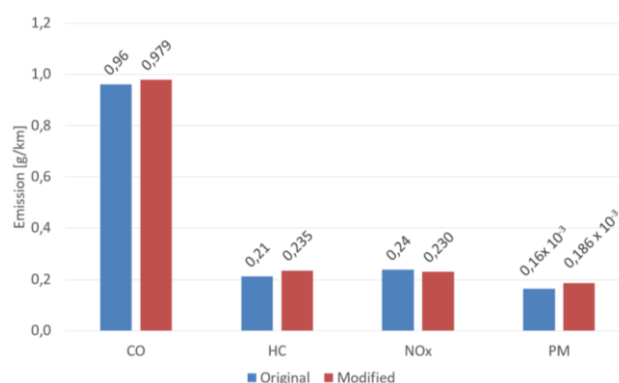


Fig. 13. Specific emission in the Extra-Urban phase of the NEDC test for the 1.3 MultiJet Euro 6 engine with factory and modified ECU calibration

It is worth noting, however, that the modification was performed by an independent tuning company, and there are currently no standardized guidelines or procedures that specify how to safely and sustainably perform this type of

modification. As a result, the results of calibration can vary significantly depending on the level of expertise, experience and tools used by the contractor in question. There is a real chance that other companies could achieve better results. At the same time, less knowledgeable or unskilled operators could cause even greater deterioration of combustion and emission characteristics and even jeopardize the life of the drive unit.

Importantly, some irregularities and simplifications are noticeable in the analyzed modification, indicating a lack of in-depth analysis of the impact of changes on engine operation. In particular, increasing the values in the maps by a fixed percentage (e.g. +8% for injection timing, +25% for maximum doses) does not take into account the actual technical capabilities of the equipment (injectors, turbocharger, cooling system or DPF). Such an approach suggests that the modification was carried out in a template fashion, without individual adjustment to a specific drive unit. Failure to consider temperature limitations, DPF regeneration conditions, mechanical tolerances or controller protection strate-

gies can result not only in increased emissions in the long run but also in the risk of engine or component damage.

In conclusion, the results presented underscore the need for caution when deciding on chiptuning and, above all, signal that, as the popularity of such services increases, it will be necessary to examine more closely the environmental impact of modified vehicles.

Without proper procedures for validating modifications, analyzing their impact on the entire powertrain, and accounting for environmental consequences, such interference can do more harm than good. Given the lack of industry regulation, it seems reasonable to develop quality standards for independent ECU modification services, taking into account operational safety and environmental compatibility. Since the measurements were performed as a single comparative test series, the results should be interpreted as a case study indicating the direction and approximate scale of emission changes, rather than as a full statistical assessment of ECU modification effects.

Nomenclature

DI direct injection
ECU engine control unit
HC hydrocarbons

NO_x nitrogen oxides
NEDC New European Driving Cycle
PM particulate matter

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A combined experimental and CFD analysis of jet-crossflow interaction in turbine cooling passages

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This study presents a combined experimental and numerical investigation of the jet-crossflow interaction relevant to gas turbine blade film cooling applications. Four staggered rows of cylindrical cooling jets inclined at 30° are examined on a flat plate subjected to an incompressible crossflow with a mainstream velocity of 2 m·s⁻¹. Experiments were conducted at a fixed blowing ratio of $M = 2$ using Particle Image Velocimetry (PIV) to obtain time-averaged velocity fields and turbulence quantities across six measurement planes. Corresponding Reynolds-Averaged Navier-Stokes (RANS) simulations were performed using the standard $k-\epsilon$ and Shear Stress Transport (SST) turbulence models under identical boundary conditions. Quantitative comparisons of normalized axial and vertical velocity profiles (U/U_e and V/U_e) reveal that both turbulence models overpredict near-wall axial velocities in the jet exit region; however, the SST model shows consistently closer agreement with PIV measurements across streamwise locations ranging from $X/D = -2$ to $X/D = 30$. In the near-field region, the SST model captures jet penetration and mixing trends more accurately, whereas the $k-\epsilon$ model underestimates downstream jet diffusion and lateral spreading. Turbulent kinetic energy distributions further demonstrate that the SST model provides improved prediction of shear-layer turbulence intensity, although both models underestimate the spatial extent of turbulence compared to experimental data. The results indicate that, under the investigated conditions ($M = 2$, $\alpha = 30^\circ$), the SST model offers superior predictive capability for jet-crossflow interaction compared to the standard $k-\epsilon$ model, particularly in regions dominated by strong shear and recirculation. These findings guide RANS turbulence model selection.

Key words: gas turbine blade cooling, film cooling, incompressible flow, jet interaction, turbulence modeling

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1. Introduction

The study centers on the analysis of film cooling performed on a test bench designed to mimic gas turbine blade-cooling scenarios. In the setup, the cooling air and hot air streams are kept at around 22°C and 40°C, respectively. The main aim is to investigate the formation and behavior of the cooling film generated by cold-air injection, which is essential for protecting turbine blades from high-temperature gases. The study is split across two primary parts: a numerical calculation and an experimental investigation.

Extensive literature research shows that significant effort has been put into understanding how film cooling works and how it interacts with the main flow. Among the many characteristics found to be significant are wall curvature, the three-dimensional structure of the outside flow, free-stream turbulence, blowing ratio, compressibility, flow unsteadiness, hole shape and location, and injection angle. Both numerical and experimental investigations have been used to assess these outcomes.

Obtaining a thorough understanding of flow patterns and heat transfer properties within turbine cascades is enabled by Computational Fluid Dynamics (CFD), a powerful tool that provides critical information for the design and optimization of gas turbines. Still, CFD models need to be validated with great care. This covers the selection of suitable turbulence models, the specification of precise inlet conditions, and grid independence tests. Local heat transfer has been shown to be especially sensitive to several varia-

bles related to secondary flows and, therefore, to the precision of physical and numerical modeling.

Under controlled conditions, experimental measurements are published for flat-plate designs, including specific Mach and Reynolds numbers and specified turbulence levels at the leading edge. Two-equation eddy viscosity models, such as $k-\epsilon$ and $k-\omega$, remain widely used from a simulation perspective because they strike a balance between computational efficiency and accuracy. Particularly good at grasping near-wall behavior and transition to turbulence, both of which are essential for gas turbine uses, these models. Most film-cooling simulations reported in the literature are based on Reynolds-averaged Navier-Stokes (RANS) equations, which are time-averaged.

Experimental investigations by Andreopoulos and Rodi [3], Johnston et al. [17], and Cao et al. [9] have highlighted the complexity of the jet's near field, marked by large-scale coherent structures. These include jet shear layer vortices dominating the initial part of the jet, horseshoe vortices wrapping around the jet base, and counter-rotating vortex pairs induced by the momentum impact of the crossflow. The turbulence properties of the jet are much affected by these structures, which cause great jet distortion and the creation of wake vortices. Figure 1 offers a summary of the intricate flow field produced by the interplay between the crossflow and the jet.

For a baseline film cooling configuration, the predictive power of different RANS models is assessed against pertinent experimental data. Although RANS methods are still

commonly used, many recent investigations have employed Large Eddy Simulation (LES) to better understand the complex flow patterns associated with film cooling.

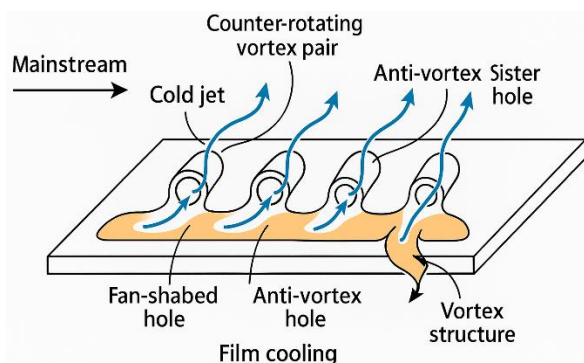


Fig. 1. Flow diagram of a row of jets based on experimental results

Yuan and Street's LES study [31] shows that a round jet discharge into a laminar crossflow is captured by large-eddy simulation. Their analysis effectively captures the coherent vortical structures observed in experimental studies, including the counter-rotating vortex pair (CVP), which emerges from the time-averaged action of sporadic vortical structures. Near the jet, wake vortices are also seen to form. The study also examines how commonly used simple boundary conditions affect simulation results and shows that the simulated mean flow fields match quite well with what is observed in real life. More recently, Leedom [20] examines how well film cooling performs using LES with different shapes for the coolant holes. The study examines discrete holes tilted at 35 degrees with a feeding plenum. The density ratio is 2, and the blowing ratios range from 0.5 to 2.0. Simulations are done for cylindrical holes with length-to-diameter ratios of 1.75 and 3.5, as well as for shaped holes (laterally diffused and console) with a ratio of 3.5, all of which come out into a laminar crossflow at a Reynolds number of about 16,000. Comparison with experimental data shows that the flow field and surface adiabatic effectiveness of the cylindrical holes are indeed correct. Important vortex structures, such as horseshoe, DSSN, and hairpin vortices, are resolved in the simulations. The findings show that jetting has a major impact on cooling performance; shaped holes, especially console holes, provide the best surface cooling. Still, in terms of aerodynamic loss, cylindrical and console holes beat out horizontally distributed ones.

In another study, Balaji et al. [5] examined how pin fins with different diameters and heights affect cooling efficiency at the corners of an aircraft turbine blade tip. Furthermore, Wang et al. [27] conducted a numerical investigation of the influence of four distinct hole layouts on film-cooling effectiveness and particle build-up on a curved surface. They investigated particle deposition using the RNG $k-\epsilon$ turbulence model, along with a discrete-phase model implemented using user-defined functions, to assess capture and impact efficiencies. The research showed that, at a blowing ratio of 1.2, combined fan-shaped and round-to-slot holes enhanced average film-cooling performance by 69.8% and 60.3%, respectively, compared with conventional cylindrical holes. Except for the round-to-slot hole, the

inclination angle between 30° and 45° strongly affected cooling performance. Particle size also influenced capture and impact efficiencies, which at blowing ratios between 0.3 and 0.9 rose, then fell. The fan-shaped and round-to-slot holes had the lowest efficiencies at a 30° tilt angle, whereas for 1 μm particles, the round-to-slot hole had the best.

To assess film-cooling efficiency, Xiaojian He et al. [16] compared the pressure-sensitive paint (PSP) and thermochromic liquid crystal (TLC) approaches. The same wind tunnel was used to test both methods across sixteen scenarios with different mainstream Reynolds numbers (35,000, 45,000, and 57,000), blowing ratios (0.5, 1.0, 1.5, and 2.0), and density ratios (0.91 and 1.44). According to the research, their cooling efficiency measurements showed certain differences, even if both PSP and TLC found an ideal blowing ratio at low to medium Reynolds numbers. Heat transmission in the solid region was thought to be responsible for these differences. Increased heat transfer was observed at high mainstream Reynolds numbers and blowing ratios, but at high Reynolds numbers and low blowing ratios, the impact declined. PSP and TLC results virtually matched under low blowing ratio (0.5) and nitrogen-cooling conditions, suggesting that at low blowing and density ratios, heat conduction effects are small.

Targeting a concave surface resembling the leading edge of a gas turbine blade, Forster and Weigand [14] conducted an experimental and numerical investigation of jet-impingement cooling. The study included nine circular jets, each encircled by three film-cooling holes arranged in a staggered pattern. Experiments at a Reynolds number of 30,000 varied the jet-to-surface separation distance and considered the impacts of crossflow. The study assessed heat transfer distributions utilizing TLC methods, whereas OpenFOAM was used to conduct complementary RANS calculations with the $k-\omega$ SST turbulence model. Crossflow was shown to lower total heat transfer and stagnation point but to produce a more consistent distribution of heat. Shorter jet separation distances also magnified crossflow effects and raised general heat transfer rates. The numerical calculations successfully caught regional flow events and fit well with empirical data. Yao et al. [30], who examined the effects of hole form, also verified this tendency. Their research revealed that non-cylindrical hole shapes provide better cooling effectiveness, particularly at lower blowing ratios ($M = 0.5$).

Using LES, Yao et al. [30] examined whether large turbulent structures formed when the injection ratio, M , was 1.7. The coolant and mainstream flows had the same volume of fluid. Their simulation results showed strong agreement with experimental data, validating the model's reliability.

By using triangular thin plates and a flop arrangement to alter the orientation of cylindrical film-cooling holes, Choi et al. [11] conducted experimental studies to improve film-cooling efficiency. The study examined several rows of 35° angled cylindrical holes in a low-speed wind tunnel at a Reynolds number of 196,167 and blow ratios ranging from 0.3 to 2.0. The authors examined several arrangements using thermographic and flow visualization methods. At a blowing ratio of 1.0, the baseline cylindrical holes had

a cooling effectiveness of about 0.1. However, even in the downstream area, the inclusion of the triangular-shaped thin plate-flop greatly increased film coverage and stopped flow separation. With only an 8% increase in discharge coefficient, this design showed up to a 242.7% improvement in cooling effectiveness over conventional cylindrical holes, making it the best option for gas turbine uses.

Toghraie et al. [25] looked at the dynamic viscosity of a magnetite-based nanofluid ($\text{Fe}_3\text{O}_4/\text{water}$) between 20 and 55°C in another study. The data revealed that while viscosity decreases with increasing temperature, it increases significantly with increasing nanoparticle volume fraction, reaching a maximum improvement of 129.7%. To accurately forecast the viscosity of this nanofluid, a new correlation was proposed.

Esfe and Toghraie [13] also investigated how thermoelectric cooling (TEC) affected the performance of solar stills. According to their research, freshwater output is directly related to the strength of solar radiation. Using TEC on the glass surface of the still boosted freshwater

production by 15% to 62%, therefore showing the promise of this enhancement approach for solar desalination devices. The research by Pouya Barnoon et al. [6] examines how flow behavior and heat transfer in a porous chamber are affected by heating and cooling generators and cryogenic conductors. Using the Finite Volume Method (FVM), the simulations were run with single-phase and two-phase models to forecast heat transfer behavior. The Newtonian, incompressible, and constant working fluid – a water–copper oxide nanofluid is thought of. Several cases with the volume fraction kept constant were studied to evaluate the rate of heat transmission and simplify the distribution. To explain fluid flow and thermal behavior inside the porous medium, the Darcy-Forchheimer model was used. Data revealed that cryogenic conductors improve heat transfer, and heating and cooling generators improve thermal performance. But in particular circumstances, traditional porous containers performed more effectively. Using current thermal control methods, the study's main goal was to maximize heat transmission efficiency.

Table 1. Summary of key studies on flow, heat transfer, and cooling

Authors	Research focus	Methods and tools	Key contributions and findings
Guo et al. [15]	Heat-transfer enhancement in jet impingement with surface protrusions.	CFD-based solver and experimental measurements; comparison of smooth, spherical, ellipsoidal, and teardrop protrusions; parametric analysis of Reynolds number, protrusion height, and spacing.	Ellipsoidal protrusions offer the highest cooling performance; they raise the average Nusselt number by 15% compared to smooth surfaces. Increasing protrusion height by 22% further improved heat transfer without raising pressure losses; lowering protrusion spacing improved cooling with pressure-loss variations under 5%. Suggested a passive augmentation plan for turbine blade cooling with low penalties.
Wang et al. [28]	Effect of rectangular jet aspect ratio (AR) on the jet-in-crossflow (JICF) flow and energy transfer.	Wind tunnel experiments, LES and turbulence analysis, temperature-field analysis, coherent vortex identification.	By delaying the onset of the hanging vortex breakdown, it was shown that increasing AR greatly improves jet penetration (46.4% gain at $X/d = 1$). Recognized the counter-rotating vortex pair (CVP) as a pseudo-sequential vortex structure. Greater AR helps stabilize downstream vortices, reduce temperature swings, accelerate turbulent kinetic energy decay, and thereby boost energy dissipation by 28.8%. The study offers guidance on maximizing jet-orifice design to improve industrial cooling and mixing efficiency.
Chatzistefanou et al. [10]	Performance analysis of a precooled air-breathing propulsion system.	One-dimensional thermodynamic modeling.	Demonstrated the importance of bypass duct modeling for improving engine efficiency and fuel economy.
Jones and Wille [18]	Simulation of a plane jet in cross-flow using LES.	LES with artificial inflow and wall boundary conditions; Smagorinsky-Lilly, one-equation, and dynamic subgrid models.	Mesh adapted to the mean velocity field improves resolution; reasonable agreement with experimental results; minor differences observed between residual-scale models.
Berkache and Rabah [8]	Film cooling analysis in gas turbine blades.	3D simulations in Fluent applied $k-\epsilon$, RSM, and Shear Stress Transport (SST) turbulence models; experimental validation with Particle Image Velocimetry (PIV).	RSM improves predictions over other RANS models; RANS limitations, due to isotropic eddy diffusivity, lead to overpredictions of the centerline cooling velocity.
Kong et al. [19]	Swirling jet flow and heat transfer in the turbine blade leading-edge chamber.	Numerical simulation with varying jet hole offsets (e/d), jet spacing (H/d), and different dimple geometries (SDs, OTDs) at $Re = 30,000$.	A jet hole offset enhances heat transfer by 15%; OTDs in a staggered arrangement yield a 24% improvement in thermal-hydraulic performance; dimples increase the area but reduce the average Nusselt number.
Vicano et al. [26]	Examined the flow characteristics of a jet in crossflow issuing from a 777-shaped cooling hole under different velocity ratios.	PIV, Refractive Index Matching (RIM), NaI-water working fluid, turbulence statistics, vorticity analysis.	Given a thorough experimental characterization of velocity fields, Reynolds stresses, turbulence intensity, and vortex development for velocity ratios ranging from 0.4 to 1.2. Determined the main vertical structures produced by the 777-shaped aperture and measured turbulence growth downstream from the hole exit. The findings provide better knowledge of complex cooling-hole designs for improved jet mixing.
Yang et al. [29]	Temperature uniformity in array jet impingement cooling.	Detailed flow and thermal field analysis under maximum cross-flow; vortex structure investigation.	Identified entrainment and counter-rotating vortex-induced nonuniformities; increasing Reynolds number or reducing spacing improves uniformity; implications for turbine cooling design.

In a linked study, Barnoon et al. [7] examined the cooling behavior, stress distribution, and displacement properties of proton exchange membrane fuel cells (PEMFCs) under natural and forced convection. The goal of the research was to lower operating temperature, thermal stress, and deformation by maximizing the quantity and thickness of bipolar plates. Results showed that forced convection decreased the working temperature by over 30 K, therefore considerably lowering the number and thickness of the plates. Natural cooling could pose thermal risks to the system while speeding up plate heating.

In another study, Baghoolizadeh et al. [4] examined how to best use energy in homes, particularly in air conditioning systems. Building models were simulated employing the EnergyPlus program under several climatic scenarios. Five main design factors were selected, and for each city, 300 Design of Experiments (DOE) points were evaluated. Constructing predictive models from the simulation data involved using Response Surface Methodology (RSM). The outcomes showed a 2% to 16% decrease in related expenses and total energy load. Significant gains were also noted in the Human Heating Comfort and Static Payback Period measures, therefore highlighting the efficiency of the optimization approach.

The objective of this research is to experimentally and numerically investigate the interaction between multiple staggered cooling jets and a turbulent crossflow representative of gas turbine blade film cooling conditions. PIV measurements are employed to quantify the mean velocity fields and turbulence characteristics of four rows of jets inclined at 30° and operating at a blowing ratio of $M = 2$. In parallel, RANS simulations using the standard $k-\epsilon$ and SST turbulence models are performed under identical operating conditions. The study aims to evaluate the ability of these turbulence models to accurately predict jet penetration, lateral spreading, near-wall flow behavior, and the distribution of turbulent kinetic energy by direct comparison with experimental data. Ultimately, the research seeks to identify the most suitable turbulence model for reliable CFD prediction of jet-crossflow interactions in turbine blade film cooling applications.

2. Problem statement

While experimental tests on the interaction between a main airflow and a secondary cooling flow over a flat wall with several holes have been conducted, the main goal of these geometric setups is to spread the cooling jet more uniformly across the surface, thereby improving overall cooling effectiveness and uniformity. The experimental tests took place in the ISAE-ENSMA wind tunnel facility, which is part of the P' Institute in Poitiers, France. This was accomplished by fitting a flat plate with a number of offset rows of holes. The test plate in Figures 2 and 3 has 81 cylindrical cooling holes set in staggered rows. They show how the ISAE-ENSMA tunnel is configured and how the plate is set up. Every hole has a 6 mm diameter and is angled 30° from the wall surface. The holes also have elliptical edge exits that measure 12 mm across. This is supposed to help the jet attach better and make the film cooler.

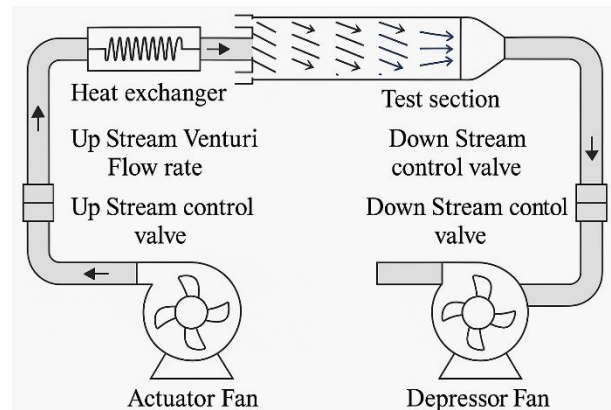


Fig. 2. General description of the tunnel facility

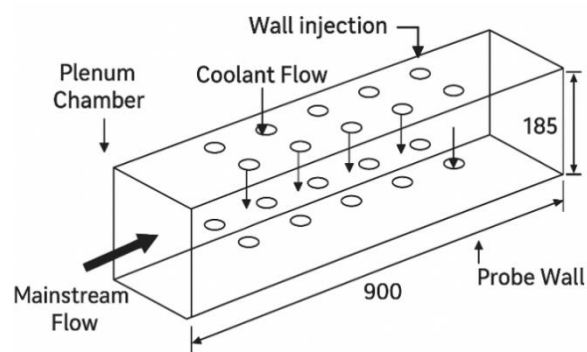


Fig. 3. General description of the test vein and the flat plate

PIV measurements were performed using a flat-plate configuration intersected by four rows of open coolant jets. As shown in Fig. 4, the jets were angled at 30° toward the axial direction and had a consistent blowing ratio of $M = 2$ during the experiments carried out at the ENSMA test facility. In the (x, y) plane, both the injection flow and the main flow are present. For data acquisition, the flat plate is split into six measurement planes. The camera system can move along the z -axis within the (x, z) plane. There is a 35 mm displacement between each measurement plane. This helps the camera capture the three-dimensional flow structure well.

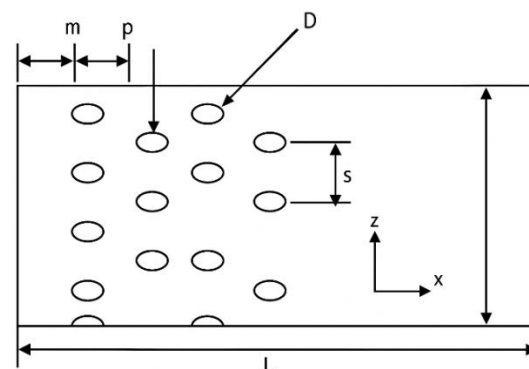


Fig. 4. Calculation domain and experimental setup

Figure 5 illustrates the six measurement planes delineated within the experimental test facility. Data acquired from these planes are used to assess the accuracy of numerical

predictions from RANS simulations conducted under equivalent operating conditions. Two turbulence modeling approaches are investigated: the standard $k-\epsilon$ model and the SST model, aiming to identify the model that exhibits superior agreement with the experimental measurements. In this configuration, the primary flow propagates along the x-axis at a uniform velocity of 2 m/s, while the coolant jets are introduced at an inclination of 30° relative to the x-axis within the (x, z) plane.

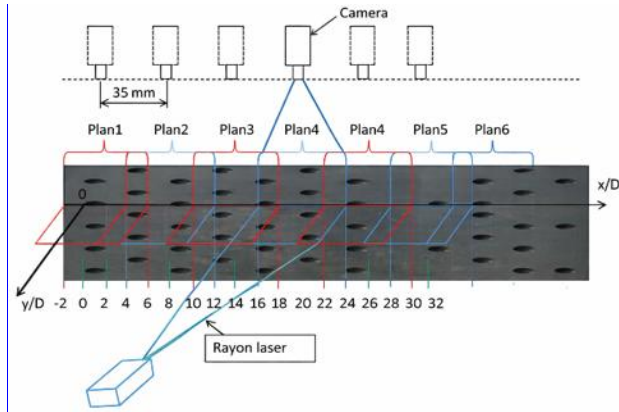


Fig. 5. Measurement plans available with PIV

3. Methodology

3.1. Experimental apparatus and protocol

To investigate the effect of secondary jet flows on the main airflow in gas turbine blade cooling, a sophisticated experimental setup was painstakingly built at the Laboratory for Thermal Studies (LET) of ENSMA in Poitiers. The experimental configuration mostly consists of a flat plate with a main airstream flowing across it. Secondary jets are pushed through several rows of staggered holes in the plate, and they interact with the main airstream. As seen in Fig. 6, the system has a PIV for high-resolution velocity field measurements. The device has these basic parts:

- The main airflow is generated by a MEIDINGER high-performance fan, model HP 98 / 10–13, with a static pressure of 20,000 Pa
- A test section with 81 cylindrical holes spaced 30° apart from the surface and measuring 6 mm in diameter houses a multi-perforated plate
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- A flow regulation valve for fine-tuning the injection parameters of the secondary air jets.

3.2. Integrated flow rate and velocity monitoring system

Located upstream of the test section, the main flow control system includes a regulatory valve (Fig. 7) that monitors the mass flow rate of the principal airflow, denoted Q_e . Located downstream of the test section, a secondary flow control system is used to measure the suction flow rate, Q_d , which matches the injection flow rate, Q_i , as shown in the

diagram. Dedicated flow control systems help to control and track the mass flow rate of every stream. Each system has a control valve to regulate flow to the intended setpoint, as well as a Venturi meter for accurate flow rate measurement. To precisely measure airflow velocity, the Venturi devices are connected to both absolute and differential pressure sensors. While the injection flow rate varied, reaching a maximum of 25 g/s, the primary flow mass rate was kept steady at 45 g/s throughout all tests.

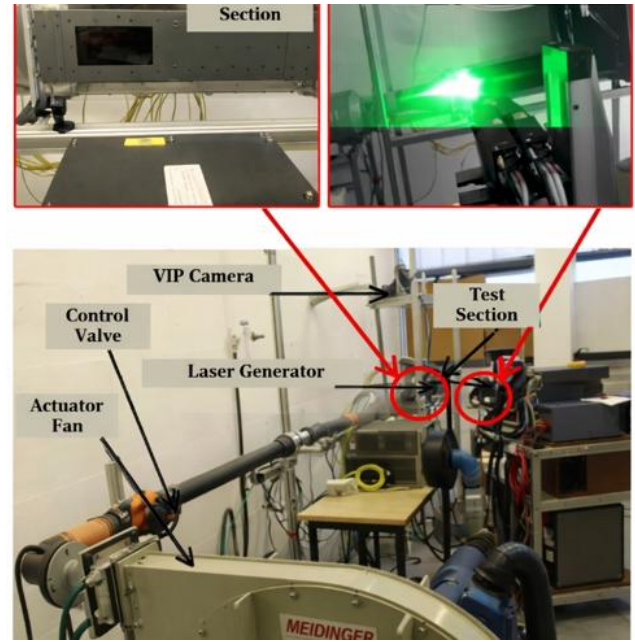


Fig. 6. Experimental apparatus

3.3. Aerodynamic conditions

Defined and determined using the following equation, the main aerodynamic parameter controlling this experimental study is the blowing ratio MM :

$$M = \frac{\rho_i U_i}{\rho_e U_e} \quad (1)$$

In this formulation, ρ_e , U_e , U_i , and ρ_i denote the density and velocity of the main flow, and the velocity and density of the injection flow, respectively. Accordingly, the blowing ratio MM can also be expressed in the following alternative form:

$$M = \frac{\rho_i U_i (n S_i)}{\rho_e U_e S_e} \times \frac{S_e}{n S_i} \quad (2)$$

where S_i is the injection area which is calculated by:

$$S_i = \frac{\pi D^2}{4} \quad (3)$$

S_e : represents the tunnel test section ($S_e = 0.0185\text{m}^2$), n : is the number of open injections.

The main mass flow rate and the injection flow rate are calculated by the relations below:

$$Q_e = \rho_e U_e S_e \quad (4)$$

$$Q_i = \rho_i U_i (n S_i) \quad (5)$$

These calculations reveal a link between the injection rate M and the primary flow rate Q_e as well as the injected air flow rate Q_i :

$$M = K \frac{Q_i}{Q_e} \text{ with } K = \frac{S_e}{nS_i} \quad (6)$$

where Q_i is the mass flow rate of the injection flow ($\text{kg} \cdot \text{s}^{-1}$), Q_e is the mass flow rate of the main flow ($\text{kg} \cdot \text{s}^{-1}$).

For all analyzed cases, the main mass flow rate is set to $45 \text{ g} \cdot \text{s}^{-1}$, corresponding to an average mainstream velocity of $2 \text{ m} \cdot \text{s}^{-1}$. Therefore, the mass flow rate of the coolant injection stream controls the blowing ratio (M).

3.4. Mathematical formulation

The numerical model was validated by a thorough CFD analysis. Among other important factors, this process involved selecting a suitable turbulence model, defining precise inlet boundary conditions, and evaluating grid independence. Among the several engineering-relevant outputs, the local heat transfer distribution is particularly sensitive to the complex dynamics of secondary flow structures. Therefore, its precise prediction relies much on the accuracy of the computational approach used and the underlying physical modeling, as well as its robustness. The Navier-Stokes equations, given here, are the governing equations for fluid flow [1, 2, 12, 22-24, 29]:

$$\begin{aligned} \frac{\partial \bar{\rho} \bar{u}_i}{\partial x_j} &= 0 \\ \bar{u}_j \frac{\partial \bar{\rho} \bar{u}_i}{\partial x_j} &= -\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\rho \bar{v} \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) - \rho \bar{u}_i' \bar{u}_j' \right] \end{aligned} \quad (7)$$

where: $-\rho \bar{u}_i' \bar{u}_j'$ are the Reynolds turbulent stresses.

3.5. Standard k-ε model

In the standard k-ε model, the Reynolds stresses are modelled as:

$$-\rho \bar{u}_i' \bar{u}_j' = -\frac{2}{3} \rho k \delta_{ij} + 2 \mu_t \bar{S}_{ij} \quad (8)$$

The eddy viscosity μ_t is related to the turbulent kinetic energy k and to its dissipation rate ϵ as

$$\mu_t = \rho C_\mu \frac{k^2}{\epsilon} \quad (9)$$

All standard linear two-equation turbulence models are based on Eq. (8), which shows a linear relationship between the Reynolds (turbulent) stress tensor and the mean strain rate tensor. Solving their respective modelled transport equations yields the spatial distributions of the turbulent kinetic energy (k) and its dissipation rate (ϵ) inside the flow field. These equations include the following source terms:

$$S_k = P - \epsilon \quad ; \quad S_\epsilon = C_{\epsilon 1} \frac{\epsilon}{k} P - C_{\epsilon 2} \frac{\epsilon^2}{k} \quad (10)$$

where P is the production of turbulence

$$P = -\rho \bar{u}_i' \bar{u}_j' \frac{\partial \bar{u}_i}{\partial x_j} \quad (11)$$

Thus, the velocity field's near-wall boundary conditions are defined using the usual wall-function technique. A Dirichlet condition requiring a zero value at the wall is used for the (k). By contrast, a local equilibrium assumption underpins the (ϵ) at the neighboring near-wall node, hence

guaranteeing coherence with the wall-function approach, as:

$$\epsilon = C_\mu^{3/4} \frac{k^{3/2}}{0.4 \delta y} \quad (12)$$

3.6. Shear Stress Transport (SST) model

Menter [21] created the SSTs turbulence model by carefully combining the best features of the k-ε and k-ω models to maximize their efficacy throughout various flow zones. To be more precise, the SST model uses the k-ω formulation in the near-wall region, where it is known to work well, and then switches to the transformed k-ε equations in the free-stream region, where the latter model usually provides better accuracy. A mixing function, F_1 , controls this change by gently interpolating between the two formulations. As Tulapurkara observes, the SST model improves its ability to capture the impacts of unfavorable pressure gradients and separation by combining Bradshaw's idea that the Reynolds shear stress is correlated with the turbulent kinetic energy [30, 31]. The resulting transport equations of the SST model are as follows:

$$\begin{aligned} \frac{\partial}{\partial t} (\rho k) + \bar{u}_j \frac{\partial}{\partial x_j} (\rho k) &= \tau_{ij} \frac{\partial \bar{u}}{\partial x_j} - \beta^* \rho \omega k + \\ &\frac{\partial}{\partial x_j} \left[(\mu + \sigma_k \mu_t) \frac{\partial k}{\partial x_j} \right] \end{aligned} \quad (13)$$

$$\begin{aligned} \frac{\partial}{\partial t} (\rho \omega) + \bar{u}_j \frac{\partial}{\partial x_j} (\rho \omega) &= \frac{\mu}{\mu_t} \tau_{ij} \frac{\partial \bar{u}}{\partial x_j} \\ &- \beta \rho \omega^2 + \frac{\partial}{\partial x_j} \left[(\mu + \sigma_\omega \mu_t) \frac{\partial \omega}{\partial x_j} \right] \\ &+ 2(1 - F_1) \rho \sigma_{\omega 2} \frac{1}{\omega} \frac{\partial k}{\partial x_i} \frac{\partial \omega}{\partial x_j} \end{aligned} \quad (14)$$

The constants in the SSTs turbulence model are determined using a blending function. If φ represents a generic constant in the SST model, and φ_1 and φ_2 are the corresponding constants in the original k-ω model and the transformed k-ε model, respectively, then φ is computed as follows:

$$\varphi = F_1 \varphi_1 + (1 - F_1) \varphi_2 \quad (15)$$

The constants in the k-ω are:

$$\sigma_{k1} = 0.850, \sigma_{\omega 1} = 0.5, \beta_1 = 0.0750, a_1 = 0.311, \beta^* = 0.090, K = 0.410$$

$$\gamma_1 = \frac{\beta_1}{\beta^*} - \frac{\sigma_{\omega 1} K^2}{\sqrt{\beta^*}} \quad (16)$$

The constants in the transformed k-ε model are:

$$\sigma_{k2} = 1.0, \sigma_{\omega 2} = 0.856, \beta_2 = 0.0828, \beta^* = 0.09, K = 0.41,$$

$$\gamma_1 = \frac{\beta_2}{\beta^*} - \frac{\sigma_{\omega 2} K^2}{\sqrt{\beta^*}} \quad (17)$$

τ_{ij} is given by Equation:

$$\begin{aligned} -\rho \bar{u}_i' \bar{u}_j' &= -\frac{2}{3} \rho k \delta_{ij} + \mu_t \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) \\ F_1 &= \tanh(\arg_1^4) : \end{aligned} \quad (18)$$

$$\arg_1 = \min \left[\max \left(\frac{\sqrt{k}}{0.09 \omega y}, \frac{500 \nu}{y^2 \omega} \right), \frac{4 \rho \sigma_{\omega 2} k}{C_{Dk \omega} y^2} \right] \quad (19)$$

$$CD_{k\omega} = \max\left(2\rho\sigma_{\omega 2} \frac{1}{\omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_i}, 10^{-10}\right), v_t = \frac{a_1 k}{\max(a_1 \omega, \Omega F_1)} \quad (20)$$

$$F_2 = \tanh(\arg_2^2)$$

$$\arg_2 = \max\left(2 \frac{\sqrt{k}}{0.09 \omega y}, \frac{500 v}{y^2 \omega}\right) \quad (21)$$

3.7. Computational domain and boundary conditions

The three-dimensional numerical flow field is shown in Fig. 7 using a symmetry plane that mimics the arrangement of the ISAE-ENSMA test facility. The computational domain consists of a flat plate with four rows of spaced cylindrical cooling holes. Each hole has a diameter of 6 mm. In the transverse ($s/D = 4$) and axial ($p/D = 4$) directions, the hole spacing ratio is fixed at 4. Using a three-dimensional steady-state formulation, the commercial CFD program FLUENT solves the governing transport equations. Figure 7 illustrates the mesh, geometry, and boundary conditions (BC) applied for both the experimental and computational investigations.

To guarantee accuracy in resolving convective terms, a 2nd order upwind discretization diagram is used. Ambient air comprises both the injected coolant jet and the main-stream flow in the current film-cooling configuration. The working fluid is treated as a perfect gas inside a macroscopic continuum framework because the thermodynamic conditions are somewhat low, that is, the temperature and pressure are both moderate.

The elliptic nature of the governing partial differential equations makes it essential to define boundary conditions along the entire boundary of the computational domain. Three different forms of boundary conditions, velocity inlet, outflow, and symmetry constraints, are applied as seen in Fig. 7. The coolant jet is introduced at an inclination angle (α) of 30°, with an injection slot width (D) of 0.006 m. The accurate representation of turbulent flow within CFD models depends on the correct specification of inlet parameters for important variables: the axial (u) and transverse (v) velocity components, turbulent kinetic energy (k), and either the turbulence dissipation rate (ϵ) or specific dissipation rate (ω), which jointly define the inflow conditions. These criteria are derived from pertinent experimental data sets [3]. The axial and vertical components of velocity are exactly specified at the jet exit interface; the spanwise velocity is considered to be insignificant. The axial velocity distribution is defined alongside the turbulence intensity variables at the inlet region upstream of the flat plate. At the upstream and downstream borders of the domain, a zero-gradient (Neumann) condition is applied for all remaining flow quantities.

Three different mesh resolutions are contrasted in Fig. 8: coarse, intermediate, and fine. Derived from the $k-\epsilon$ turbulence model at the station $X/D = 0$, the findings center on the normalized axial mean velocity profiles U/U_e . Two particular mesh setups are emphasized: a coarse mesh with 557,381 cells and a fine mesh with 2,575,501 cells. The goal of this mesh sensitivity analysis was to find the best mesh that would yield accurate results while requiring the

least computer time. It also aimed to see how mesh density affected simulation accuracy.

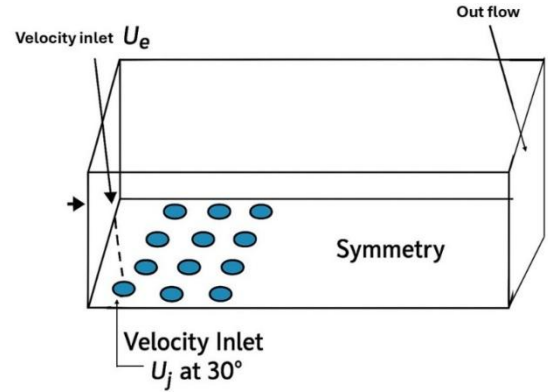


Fig. 7. Experimental and numerical studies: 3-D domain and boundary conditions

The comparison shows that the coarse and fine meshes have practically identical normalized velocity profiles, indicating mesh independence in the near-field. Therefore, the coarse mesh was chosen for the following simulations to cut computer time without compromising accuracy. Using the three-dimensional CFD commercial solver, ANSYS FLUENT and the SIMPLC pressure-velocity coupling method, the governing transport equations were found and solved. To better control numerical stability and accuracy, the convective terms were discretized using a second-order upwind scheme.

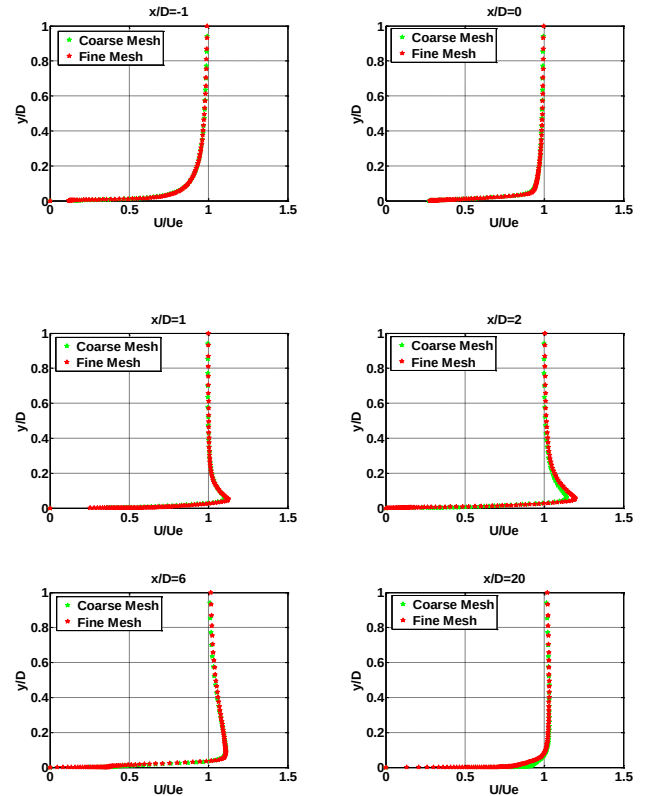


Fig. 8. Profiles of the average reduced axial speeds U/U_e obtained by the model turbulence $k-\epsilon$ for two coarse and fine meshes

Except at the downstream point ($x/D = 20$), where modest differences were observed, the profiles of the normalized axial velocity U/U_e remained constant across the assessed stations for the two mesh resolutions. These differences are due to numerical accuracy rather than mesh quality. Therefore, all the following computations were carried out using a coarse mesh with about 24,000 cells.

3.8. Mesh and computational domain

The two-dimensional (2D) variant of the CFD commercial solver, ANSYS FLUENT, was used to model and solve the transport equations. The flat plate domain had a structured, non-uniform mesh applied. Two different mesh arrangements were tested to determine a suitable mesh resolution. The first had roughly 24,000 cells; the second, around 38,000. Finer resolution near the wall showed grid spacing up to ten times smaller in that area.

The comparative study revealed few differences in the findings from the two mesh setups, suggesting that the smaller near-wall mesh had little effect on accuracy. As a result, the coarser mesh with less refinement near the wall was chosen for all subsequent simulations to reduce computational cost and time. Figure 9 shows the last mesh layout.

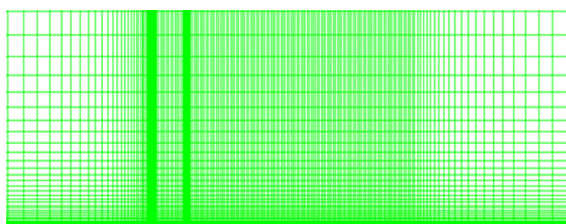


Fig. 9. Mesh of the plane plate

4. Results and discussion

4.1. Results analysis

To evaluate the predictive correctness of the numerical method, representative results of the time-averaged flow field derived from both PIV measurements and RANS simulations are compared in the present study. To determine the most appropriate turbulence model, FLUENT, a commercial CFD program, is used to simulate turbulent flow across a flat plate. Two often-used $k-\epsilon$ turbulence model equations are assessed. The results section provides an understanding of the reliability and performance of the computational models by contrasting experimental data with numerical predictions. The jet exit region, the dispersion behavior of the coolant jets, and the flow mixing mechanisms merit close scrutiny. The flow vector was zoned across six mixed streamwise and spanwise measurement planes to get statistically significant mean time numbers. Each combined plane corresponds to a different measurement, so there are six sets of measurements. This arrangement guarantees crossflow fluid travel throughout the full measurement domain, including all six planes.

4.2. Flow calculation results

4.2.1. Comparison between experimental and the numerical results of reduced speeds

Simulations of the mean and turbulent velocity profiles along the centerline ($x-z$ baseline) were conducted over the

flat plate and compared with experimental data, as seen in Fig. 10 and Fig. 11, to help one visualize the interaction dynamics between the coolant jets and the crossflow. With an eye on determining which model best reflects the complex jet-crossflow interactions from the jet exit to the far downstream region, these comparisons seek to evaluate the predictive accuracy of the two models of Standard turbulence $k-\epsilon$ and SSTs with respect to the experimental PIV data. The comparisons were made at a blowing ratio of $M = 2$.

At several streamwise locations $X/D = -2, 0, 2, 8, 22$, and 30 the graphs show the decreased axial (U/U_e) and vertical mean velocity components (V/U_e) across the spanwise symmetry plane ($Z/D = 0$). The examination shows that the jet's momentum significantly disturbs and deflects the crossflow as it approaches the jet exit. Significant differences between the experimental data and numerical calculations are observed in these near-wall regions, particularly in the prediction of axial velocity profiles. Although the SST model closely fits experimental trends across most sites, both turbulence models tend to overpredict the decrease in axial velocity near the wall. There are clear differences between the two models right after the jets, especially in how well they capture the regions where the flow separates and reattaches. The better performance of the SST model is ascribed to its greater ability to handle energy generation and transport processes associated with coherent structures, yet it has a tendency to somewhat overestimate their impact.

Conversely, the standard $k-\epsilon$ model omits important flow characteristics, especially in the downstream area, where it forecasts insufficient jet dispersion and little lateral dispersion. Particularly in regions affected by jet-crossflow interaction dynamics, this deficit might account for the clear departure from experimental data.

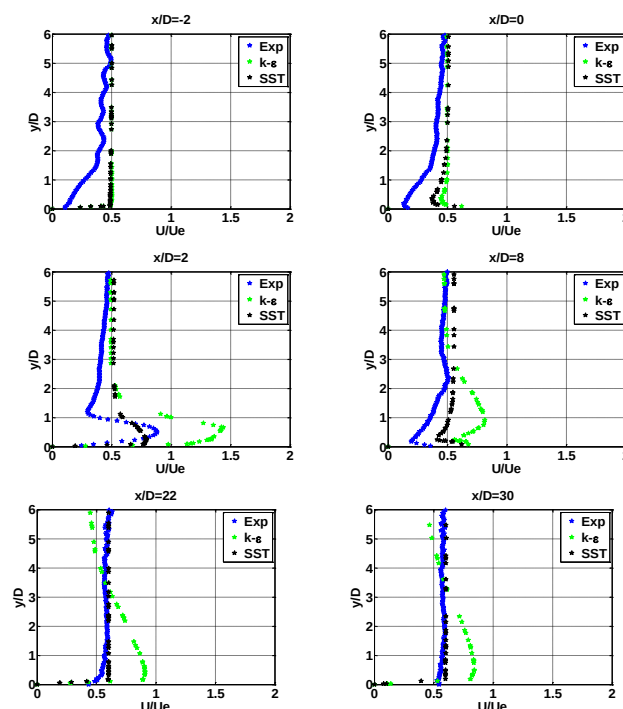


Fig. 10. Comparison of experimental and numerical results of reduced mean velocity profiles (U/U_e) along the central plane $Z/D = 0$

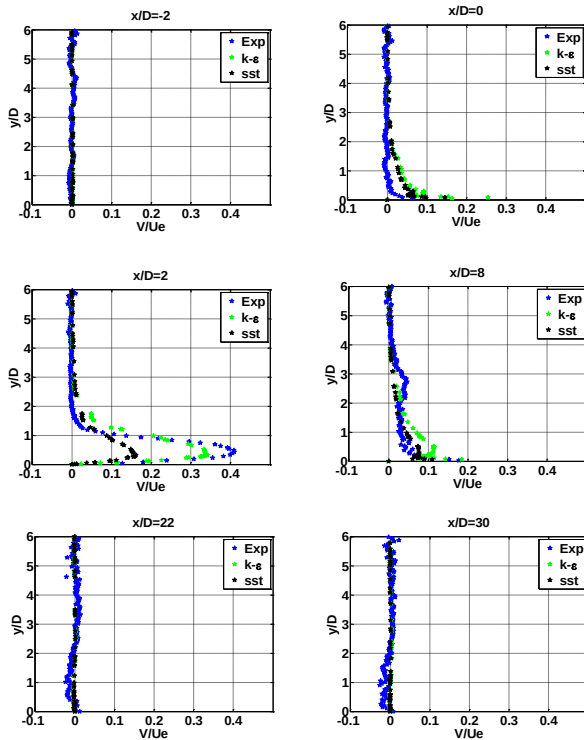


Fig. 11. Comparison of experimental and numerical results of reduced mean velocity profiles (V/U_e) along the central plane $Z/D = 0$

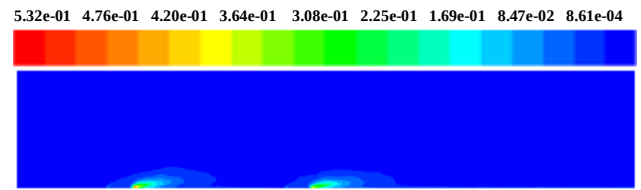
4.2.2. Comparison of turbulence energy along the axis of the jet

For a jet shot into a boundary layer at an injection rate of $M = 2.0$, Fig. 12 shows the contours of turbulent kinetic energy (TKE) predicted by the two models, Standard turbulence $k-\epsilon$ (Fig. 12a) and SSTs (Fig. 12b), compared to the experimental PIV measurements (Fig. 12c). The visualizations show how a fully turbulent jet interacts with the flow in the boundary layer. Though both computational models reflect the basic characteristics of the jet development and turbulence generation, neither matches the experimental results well. Near-wall areas and the degree of turbulent diffusion are where inconsistencies are especially obvious. Compared to the $k-\epsilon$ model, which often underestimates the spatial extent and intensity of turbulent structures, the SST model captures the shear-layer turbulence of the jet more accurately. Still, neither model can correctly reproduce the precise TKE distribution seen in the PIV data, therefore limiting their ability to capture the complex jet-crossflow interaction at this blowing ratio.

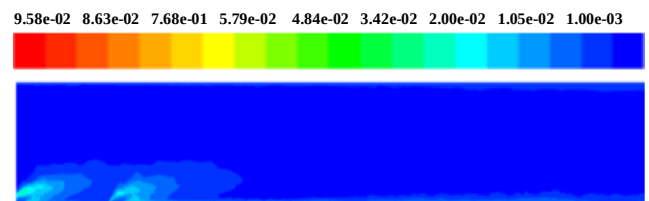
4.2.3. The contour lines of velocity magnitude derived from simulation outputs

Figure 13 shows the reduced mean velocity profiles from the three-dimensional (3D) simulations of the jet-crossflow interaction in the spanwise direction (Z/D) to investigate the lateral spreading behavior of the coolant jet. To evaluate the degree of jet diffusion and symmetry of the flow field, velocity profiles are derived at two streamwise locations, $X/D = 12$ and $X/D = 16$, then plotted along the spanwise direction. These graphs show how well the jet

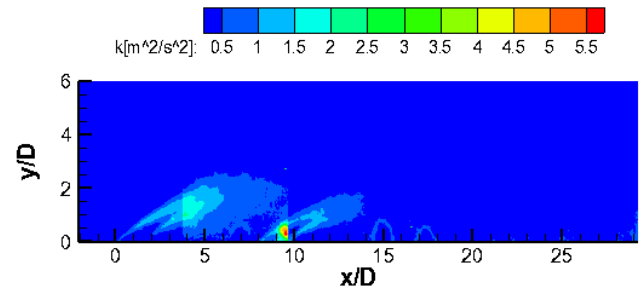
spreads and how well the fluid mixes as it moves away from the injection point.



(a) Contours of the kinetic energy of turbulence obtained by the turbulence model $k-\epsilon$



(b) Contours of the kinetic energy of turbulence obtained by the turbulence model SST



(c) Contours of the kinetic energy of turbulence obtained by the PIV

Fig. 12. Contours of the kinetic energy obtained by numerical calculation and by experiment

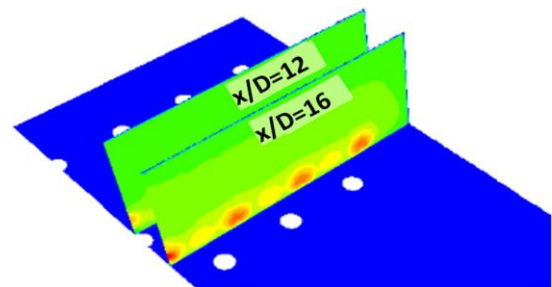


Fig. 13. Center planes $x/D = 12$ and $x/D = 16$

Transverse-plane measurements using the PIV technique could not be performed due to limitations in camera placement and movement. Therefore, only quantitative data from Fig. 14 for both turbulence models may be examined. The image shows isocontours of constant reduced velocity, which help show how the coolant film moves laterally. In terms of both the shape and intensity of the velocity contours, the standard $k-\epsilon$ and SST models provide rather similar estimates. For instance, the SST model yields a range of 1.79 to 4.87 for the reduced axial velocity U/U_e at $X/D = 16$; the $k-\epsilon$ model yields a range of 1.78 to 4.86. The RSM forecasts a range of 2.35 to 4.70 for comparison, showing a somewhat improved ability to capture the peak velocity zones. These findings imply that, while the $k-\epsilon$ and

SST models provide generally similar predictions, they might underpredict certain flow structures compared to more sophisticated models such as the RSM.

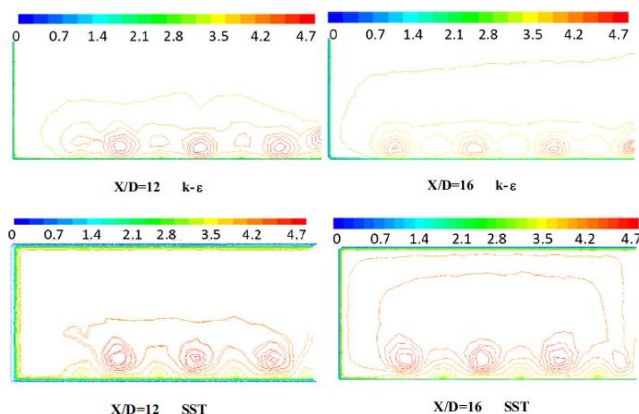


Fig. 14. Contours of velocity magnitude for $k-\epsilon$ and SST models in $X/D = 12$ and $X/D = 16$ center plane

Conclusions

This study experimentally and numerically examined the interaction of multiple staggered cooling jets with

a turbulent crossflow representative of gas turbine blade film cooling at a blowing ratio of $M = 2$. Comparisons between PIV measurements and RANS simulations indicate that the SST turbulence model predicts jet penetration, velocity decay, and turbulence distribution more accurately than the standard $k-\epsilon$ model, particularly in the near-hole region dominated by strong shear and recirculation. The $k-\epsilon$ model was found to overestimate near-wall velocities and to underestimate jet spreading, potentially leading to non-conservative cooling predictions. For engineering applications, the SST model is recommended for CFD-based analysis and the preliminary design of multirow film-cooling configurations operating at moderate blowing ratios, whereas the standard $k-\epsilon$ model should be used only for qualitative assessments or supported by experimental validation. Future work should focus on the application of advanced turbulence modeling approaches, such as Reynolds stress models or hybrid RANS LES methods, to better capture unsteady jet-crossflow interactions. In addition, coupling aerodynamic analysis with thermal simulations is recommended to directly evaluate film-cooling effectiveness and heat transfer performance under realistic operating conditions.

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The results of a hybrid vehicle testing in real-world conditions

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This paper presents the results of tests on a hybrid petrol-electric vehicle with external additional charging of the electric battery (a so-called "PHEV" solution). The tests were primarily intended to measure energy consumption, but vehicle speed was also recorded. The tests were conducted on test routes representing urban and rural driving. The tests were conducted for various vehicle control strategies. The hybrid drive and controlled battery charging strategies were implemented for zero and full initial battery charge. Additional tests were conducted in "sport" mode to achieve the highest power output. The tests were conducted with various measurement systems: based on measurements from the fuel dispenser, the on-board computer, the MyBMW application, and the external EOB interface. It was found that there were no significant differences in the measurement results between the systems used. The results showed that the sensitivity of average speed measurements to the control strategy for all test route stages was on the order of 6–16%. The sensitivity of average volumetric fuel consumption measurements to the control strategy for all test route stages was on the order of 4–80%. Whereas the sensitivity of the average speed measurement results to test route stages for all control strategies was on the order of (33–53)%. The sensitivity of average volumetric fuel consumption measurements to test route stages across all control strategies was on the order of 19–60%.

Key words: road vehicles, driving tests, fuel consumption, hybrid drive

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1. Introduction

The study focused on a hybrid vehicle with a combustion engine and an electric motor, with an external battery charging system [5–7, 12, 13, 15–17].

Hybrid combustion-electric drives offer many advantages, combining the characteristics of both combustion engine and electric motor drives [9].

The most important advantages of a hybrid vehicle's electric motor drive are as follows [9–11, 14]:

- Electric motor drive avoids the emission of pollutants contained in the combustion engine exhaust gases. Pollution from electric-vehicle motors is limited to particulate matter from tribological vapors, primarily from the vehicle's braking system and the friction between the tires and the road surface. Reducing exhaust emissions by using electric drives is particularly important for motorization in urban centers.
- Vehicle drives with electric motors also contribute to reducing noise emissions, which is particularly important in urban operation.
- The combined use of a combustion engine and electric motor drive improves the vehicle's dynamic operating characteristics – this is particularly important in urban traffic.

The key advantages of using an internal combustion engine in a hybrid vehicle are as follows [9]:

- The most important advantage of using an internal combustion engine in a hybrid vehicle is the increased vehicle range compared to a vehicle powered solely by an electric motor.
- Another advantage is the combined use of an internal combustion engine in the drive system with an electric motor, which improves the vehicle's dynamic characteristics.

There are various design solutions for hybrid drives – primarily in series, parallel, and mixed systems [13].

Among hybrid drive system solutions, a particularly important one is an intermediate form between a classic hybrid drive and an electric drive – one that includes additional external charging of the electric battery. This solution is called a PHEV [1, 5–7]. This solution has a significant advantage over an electric drive: it mitigates the notable range limitation of an electric vehicle.

A passenger vehicle with a PHEV drive is the subject of research, the results of which are presented in this paper.

2. Research method

The test vehicle was a BMW 5 Series Touring 530e xDrive passenger car (G31 generation). Table 1 presents the characteristics of the car's drive system [3].

Table 1. Drivetrain characteristics of the BMW 5 Series Touring 530e xDrive (G31) [3]

Parameter	Characteristic
Drivetrain type	Full PHEV hybrid system All-wheel drive
Combustion engine	BMW TwinPower Turbo technology R4, $V_{ss} = 1998 \text{ cm}^3$, $N_{eN} = 135 \text{ kW}/5000\text{--}6500 \text{ min}^{-1}$, $M_{eM} = 300 \text{ N}\cdot\text{m}/1350\text{--}4000 \text{ min}^{-1}$
Electric motor	Synchronous integrated with the 8-speed Steptronic gearbox, $N_{max} = 80 \text{ kW}/3140 \text{ min}^{-1}$, $M_{max} = 265 \text{ N}\cdot\text{m}/100\text{--}2500 \text{ min}^{-1}$
System power	215 kW
System torque	420 N·m
Transmission	8-speed Steptronic automatic transmission
Accumulator	Lithium-ion, 354 V, 12 kWh (10.4 kWh net)
Fuel consumption, CO ₂ emissions	2.4–2.2 dm ³ /100 km, 54–50 g CO ₂ /km
Range	48–53 km (EV), ~ 700 km (ICE & EV)
Top speed	225 km/h (limited electronically)

The study focused on fuel and energy consumption. The study was conducted based on:

- vehicle control strategies
- route characteristics
- fuel consumption measurement system.

The following vehicle control strategies were considered [3]:

- Strategy 1: start SOC = 100% – „hybrid” mode
- Strategy 2: start SOC = 0% – „hybrid” mode
- Strategy 3: start SOC = 100% – „battery control” mode
- Strategy 4: start SOC = 0% – „battery control” mode
- Strategy 5: start SOC = 0% – „sport” mode.

The following hybrid vehicle operating modes were considered [3]:

- „Hybrid” – the default operating mode, in which the system optimizes the interaction of both power sources: the combustion engine and the electric motor. At low speeds, electric drive is preferred, while at high speeds, the combustion engine is used.
- „Battery control” – the driver sets the battery charge level to be maintained or reached. Electric energy can be saved for urban driving after an extra-urban section of the route.
- „Sport” – both engines operate simultaneously to achieve maximum power.
- Vehicle control strategies vary significantly. There is a particularly significant difference in the degree of initial battery charging, especially in "battery control" mode.

The tests were conducted on the following routes:

- Stage 0 – technical section (Fig. 1)
- Stage I – extra-urban section (Fig. 2)
- Stage II – city loop (Fig. 3)
- Stage III – extra-urban return section (Fig. 4).

Vehicle movement characteristics during selected sections vary significantly, particularly on extra-urban sections, on city loops and technical sections. Vehicle movement conditions typical of driving in urban centers, especially in traffic congestion, are characterized by low average speed, high acceleration, and a significant proportion of idle time. Driving in moderate traffic congestion at medium speeds is characterized by low to medium load on the drive system. The average vehicle speed and low acceleration are also moderate. Driving at high speeds places a high load on the drive system, but at the same time provides high operating stability [8].

The above tests are similar to the RDE (Real Driving Emissions) test and, further, to the WLTC (Worldwide harmonized Light duty Test Cycle) test [2, 15].

The following speed and fuel consumption measurement systems are tested [3]:

- based on measurements from the fuel dispenser
- from the on-board computer
- from the MyBMW application
- from the external EOB interface.

Detailed analyses were performed for individual vehicle control strategies and routes, which determine the nature of the traffic:

- average vehicle speed – v_{AV}
- average volumetric fuel consumption – q_{AV} .

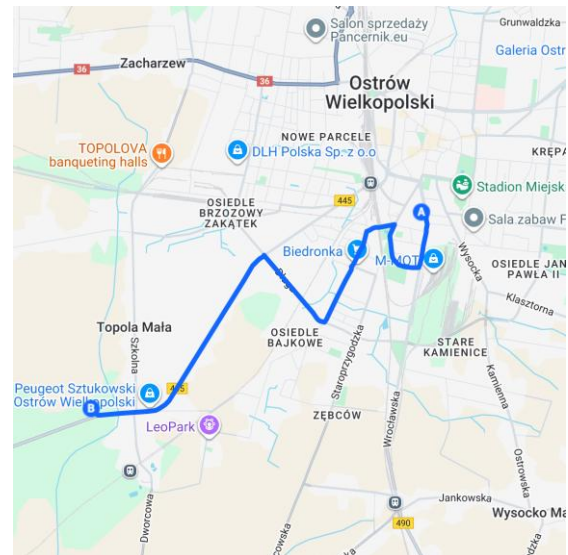


Fig. 1. Stage 0 – technical section [Google Maps]

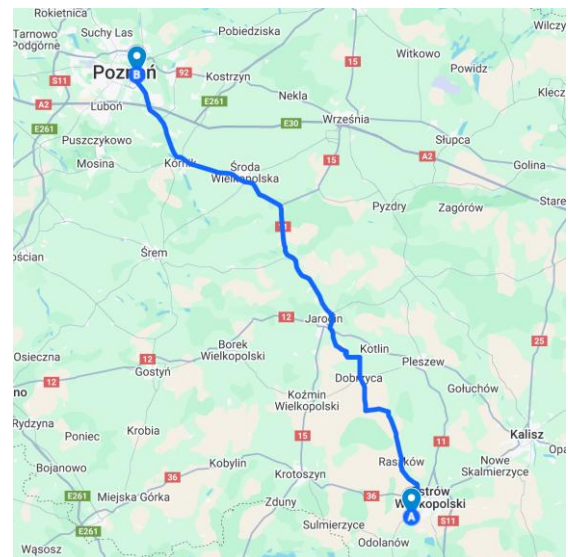


Fig. 2. Stage I – extra-urban section [Google Maps]

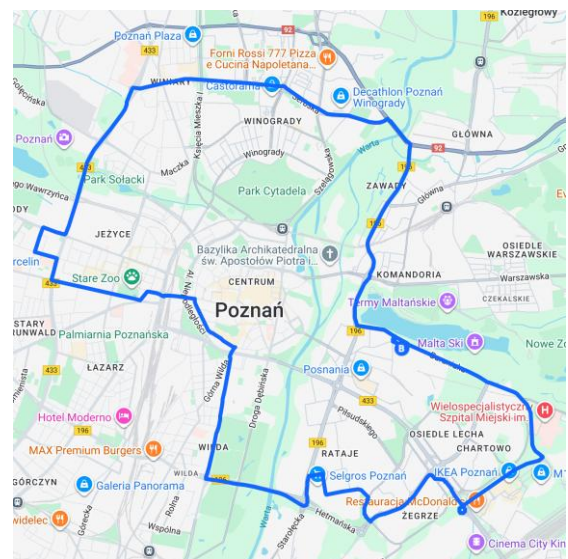


Fig. 3. Stage II – city loop [Google Maps]

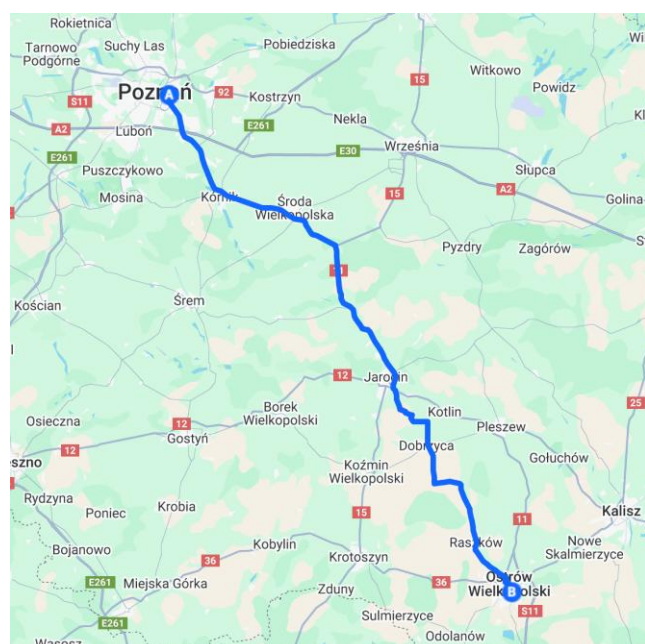


Fig. 4. Stage III – extra-urban return section [Google Maps]

3. Research results

Table 2 presents the test results for hybrid vehicle control strategy 1.

Table 2. Research results for the hybrid vehicle control strategy 1: S – distance, v_{AV} – average velocity, average distance volumetric fuel consumption – q_{AV} , volumetric fuel consumption – Q

Stage	S [km]			
	On-board computer	App MyBMW	EOBD	
Stage 0	7.5	7.7	7.7	
Stage I	120.7	121.0	115.7	
Stage II	29.1	29.0	29.2	
Stage III	121.8	121.0	115.7	
Together	279.1	278.7	260.6	
Stage	v_{AV} [km/h]			
	On-board computer	App MyBMW	EOBD	
Stage 0	41.2	39.9	39.9	
Stage I	67.5	67.6	61.7	
Stage II	20.5	19.8	20.4	
Stage III	65.3	66.0	61.4	
Together	60.9	61.2	58.1	
Stage	q_{AV} [dm ³ /100 km]			
	On-board computer	App MyBMW	EOBD	
Stage 0	13.6	13.6	13.6	
Stage I	4.0	3.9	4.3	
Stage II	10.3	10.3	10.6	
Stage III	6.7	6.6	7.2	
Together	6.1	6.0	6.7	
Stage	Q [dm ³]			
	On-board computer	App MyBMW	EOBD	Refueling
Stage 0	1.0	1.0	1.0	
Stage I	4.8	4.7	5.0	
Stage II	3.0	3.0	3.1	
Stage III	8.2	8.0	8.3	
Together	17.0	16.7	17.5	17.3

Table 3 presents the test results for hybrid vehicle control strategy 2. Table 4 presents the test results for the hybrid vehicle control strategy 3. Table 5 presents the test results for the hybrid vehicle control strategy 4.

Table 3. Research results for the hybrid vehicle control strategy 2: S – distance, v_{AV} – average velocity, average distance volumetric fuel consumption – q_{AV} , volumetric fuel consumption – Q

Stage	S [km]			
	On-board computer	App MyBMW	EOBD	
Stage 0	7.5	7.5	7.2	
Stage I	121.9	122.0	123.5	
Stage II	28.2	28.0	28.4	
Stage III	121.0	121.0	122.6	
Together	278.6	278.5	274.5	
Stage	v_{AV} [km/h]			
	On-board computer	App MyBMW	EOBD	
Stage 0	42.5	42.6	38.4	
Stage I	58.3	58.3	55.9	
Stage II	23.8	23.8	22.9	
Stage III	61.7	62.0	60.2	
Together	55.9	56.0	55.4	
Stage	q_{AV} [dm ³ /100 km]			
	On-board computer	App MyBMW	EOBD	
Stage 0	8.3	8.3	9.1	
Stage I	6.8	6.6	7.1	
Stage II	9.5	9.4	9.7	
Stage III	6.7	6.6	6.8	
Together	7.1	6.9	7.5	
Stage	Q [dm ³]			
	On-board computer	App MyBMW	EOBD	Refueling
Stage 0	0.6	0.6	0.6	
Stage I	8.3	8.1	8.7	
Stage II	2.7	2.6	2.8	
Stage III	8.1	8.0	8.4	
Together	19.7	19.3	20.5	20.7

Table 4. Research results for the hybrid vehicle control strategy 3: S – distance, v_{AV} – average velocity, average distance volumetric fuel consumption – q_{AV} , volumetric fuel consumption – Q .

Stage	S [km]			
	On-board computer	App MyBMW	EOBD	
Stage 0	7.5	7.5	7.2	
Stage I	122.0	122.0	122.6	
Stage II	29.0	29.0	27.7	
Stage III	118.7	119.0	113.7	
Together	277.2	277.5	264.1	
Stage	v _{AV} [km/h]			
	On-board computer	App MyBMW	EOBD	
Stage 0	36.5	36.6	26.5	
Stage I	68.4	68.2	60.2	
Stage II	32.0	32.0	28.9	
Stage III	69.4	69.4	62.3	
Together	64.2	64.1	58.5	
Stage	q _{AV} [dm ³ /100 km]			
	On-board computer	App MyBMW	EOBD	
Stage 0	5.6	5.5	6.0	
Stage I	6.5	6.4	6.8	
Stage II	0.0	0.0	0.0	
Stage III	6.6	6.4	7.1	
Together	5.8	5.7	6.4	
Stage	Q [dm ³]			
	On-board computer	App MyBMW	EOBD	Refueling
Stage 0	0.4	0.4	0.4	
Stage I	7.9	7.8	8.4	
Stage II	0.0	0.0	0.0	
Stage III	7.8	7.6	8.1	
Together	16.2	15.8	16.9	17.2

Table 5. Research results for the hybrid vehicle control strategy 4: S – distance, v_{AV} – average velocity, average distance volumetric fuel consumption – q_{AV} , volumetric fuel consumption – Q

Stage	S [km]			
	On-board computer	App MyBMW	EOBD	
Stage 0	7.5	7.6	7.6	
Stage I	120.7	121.0	122.3	
Stage II	28.2	28.0	26.8	
Stage III	118.7	119.0	113.8	
Together	275.1	275.6	262.8	
Stage	v _{AV} [km/h]			
	On-board computer	App MyBMW	EOBD	
Stage 0	36.3	36.3	31.6	
Stage I	61.0	61.0	59.0	
Stage II	24.8	24.8	22.3	
Stage III	56.6	56.4	51.8	
Together	54.7	54.7	53.0	
Stage	q _{AV} [dm ³ /100 km]			
	On-board computer	App MyBMW	EOBD	
Stage 0	6.6	6.5	6.7	
Stage I	8.9	8.6	9.1	
Stage II	0.0	0.0	0.0	
Stage III	6.2	6.0	6.8	
Together	6.8	6.5	7.3	
Stage	Q [dm ³]			
	On-board computer	App MyBMW	EOBD	Refueling
Stage 0	0.5	0.5	0.5	
Stage I	10.7	10.4	11.1	
Stage II	0.0	0.0	0.0	
Stage III	7.4	7.1	7.7	
Together	18.6	18.0	19.3	18.8

Table 6 presents the test results for the hybrid vehicle control strategy 5.

Table 6. Research results for the hybrid vehicle control strategy 5: S – distance, v_{AV} – average velocity, average distance volumetric fuel consumption – q_{AV} , volumetric fuel consumption – Q

Stage	S [km]			
	On-board computer	App MyBMW	EOBD	
Stage 0	7.5	7.5	7.1	
Stage I	120.7	121.0	122.3	
Stage II	28.1	28.0	26.8	
Stage III	118.6	119.0	113.7	
Together	274.9	275.5	262.8	
Stage	v _{AV} [km/h]			
	On-board computer	App MyBMW	EOBD	
Stage 0	25.7	25.6	23.3	
Stage I	68.0	68.1	65.8	
Stage II	22.2	22.4	20.1	
Stage III	61.9	62.5	56.4	
Together	59.5	59.9	57.7	
Stage	q _{AV} [dm ³ /100 km]			
	On-board computer	App MyBMW	EOBD	
Stage 0	10.1	10.1	11.0	
Stage I	6.8	6.4	6.8	
Stage II	10.1	10.0	11.4	
Stage III	7.0	6.8	7.5	
Together	7.3	7.0	7.9	
Stage	Q [dm ³]			
	On-board computer	App MyBMW	EOBD	Refueling
Stage 0	0.8	0.8	0.8	
Stage I	8.2	7.7	8.3	
Stage II	2.8	2.8	3.0	
Stage III	8.3	8.1	8.6	
Together	20.1	19.4	20.7	21.0

As the first step in analyzing the test results, it was decided to check the sensitivity of the test results to the measurement methods used. The coefficient of variation of the test results was used as a measure of sensitivity.

Table 7 presents the coefficient of variation of the average vehicle speed for all route stages and all vehicle control strategies. The test results were also presented in Fig. 5.

Table 7. Coefficient of variation – W of average velocity – v_{AV} for all stages and all strategies of vehicle control

	Strategy 1	Strategy 2	Strategy 3	Strategy 4	Strategy 5
Stage	W [v_{AV}]				
Stage 0	0.015	0.047	0.143	0.064	0.045
Stage I	0.042	0.020	0.058	0.016	0.016
Stage II	0.015	0.019	0.047	0.049	0.049
Stage III	0.031	0.013	0.050	0.040	0.045
Together	0.023	0.005	0.042	0.014	0.016

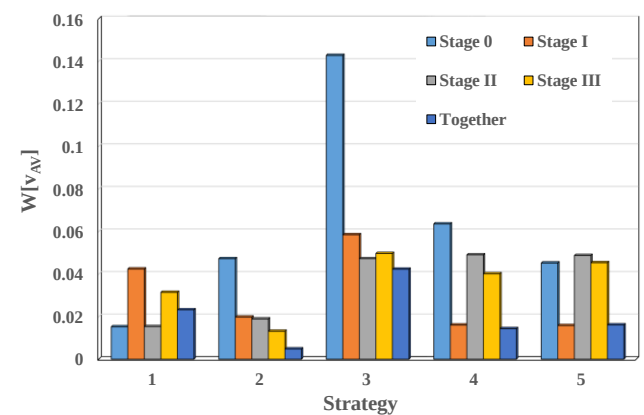


Fig. 5. Coefficient of variation – W of average velocity – v_{AV} for all stages and all strategies of vehicle control

Table 8 presents the coefficient of variation of the average volumetric fuel consumption for all route stages and all vehicle control strategies. The test results were also presented in Fig. 6.

Table 8. Coefficient of variation – W of average distance volumetric fuel consumption – q_{AV} for all stages and all strategies of vehicle control

	Strategy 1	Strategy 2	Strategy 3	Strategy 4	Strategy 5
Stage	W [q_{AV}]				
Stage 0	0.000	0.042	0.041	0.015	0.045
Stage I	0.046	0.028	0.027	0.022	0.016
Stage II	0.015	0.013	0.000	0.000	0.049
Stage III	0.038	0.013	0.045	0.051	0.045
Together	0.051	0.031	0.050	0.049	0.016

The highest coefficient of variation for average vehicle speed was 0.037, while the average value for volumetric fuel consumption on the road was 0.030. Therefore, the sensitivity of the test results to the measurement methods used was low, so it was decided to use the test results from the on-board computer for further consideration.

Figure 7 presents the average vehicle speed, and Fig. 8 the average volumetric fuel consumption on the road for all route stages and all vehicle control strategies.

To scale fuel consumption measurement methods, we compared measurements from the on-board computer with those obtained during refueling. Figure 9 presents the re-

sults of a comparison of fuel consumption between the on-board computer and refueling.

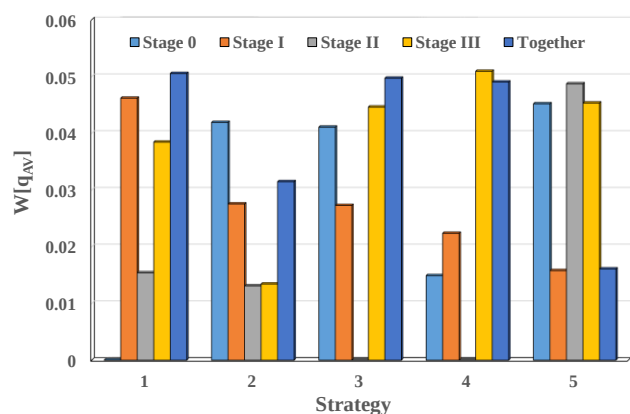


Fig. 6. Coefficient of variation – W of average distance volumetric fuel consumption – q_{AV} for all stages and all strategies of vehicle control

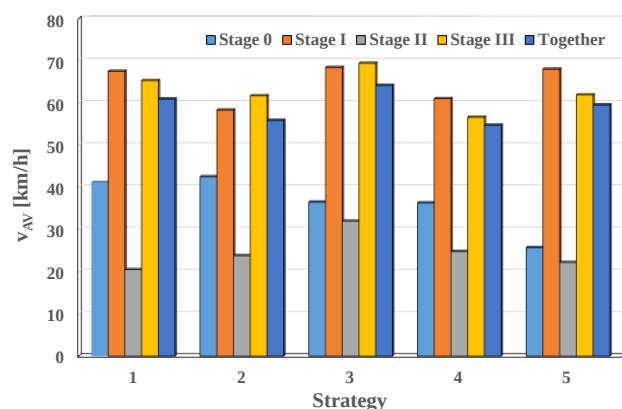


Fig. 7. Average vehicle velocity – v_{AV} for all stages and all strategies of vehicle control

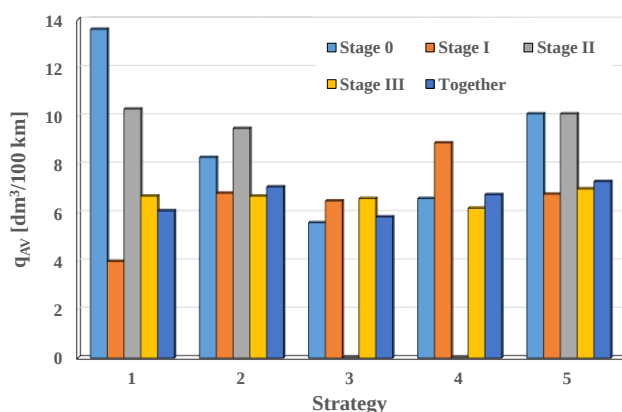


Fig. 8. Average distance volumetric fuel consumption – q_{AV} for all stages and all strategies of vehicle control

The consistency between fuel consumption measurements based on the on-board computer and those based on vehicle refueling was found to be good. The relative difference between fuel consumption measurements based on the on-board computer and those based on vehicle refueling was within the range of 0.009 to 0.053, with an average value of 0.033.

Next, the sensitivity of the average vehicle speed and average volumetric fuel consumption measurements to the control strategy was tested for all route stages. The results for the coefficient of variation of average vehicle speed and average volumetric fuel consumption were presented in Table 9.

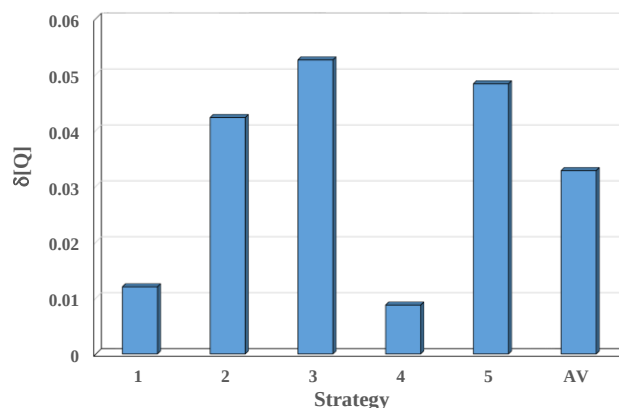


Fig. 9. Relative difference in fuel consumption based on the on-board computer readings and based on refueling the vehicle – δ

Table 9. Coefficient of variation – W of average vehicle velocity – v_{AV} and average distance volumetric fuel consumption – q_{AV} for all stages

Stage	$W[v_{AV}]$	$W[q_{AV}]$
0	0.162	0.320
1	0.065	0.236
2	0.160	0.818
3	0.067	0.039
Together	0.058	0.085

The results of the coefficient of variation of the average vehicle speed were shown in Fig. 10 and the average volumetric fuel consumption in Fig. 11.

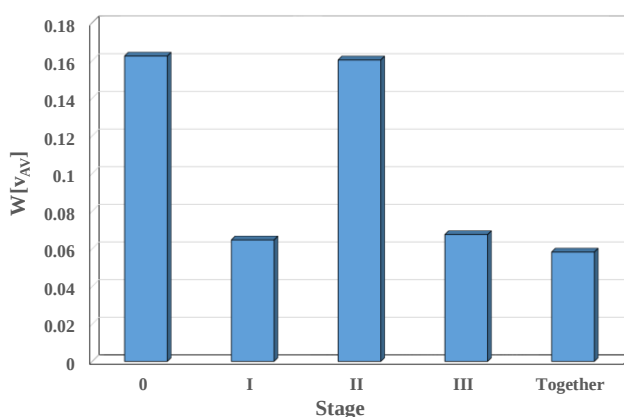


Fig. 10. Coefficient of variation – W of average vehicle velocity – v_{AV} for all strategies of vehicle control

The sensitivity of average vehicle speed measurement results to the control strategy for all route stages was on the order of 6–16%.

The sensitivity of average volumetric fuel consumption measurements to the control strategy across all route stages was on the order of 4–80%.

The sensitivity of average vehicle speed and average volumetric fuel consumption measurement results to the route stages for all control strategies is presented in Table 10.

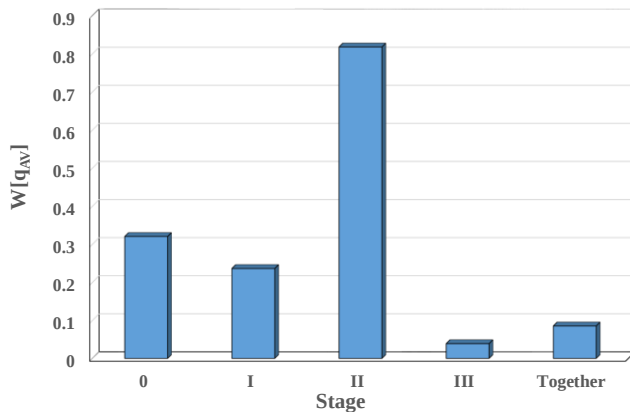


Fig. 11. Coefficient of variation – W of average distance volumetric fuel consumption – q_{AV} for all stages

Table 10. Coefficient of variation – W of average vehicle velocity – v_{AV} and average distance volumetric fuel consumption – q_{AV} for all strategies of vehicle control

Strategy	$W[v_{AV}]$	$W[q_{AV}]$
1	0.396	0.419
2	0.526	0.446
3	0.337	0.583
4	0.331	0.608
5	0.465	0.189

The results of the coefficient of variation of the average speed were shown in Fig. 12 and the average volumetric fuel consumption in Fig. 13.

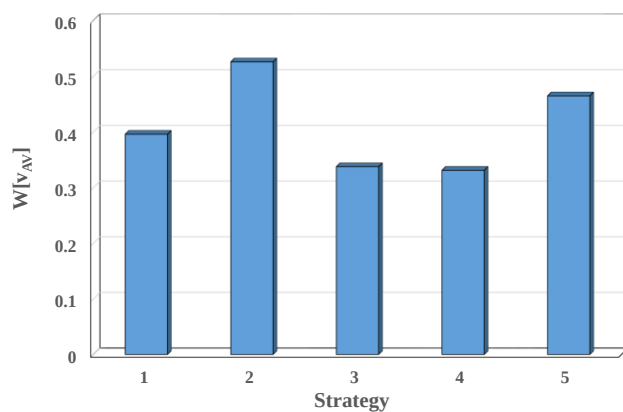


Fig. 12. Coefficient of variation – W of average vehicle velocity – v_{AV} all strategies of vehicle control

The sensitivity of average vehicle speed measurements to route stages across all control strategies was on the order of 33–53%. The sensitivity of the average volumetric fuel consumption measurement results at the route stages across all control strategies was on the order of 19–60%.

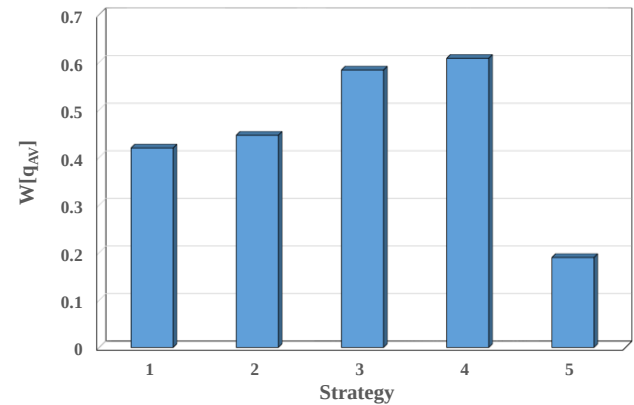


Fig. 13. Coefficient of variation – W of average distance volumetric fuel consumption – q_{AV} all strategies of vehicle control

4. Summary

Based on the research conducted, the following conclusions can be drawn regarding the use of PHEV:

1. Vehicle traffic conditions typical of driving in city centers, especially in traffic jams, are characterized by low average speeds, high accelerations, and a high proportion of idle time.

Driving in moderate traffic conditions at medium speeds is characterized by low to medium load on the drive system. The average vehicle speed is moderate and accelerations are low.

Driving at high speeds places high loads on the drive system, yet it maintains high operating stability.

The control of a PHEV significantly impacts the drive system's operation. The initial state of charge of the electric battery is a key factor. The initial state of charge of the electric battery determines the vehicle's electric range, with significant consequences for fuel consumption and exhaust emissions. Exhaust emissions from electric drive are limited to particulate matter from tribological vapors, primarily from the braking system and tire-road surface interaction.

2. The impact of the PHEV's drive system operating conditions is assessed by its impact on driving speed and fuel consumption for various route characteristics and control strategies.

The highest average speed is for extra-urban sections, the lowest for the city loop.

The highest volumetric fuel consumption was observed in the technical section, followed by the city loop, and the lowest in the extra-urban section.

There was no significant effect of the control strategy on the average vehicle speed and volumetric fuel consumption. Of course, in strategy 3, with the battery fully charged and the electric motor operating, fuel consumption was lowest.

3. The sensitivity of the measurement results to the use of the vehicle speed and fuel consumption measurement system was found to be low.
4. The consistency between the fuel consumption measurements based on the on-board computer and those based on refueling was found to be good. The relative difference between fuel consumption measurements

- from the on-board computer and those from refueling was 0.009–0.053, with an average of 0.033.
- The sensitivity of average vehicle speed measurement results to the control strategy for all route stages was on the order of 6–16%. The sensitivity of average volumetric fuel consumption measurements to the control strategy across all route stages was on the order of 4–80%.
 - The sensitivity of average vehicle speed measurement results to the route stages for all control strategies was on the order of 33–53%. The sensitivity of average volumetric fuel consumption measurements to route stages across all control strategies is on the order of 19–60%.
- It is advisable to continue this work at a significantly expanded scope:
- Continuous recording of vehicle speed and fuel consumption. This would enable the design of driving tests corresponding to vehicle motion under empirical test

conditions, the synthetic results of which are presented in this paper, as well as the analysis of measurement results in the following domains:

- time (including statistical analysis and analysis of dynamic properties using the average speed product of speed and acceleration modulus as a measure of the dynamic properties of the velocity process [4])
 - frequency
 - process values.
- Conducting exhaust emission tests under the following conditions:
 - actual conditions on the test routes selected, consistent with those selected in this work
 - on a chassis dynamometer in tests corresponding to actual driving conditions in the empirical tests.

Nomenclature

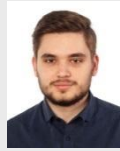
AV	average value	Q	volumetric fuel consumption
CO ₂	carbon dioxide	R	in-line arrangement of cylinders
EOBD	European On Board Diagnostic	S	distance
EV	electric drive	SOC	state of charge
ICE	combustion engine	v	velocity
M _{eM}	maximum torque of the combustion engine	V _{ss}	displacement of the combustion engine
M _{max}	maximum torque of the electric motor	W	coefficient of variation
N _{eN}	rated power of the combustion engine	δ	relative difference
N _{max}	maximum power of the electric motor	q	distance volumetric fuel consumption

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Transient combustion behaviour of a high-performance W12 twin turbo spark ignition drift car spec engine

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The paper presents an analysis of indicators of a high-performance W12 twin-turbo engine used in drifting applications, supplemented with time-series data on engine parameters. The tests were conducted on a chassis dynamometer using indicating and data logging equipment. The engine external characteristics were determined, yielding 898 kW and 1488 Nm. To enable safe in-cylinder pressure measurements, the turbocharger boost control duty cycle was limited to 30%, resulting in reduced thermodynamic loading without compromising the representativeness of the combustion process. The 450 consecutive operating cycles were recorded under transient conditions. Under full-load conditions, the average $MBF_{50} = 13.8^{\circ}CA$ and $COV(MBF_{50}) = 17.3\%$, indicating conservative phasing relative to MBT and significant cycle-to-cycle variability. It was demonstrated that indexing enables precise correlation of transient parameters with combustion evolution and is an effective tool for optimizing the calibration of high-performance engines.

Key words: high-performance engine, dyno testing, engine indication, data logging, boost control

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1. Introductions

Currently, combustion engines remain the dominant power source in most machines, vehicles, and other means of transport. The development of the engine sparked the automotive industry's dynamic growth, and the widespread use of cars led to the emergence of a new competitive discipline: motorsport. Consequently, the internal combustion engine became one of the key areas of technological development and the main object of design competition.

Originally, motorsport was based mainly on commercial vehicles that were only slightly modified, and competition took the form of races between designated destinations or on closed-circuit tracks. Along with the development of automotive technology, numerous motorsport disciplines began to separate (Fig. 1), and vehicle designs became increasingly specialized and designed for specific competition conditions.

The construction of internal combustion engines used in motorsports differs significantly from that of commercial engines. These differences arise from distinct operating conditions and regulatory constraints. The design and operating characteristics of such engines are closely tailored to the specific discipline, and the main design differences include, among others, forced induction systems, lubrication systems, construction materials, and combustion control strategies. In the highest categories of circuit racing and single-seater, such as Formula 1, Formula 2, Formula 3, Hypercar in the FIA World Endurance Championship, and IndyCar, prototype engines are used that are designed from scratch for a specific vehicle and technical regulations [8, 24, 26, 36, 39].

The development of a prototype motorsport engine involves concept definition in accordance with technical regulations, followed by computer-aided design (CAD) as well

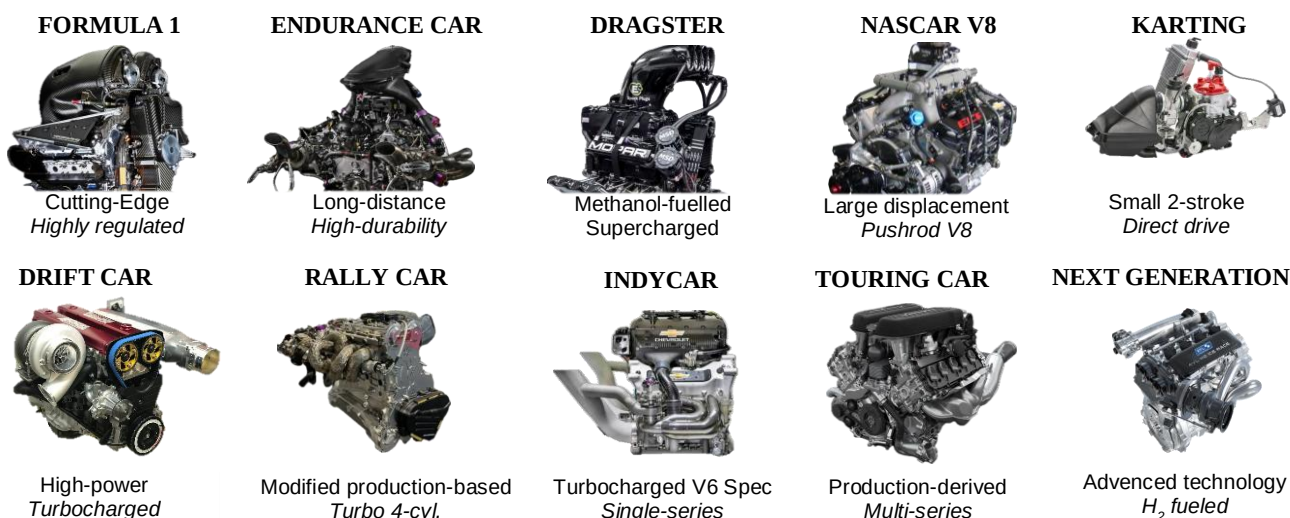


Fig. 1. Representative engine configurations used in competitive motorsport applications [5, 6, 19, 25, 29, 30, 33–35, 38]

as finite element method (FEM) and computational fluid dynamics (CFD) analyses of the engine and its components. The next step is to build a prototype and conduct tests on an engine dynamometer, including data acquisition during operation. After integration with the vehicle, development is iterative and data-driven, based on experimental data and subsequent design modifications [1]. Factory engine development programs incur extremely high costs, reaching \$1.4 billion [37] in F1 and \$200 million [12] in LMP1.

These costs often contribute to the decline of a racing series. An example of this is the final phase of the Class 1 regulations of the Deutsche Tourenwagen Masters series, which saw a marked escalation in the technical sophistication of the vehicles, directly translating into increased participation costs [15].

The cars were equipped with carbon monocoques and highly powerful, prototype 2.0 L R4 turbo engines (BMW P48, Audi 2.0 L R4 TFSI DTM), designed from scratch as structural elements of the vehicle (Audi RS5 DTM – Fig. 2) [10, 27].

The use of such advanced designs significantly increased development and operating costs to a level comparable to other top series based on prototypes. For comparison, the estimated budgets for the Audi and BMW factory programs for car construction in the final phase of the Class 1 regulations were around €30–50 million per year [14].

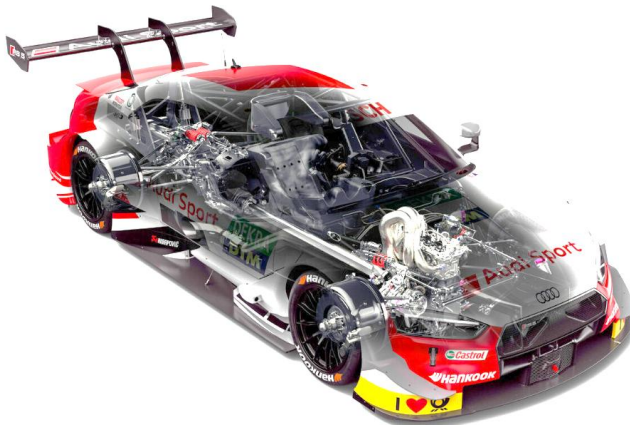


Fig. 2. Fully prototype, centrally single-seat carbon-monocoque Audi RS5 DTM [3]

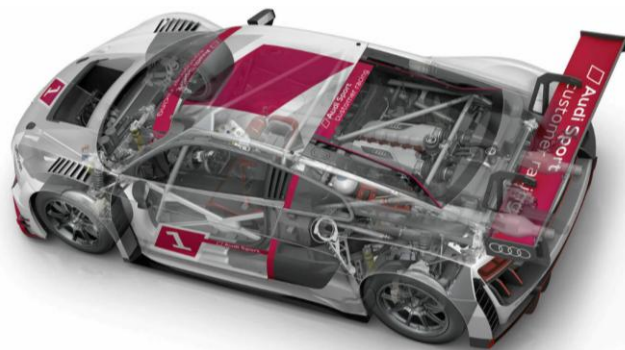


Fig. 3. Production-based Audi R8 LMS GT3 with a series-derived V10 and race-modified suspension [4]

In contrast, in the first season of the new regulations, they could reach as much as €80 million. The DTM pro-

gram for private teams was estimated at €5–6 million per year [14]. Following the rule changes from the 2021 season, which allowed only FIA GT3 cars with production-derived powertrains (including the BMW S58 inline six-cylinder, the naturally aspirated Audi R8 LMS V10 (Fig. 3), and Mercedes-AMG V8 units) and the simultaneous application of the Balance of Performance system, budget requirements have decreased to around €3 million. The number of competitors has increased to around 24 [20].

Engines prepared for drifting and lower-level touring car series, such as TCR, GT, and NASCAR, as well as for selected national single-seater applications, use production-based engines due to limited budgets [20]. Using ready-made engines allows skipping many R&D stages, limiting engine preparation to modifications to the equipment and control system.

Drifting is a relatively new discipline, in which mainly private competitors participate, or factory support is only available in the highest class of formula drift (Papadakis Racing supported by Toyota Gazoo Racing [32], Ford Racing support for Vaughn Gittin Jr, Chelsea DeNofa, and most recently, James Dean [16]), which significantly affects much smaller budgets and access to factory programs.

In drift racing, the engine is exposed to frequent dynamic loads, where quick throttle response, high torque across a wide range of engine speeds, and the ability to maintain stable operation under non-steady flow and cooling conditions are crucial. Unlike factory motorsport programs, where engine dynamometers and indicator measurements are the primary tools for engine development, in drifting projects based on modified production engines, the process of validation, tuning, and performance control is usually limited to a chassis dynamometer, less often to an engine dynamometer, and advanced indicator testing is rare.

Drifting designs most often use engines derived from high-performance production models, such as the Toyota 2JZ-GTE and Nissan RB26DETT inline six-cylinder engines, modern BMW R6 engines (e.g., S55/S58), or naturally aspirated and supercharged V8s from the GM LS family, which, thanks to their rigid block architecture and high availability of parts, provide a convenient base for competitive applications [13]. In adapting production engine blocks for motorsport applications, it is crucial to increase structural rigidity and resistance to high thermal and mechanical loads. Among other things, open-deck blocks are pinned to stabilize the cylinder liners, and additional reinforcements are added to the lower part of the engine in the form of girders or stiffening plates for the crankshaft supports [22]. These design changes are often complemented by advanced functional coatings, such as ceramic thermal barrier coatings (TBCs) applied to piston crowns and low-friction coatings (LFCs) used on piston skirts or bearing surfaces, which reduce friction losses and improve durability under high-load operating conditions [17, 40].

An integral part of the powertrains used in drifting is forced induction, which allows for very high specific power while maintaining a wide torque range. In practice, single-turbocharger systems or high-performance twin-turbo configurations offering high mass flow rate and high pressure ratio (Garrett G40/G42, BorgWarner EFR 9280) dominate.

The construction of the high-performance ERF series turbocharger equipped with advanced regulation and control systems is shown in Fig. 4. The performance map of the 9280 model (Fig. 5) demonstrates the capability to achieve pressure ratios exceeding 4.0 and airflow rates above 100 lb/min, significantly higher than those typical of standard passenger-car turbochargers.

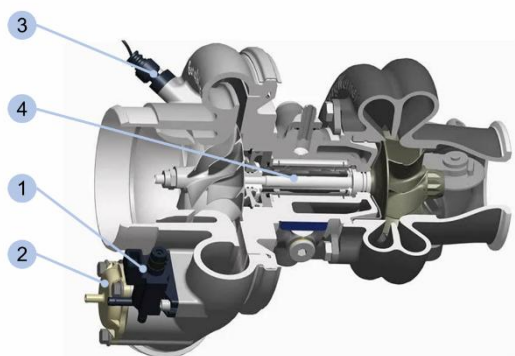


Fig. 4. Construction of twin scroll BorgWarner EFR series turbocharger, equipped with a boost pressure control solenoid valve (BCSV) (1), integrated compressor bypass valve (CRV) (2), rotor speed sensor (3), and ceramic ball bearings (4) [18]

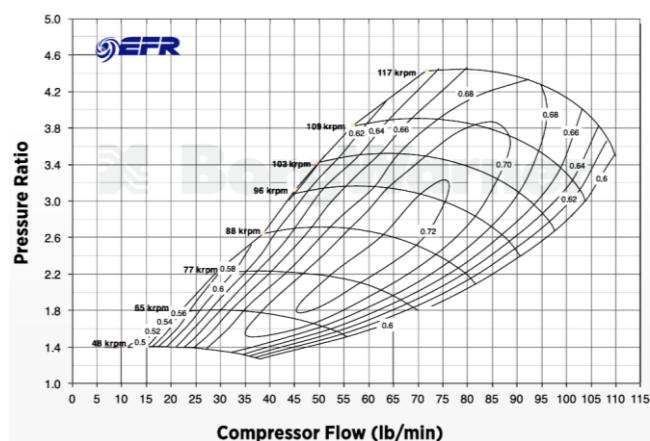


Fig. 5. BorgWarner EFR 9280 turbocharger compressor performance map [9]

At the same time, mechanical compressors are used much less frequently and are mainly found in selected projects based on V8 engines. The most commonly used turbochargers are those with ball bearings and twin-scroll housings, working with equal-length exhaust manifolds and external wastegates [28]. Blow-off valves are used on the compressor side to limit surge during rapid throttle closure.

In addition to a reinforced engine housing and a high-capacity turbocharging system, a control system based on programmable standalone ECUs with individual electrical harnesses is required. The official controller used in the Formula Drift series is Link ECU [23], but in practice, teams also use other programmable standalone units such as MoTeC, ECUMaster, Haltech, and AEM [11]. These solutions are essential in drifting – they provide full control over engine operation and the implementation of motorsport strategies such as flex-fuel using an ethanol content sensor, anti-lag and launch control systems for better use of forced induction, advanced engine thermal man-

agement and lubrication control strategies, as well as extensive data recording and integration with additional vehicle systems [2].

In addition to the three fundamentals of a drifting engine described above, independent cooling and lubrication systems play an important role in maintaining stable engine operating conditions under prolonged loads and during high-lateral-force driving.

In practice, complex oil-cooling systems, electric cooling pumps, additional oil and coolant radiators, and larger oil pans or dry-sump systems are used to prevent temporary drops in lubrication pressure. These are complemented by reinforced ignition systems, high-performance fuel injectors, temperature and pressure sensors at key points in the engine, and ECU-protected strategies that limit torque when the permissible operating parameters are exceeded.

Drifting engine calibration is typically performed on a chassis dynamometer using a standalone ECU [7,21]. Calibration begins by setting the base Fuel Table (VE or Injector Pulse Width) and the Ignition Timing Map, then adjusting the Boost Control Table based on gear or throttle position. Under these conditions, the operating limits are determined based on parameters such as lambda target (usually $\lambda \approx 0.78$ – 0.85 at high load), exhaust gas temperature (EGT), knock level, intake air temperature (IAT), and torque values, with ignition and fuel dose adjustments made iteratively based on ECU logs. After dyno calibration, motorsport strategies are implemented, including Anti-Lag Control, Launch Control, Gear-based Boost, and Ignition Cut during gear changes, with final tuning taking place under track conditions through analysis of telemetry data and correction maps.

Typically, drifting engines are tuned without indicator measurements, which is why the unit's operating limits are determined based on indirect indicators available in the ECU and on a chassis dynamometer [31]. Ignition calibration is performed using the "torque plateau" method, which involves gradually increasing the ignition advance angle until the torque increase disappears, while observing the knock level, exhaust gas temperature and lambda stability. In the absence of parameters such as MBF50, APmax, or MAPO, tuners rely on analyzing logs covering knock retard, speed increase, intake air temperature, and ignition corrections to determine a safe operating margin for the engine. However, this does not guarantee maximum thermodynamic efficiency.

Considering the limited scope of thermodynamic research on this type of design (motorsport drifting ICE +1000 HP), the authors decided to conduct experimental verification and identification tests involving indicator measurements, enabling direct analysis of the combustion process in the cylinder and engine operating parameters. Comparing the indicator results with the ECU operating data enables an assessment of the impact of the adopted tuning strategies on combustion-chamber processes and indicates potential directions for further modification of the drive unit.

2. Engine and test object description

The research subject was a high-performance race engine built by Gregor Performance for drifting-type competi-

tion. The engine was based on a W12 spark ignition engine with twin turbochargers. The tested engine was installed in a Nissan 350Z (Z33) body, fully prepared for motorsport (Fig. 6). The aim of the modification was to achieve a high engine power output under specific racing operating conditions.

The engine technical data are presented in Table 1. The main features include a 48-valve DOHC system, PFI (port fuel injection), and a dry-sump lubrication system. Forced induction was achieved via a twin-turbo system with Garrett GT35 units. The engine was controlled by two independent Ecumaster EMU Black controllers. This standalone solution allowed for the operation of 12 cylinders.

Table 1. High-performance drift car Engine specifications

Parameter	Value
Engine type	W12 DOHC PFI
Displacement	5998 cm ³
Number of valves per cylinder	4
Cylinder bore	84.0 mm
Piston stroke	90.2 mm
Compression ratio	10:1
Engine aspiration	twin-turbo aspirated engine

a)



b)



Fig. 6. Drift-spec Nissan 350z (a) with W12 high-performance engine (b)

A security algorithm based on knock detection, exhaust gas temperature control, mixture composition, and oil pressure was used. The fuel system was adapted to operate in Flex Fuel mode based on alcohol concentration measurements in the fuel. The engine control system is adapted to

high-octane gasoline, alcohol fuels, and mixtures (E85). The engine temperature was regulated by an independent cooling system. System output was regulated by EWP electric pumps, which were controlled directly by the ECU. The drive train is based on a modified ZF 8HP eight-speed transmission, controlled by an external standalone controller. This control method enabled independent management of the gear shift strategy. The tests were conducted with E85 racing fuel, and the fuel composition sensor indicated an ethanol content of 69%.

3. Experimental setup and research methodology

3.1. Engine dynamometer test bench

The engine tests were carried out on a hub dynamometer test bench (Fig. 7).

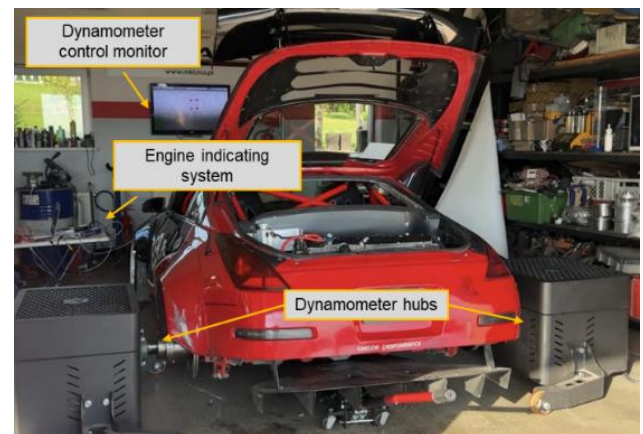


Fig. 7. Vehicle mounted on a chassis dynamometer during experimental testing under real load conditions

The specific characteristics of the test bench enabled the elimination of undesirable wheel slip. It is particularly important when testing engines that generate high torque. The Dyno Revolt HR-2 dynamometer was employed, enabling torque measurements in the 0–4000 Nm $\pm 0.1\%$ FS and wheel rotational speed 0–3500 rpm. The dynamometer measuring range enabled stable testing across the entire engine operating range. The combustion engine was tested on a chassis dynamometer during the car's operation. That enabled simultaneous acquisition of indicator parameters and powertrain operating values under real load conditions.

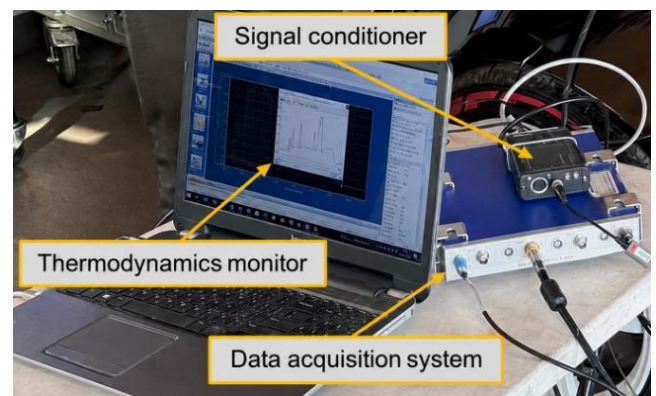


Fig. 8. View of AVL data acquisition system for thermodynamic combustion process analysis

3.2. Data acquisition system and measurements apparatus

The AVL IndiMicro 602, a four-channel system dedicated to the analysis of in-cylinder processes (Fig. 8), and an EcuMaster EDL-1 data logger were used to record data. The indicator measurement system for the crankshaft angular position included a piezoelectric pressure transducer integrated with an AVL ZI45 spark plug, a Pico TA398 high-voltage probe, and Pico TA018 current clamps. The cylinder pressure, high voltage in the secondary circuit of the ignition coil, and the injection control current were recorded. All measurements were performed within a single engine cylinder. Engine operating parameters were acquired from time traces using a dedicated data logger connected to the ECU. A total of 32 parameters were selected for recording, with a sampling frequency of 25 Hz. The specifications of the measuring equipment used are presented in Table 2.

Table 2. Measurement equipment and technical specifications

Parameter	Description	Specification
Combustion analysis	AVL IndiMicro 602	4 channels $\Delta\alpha = 0.5^\circ \text{CA}$
Data logger	EDL-1	200 channels $f_s = 25 \text{ Hz}$
Cylinder pressure	AVL ZI33	0–250 bar $\pm 0.3\%$ FS
Ignition high voltage	Pico TA398	Attenuation ratio 10,000:1
Injector signal	Pico TA018	0–60A 2.0%; $\pm 5 \text{ mA}$

3.3. Experimental plan and data processing

The tests were divided into two stages. The first stage involved determining the engine's external characteristics at full boost. The second stage involved determining the external characteristics, supplemented by a thermodynamic analysis of the combustion process under limited boost (30%) to mitigate the risk of damage to the spark plug integrated with the pressure transducer. During the second stage, the engine was diagnosed at idle speed.

The processing of measurement data recorded in the crankshaft angle function began with low-pass filtering at a cutoff frequency of 4 kHz. Next, the heat dissipation characteristics were determined in accordance with a simplified model based on the first law of thermodynamics [18]:

$$\frac{dQ}{d\alpha} = \frac{\gamma}{\gamma-1} P \frac{dV}{d\alpha} + \frac{1}{\gamma-1} V \frac{dP}{d\alpha} \quad (1)$$

where: γ , the polytropy exponent, was set at 1.32 for compression and 1.27 for expansion stroke; P is the measured pressure in the cylinder, and V is the current cylinder volume relative to the angular position of the crankshaft.

Based on the current volume and pressure in the cylinder, the average indicated mean effective pressure was determined according to the formula:

$$\text{IMEP} = \frac{1}{V_d} \cdot \int P dV \quad (2)$$

where: V_d is the displacement volume, P is the instantaneous pressure value for the instantaneous cylinder volume V .

Maximum values and their angular positions were determined for selected parameters. The ignition timing

SOIgn was determined from the high-voltage waveform in the secondary circuit of the ignition coil.

4. Results

4.1. Full-load engine performance characteristics

Figure 9 illustrates the full-load characteristics of the engine operating at 100% boost (solid line) and at a turbocharger boost control duty cycle limited to 30% (dashed line). Under full boost conditions, the engine produced 889 kW at 6222 rpm and 1488 Nm of torque at 4393 rpm. These results were used to limit the boost during indicator measurements. Under limited boost conditions, the engine produced 508 kW and 889 Nm of torque.

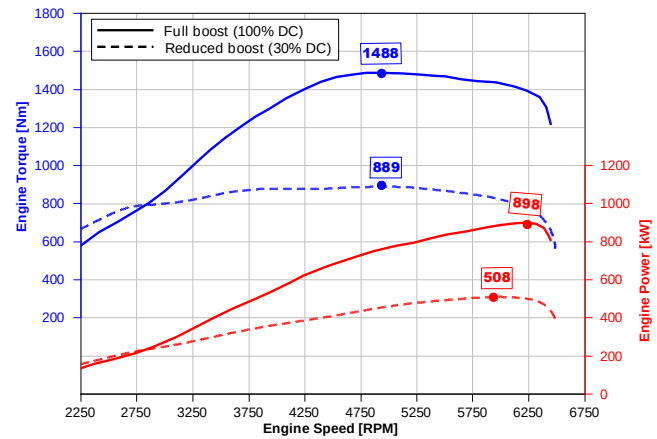


Fig. 9. Engine torque and power characteristics measured on a chassis dynamometer under full and partial boost conditions

The obtained results reflect the high-performance configuration of the tested engine, including turbocharging, a modified intake system, standalone ECU control, and operation on ethanol-blended fuel. High engine torque is required in drifting to maintain controlled rear-wheel slip during sustained oversteer conditions.

4.2. Engine control variables under test conditions

The control method is important for engine performance under the specific operating conditions dictated by motorsport. The engine calibration strategy was based on external operating parameters available from the ECU, without direct feedback from in-cylinder combustion analysis. Figure 10 illustrates the time-based selection of engine control variables and operating parameters under partial boost conditions.

Figure 10a presents the time history of absolute manifold pressure in the intake manifold (MAP) after the turbocharger (green line) and the boost control duty cycle of the solenoid valve regulating the turbocharger output flow (blue line). During the initial phase of engine acceleration and gradual throttle opening, at engine speeds from 1650 to 2400 rpm, the increase in boost pressure is regulated by the rising exhaust gas flow driving the turbine. Above 2400 rpm, the engine enters wide-open throttle (WOT), and the intake manifold pressure rises above atmospheric pressure. At the same time, the boost is limited to approximately 30%, corresponding to a partial exhaust gas bypass before the turbine blades. Despite the use of exhaust gas bypass,

further increases in engine speed result in boost pressure rising to 160 kPa, with the valve control ratio simultaneously corrected to 23%. Once the maximum engine speed of 6335 rpm is reached, the throttle closes, and the boost air relief valve takes over the boost pressure control function.

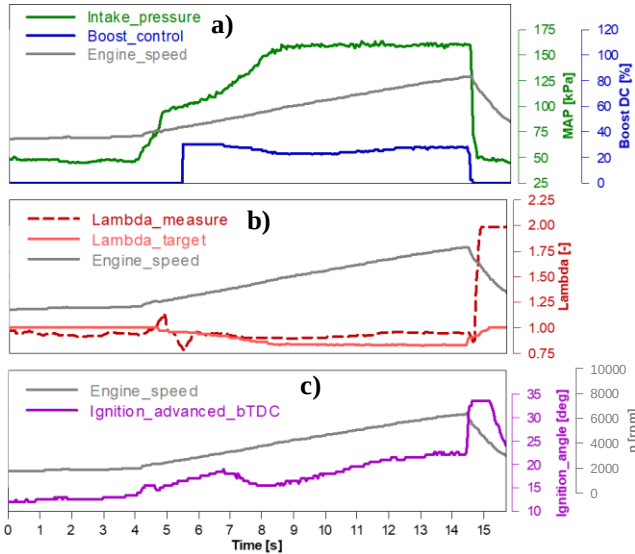


Fig. 10. Transient engine control variables and operating parameters recorded under partial boost conditions

Figure 10b shows changes in the air excess ratio λ during acceleration. The dotted line represents the actual value of λ measured by the oxygen sensor at the exhaust, while the solid line represents the value set by the engine control unit. The set value at idle speed was determined to achieve stoichiometric combustion, aiming for a rich combustion $\lambda \approx 0.83$ during engine acceleration. The transient boost control during the initial phase of acceleration produced a temporary lean mixture with $\lambda \approx 1.15$, followed by enrichment to $\lambda \approx 0.95$ due to fuel metering. During further acceleration, the setpoint and actual values were the same. However, this trend changed towards running on a leaner mixture than the setpoint due to insufficient turbocharger boost control duty cycle despite the current correction from 30 to 24%, but without exceeding $\lambda = 1.00$. In the range from 1600 rpm to 3000 rpm, the ignition advance angle increased classically with rotational speed (Fig. 10c). Subsequently, up to 3900 rpm, there was a correction related to the boost limitation and a linear increase correlating with the engine speed.

4.3. Transient thermodynamic indicator analysis

The test was conducted for 0 to 15 seconds, and, in parallel with the time signal recording, a total of 450 consecutive engine cycles were recorded and presented as cylinder pressure curves (Fig. 11).

The first 70 cycles corresponded to transitional states associated with throttle opening. In this range, a gradual increase in maximum combustion pressure was observed, reaching approximately 33 bar. Cycles 70 to 197 represented the engine-load increase phase, during which maximum pressure increased to approximately 85 bar. The subsequent cycles corresponded to the engine operating at full load,

characterized by relatively stable maximum pressure values.

The observed changes in cylinder pressure represent the transition from low-load operation through a phase of increasing load to stable high-load conditions.

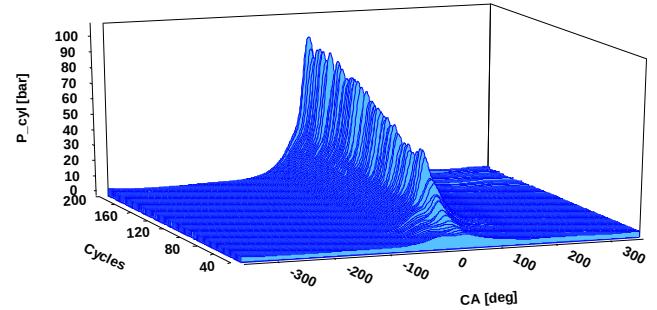


Fig. 11. In-cylinder pressure evolution versus crank angle and subsequent cycles under transient partial boost conditions

4.4. Combustion process characteristics under transient operating conditions

Based on the cylinder pressure curves, the time-dependent changes in the combustion process's characteristic parameters were determined (Fig. 12), including IMEP, Pmax, maximum heat release rate, MBF10, MBF50, and MBF90, which enabled the evaluation of combustion phasing evolution under transient engine operating conditions.

Over a period of up to 4 seconds, the engine idled at a load range of 1.2 to 1.9 bar IMEP, with a constant maximum combustion pressure of approximately 6.5 bar and a maximum heat release rate of 20 kJ/m³ °CA. Operation at the load necessary to maintain idling, despite the small variability of Pmax and dQmax, was characterized by high variability in the angle of maximum pressure APmax. The irregularity of the APmax position determined the variability in the start of combustion (MBF10), but did not significantly affect the duration of the first phase of combustion (MBF10–MBF50) or the second phase (MBF50–MBF90). It was observed that during idling, both combustion phases had a similar duration.

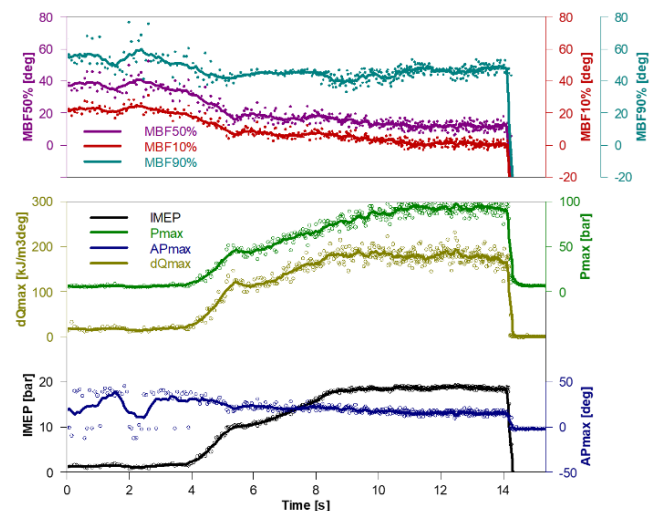


Fig. 12. Transient in cylinder thermodynamics indicators of the combustion process under partial boost conditions

The increase in engine load from 4 to approximately 8.2 s, corresponding to an increase in IMEP from 1.9 to less than 18 bar, significantly affected the stabilization of the APmax angle and the change in the share of individual combustion phases. As the load increased, the values of Pmax (up to approx. 72 bar) and dQmax (up to approx. 180 kJ/m³ °CA) increased, while the angles APmax and MBF10 accelerated systematically from 31 to 25°CA aTDC and from 16 to 7°CA aTDC, respectively. The acceleration of combustion initiation was accompanied by a corresponding shift in the MBF50 center of combustion. At the same time, an elongation of the second combustion phase from approximately 15 to 30°CA was observed, while the MBF90 angle showed an upward trend (from approx. 42°CA to 45°CA), opposite to the changes in the start and center of combustion. With increasing load, there was a clear reduction in the MBF90 spread, but not in MBF10 or MBF50.

The combustion process at full load, up to approximately 14 seconds of the test and at IMEP $\approx$ 18.5 bar, was characterized by roughly constant values of Pmax and dQmax, amounting to about 92 bar and 182 kJ/m³ °CA, respectively. The APmax angle accelerated further from 22 to 18.5°CA aTDC, accompanied by a shift in MBF10 from 7 to around 0°CA and an acceleration in MBF50 from 18 to 11°CA aTDC. At full load, a renewed rise in MBF90 dispersion was observed despite earlier stabilization. Full-load operation extended the second combustion phase from around 23 to 36°CA, while the first phase remained almost constant at around 11°CA.

The observed redistribution of the duration of individual combustion phases is confirmed by the analysis of pressure and heat release rate curves presented in the next subsection. Figure 13 shows the changes in the angular duration of individual combustion phases, determined from MBF curves. Blue triangles mark the MBF10–50 range, red squares mark MBF50–90, and black diamonds mark the total combustion period MBF10–90.

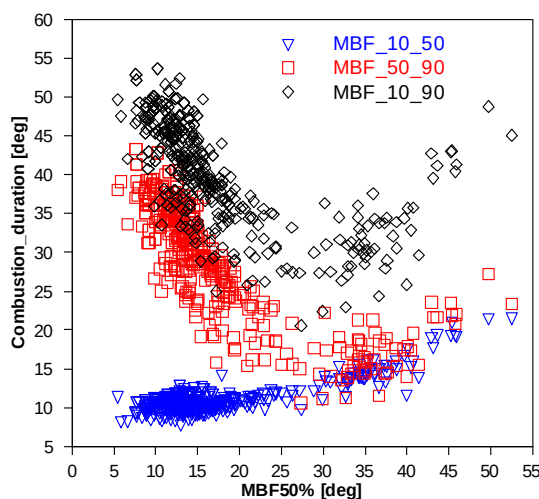


Fig. 13. Relationship between combustion duration intervals and MBF50 phasing under transient partial boost conditions

In the initial phase of engine operation, a wide spread of MBF10–90 values was observed, due to combustion instability under transient conditions. As the load increased, the

total combustion time shortened gradually, and the proportion of the second combustion phase (MBF50–90) decreased markedly. At the same time, the first combustion phase (MBF10–50) showed a slight prolongation and stabilization of the course.

The observed changes indicate a transition from angularly extended combustion at low load to a more concentrated combustion at higher IMEP values.

Figure 14 shows the relationship between the selected parameter describing the combustion process and the engine operating conditions during transient states. In the initial phase, a large spread of values was observed, characteristic of unstable combustion at low load. As the load increased, the curve gradually became more orderly and shifted to a region of lower variability, corresponding to stable combustion at higher IMEP values.

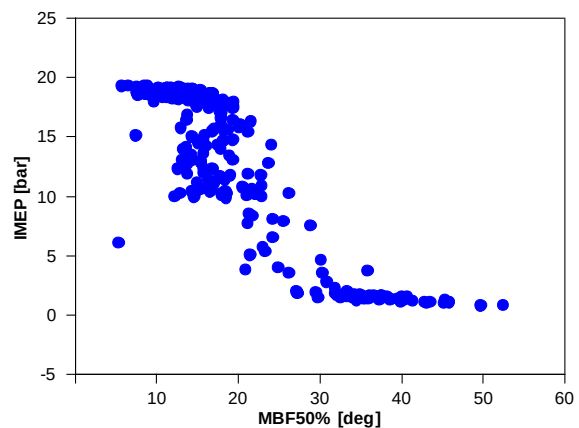


Fig. 14. Relationship between combustion duration intervals and MBF50 phasing under transient partial boost conditions

4.5. In-cylinder pressure phenomena in different states of engine operation

When the throttle was opened and the fuel dose was increased to maintain the set λ value, a gradual increase in cylinder pressure and heat release rate was observed (Fig. 15). Despite the acceleration of the ignition advance angle, the combustion process clearly proceeded after TDC, and the position of the maximum pressure angle APmax did not show a clear tendency to occur earlier. At the same time, the maximum heat release rate, dQ/d α , occurred earlier in each subsequent cycle. The visible irregularity of the dQ/d α curve indicated instability in the combustion process, characteristic of transitional states of engine operation.

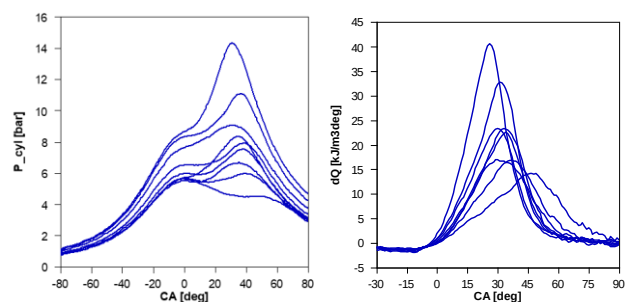


Fig. 15. In-cylinder pressure and heat release rate for selected initial acceleration cycles under partial boost conditions

A detailed thermodynamic analysis of the selected engine operating cycle at idle (Fig. 16) showed an extended discharge phase lasting approximately 11.5°CA. As a result, the combustion process was prolonged, leading to delayed combustion. This was caused by a significant proportion of combustion occurring after TDC and by an extended period of increased heat release rate. This pattern indicates the dominance of combustion initiated by ignition under conditions of heterogeneous charge preparation.

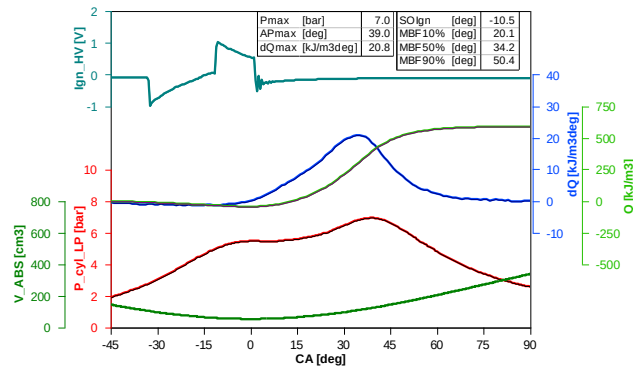


Fig. 16. Thermodynamic analysis of a representative engine cycle during idle operation

Figure 17 shows the pressure curves in the cylinder and the heat release rates for selected cycles, illustrating the changes during an increase in engine load. Increasing the load causes a clear increase in the maximum value of $dQ/d\alpha$ and a shift in the combustion process towards TDC. At the same time, the irregularity of the $dQ/d\alpha$ curves disappeared, indicating stabilization of the combustion process and a more kinetic reaction.

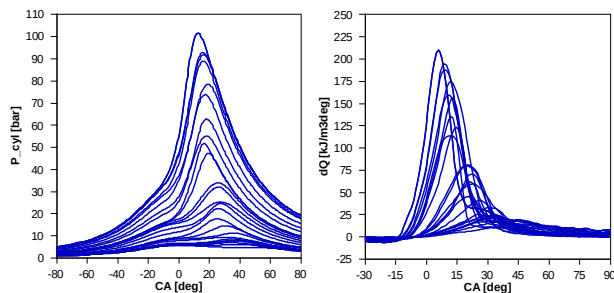


Fig. 17. In-cylinder pressure and heat release rate for selected cycles representing increasing load over the entire test

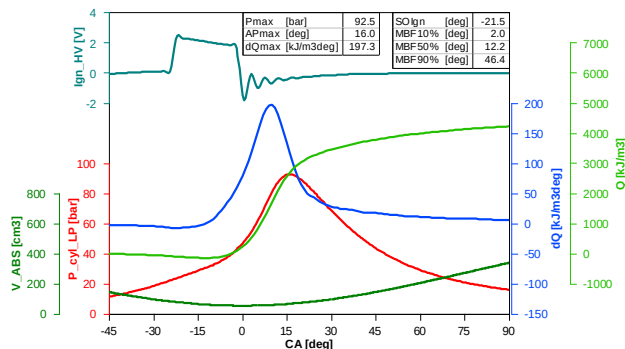


Fig. 18. Thermodynamic analysis of a representative engine cycle during full-load operation

A detailed thermodynamic analysis (Fig. 18) indicated an extension of the angular discharge period to approximately 18°CA. Combustion was kinetic, with a homogeneous mixture, and the increase in heat release rate was more rapid, although the maximum Q values were reached slightly later than in the initial acceleration phase.

5. Summary and conclusions

The performed thermodynamic indicator analysis demonstrated that in-cylinder pressure measurements constitute a significant extension of traditional motorsport engine calibration procedures. Conventional tuning based on external parameters such as torque, air-fuel ratio or exhaust gas temperature enables safe engine operation; however, it does not provide direct information about combustion phasing or thermodynamic loading of engine components. The results obtained in this study confirm that indicator measurements allow a more precise evaluation of combustion behaviour in a highly modified production-based racing engine and create a link between empirical dyno tuning and advanced experimental diagnostics. The main findings of the study are summarized as follows:

- 1) In the tested setup, the engine achieved 889 kW at 6222 rpm and 1488 Nm at 4393 rpm, confirming the high-performance characteristics required in drifting applications.
- 2) Limiting the boost control duty cycle to 30% reduced the engine output to 508 kW and 889 Nm. Under these conditions, the measured in-cylinder pressure and overall engine load were significantly lower, thereby minimizing the risk of damage to the spark plug-integrated pressure transducer during indication measurements.
- 3) The ability to directly link transient (temporal) parameters with the analysis of successive operating cycles was achieved, enabling the evaluation of combustion evolution during load ramp-up.
- 4) Under idle conditions, significant variability in the APmax angle and MBF phases was observed, with high repeatability in the heat release rate (HRR) curve.
- 5) Under transitional conditions, with increasing load, stabilization of the APmax angle and a reduction in MBF90 dispersion were observed, while MBF50 variability remained significant in the initial phase of load increase.
- 6) At full load, increased variation in maximum heat release rate (HRR) and Pmax was observed, indicating high sensitivity of the combustion process under WOT conditions.
- 7) In the case of traditional tuning without engine indication, at full load, an average MBF50 value of 13.8°CA was obtained, slightly delayed relative to MBT, but ensuring a safe operating margin; at the same time, $COV(MBF50) = 17.3\%$ indicated significant cycle-to-cycle variability. The use of indicator-based tuning could bring the combustion phasing closer to MBT and reduce uniqueness at maximum IMEP and WOT.
- 8) Engine tuning is a significant extension of traditional calibration methods, enabling improved tuning quality, increased combustion process repeatability, and potential improvements in engine performance and stability.

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Nomenclature

APmax	angle of maximum in-cylinder pressure	ICE	internal combustion engine
aTDC	after top dead centre	IMEP	indicated mean effective pressure
BDC	bottom dead centre	IAT	intake air temperature
CAD	computer-aided design	LMP1	Le Mans Prototype 1
CI	compression ignition	MAP	manifold absolute pressure
CNG	compressed natural gas	MAPO	maximum amplitude of pressure oscillation
COV	coefficient of variation	MBF	mass burn fraction
DI	direct injection	MBT	minimum advance for best torque
DOHC	double overhead camshaft	PFI	port fuel injection
DTM	Deutsche Tourenwagen Masters	Pmax	maximum in-cylinder pressure
ECU	engine control unit	SI	spark ignition
EGT	exhaust gas temperature	SOC	start of combustion
FIA	Fédération Internationale de l'Automobile	TCR	Touring Car Racing
GT3	Grand Touring class 3	TDC	top dead centre
HRR	heat release rate	WOT	wide open throttle

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Injection timing – driven redistribution of combustion intensity and pressure-induced mechanical loading in a turbocharged diesel engine

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Injection timing effects on combustion intensity redistribution and pressure-induced mechanical loading were evaluated for a turbocharged diesel engine using a calibrated Diesel-RK simulation model. Engine simulations were performed at multiple load conditions with systematic variation of injection timing. Advancing injection timing shifted combustion toward top dead center, accelerating pressure development and increasing the maximum pressure rise rate from approximately 1.2–1.6 bar/°CA to values exceeding 5 bar/°CA under advanced conditions. The maximum piston gas force increased from 55–70 kN to over 100 kN at high load. The results demonstrated that mechanical loading depended on both pressure magnitude and pressure-development rate, indicating a coupled combustion–pressure interaction rather than a single-parameter response. The results showed that mechanical loading depended on both pressure magnitude and pressure-rise behaviour rather than on a single pressure parameter. Increasing pressure-rise rate intensified piston gas-force response, particularly under advanced injection and high-load operation. The results also indicated that combustion-optimal injection timing did not necessarily correspond to mechanically favourable operating conditions.

Key words: injection timing, combustion–pressure coupling, pressure-rise dynamics, turbocharged diesel engine

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1. Introduction

Diesel engines operating with compression ignition remain the preferred choice for many heavy-duty transport and power-generation applications, largely because they combine efficient fuel conversion with dependable performance across a broad range of operating conditions [1]. The timing of fuel injection is one of the calibration parameters that most directly affects combustion behavior [2]. Advancing or retarding the fuel injection tends to shift the combustion timing with respect to top dead center, which affects the in-cylinder pressure history and the rate at which pressure builds during combustion [3].

Changes in cylinder pressure provide a direct indication of how combustion develops throughout the engine cycle and of the mechanical loads experienced by engine components [4]. The maximum cylinder pressure represents the highest load transferred through the piston, connecting rod, and crankshaft, while the pressure rise rate describes how rapidly this load is generated during combustion [5]. As a result, the choice of injection timing affects both engine performance and emissions, as well as the magnitude and characteristics of the resulting mechanical loading [6].

In conventional engine calibration, limits on peak cylinder pressure and pressure rise rate are applied to ensure mechanical reliability [7]. These limits are generally considered design constraints rather than responses to changes in the combustion process [8]. Different injection timings result in distinct combustion patterns and cylinder pressure traces [9]. As a result, the resulting mechanical loading is determined by the combustion process rather than treated as an independent operating constraint [10].

Despite extensive research on injection timing effects, the relationship among combustion intensity, pressure development, and mechanical loading remains poorly understood across different operating conditions. The present

study evaluates combustion intensity and pressure-based loading behavior under variations in injection timing using pressure-derived indicators and operating-condition mapping.

2. Literature review

Injection timing has been examined extensively in experimental studies and numerical simulations [11]. The reported results usually describe changes in ignition delay, heat release rate, peak cylinder pressure, pressure rise rate, fuel consumption, and nitrogen oxide emissions [12]. Earlier fuel injection generally produced stronger premixed combustion and higher cylinder pressure [13]. When injection was delayed, pressure developed more gradually, while thermal efficiency declined [14].

Peak cylinder pressure and maximum pressure rise rate are widely used as calibration constraints because they reflect combustion severity and mechanical loading [15]. These parameters define the acceptable operating range to limit structural loading and combustion noise [16]. Most previous studies assessed engine operation by comparing the calculated values with these limits.

However, in most cases, combustion development and mechanical constraints are treated separately. Previous investigations characterized combustion intensity using heat release or efficiency-based measures [17]. Peak pressure and pressure rise rate were typically considered calibration limits rather than variables linked to combustion development. Consequently, the effect of injection timing on the redistribution of combustion intensity and the associated pressure-induced loading has not been systematically evaluated [18].

Previous research has investigated diesel combustion under different injection strategies and operating conditions [19–21]. Most of these studies evaluated combustion characteristics and stability, whereas the associated pressure loading received less attention. Their analyses were gener-

ally limited to individual operating conditions, making it difficult to describe the overall response to changes in injection timing. Consequently, the transition from moderate to severe combustion could not be identified consistently from pressure-based measurements.

This work examines the influence of injection timing on combustion intensity and the associated pressure-induced mechanical loading. Pressure-derived indicators are used to evaluate the evolution of combustion and pressure development under different operating conditions. The analysis identifies load-dependent trends in the pressure response over the investigated injection timing range.

4. Novelty and contribution

The present study evaluates the relationships among combustion intensity, pressure-rise response, and mechanical loading under injection timing variations using pressure-based indicators. Unlike conventional approaches that rely mainly on peak cylinder pressure, the analysis considers both pressure magnitude and pressure-rise response during combustion development.

The analysis also considers the influence of engine load under different injection-timing conditions to distinguish operating regions associated with favorable performance characteristics from those characterized by elevated mechanical loading. The novelty of the present work lies in the combined interpretation of combustion intensity and pressure-based loading response across different operating conditions.

4. Materials and methods

4.1. Engine model, operating conditions and injection strategy

Simulations were performed in Diesel-RK for a turbocharged four-cylinder diesel engine under steady-state conditions. Engine geometry, turbocharging parameters, and boundary conditions were taken from manufacturer data and a validated calibration dataset (Appendix Table A1). The model was calibrated at the rated peak-torque condition (1600 rpm) by adjusting the injected fuel mass per cycle until the predicted brake torque matched 480 N·m. Using a compressor pressure ratio of $PR = 1.33$ and $SOI = 8^\circ$ bTDC, a fuel quantity of approximately 0.1059 g/cycle produced a torque of about 482 N·m, indicating close agreement with the target condition. The operating matrix was constructed over five engine speeds (1000, 1500, 2000, 2500, and 3000 rpm) and four load levels (25%, 50%, 75%, and 100%), resulting in 120 steady-state simulation cases ($5 \times 4 \times 6$). For each speed–load combination, injection timing (SOI) was varied from 0° to 20° CA bTDC in 4° CA increments.

During each injection timing sweep, the injected fuel quantity, injection pressure, and boundary conditions were held constant, and only the injection timing was varied. The recorded outputs included peak cylinder pressure (p_{max}), maximum pressure rise rate ($(dp/d\theta)_{max}$), IMEP, brake power, torque, BSFC, and NO_x emissions.

4.2. Combustion intensity and mechanical loading indicators

Combustion behavior and mechanical loading were characterized using pressure-derived indicators obtained

from the in-cylinder pressure trace. The maximum pressure rise rate $(dp/d\theta)_{max}$, was used as a measure of combustion intensity, representing the rate of pressure development during the premixed combustion phase. Peak cylinder pressure (p_{max}) was used to represent the maximum thermodynamic loading acting on the piston–connecting rod–crankshaft system.

The loading response was analyzed as a function of injection timing to identify operating regions where small phasing variations resulted in significant increases in pressure gradients or peak pressure.

4.3. Mathematical formulation of combustion intensity and mechanical loading metrics

The apparent heat release rate (aHRR) was calculated using the first law of thermodynamics as:

$$\frac{d\dot{Q}_{app}}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta} \quad (1)$$

The pressure rise rate was determined numerically as:

$$\frac{dp}{d\theta} \approx \frac{p_{i+1} - p_i}{\theta_{i+1} - \theta_i} \quad (2)$$

where: p_i and p_{i+1} represent consecutive pressure values at crank angles θ_i and θ_{i+1} , respectively. Combustion phasing (CA50) was defined as:

$$\frac{\int_{SOC}^{CA50} \dot{Q}_{app} d\theta}{\int_{SOC}^{EOC} \dot{Q}_{app} d\theta} = 0.5 \quad (3)$$

where EOC denotes the end of combustion. The mechanical loading index (MLI) was defined as the product of maximum in-cylinder pressure and maximum pressure-rise rate to represent combined pressure magnitude and loading severity:

$$MLI_{raw} = p_{max} \left(\frac{dp}{d\theta} \right)_{max} \quad (4)$$

The raw MLI values were normalized using min–max scaling before their incorporation into the Balanced Score formulation.

$$MLI_{norm} = \frac{MLI_{raw} - MLI_{min}}{MLI_{max} - MLI_{min}} \quad (5)$$

The combustion intensity index (CII) was defined to characterize the temporal concentration of heat release by relating the peak heat-release rate to the combustion duration:

$$CII_{raw} = \frac{HRR_{max}}{(CA95 - CA05)} \quad (6)$$

where HRR_{max} denotes the maximum net heat-release rate:

$$HRR_{max} = \left(\frac{dQ}{d\theta} \right)_{max} \quad (7)$$

CA05, CA50, and CA95 correspond to 5%, 50%, and 95% cumulative heat release, respectively. CA05 was adopted as the combustion-onset indicator. The CII values were normalized using min–max scaling according to:

$$CII_{norm} = \frac{CII_{raw} - CII_{min}}{CII_{max} - CII_{min}} \quad (8)$$

The analysis was based on a single-zone thermodynamic model assuming spatially uniform thermodynamic properties within the cylinder. Heat transfer, blow-by, and related losses were incorporated within the apparent heat-release formulation.

4.4. Performance normalization and evaluation criteria

To enable consistent comparison across operating conditions, performance parameters were normalized using a min–max scaling approach as:

$$X^* = \frac{X - X_{\min}}{X_{\max} - X_{\min}} \quad (9)$$

where $X_{\min}$ and $X_{\max}$ represent the minimum and maximum values over the complete simulation dataset. The normalized evaluation parameters consisted of brake thermal efficiency (BTE*), brake power (BP*), and brake specific fuel consumption (BSFC*). BTE* and BP* were treated as performance-enhancing parameters, whereas BSFC* was minimized. The normalized brake specific fuel consumption was defined as:

$$\text{BSFC}^* = 1 - \frac{\text{BSFC} - \text{BSFC}_{\min}}{\text{BSFC}_{\max} - \text{BSFC}_{\min}} \quad (10)$$

An aggregated performance score was introduced as:

$$\text{Perf}_{\text{Score}} = \frac{\text{BTE}^* + \text{BP}^* + \text{BSFC}^*}{3} \quad (11)$$

The performance score ($\text{Perf}_{\text{Score}}$) was used for comparative evaluation across operating conditions. To evaluate the operating conditions by simultaneously considering performance characteristics and mechanical loading severity, a balanced score was introduced as follows:

$$\text{Balanced Score} = 0.6 \text{Perf}_{\text{Score}} + 0.4(1 - \text{MLI}_{\text{norm}}) \quad (12)$$

The weighting factors in the Balanced Score were selected to reflect a performance-oriented calibration objective, prioritizing engine efficiency and output, while mechanical loading serves as a limiting constraint. A weight of 0.6 was assigned to the performance score, while a weight of 0.4 was assigned to the loading term.

4.5. Operating envelope construction

An operating envelope was constructed in a two-dimensional pressure space using peak cylinder pressure ($p_{\max}$) and maximum pressure rise rate $(dp/d\theta)_{\max}$. The injection timing sweep was mapped to identify transitions between moderate and high-intensity combustion regimes. The envelope was used to identify operating regions characterized by rapid increases in pressure gradients.

5. Results and discussion

Simulations were performed using the engine specifications (Appendix, Table 1) and operating conditions (Appendix, Table 2). Calibrated Diesel-RK outputs were used to extract performance, combustion, pressure development and gas force parameters. Raw and post-processed data are provided in Appendix Tables A3 and A4. The main analysis focuses on 2000 rpm for all loads with injection timing varied from 0°CA to 20°CA, while the remaining speed cases are presented in the Appendix.

5.1. Effect of injection timing on combustion structure evolution

At 2000 rpm, CA50 showed a greater response to injection-timing variation under low-load operation than under high-load conditions (Fig. 1). At 25% and 50% load, CA50 changed by only about 1°CA despite injection advance. As the load increased further, the response weakened considerably, and CA50 remained almost unchanged, staying near 18.8°CA at 75% load and about 23.5°CA at full load.

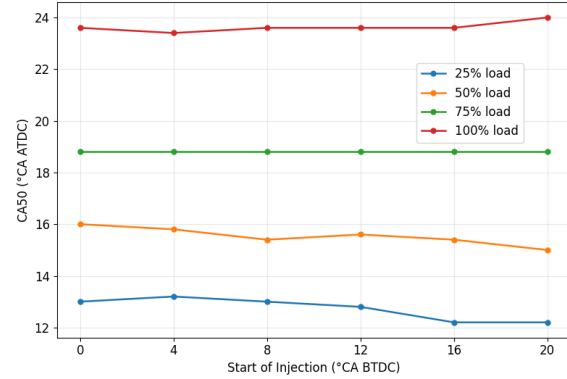


Fig. 1. Variation of combustion phasing (CA50) with injection timing at different engine loads at 2000 rpm

At low load, the longer ignition delay increases the premixing duration before ignition. Advancing the injection timing extends this period, producing a more concentrated heat-release event near top dead center and shifting the combustion centroid to earlier crank angles. With increasing load, the shorter ignition delay limits the accumulation of premixed fuel. Under these conditions, variations in injection timing no longer produce a proportional shift in the combustion centroid, leading to the observed clustering of CA50.

Figure 2 shows that injection timing altered both combustion phasing and combustion duration. At early combustion phasing ($\text{CA50} \approx 12\text{--}13^\circ\text{CA aTDC}$), the combustion duration remains short, indicating a concentrated heat-release event. As CA50 shifts toward $15\text{--}16^\circ\text{CA aTDC}$, combustion duration increases, indicating a transitional combustion regime. A markedly different response emerges

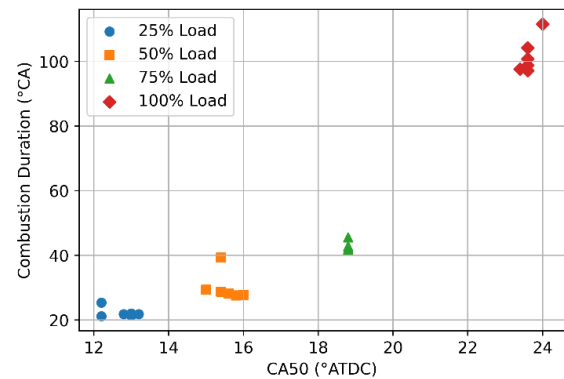


Fig. 2. Combustion structure evolution represented by the relationship between CA50 and combustion duration under injection timing variations at 2000 rpm

beyond approximately 18.8°CA aTDC, where the combustion duration increases further and eventually expands sharply at later phasing conditions.

4.2. Redistribution of combustion intensity

Figure 3 shows that the response of normalized combustion intensity to injection timing was strongly load-dependent and distinctly non-monotonic. At 25% load, it remained relatively high over the entire SOI range. At 50% load, a clear minimum appears at intermediate SOI, followed by a marked increase at more advanced timing, indicating a change in the heat-release pattern. At 75% load, the variation is smaller, with a gradual increase and a slight decline. At 100% engine load, the values remain low across all conditions. The relatively high values at low and intermediate loads indicate more temporally concentrated heat release.

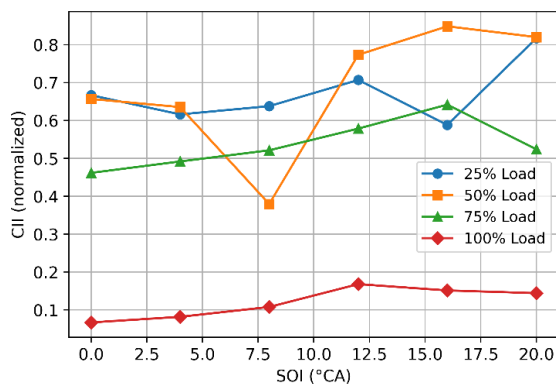


Fig. 3. Variation of normalized combustion intensity index (CII) with injection timing at 2000 rpm for different engine loads

Increasing load raised HRR_{max} , but also extended combustion duration through a stronger diffusion-controlled phase. At high load, the extended combustion duration reduced CII_{norm} despite the higher fuel quantity. Non-monotonic behaviour appeared mainly at low and intermediate loads because injection timing affected heat-release development differently across load conditions. At certain SOI values, advancing injection increased premixed combustion intensity, whereas at others it broadened the heat-release process and increased combustion duration.

Figure 4 summarizes the maximum attainable combustion intensity across the speed-load domain together with the corresponding optimal injection timing. At low load, high normalized CII values approaching unity were obtained, particularly at 1000–1500 rpm, whereas at high load the achievable intensity remained substantially lower, with values typically below 0.3 at low and intermediate speeds and reaching only about 0.4 at higher speeds.

At low load, the combustion event remained temporally compact, allowing a large fraction of the energy to be released within a narrow crank-angle interval. At high load, the extended diffusion-controlled phase broadened the combustion interval and reduced temporal concentration despite the higher energy release.

The optimal combustion structure varied across the investigated operating conditions. At lower speeds, higher CII values were sustained due to a longer available time for

rapid energy release, whereas higher speeds reduced the achievable concentration, particularly when combined with high load.

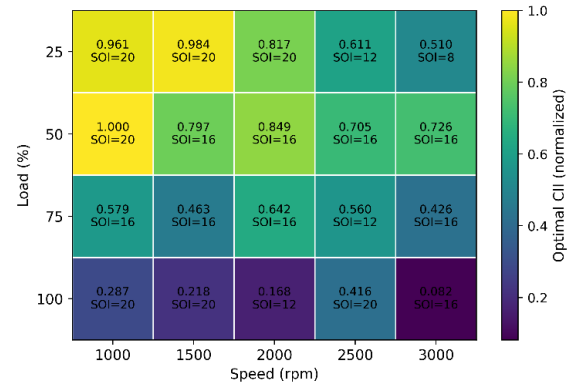


Fig. 4. Speed-load map of combustion intensity redistribution at the optimal injection timing condition

The corresponding optimal injection timing values, indicated in Fig. 4, vary systematically across the operating domain. At low load (25%), the optimal timing is predominantly advanced ($SOI \approx 20^\circ CA$) at low and intermediate speeds, shifting toward more retarded values ($SOI \approx 12-8^\circ CA$) at higher speeds. At medium load (50%), the optimal timing stabilizes around $SOI \approx 16^\circ CA$ across most speeds. A similar trend is observed at 75% load, where $SOI \approx 16^\circ CA$ remains dominant, with slight shifts toward $12^\circ CA$ at higher speeds. At full load (100%), the optimal timing spans a broader range ($SOI \approx 12-20^\circ CA$), reflecting the increased sensitivity of combustion structure to operating conditions. Accordingly, the map illustrates how combustion characteristics varied across the speed-load domain under different injection timing conditions.

5.3. Injection timing – induced pressure evolution

5.3.1. Evolution of maximum cylinder pressure

Following the redistribution of combustion intensity described in Section 4.2, the resulting response of in-cylinder pressure development was evaluated under different injection timing conditions. The analysis considers peak cylinder pressure and pressure rise rate as the main indicators of pressure-driven engine response.

The variation of maximum cylinder pressure with injection timing at 2000 rpm exhibits a monotonic increase with injection advance. This response became more pronounced at higher loads as heat release shifted toward higher-pressure conditions. As shown in Fig. 5, peak pressure increases from about 53–61 bar at retarded injection to around 82–106 bar at advanced injection, with the highest values at full load. Advancing injection timing shifted heat release toward the compression–expansion transition at higher pressure. The pressure increase across the injection timing range grew from about 25–30 bar at low load to more than 40 bar at full load. However, peak pressure reflects the overall pressure magnitude rather than the temporal distribution of heat release. Comparison with Fig. 3 shows that while CII varied strongly with load and injection timing, peak pressure increased more uniformly with injection advance. Although peak pressure increased monotonically with injection advance, it did not

fully reflect the variation in heat-release intensity and pressure-rise response.

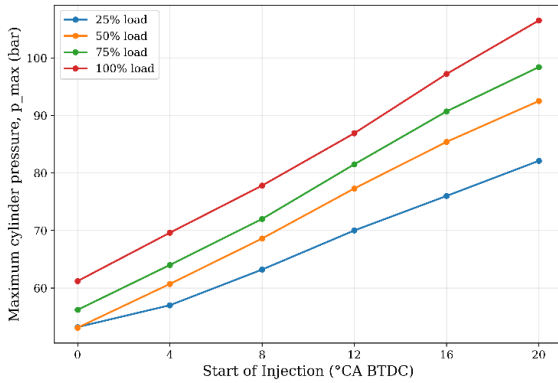


Fig. 5. Variation of maximum cylinder pressure with injection timing at different engine loads (2000 rpm)

5.3.2. Development of maximum pressure rise rate

The variation in the maximum pressure rise rate with injection timing at 2000 rpm showed a strong, load-dependent, nonlinear response. As shown in Fig. 6, $(dp/d\theta)_{max}$ increased from approximately 1.2–1.5 bar/°CA under retarded injection conditions to about 3.4–5.5 bar/°CA at advanced injection timing, with the highest values observed at high load. Earlier injection extended the premixed fuel–air preparation period before ignition, increasing the fraction of rapidly released energy near the start of combustion. Consequently, relatively small phasing shifts produced disproportionately large increases in pressure-rise rate, particularly under high-load operation, where the larger fuel quantity intensified premixed combustion. Comparison with Fig. 3 indicates that the pressure-rise response followed combustion-intensity redistribution more closely than peak cylinder pressure. Accordingly, the pressure-rise response provided clearer differentiation of load-severity than peak pressure alone.

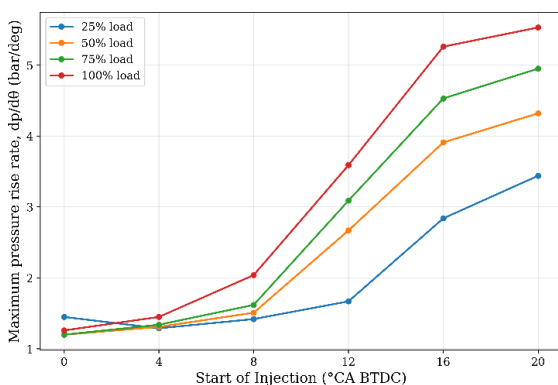


Fig. 6. Effect of injection timing on maximum pressure rise rate at different engine loads (2000 rpm)

5.4. Combustion intensity–mechanical loading coupling

5.4.1. Combustion intensity influence on pressure development

The following analysis examines the relationships among combustion intensity, pressure development, and mechanical loading as injection timing varies. Mechanical

loading was evaluated using peak pressure, pressure-rise rate, and the mechanical loading index (MLI).

Figure 7 shows that the relationship between normalized combustion intensity and maximum pressure rise rate remained strongly dependent on engine load across the investigated operating conditions. At 25% and 50% load, relatively high CII_{norm} values were generally associated with higher pressure rise rates, whereas at 75% and especially 100% load, the pressure rise rate continued to increase, even though CII_{norm} remained moderate or very low. For example, at 100% load, CII_{norm} remained below about 0.17, while the maximum pressure rise rate exceeded 5 bar/°CA.

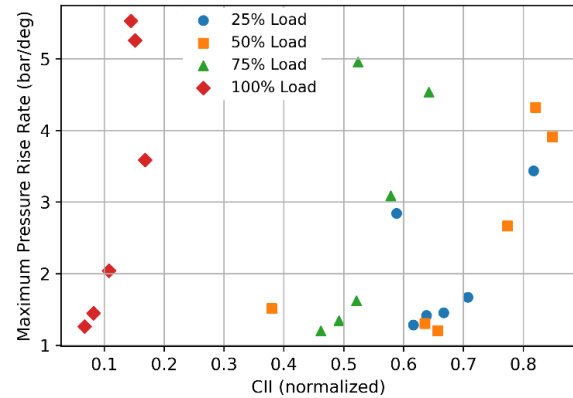


Fig. 7. Relationship between combustion intensity index and maximum pressure rise rate at different engine loads (2000 rpm)

At low and intermediate loads, a higher CII_{norm} reflects a more compact heat-release event, in which a relatively large fraction of the released energy is concentrated within a short crank-angle interval. Accordingly, low- and medium-load conditions showed higher pressure-rise response with increasing CII_{norm} .

At higher loads, the combustion characteristics change. The duration of combustion becomes longer, and the diffusion-controlled phase becomes more pronounced, which lowers CII_{norm} by reducing the temporal concentration of heat release. At the same time, the larger fuel quantity and stronger overall heat release increase the absolute pressure response, so the maximum pressure rise rate continues to increase even when the normalized combustion intensity remains low.

Accordingly, similar combustion intensities produced different pressure-rise responses across load conditions because both the heat-release concentration and the total released energy influenced pressure development. As a result, the 100% load cases occupied the low- CII_{norm} , high- $(dp/d\theta)_{max}$ region, whereas lower-load conditions remained associated with higher CII_{norm} values at comparatively lower pressure-rise rates. These results indicate that the relationship between combustion intensity and pressure-rise response depended strongly on operating conditions.

5.4.2. Pressure rise–driven mechanical loading response

Figure 8 shows the variation of maximum gas force with the maximum pressure-rise rate under the investigated operating conditions. Gas force increased from approximately 50–70 kN at low pressure-rise rates (~ 1 – 2 bar/°CA) to values exceeding 100 kN as $(dp/d\theta)_{max}$ approached 5 bar/°CA.

Despite the overall increase, the relationship remained load-dependent. Similar pressure-rise rates produced different gas-force levels across the investigated loads. High-load operation occupied the upper gas-force region because larger fuel quantities generated higher absolute cylinder pressure even under comparable pressure-rise conditions.

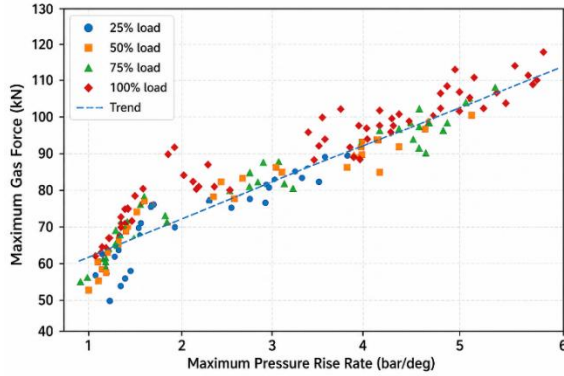


Fig. 8. Relationship between maximum pressure-rise rate and maximum gas force under different operating conditions

The spread of operating points reflects variations in combustion phasing and pressure development. The results also show that the pressure-rise rate alone could not fully represent mechanical loading severity, since gas-force generation depended on both the pressure-rise response and the absolute pressure level. This combined effect is incorporated later through the mechanical loading index (MLI).

5.4.3. Mechanical loading index response

Figure 9 shows that MLI_{norm} increased with injection advance across all loads, with a stronger response at high loads. At 25% load, MLI_{norm} varied within a limited range (≈ 0.04 – 0.40), whereas at full load it increased to approximately 0.93, indicating a stronger mechanical loading response at high load.

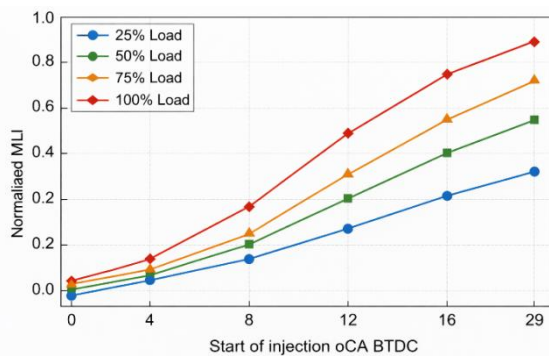


Fig. 9. Mechanical loading index variation with injection timing at different engine loads (2000 rpm)

At high load, identical injection-timing shifts produced larger increases in MLI_{norm} because of the combined effect of pressure-rise response and energy release. Combustion phasing influenced pressure development, whereas load controlled the magnitude of the pressure response. Higher fuel energy intensified pressure development and accelerated the increase in mechanical loading at high load.

4.5. Trade-off between performance and mechanical loading

Figure 10 shows the trade-off between performance and mechanical loading at 2000 rpm for different load conditions. Higher performance was accompanied by increased mechanical loading. Operating points with moderate performance (~ 0.45) corresponded to low loading levels (≤ 0.1), whereas high-performance conditions (~ 0.7) were associated with substantially higher loading (≥ 0.7). At 25% load, operation remained within a low-loading region with limited performance gains, whereas at 100% load, both performance and loading increased substantially.

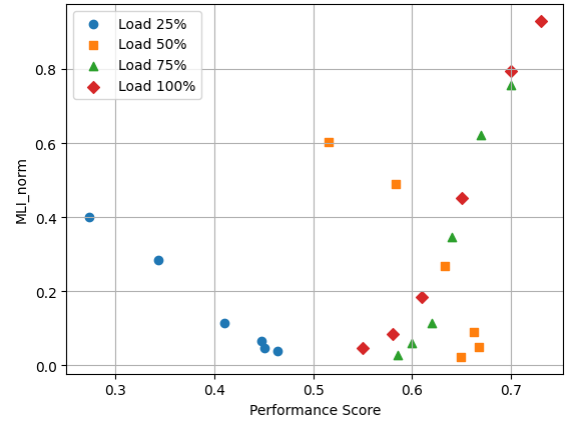


Fig. 10. Performance–mechanical loading trade-off map at 2000 rpm under different engine load conditions

Higher performance was associated with stronger pressure development and increases in both p_{max} and $(dp/d\theta)_{max}$, which amplified mechanical loading. Advanced injection produced stronger pressure development at high load.

At 25% load, the lower fuel quantity limited pressure development. At 50–75% load, sensitivity to injection timing increased as premixed combustion became more pronounced. At 100% load, the response became steeper because of the simultaneous increase in p_{max} and $(dp/d\theta)_{max}$. Higher CII values were associated with more concentrated heat release and steeper pressure gradients, whereas at high load, the increase in loading became increasingly influenced by energy release.

4.6. Optimal injection timing envelope

Two optimality criteria were considered in this study: a combustion-based optimum derived from CII and a performance-loading optimum based on the Balanced Score. The optimal injection timing varied with engine speed and load. Figure 4 represents combustion-based optima, whereas the present section considers combined performance and loading conditions.

Figure 11 shows the optimal injection timing at 2000 rpm based on the Balanced Score, which combines performance and mechanical loading. The optimal operating conditions were identified using the Balanced Score ranking. The data formed distinct load-dependent trajectories across the investigated conditions. The Balanced Score response varied with load and injection timing.

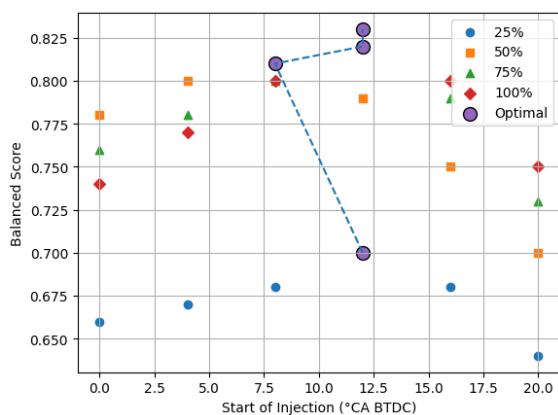


Fig. 11. Optimal injection timing envelope at 2000 rpm

At low load (25%), the response to injection timing remains limited, reflecting insufficient energy release to generate strong pressure development, and thus only marginal variation in the Balanced Score. In contrast, at intermediate loads (50–75%), a well-defined optimal region emerges around SOI $\approx$ 8–12°CA bTDC, where the balance between performance and mechanical loading was most favorable. At full load (100%), the trend shifts: excessive injection advancement leads to a deterioration in the Balanced Score, despite high performance levels, due to rapid mechanical loading.

This response arises from the coupling between combustion structure and pressure development. Advancing the injection increased combustion intensity and the pressure-rise rate, thereby increasing mechanical loading. At high load, larger fuel quantities increased the pressure response even under less concentrated combustion conditions. Similar $p_{\max}$ values were associated with different pressure-rise responses under different load conditions. The optimal points shifted with load due to the combined effects of combustion phasing and energy release.

These results indicate that performance improvement must be considered together with mechanical loading during engine calibration. The optimal injection timing identified in Fig. 11 differed from the combustion-based optimum shown in Fig. 4 because the Balanced Score included both performance and mechanical loading. Combustion-optimal conditions did not always correspond to the lowest mechanical loading conditions.

5.7. Physical interpretation of phasing-driven loading mechanism

The preceding results show that mechanical loading was influenced by changes in combustion intensity and pressure-rise response under injection timing variations. Advancing injection timing increased combustion intensity

(CII) and pressure-rise rate, $(dp/d\theta)_{\max}$. Similar peak-pressure levels were associated with different pressure-rise responses and mechanical loading responses.

Injection timing controls structural loading by influencing combustion-zone evolution. Moderate injection advance improved combustion phasing and engine performance. Further injection advance increased $(dp/d\theta)_{\max}$ more rapidly than performance improvement, leading to higher mechanical loading.

6. Conclusions

The present study investigated the relationships among combustion characteristics, pressure development, and mechanical loading in a turbocharged diesel engine under injection timing variations. The results showed that combustion-optimal injection timing did not always correspond to the lowest mechanical loading conditions, indicating that injection timing selection depends strongly on operating condition and loading response. Mechanical loading was more strongly influenced by pressure-rise response than by peak pressure alone, particularly under advanced injection and high-load conditions. The main conclusions are summarized as follows:

1. Injection timing modified combustion phasing and changed the distribution of heat release during the engine cycle. Advancing the injection moved the main combustion phase toward earlier crank angles and increased the temporal concentration of heat release.
2. Combustion intensity strongly influenced pressure development, where the maximum pressure rise rate increased from approximately 1.2–1.5 bar/°CA under retarded injection to values exceeding 5 bar/°CA at advanced injection and high load.
3. The maximum gas force exceeded 100 kN under full-load operation. The results further showed that pressure-rise response provided a more representative indication of combustion-driven loading severity than peak pressure alone.
4. The mechanical loading index (MLI) increased with injection advance across the investigated conditions. Advanced injection improved engine performance but simultaneously increased pressure-induced mechanical loading.
5. The optimal injection timing therefore, varied with load and operating condition. Moderate injection advance produced near-maximum performance while maintaining lower mechanical loading than highly advanced injection conditions.

The CII and MLI indicators enabled comparison of combustion characteristics and mechanical loading across operating conditions.

Nomenclature

aHRR	apparent heat release rate, J/°CA
BP	brake power, kW
BSFC	brake specific fuel consumption, kg/kWh
BTE	brake thermal efficiency, %
CA05	crank angle at 5% heat release, °CA
CA50	crank angle at 50% cumulative heat release, °CA

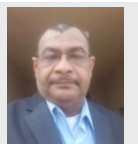
CA95	crank angle at 95% heat release, °CA
CII	combustion intensity index, –
CII _{norm}	normalized combustion intensity index, –
dQ/dθ	heat release rate per crank angle, J/°CA
EOC	end of combustion, °CA
F _{max}	maximum piston gas force, kN

HRR	heat release rate, J/°CA	$p_{\max}$	maximum in-cylinder pressure, bar
$HRR_{\max}$	maximum heat release rate, J/°CA	$Perf_{\text{Score}}$	performance score, –
IMEP	indicated mean effective pressure, bar	PR	turbocharger pressure ratio, –
Load (L)	engine load, %	Q	heat release, J
MLI	mechanical loading index, –	SOI	start of injection, °CA bTDC
MLI_{norm}	normalized mechanical loading index, –	TDC	top dead center, –
N	engine speed, rpm	$(dp/d\theta)_{\max}$	maximum pressure rise rate, bar/°CA
$p(\theta)$	instantaneous in-cylinder pressure, bar		

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Appendix

Table A1. Specifications of the investigated engine and applied numerical settings

Engine type	4HK1 – 5.2L Turbo Diesel	EVO / EVC	48°C a bBDC / 14°C aTDC
No. of cylinder	4-cylinder, In-line	Connecting rod length	220 mm
Valve mechanism	16-valve, chain drive	Injection timing (SOI)	20°C a bTDC (baseline setting)
Combustion Chamber	Bowl-in-piston	Injection duration (equivalent crank angle, based on 4 ms)	At 1000 rpm → 24 °CA
Fuel system	Direct injection (common rail)		At 2000 rpm → 48 °CA
Displacement	5193 cm ³		At 3000 rpm → 72 °CA
Bore × Stroke	115 mm × 125 mm		At 4000 rpm → 96 °CA
Compression ratio	16.5: 1	Nozzle hole diameter	≈ 0.18 mm
Max output power	136 kW at 2600 rpm	Spray cone angle	150°
Max torque	480 N · m at 1600 rpm	Number of nozzle holes	7
IVO / IVC	19°C a bTDC / 53°C a bBDC	Ambient parameters	P ₀ = 1 bar, T ₀ = 313 K

Table A2. Load definition and corresponding fuel injection quantities at various engine speeds

Engine speed [rpm]	Load [%]	Target torque [N · m]	Fuel mass per cycle [g/cycle]
1000	25	107.8	0.0295
	50	215.5	0.0497
	75	335.3	0.0700
	100	437.5	0.1200
1500	25	135.8	0.0307
	50	242.5	0.0498
	75	358.3	0.0735
	100	475.2	0.1100
1600 (Reference)	25	120.5	0.0245
	50	241.0	0.0457
	75	361.5	0.0730
	100	482.0	0.1059
2000	25	122.3	0.0296
	50	241.5	0.0508
	75	361.8	0.0727
	100	478.0	0.1033
2500	25	116.2	0.0307
	50	242.0	0.0520
	75	350.2	0.0730
	100	470.9	0.1000
3000	25	091.1	0.0282
	50	245.0	0.0547
	75	346.7	0.0768
	100	462.3	0.1100

Table A3. Raw performance, combustion phasing, and pressure-trace indicators for the SOI sweep matrix (1000–3000 rpm; 25–100% load)

Case n_Load_SOI	BP	BTE	BSFC	Ignition delay	CA5	CA50	CA95	pmax	Apmax	dp/dθ	HRR _{max}
rpm_ %_ °CA bTDC	kW	–	kg/kWh	°CA	°CA aTDC	°CA aTDC	°CA aTDC	bar	°CA aTDC	bar/°CA	J/°CA
1000rpm_25%_SOI=0	13.0	0.312	0.2716	4.1	2	12.4	24	51.4	0	1.41	67.30
1000rpm_25%_SOI=4	13.2	0.315	0.2689	4.0	1.8	12.4	23.8	57.2	8	1.30	65.30
1000rpm_25%_SOI=8	13.0	0.311	0.2722	4.1	1.8	12	23.2	64.5	8	1.45	66.00
1000rpm_25%_SOI=12	12.6	0.302	0.2803	4.7	1.8	11.4	22.2	72.2	7	2.36	73.60
1000rpm_25%_SOI=16	12.0	0.287	0.2954	5.9	2.8	11.2	22.4	78.9	5	3.22	79.30
1000rpm_25%_SOI=20	11.3	0.271	0.3123	7.7	3.6	10.8	24.8	83.4	3	3.82	90.70
1000rpm_50%_SOI=0	25.1	0.357	0.2373	3.9	2.2	14.6	29.8	53.7	11	1.19	76.50
1000rpm_50%_SOI=4	25.5	0.362	0.2342	3.7	2.2	14.6	29.6	62.1	11	1.31	92.00
1000rpm_50%_SOI=8	25.4	0.361	0.2349	3.8	2	14.4	29.4	71.3	9	1.59	90.40
1000rpm_50%_SOI=12	24.8	0.353	0.2402	4.3	2	13.8	29	80.6	8	3.05	89.90
1000rpm_50%_SOI=16	23.9	0.340	0.2495	5.4	2.6	13.6	29	89.3	6	4.08	113.89
1000rpm_50%_SOI=20	22.7	0.322	0.2627	7.1	3.6	13.2	32.4	96.3	4	4.71	126.90
1000rpm_75%_SOI=0	34.6	0.349	0.2425	4.1	2.2	18	53.4	50.5	11	1.01	106.00
1000rpm_75%_SOI=4	35.5	0.358	0.2365	3.9	2.2	17.8	51.4	58.2	10	1.19	103.50
1000rpm_75%_SOI=8	35.2	0.355	0.2389	4.1	2.2	17.6	62.2	66.6	10	1.87	112.40
1000rpm_75%_SOI=12	35.5	0.358	0.2368	5.0	2	17	50.4	75.1	9	3.22	130.60
1000rpm_75%_SOI=16	34.7	0.350	0.2422	6.3	3.4	17.6	51	83.5	7	4.59	143.90
1000rpm_75%_SOI=20	33.3	0.336	0.2519	8.1	4.2	17.6	57.2	90.2	5	4.74	146.90

Table A3. cont

Case n_Load_SOI	BP	BTE	BSFC	Ignition delay	CA5	CA50	CA95	pmax	Apmax	dp/dθ	HRR _{max}
rpm_°CA bTDC	kW	–	kg/kWh	°CA	°CA aTDC	°CA aTDC	°CA aTDC	bar	°CA aTDC	bar/°CA	J/°CA
1000rpm_100%_SOI=0	44.0	0.259	0.3273	2.8	3	23.6	119	57.0	11	1.17	130.60
1000rpm_100%_SOI=4	45.2	0.266	0.3189	2.8	3	23.6	117.8	65.5	9	1.37	129.10
1000rpm_100%_SOI=8	45.8	0.270	0.3143	3.0	2.8	23.4	115.4	74.0	8	2.21	127.90
1000rpm_100%_SOI=12	45.9	0.270	0.3136	3.4	2.4	22.8	111.8	83.0	7	3.91	142.90
1000rpm_100%_SOI=16	45.3	0.267	0.3177	4.4	2.2	22	107.8	92.0	7	4.45	176.20
1000rpm_100%_SOI=20	44.0	0.259	0.3273	5.7	3.4	22.4	107.6	100.5	6	4.93	215.50
1500rpm_25%_SOI=0	20.7	0.317	0.2674	6.5	3.2	12.4	24.2	45.7	0	1.25	77.40
1500rpm_25%_SOI=4	21.2	0.325	0.2608	5.8	2.8	12.4	24.4	51.8	8	1.12	74.00
1500rpm_25%_SOI=8	21.3	0.327	0.2589	6.0	2.8	12.4	24.4	58.3	8	1.27	76.20
1500rpm_25%_SOI=12	21.1	0.323	0.2619	7.0	3.4	12.2	24.6	65.0	7	1.93	82.40
1500rpm_25%_SOI=16	20.7	0.317	0.2669	8.7	4	11.6	27	71.0	6	2.92	84.40
1500rpm_25%_SOI=20	20.2	0.309	0.2738	11.3	4	12	23.8	76.6	5	3.48	86.20
1500rpm_50%_SOI=0	36.8	0.348	0.2434	4.6	2	15	31.6	49.0	11	1.05	90.50
1500rpm_50%_SOI=4	37.7	0.357	0.2376	4.6	2	15.2	31.6	55.9	9	1.17	91.40
1500rpm_50%_SOI=8	38.1	0.360	0.2354	4.9	2	14.8	31.8	63.6	9	1.41	96.30
1500rpm_50%_SOI=12	37.9	0.358	0.2365	5.8	3	15.2	33	71.6	8	2.60	108.60
1500rpm_50%_SOI=16	37.3	0.352	0.2405	7.4	3.8	15	34.4	79.0	7	4.10	114.40
1500rpm_50%_SOI=20	36.4	0.344	0.2463	9.6	4.4	14.2	36	85.9	5	4.29	113.50
1500rpm_75%_SOI=0	53.9	0.345	0.2452	3.9	2.4	19.2	53.4	51.6	11	1.07	105.30
1500rpm_75%_SOI=4	55.4	0.355	0.2387	4.0	2.4	19.2	54.2	59.1	9	1.22	107.80
1500rpm_75%_SOI=8	56.4	0.361	0.2348	4.2	2.2	19.2	54.2	67.4	10	1.84	115.30
1500rpm_75%_SOI=12	56.5	0.362	0.2340	4.9	2.2	18.6	54.4	76.0	9	3.13	126.40
1500rpm_75%_SOI=16	56.0	0.358	0.2363	6.3	3.4	19	56.4	84.5	7	4.50	140.10
1500rpm_75%_SOI=20	54.8	0.351	0.2414	8.1	4.2	19.2	59.4	91.7	6	4.57	142.30
1500rpm_100%_SOI=0	70.8	0.303	0.2797	3.8	2.6	23	110	59.4	11	1.21	129.70
1500rpm_100%_SOI=4	73.2	0.313	0.2705	3.7	2.6	23.2	108.2	67.6	9	1.40	129.80
1500rpm_100%_SOI=8	74.7	0.319	0.2652	4.0	2.6	23	108.2	76.0	8	2.14	137.90
1500rpm_100%_SOI=12	75.2	0.322	0.2631	4.4	2.2	22.6	107.8	85.4	8	3.83	153.40
1500rpm_100%_SOI=16	72.8	0.312	0.2718	5.5	3	22.8	136.8	95.0	8	4.98	179.20
1500rpm_100%_SOI=20	73.4	0.314	0.2699	7.2	4	22.8	113.2	104.2	6	5.61	200.80
2000rpm_25%_SOI=0	24.4	0.290	0.2917	6.7	3.4	13	24.8	53.2	0	1.45	70.90
2000rpm_25%_SOI=4	25.2	0.301	0.2817	6.1	3	13.2	24.8	57.0	8	1.29	68.60
2000rpm_25%_SOI=8	25.6	0.305	0.2773	6.3	3.2	13	25.2	63.2	6	1.42	70.80
2000rpm_25%_SOI=12	25.6	0.306	0.2772	7.1	3.6	12.8	25.4	70.0	6	1.67	75.10
2000rpm_25%_SOI=16	25.3	0.302	0.2805	8.7	4.2	12.2	29.6	76.0	5	2.84	77.60
2000rpm_25%_SOI=20	24.9	0.297	0.2850	11.2	3.8	12.2	25	82.1	5	3.44	80.70
2000rpm_50%_SOI=0	49.1	0.341	0.2483	4.6	2.4	16	30.2	53.1	10	1.20	91.20
2000rpm_50%_SOI=4	50.6	0.352	0.2409	4.6	2.2	15.8	29.8	60.7	8	1.31	88.60
2000rpm_50%_SOI=8	50.6	0.352	0.2409	4.9	2.2	15.4	41.6	68.6	9	1.51	93.40
2000rpm_50%_SOI=12	51.6	0.359	0.2362	5.4	2.8	15.6	31	77.3	8	2.67	103.30
2000rpm_50%_SOI=16	51.0	0.355	0.2389	6.7	3.4	15.4	32.2	85.4	7	3.91	112.60
2000rpm_50%_SOI=20	50.0	0.348	0.2437	8.7	4.4	15	33.8	92.5	6	4.32	112.20
2000rpm_75%_SOI=0	71.5	0.347	0.2441	4.5	2.4	18.8	45.2	56.2	10	1.20	113.00
2000rpm_75%_SOI=4	74.1	0.360	0.2354	4.5	2.4	18.8	44.2	64.0	8	1.34	114.50
2000rpm_75%_SOI=8	75.7	0.368	0.2305	4.7	2.4	18.8	44.4	72.0	9	1.62	119.10
2000rpm_75%_SOI=12	76.5	0.371	0.2282	5.2	2.8	18.8	44.8	81.5	9	3.09	127.00
2000rpm_75%_SOI=16	76.2	0.370	0.2289	6.5	3.6	18.8	46.4	90.7	7	4.53	138.30
2000rpm_75%_SOI=20	75.1	0.364	0.2325	8.5	4.2	18.8	49.8	98.4	6	4.95	129.70
2000rpm_100%_SOI=0	93.7	0.320	0.2647	4.3	2.6	23.6	99.8	61.2	11	1.26	130.70
2000rpm_100%_SOI=4	97.4	0.333	0.2544	4.3	2.6	23.4	100.2	69.6	8	1.45	136.00
2000rpm_100%_SOI=8	100.2	0.342	0.2474	4.5	2.6	23.6	101.4	77.8	7	2.04	146.10
2000rpm_100%_SOI=12	101.9	0.348	0.2433	5.0	2.8	23.6	103.6	86.9	8	3.59	169.00
2000rpm_100%_SOI=16	102.2	0.349	0.2427	6.1	3.4	23.6	107.6	97.2	7	5.26	169.00
2000rpm_100%_SOI=20	101.0	0.345	0.2456	7.8	4.2	24	115.8	106.5	6	5.53	178.40
2500rpm_25%_SOI=0	28.5	0.262	0.3230	5.7	3	14	26.2	49.5	0	1.35	64.40
2500rpm_25%_SOI=4	29.7	0.273	0.3101	5.7	3	14.2	26.4	53.3	8	1.21	64.50
2500rpm_25%_SOI=8	30.4	0.280	0.3029	6.0	3.2	14	26.4	59.3	7	1.33	67.90
2500rpm_25%_SOI=12	30.7	0.283	0.2997	6.8	3.6	13.8	26.8	65.8	7	1.57	72.60
2500rpm_25%_SOI=16	30.7	0.282	0.3003	8.5	4.2	13	32.2	71.7	5	2.74	75.40
2500rpm_25%_SOI=20	30.4	0.279	0.3033	11.0	4	12.8	28.6	77.5	5	3.29	77.00
2500rpm_50%_SOI=0	59.6	0.324	0.2616	5.2	2.8	16.8	32.8	50.7	11	1.14	91.40
2500rpm_50%_SOI=4	62.0	0.337	0.2515	5.1	2.8	17	32.2	57.7	8	1.23	91.90

Table A3. cont

Case n_Load_SOI	BP	BTE	BSFC	Ignition delay	CA5	CA50	CA95	pmax	Apmax	dp/dθ	HRR _{max}
rpm_°CA bTDC	kW	–	kg/kWh	°CA	°CA aTDC	°CA aTDC	°CA aTDC	bar	°CA aTDC	bar/°CA	J/°CA
2500rpm_50%_SOI=8	63.5	0.345	0.2458	5.4	3	17	32.6	64.6	8	1.43	97.10
2500rpm_50%_SOI=12	64.1	0.348	0.2435	6.1	3.4	17	33.6	72.5	8	2.39	103.30
2500rpm_50%_SOI=16	63.9	0.347	0.2442	7.6	4.2	16.8	35.4	80.3	7	3.76	107.30
2500rpm_50%_SOI=20	62.6	0.340	0.2491	9.9	4.4	16	44.4	86.7	5	3.98	105.30
2500rpm_75%_SOI=0	85.6	0.331	0.2559	4.8	2.6	19.4	45.2	57.0	10	1.24	111.90
2500rpm_75%_SOI=4	89.1	0.345	0.2459	4.8	2.6	19.6	44.8	64.5	8	1.37	112.20
2500rpm_75%_SOI=8	91.7	0.355	0.2389	5.0	2.8	19.8	44.6	72.3	8	1.62	116.70
2500rpm_75%_SOI=12	93.1	0.360	0.2353	5.6	3.2	19.8	45.4	81.2	8	2.90	125.00
2500rpm_75%_SOI=16	90.9	0.351	0.2410	6.7	3.8	19.8	62.6	90.3	7	4.29	133.20
2500rpm_75%_SOI=20	92.1	0.356	0.2378	8.7	4.4	19.8	51.4	98.0	6	4.69	134.30
2500rpm_100%_SOI=0	114.0	0.322	0.2631	4.4	2.8	23.2	67.6	65.7	10	1.44	126.00
2500rpm_100%_SOI=4	119.4	0.337	0.2513	4.4	2.8	23	65.6	74.4	8	1.58	126.30
2500rpm_100%_SOI=8	123.3	0.348	0.2434	4.6	2.6	22.8	65.6	83.1	7	1.91	134.00
2500rpm_100%_SOI=12	125.7	0.355	0.2386	5.1	3	23.2	66.6	92.3	8	3.52	142.50
2500rpm_100%_SOI=16	126.6	0.357	0.2370	5.8	3.4	23.2	68.6	103.1	7	4.78	159.50
2500rpm_100%_SOI=20	125.7	0.355	0.2388	7.5	4.2	23.4	73.2	113.2	6	5.56	171.70
3000rpm_25%_SOI=0	26.2	0.218	0.3879	5.6	3	14	26.2	50.1	0	1.37	58.80
3000rpm_25%_SOI=4	27.7	0.231	0.3659	5.6	3	14	25.2	53.0	7	1.24	59.20
3000rpm_25%_SOI=8	28.6	0.239	0.3548	5.9	3.2	13.8	25.6	58.7	6	1.34	62.70
3000rpm_25%_SOI=12	28.4	0.237	0.3581	6.6	3.6	14	34.2	64.4	6	1.53	65.20
3000rpm_25%_SOI=16	28.3	0.236	0.3582	8.3	4.2	13.6	33.8	69.8	5	2.58	68.30
3000rpm_25%_SOI=20	28.1	0.235	0.3612	10.7	4.2	13.2	30.8	74.9	5	2.96	68.60
3000rpm_50%_SOI=0	71.4	0.307	0.2757	5.1	2.6	17.2	33.2	54.0	10	1.24	94.40
3000rpm_50%_SOI=4	74.6	0.321	0.2641	5.1	2.6	17.4	33.4	61.1	7	1.33	95.10
3000rpm_50%_SOI=8	77.0	0.331	0.2558	5.3	2.8	17.4	33.2	68.0	7	1.52	98.80
3000rpm_50%_SOI=12	78.4	0.337	0.2512	5.9	3.4	17.6	34	75.9	8	2.43	102.90
3000rpm_50%_SOI=16	78.5	0.338	0.2508	7.2	4	17.6	35	84.1	7	3.90	108.70
3000rpm_50%_SOI=20	77.7	0.334	0.2534	9.2	4.6	17.2	37.6	90.9	6	4.20	106.40
3000rpm_75%_SOI=0	99.8	0.306	0.2770	4.9	2.8	21.2	54.4	55.8	10	1.23	110.90
3000rpm_75%_SOI=4	105.1	0.322	0.2630	4.9	2.6	21.4	52.4	63.2	8	1.34	110.00
3000rpm_75%_SOI=8	108.9	0.334	0.2539	5.1	3	21.6	53	70.3	7	1.58	113.60
3000rpm_75%_SOI=12	111.5	0.342	0.2480	5.7	3.4	21.8	53.6	78.4	8	2.76	122.30
3000rpm_75%_SOI=16	112.3	0.344	0.2462	7.0	4	22	56.4	87.0	7	4.45	132.20
3000rpm_75%_SOI=20	111.4	0.341	0.2483	9.0	4.6	22	61.6	94.6	6	4.45	132.20
3000rpm_100%_SOI=0	131.4	0.281	0.3013	4.7	2.8	26.2	120.8	64.2	10	1.38	135.10
3000rpm_100%_SOI=4	139.3	0.298	0.2843	4.7	2.8	26.2	114.8	72.6	8	1.54	134.10
3000rpm_100%_SOI=8	145.3	0.311	0.2725	4.9	2.8	26.2	113	80.4	6	2.30	139.40
3000rpm_100%_SOI=12	149.3	0.319	0.2652	5.4	3.2	26.6	115.6	88.8	7	3.39	149.90
3000rpm_100%_SOI=16	151.5	0.324	0.2613	6.2	3.8	27	120.2	98.6	7	5.16	162.20
3000rpm_100%_SOI=20	151.1	0.323	0.2620	7.8	4.4	27.2	129.4	108.2	6	5.32	169.50

Table A4. Calculated combustion duration, gas-force-based loading indicators (F_{max} , MLI), combustion intensity index (CII), and normalized performance metrics (BTE*, BP*, BSFC*) with the resulting Perf_Score and Balanced Score for all SOI sweep cases

Case n (rpm)_Load (%)_SOI°	Combustion duration °CA	Fmax kN	CII _{raw} J/°CA ²	CII _{norm}	MLI _{raw} bar ² /°CA	MLI _{norm}	BP*	BTE*	BSFC*	Perf _{Score}	Balanced score
1000rpm_25%_SOI=0	22	53.46	3.059	0.589	72.24	0.037	0.012	0.612	0.728	0.4507	0.6558
1000rpm_25%_SOI=4	22	59.47	2.968	0.562	74.21	0.040	0.013	0.632	0.745	0.4635	0.6621
1000rpm_25%_SOI=8	21.4	67.07	3.084	0.597	93.69	0.074	0.012	0.607	0.724	0.4478	0.6391
1000rpm_25%_SOI=12	20.4	75.02	3.608	0.757	170.35	0.206	0.009	0.549	0.674	0.4105	0.5637
1000rpm_25%_SOI=16	19.6	82.08	4.046	0.890	254.28	0.352	0.005	0.447	0.579	0.3436	0.4655
1000rpm_25%_SOI=20	21.2	86.74	4.278	0.961	319.00	0.464	0.000	0.346	0.474	0.2733	0.3785
1000rpm_50%_SOI=0	27.6	55.82	2.772	0.502	63.62	0.022	0.098	0.907	0.943	0.6493	0.7808
1000rpm_50%_SOI=4	27.4	64.61	3.358	0.680	81.12	0.052	0.101	0.938	0.962	0.6671	0.7794
1000rpm_50%_SOI=8	27.4	74.14	3.299	0.663	113.60	0.108	0.100	0.931	0.958	0.6630	0.7544
1000rpm_50%_SOI=12	27	83.79	3.330	0.672	246.10	0.338	0.096	0.879	0.925	0.6335	0.6451
1000rpm_50%_SOI=16	26.4	92.86	4.314	0.972	364.44	0.542	0.090	0.793	0.867	0.5832	0.5330
1000rpm_50%_SOI=20	28.8	100.11	4.406	1.000	453.15	0.696	0.081	0.681	0.784	0.5155	0.4310
1000rpm_75%_SOI=0	51.2	52.48	2.070	0.288	51.00	0.000	0.166	0.857	0.910	0.6445	0.7867
1000rpm_75%_SOI=4	49.2	60.51	2.104	0.298	69.01	0.031	0.172	0.915	0.948	0.6786	0.7947
1000rpm_75%_SOI=8	60	69.29	1.873	0.228	124.35	0.127	0.170	0.891	0.933	0.6647	0.7481
1000rpm_75%_SOI=12	48.4	78.08	2.698	0.480	242.19	0.331	0.172	0.912	0.946	0.6766	0.6737
1000rpm_75%_SOI=16	47.6	86.85	3.023	0.579	383.22	0.575	0.166	0.860	0.912	0.6460	0.5577
1000rpm_75%_SOI=20	53	93.73	2.772	0.502	427.34	0.651	0.157	0.772	0.852	0.5934	0.4956
1000rpm_100%_SOI=0	116	59.24	1.126	0.000	66.38	0.027	0.233	0.264	0.379	0.2922	0.5647
1000rpm_100%_SOI=4	114.8	68.12	1.125	0.000	89.46	0.067	0.241	0.309	0.432	0.3277	0.5700
1000rpm_100%_SOI=8	112.6	76.93	1.136	0.003	163.57	0.195	0.246	0.335	0.461	0.3472	0.5304
1000rpm_100%_SOI=12	109.4	86.26	1.306	0.055	324.67	0.473	0.247	0.338	0.465	0.3499	0.4206
1000rpm_100%_SOI=16	105.6	95.70	1.669	0.166	409.55	0.620	0.242	0.316	0.439	0.3325	0.3514
1000rpm_100%_SOI=20	104.2	104.45	2.068	0.287	495.32	0.769	0.233	0.265	0.380	0.2925	0.2680
1500rpm_25%_SOI=0	21	47.48	3.686	0.780	57.01	0.010	0.067	0.644	0.754	0.4882	0.6888
1500rpm_25%_SOI=4	21.6	53.86	3.426	0.701	57.91	0.012	0.070	0.697	0.796	0.5209	0.7078
1500rpm_25%_SOI=8	21.6	60.62	3.528	0.732	74.24	0.040	0.071	0.712	0.808	0.5302	0.7021
1500rpm_25%_SOI=12	21.2	67.59	3.887	0.842	125.29	0.129	0.070	0.688	0.789	0.5155	0.6579
1500rpm_25%_SOI=16	23	73.78	3.670	0.776	207.45	0.271	0.067	0.648	0.757	0.4906	0.5861
1500rpm_25%_SOI=20	19.8	79.62	4.354	0.984	266.62	0.373	0.063	0.595	0.714	0.4575	0.5253
1500rpm_50%_SOI=0	29.6	50.92	3.057	0.589	51.26	0.000	0.182	0.849	0.905	0.6452	0.7869
1500rpm_50%_SOI=4	29.6	58.17	3.088	0.598	65.23	0.025	0.188	0.904	0.941	0.6780	0.7969
1500rpm_50%_SOI=8	29.8	66.10	3.232	0.642	89.82	0.067	0.191	0.926	0.955	0.6907	0.7876
1500rpm_50%_SOI=12	30	74.48	3.620	0.760	186.57	0.235	0.189	0.915	0.948	0.6842	0.7167
1500rpm_50%_SOI=16	30.6	82.16	3.739	0.797	323.73	0.472	0.185	0.876	0.923	0.6614	0.6081
1500rpm_50%_SOI=20	31.6	89.26	3.592	0.752	368.62	0.550	0.179	0.822	0.887	0.6291	0.5577
1500rpm_75%_SOI=0	51	53.68	2.065	0.286	55.00	0.007	0.304	0.831	0.893	0.6762	0.8030
1500rpm_75%_SOI=4	51.8	61.45	2.081	0.291	72.27	0.037	0.315	0.894	0.934	0.7142	0.8138
1500rpm_75%_SOI=8	52	70.07	2.217	0.333	124.03	0.126	0.321	0.932	0.959	0.7375	0.7920
1500rpm_75%_SOI=12	52.2	79.00	2.421	0.395	238.01	0.324	0.322	0.940	0.963	0.7419	0.7157
1500rpm_75%_SOI=16	53	87.90	2.643	0.463	380.15	0.569	0.318	0.917	0.949	0.7281	0.6091
1500rpm_75%_SOI=20	55.2	95.33	2.578	0.443	419.10	0.637	0.310	0.867	0.917	0.6980	0.5641
1500rpm_100%_SOI=0	107.4	61.71	1.208	0.025	71.58	0.036	0.424	0.553	0.677	0.5513	0.7166
1500rpm_100%_SOI=4	105.6	70.32	1.229	0.032	94.74	0.076	0.441	0.621	0.735	0.5991	0.7292
1500rpm_100%_SOI=8	105.6	79.04	1.306	0.055	162.92	0.194	0.452	0.661	0.768	0.6271	0.6988
1500rpm_100%_SOI=12	105.6	88.80	1.453	0.100	327.06	0.478	0.456	0.678	0.781	0.6383	0.5919
1500rpm_100%_SOI=16	133.8	98.72	1.339	0.065	472.88	0.730	0.439	0.610	0.727	0.5919	0.4632
1500rpm_100%_SOI=20	109.2	108.35	1.839	0.218	584.72	0.923	0.442	0.625	0.739	0.6022	0.3920
2000rpm_25%_SOI=0	21.4	55.33	3.313	0.667	77.39	0.046	0.093	0.471	0.602	0.3887	0.6149
2000rpm_25%_SOI=4	21.8	59.22	3.147	0.616	73.20	0.038	0.099	0.539	0.665	0.4341	0.6451
2000rpm_25%_SOI=8	22	65.74	3.218	0.638	89.59	0.067	0.102	0.570	0.692	0.4546	0.6461
2000rpm_25%_SOI=12	21.8	72.75	3.445	0.707	116.85	0.114	0.102	0.571	0.693	0.4551	0.6275
2000rpm_25%_SOI=16	25.4	79.04	3.055	0.588	215.97	0.285	0.100	0.547	0.672	0.4396	0.5496
2000rpm_25%_SOI=20	21.2	85.40	3.807	0.817	282.16	0.400	0.097	0.516	0.645	0.4193	0.4916
2000rpm_50%_SOI=0	27.8	55.25	3.281	0.657	64.02	0.023	0.269	0.803	0.874	0.6488	0.7803
2000rpm_50%_SOI=4	27.6	63.12	3.210	0.636	79.23	0.049	0.280	0.873	0.921	0.6912	0.7952
2000rpm_50%_SOI=8	39.4	71.27	2.371	0.380	103.68	0.091	0.280	0.872	0.921	0.6910	0.7781
2000rpm_50%_SOI=12	28.2	80.32	3.663	0.774	205.98	0.268	0.287	0.918	0.950	0.7183	0.7237
2000rpm_50%_SOI=16	28.8	88.82	3.910	0.849	334.32	0.490	0.283	0.891	0.933	0.7023	0.6253
2000rpm_50%_SOI=20	29.4	96.12	3.816	0.820	399.76	0.603	0.276	0.845	0.903	0.6747	0.5635
2000rpm_75%_SOI=0	42.8	58.41	2.640	0.462	67.55	0.029	0.429	0.842	0.900	0.7238	0.8229
2000rpm_75%_SOI=4	41.8	66.55	2.739	0.492	85.98	0.061	0.448	0.926	0.955	0.7763	0.8416
2000rpm_75%_SOI=8	42	74.88	2.836	0.521	116.87	0.114	0.459	0.976	0.986	0.8070	0.8386
2000rpm_75%_SOI=12	42	84.71	3.024	0.579	251.58	0.347	0.464	1.000	1.000	0.8214	0.7541
2000rpm_75%_SOI=16	42.8	94.26	3.231	0.642	410.98	0.623	0.463	0.993	0.996	0.8170	0.6411
2000rpm_75%_SOI=20	45.6	102.35	2.844	0.524	487.74	0.756	0.455	0.956	0.973	0.7945	0.5745
2000rpm_100%_SOI=0	97.2	63.64	1.345	0.067	77.37	0.046	0.587	0.666	0.772	0.6749	0.7867
2000rpm_100%_SOI=4	97.6	72.38	1.393	0.082	100.77	0.086	0.614	0.750	0.836	0.7332	0.8055
2000rpm_100%_SOI=8	98.8	80.85	1.479	0.108	158.71	0.186	0.634	0.812	0.880	0.7751	0.7905
2000rpm_100%_SOI=12	100.8	90.37	1.677	0.168	311.79	0.451	0.646	0.849	0.905	0.8001	0.6996
2000rpm_100%_SOI=16	104.2	101.02	1.622	0.151	510.81	0.796	0.648	0.855	0.909	0.8041	0.5642
2000rpm_100%_SOI=20	111.6	110.71	1.599	0.144	588.82	0.930	0.639	0.828	0.891	0.7862	0.4995
2500rpm_25%_SOI=0	23.2	51.47	2.776	0.503	66.94	0.028	0.123	0.287	0.406	0.2721	0.5522
2500rpm_25%_SOI=4	23.4	55.41	2.756	0.497	64.70	0.024	0.131	0.358	0.487	0.3255	0.5858
2500rpm_25%_SOI=8	23.2	61.64	2.927	0.549	78.68	0.048	0.136	0.401	0.532	0.3564	0.5947

Table A4cont.

Case n (rpm)_Load (%)_SOI ⁰	Combustion duration °CA	Fmax kN	CII _{raw} J/°CA ²	CII _{norm}	MLI _{raw} bar ² /°CA	MLI _{norm}	BP*	BTE*	BSFC*	Perf _{Score}	Balanced score
2500rpm_25%_SOI=12	23.2	68.36	3.129	0.611	103.08	0.090	0.138	0.421	0.552	0.3704	0.5862
2500rpm_25%_SOI=16	28	74.57	2.693	0.478	196.68	0.252	0.138	0.417	0.548	0.3677	0.5198
2500rpm_25%_SOI=20	24.6	80.53	3.130	0.611	255.06	0.353	0.136	0.399	0.530	0.3547	0.4716
2500rpm_50%_SOI=0	30	52.75	3.047	0.586	57.63	0.011	0.344	0.690	0.791	0.6085	0.7605
2500rpm_50%_SOI=4	29.4	59.95	3.126	0.610	71.06	0.035	0.361	0.775	0.854	0.6634	0.7841
2500rpm_50%_SOI=8	29.6	67.17	3.280	0.657	92.34	0.072	0.372	0.827	0.890	0.6962	0.7891
2500rpm_50%_SOI=12	30.2	75.37	3.421	0.700	173.31	0.212	0.376	0.847	0.904	0.7093	0.7409
2500rpm_50%_SOI=16	31.2	83.49	3.439	0.705	302.19	0.435	0.375	0.841	0.900	0.7054	0.6494
2500rpm_50%_SOI=20	40	90.18	2.633	0.459	345.55	0.510	0.366	0.797	0.869	0.6773	0.6025
2500rpm_75%_SOI=0	42.6	59.26	2.627	0.458	70.80	0.034	0.530	0.737	0.826	0.6977	0.8049
2500rpm_75%_SOI=4	42.2	67.10	2.659	0.467	88.35	0.065	0.554	0.826	0.889	0.7565	0.8280
2500rpm_75%_SOI=8	41.8	75.22	2.792	0.508	116.97	0.114	0.573	0.891	0.933	0.7991	0.8338
2500rpm_75%_SOI=12	42.2	84.44	2.962	0.560	235.38	0.319	0.583	0.927	0.955	0.8216	0.7654
2500rpm_75%_SOI=16	58.8	93.85	2.265	0.348	387.50	0.582	0.567	0.871	0.920	0.7858	0.6386
2500rpm_75%_SOI=20	47	101.87	2.857	0.528	459.93	0.707	0.576	0.902	0.940	0.8061	0.6007
2500rpm_100%_SOI=0	64.8	68.26	1.944	0.250	94.26	0.075	0.733	0.678	0.781	0.7307	0.8085
2500rpm_100%_SOI=4	62.8	77.34	2.011	0.270	117.65	0.115	0.771	0.777	0.855	0.8011	0.8345
2500rpm_100%_SOI=8	63	86.44	2.127	0.305	158.82	0.187	0.798	0.849	0.905	0.8508	0.8359
2500rpm_100%_SOI=12	63.6	96.01	2.241	0.340	324.79	0.474	0.816	0.894	0.935	0.8817	0.7395
2500rpm_100%_SOI=16	65.2	107.17	2.446	0.403	492.70	0.764	0.822	0.910	0.945	0.8925	0.6298
2500rpm_100%_SOI=20	69	117.64	2.488	0.416	628.94	1.000	0.815	0.893	0.934	0.8806	0.5284
3000rpm_25%_SOI=0	23.2	52.06	2.534	0.430	68.47	0.030	0.106	0.000	0.000	0.0353	0.4091
3000rpm_25%_SOI=4	22.2	55.13	2.667	0.470	65.64	0.025	0.117	0.086	0.137	0.1133	0.4579
3000rpm_25%_SOI=8	22.4	61.06	2.799	0.510	78.49	0.048	0.123	0.133	0.207	0.1545	0.4737
3000rpm_25%_SOI=12	30.6	66.95	2.131	0.307	98.68	0.082	0.121	0.119	0.187	0.1423	0.4524
3000rpm_25%_SOI=16	29.6	72.60	2.307	0.360	180.36	0.224	0.121	0.118	0.186	0.1417	0.3955
3000rpm_25%_SOI=20	26.6	77.92	2.579	0.443	221.80	0.296	0.120	0.106	0.167	0.1309	0.3603
3000rpm_50%_SOI=0	30.6	56.10	3.085	0.597	66.80	0.027	0.429	0.582	0.703	0.5710	0.7317
3000rpm_50%_SOI=4	30.8	63.51	3.088	0.598	81.18	0.052	0.451	0.670	0.775	0.6323	0.7585
3000rpm_50%_SOI=8	30.4	70.72	3.250	0.648	103.42	0.091	0.468	0.738	0.827	0.6778	0.7704
3000rpm_50%_SOI=12	30.6	78.94	3.363	0.682	184.66	0.231	0.478	0.777	0.856	0.7038	0.7298
3000rpm_50%_SOI=16	31	87.44	3.506	0.726	328.02	0.479	0.479	0.782	0.859	0.7066	0.6322
3000rpm_50%_SOI=20	33	94.54	3.224	0.640	382.22	0.573	0.473	0.759	0.842	0.6915	0.5857
3000rpm_75%_SOI=0	51.6	58.03	2.149	0.312	68.57	0.030	0.631	0.572	0.694	0.6324	0.7673
3000rpm_75%_SOI=4	49.8	65.69	2.209	0.330	84.92	0.059	0.669	0.678	0.782	0.7097	0.8024
3000rpm_75%_SOI=8	50	73.08	2.272	0.350	110.78	0.103	0.696	0.755	0.839	0.7632	0.8166
3000rpm_75%_SOI=12	50.2	81.55	2.436	0.400	216.76	0.287	0.715	0.807	0.876	0.7992	0.7648
3000rpm_75%_SOI=16	52.4	90.46	2.523	0.426	387.38	0.582	0.720	0.822	0.887	0.8099	0.6531
3000rpm_75%_SOI=20	57	98.33	2.319	0.364	420.68	0.640	0.714	0.804	0.874	0.7973	0.6225
3000rpm_100%_SOI=0	118	66.75	1.145	0.006	88.76	0.065	0.856	0.410	0.542	0.6030	0.7357
3000rpm_100%_SOI=4	112	75.48	1.197	0.022	111.92	0.105	0.913	0.521	0.649	0.6941	0.7743
3000rpm_100%_SOI=8	110.2	83.60	1.265	0.043	185.34	0.232	0.956	0.605	0.723	0.7613	0.7638
3000rpm_100%_SOI=12	112.4	92.28	1.334	0.064	300.47	0.432	0.984	0.661	0.768	0.8046	0.7101
3000rpm_100%_SOI=16	116.4	102.49	1.393	0.082	508.67	0.792	1.000	0.692	0.793	0.8282	0.5802
3000rpm_100%_SOI=20	125	112.44	1.356	0.070	574.91	0.906	0.997	0.687	0.788	0.8239	0.5318

Analysis of the variability of daily and monthly exposure to aircraft noise depending on the fleet structure – example of the 737-800 and 737 MAX 8

ARTICLE INFO

Aircraft noise is one of the most significant environmental concerns around airports, generating both social conflicts and financial consequences. This study examined how the fleet composition influences daily exposure to aircraft noise in the vicinity of Poznań-Ławica Airport, focusing on two widely operated aircraft types: the Boeing 737-800 and the newer 737 MAX 8. Measurements were carried out at six monitoring points during 78 operations, and the analysis was based on the sound exposure level expressed on the physical scale and in decibels. The results show that the 737-800 consistently produces higher noise levels than the MAX 8, with mean $L_{A_{SEL}}$ values of $0.89 \text{ Pa}^2\cdot\text{s}$ (94.5 dB) compared to $0.57 \text{ Pa}^2\cdot\text{s}$ (92.6 dB), a difference of 56%. The gap is particularly evident during take-off, where the 737-800 exceeds the MAX 8 by 137% (2.7 dB). Regression analysis confirmed a decreasing trend in SEL with increasing altitude; in the revised analysis, this dependence is modeled using a log-distance function in dB units, with altitude treated as a proxy for source–receiver distance. The simulation of nine fleet composition scenarios for a representative month showed that increasing the share of MAX 8 aircraft substantially reduces the number of days when the daily equivalent sound level (L_{AeqD}) exceeds the legal threshold of 60 dB. Economic analysis using official Polish penalty tariffs for 2025 also showed that monthly environmental fines could be reduced by up to 69% if the fleet were fully modernised to MAX 8. Even partial renewal, with a 60–70% share of MAX 8, reduced costs by 60–70% relative to a fleet dominated by the 737-800. These findings highlight that fleet modernisation is a highly effective strategy for reducing community exposure to aircraft noise and mitigating the financial burden on airports, although the analysis was limited to daytime operations and one seasonal period.

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1. Introduction

The increase in air traffic in recent decades has led to a growing problem of noise impacts in the vicinity of airports, alongside the harmful gaseous emissions [20]. One of the most striking examples of this phenomenon in Poland is Poznań-Ławica Airport, which has become the subject of broad public debate, numerous court proceedings, and environmental analyses regarding the exceedances of permissible noise levels in residential areas [7, 38].

The largest international airport in the region is located about 7 km west of the center of Poznań, while still within the city limits. It lies along Bukowska Street, which is also provincial road no. 307. The ICAO code of Poznań Airport is EPPO, while its IATA code is POZ [32, 38]. Both the runway and its apron, along with all airport facilities, are on flat terrain. The passenger terminal building was built at an elevation of 85–88 m above sea level, while the runway lies predominantly between 80 and 90 m above sea level.

The immediate surroundings of the airport consist mainly of residential areas with single-family houses and, to a lesser extent, of commercial and service areas. To the east of the airport, there are allotment garden complexes. To the south and southwest, a large portion of the area near the passenger terminal is occupied by large-capacity car parks for passengers. In addition, the southeast section is home to cargo and general aviation facilities.

In Poznań, in the Ławica district, as in other European airports, continuous, accredited monitoring of aircraft noise is carried out [6]. According to recent years, daily noise levels in protected zones fluctuated between 55 and 60 dB [10], with incidental exceedances of up to 5 dB above the

norm. In areas not covered by statutory protection, values can locally reach 65–70 dB, although no legal limits apply there [17]. The issue of aircraft noise has generated significant social consequences, as reflected in thousands of lawsuits filed against the airport operator. In 2022 alone, residents had filed more than 1700 claims, resulting in the award of substantial compensation for loss of property value and the costs of adapting buildings to higher noise levels (e.g., installation of soundproof windows) [1]. The total amount of compensation paid by Poznań-Ławica Airport between 2014 and 2023 exceeded PLN 142 million. This sum is more than three times the amount spent by all other Polish airports combined during the same period. Until mid-2022, the basis for granting compensation was solely the location of a property within the so-called limited use area (OOU) [11]. However, a landmark Supreme Court ruling in June 2022 clarified that location within the OOU alone is not sufficient – it is necessary to demonstrate an actual decrease in property value or the incurrence of real acoustic adaptation costs [32, 38]. This case law significantly narrowed the scope of airport claims and led to a considerable decline in the number of new lawsuits.

Exceedances of permitted noise levels at measurement points also result in the imposition of administrative fines, the amount of which depends on the scale of the exceedance and the number of points [8]. The problem of aircraft noise is the subject of numerous resident complaints, as illustrated by the scale of legal proceedings. Some members of the local community participate in initiatives aimed at developing compromise solutions, such as the introduction of designated approach routes or operating bans. However,

noise remains one of the main factors of social conflict surrounding the airport [2, 31].

In recent years, a clear trend has emerged in Europe toward the modernization of short-haul passenger aircraft fleets, driven both by increasingly stringent environmental requirements and by economic pressures associated with high fuel costs. Particular attention has been paid to the introduction of new aircraft variants, such as the Boeing 737 MAX 7, which, through the use of next-generation engines and optimized aerodynamics, offer substantial reductions in fuel consumption and noise emissions [5, 15].

Predictive models suggest that a 10% increase in fuel prices can accelerate fleet renewal by up to a factor of two within a 2–3-year horizon [28]. Market analyses indicate that European carriers such as Lufthansa and Ryanair are already placing significant orders for Boeing 737 MAX aircraft, and the MAX 7 variant is expected to play an important role in regional operations in the coming years. Boeing's strategy to expand the MAX family as a direct response to the Airbus A320neo is reflected in the purchase decisions of key European operators [15].

2. Methodology and research objects

2.1. Research objects

The Boeing 737-800 is a twin-engine, narrow-body, medium-range aircraft (with a range of approximately 5,500 km) belonging to the Next Generation family, the most recent variants of the 737 introduced in the late 1990s. It is capable of operating medium-haul routes, for example, holiday flights or services between European capitals. In an all-economy configuration, the aircraft can accommodate up to 189 passengers. Its cruising speed is 838 km/h, and its maximum take-off weight is 79 tons [1, 16, 22].

The Boeing 737 MAX 8 is also a twin engine, narrow-body, medium-range aircraft, with an extended range of approximately 6,500 km, developed under a program launched in 2011. The first delivery took place in May 2017 to Malindo Air of Malaysia. Compared to the 737-800, the newer model incorporates a few modern features, such as improved cabin soundproofing, larger overhead bins, and more ergonomic seating – depending on customer specifications. In addition, the manufacturer has improved aerodynamic performance, for example, by introducing redesigned winglets [1, 12, 15, 27].

Tables 1a and 1b present a comparison of the key technical and acoustic parameters (from the perspective of certification) of both types of aircraft. It includes numerical data for the Effective Perceived Noise Level (EPNdB), a parameter used to assess the environmental impact of aircraft noise. The comparison considers two selected engines: the CFM56-7B26 (installed on the 737-800) and the LEAP-1B27 (installed on the 737 MAX 8). In addition to the 1B27, the manufacturer also offers the 1B25 and 1B28 variants, which differ in terms of maximum take-off thrust. Since the 1B27 version is the most common among the aircraft studied, its acoustic parameters were chosen for the analysis.

Table 1a. Operating and acoustic parameters of CFM56-7B26 engine in respect of noise levels and noise limits [3, 19]

Aircraft type	B737-800		
Engine	CFM56-7B26		
Maximum take-off thrust [kN]	117.0		
Bypass ratio [-]	5.1:1		
Compression ratio [-]	32.7:1		
Fan diameter [m]	1.55		
Number of fans/low-pressure compressors/high-pressure compressors/high-pressure turbines/low-pressure turbines [-]	1/3/9/1/4		
Dry engine mass [kg]	2370		
External view of the engine			
Considered maximum take-off mass [kg]	70,985		
Considered maximum landing mass [kg]	65,317		
Measuring points	Lateral [dB]	Fly-over [dB]	Approach [dB]
Noise level (EPNdB)	94.2	84.6	96.4
Noise limit (EPNdB)	96.6	91.3	100.4
Difference (EPNdB)	2.4	6.7	4.0

Table 1b. Operating and acoustic parameters of CFM LEAP-1B27 engine with respect to noise levels and noise limits [18, 19, 33]

Aircraft type	B737-MAX8		
Engine	CFM LEAP-1B27		
Maximum take-off thrust [kN]	124.7		
Bypass ratio [-]	9.0:1		
Compression ratio [-]	40.0:1		
Fan diameter [m]	1.76		
Number of fans/low-pressure compressors/high-pressure compressors/high-pressure turbines/low-pressure turbines [-]	1/3/10/2/5		
Dry engine mass [kg]	2780		
External view of the engine			
Considered maximum take-off mass [kg]	70,306		
Considered maximum landing mass [kg]	65,056		
Measuring points	Lateral [dB]	Fly-over [dB]	Approach [dB]
Noise level (EPNdB)	88.3	79.2	93.9
Noise limit (EPNdB)	96.6	91.2	100.3
Difference (EPNdB)	8.3	12.0	6.4

For comparable airport operations (without acknowledging the mass of analyzed aircraft, what was a limitation of this study), the calculated differences between the measured noise levels and the certification limits at each reference point (lateral, fly-over, and approach) are greater in the case of the Boeing 737 MAX 8: approximately 3.5 times greater at the lateral reference point, 1.8 times greater at the fly-over point, and 1.6 times greater at the approach point. Newer aircraft generations produce lower noise emissions, resulting in lower recorded sound levels and a less disruptive impact on humans and wildlife.

2.2. Methodology

The study was based on experimental data from field measurements conducted at six acoustic monitoring points located around Poznań-Ławica Airport (ICAO: EPPO). The points were placed at locations representative of residential exposure to aircraft noise and of typical operational flight paths. The spatial distribution of the measurement points relative to the airport is shown in Fig. 1.

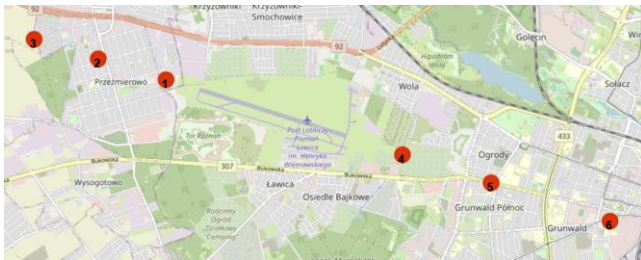


Fig. 1. Location of the measurement points in relation to the airport studied

Measurements were carried out between July and September 2024, covering 78 aircraft operations ($n = 468$), of which 71% involved the Boeing 737-800 and 29% the Boeing 737 MAX 8. Data were collected using a Brüel & Kjær Type 2250 class 1 sound level meter [21]. The instrument was mounted on the end of a telescopic microphone stand at a minimum height of 3.5 m above ground level, following the requirements specified in the Polish Regulation of Minister of the Environment [29], with the microphone oriented vertically upwards. All monitoring points were located at an elevation of 86.7 m (± 2.3 m) and at a distance of approximately 40 m from nearby buildings or obstacles, although the exact distance varied considerably depending on the local land use (see point 4 located directly next to a residential building as an example). The tested aircraft flew over the sound level meter at a distance of up to several meters (horizontally), which depended on the adopted descent path (and which was independent of the authors). The combined uncertainty of $L_{A_{Sel}}$ values at the measurement points ranged from 0.7 to 1.6 dB, with the dominant contribution coming from statistical variability, supplemented by wind-related uncertainty (0.1–0.5 dB for wind speeds of 1.0–5.1 m/s). Other components – including instrumental (0.5 dB), calibration (0.2 dB), microphone positioning (0.2 dB), climatic (0.2 dB), and background noise [4, 13] (0.3 dB) – were considered in accordance with the PN-ISO 1996-2 and AIR1845A standards [34], recommendations of the Polish Central Office of Measures. The over-

all uncertainty did not exceed 1.6 dB, allowing the results to be considered reliable.

The study was carried out in accordance with the Regulation of the Minister of the Environment (Journal of Laws, 2007), Directive 2002/49/EC, and the PN-ISO 1996-2:2019-01 standard [9, 14]. To compare the impact of the two aircraft types, only $L_{A_{Sel}}$ levels were analyzed. This energy-based indicator enables comparisons of operations regardless of their duration or number and is also required for models used to evaluate environmental exposure. To determine both the energy and the statistical exposure levels, the $L_{A_{Sel}}$ values were converted into physical units ($\text{Pa}^2 \cdot \text{s}$) [24].

The analysis was supplemented with selected acoustic indicators (L_{Aeq} , L_{50} , L_{AFmax} , L_{Cpeak} , L_{AFsd} , L_{Ae}) [30, 35], as well as with an examination of mutual correlations between these indicators in physical units (Spearman rank correlation coefficient). The significance of the differences was assessed using statistical tests (Shapiro-Wilk, Student's t test and Mann-Whitney U test).

In the environmental simulations, for each monitoring point and each day of August, the equivalent daily sound level L_{AeqD} [36] was calculated using the formula:

$$L_{AeqD} = 10 \log_{10} \left(\frac{1}{T_D} \sum_{i=1}^N E_i \right) \quad (1)$$

where: L_{AeqD} – daily equivalent A-weighted sound level [dB], T_D – number of seconds in a reference daytime between 6 a.m. and 10 p.m. (16 h, 57,600 s), N – the number of operations executed at that time, E_i – linear exposure contribution defined derived from $L_{A_{Sel}}$ (acoustic energy of a single operation) as:

$$E_i = 10^{0.1 A_{Sel_i}} \quad (2)$$

The background energy influence was analyzed and corrected using the following formula:

$$L_{AeqD,corr} = 10 \log_{10} (10^{0.1 L_{AeqD}} - 10^{0.1 L_{Aeq,bg}}) \quad (3)$$

where $L_{AeqD,corr}$ is the daytime equivalent A-weighted sound level after background correction [dB], L_{AeqD} is the uncorrected daytime equivalent A-weighted sound level [dB], and $L_{Aeq,bg}$ is the equivalent background A-weighted sound level [dB] determined for the same reference period. Background correction was therefore performed by energetic subtraction of the equivalent background contribution, rather than by adding an undefined background term inside Eq. (1).

Nine fleet composition scenarios were modeled (ranging from 100% 737-800 to 100% MAX 8, with increments of 10–20%). For each monitoring point and each day when the daily L_{AeqD} exceeded the threshold value of 60 dB, the excess was calculated in decibels, after which the financial penalty was determined according to the tariff thresholds applicable in 2025: EUR 3.92 for each dB above the limit in the 1–5 dB range, EUR 6.86 per dB in the 5–10 dB range, EUR 9.81 per dB in the 10–15 dB range, and EUR 14.71 per dB above 15 dB. This approach enabled an assessment of both the environmental and economic consequences of different fleet configurations.

3. Results

3.1. Comparison of the characteristics of both aircraft types

The general analysis (Fig. 2) showed that the mean value of $L_{A_{SEL}}$ for the Boeing 737-800 was $0.89 \text{ Pa}^2\cdot\text{s}$ (94.5 dB), while for the Boeing 737 MAX 8 it was $0.57 \text{ Pa}^2\cdot\text{s}$ (92.6 dB). This difference was statistically sig-

nificant (Mann–Whitney U test, $p = 0.001$). In practical terms, the 737-800 generated an $L_{A_{SEL}}$ level $0.32 \text{ Pa}^2\cdot\text{s}$ higher, corresponding to a 56% increase over the MAX 8, i.e., a difference of 1.9 dB.

The distribution of $L_{A_{SEL}}$ by flight phase (take-off/landing) is presented in Fig. 3. During take-off, the difference between aircraft types amounted to $0.52 \text{ Pa}^2\cdot\text{s}$ (2.7 dB), while during landing it was $0.07 \text{ Pa}^2\cdot\text{s}$ (0.8 dB).

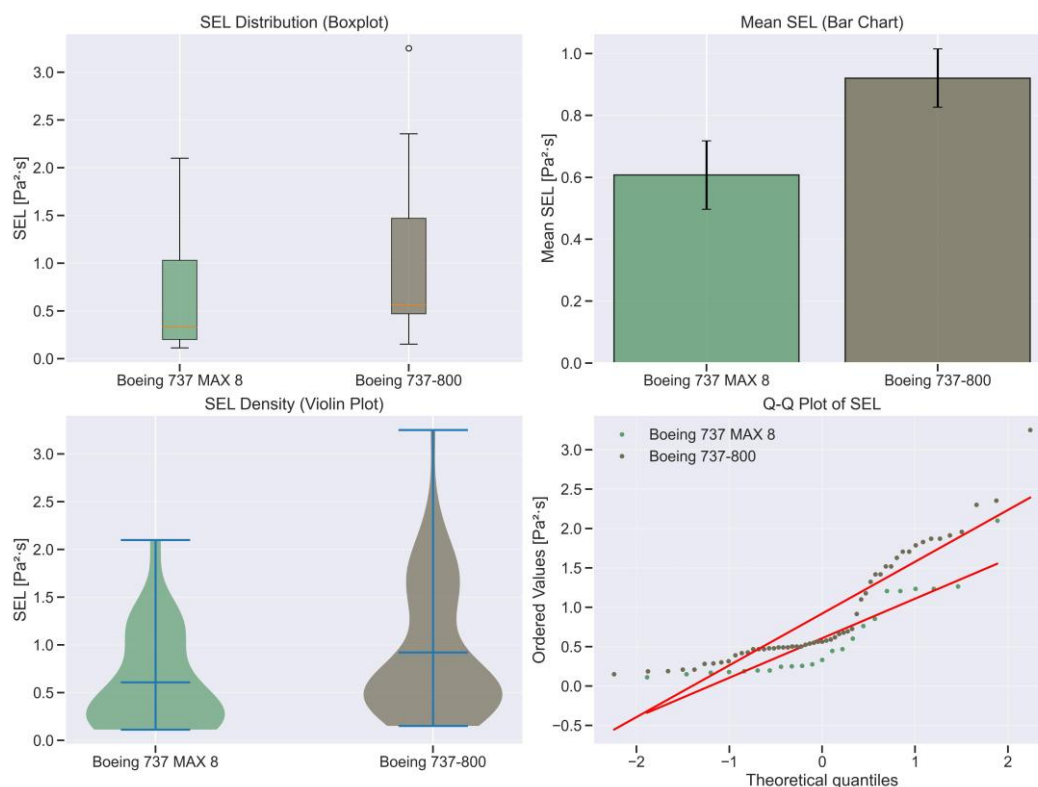


Fig. 2. General analysis of $L_{A_{SEL}}$ levels

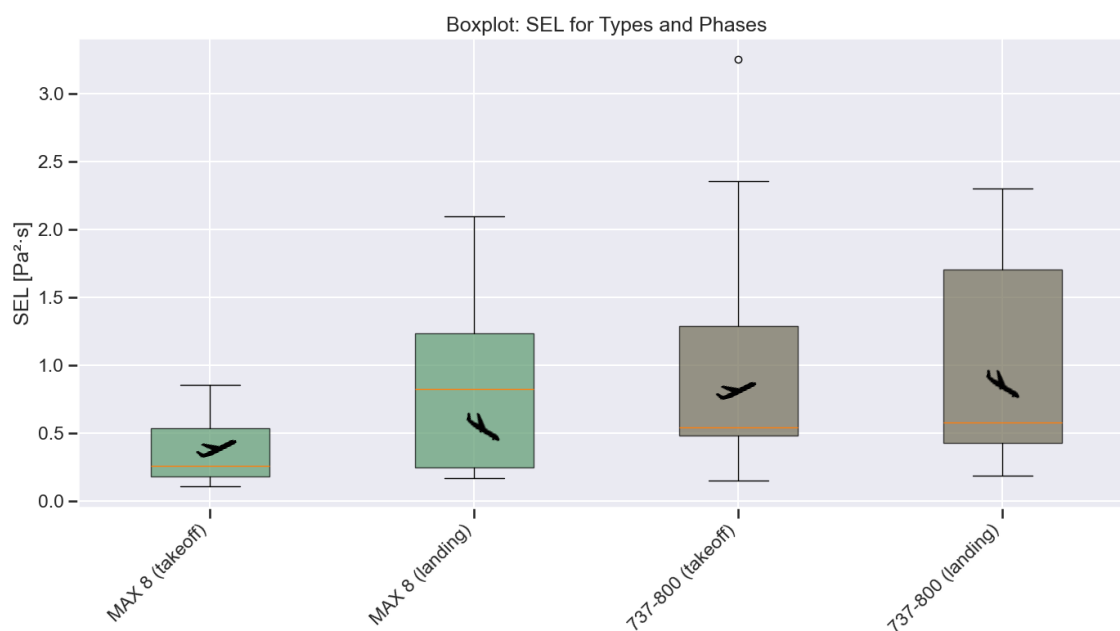


Fig. 3. Distribution of $L_{A_{SEL}}$ by take-off and landing phases

The relationship between sound exposure level and flight altitude is shown in Fig. 4 using a log-distance representation in the dB domain. In the revised analysis, $L_{A\text{Sel}}$ was modeled as a linear function of $\log_{10}(d)$

$$L_{A\text{Sel}}(d) = \beta_0 + \beta_1 \log_{10} d \quad (4)$$

where d denotes the source–receiver distance. Because the available dataset did not include the horizontal offset of the aircraft relative to the monitoring point, altitude above the

monitoring point was used as a proxy for distance in this analysis. This representation is more consistent with acoustic propagation theory than the previous linear-in-altitude model in meters, because under free-field spreading a level expressed in decibels is expected to vary approximately linearly with $\log_{10}(d)$. The larger dispersion visible in the Boeing 737-800 subset likely reflects both greater operational variability and the fact that the aggregated figure includes both landings and departures.

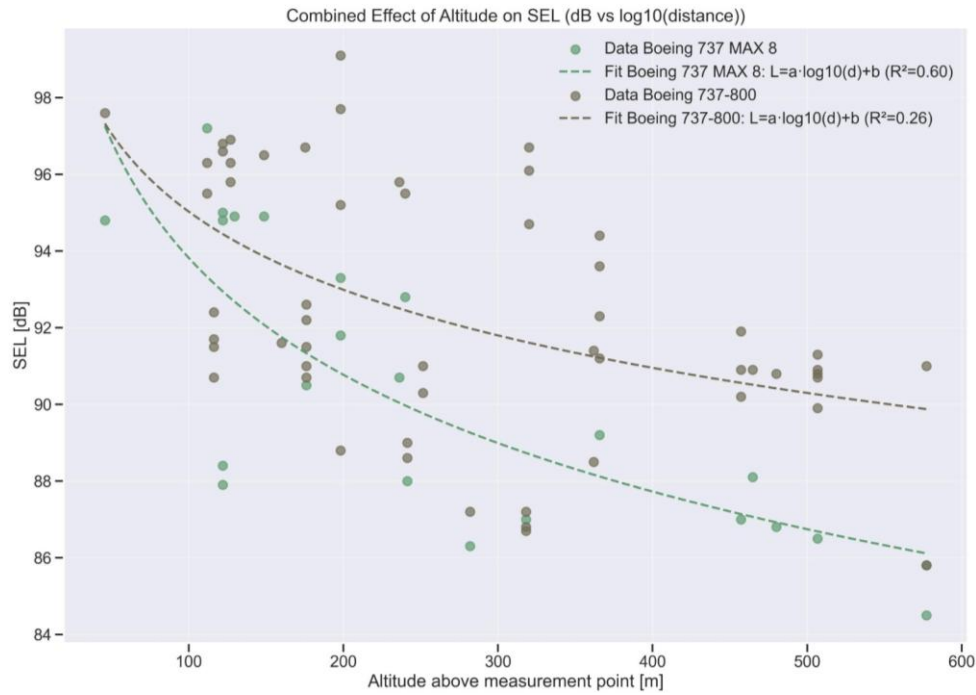


Fig. 4. Relationship between sound exposure level and flight altitude using a log-distance representation in the dB domain for Boeing 737-800 and Boeing 737 MAX 8. Data shown here include both landings and departures

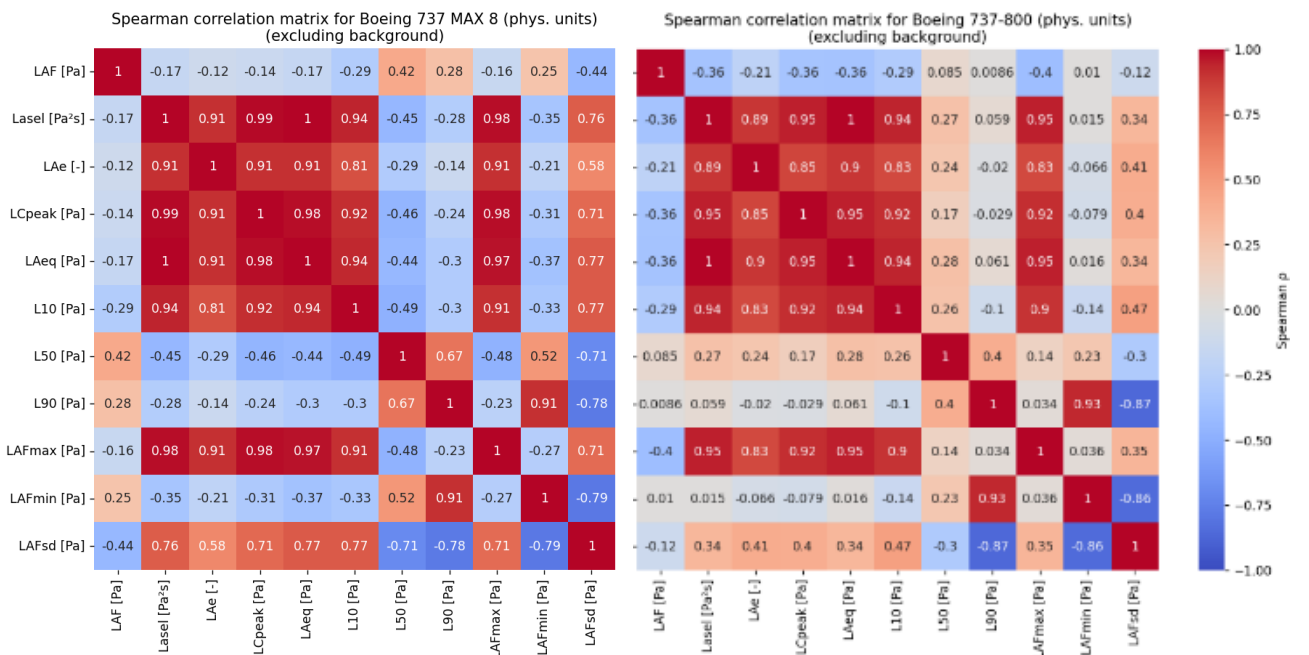


Fig. 5. Statistical summary of correlation values for acoustic indicators

The larger dispersion observed for the Boeing 737-800 subset likely reflects greater operational heterogeneity in that sample and the fact that the aggregated figure combines both landings and departures. As noted above, the Boeing 737-800 generates, on average, 56% higher $L_{A_{Sel}}$ values ($0.32 \text{ Pa}^2 \cdot \text{s}$, 1.9 dB) than the Boeing 737 MAX 8, with the difference increasing to 137% ($0.52 \text{ Pa}^2 \cdot \text{s}$, 2.7 dB) during take-off. The strong dependence of $L_{A_{Sel}}$ on both altitude and aircraft type was statistically confirmed. The findings indicate that the 737-800 generates significantly more acoustic energy around the airport than the newer 737 MAX 8. The difference in $L_{A_{Sel}}$ is not only statistically significant but also practically relevant for environmental exposure assessment [23].

3.2. Comparison of acoustic characteristics based on indicators and their mutual correlations

The correlation analysis of the acoustic indicators for both aircraft types was carried out using Spearman's rank correlation coefficient, summarized in Table 2. The coefficient was calculated exclusively in physical units (Fig. 5).

Table 2. Basic statistics of Spearman correlations (physical scale)

Aircraft type	Spearman ρ – mean	Spearman ρ – median	Std. dev.
737 MAX 8	0.61	0.67	0.31
737-800	0.49	0.36	0.38

For the Boeing 737 MAX 8, the mean Spearman correlation was 0.61 (median 0.67, standard deviation 0.31), while for the 737-800 the corresponding values were 0.49 (median 0.36, standard deviation 0.38). The difference in mean correlation between the two types was 0.12, representing a 24% increase in favor of MAX 8.

To account for the differences between aircraft types, a matrix of absolute differences (corr) was calculated according to Eq. 4:

$$\Delta_{\text{corr}} = r_{\text{Spearman}_{800}} - r_{\text{Spearman}_{\text{MAX}8}} \quad (4)$$

The mean value of Δ_{corr} was 0.12 (median 0.086), with extreme values ranging from -0.25 to 0.43 (Table 3). This indicates that, in most cases, correlations are higher for the 737-800, although approximately 25% of the metric pairs show higher values for the MAX 8. A detailed breakdown of these differences is presented in Tables 4 and 5.

Table 3. Statistics of Spearman correlation differences (Δ_{corr})

Statistics	Mean	Median	Std. Dev.	Min.	Max.
Δ_{corr}	0.12	0.086	0.167	-0.25	0.43

The highest positive Δ_{corr} values (0.33 – 0.43 , Table 5) were observed for pairs of indicators related to energetic exposure ($L_{A_{eq}}$, $L_{A_{Sel}}$) and variability measures ($L_{A_{Fsd}}$, $L_{A_{Fmax}}$, $L_{A_{Fmin}}$, $L50$). For example, the correlation between $L_{A_{eq}}$ and $L_{A_{Fsd}}$ was higher by 0.43 for the 737-800, and the correlation between $L_{A_{Sel}}$ and $L_{A_{Fsd}}$ was higher by 0.42 . In relative terms, this means that the 737-800 exhibits up to 50–100% stronger dependencies between these indicators compared with the MAX 8. This reflects greater predictability of variability increases alongside rising energy levels in the

case of the 737-800. In contrast, for certain pairs of metrics (Table 5), MAX 8 demonstrated stronger correlations (the largest difference was 0.25 , for example, between L_{AF} and L_{AFmax}), although these relationships are less critical from an environmental protection perspective.

Table 4. Indicator pairs with the largest correlation differences ($\Delta_{\text{corr}} > 0.3$)

Pair of measures	Difference Δ_{corr} (800 – MAX8)
$L_{A_{eq}} [\text{Pa}] - L_{A_{Fsd}} [\text{Pa}]$	0.43
$L_{A_{Sel}} [\text{Pa}^2 \cdot \text{s}] - L_{A_{Fsd}} [\text{Pa}]$	0.42
$L50 [\text{Pa}] - L_{A_{Fsd}} [\text{Pa}]$	0.41
$L_{A_{Fmax}} [\text{Pa}] - L_{A_{Fsd}} [\text{Pa}]$	0.36
$L_{A_{eq}} [\text{Pa}] - L_{A_{Fmin}} [\text{Pa}]$	0.36
$L50 [\text{Pa}] - L_{A_{Fmax}} [\text{Pa}]$	0.34
$L_{A_{Sel}} [\text{Pa}^2 \cdot \text{s}] - L_{A_{Fmin}} [\text{Pa}]$	0.34
$L_{AF} [\text{Pa}] - L50 [\text{Pa}]$	0.33

Table 5. Pairs of indicators with higher correlations for MAX 8 ($\text{corr} < 0.3$)

Pair of measures	Difference Δ_{corr} (800 – MAX8)
$L_{AF} [\text{Pa}] - L_{AFmax} [\text{Pa}]$	-0.25
$L_{AF} [\text{Pa}] - L_{C_{peak}} [\text{Pa}]$	-0.22
$L_{AF} [\text{Pa}] - L_{A_{eq}} [\text{Pa}]$	-0.19
$L_{AF} [\text{Pa}] - L_{A_{Sel}} [\text{Pa}^2 \cdot \text{s}]$	-0.19
$L_{AF} [\text{Pa}] - L_{A_e} [-]$	-0.10

In this respect, the MAX 8 surpasses the 737-800 in the strength of correlations between these pairs; however, such relationships are of limited relevance for environmental exposure assessment.

In summary, the Boeing 737 MAX 8 shows higher mean and median associations among most acoustic indicators than the 737-800. The relationships between energetic and peak measures ($L_{A_{eq}}$, $L_{A_{Sel}}$, $L_{C_{peak}}$, L_{AFmax}) are particularly consistent and predictable. On the contrary, the 737-800 shows greater variability and heterogeneity in correlations, especially for indicators of noise distribution variability. In practice, this implies that for a fleet dominated by MAX 8 aircraft, environmental exposure can be predicted with high precision based on energetic indicators, whereas for the 737-800, it is also necessary to consider statistical distribution parameters of noise.

4. Impact of fleet composition and number of operations on environmental exposure

For the simulations, the number of flight operations was assumed as follows: 115 on weekdays (± 10) and 95 on weekends (± 10). This approach reflects the actual traffic load and ensures the month's representativeness for the analysis (August). It was assumed that the total number of operations in each variant remained constant, with only the percentage share of individual aircraft types varying between fleet configurations. Nine variants of the fleet structure were analyzed, ranging from extreme cases consisting exclusively of one aircraft type (100% Boeing 737-800 or 100% Boeing 737 MAX 8), through mixed configurations covering successive percentage intervals (10% to 90% for each type, in steps of 10% or 20%), to evenly distributed variants (e.g. 50% 737-800 and 50% MAX 8). Such a broad spectrum of variants enables a comprehensive assessment

of the impact of gradually introducing a more modern fleet on the acoustic environment surrounding the airport. These variants reflect both operational scenarios observed during transitional periods of fleet modernization and analytical extremes that allow identification of boundary cases [7, 31]. Consequently, it was possible not only to assess current exposure, but also to predict the environmental effects of progressively increasing the share of Boeing 737 MAX 8 aircraft in total operations. The configurations adopted and the distribution of aircraft types in each structure analyzed structure are shown in Fig. 6.

The results of the L_{AeqD} simulation for different variants of the fleet structure are presented as heat maps (Fig. 7), allowing for a detailed assessment of the noise exposure depending on the measurement point and the proportion of specific aircraft types in flight operations. The analysis is

based on a direct comparison of days when the daily L_{AeqD} level exceeded the 60 dB threshold [26].

The color of each block in the heatmap represents the mean daily L_{AeqD} level obtained at a given measurement point and the fleet configuration variant: the more intense the color, the higher the simulated noise level in the analyzed case. Blocks with light neutral colors correspond to measurement points and fleet variants where the L_{AeqD} level did not significantly exceed the acoustic background level, meaning that the impact of aircraft noise was negligible compared to local background noise. At more distant points (5, 6), no exceedances occurred, which is represented by empty blocks. The cells outlined with a purple border indicate cases in which the corrected daily L_{AeqD} level exceeded the 60 dB threshold adopted as a criterion for environmental noise.

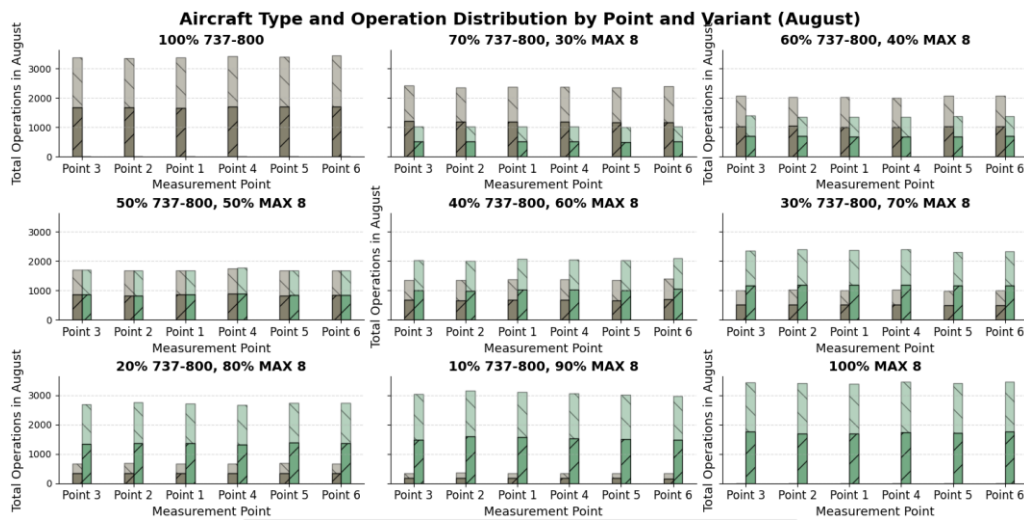


Fig. 6. Percentage structure of aircraft types (Boeing 737-800 and Boeing 737 MAX 8) in each fleet configuration variant used in the simulation analysis (lighter colors are landings, darker colors are take-offs)

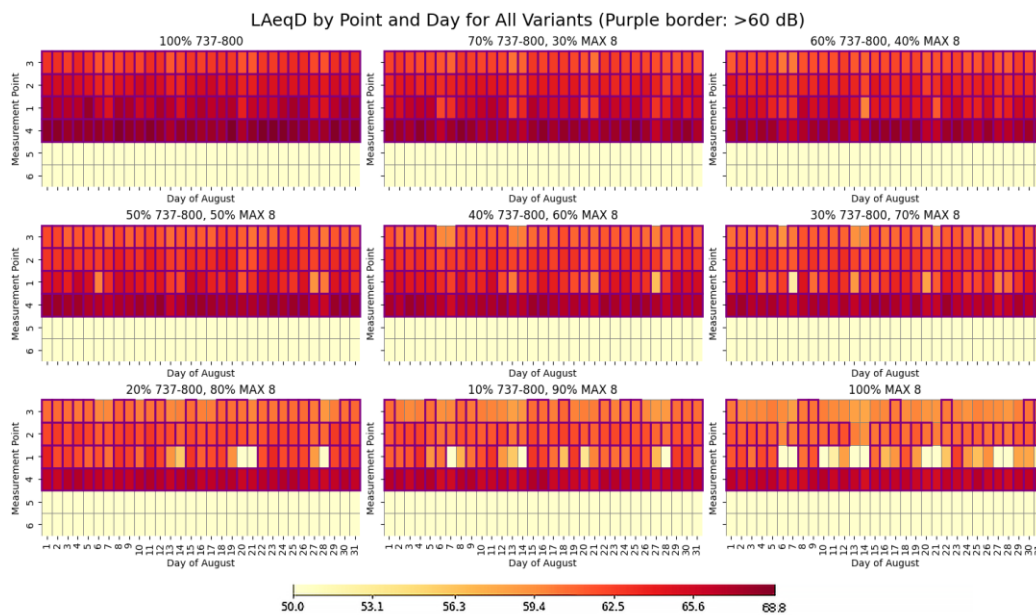


Fig. 7. Heatmap of L_{AeqD} values for different scenarios of fleet composition in flight operations

Regardless of the fleet scenario, measurement point 4, located closest to the airport, always exceeded the established 60 dB threshold. However, from a share of 80% MAX 8, the exceedances became noticeably smaller. For all configurations at points 5 and 6, the L_{AeqD} level did not exceed the background noise level. A strong spatial gradient of exposure is evident: the closer the airport, the greater the frequency and intensity of threshold exceedances. This effect is clearly moderated by the proportion of MAX 8 aircraft, which demonstrate more favorable acoustic and energetic parameters.

The transition from a fleet dominated by 737-800 to one with a higher share of MAX 8 results in a substantial reduction in noise exposure and a marked decrease in the number of days exceeding the environmental threshold. A particularly beneficial effect is observed even with a share of 60–70% MAX 8 [7, 31, 36].

Variants dominated by 737-800 (100%, 70%, 60%) and 50% share lead to threshold exceedances (60 dB) on all days of the month at points 1–4 (the locations closest to the airport). As the proportion of 737 MAX 8 increases, the number of exceedance days decreases, especially at points 1–3 (Fig. 8). With a 100% share of days of MAX 8, the number of exceedances at point 1 is incidental (7 days).

To gather the number of exceedances during the analyzed month (August), a simple Table 6 was created. The greatest benefits of using the MAX 8 aircraft for operations at Poznań Airport can be seen in points 1 and 3, especially with the 100% share in the fleet configuration.

To evaluate the economic consequences of noise exceedances, a study adopted the approach consistent with the penalty rates for exceeding the permitted levels of environmental noise currently in force in Poland [25], as described in the research methodology. The total monthly environmental penalty was calculated as the sum of the values accrued across all days in August at each measurement point and for each fleet composition scenario. Figure 9 presents a comparison of these results for points 1–4, that is, the locations most exposed to aircraft noise.

Table 6. Number of daily 60 dB threshold exceedances in August

Each variant share	Point no. 1	Point no. 2	Point no. 3	Point no. 4
100% 737-800	31	31	31	31
0% MAX 8				
70% 737-800	31	31	31	31
30% MAX 8				
60% 737-800	31	31	30	31
40% MAX 8				
50% 737-800	31	31	28	31
50% MAX 8				
40% 737-800	26	31	28	31
60% MAX 8				
30% 737-800	27	31	26	31
70% MAX 8				
20% 737-800	21	31	24	31
80% MAX 8				
10% 737-800	13	31	20	31
90% MAX 8				
0% 737-800	7	27	10	31
100% MAX 8				

For the baseline variant (100% 737-800), the total monthly environmental penalty (for all points 1–6) amounts to 6069 EUR. In the extreme case of 100% MAX 8, this sum decreases to 1902 EUR, representing a reduction of 4167 EUR ($\approx 69\%$). Mixed variants confirm a gradual decline in costs. With a 70% share of the 737-800, the penalty amounts to 4979 EUR (a reduction of 18% relative to the baseline); with a 50% share, 4136 EUR (a reduction of 32%); with a 30% share, 3225 EUR (a reduction of 47%); and already at 10% share, the penalty falls to 2353 EUR, corresponding to a reduction of 61% compared to a fleet composed entirely of 737-800 aircraft.

The highest penalty values occur systematically at points 1 and 4, located closest to the airport – for example, in the 100% 737-800 variant, the monthly penalty amounts to 1248 EUR at point 4 and 901 EUR at point 1. At more distant points (5 and 6), the total penalty is zero in every variant, indicating that there are no exceedances of the 60 dB threshold at these locations.

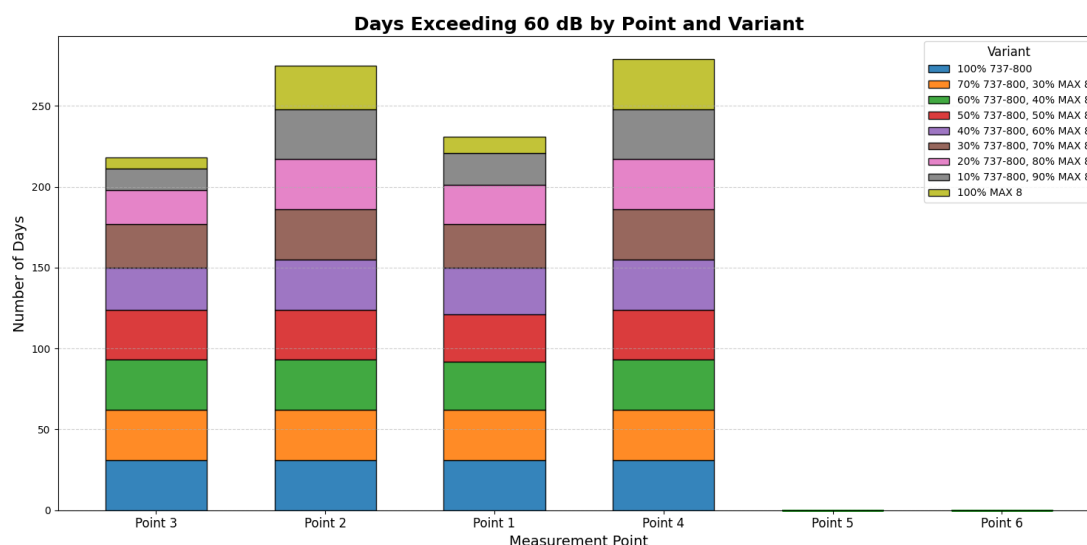


Fig. 8. Number of days that exceeded the environmental threshold by aircraft type and measurement point

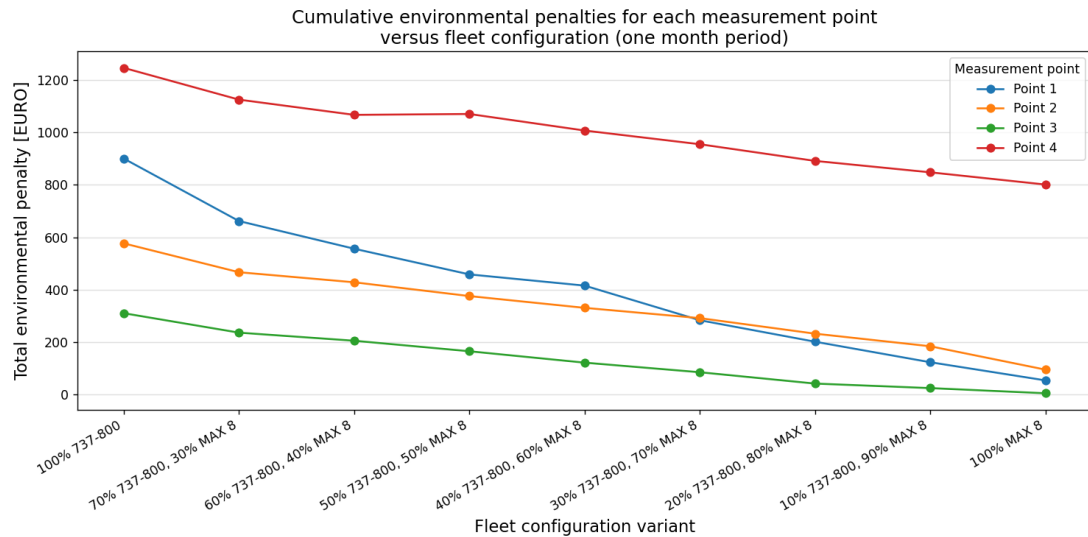


Fig. 9. Total monthly environmental penalty (in EUR) for each measurement point as a function of the fleet configuration variant. Each line corresponds to a different measurement point; values calculated for the sum of all days in August, with daily L_{AeqD} exceeding 60 dB

5. Discussion

When interpreting the results, several important limitations of the study must be acknowledged. First, the simulations were conducted for a single month (August), which allows assessment of seasonal effects but does not account for the variability of acoustic characteristics or the distribution of operations throughout the year. Incorporating annual data could yield different, more nuanced conclusions, particularly regarding air traffic variability and meteorological conditions. The assumed fleet shares in the simulation variants reflect both the actual proportions observed during the measurement campaign and the potential modernization scenarios. In practice, the actual distribution of aircraft types depends on the decisions of airline operators, over which the airport has limited influence, although it is the airport that bears responsibility for meeting environmental obligations. Therefore, the configurations were designed to be representative of both current and potential future fleet composition. Another limitation concerns the acquisition of $L_{A_{SEL}}$ and background noise data. The analysis was based on the results of a selected period, and particularly at points 5 and 6, the background noise level was elevated due to strong urban noise, which may have limited the detection of exceedances attributable to aircraft. A different measurement campaign or a different season could produce different results at these points. The environmental cost model used in the analysis considered only exceedances of the daily L_{AeqD} 60 dB and did not encompass all possible consequences of airport operations, such as resident claims, compensations, or investments in noise protection. Therefore, the actual economic balance may be even more favorable under effective fleet modernization. It should also be noted that the study focused on daytime hours, which represent the main phase of air traffic at Poznań Airport. Assessing the impact on the acoustic climate during nights or weekends would require separate analyses, accounting for regulatory specifics and the distribution of operations. In addition, the dataset did not include the full slant distance between the aircraft and the receiver, nor operational

metadata such as aircraft mass, payload, destination, or flight phase; therefore, altitude was used only as a proxy for distance, and weight- or phase-related effects could not be quantified explicitly in the present model. Despite these limitations, the findings highlight a strong dependence of noise impact on the type of aircraft operated and confirm that fleet modernization may be one of the most effective tools for managing environmental risk around airports.

6. Conclusions

The results showed that the Boeing 737-800 aircraft generates significantly higher energetic noise levels than the newer 737 MAX 8. The mean $L_{A_{SEL}}$ value for the 737-800 was $0.89 \text{ Pa}^2 \cdot \text{s}$ (94.5 dB), while for the MAX 8 it was lower at $0.57 \text{ Pa}^2 \cdot \text{s}$ (92.6 dB), representing a difference of 56% (1.9 dB). These differences are particularly pronounced during the take-off phase, where the $L_{A_{SEL}}$ level for the 737-800 exceeded that of the MAX 8 by up to 137% (2.7 dB). A statistically significant decrease in $L_{A_{SEL}}$ with increasing altitude was observed; however, in the revised interpretation, altitude is treated as a proxy for source–receiver distance, and the fitted relationship is presented in a physically motivated log-distance form in dB.

The correlation analysis of the acoustic indicators revealed that the Boeing 737 MAX 8 is characterized by greater consistency and predictability in the relationships between energetic and peak measures (mean Spearman correlation of 0.61, compared to 0.49 for the 737-800), indicating a more stable acoustic profile for these operations. In contrast, the 737-800 exhibited greater variability and higher correlations in distribution-related noise metrics, suggesting a more heterogeneous acoustic emission pattern for this aircraft type.

Environmental simulations conducted for nine fleet composition scenarios (ranging from 100% 737-800 to 100% MAX 8) demonstrated a marked reduction in the number of days exceeding the L_{AeqD} threshold of 60 dB, as well as a significant decrease in monthly environmental penalties imposed for such exceedances. The findings clearly indicate that increasing the share of Boeing 737 MAX 8

aircraft in the fleet results in a substantial reduction in environmental impacts associated with aircraft noise. For measurement points located closest to the airport, transitioning from a fleet entirely composed of Boeing 737-800s to one fully dominated by 737 MAX 8 results in a reduction in threshold (L_{AeqD}) exceedance days by 80–90% and lowers total monthly penalties from approximately 5000–7000 EUR to 600–900 EUR. Even partial modernization – at a share of 60–70% MAX 8 – reduces costs by 60–70% compared to the baseline scenario. Across the entire study area, the fleet results in a 69% reduction in total penalties, confirming tangible economic benefits from the introduction of aircraft with improved acoustic characteristics. In particular, even with about 50% MAX 8 in the fleet, total costs fall by one-third, while at a share of 70–80% the reduction exceeds 50%.

The findings indicate that fleet modernization toward a greater share of Boeing 737 MAX 8 aircraft with superior acoustic performance is a highly effective tool for reducing

community exposure to aircraft noise and minimizing environmental costs incurred by the airport. The analysis accounted for not only acoustic impacts but also for economic and legal dimensions, drawing on national and European environmental protection regulations and normative guidelines.

In summary, the applied methodology and the proposed analytical tools (including $L_{A_{SEL}}$ analysis, correlation metrics, and fleet scenario modelling) may serve as effective support for environmental management at airports and the optimization of operational decisions in the context of protecting residents from the adverse effects of transportation noise.

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Nomenclature

L_{Ae}	A-weighted sound exposure level	L_{AFsd}	standard deviation of A-weighted sound level with fast time weighting
L_{Aeq}	A-weighted equivalent continuous sound level	$L_{A_{SEL}}$	A-weighted sound exposure level (SEL)
L_{AeqD}	Daytime A-weighted equivalent continuous sound level	L_{Cpeak}	Maximum A-weighted sound level with fast time weighting
L_{AFmax}	Maximum A-weighted sound level with fast time weighting	$L50$	A-weighted sound level exceeded for 50% of the measurement time

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A review of internal combustion engines applied to unmanned aerial vehicles

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Internal combustion engines constitute one of the two fundamental propulsion categories used in unmanned aerial vehicles (UAVs). This article analyzes the occurrence and application of individual types of internal combustion engines across unmanned platforms, with particular emphasis on their influence on aircraft performance and mission profiles. The advantages and limitations of specific propulsion system configurations are identified in the context of unmanned aviation, including factors such as energy efficiency, mass, structural complexity, endurance, and acoustic and thermal signatures. In addition, representative UAVs employing different types of internal combustion engines are presented, illustrating the practical implications of propulsion system selection in real-world operational applications.

Key words: *unmanned aerial vehicles (UAVs), internal combustion engines, aircraft propulsion systems, piston and turbine engines, endurance-oriented propulsion*

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1. Introduction

The development of unmanned aerial vehicles (UAVs) is closely linked to the evolution of propulsion systems, as propulsion determines the key operational parameters of a platform, including endurance, range, service ceiling, airspeed, payload capacity, and overall reliability. In the early stages of UAV development – primarily focused on aerial targets and simple surveillance platforms – relatively uncomplicated piston engines adapted from light aviation or other small-engine applications predominated. These power plants were selected primarily for their favorable power-to-weight ratio and ease of integration. As the requirements of reconnaissance, surveillance, and later strike missions increased, greater emphasis was placed on cruise efficiency, stable operation over long duty cycles, and the ability to maintain performance across a wider range of altitudes and ambient temperatures.

In recent years, the rapid expansion of unmanned aerial vehicle applications in both civilian and military sectors has significantly increased the importance of propulsion system optimization. Modern UAVs are expected to perform increasingly demanding missions, including long-endurance intelligence, surveillance and reconnaissance (ISR), precision strike operations, cargo transport, border monitoring, environmental sensing, and autonomous cooperative missions. These operational requirements impose growing demands on propulsion systems for fuel efficiency, reliability, endurance, payload capability, reduced thermal and acoustic signatures, and operational flexibility. Simultaneously, recent research trends in UAV propulsion have increasingly focused on hybrid-electric architectures, heavy-fuel compatibility, advanced engine control systems and mission-oriented propulsion optimization.

Despite the dynamic development of electric propulsion in the micro- and mini-UAV classes, the energy density limitations of batteries mean that liquid fuels remain the dominant energy carrier for medium- and large-scale platforms. Consequently, internal combustion propulsion systems continue to form the backbone of UAVs requiring

multi-hour endurance and the capability to carry significant payloads. In parallel, turbine-based propulsion systems have been developed for high-speed platforms, for which operation in flight regimes unattainable for conventional propeller-driven configurations is a primary requirement. In recent years, increasing attention has also been paid to hybrid propulsion architectures, in which an internal combustion engine serves as an onboard energy source while electric motors provide propulsion.

From a systems perspective, internal combustion engines used in UAVs can be classified into three main groups: spark-ignition piston engines, compression-ignition piston engines, and gas turbine engines. Two-stroke spark-ignition engines are employed primarily in lightweight platforms, where a high power-to-weight ratio and mechanical simplicity are critical, often at the expense of higher specific fuel consumption and less favorable acoustic characteristics. Four-stroke spark-ignition engines dominate many tactical UAVs and MALE-class systems, where efficiency during prolonged cruise and stable operation in multi-hour missions is essential. Compression-ignition and heavy-fuel engines are being developed to further improve efficiency and enable fuel commonality in military applications; however, their implementation requires advanced fuel-injection control, turbocharging, and design solutions that ensure reliable cold- and high-altitude-starting. Turbine propulsion systems, in turn, represent a natural choice for high-speed UAVs, where flight dynamics and a high thrust-to-weight ratio take precedence over maximum fuel economy.

The purpose of this review article is to provide an overview of current knowledge and development of internal combustion propulsion systems for unmanned aerial vehicles, with particular focus on the relationships among propulsion system characteristics, operational effectiveness, mission requirements, and platform integration limitations.

This review contributes to the existing body of research by providing a comprehensive evaluation and comparison of internal combustion propulsion technologies across dif-

ferent UAV categories and operational applications. In contrast to many previously published studies concentrating on individual engine types or selected groups of unmanned platforms, the present work combines the historical evolution of UAV propulsion, propulsion system classification, operational properties, representative aircraft examples, and mission-oriented performance analysis within a single coherent framework. Special emphasis is placed on the interdependence between propulsion architecture, endurance, fuel economy, operational altitude, payload capability, detectability, and integration-related complexity. Additionally, the review highlights current technological developments and outlines prospective trends in UAV propulsion, including hybrid propulsion concepts, heavy-fuel engines, and low-observability propulsion solutions.

2. History of UAV propulsion systems

The earliest recorded instance of unmanned aerial vehicles is commonly associated with the aerial attack on Venice on 12 July 1849 by Austro-Hungarian forces. The means of attack consisted of approximately 200 small hot-air balloons carrying explosive charges. Although these aerostats may be regarded as the earliest unmanned aerial systems, they were extremely simple constructs, devoid of propulsion and any form of control [39]. The first unmanned aerial vehicle equipped with its own propulsion system did not appear until World War I. At that time, as manned aviation was rapidly developing, the first concepts of crewless aircraft began to emerge. Great Britain assumed a leading role in this field, where several independent radio-controlled fixed-wing projects were developed under the Aerial Target program, using a control system designed by the RFC's Experimental Works, led by Archibald Low. After three years of research, a prototype fixed-wing UAV conducted its first unmanned flight on 21 March 1917. Propulsion was provided by an air-cooled, two-cylinder ABC Gnat engine with a displacement of 2.3 dm³, producing 34 kW of power [45, 55].

A similar line of development was pursued by Elmer Sperry, who, in cooperation with Peter Hewitt, undertook the conversion of the Curtiss N-9 seaplane into an unmanned aircraft. As in the original manned version, the UAV was powered by a water-cooled Wright-Hisso A V8 engine, structurally derived from the Hispano-Suiza 8. The powerplant, with a displacement of 11.8 dm³, delivered a maximum power output of 110 kW [4, 45]. Further work by Sperry, this time in collaboration with Charles Kettering and the Dayton-Wright Airplane Company, led to the development of the Kettering Bug unmanned attack aircraft. In contrast to earlier designs, the propulsion system represented a significant departure. The primary engine selection criteria for this precursor of modern cruise missiles were simplicity of construction, low production cost, and a high power-to-weight ratio. Consequently, a two-stroke DePalma V4 engine manufactured by the Ford Motor Company was selected, delivering 28 kW of power, sufficient to propel the relatively small airframe [47, 58].

Another breakthrough in UAV propulsion occurred in 1921 with the development of the RAE Larynx prototype, the first unmanned aircraft powered by a radial engine. The propulsion system was a seven-cylinder, air-cooled Arm-

strong Siddeley Lynx IV radial engine. With a displacement of 12.4 dm³, it produced 150 kW of power [64, 83]. The first inline engine applied to unmanned aircraft followed in 1933, when the de Havilland Gipsy Major was used to power the de Havilland DH.82 Queen Bee, an unmanned variant of the de Havilland DH.82 Tiger Moth. This air-cooled inline four-cylinder engine, with a displacement of 6.1 dm³, generated a maximum power output of 96 kW [25, 26]. Another important milestone in the development of unmanned aviation was the establishment of Reginald Denny Industries in 1935, later transformed into the Radioplane Company – the first dedicated drone manufacturer in history. Although its OQ-2 and OQ-3 UAVs became the first unmanned aircraft to be mass-produced, they did not introduce significant innovations in propulsion, which remained based on two-cylinder spark-ignition engines.

A further major advance occurred with the German V-1 research programme, which resulted in the first mass-produced cruise missile, the Fieseler Fi 103. This system was also the first unmanned aerial vehicle powered by an air-breathing jet engine. The Argus As 014 pulsejet engine produced 2.7 kN of thrust, accelerating the vehicle to approximately 640 km/h. Following the end of World War II, jet propulsion technology and its aviation applications developed rapidly, with corresponding effects in the unmanned domain. In 1948, the U.S. Air Force awarded Ryan Aeronautical Company a contract to develop and manufacture the first turbojet-powered unmanned aircraft. The first flight of the XQ-2 Firebee prototype, powered by a Continental J69-T-19B engine, took place in 1951, with the engine producing a maximum thrust of 4.7 kN. In 1960, the BQM-34A production variant entered service, equipped with the Continental J69-T-29A engine delivering up to 7.6 kN of thrust, enabling maximum speeds of approximately 1,110 km/h [26, 59]. Similar developments occurred on the opposite side of the Iron Curtain. Production of the first Soviet turbojet-powered UAV, the Lavochkin La-17, began in 1954. Propulsion was provided by the Mikulin RD-9BK engine, a simplified derivative of the powerplant used in the MiG-19 fighter aircraft. From that point onward, unmanned aviation increasingly shifted to jet propulsion, reflecting the broader aviation trend of the era toward ever-higher flight speeds. This trajectory culminated in the development of the first unmanned aircraft powered by a ramjet engine. On 5 March 1966, the Lockheed D-21, equipped with the RJ43-20S4 ramjet engine, conducted its first test flight.

During the 1970s, unmanned aviation experienced a return to smaller platforms intended for tactical reconnaissance missions. Due to the operational characteristics of such missions, piston engines once again became the preferred propulsion solution. Israel emerged as a leading center of research and development in this field, particularly through the state-owned Israeli Aircraft Industries. Between 1970 and 1990, three major UAV programmes of this type were developed: the Tadiran Mastiff, IAI Scout, and RQ-2 Pioneer [17]. During the same period, rotary engines entered the unmanned aviation domain. The BTT3 Banshee, developed in the early 1980s by Target Technology Ltd., is generally recognized as the first UAV to employ a rotary engine. Early versions of the fixed-wing platform

were powered by a two-stroke, two-cylinder Normalair-Garrett engine rated at 19 kW. Subsequent development led to the Banshee Whirlwind variant, which used a Norton P73 Wankel rotary engine producing 28 kW. Rotary engines have since remained present in the UAV market, finding application in several projects over the following four decades [66].

A new avenue of development was opened by tests initiated in the United States in 2004. Until that time, all piston-engine-powered UAVs, regardless of engine type, relied on gasoline as their primary fuel. In 2004, however, the U.S. Air Force initiated efforts to employ compression-ignition engines in unmanned aerial vehicles. The key motivation was a significant simplification of logistics: fuel supplied to ground vehicles could also power the aerial component. The RQ-5 Hunter UAV, previously powered by two Moto Guzzi spark-ignition engines, was selected for the pilot program. In the MQ-5 Hunter variant, propulsion was changed to two Mercedes HFE Diesel inline three-cylinder engines. Although the logistical advantages were substantial, compression-ignition engines have not been widely adopted in unmanned aviation [16, 37].

The most recent major transformation in UAV propulsion has been the widespread adoption of electric propulsion systems. Achieving this shift required sufficient technological maturity, particularly in electrical energy storage and electronic component miniaturization. Early attempts to employ electric propulsion in unmanned aircraft date back to the 1950s, primarily in experimental platforms. A decisive breakthrough occurred in the 1990s with the development of brushless DC motors, advances in onboard electronics miniaturization, and the introduction of more efficient power controllers. The widespread availability of lithium-polymer and lithium-ion batteries after 2000 enabled a substantial increase in the energy density of onboard power systems. As a result, electric propulsion became viable for small-scale unmanned aerial vehicles. Owing to its mechanical simplicity, low cost, and ease of operation, electric propulsion remains the most prevalent propulsion type in micro- and mini-class UAVs to this day.

3. Engine types

3.1. Two-stroke spark-ignition engines

Two-stroke spark-ignition engines constitute an important group of propulsion sources for small- and medium-class unmanned aerial vehicles, particularly in applications where a high power-to-weight ratio and the structural simplicity of the propulsion system are prioritized. In a two-stroke engine, the complete working cycle is accomplished within a single crankshaft revolution, which, from a kinematic standpoint, enables relatively high power density with fewer mechanical components than in four-stroke units. From the perspective of UAV airframe integration, this translates into a favorable propulsion system mass and potentially reduced installation volume requirements, which is especially important in designs with limited internal fuselage space and stringent maximum take-off mass constraints. Two-stroke piston engines are therefore particularly attractive in UAV classes with a maximum take-off mass

below approximately 100 kg, where the trade-off between performance, cost, and system simplicity is critical [24].

At the same time, the specific characteristics of the charge-exchange process in two-stroke engines (scavenging), as well as the typical lubrication solutions employed, impose several operational limitations. The most significant of these include elevated emissions of toxic exhaust components, high sensitivity to intake system operating conditions, and less favorable acoustic characteristics compared to four-stroke engines. In UAV applications, these factors are relevant not only from an environmental standpoint, but also due to increased acoustic detectability – particularly in military missions – and the dynamic loading of the airframe structure resulting from higher vibration amplitudes. Furthermore, conventional carburetor-based fuel delivery systems may degrade the mixture with increasing flight altitude, increasing the risk of power loss and incomplete combustion. One approach to mitigating this effect involves using fuel injection systems and more advanced mixture control strategies that vary with ambient conditions [24, 48].

From a systems-level perspective, it should be emphasized that the advantage of two-stroke engines in terms of specific power does not necessarily translate into superior mission-level energy efficiency. In endurance-oriented missions, specific fuel consumption and the stability of operating parameters over a wide range of loads are of paramount importance. Despite their favorable mass characteristics, two-stroke engines may require a higher fuel mass flow to deliver a comparable amount of useful shaft energy under cruise conditions, thereby reducing the overall benefit associated with engine mass reduction. For this reason, in many modern UAV classes, a gradual shift can be observed in which two-stroke engines are increasingly confined to platforms that prioritize short-duration flight, low acquisition cost, operational simplicity, and ease of maintenance over maximum endurance.

The literature on UAV propulsion systems indicates that the RQ-2 Pioneer (Fig. 1) platform employed a spark-ignition, two-stroke, two-cylinder piston engine manufactured by Fichtel & Sachs, installed in a pusher propeller configuration. The engine features a horizontally opposed layout and delivers approximately 20 kW of power. This propulsion system was selected for its favorable power-to-weight ratio and ease of integration into a short-range reconnaissance UAV [24, 57].



Fig. 1. RQ-2 Pioneer [57]

Another example of a two-stroke propulsion application is the Sperwer A/B unmanned aerial vehicle, which is equipped with a spark-ignition piston engine operating on the two-stroke cycle – the Bombardier-Rotax 582/562UL [12] (Fig. 2).



Fig. 2. Engine Rotax 582/562UL [12]

Technical sources emphasize the use of gasoline-class fuel, consistent with the operational practice for small spark-ignition engines in UAVs of this category. Such a propulsion choice aligns with the requirements of medium-mass tactical UAVs, providing relatively high specific power at moderate system complexity, at the expense of higher specific fuel consumption compared to four-stroke or heavy-fuel engines [81].

3.2. Four-stroke spark-ignition engines

Four-stroke spark-ignition engines are employed in UAVs primarily in applications where higher cruise efficiency, increased operational durability, and improved controllability of operating parameters over a wide load range are critical. Unlike two-stroke units, the four-stroke engine completes its working cycle over two crankshaft revolutions, with charge exchange controlled by a valve train, enabling more precise shaping of the intake and exhaust processes. In UAV applications, this translates into more stable operation during long-endurance missions, where low specific fuel consumption, repeatable performance, and relatively long maintenance intervals are essential.

The literature on propulsion system selection as a function of UAV mass indicates that four-stroke engines are particularly common in platforms with a maximum take-off mass exceeding approximately 50 kg, as they offer a favorable compromise between fuel economy and reliability, despite their higher mass and more complex mechanical architecture compared to two-stroke engines. Additional advantages include a dedicated lubrication system that eliminates the need for fuel oil premixing and reduces oil combustion emissions. Consequently, four-stroke engines are better suited to long-duration operations, where maintenance predictability and the mitigation of performance degradation over time are key considerations [24].

Conversely, the application of four-stroke engines in UAVs necessitates a deliberate approach to mechanical and thermal integration. The greater number of moving compo-

nents, the inclusion of a valve train, and often more elaborate cooling and fuel supply systems increase overall mass and system complexity. This may require reinforcement of the engine mounting structure, careful consideration of vibration isolation, and the provision of stable cooling conditions during prolonged low-airspeed flight. An additional issue is susceptibility to intake system icing in carbureted configurations, as well as the need to ensure mixture control as a function of altitude, ambient temperature, and pressure. In many applications, these challenges are addressed through the use of fuel injection systems and turbocharging, which help to maintain power at altitude and improve operational flexibility under varying mission conditions [24].

In practice, four-stroke spark-ignition engines are widely represented in the MALE class, as well as in tactical and logistical platforms where endurance and payload capability must be achieved within reasonable operating costs. From the standpoint of acoustic and thermal detectability, four-stroke engines often offer a more favorable signature than two-stroke units at comparable power levels; however, actual detectability depends on a range of system-level factors, including propeller characteristics, exhaust attenuation, thermal insulation, and mission-specific operating regimes.

One of the most recognizable UAVs with a four-stroke propulsion system is the MQ-1 Predator, primarily employed in military missions. The MQ-1 is powered by a four-cylinder Rotax 914F engine (Fig. 3) rated at 115 hp. The platform is used for medium-range missions with an operational radius of up to 1,200 km, with its primary roles including reconnaissance and precision strikes using onboard weaponry [24, 31, 52].

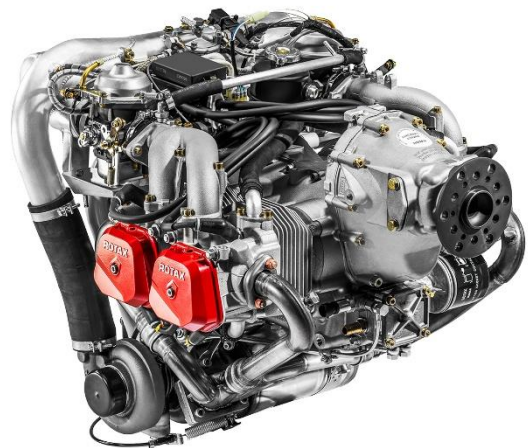


Fig. 3. Engine Rotax 914F [31]

Another example is the Bayraktar TB2, designed and manufactured in Turkey, which is equipped with a four-stroke Rotax 912 engine [81]. The Bayraktar TB2 is a MALE-class tactical UAV intended for intelligence, surveillance, and reconnaissance missions, as well as target acquisition and designation using an electro-optical payload that includes a laser rangefinder. In a typical operational profile, the platform conducts long-duration area surveillance, identifies and tracks targets, and transmits near-real-

time data to a ground control station, thereby supporting tactical-level decision-making. In its armed configuration, the TB2 is also employed for limited-scale precision-strike missions using lightweight guided munitions carried on underwing hardpoints, combining ISR with strike capabilities.

Another configuration is the RQ-5 Hunter (Fig. 4), equipped with two four-stroke, two-cylinder, spark-ignition engines manufactured by Moto Guzzi, delivering a total power output of approximately 100 kW [24]. The RQ-5 Hunter is a UAV primarily intended for reconnaissance, surveillance, and target designation in support of ground forces. In operational use, the platform served as a real-time imagery source, assisting tactical commanders with situational assessment, monitoring adversary activity, and surveillance of selected areas and lines of communication. The system was employed in long-endurance observation missions, including expeditionary operations. A particularly significant and frequently emphasized role of the Hunter was target acquisition and artillery fire correction. In this mission profile, the UAV provided continuous updates on target position and movement, enabling battle damage assessment and shortening the “detect-decide-engage” cycle in land operations [11].



Fig. 4. RQ-5 Hunter [11]

Four-stroke spark-ignition piston engines in UAVs are therefore primarily used on platforms that require high cruise efficiency, stable operation during long-endurance missions, and enhanced reliability. Compared to two-stroke units, they generally offer lower specific fuel consumption, improved combustion control, and reduced vibration amplitudes, at the expense of increased mass and greater mechanical complexity. As a result, they dominate in tactical fixed-wing UAVs and MALE-class systems, where endurance and operational predictability of the propulsion system are the primary design drivers.

3.3. Compression-ignition engines

Compression-ignition (CI) engines in UAVs are developed and implemented primarily in response to two major categories of requirements:

- increased energy efficiency in long-endurance missions
- fuel logistics commonality in military and specialized applications.

The combustion process in CI engines is based on the auto-ignition of the fuel–air mixture, achieved by compressing the intake air to a pressure and temperature suffi-

cient to initiate ignition, thereby eliminating the need for spark ignition. This mechanism, however, requires highly precise fuel injection and accurate control of the combustion process. Under typical UAV mission profiles, characterized by extended cruise segments and steady-state operating points, CI engines can offer a significant advantage through lower specific fuel consumption, which directly translates into increased endurance or a higher useful payload for a given onboard fuel mass.

From the perspective of operational safety and ground handling, heavy fuels generally exhibit lower volatility than gasoline, which may reduce fire hazards during maintenance operations and emergency situations. For military users, a particularly important argument is the ability to power multiple systems with a single fuel class (e.g., JP-8 or Jet A), thereby simplifying supply chains in expeditionary operations. Consequently, in the literature and in review studies on UAV propulsion systems, heavy-fuel engines are frequently associated with platforms intended for reconnaissance-strike roles and long-duration patrol missions [81].

These benefits, however, are achieved at the cost of substantial design and integration challenges. Compared to spark-ignition engines, CI units typically operate at higher peak in-cylinder pressures, resulting in increased mechanical loads and, consequently, potentially higher component mass or the need for advanced materials and structural solutions to ensure adequate strength. In addition, the CI combustion process is strongly dependent on the quality of fuel atomization and air-fuel mixing, which, under high-altitude flight conditions characterized by low air density and low ambient temperatures, may degrade ignition stability and lead to starting difficulties or unstable operation at low temperatures. These issues are sufficiently critical that studies of heavy-fuel propulsion emphasize technical barriers related to combustion stability, reliable ignition under adverse conditions, and the need for auxiliary systems to support engine start-up and sustained operation [70].

From a systems-level perspective, the use of a CI engine also affects the overall UAV architecture. Such engines often require advanced fuel injection control systems, effective boosting solutions to maintain power at altitude, and careful thermal management. These requirements influence mass, system complexity, and life-cycle costs; therefore, selecting CI propulsion is most justified in applications where the efficiency gains and heavy-fuel logistics advantages outweigh the increased integration complexity.

A 2023 review study [81] identifies several representative UAV platforms employing heavy-fuel propulsion systems. Among them is the MQ-1C Gray Eagle, for which the compilation reports the use of an HFE-class engine rated at approximately 180 hp, capable of operating on diesel fuel as well as aviation jet fuels, with reported endurance of approximately 42 hours and a payload capacity of 227 kg. The same review includes the ScanEagle 2 UAV (Fig. 5), equipped with a piston propulsion system adapted to heavy fuels such as JP-5 or JP-8, with reported endurance of 20 to 28 hours and a payload capacity of 5 kg [44]. The authors interpret this example as indicative of the trend toward fuel commonality even in smaller UAV systems. In addition, the CH-4 platform is cited as employing a propulsion unit de-

scribed as an HFE engine operating on heavy fuel, with reported endurance of approximately 30 hours and a payload capacity of 115 kg, confirming the adoption of the heavy-fuel concept beyond Western UAV systems.

In summary, compression-ignition engines represent a propulsion solution for UAVs that maximizes the effectiveness of long-endurance missions and simplifies fuel logistics. However, their successful implementation requires a high level of design maturity in combustion control, boosting, cold- and high-altitude-start capability, and the mechanical and thermal integration of the entire propulsion system [64].



Fig. 5. ScanEagle 2 [44]

3.4. Rotary piston engines (Wankel engines)

The rotary piston engine, commonly referred to as the Wankel engine, was developed by Felix Wankel in the 1950s as an alternative to conventional reciprocating piston engines. Its design is based on the rotation of a triangular rotor within an epitrochoid chamber, eliminating the standard crankshaft mechanism, valves, and most of the moving components typical of four-stroke piston engines. While Wankel engines did not achieve dominance in automotive applications, their unique design characteristics make them particularly attractive for unmanned aerial vehicle applications.

A primary limitation of conventional piston engines is the need to convert reciprocating motion into rotational motion. In Wankel engines, this conversion is unnecessary, reducing parts and simplifying the overall engine structure. The reduced component count translates into lower mass and smaller dimensions. In aviation applications, the most significant advantages of rotary engines include a high power-to-weight ratio, relatively low total engine mass, and reduced vibration levels compared to conventional piston engines. However, their use in UAVs also entails drawbacks, including lower thermal efficiency, higher emissions, and accelerated component wear. Rotary piston engines are most commonly used in medium-sized UAVs intended for tactical operations, where electric propulsion is insufficient in power and energy density, while conventional piston engines may be too heavy for relatively light-weight airframes [1, 17, 49].

Representative applications of rotary piston engines in UAVs include:

- Hermes 450 (Fig. 6) – An Israeli medium-range, multi-role UAV developed in the mid-1990s by Elbit Systems. The airframe, in a characteristic parasol-wing configuration, is powered by a UEL AR801 engine producing 40 kW. Post-2008 production units, designated Hermes 450W, were equipped with the upgraded UEL AR902(W) engine rated at 49 kW. The primary advantage of this UAV is its long endurance relative to other platforms in its class, achieving up to 17 hours, or even 30 hours in the Hermes 450LE variant. The program has been highly successful in export markets, with units delivered to the armed forces of multiple countries [7, 23, 30, 78].



Fig. 6. Elbit Hermes 450 [7]

- AAI RQ-7 Shadow – The most widely produced UAV with a rotary piston engine. Since its market introduction in 2002, over 500 units have been delivered, predominantly to the U.S. Army. The twin-boom airframe is powered by an AR741-1101 engine (Fig. 7) producing 28 kW, driving a two-blade pusher propeller. Designed for short-range reconnaissance, the UAV can fly for 6 hours at 130 km/h. Missions are supported by the onboard POP300 electro-optical payload produced by Israel Aerospace Industries, which includes both a daylight camera and an infrared sensor [2, 31, 36, 77].



Fig. 7. AR741-1101 engine [2]

- Nostromo Yará – The first serially produced UAV designed in South America, introduced in 2006 by the Argentine company Nostromo Defensa. Its propulsion system consists of an air-cooled Sonic 35 rotary piston engine, the first rotary unit designed to operate on JP-8, the standard turbine fuel for NATO military forces. The engine produces 5.9 kW at the three-blade pusher propeller. With a cruise speed of 115 km/h, the maximum flight endurance is six hours [9].
- Schiebel Camcopter S-100 – An Austrian-produced rotary-engine UAV helicopter, introduced in 2005 to partially replace manned helicopters aboard naval vessels. The UAV, with a maximum take-off mass of 200 kg, is powered by a Schiebel S1 engine rated at 41 kW. Since 2012, an upgraded S2 engine version has been offered, compatible with JP-5, Jet A-1, and JP-8 fuels. The 57 L internal fuel capacity enables a 6-hour flight, which can be extended to 10 hours with additional external fuel tanks. The Camcopter S-100 has gained international recognition, with units delivered to military and civilian institutions in 29 countries [6, 69].

3.5. Pulsejet engines

Pulsejet engines constitute a distinct class of jet propulsion systems, characterized by mechanical simplicity and the absence of complex components such as compressors or turbines. Their operation is based on cyclic combustion of a fuel–air mixture within a combustion chamber, with or without valves, generating periodic bursts of hot gases that produce thrust in an impulsive manner. This design results in extremely low production costs, while the lack of rotating components makes the engine tolerant to ingestion of small debris and particulate matter, reducing the risk of damage and simplifying maintenance [62, 65].

The efficiency of pulsejet engines is generally very low, which limits their application to cases where simplicity and low manufacturing cost outweigh propulsion efficiency. Typically, these engines are used in expendable UAVs, such as strike drones or target drones, where the one-time nature of the mission renders high engine efficiency unnecessary, and the production cost of propulsion units is a critical factor. Historically, pulsejets were also employed before the development of reliable turbojet engines [62, 65].

Since the introduction of the first pulsejet-powered UAV, the German V-1, several unmanned aircraft have employed this type of propulsion:

- TD2D Katydid – An American target drone introduced by the U.S. Navy in 1942. To achieve higher maximum speeds that better simulate contemporary manned combat aircraft, a pulsejet engine was selected over the piston engines commonly used at the time. The McDonnell XPJ40-MD-2 engine was developed specifically for this application, generating 2.9 kN of thrust. The UAV could reach 400 km/h, compared to 200–220 km/h for piston-engine drones of the same era. Fuel capacity allowed for a 40-minute flight [56].
- Aviolanda AT-21 – A target UAV developed in 1955 by Aviolanda, becoming the first Dutch UAV produced in series. The aircraft, in a conventional layout, was powered by a SNECMA AS-11 Écrevisse pulsejet producing

0.85 kN of thrust. Maximum speed reached 346 km/h, while a cruise speed of 285 km/h allowed a 70-minute flight on 120 L of fuel [21].

- Scitor-D (Fig. 8) – A UAV presented in March 2024 by Wave Engine Corp. as a technology demonstrator for the modern J-1 pulsejet engine. The primary advantages of this propulsion unit are its extreme mechanical simplicity, absence of moving parts, and wide range of compatible fuels. According to the manufacturer, the engine can operate on gasoline, aviation fuels such as JET-A and JP-8, and E85 biofuel. The unit has a mass of 7.5 kg and generates up to 0.24 kN of thrust, suitable for small UAVs. Additionally, the company is developing the K-1 engine, intended for larger airframes, capable of producing approximately 1 kN of thrust [22, 80, 82].



Fig. 8. Wave Engine Corp. Scitor-D [22]

3.6. Turbojet engines

Turbojet engines generate thrust by accelerating a stream of exhaust gases produced by the combustion of aviation fuel in the combustion chamber and subsequent expansion through the turbine and exhaust nozzle. Compared to conventional piston engines, they are characterized by a very high thrust-to-weight ratio, a mechanically simple architecture, and the capability to operate over a wide range of altitudes and flight speeds. These features make turbojet engines suitable for unmanned aerial vehicles intended for high-speed, high-altitude flight [1, 53].

The principal disadvantage of turbojet engines is their high specific fuel consumption, which significantly limits aircraft endurance. For this reason, turbojets are most commonly employed in strike UAVs and aerial target drones. Both categories share a requirement for high flight speed combined with relatively short mission durations, typically not exceeding several hours. In addition, the expendable or single-use nature of such missions favors propulsion systems that are inexpensive and structurally uncomplicated—criteria that turbojet engines satisfy well [1, 17].

In reconnaissance UAV applications, turbojet engines are used far less frequently and are typically replaced by turbofan engines. This is due not only to turbojet high fuel consumption but also to their elevated acoustic output and infrared signature, which substantially increase platform detectability. The application of turbojet engines in smaller tactical-class UAVs is also problematic, as most such platforms are designed to operate at relatively low airspeeds, typically between 50 and 150 km/h. At these speeds, turbojet efficiency is markedly lower than at higher velocities,

rendering their use technically and operationally unjustified [1, 17].

Representative examples of UAVs powered by turbojet engines include:

- Ryan BQM-34 Firebee – An aerial target drone produced by the Ryan Aeronautical Company starting in 1960. It is powered by a Continental J69-T-29A turbojet engine delivering a maximum thrust of 7.6 kN, enabling the aircraft to reach speeds of up to 1,110 km/h. Launch is performed either from a ground-based launcher using a rocket-assisted takeoff system or airborne from pylons mounted on specially modified Lockheed DC-130 aircraft. At a cruise speed of approximately 1010 km/h, the onboard fuel supply supports a flight endurance of about 75 minutes. More than 6,500 Firebee units were produced, with some variants remaining in service to this day [59].
- Nord Aviation CT20 – A French target drone developed in 1957 and powered by a Turbomeca Marboré II turbojet engine generating 3.9 kN of thrust. Launch was conducted from a ramp using two auxiliary rocket boosters. The UAV could sustain a maximum flight duration of approximately 55 minutes and reach speeds of up to 900 km/h. Produced in around 1500 units, the CT20 served as one of NATO's primary aerial target systems during the second half of the 20th century [13, 74].
- Shahed-238 (Fig. 9) – An Iranian strike UAV introduced in 2023. The platform, carrying a 50-kg warhead, is powered by a Toloue-10 turbojet engine, a derivative of the Czech PBS TJ-100 (Fig. 10). The engine produces up to 1.5 kN of thrust, enabling a maximum airspeed of approximately 600 km/h. Flight endurance is estimated at around 2 hours. Compared to the piston-engine-powered Shahed-136, the adoption of turbojet propulsion increased maximum speed by more than 400 km/h, significantly reducing transit time over hostile territory and complicating air defense countermeasures [3, 71, 73].



Fig. 9. Shahed-238 [73]

- UJ-23 Topaz – A multirole UAV produced by UKRJET since 2023, featuring forward-swept wings and a V-tail configuration. The platform is powered by a turbojet engine, although specific engine type and parameters remain classified. Publicly available data indicate a maximum speed of approximately 800 km/h and a maximum flight time of about 20 minutes. A deriva-

tive version, the UJ-25 Skyline, has been developed as a loitering munition equipped with a 10-kg warhead [79].



Fig. 10. PBS TJ-100 turbojet engine [3]

3.7. Turboprop engines

A turboprop engine is a type of gas turbine propulsion system in which most of the fuel's chemical energy is converted into mechanical power to drive a propeller, rather than directly generating thrust from the exhaust stream. Ambient air is drawn into the engine and compressed in a compressor, then directed into the combustion chamber, where fuel and air are burned. The hot exhaust gases expand through a turbine, which extracts mechanical energy. In a turboprop engine, most of the turbine's power is transmitted to a driveshaft and then, through a reduction gearbox, to the propeller. The gearbox reduces rotational speed while increasing torque, allowing the propeller to rotate at an optimal speed below the turbine shaft speed [1, 18].

The primary advantage of turboprop engines is their low fuel consumption during flights at moderate speeds. Within the typical speed range of 200–450 km/h, turboprop propulsion offers the highest fuel efficiency among all available propulsion types. This characteristic makes turboprops the preferred choice for medium- and long-range reconnaissance UAVs, where maximum speed is secondary to extended endurance [18, 84].

Turboprop propulsion, however, imposes limitations. Maximum airspeed is considerably lower than that achievable with turbojet or turbofan engines. Due to the propeller's operational characteristics, turboprop UAVs generally do not exceed speeds of approximately 500 km/h. They are also typically constrained to medium altitudes, generally below 30–35,000 ft. Consequently, turboprops occupy a performance niche between piston and jet engines: they operate faster than piston-powered UAVs but slower than jet-powered systems, and they achieve peak efficiency at altitudes too high for piston engines but below optimal turbofan altitudes [18, 84].

Representative UAVs employing turboprop propulsion include:

- MQ-9A Reaper (Fig. 11) – A U.S. MALE-class (Medium Altitude, Long Endurance) UAV developed by General Atomics Aeronautical Systems in 2001 as an evolution of the MQ-1 Predator. The platform is designed for long-endurance flights, advanced reconnaissance, and precision strike missions over large areas at medium-to-high altitudes. Propulsion is provided by a Honeywell TPE331-10GD turboprop engine (Fig. 12) delivering up

to 670 kW to a pusher propeller. The aircraft achieves a maximum speed of 480 km/h, while endurance reaches approximately 27 hours (up to 34 hours in the MQ-9 ER variant). Coupled with advanced sensor suites, including radar, electro-optical payloads, and electronic warfare modules, the MQ-9 represents a versatile and highly capable system for modern operations [42, 75, 85].



Fig. 11. MQ-9A Reaper [75]

- Bayraktar Akinci – A Turkish HALE-class (High Altitude, Long Endurance) multirole UAV developed in 2019 by Baykar Makina. The platform uses two turbo-prop engines, with specific models depending on the variant: AI-450T (340 kW each) in Version A, PT6A-135A (560 kW) in Version B, and Motor Sich MS-500V (630 kW) in Version C. The UAV has an operational ceiling of 30,000 ft and a cruise speed of 280 km/h. Onboard fuel allows for a 25-hour flight with a maximum range of 7500 km. Eight underwing hardpoints enable the deployment of a wide range of armaments, with a total payload of up to 1500 kg [19, 38].



Fig. 12. Honeywell TPE331-10GD [42]

- IAI Heron TP – An Israeli HALE-class UAV introduced in 2007 by Israel Aerospace Industries. The aircraft features a twin-boom configuration and is powered by a Pratt & Whitney Canada PT6A-67A turboprop engine producing 883 kW and driving a four-blade pusher propeller. The platform has an operational ceiling of 41,000 ft and endurance exceeding 30 hours, positioning it among the leading long-range reconnaissance UAVs globally [8, 43].
- TAI Turna – A delta-winged target drone introduced in 2001 by Turkish Aerospace Industries. The original version was powered by a rotary piston engine (AR741), while a 2008 upgrade introduced a turboprop engine, the TEI-TP-1X, rated at 28 kW. The UAV reaches a maximum speed of 330 km/h and has an endurance of up to 90 minutes. The use of turboprop propulsion in a

small, low-altitude UAV makes this a unique design within its class [8, 46, 54, 77].

3.8. Turboshaft engines

Turboshaft engines belong to the class of turbine powerplants designed to deliver mechanical power to an output shaft rather than producing direct thrust. Essentially, they are gas turbines adapted for continuous rotational power output, which can then be utilized for mechanical propulsion. Unlike turboprop engines, turboshafts do not include a gearbox or propeller, although the operating principle of the core engine is the same. In aviation applications, turboshhaft engines are often converted into turboprops by adding a reduction gearbox and a propeller. A notable example is the Rolls-Royce M250 turboshhaft engine, which was adapted for unmanned aviation into a turboprop unit used in demonstrators such as the BAE Mantis [61, 84].

Turboshhaft engines are primarily used in rotorcraft UAVs, powering the main rotor. They are also employed in hybrid UAV propulsion systems, where the turboshhaft generates electrical power that drives electric motors connected to rotors or propellers [61, 84].

Representative UAVs employing turboshhaft technology include:

- Bell Eagle Eye – A VTOL UAV developed in the late 1990s for the U.S. Navy VT-UAV program. A Pratt & Whitney Canada PW207D turboshhaft engine housed in the fuselage transmitted power via a gearbox to two rotors mounted in movable wingtip pods. The orientation of these pods controlled the flight mode: in vertical position, the rotors enabled VTOL operation, and in horizontal position, they provided forward flight. The program was ultimately discontinued following a prototype accident in 2006 [32, 60].
- Boeing A160T Hummingbird (Fig. 13) – A UAV helicopter first flown in 2007, designed for extended endurance and long-range operations. Early A160 variants used Subaru piston engines, but in 2007, the platform was upgraded with Pratt & Whitney Canada PW207D turboshhaft engines (Fig. 14). During testing, the UAV achieved 20 hours of endurance and over 4,000 km of range. The project also featured an advanced autonomous flight control system. Despite technical achievements, the program was suspended in December 2012 [5, 27, 41].



Fig. 13. Boeing A160T Hummingbird [27]

- MQ-8B Fire Scout – An unmanned helicopter introduced in 2006. Its main rotor is powered by a Rolls-Royce 250-C20W turboshhaft engine, enabling 8-hour

flights with a 77 kg payload. Lessons learned from this program led to the development of the larger MQ-8C, powered by a Rolls-Royce 250-C47 B engine. Between 2009 and 2023, the U.S. Navy acquired 30 MQ-8B units and 38 MQ-8C units [15, 40].



Fig. 14. Pratt & Whitney Canada PW207D [41]

- S-70UAS U-Hawk – An unmanned variant of the UH-60L Black Hawk transport helicopter developed by Lockheed Martin. Power is provided by two General Electric T700-GE-701C turboshaft engines, as in the manned version. The cockpit was replaced with cargo space and a loading ramp, increasing usable cargo volume by 25% and simplifying loading operations. Prototype flights are scheduled for 2026 [50, 51].

3.9. Turbofan engines

A turbofan engine is a type of gas turbine engine in which a large fan at the inlet generates a substantial airflow that bypasses the engine core. This configuration enables higher propulsive efficiency and greater thrust at subsonic speeds compared with classical turbojet engines. The high bypass ratio reduces specific fuel consumption and noise levels, which are particularly advantageous for long-endurance, long-range UAV missions. In addition, the reduced acoustic and thermal signatures improve survivability by reducing detectability.

Turbofan engines are primarily employed in larger, technologically advanced UAV platforms where aerodynamic performance, range, and high-altitude capability are key design requirements. However, their structural complexity, relatively high mass, and large physical dimensions generally preclude their application in small UAVs [72, 84]. UAVs powered by turbofan engines are predominantly represented by MALE- and HALE-class platforms:

- MQ-20 Avenger – A U.S. multi-role MALE-class UAV developed by General Atomics as an evolution of the MQ-1 Predator, with its maiden flight in 2009. The airframe features a flattened, blended configuration that enhances stealth characteristics. Propulsion is provided by a Pratt & Whitney Canada PW545B turbofan engine generating 21.4 kN of thrust with a bypass ratio of 3.8. Internal fuel tanks accommodate approximately 3500 kg of fuel, enabling endurance of more than 18 hours at altitudes up to 50,000 ft. The maximum speed is 740 km/h, with a cruise speed of approximately 650 km/h. In 2016, the extended-range MQ-20 ER variant conducted its first flight, featuring increased wingspan and

enlarged fuel tanks, which extended endurance to approximately 23 hours [33, 34, 35].

- RQ-4 Global Hawk (Fig. 15) – A HALE-class strategic reconnaissance UAV introduced in 2003 by Northrop Grumman Corporation. The primary design objective was to create an unmanned platform capable of persistent, multi-sensor strategic surveillance. A Rolls-Royce AE 3007H turbofan engine (Fig. 16) producing 34 kN of thrust accelerates the 14-ton air vehicle to a maximum speed of 629 km/h. At a cruise speed of 570 km/h and an altitude of 65,000 ft, the 7800 kg fuel load supports missions exceeding 34 hours and a range of approximately 22,800 km, making the RQ-4 a unique solution on a global scale. Its reconnaissance capability is enabled by an onboard sensor suite including electro-optical and infrared cameras, an AESA radar, and a ground moving target indicator (GMTI) [10, 58, 61, 63, 68, 76].



Fig. 15. RQ-4 Global Hawk [61]

- Bayraktar Kızılelma – An unmanned combat aircraft developed in 2022 by the Turkish company Baykar. The airframe incorporates stealth technologies, while propulsion is provided by an Ivchenko-Progress turbofan engine. The Kızılelma-A variant is powered by the AI-25TLT engine delivering 16.9 kN of thrust, whereas the Kızılelma-B variant uses the AI-322F engine generating 24.5 kN. Due to the mission profile envisioned for this UAV, high speed and low observability were prioritized at the expense of endurance, which is approximately 3 hours. A key advantage is the maximum speed of up to 1100 km/h. Two internal hardpoints and six external wing stations allow the carriage of up to 1500 kg of ordnance [14, 20, 46].

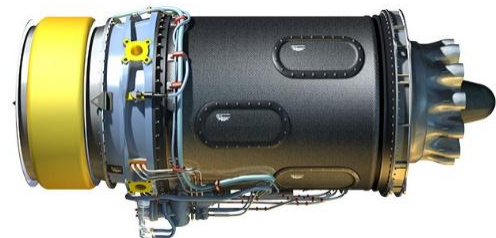


Fig. 16. Rolls-Royce AE 3007H [68]

4. Conclusions

This article provides a comprehensive review of the development and current applications of combustion-based propulsion systems in unmanned aerial vehicles (UAVs),

highlighting that engine selection remains a key factor shaping mission profiles, including endurance, range, operational altitude, speed, and payload capacity. Against the backdrop of historical evolution from early UAVs with simple piston engines, through the era of flow engines used in high-speed systems, to the contemporary diversification of propulsion solutions, it is evident that combustion engines continue to play a critical role, particularly in medium and large UAV classes, where liquid fuels provide an energy advantage over batteries.

The conducted review demonstrates that no single propulsion architecture is universally optimal for all UAV categories and mission profiles. Instead, propulsion system selection is strongly dependent on operational requirements, including endurance, flight speed, payload capacity, altitude, detectability, and logistical constraints.

Two-stroke spark-ignition engines remain particularly suitable for lightweight tactical UAVs due to their high power-to-weight ratio, mechanical simplicity, and low acquisition cost. However, their relatively high specific fuel consumption, elevated acoustic signature, and reduced operational efficiency limit their applicability in long-endurance missions.

Four-stroke spark-ignition engines currently represent one of the most balanced propulsion solutions for medium-class and MALE UAVs, offering favorable cruise efficiency, improved reliability, and stable long-duration operation. Compression-ignition and heavy-fuel engines offer additional operational advantages, including lower fuel consumption and standardized fuel in military logistics, though their implementation requires advanced fuel injection systems, turbocharging, and sophisticated thermal management.

Turbine propulsion systems dominate high-speed and high-altitude UAV applications. Turbojet engines are primarily used in strike UAVs and target drones where speed and simplicity are prioritized over endurance. Turbofan and turboprop engines, in turn, enable long-endurance HALE and MALE operations while maintaining relatively high cruise efficiency and operational ceilings.

Rotary piston engines occupy an important niche in medium tactical UAVs due to their compact size, low vibration, and favorable power density. At the same time, pulsejet propulsion remains limited to expendable, low-cost systems due to poor fuel efficiency and a high acoustic signature.

Table 1 summarizes the key characteristics, advantages, and operational limitations of the major propulsion system categories used in UAVs.

Identified development trends include:

- Continued optimization of piston engine efficiency and reliability for long-endurance flight profiles
- Increasing importance of compatibility with heavy-fuel types in military applications
- Gradual adoption of hybrid solutions, where a combustion engine operates near peak efficiency as an energy source, while the primary propulsion is delivered electrically.

The identified development trends indicate continued efforts to improve the efficiency, reliability, and operational

Table 1. Comparison of engine characteristics

Engine type	Typical UAV Class	Main advantages	Main limitations
Two-stroke SI	Small tactical UAVs	Low mass, simplicity, low cost	High fuel consumption, noise, emissions
Four-stroke SI	Tactical / MALE UAVs	Reliability, stable operation	Higher mass and complexity
Compression-ignition (HFE)	MALE / HALE UAVs	Fuel economy, JP-8 compatibility	Complex injection and thermal systems
Rotary piston (Wankel)	Medium tactical UAVs	Compact size, low vibration	Lower thermal efficiency
Pulsejet	Expendable UAVs	Extremely simple structure	Very poor efficiency, noise
Turbojet	Strike UAVs	High speed, compactness	High fuel consumption
Turboprop	MALE / HALE UAVs	Excellent cruise efficiency	Lower maximum speed
Turboshaft	Rotary-wing UAVs	Rotorcraft capability	Complex transmission systems
Turbofan	HALE / UCAV platforms	High altitude and endurance	High cost and dimensions

durability of piston propulsion systems for long-endurance UAV missions. Increasing attention is also being directed toward propulsion systems compatible with heavy fuels used in military logistics, particularly JP-8 and Jet-A. Another important direction involves the progressive implementation of hybrid propulsion architectures, in which an internal combustion engine functions primarily as an onboard power generation unit operating near its optimal efficiency range, while thrust is produced by electrically driven propulsion systems.

These developments suggest that future research should increasingly focus on mission-oriented propulsion integration, reduction of acoustic and thermal signatures, enhanced operational resilience under demanding environmental conditions, and lifecycle cost optimization in response to growing operational and environmental expectations.

Future advancements in UAV propulsion are likely to be closely linked to the expansion of hybrid and integrated propulsion concepts combining combustion engines with electric propulsion technologies. Such systems may improve overall energy management efficiency while simultaneously reducing detectability during critical phases of operation through lower acoustic and infrared emissions.

Another significant technological direction is the broader adoption of heavy-fuel propulsion systems aligned with military fuel standardization requirements. At the same time, tightening environmental regulations and sustainability objectives are expected to stimulate further research into sustainable aviation fuels, synthetic fuels, and low-emission combustion strategies applicable to unmanned aircraft.

In addition, the role of intelligent propulsion management systems is expected to increase substantially. Future UAV propulsion architectures will likely incorporate autonomous engine control, artificial intelligence-supported

energy management, predictive maintenance systems, and adaptive optimization algorithms designed to improve reliability and operational efficiency. Parallel research efforts will also focus on minimizing the detectability of the propulsion system, increasing robustness in harsh operating environments, and reducing long-term operational costs.

Consequently, future investigations should address not only improvements in propulsion efficiency but also broader issues related to mission integration, autonomy, survivability, operational flexibility, and the sustainability of UAV propulsion systems.

Nomenclature

CI	compression ignition	IR	infrared
SI	spark ignition	ISR	intelligence, surveillance and reconnaissance
UAV	unmanned aerial vehicle	JP-5	jet propellant 5
ABC	British engine manufacturer	JP-8	jet propellant 8
AESA	Active Electronically Scanned Array	Jet A	aviation turbine fuel
AFIS	Aerodrome Flight Information Service	MALE	medium altitude long endurance
DC	direct current	MTOM	maximum take-off mass
EO	electro-optical	RFC	Royal Flying Corps
GMTI	ground moving target indicator	UAV	unmanned aerial vehicle
HALE	high altitude long endurance	UEL	UAV Engines Ltd.
HFE	heavy fuel engine	VTOL	vertical take-off and landing

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The integration of the fuel additives and the mathematical modeling approach to enhance the engine exhaust emissions

ARTICLE INFO

The emission characteristics of a Perkins AD4.248 four-cylinder, four-stroke diesel engine with Diesel Smoke Stop (DSS) enhancement were evaluated for several loads. The engine was operated using four parameters, including a DSS additive concentration of 3 to 6 ml×l⁻¹ added to diesel fuel, a machine body vibration (MBV) of 100 to 300 ms⁻², an engine speed (ES) of 1500 to 2000 rpm, and an engine operation time (EOT) of 1 to 3 hours. To improve engine performance, the effects of engine optimization on engine exhaust gases, including carbon monoxide (CO), nitrogen monoxide (NO), and hydrogen sulfide (H₂S), were evaluated using Response Surface Methodology (RSM). The results showed that adding 3 ml of DSS fuel improver per liter of diesel fuel resulted in the lowest overall CO, NO, and H₂S emissions. The lowest exhaust gas emissions were achieved at a DSS additive concentration of 3 ml×l⁻¹, at an engine speed of 1500 rpm, an engine operating time of 1 hour, and a vibration of 100 ms⁻². The RSM models demonstrated acceptable performance in predicting each exhaust emission with an accuracy of 79.13%, 79.35%, and 79.95% for CO, NO, and H₂S, respectively.

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1. Introduction

The global community relies on a continuous energy supply because it is vital to many critical sectors, including manufacturing, transportation, and power generation. Fossil fuels, such as natural gas, coal, gasoline, and diesel, remain the primary sources of energy to meet today's global energy demand. However, concerns about their sustainability have led to the use of alternative fuels, which have the potential to reduce environmental pollution and decrease dependence on depleting, non-renewable fossil fuels [14].

As the global community increasingly strives to achieve net-zero emissions and sustainable development goals, fuel additives have emerged as suitable adjuncts to fossil fuels. Fuel improvers are chemical additives added to fuel to enhance combustion efficiency, improving engine performance and reducing fuel consumption. Their effect on exhaust emissions can be positive in many cases, as they reduce pollutants resulting from fuel combustion and can substantially decrease greenhouse gas emissions. Using diesel fuel improvers can help many professions transition to a more sustainable, low-carbon society while offering social and economic benefits. However, it is essential to carefully consider the impacts of diesel fuel additives on many aspects, including agricultural operations and agricultural production security, and to ensure that their effects on fuel are managed and used sustainably, while accounting for the pros and cons of these additives [3, 4]. Fuel additives mixed with conventional diesel fuels offer several benefits over pure fossil fuels, as they are non-toxic and environmentally friendly owing to their physicochemical properties [1, 4].

Alcohols can be blended as improvers with gasoline and diesel fuel. Several studies have investigated the effects of alcohol-based improvers on petrol and diesel engines. It is known that the low cetane number of alcohol fuels provides a homogeneous, lean mixture that can reduce NO_x emis-

sions and smoke and achieve good thermal efficiency in a high-load engine [10]. However, pure alcohol has a very low cetane number and cannot be run in diesel engines. Therefore, for alcohol to operate without modification in a compression ignition (CI) engine's fuel system, it must be blended with diesel fuel. Short-chain alcohols, such as methanol and ethanol, can be blended with diesel fuel. However, longer-chain alcohols, such as butanol, are more suitable due to their higher energy content, greater stability in the mixture, and lower corrosiveness [17]. Diesel fuel additives can significantly reduce exhaust emissions by reducing pollutants from fuel combustion, particularly particulate matter. Additionally, some additives improve combustion, thereby reducing emissions of suspended particles that can harm both health and the environment. Moreover, reducing harmful pollutants and health hazards in the atmosphere, such as carbon dioxide, carbon monoxide, hydrocarbons, and NO_x emissions, and more efficient combustion can lead to lower temperatures inside the engine, which reduces the formation of nitrogen oxides and other harmful gases. However, it should be noted that the effectiveness of improvers depends on the quality of the product and its use. It is always advisable to use additives approved by the vehicle manufacturer and to follow the recommendations in the vehicle manual regarding appropriate fuel types and additives [8, 9].

Lower loads mean lower temperatures and incomplete combustion. This, together with the reduced volatility of rapeseed oil with a 20% n-hexane additive, results in inferior fuel evaporation. The higher viscosity relative to diesel fuel also adversely affects atomization. As a result, the combustion process is less efficient, and CO concentrations are higher. This effect is offset by the oxygen content in the rapeseed oil-n-hexane mixture with a 20% n-hexane addition, whereas, unlike diesel fuel, in diesel, the mixture becomes richer with increasing load and combustion tempera-

ture. A decrease in oxygen content leads to an increase in CO concentration. The NO₂ concentration is higher than that of diesel fuel due to the oxidation of NO to NO₂ caused by the oxygen content of rapeseed oil [18].

RSM is a powerful statistical method for designing experiments and optimizing the effects of process variables. It has been widely used to improve and optimize processes and products across various fields, providing a structured approach to solving critical issues. Therefore, RSM is a valuable tool in industrial and scientific research settings, helping researchers determine relationships between input and response variables by systematically varying the input variables [6, 11].

Veza et al. [38] concluded that RSM has many advantages for designing, modeling, estimating, and optimizing biofuels for internal combustion engines with satisfactory accuracy. With its predictive and optimization capabilities, the response surface methodology has the potential to help advance the optimization of biofuel-powered internal combustion engines for the transition to sustainable energy. Despite RSM's potential to predict and optimize engine performance and diesel engine exhaust emissions, comprehensive research on RSM for fuel improvers, particularly for internal combustion engines, is challenging to find. Accurately predicting exhaust gas emissions from internal combustion engines remains difficult due to the complex nature of engine operation and the various factors that influence emissions. Existing models are unable to comprehensively predict emissions across diverse operating conditions. Many current models are limited by assumptions that restrict their applicability to specific types of engines or fuels, and they often fail to account for all the dynamic interactions within an engine system [13, 41, 43]. One primary limitation in modeling exhaust emissions is the difficulty of capturing the nonlinear interactions among various engine parameters, such as air-fuel ratios, ignition timing, and post-combustion treatments. Traditional models often rely on simplified linear correlations that may not accurately reflect these intricate relationships [15].

As a review, there is a growing interest in developing models that combine empirical data and advanced statistical methods to improve prediction accuracy and adapt to varying engine conditions [25]. While significant progress has been made in modeling engine exhaust emissions, gaps persist in model accuracy and applicability across different scenarios [19]. Bridging these gaps requires more sophisticated approaches that embrace the complexity of engine dynamics and leverage advancements in data analytics and computational power [33]. Very few studies have been conducted on the combustion and emissions characteristics of an engine using Diesel Smoke Stop (DSS) fuel improver. Therefore, this study uses a DSS mixed with diesel fuel to conduct performance and emissions tests on a Perkins AD4.248 four-cylinder, four-stroke diesel engine. Furthermore, no experiments in the literature determine the optimal concentration level of the improver-diesel mixture, and no models predict exhaust gas emissions. Therefore, an RSM was used in this study to optimize and parameterize the emission parameters of a diesel engine operating on a DSS-diesel mixture.

2. Materials and methods

2.1. Experimental site and machines

The experiment was conducted in 2022, 2023, and 2024 in the open fields of Mosul University in Mosul District – Nineveh Governorate, Iraq. The site coordinates are (E43,07'59.88 – N36.23'08.26) above sea level. The experimental site was abandoned land that had not been cultivated for 8 years. The area of the experimental field was two hectares. The field's topography was flat, and the soil had a clay texture (Clay, Silt, and Sand proportions were 48.45, 26, and 25.55, respectively).

The experimental work was conducted on a four-cylinder, four-stroke engine. Table 1 shows the test engine specifications. The vibration source for the subsoiler plow is a single-armed vibrating wing plow manufactured in Turkey by UZEL Company. The wings' vibration is directly in contact with the tractor's power take-off shaft. It has the following specifications: (1) Maximum working depth of 60 cm, (2) Vibrating body width of 35.8 cm, (3) Plow weight of 260 kg, (4) PTO of 540 rpm, and (5) Required traction power of 60 hp.

Table 1. Engine specification

Parameters	Specification
Model	Perkins AD4.248 diesel
Description	4-cylinder four-stroke liquid-cooled
Rated speed	2500 rpm
Compression ratio	16:1
Rated power	44.7 kW
Bore × stroke	101 × 127 mm
Displacement	4.1 L
Electrical system voltage	12 V
Lubrication system capacity	7.6 L
Coolant system capacity	10.4 L

2.2. Fuel test and experimental setup

The diesel fuel was prepared by blending diesel with Diesel Smoke Stop DSS at concentrations of 3–6 ml/l. Table 2 lists the properties of DSS, Table 3 lists the properties of the neat diesel and fuel diesel after blending, and Table 4 lists the experimental factors.

Table 2. Characteristics of the DSS enhancer

Parameters	Qualifications
Mixture classification	According to Regulation (EC) (CLP) 1272/2008 European Commission
Ingredients of the improver	Isoalkanes, Cyclics, Aromatics n-alkanes C10-C13, Ethylhexanoic acid, Iron salt, Ethylhexanoic acid
Hazard and toxicity of the mixture	It may be fatal if swallowed and enters the airways, and does not contain any bioaccumulative or toxic substances

The basic chemical mechanism is based on catalysis and cleaning during combustion: Catalysis: Contains organometallic compounds (e.g., barium or iron additives) which act as combustion catalysts. As the fuel burns, these metallic compounds lower the activation energy required for the oxidation of carbon molecules. This means that soot burns more rapidly and completely, even at lower temperatures, and there is less visible black smoke. Cleaning: Ashless

polymeric amino detergents to dissolve and prevent carbon build-up in fuel injectors. Clean injectors produce a perfect high-precision fuel spray and do not create oxygen-deficient pockets that cause thick smoke.

Table 3. Properties of diesel fuel before and after DSS addition

Properties	Neat diesel	Diesel with DSS (3 ml×l ⁻¹)	Diesel with DSS (6 ml×l ⁻¹)
Cetane number	47	48	48
Density	0.8270	0.8250	0.8241
Flash point	76	74	73
Viscosity	2.3	2.0	1.8
Color	1.5	1.0	0.5
Sulfur content	0.863	0.854	0.835

Table 4. Factors and ranges used in the experimental design

Parameters	Values
FACR, ml×l ⁻¹	3 to 6
MBV, ms ⁻²	100 to 300
ES, rpm	1500 to 2000
EOT, hr	1 to 3
Total number of tests	80

The experiments involved blending DSS and diesel to form three blends and testing their performance and emissions. They were performed at two engine speeds, ranging from 1500 to 2000 rpm, with machine body vibration between 100 and 300 ms⁻² under the load of a subsoiler plow for 1 to 3 hours of engine operation. The results were averaged across five repeated experiments. Exhaust gas emissions were measured using a Dräger X-am 5000 three-gas analyzer that measured CO, NO, and H₂S. Technical Data of Dräger X-am 5000, operation time > 12 hr, the dimensions (W × H × D) 48 × 130 × 44 mm, weight about 250 mg, battery type of NiMH, and IP 67 degree of protection. The Dräger X-am 5000 is shown in Fig. 1.



Fig 1. Dräger X-am 5000 exhaust gas emissions measuring device

2.3. Developing RSM modeling

The study aims to build multiple models to predict CO, NO, and H₂S emissions using FACR, MBV, ES, and EOT. The factors have different levels that can be divided, as described in Table 4. The steps of developing and building multiple models are as follows:

1. RSM's first goal is to find the relationship between the independent variables and the response variable; in this study, Design-Expert Version 12 software (Stat-Ease

Inc., Minneapolis, Minnesota, USA) was used to analyze the Data. Figure 2 shows a schematic diagram of the RSM model development process.

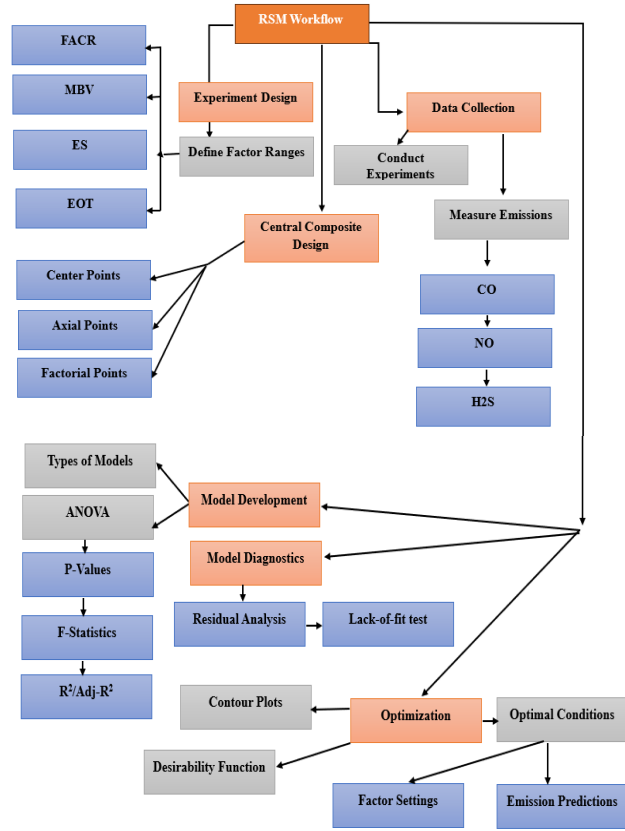


Fig. 2. Schematic diagram of the RSM model development process

2. The most appropriate polynomial mathematical models were chosen to predict the interactive effects of FACR, MBV, ES, and EOT parameters on CO, NO, and H₂S emissions.
3. Equation (1)–(6) is used to compute the R², the adjusted correlation coefficient (Adj. R²), predicted correlation coefficient (Pred. R²) and Adeq. precision.

$$R^2 = 1 - \left[\frac{SS_{\text{residual}}}{SS_{\text{residual}} + SS_{\text{model}}} \right] \quad (1)$$

$$\text{Adj. } R^2 = 1 - \left[\frac{\frac{SS_{\text{residual}}}{df_{\text{residual}}}}{\frac{SS_{\text{residual}} + SS_{\text{model}}}{df_{\text{residual}} + df_{\text{model}}}} \right] \quad (2)$$

$$\text{Pred. } R^2 = 1 - \left[\frac{\text{PRESS}}{SS_{\text{residual}} + SS_{\text{model}}} \right] \quad (3)$$

$$\text{PRESS} = \sum_{i=1}^N (e_i - 1)^2 \quad (4)$$

$$e_i - 1 = \frac{e_i}{1 - h_{ii}} \quad (5)$$

$$\text{Adeq Precision} = \frac{(\max(\text{predicted values}) - \min(\text{predicted values}))}{\sqrt{\text{average variance of the predicted values}}} \quad (6)$$

4. An analysis of variance table was used to evaluate the fitness of each developed model. The fit statistics, presented in Table 5, allowed the software to suggest the best models.

Table 5. Models proposed by the program to predict CO, NO, and H₂S emissions

Source	CO emission				
	p-value	Fit p-value	R ² predicted	R ² -adjusted	Model status
Linear model	< 0.0001	< 0.0001	0.5812	0.5320	Suggested
2FI model	0.0009	< 0.0001	0.7695	0.7235	
Quadratic model	0.9128	< 0.0001	0.6543	0.1311	
Cubic model	< 0.0001	< 0.0001	0.9138	−5.5535	Aliased
Source	NO emission				
	p-value	Fit p-value	R ² predicted	R ² -adjusted	Model status
Linear model	< 0.0001	< 0.0001	0.5872	0.5389	Suggested
2FI model	0.0004	< 0.0001	0.7834	0.7361	
Quadratic model	0.5261	< 0.0001	0.6798	0.2132	
Cubic model	< 0.0001	0.0024	0.334	0.135	Aliased
Source	H ₂ S emission				
	p-value	Fit p-value	R ² predicted	R ² -adjusted	Model status
Linear model	< 0.0001	< 0.0001	0.5489	0.5021	Suggested
2FI model	0.0028	< 0.0001	0.7857	0.7187	
Quadratic model	0.6605	< 0.0001	0.5760	0.1316	
Cubic model	< 0.0001	0.0059	0.7862	−1.5213	Aliased

Table 6. ANOVA for CO, NO, and H₂S emissions models

Source	df	CO		NO		H ₂ S		Sig.
		F-value	P-value	F-value	P-value	F-value	P-value	
Model	10	17.00	< 0.0001	18.06	< 0.0001	12.17	< 0.0001	Sig.
A-FACR	1	98.09	< 0.0001	111.44	< 0.0001	88.79	< 0.0001	
B-MBV	1	32.33	< 0.0001	27.76	< 0.0001	13.95	0.0004	
C-ES	1	13.52	0.0005	12.38	0.0008	4.53	0.0369	
D-EOT	1	0.0574	0.8114	0.1675	0.6836	1.77	0.1876	
AB	1	4.88	0.0305	9.50	0.0030	5.85	0.0182	
AC	1	1.49	0.2265	5.77	0.0190	0.2830	0.5965	
AD	1	1.46	0.2315	1.68	0.1999	1.59	0.2123	
BC	1	12.28	0.0008	7.06	0.0098	1.20	0.2781	
BD	1	2.76	0.1013	2.83	0.0969	3.74	0.0574	
CD	1	3.18	0.0791	1.97	0.1645	0.0186	0.8920	
Residual	69							
Lack of Fit	14	50.02	< 0.0001	24.23	< 0.0001	7.18	< 0.0001	Sig.
Pure Error	55							
Cor Total	79							

3. Results and discussion

3.1. Analysis of variance (ANOVA) for CO, NO, and H₂S emissions models

Table 6 shows the ANOVA models for the CO, NO, and H₂S emissions by FACR, MBV, and ES. The CO, NO, and H₂S models are significant, with recorded F-values of 17.00, 18.06, and 12.17, respectively. There is only a 0.01% chance that an F-value of this size caused an effect. P-values for the CO, NO, and H₂S emissions model of less than 0.0500 indicate that most terms of the model are significant. In the CO model, cases A, B, C, AB, and BC are significant model terms. The A, B, C, AB, AC, and BC are significant model terms for NO emission. While the A, B, C, and AB are significant model terms for H₂S emission. The model terms are insignificant when p-values are more than 0.1. Values greater than 0.1 indicate the model terms are not necessary. If many insignificant model terms are not counted, they are not required to support the hierarchy. In improved models, excluding these non-significant terms improves performance [22]. In the CO, NO, and H₂S models, the Lack of Fit F-values of 50.02, 24.23, and 7.18 indicate significant Lack of fit relative to the pure error, respec-

tively. There is only a 0.01% chance that a Lack of Fit F-value this large could occur by chance.

3.2. Model of CO emissions

Accurate prediction of CO emissions is crucial for understanding their real-time dynamics and for formulating emission-reduction policies. The model determines whether the recommended model is compatible with the test results. In the CO emissions model (Table 7), the R² predicted of 0.7235 is in reasonable agreement with the R² Adjusted of 0.7695, with the difference being less than 0.2 between R² predicted and R² Adjusted. At the same time, Adeq Precision measures the signal-to-effect ratio, which should be greater than four and is desirable [20]. It achieved about 18.233 on the 2FI model (Table 7). The model in actual factor terms can be applied to predict the response for given levels of each factor. The value, in the form of adjusted and predicted values with a difference of 0.2, ensures the research follows the model and can be used to navigate the design space described. The equation in terms of actual factors can be used to predict responses for given levels of each factor, indicating that the model is highly significant and that the responses are accurate.

Table 7. 2FI model of CO emissions

Model of CO emissions	Std. Dev.	C.V. %	R ²	Adjusted R ²	Predicted R ²	Adeq Precision
	1.86	7.25	0.7913	0.7695	0.7235	18.233
$\text{CO} = 16.84852 - 0.963252\text{FACR} - 0.195023\text{MBV} - 0.013086\text{ES} \\ + 0.012620\text{FACR} * \text{MBV} + 0.002789\text{FACR} * \text{ES} \\ - 0.689815\text{FACR} * \text{EOT} + 0.000120\text{MBV} * \text{ES} \\ - 0.014236\text{MBV} * \text{EOT} - 0.006111\text{ES} * \text{EOT}$						

3.3. Effect of parameters on CO emissions

The interaction effect between FACR and the MBV significantly influences the engine CO emissions, as illustrated in the response plot in Fig. 3a. The figure shows that CO emissions increase as the FACR increases. Increasing the concentration of a fuel additive may lead to increased carbon monoxide (CO) emissions due to incomplete fuel combustion. It occurs when there is insufficient oxygen in the combustion process, leading to the formation of CO rather than carbon dioxide (CO₂). Little air, incomplete combustion, and an unbalanced mixer ratio are all reasons for CO emission from the engine exhaust [21]. The widespread oxygenation and flammability properties of the Diisopropyl ether blend can reduce CO emissions [36]. In addition, Figure 3a shows that an increase in MBV due to increased load does not improve CO conversion to CO₂. The engine experiences vibrations due to overload. The fuel may not be fully combusted in the engine cylinders, leading to increased carbon monoxide formation, a gas produced by the incomplete combustion of carbon [37].

The concentration rate of the fuel additive is one of the significant influencing parameters in CO emissions. From the RSM interaction plots in Fig. 3b, the CO tends to increase as the FACR and ES increase. CO emissions in the exhaust vary with ES, but the difference is close to that with an increase in ES from 1500 to 2000 rpm. As engine speed increases, fuel consumption increases; thus, the amount of exhaust gases, including carbon monoxide, increases. Sudrajat et al. [32] found that CO exhaust emissions increase with increasing engine speed. CO is produced by burning fuel in an excess of oxygen. The ratio of carbon that cannot bind to oxygen increases fuel demand, thereby reducing thermal efficiency and increasing CO emissions. The results of Sekar et al. [27] showed that pure diesel fuel has the highest CO emissions, and a 15% blend of Diethylene Glycol and Dimethyl ether additives has the lowest.

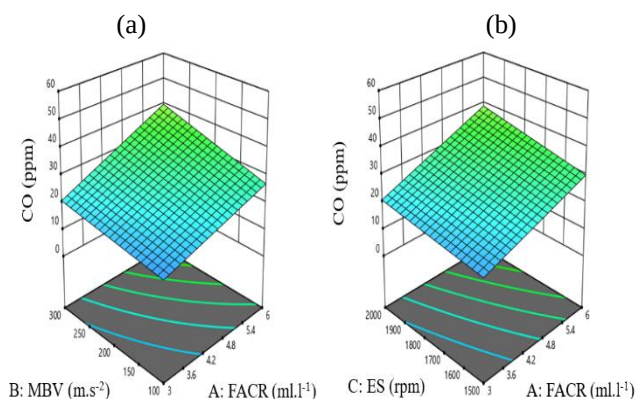


Fig. 3. Variation of CO emissions against FACR, MBV, and ES: a) the effect of interactions between MBV and FACR on CO emissions; b) the effect of interactions between ES and FACR on CO emissions

The CO emissions in the exhaust gas indicate incomplete combustion, which causes air pollution and reduces thermal efficiency. Figure 4a shows that CO emissions result from the interaction between EOT and FACR. The figure shows that CO emissions increase as EOT and FACR increase. CO emissions in the exhaust vary with EOT, but the difference is close to that with an increase in EOT from 1 to 3 hr. CO emissions are relatively low but increase by about 50% with increased engine operation time [39]. The increased CO emissions were confirmed during the test due to the greater chance of incomplete combustion. The lowest CO emissions (14 ppm) were achieved when the engine was operated at 1 hour and 3 ml×l⁻¹ of FACR was added to the fuel. Measuring engine exhaust CO emissions can indicate engine performance. High CO levels indicate incomplete fuel combustion, resulting in lost power, increased fuel consumption and reduced engine efficiency [24]. Figure 4b presents a contour plot of the model's interaction between ES and MBV, showing CO emissions concentrations for different changes in ES and MBV. The figure shows that CO emissions tend to increase as ES and MBV increase in all noted speed and vibration conditions. The lowest CO emissions were achieved at an interaction between the engine's speed of 1500 rpm and a vibration rate of 100 ms⁻².

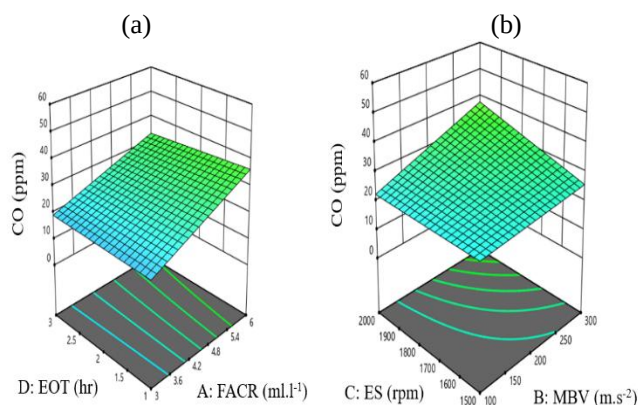


Fig.4. Variation of CO emissions against EOT, FACR, ES, and MBV: a) the effect of interactions between EOT and FACR on CO emissions; b) the effect of interactions between ES and MBV on CO emissions

Figure 5a depicts the CO emissions in the exhaust gas output from the exchanges between EOT and MBV. The curve shows CO emissions increasing as the EOT and MBV increase. The emission increase due to increased MBV is significantly greater than that due to EOT, because EOT causes a higher temperature rise that improves the conversion of CO to CO₂. Thiagarajan et al. [35] explained that CO is converted to CO₂ in the presence of oxygen and at high temperatures, where oxygen oxidizes CO. Furthermore, the rate of chemical reactions increases with increasing temperature; higher temperatures generated by complete combustion resulted in a higher conversion rate of CO to CO₂. The interaction effect between EOT and ES significantly influences the engine's CO emissions, as illustrated in the response plot in Fig. 5b. CO emissions increase as the EOT and ES increase; the increase due to increased ES is significantly greater than the increase due to EOT. The

lowest CO emissions were achieved at an engine speed of 1500 rpm and an engine operating time of 1 hr.

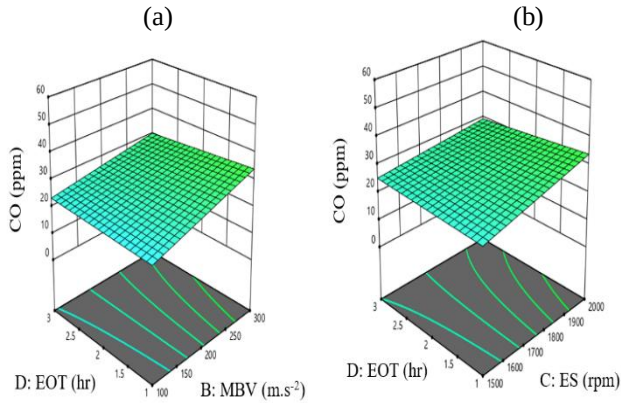


Fig. 5. Variation of CO emissions against EOT, MBV, and ES: a) the effect of interactions between EOT and MBV on CO emissions; b) the impact of interactions between EOT and ES on CO emissions

3.4. Model of NO emissions

In the NO emissions in exhaust gases model, the R^2 -prediction was 0.7834, which reasonably agrees with the R^2 -Adjusted of 0.7361. The difference is less than 0.2; at the same time, Adeq precision measures the signal-to-effect ratio. A ratio greater than 4 is desirable, which means the model can be used to navigate the design space. A model ratio of 9.33 indicates an adequate signal (Table 8). The model ($R^2 = 0.7935$) in actual-factor terms can be used to predict the response for given levels of each factor. Predicting nitrogen oxides (NO_x) emissions from engine exhaust using conventional mathematical and physical methods is challenging due to the complex, variable operating conditions of engines. The accuracy of emission predictions related to the model optimization [23, 28]. Zhang et al. [42] developed a model to predict nitrous oxide emissions, and the results showed moderate predictive ability for NO_x emissions ($R^2 = 0.779878$). However, compared to previous models, it has better generalization and higher R^2 values. RSM is used to predict and optimize the NO_x emissions of a diesel engine under three experimental conditions. The high R-squared values, close to unity, indicate high model accuracy [9].

Table 8. 2FI model of NO emissions

Model of NO emissions	Std. Dev.	C.V. %	R^2	Adjusted R^2	Predicted R^2	Adeq Precision
	2.08	9.33	0.7935	0.7834	0.7361	19.1802
$NO = 32.09164 - 5.54658 \text{ FACR} - 0.150099 \text{ MBV} - 0.019945 \text{ ES} \\ + 12.69598 \text{ EOT} + 0.015606 \text{ FACR} * \text{MBV} \\ + 0.004867 \text{ FACR} * \text{ES} - 0.655336 \text{ FACR} * \text{EOT} \\ + 0.000081 \text{ MBV} * \text{ES} - 0.012780 \text{ MBV} * \text{EOT} \\ - 0.004268 \text{ ES} * \text{EOT}$						

3.5. Effect of parameters on NO emissions

The interaction effect between MBV and FACR significantly influences the engine's NO emissions, as illustrated in the response plot in Fig. 6a. The figure shows that NO emissions increase as the MBV and FACR increase. However, the increase in the FACR effect is greater than that due to MBV. The lowest NO emission was recorded with 3 ml×l⁻¹, due to the latent heat of the 3 ml×l⁻¹ additive percentage, which is higher than that of diesel with a high

additive rate. Thus, using the 3 ml×l⁻¹ amount of additives decreased the peak temperature in the cylinder. Therefore, NO_x emissions are reduced with the use of a fuel additive. Elkelawy et al. [8] explained that fuel additives significantly reduce emissions in various engine types. Some engine additives can increase nitrogen oxide emissions if not used properly. Furthermore, diesel blends with appropriate additives have been shown to reduce nitrogen oxide emissions. In general, the application of fuel additives can significantly reduce harmful emissions in various engine types if used in the correct proportions. In addition, Figure 6a shows that increased MBV due to increased load will not improve NO emissions. The engine experiences vibrations due to overload. The variation in NO_x emissions with engine load across all test fuels showed that NO_x emissions increased with increasing engine load. High temperatures are the main factor in NO_x emissions. The nitrogen molecules react with oxygen, producing NO_x after the required reaction time is reached [30].

The RSM interaction plots in Fig. 6b show that NO tends to increase with increasing ES and FACR. NO emissions in the exhaust vary with ES, but the difference is close to that with an increase in ES from 1500 to 2000 rpm. As engine speed increases, the amount of NO_x emitted increases. The combustion temperature inside the engine increases with speed, leading to increased NO_x formation [34]. When the engine runs at higher speed, the number of combustion cycles per unit time increases, resulting in higher combustion temperatures and increased NO_x emissions [7].

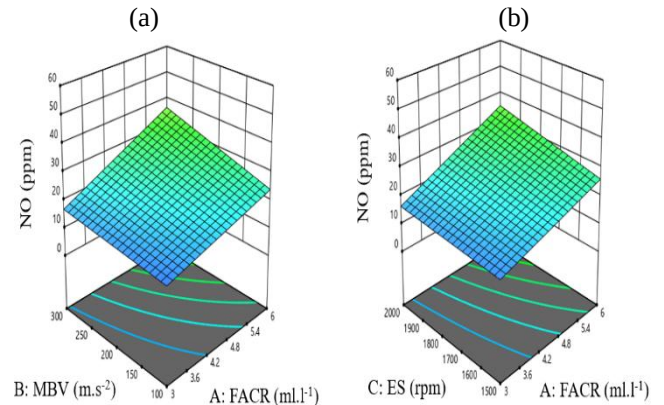


Fig. 6. Variation of NO emissions against MBV, ES, and FACR: a) the effect of interactions between MBV and FACR on NO emissions; b) the effect of interactions between ES and FACR on NO emissions

NO_x is an air pollutant that contributes to smog and acid rain and harms human health. When fuel does not burn completely, it produces undesirable compounds such as NO_x. The main reasons for incomplete combustion include insufficient oxygen in the fuel mixture, suboptimal combustion temperatures, and inefficient engine design. Practical steps to improve combustion efficiency and reduce emissions include adjusting the fuel mixture and adding fuel additives [40]. Figure 7a shows that NO emissions result from the interaction between EOT and FACR. The figure shows that NO emissions increase as EOT and FACR increase. NO emissions in the exhaust vary with EOT, but the

difference is close to that with an increase in EOT from 1 to 3 hr. The lowest NO emissions (12 ppm) were achieved when the engine was operated for 1 hour, and 3 ml $\times$ l $^{-1}$ of FACR was added to the fuel. NO $_x$ is formed when nitrogen and oxygen in the air react at very high temperatures, conditions common in engine combustion chambers. The longer an engine runs, the greater the chance of these reactions occurring and the more NO $_x$ will be formed [2]. NO makes up the majority of the total NO $_x$ produced in internal combustion engines, while NO $_2$ makes up the remaining percentage [26].

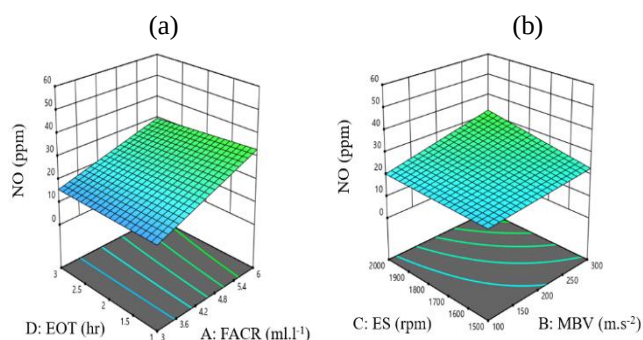


Fig. 7. Variation of NO emissions against EOT, FACR, ES, and MBV: a) the effect of interactions between EOT and FACR on NO emissions; b) the effect of interactions between ES and MBV on NO emissions

Figure 7b presents a contour plot of the model's interaction between ES and MBV, showing NO emissions concentrations for different changes in ES and MBV. The figure shows NO increase in emissions as ES and MBV increase across all noted speed and vibration conditions. The lowest NO emissions were achieved at an interaction between the engine's speed of 1500 rpm and a vibration rate of 100 ms $^{-2}$.

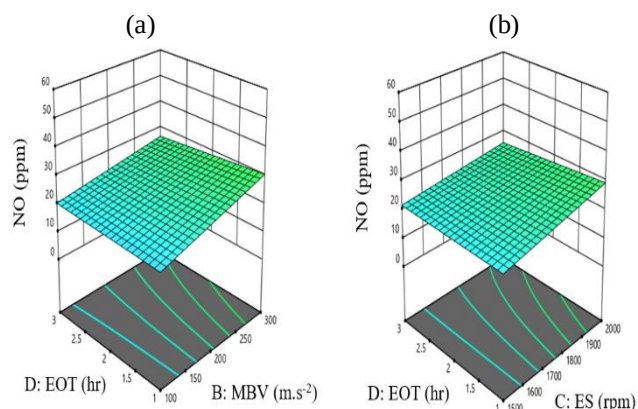


Fig. 8. Variation of NO emissions against EOT, MBV, and ES: a) the effect of interactions between EOT and MBV on NO emissions; b) the effect of interactions between EOT and ES on NO emissions

Figure 8a depicts the NO emissions in the exhaust gas output from the interactions between EOT and MBV. The response curve shows NO emissions increasing as the EOT and MBV increase. The emission increase due to increased MBV is significantly greater than that due to EOT because EOT causes a higher temperature rise, which improves NO conversion. Janulevičius et al. [16] tested the ten Massey Ferguson 6499 tractors studied. They showed that most

tractors (approximately 76%) emitted nitrogen oxides during engine loads exceeding 50% of maximum torque, 21.6% between 30 and 50% of maximum torque, and only 2.3% at loads up to 30% of maximum torque. The highest nitrogen oxide emissions during actual operating cycles were recorded at speeds between 700 and 1100 rpm. The interaction effect between EOT and ES significantly influences the engine's NO emissions, as illustrated in the response plot in Fig. 8b. The figure shows that NO emissions increase with both EOT and ES; the increase due to ES is significantly greater than that due to EOT. The lowest NO emissions were achieved at an engine speed of 1500 rpm and an engine operation time of 1 hr.

3.6. Model of H₂S emissions

The accuracy of responses predicted by the software's experimental model was analyzed using variance analysis. The C.V. and R² values were used to determine the best-fitting model. The best-fitting model had a lower variation coefficient and a higher determination coefficient. Its significance, or lack thereof, determined the adequacy of the model. Meanwhile, the model's lack of fit must be statistically insignificant ($p > 0.05$) to fit the data well [5]. The R² value for the H₂S emissions model was 0.7857. The R²-prediction reasonably agrees with the R²-adjusted, with the difference of less than 0.2; at the same time, Adeq precision measures the signal-to-effect ratio. A ratio greater than 4 is desirable, indicating the model can effectively navigate the design space. A model ratio of 16.0715 indicates an adequate signal, as shown in Table 9.

The developed model can be used to navigate the design space in the H₂S emissions model. The predicted data in all models agreed with the experimental data, with R² values in these three designs close to 0.8. As stated by [29], the R² value should be at least 0.8 for the model to be adequate. In addition, C.V. is the standard error ratio used to predict the mean value of the actual response. In other words, C.V. could represent the model's repeatability. The model's C.V. value was 14.57. Lower values of the coefficient of variation indicated better precision and reliability of the experiment [19]. The above statistical analysis found that the 2FI model performed well in prediction. Furthermore, using the optimal model, the R² value for the predicted response was higher than that of most other output models. Data generated from the experimental model may have lower error than the raw data from the experiment.

Table 9. 2FI model of H₂S emissions

Model of H ₂ S emissions	Std. Dev.	C.V. %	R ²	Adjusted R ²	Predicted R ²	Adeq Precision
	1.02	14.57	0.7995	0.7857	0.7187	16.0715
$\begin{aligned} \text{H}_2\text{S} = & -3.43126 + 1.27701\text{FACR} - 0.015013 \text{ MBV} + 0.001235\text{ES} \\ & + 1.44304\text{EOT} + 0.003142\text{FACR} * \text{MBV} \\ & - 0.000276 \text{ FACR} * \text{ES} - 0.163542 \text{ FACR} * \text{EOT} \\ & + 8.52083\text{E} - 06 \text{ MBV} * \text{ES} - 0.003766 \text{ MBV} \\ & * \text{EOT} - 0.000106\text{ES} * \text{EOT} \end{aligned}$						

3.7. Effect of parameters on H₂S emissions

Diesel fuel contains some sulfur. During fuel combustion, sulfur reacts with oxygen to form sulfur dioxide (SO₂). Under combustion conditions, some of the SO₂ can be converted to hydrogen sulfide (H₂S) [5]. H₂S is hazardous to human health, with concentrations of 100 ppm and above

posing a risk if exposure continues for 1 to 4 hours [31]. The interaction effect between MBV and FACR significantly influences the engine's H₂S emissions, as illustrated in the response plot in Fig. 9a. The figure shows that H₂S emissions increase as the MBV and FACR increase. However, the increase in the FACR effect is greater than that due to MBV. The lowest H₂S emission was recorded at 3 ml×l⁻¹, due to the latent heat of the 3 ml×l⁻¹ additive concentration. Thus, using the 3 ml×l⁻¹ amount of additives decreased the peak temperature in the cylinder. Fuel additives may include compounds that reduce the sulfur content of diesel fuel, thereby reducing hydrogen sulfide emissions. Improving the combustion process also reduces the production of harmful gases, including H₂S [38].

In addition, Fig. 9a shows that increased MBV due to increased load will not improve H₂S emissions. The engine experiences vibrations due to overload. Fayad et al. [12] showed that H₂S concentrations in the exhaust are fundamentally determined by the amount of fuel injected into the combustion chamber. As more fuel is injected in these cases, H₂S concentrations increase with increasing engine load or speed. Hydrogen reacts rapidly with sulfur, leading to high concentrations of H₂S. In the case of constant loads and variable engine speeds, H₂S decreased by 43.56% with additives added to the fuel compared to diesel. However, the H₂S concentrations did not reach hazardous levels. The RSM interaction plots in Figure 9b show that H₂S increases with increasing ES and FACR. H₂S emissions in the exhaust vary with ES, but the difference is close to that with an increase in ES from 1500 to 2000 rpm. As engine speed increases, the amount of H₂S emitted increases.

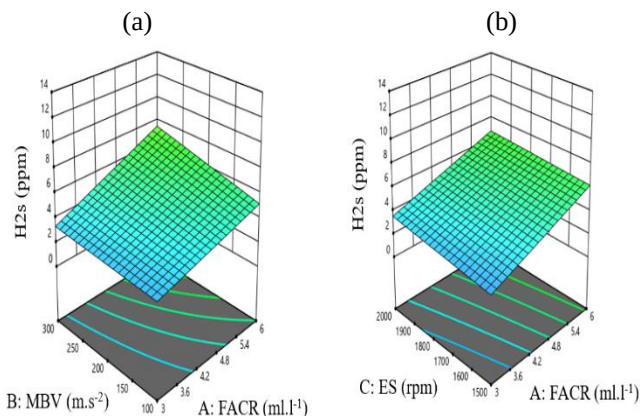


Fig. 9. Variation of H₂S emissions against FACR, MBV, and ES: a) the effect of interactions between MBV and FACR on H₂S emissions b) the effect of interactions between ES and FACR on H₂S emissions

Figure 10a shows that H₂S emissions are due to the interaction between EOT and FACR. The figure shows that H₂S emissions increase as EOT and FACR increase. H₂S emissions in the exhaust vary with EOT, but the difference is close to that with an increase in EOT from 1 to 3 hr. The increased H₂S emissions were confirmed during the test due to the greater chance of incomplete combustion. The lowest H₂S emissions were achieved when the engine was operated at 1 hour, and 3 ml×l⁻¹ of FACR was added to the fuel. Measuring engine exhaust H₂S emissions can indicate engine performance. Although the H₂S emission rate was low,

its accumulation could have a profound impact on the environment and human health. Figure 10b presents a contour plot of the model interaction between ES and MBV, showing H₂S emission concentrations for different changes in ES and MBV. The figure shows H₂S emissions increase as ES and MBV increase in all noted speed and vibration conditions. The lowest H₂S emissions were observed at an interaction between an engine speed of 1500 rpm and a vibration rate of 100 ms⁻².

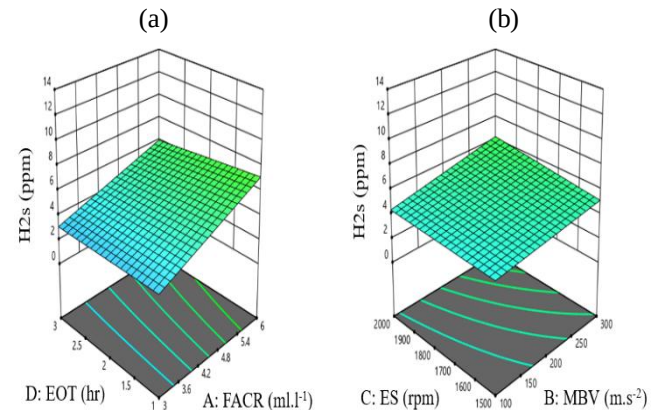


Fig. 10. Variation of H₂S emissions against EOT, FACR, ES, and MBV: a) the effect of interactions between EOT and FACR on H₂S emissions; b) the effect of interactions between ES and MBV on H₂S emissions

Figure 11a depicts the H₂S emissions in the exhaust gas output from the interactions between EOT and MBV. The response curve shows that H₂S emissions increase as EOT and MBV increase. The increase in emissions due to increased MBV is significantly greater than that due to EOT.

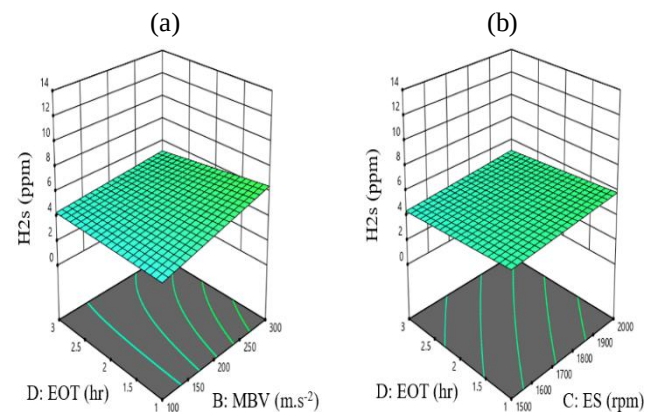


Fig. 11. Variation of H₂S emissions against EOT, MBV, and ES: a) the effect of interactions between EOT and MBV on H₂S emissions; b) the effect of interactions between EOT and ES on H₂S emissions

The interaction effect between EOT and ES significantly influences the engine's H₂S emissions, as illustrated in the response plot in Fig. 11b. The figure shows that H₂S emissions increase as EOT and ES increase; the increase due to ES is significantly greater than that due to EOT. For the engine exhausts of the fuels tested under both variable-load and variable-speed conditions, Fayad et al. [12] reported that increasing engine speed from 1250 to 3000 rpm increased H₂S emissions from 8 to 23 ppm. The results

show that the H₂S concentrations in the exhaust mainly depend on the amount of fuel injected into the combustion chamber, load, and variable speed. The lowest H₂S emissions (2 ppm) were achieved at an engine speed of 1500 rpm and an engine operating time of 1 hr.

4. Conclusion

The RSM models were established to predict CO, NO, and H₂S when Diesel Smoke Stop powers an engine. The RSM results provided insights into the effects on exhaust gas and emission characteristics. Experimental data were used to develop the RSM model and study the effect of the parameters. The research results and factor impact analysis led to the following conclusions:

1. The results showed that adding a 3 ml DSS fuel improver per liter of diesel fuel recorded the lowest overall CO, NO, and H₂S emissions.
2. The DSS additive concentration of 3 ml×l⁻¹ overlapped with an engine speed of 1500 rpm, an engine operation time of 1 hour, and a vibration of 100 ms⁻², which recorded the lowest exhaust gas emissions.

3. CO, NO, and H₂S emissions increased as the Machine body vibration increased in all other parameters.
4. CO, NO, and H₂S emissions increased with increased engine speed and engine operation time.
5. CO, NO, and H₂S emissions increased with increasing fuel additive concentration rate (Diesel Smoke Stop), which was higher than for all other parameters.
6. It can be deduced that RSM models have many advantages in predicting and investigating emission characteristics. RSM, among other things, provides an acceptable level of performance in predicting each exhaust emission.

Intriguing research questions for future studies will focus on machine learning algorithms and computational fluid dynamics modeling to improve emissions prediction and better understand in-cylinder phenomena, along with broader practical and engineering implications for industrial and marine engine applications.

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Nomenclature

CO	carbon monoxide
DSS	diesel Smoke Stop enhancement
EOT	engine operation time
ES	engine speed
FACR	fuel additive concentration rate

H ₂ S	hydrogen sulfide
MBV	machine body vibration
NO	nitrogen monoxide
RSM	response surface methodology

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Discrepancies between inspection results and actual technical condition of passenger cars: a Polish case study

ARTICLE INFO

Ensuring the technical reliability of vehicles through periodic inspections is one of the key elements of road safety policy. However, discrepancies between inspection results and the actual condition of vehicles raise concerns about the effectiveness of current diagnostic practices. This study evaluates the technical condition of passenger cars operating in western Poland, based on data from over one thousand periodic inspections conducted between 2024 and 2025. Statistical analyses were performed to identify relationships between vehicle age, mileage, and defects such as suspension looseness, corrosion, oil leakage, and lighting adjustment. In addition, a detailed technical assessment was conducted on a Volkswagen Golf V 1.4 TSI that had a valid inspection certificate. Despite compliance with formal requirements, the vehicle exhibited severe mechanical faults, including worn timing components and incorrect chassis geometry. The results demonstrate that vehicles often pass inspections despite having defects that can compromise road safety. The findings underline the need to improve diagnostic accuracy, enhance inspector training, and ensure the use of standardized repair procedures. Strengthening inspection quality control would contribute to increased traffic safety and a more reliable assessment of vehicle technical condition.

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1. Introduction

Ensuring that vehicles remain in proper technical condition is one of the fundamental pillars of road safety and environmental protection. The purpose of periodic vehicle inspections is to confirm that each vehicle operating on public roads complies with legal requirements and safety standards [4]. In practice, however, the inspection process is influenced by numerous factors, including the applied diagnostic procedures, the technical condition of the measuring equipment, and – perhaps most importantly – the experience and competence of the inspector.

The reliability of a technical inspection depends not only on the diagnostic technology used but also on the human factor. Inspectors must be able to identify faults whose symptoms may be subtle or intermittent, and they must properly interpret diagnostic readings. Many defects, particularly those related to wear, improper assembly, or material fatigue, can only be identified through experience and thorough mechanical verification. The increasing complexity of vehicle systems, the use of electronic control units, and the integration of driver-assistance technologies further complicate the diagnostic process. Previous studies and reports have also emphasized that the quality and reliability of vehicle inspections depend not only on diagnostic equipment, but primarily on the proper execution of procedures and effective supervision of inspection stations [8, 20, 22, 24].

Modern inspection stations are equipped with advanced devices for measuring braking forces, suspension parameters, and exhaust emissions. Vehicles produced over the past two decades are also equipped with OBD/EOBD systems, enabling partial electronic diagnostics. Despite this, numerous studies indicate that even vehicles with a valid inspection certificate may still pose safety or environmental

risks. The inspector's role remains decisive, as human error, time constraints, or procedural shortcuts can lead to undetected defects that later contribute to road accidents or mechanical failures.

Vehicle inspection procedures across the European Union have been harmonized under Directive 2014/45/EU, which specifies the scope, frequency, and technical requirements of periodic inspections. These include verification of vehicle identification, braking and steering efficiency, lighting and visibility, suspension and axle components, and environmental aspects such as pollutant emissions and fluid leakage. However, the implementation and supervision of these inspections remain the responsibility of individual member states, which results in variations in diagnostic quality and enforcement.

Several researchers have addressed these issues. Leoniev et al. [9] analysed mathematical models of deceleration for vehicles with brake system failures, showing that small differences in measurement assumptions can lead to significant errors in determining braking efficiency. Shoin [23] discussed the role of automotive expertise in legal proceedings, emphasizing the responsibility of experts and the importance of reliable documentation in judicial and insurance contexts. Karev and Volkov [6] examined the uncertainties arising from technical evaluations used as court evidence and proposed improved analytical models for assessing braking dynamics.

Malinowski et al. [12] demonstrated that braking test results can vary significantly across inspection stations, even for the same vehicle tested under similar conditions. This variability highlights the influence of human factors and station-specific calibration procedures. Suyanto et al. [25] emphasized that professional competence, discipline, and honesty are essential soft skills determining the quality of

work in the automotive service and inspection sectors. Driving behavior, maintenance practices, and vehicle age all have a strong impact on technical condition and operational safety [15]. Studies from several European countries show that privately owned vehicles are generally better maintained than company fleets due to lower usage intensity and greater owner responsibility. Ioniță et al. [5] found that even vehicles compliant with inspection standards may pose risks when overloaded or modified. Christensen and Elvik [3] reached similar conclusions in Norway, showing that while inspections reduce the number of technically defective vehicles, they do not necessarily reduce accident rates – likely due to compensatory driver behavior or undetected faults.

In many Central and Eastern European countries, the aging vehicle fleet remains a major challenge. In Ukraine, the average age of passenger cars exceeds 22 years [17], while in Poland it is approximately 14 years, increasing annually. Older vehicles are more prone to corrosion, suspension looseness, and fluid leaks, all of which reduce their operational safety. Statistical analyses conducted in western Poland [18] have demonstrated strong correlations between vehicle age, mileage, and the presence of mechanical or structural defects, emphasizing the importance of reliable inspection procedures.

Other studies have addressed specific aspects of vehicle condition assessment, including post-accident repairs [10], exhaust emissions [13, 14, 26], wheel and tire diagnostics [2, 19], bearing wear [10], and suspension degradation [28]. Recent work has also explored diagnostic challenges related to electric and hybrid vehicles [1, 27], highlighting the need to modernize inspection systems and provide specialized training for technicians. Earlier studies have also addressed the effectiveness of vehicle inspection systems and their impact on road safety, including analyses presented in Polish technical journals [16, 26].

Despite regulatory frameworks and technological progress, significant shortcomings remain in the inspection process. The most common problems identified in the literature include procedural limitations, where standard inspection methods are insufficiently sensitive to detect hidden or intermittent faults. Human factor variability, resulting from differences in inspector expertise, diligence, and interpretation of results. Poor-quality repairs, often involving non-original or improperly installed components, which may pass inspection but reduce long-term reliability.

Therefore, there is a growing need to evaluate the actual effectiveness of technical inspections in identifying safety-critical vehicle defects. It is particularly important to determine whether vehicles with valid inspection certificates can still pose potential hazards due to undetected mechanical issues.

The present study addresses these issues by analyzing the technical condition of passenger cars operating in western Poland. The dataset includes vehicles inspected between 2024 and 2025 and covers key parameters such as vehicle age, mileage, corrosion, oil leakage, and suspension condition. Statistical analyses were conducted to identify significant relationships between these variables and to

determine how factors such as brand, mileage, and age influence inspection outcomes.

To complement the statistical analysis, a detailed case study was performed on a Volkswagen Golf V 1.4 TSI. The selected vehicle, despite holding a valid inspection certificate, underwent a comprehensive technical evaluation that revealed multiple defects undetected during the official inspection.

By combining quantitative data analysis with an in-depth technical case study, this research aims to assess the reliability of periodic inspection procedures and to verify whether a vehicle certified as roadworthy can, in fact, meet the required safety standards. The findings contribute to improving the effectiveness of inspection systems, enhancing inspector training and supporting policy initiatives to increase road safety and environmental protection.

While numerous studies have proposed mathematical and analytical models describing selected aspects of vehicle diagnostics and braking processes [6, 9], these approaches are typically developed under simplified assumptions and do not fully reflect the complexity of real inspection conditions. In particular, they do not capture human factors, variability between inspection stations, or the impact of repair quality on actual vehicle condition.

For this reason, the present study deliberately does not focus on the development of new mathematical models. Instead, it addresses a clearly identified gap in the literature by providing large-scale empirical evidence from over 1000 real inspection records, complemented by an in-depth technical case study. Such an approach enables verification of how inspection systems perform in practice, which cannot be reliably assessed using model-based methods alone.

The novelty of this study lies in the use of large-scale, real-world inspection data combined with a detailed technical case study. Unlike model-based or purely theoretical analyses, this approach enables direct verification of discrepancies between formal inspection results and the actual technical condition of vehicles. The study contributes to the field by providing practical evidence of limitations in current inspection procedures and identifying factors that influence the occurrence of undetected defects.

2. Materials and methods

All statistical calculations were conducted using the STATISTICA software package from StatSoft Inc. (Tulsa, OK, USA, 2020), version 13.3 [30]. Quantitative variables were described by arithmetic mean and standard deviation (SD), median, minimum and maximum values (range), and a 95% confidence interval (95% CI). Due to deviations from normality and the presence of non-unimodal distributions, non-parametric tests were applied in the analysis.

The study was carried out at a vehicle inspection station between September 2024 and September 2025, similarly to the survey conducted two years earlier. The statistical analysis focused primarily on identifying relationships among vehicle age, mileage, and defect occurrence, while brand-related comparisons were treated as supplementary.

2.1. Results

The analysis primarily focuses on relationships between operational parameters (age and mileage) and defect occur-

rence, while brand-based comparisons are included for complementary interpretation.

The observed distributions were not assumed to be normal; therefore, non-parametric statistical methods were applied. As in the 2023 study [18], the majority of vehicles were powered by gasoline – 592 vehicles (57.32%) – followed by diesel vehicles, numbering 383 (37.08%). The mean vehicle age was slightly higher (by approximately two months) than in the 2023 analysis, while the average mileage was greater by about 4500 km (Table 1).

Table 1. Baseline characteristics of the study population (N = 1033); mean (SD), median; Range; [95% CI $\bar{X}_{mean}$]

Age (years)	14.6 (6.3) 15 1–37 [14.2; 15.0]	Diesel (n = 383)	14.5 (5.3) 15 1–30 [14.0; 15.1]
		Fuel (n = 592)	14.7 (6.9) 15 1–37 [14.2; 15.3]
		Fuel + Gas (n = 50)	15.3 (6.2) 15.5 2–27 [13.5; 17.1]
		Hybrid (n = 7)	4.6 (1.8) 4 3–7 [2.9; 6.2]
		p-value	0.0004 ¹
Mileage (km)	172999 (93465) 162974 9783–646497 [167293; 178705]	Diesel (n = 383)	226951 (91957) 224871 23116–646497 [217712; 236190]
		Fuel (n = 592)	136467 (76316) 128499 9783–428612 [130307; 142628]
		Fuel + Gas (n = 50)	203750 (81403) 198464 44250–384736 [18616; 226885]
		Hybrid (n = 7)	108195 (60824) 132978 34368–169667 [51942; 164448]
		p-value	< 0.0001 ¹
Capacity (cm ³)	1605.4 (454.9) 1591 65–5461 [1577.6; 1633.1]	Diesel (n=383)	1866.2 (351.2) 1896 1199–3996 [1830.9; 1901.5]
		Fuel (n = 592)	1441.2 (437.8) 1389 652–5461 [1405.9; 1476.6]
		Fuel + Gas (n = 50)	1525.6 (459.2) 1396 995–3604 [1395.1; 1656.1]
		Hybrid (n = 7)	1725.0 (205.3) 1798 1395–1991 [1535.1; 1914.9]
		p-value	< 0.0001 ¹

1 – Kruskal-Wallis's test

When comparing vehicle brands, a statistically significant difference in age was observed only between Opel and the other brands (Fig. 1). For mileage, significant differences were observed between Toyota and Volkswagen, as well as between Toyota and Ford. The mileage of

Volkswagen vehicles also differed significantly from that of cars classified as “other brands.” Engine capacity values varied significantly across the analysed brands.

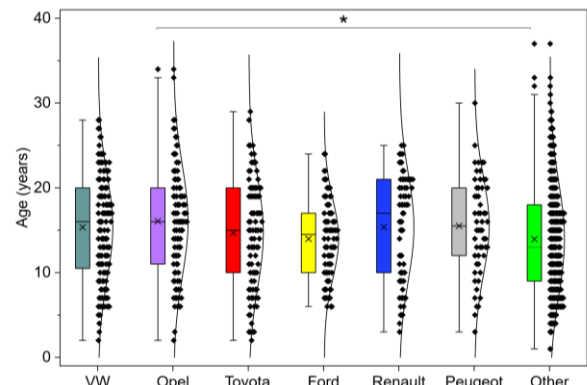


Fig. 1. Age vs brand of the 1033 personal cars

Table 2. Baseline characteristics of car brand (N = 1033); mean (SD), median; Range; [95% CI $\bar{X}_{mean}$]

Brand	Age [years]	Mileage [km]	Capacity [cm ³]
VW (n = 144)	15.3 (6.1)	212187	1683.5 (371.3)
	16	(110150)	1896
	2–28	209334	999–2967
	[14.3; 16.4]	19487–646497 [194042; 230331]	[1622.3; 1744.6]
Opel (n = 93)	16.1 (6.5)	173182	1499.1 (270.7)
	16	(90809)	1443.5
	2–34	171987	973–2172
	[14.8; 17.4]	15509–529722 [154976; 191388]	[1444.9; 1553.4]
Toyota (n = 93)	14.7 (6.5)	143958	1438.5 (377.4)
	15	(84170)	1329
	2–29	143247	998–2231
	[13.3; 16.0]	14494–404067 [126624; 161293]	[1360.7; 1516.2]
Ford (n = 84)	14.0 (4.6)	189152	1689.0 (439.3)
	14.5	(79432)	1596
	6–24	197574	998–3700
	[13.0; 15.0]	34795–344802 [171914; 206390]	[1593.6; 1784.3]
Renault (n = 69)	15.4 (6.3)	166513	1456.2 (308.7)
	17	(79612)	1461
	3–25	161570	898–1998
	[13.8; 16.9]	28961–375903 [147389; 185638]	[1382.0; 1530.4]
Peugeot (n = 56)	15.5 (5.7)	188528	1492.7 (259.4)
	15.5	(99089)	1560
	3–30	177304	954–1997
	[14.0; 17.0]	16864–434568 [161992; 215064]	[1423.2; 1562.1]
Other (n = 489)	14.0 (6.6)	163308	1655.0 (533.4)
	13	(89970)	1591
	1–37	152474	652–5461
	[13.4; 14.5]	9783–475066 [155314; 171302]	[1607.6; 1702.4]
p-value	0.0056 ¹	< 0.0001 ¹	< 0.0001 ¹

1 – Kruskal-Wallis's test

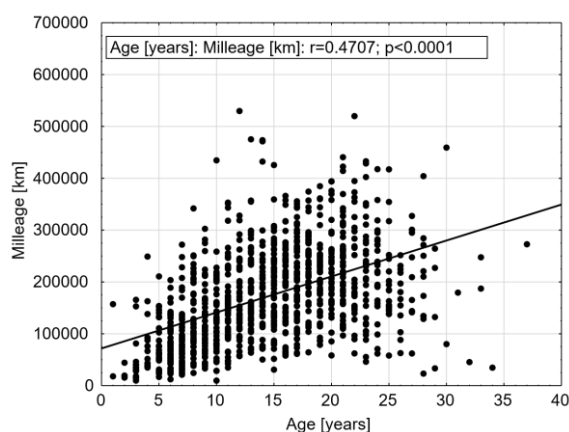


Fig. 2. Mileage vs. age of the 1033 personal cars

The braking force of the left front wheel differed significantly between Toyota and Volkswagen ($p = 0.0324$), Ford ($p = 0.0001$), and other brands ($p = 0.0012$). A statistically significant difference was also observed between Opel and Ford ($p = 0.0027$). The same pattern was found for the right front wheel. For the left rear wheel, braking force differed significantly between Toyota and Volkswagen ($p < 0.0001$), Ford ($p = 0.0002$), and other brands ($p = 0.0020$). Significant differences were also found between Volkswagen and Peugeot ($p = 0.0117$) and between Volkswagen and Renault ($p = 0.0123$). The braking force of the right rear wheel in Toyota vehicles differed from that in Volkswagen ($p = 0.0001$), Ford ($p = 0.0002$), and other brands ($p = 0.0030$). Peugeot vehicles also showed differences compared with Volkswagen ($p = 0.0015$), Ford ($p = 0.0022$), and other brands ($p = 0.0405$). For the parking brake, significant differences were noted between Volkswagen and Toyota

($p = 0.0285$), Opel ($p < 0.0001$), and Renault ($p = 0.0459$). Opel also differed from Ford ($p = 0.0023$) and from other brands ($p = 0.0007$), as illustrated in Fig. 3.

Brake performance was analysed as a key parameter routinely assessed during periodic technical inspections, providing an objective measure of vehicle safety and diagnostic effectiveness.

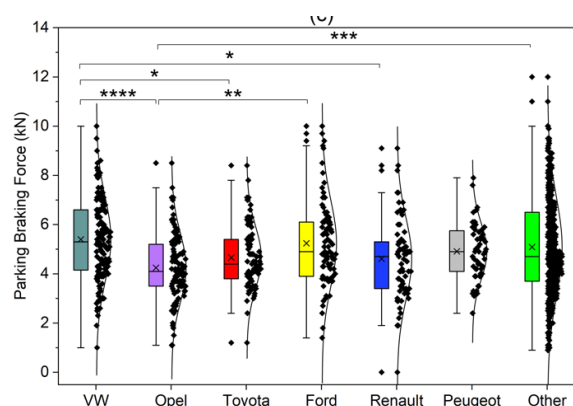


Fig. 3. Force of parking brake (kN) of the 1033 personal cars

Headlight alignment was corrected in 469 vehicles (45.4%). The highest proportions of vehicles requiring adjustment were Ford (54.8%) and Opel (50.0%), while the lowest was recorded for Toyota (34.4%). In this case, vehicle age had no significant effect on whether adjustment was required ($p = 0.2560$; Mann-Whitney U test), whereas mileage was a decisive factor. Vehicles requiring adjustment had an average mileage of 184,339 km, compared to 163,570 km for those not requiring adjustment ($p = 0.0005$; Mann-Whitney U test).

Table 3. Braking force: LF – left front, RF – right front, LR – left rear, RR – right rear, parking brake

Brand	Braking force LF [kN]	Braking force RF [kN]	p-value	Braking force LR [kN]	Braking force RR [kN]	p-value	Braking force parking [kN]
VW (n = 144)	6.5 (1.4) 6.4 3.4–11.2 [6.3; 6.7]	6.5 (1.3) 6.5 3.3–11.3 [6.3; 6.7]	0.8308 ²	2.0 (0.6) 2.1 0.5–3.5 [1.9; 2.1]	2.0 (0.6) 2.0 0.4–3.6 [1.9; 2.1]	0.1798 ²	5.4 (1.7) 5.3 1.0–10.0 [5.1; 5.7]
Opel (n = 93)	6.1 (1.6) 5.9 2.7–11.0 [5.8; 6.4]	6.0 (1.6) 5.8 2.9–10.0 [5.7; 6.3]	0.6553 ²	1.9 (0.6) 1.8 0.6–3.4 [1.8; 2.0]	1.8 (0.5) 1.7 0.6–3.8 [1.7; 1.9]	0.1980 ²	4.2 (1.4) 4.1 1.1–8.5 [4.0; 4.5]
Toyota (n = 93)	5.9 (1.4) 5.5 3.0–9.6 [5.6; 6.2]	5.8 (1.4) 5.7 3.2–9.6 [5.5; 6.1]	0.8190 ²	1.7 (0.5) 1.6 0.6–2.8 [1.6; 1.8]	1.7 (0.9) 1.5 0.7–9.0 [1.5; 1.9]	0.1981 ²	4.7 (1.3) 4.4 1.2–8.4 [4.4; 4.9]
Ford (n = 84)	6.9 (1.4) 6.8 3.9–10.0 [6.6; 7.2]	6.8 (1.4) 6.7 3.9–9.9 [6.5; 7.1]	0.7067 ³	2.1 (0.6) 2.1 0.5–3.5 [1.9; 2.2]	2.0 (0.6) 2.0 0.5–3.6 [1.9; 2.1]	0.2856 ³	5.2 (1.7) 4.9 1.4–10.0 [4.9; 5.6]
Renault (n = 69)	6.4 (1.5) 6.3 3.7–10.0 [6.0; 6.7]	6.3 (1.5) 6.0 3.7–10.0 [5.9; 6.6]	0.5872 ²	1.8 (0.4) 1.7 0.9–2.9 [1.6; 1.9]	1.7 (0.4) 1.6 0.7–3.4 [1.6; 1.8]	0.2331 ²	4.6 (1.6) 4.7 0.0–9.1 [4.2; 5.0]
Peugeot (n = 56)	6.4 (1.4) 6.5 4.0–10.0 [6.0; 6.8]	6.3 (1.3) 6.2 3.7–10.0 [5.9; 6.6]	0.6935 ³	1.7 (0.5) 1.7 0.4–3.1 [1.6; 1.9]	1.6 (0.4) 1.6 0.3–2.9 [1.5; 1.7]	0.0845 ²	4.9 (1.1) 4.9 2.4–7.9 [4.6; 5.2]
Other (n = 489)	6.6 (1.4) 6.5 4.0–10.0 [6.0; 6.8]	6.5 (1.7) 6.4 1.1–11.0 [6.4; 6.7]	0.7931 ¹	2.0 (0.8) 1.9 0.3–6.1 [1.9; 2.1]	1.9 (0.8) 1.8 0.1–9.0 [1.8; 2.0]	0.3263 ¹	5.1 (2.0) 4.7 0.9–12.0 [4.9; 5.3]
p-value	< 0.0001 ¹	< 0.0001 ¹	---	< 0.0001 ¹	< 0.0001 ¹	---	< 0.0001 ¹

¹ – Kruskal-Wallis's test; ² – U Mann-Whitney test; ³ – t Student test

Suspension play was detected in 133 vehicles (12.9%), occurring most frequently in Renault cars (18.8%) and only once in Toyota vehicles (1.1%). Mileage was again a significant factor: vehicles with suspension play had an average mileage of 205,669 km, while those without play averaged 168,171 km ($p < 0.0001$; Mann-Whitney U test).

Corrosion spots were found in 52 vehicles (5.0%), and through-holes in 16 vehicles (1.5%). Vehicles without any corrosion defects were on average 14.2 years old, those with corrosion spots 22.5 years, and those with perforations 24.6 years ($p < 0.0001$; Kruskal-Wallis test). Corrosion was most frequent among Volkswagen (8.3%) and Opel (6.1%) vehicles. The average mileage of vehicles with corrosion was 262,891 km, compared to 166,617 km for corrosion-free cars ($p < 0.0001$; Kruskal-Wallis test).

Visible oil drops were detected in 78 vehicles (7.6%), while oil sweating was observed in 125 vehicles (12.1%). Vehicles without any oil leakage were on average 13.1 years old, those with oil drops 21.1 years, and those with sweating 19.1 years ($p < 0.0001$; Kruskal-Wallis test). Both vehicle age and mileage significantly influenced the presence of oil leaks ($p < 0.0001$; Kruskal-Wallis test). Vehicles without leaks had an average mileage of 147,690 km, those with visible oil drops 210,879 km, and those with sweating 237,705 km. Oil drops occurred most frequently in Opel (19.4%), Volkswagen (10.4%), and Renault (10.1%) vehicles.

Lighting faults were recorded in 55 vehicles (5.3%), occurring most often in Ford (7.1%), Volkswagen (6.3%), and Opel (6.1%) cars, and least frequently in Toyota (3.2%).

3. Case study

Although such components are not directly assessed during standard vehicle inspections, their condition provides valuable insight into the limitations of current diagnostic procedures. To compare the results of statistical analyses conducted at the vehicle inspection station, a detailed technical assessment was performed on a selected passenger car – a Volkswagen Golf V 1.4 TSI, model year 2008, with the CAXA engine code, operated in the western region of Poland. The vehicle's odometer reading at the time of disassembly was 216,514 km. The car held a valid inspection certificate and had undergone a series of repairs at a service associated with the network of workshops eight months earlier. Since that repair, the vehicle had been driven approximately 20,000 km and had subsequently passed another periodic inspection with a positive result.

Before verification of its technical condition, the user reported abnormal noise during engine start-up. It was also noted that the engine tended to run unevenly at idle speed. A computer-based diagnostic test of the warmed-up engine was conducted using the OBD-II (On-Board Diagnostics) system under normal operating temperature. The analysis revealed irregular engine operation at idle and abnormal readings from both lambda sensors – the first sensor indicated a constant voltage, while the second showed no response to increased engine speed, suggesting sensor malfunction or circuit error. It should be noted that not all identified defects are directly detectable during standard vehicle inspections; however, their presence indicates broader limi-

tations in the inspection process and the potential coexistence of detectable and non-detectable faults.

The timing drive mechanism was disassembled to verify the technical condition of its components and subassemblies. Figure 4 shows the components of the engine timing system after disassembly. It was found that aftermarket components of unidentified manufacture had been installed, including the timing chain, camshaft sprocket, and chain guides. The crankshaft sprocket and timing chain tensioner were also of unknown origin. Moreover, the sprocket equipped with the variable valve timing mechanism and the sprocket driving the oil pump chain had not been replaced during the repair. The oil pump drive chain itself appeared to be an original factory component (its markings corresponded with those on the oil pump sprocket), and its guide rail exhibited visible frictional wear. It should be emphasized that the use of non-original components does not imply non-compliance with homologation requirements; however, incomplete replacement of cooperating elements and uncertain component history may adversely affect system reliability.

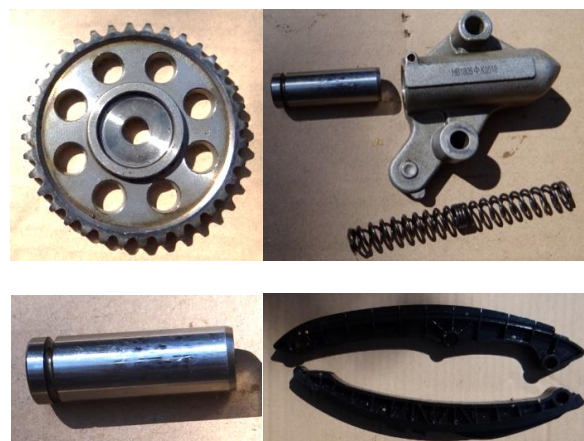


Fig. 4. Components of the engine timing drive mechanism: timing sprocket, chain tensioner (visible scuff marks on the tensioner piston surface), chain tensioner piston (signs of frictional wear), and timing chain guides

Figure 5 shows the gear assembly integrated with the variable valve timing control system. The component dates from 2008 and is evidently the original part installed during the vehicle's production. With a mileage exceeding 200,000 km, and in accordance with the manufacturer's repair technology and commonly available service guidelines, this subassembly should have been unquestionably replaced with a new one during the timing drive repair.

The variable valve timing mechanism forms an integral part of the camshaft drive system, and replacement of this component is mandatory when the timing chain is renewed. According to the repair procedures for chain-driven systems – including those specified by the Volkswagen manufacturer – all cooperating elements of the chain transmission should be replaced with new components. The rationale for this requirement lies in the gradual change in gear tooth geometry that occurs naturally over time.

The CAXA engine oil pump is driven by a steel chain, which, in this case, was replaced with a new component from an unidentified manufacturer. Consequently, a new timing chain was combined with sprockets and chain guides that had been in continuous use since the vehicle's production. Such a combination of new and worn components may lead to improper load distribution and accelerated wear within the system.



Fig. 5. Variable valve timing gear assembly integrated with the camshaft drive sprocket, and oil pump drive gear

Figure 6 shows the guide rail of the oil pump drive chain, which is an original component from the vehicle's production period. Distinct traces of frictional wear are visible on the sliding surface.



Fig. 6. Guide rail of the oil pump drive chain, identified as an original component from the vehicle's production period. Distinct frictional wear marks are visible on the sliding surface

The CAXA engine oil pump is driven by a steel chain, which in this case was replaced with a new but non-original component. Consequently, the new timing chain of un-

known quality was combined with sprockets and chain guides that had been in continuous use since the vehicle's production.

The left rear wheel bearing produced noticeable noise during driving – a symptom typical of a bearing that has either undergone prolonged wear or been incorrectly installed, for example, due to mechanical damage or contamination. When the vehicle was lifted, significant longitudinal play in the left-rear wheel hub assembly was detected. It should be emphasized that operating a vehicle with such a defective bearing is unacceptable and poses a serious safety hazard.

The rear left wheel bearing assembly consists of the hub, two rows of rolling balls, and two inner rings. Visual inspection revealed clear signs of long-term operation, including traces of lubricant leakage and grease with a dense, plastic consistency typical of bearings used for an extended period. Figures 7–9 show the examined bearing, where distinct fatigue damage to the raceways and surface wear of the rolling elements are clearly visible.



Fig. 7. Left rear wheel bearing removed from the vehicle, showing leakage of lubricating grease with a dense, viscous consistency typical of long-term exploitation



Fig. 8. Elements of the left rear wheel bearing: rolling balls on the right display characteristic signs of fatigue-related surface damage



Fig. 9. Outer and inner rings of the left rear wheel bearing with visible surface fatigue and pitting damage on the raceways

Based on the dismantled components, it can be concluded that the wheel bearing had been in operation for much

longer than indicated by the repair documentation, which stated a period of 9 months and approximately 20,000 km. The damage observed on the bearing is typical of wheel bearings that have been used for a very long time. Clear fatigue-related wear was visible on the contact surfaces between the rolling elements and the bearing raceways. Such damage patterns are characteristic of bearings subjected to prolonged loading.

The chassis geometry was measured using a certified computer-based wheel alignment system (four-wheel 3D geometry analyzer, accuracy $\pm 0.01^\circ$), commonly used in professional vehicle inspection stations in Poland. The measurement revealed significant deviations from the manufacturer's specified tolerances (Fig. 10). This raises an important question: can a vehicle that has undergone workshop repairs and passed an official periodic inspection still exhibit such large discrepancies in chassis geometry? Furthermore, can a vehicle with a valid inspection certificate but with suspension geometry values outside the manufacturer's tolerance range be considered safe?

The answer is clear – the vehicle was unsafe and did not provide the required level of safety for its occupants or other road users.

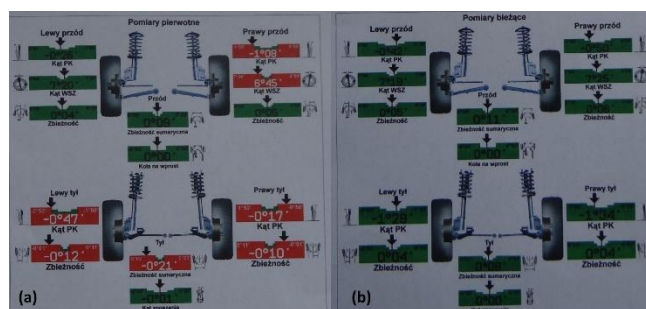


Fig. 10. Results of chassis geometry measurement for the Volkswagen Golf 2004-09 (1K), standard suspension (2UA/1JA), mileage at the time of testing: 216,738 km. (a) Initial measurement before repair; (b) values after suspension repair and alignment adjustment

4. Discussion

The results of the statistical analysis and the detailed technical inspection of the selected passenger car together provide a comprehensive, evidence-based picture of the actual roadworthiness of vehicles operating in western Poland. Unlike model-based approaches, this study is grounded in real inspection data and direct technical verification. The findings demonstrate that even vehicles that successfully pass periodic inspections may exhibit significant technical issues that could compromise operational safety. This observation indicates that the current framework of vehicle technical inspection procedures, although legally regulated and standardized, is not fully effective in identifying critical safety-related defects.

From the statistical data collected on 1033 vehicles inspected between October 2024 and September 2025, several key trends emerge. The mean vehicle age was 14.6 years, which is consistent with the aging vehicle fleet in Poland and significantly above the average for Western Europe. Mileage, averaging approximately 173,000 km, showed a strong correlation with the occurrence of certain technical defects, such as suspension looseness, oil leakage, and

corrosion. Vehicles showing signs of corrosion were, on average, eight years older and had 100,000 km higher mileage than corrosion-free cars ($p < 0.0001$). Similarly, oil leaks and sweating were associated with both increased vehicle age and mileage, indicating that wear-related failures are a major factor influencing vehicle safety and environmental impact. Although the analysis is based on standard statistical methods, the strength of the study lies in the use of real-world inspection data, which enables direct verification of observed relationships.

The data also revealed statistically significant differences among vehicle brands in age, mileage, and braking performance. For example, although younger, Toyota vehicles demonstrated weaker braking forces than Volkswagen and Ford vehicles ($p < 0.05$). This may reflect variations in brake system maintenance practices or differences in design and component quality among brands. Brake test results are particularly relevant in this context, as they represent standardized, measurable parameters used in inspection procedures, enabling comparisons across vehicles and stations. Furthermore, the need for headlight adjustment was strongly associated with vehicle mileage rather than age, suggesting that lighting alignment degradation occurs primarily due to vibration and long-term use rather than material aging alone. These results collectively point to a clear link between vehicle wear and the frequency of defects detectable during periodic inspections. Although the analysis is based on standard statistical methods, the strength of the study lies in the use of real-world inspection data, which enables direct verification of observed relationships. These brand-related differences should be interpreted with caution and do not constitute an evaluation of manufacturer performance, but rather reflect the characteristics of the analysed sample and real-world operating conditions.

However, the most critical insight emerges from the practical verification of a real vehicle case – a Volkswagen Golf V 1.4 TSI (2008) – which clearly demonstrates limitations that cannot be captured by statistical aggregation or theoretical modelling alone. The analysed vehicle had undergone several repairs approximately eight months prior to the inspection and had accumulated an additional 20,000 km during that period. Despite this, the technical evaluation revealed multiple irregularities that should have disqualified the car from being considered roadworthy. The examination of the timing drive system showed the use of non-original replacement parts of unverified quality, including the timing chain, guides, and sprockets. Moreover, key elements such as the variable valve timing gear and the oil pump drive gear remained original from 2008, despite being subject to wear beyond their design limits. The presence of worn chain guides and a scored chain tensioner piston indicated that the repair was not carried out in accordance with manufacturer recommendations. Such practices, unfortunately common in independent workshops, can lead to a substantial reduction in reliability and increase the risk of mechanical failure.

The inspection of the rear left wheel bearing further emphasized the limitations of current inspection procedures. The bearing displayed advanced fatigue damage and excessive longitudinal play, conditions that should have

been easily detectable during a standard suspension and wheel test. The degraded lubricant consistency and the visible pitting of the rolling elements were indicative of prolonged use well beyond the expected exploitation life. These findings demonstrate that the defect was not a result of recent deterioration but of an earlier oversight during both repair and inspection processes. Some of the identified defects fall outside the scope of standard inspection procedures, which further highlights the inherent limitations of such systems. Operating a vehicle in such a condition poses a direct risk to road safety, as bearing failure can lead to wheel lock-up or loss of vehicle stability.

Equally concerning were the findings from the chassis geometry measurements. The alignment values significantly exceeded manufacturer tolerances prior to corrective adjustment, despite the vehicle's successful completion of its last technical inspection. Incorrect wheel geometry can lead to poor handling, uneven tire wear, and increased fuel consumption – all of which have safety and economic consequences. These deviations should have been detected during the official inspection, as the alignment reflects both the integrity of suspension components and the quality of any previous post-accident repairs. The inability of standard inspection stations to detect such issues suggests a need for stricter diagnostic procedures and better-trained inspectors.

Combining these practical results with the statistical analysis leads to several important implications. First, while the majority of inspected vehicles are nominally compliant with regulations, a non-negligible proportion still presents latent defects that directly affect safety. The 12.9% incidence of suspension looseness and the 7.6% rate of active oil leaks, for example, illustrate that mechanical wear and insufficient maintenance are widespread. Second, the high variability in defect detection among brands and stations confirms earlier studies showing that the human factor – diagnostic skill, diligence, and experience – remains a decisive element of inspection reliability [12]. Even though diagnostic equipment has become increasingly sophisticated, its effectiveness depends on the competence of the operator.

The findings of the present study align with earlier international observations [3], which reported that while periodic inspections reduce the number of defective vehicles, they do not necessarily translate into a direct reduction in accident rates. This paradox may be explained by compensatory driver behavior and the fact that certain defects remain undetected during standard inspections. The detailed case analysis conducted in this research further supports the conclusion that the official inspection procedures are insufficiently sensitive to detect many safety-critical issues, particularly those related to wear-induced degradation or improper repairs.

Another crucial aspect revealed by this study concerns the quality of automotive repairs and the responsibility of workshops. The use of non-genuine components of uncertain origin, as observed in the analyzed vehicle, compromises both performance and safety. Although legal mechanisms such as warranty and consumer protection regulations exist, they are often ineffective in practice, especially when technical expertise is required to establish fault.

Therefore, there is a pressing need for a system that ensures greater accountability for repair quality and for integrating workshop data into the official vehicle inspection database.

While advanced diagnostic methods may improve defect detection, their implementation is associated with increased costs, longer inspection times, and the need for specialized training. Moreover, their effectiveness depends strongly on proper execution and interpretation by inspectors. Therefore, improving inspection quality through better supervision, standardized procedures, and sufficient inspection time may be equally important as introducing new technologies.

Finally, the results highlight the need to enhance both procedural and technological aspects of vehicle inspections. Integrating advanced diagnostic methods, such as vibration analysis or on-board sensor data interpretation, could significantly improve defect detection rates. Additionally, a comprehensive training and certification program for inspectors, emphasizing both theoretical knowledge and practical experience, is essential to reduce the influence of the human factor. This perspective is consistent with findings reported in the literature and institutional analyses, which identify insufficient supervision and variability in inspection quality as key limitations of current systems [9, 20, 22, 24]. Given the rapid technological evolution of vehicles – including hybrid and electric powertrains – periodic inspection systems must adapt accordingly to maintain their relevance and effectiveness.

In summary, this study confirms that the technical condition of passenger cars in western Poland remains a significant factor influencing road safety. The results emphasize the importance of continuous improvement of inspection procedures, higher competence standards for inspectors, and stricter quality control in automotive repairs. The combination of statistical evidence and practical diagnostics clearly demonstrates that a valid inspection certificate does not always guarantee actual vehicle safety. Addressing these shortcomings is vital to reducing the risk of accidents and ensuring the long-term reliability of vehicles operating on public roads.

5. Limitations and future work

The present study, while comprehensive in scope, has several limitations that should be acknowledged. First, the data were obtained from a single regional inspection station network in western Poland, which may not fully represent the national vehicle population or inspection practices in other regions. Additionally, although over 1000 vehicles were analyzed statistically, the practical verification involved only one detailed technical case study, limiting the generalizability of the specific mechanical findings. The study also relied on periodic inspection data that do not capture intermediate maintenance or repair activities performed between inspections. Future research should therefore include a broader geographic sample and employ longitudinal monitoring of selected vehicles to observe defect development over time. Expanding the analysis to include vehicle types such as light commercial vehicles, hybrids, and electric cars would also provide valuable insights into evolving maintenance challenges. Furthermore, integrating advanced diagnostic tools – for example, non-invasive

vibration analysis, thermal imaging, or OBD-based fault pattern recognition – could significantly enhance defect-detection sensitivity and support the development of an improved, data-driven framework for vehicle safety inspection.

6. Conclusions

The study revealed that the actual technical condition of many passenger cars in western Poland does not fully correspond to their official inspection results. Statistical analysis of over one thousand vehicles showed that defects such as corrosion, suspension looseness, and oil leakage are significantly related to vehicle age and mileage. The detailed inspection of a Volkswagen Golf V confirmed that even vehicles with valid inspection certificates may exhibit

serious faults, including worn components, incorrect chassis geometry, and non-original parts of uncertain quality. The study provides practical, evidence-based insight into the real effectiveness of vehicle inspection systems, complementing existing model-based approaches. This study contributes to the field by providing empirical, evidence-based insight into the real effectiveness of vehicle inspection systems and highlighting their practical limitations.

7. Practical implications

The results can support policymakers, inspection authorities, and vehicle service organizations in developing more effective inspection standards and training programs for automotive diagnosticians, ultimately contributing to higher safety and technical reliability on public roads.

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Engine NO_x control of the SCR system using a sliding mode controller

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The selective catalytic reduction system (SCR) uses aqueous NH₃ and a catalyst for reducing the NO_x emission emitted by the compression ignition (CI) engine exhaust. Due to stringent emission regulations and the growing demand for SCR systems, significant improvements in urea injection strategies and catalyst activity are required to reduce exhaust gas by-products. In this study, an SCR system combined with a super-twisting sliding mode controller (STSMC) is employed to regulate the exhaust gases of CI engines. The SCR system is represented by linearized state-space models of third and fourth order. The US06 emission cycle is used to determine the engine exhaust NO_x characteristics, which are then fed into the SCR system. According to the simulation results, the fourth order SCR model has a better de-NO_x conversion rate and less NH₃ slip than the third order SCR model, due to the fast-SCR kinematics.

Key words: CI engine, NO_x, SCR, super-twisting sliding mode controller, US06This is an open access article under the CC BY license (<http://creativecommons.org/licenses/by/4.0/>)

1. Introduction

IC engines remain the most popular for automotive applications [8, 21]. The primary pollutant emitted from the exhaust of CI engines is NO_x, and its stringent emission norms have become increasingly stringent in recent decades [7, 18]. The urea-based SCR system has great potential and widespread acceptance for reducing NO_x emission [4, 13]. The SCR system uses aqueous NH₃, which reacts with exhaust NO_x in the presence of a catalyst to convert them into N₂ and H₂O, reducing tailpipe emissions [10]. Controlling exhaust NO_x and NH₃ slip under engine operating conditions with an effective catalyst remains a major challenge for the SCR system in real time [26]. Excessive NH₃ dosing causes ammonia slip, and inadequate NH₃ dosing causes NO_x slip by the SCR system [25].

A dual-cell SCR system for a diesel engine is designed, and a fuzzy logic controller is used for urea dosing to enhance NO_x conversion efficiency and reduce NH₃ tailpipe emissions [27]. A three state SCR model is developed and SMC is designed to control the NO_x and NH₃ pollutants [3]. A singular perturbation approach is used to separate tailpipe NO_x from NH₃ and to predict the NH₃ coverage factor of the urea-SCR system using the NO_x sensor [15]. An SMC is considered a reliable controller for advanced nonlinear systems, subjected to parametric uncertainties and external perturbations [1, 17]. The primary aspect of SMC is that it modifies the system dynamics by selecting a suitable switching function, keeping its closed-loop response unaffected by system uncertainties [5]. Super Twisting is one of the higher order SMC that minimizes the chattering effect without using the time derivatives of the sliding variable. [16].

Numerous investigations into SCR systems for IC engines have been carried out. However, stringent emission regulations have necessitated an advanced SCR system controller in order to reduce exhaust emissions. This study presents the use of DSTSMC in the SCR system to control NO_x emissions from the CI engine. The paper is organized as follows. NO_x emissions from the CI engine are discussed in Section 2. The methodology of the SCR system is briefed

in Section 3. Sections 4 and 5 discuss the state space models of the SCR system. The controller design is briefed in Section 6. Section 7 discusses the results. Section 8 presents conclusions.

2. Exhaust NO_x emission for CI engine

This work follows the procedure published in [3] and [20]. The test bench consists of a medium-duty CI engine connected to a dynamometer. The test bench is started, and then the dynamometer load is applied to the engine. The diesel engine is tested over a sequence of US06 emission cycles, and the NO_x emissions are obtained [9] and [15]. The engine performance characteristics during the US06 cycle are listed in Table 1.

Table 1. CI engine performance characteristics

Time [s]	Engine speed [rpm]	Engine torque [Nm]	Exhaust gas flow rate [m ³ /s]	Exhaust gas temp [K]
0	1000	50	0.68	485
50	1200	100	0.05	508
100	2500	150	0.35	523
150	3000	200	0.05	548
200	2000	100	0.15	563
250	3500	250	0.05	588
300	4000	300	0.15	600
350	4500	280	0.05	623
400	3000	150	0.15	635
450	2000	100	0.05	645
500	1500	80	0.15	660
550	1000	60	0.05	675
600	800	40	0.15	685

As engine load increases, its combustion temperature rises, resulting in higher concentrations of C_{NO}, C_{NO2} emissions. Engine NO_x during US06 emission cycle is displayed in Fig. 1. The engine exhaust emission of C_{NO} is only considered for third order model, whereas the engine exhaust emission of both C_{NO} and C_{NO2} are considered for fourth

order model [3]. The average $C_{NO,in}$ is 212 ppm for third-order model for US06 cycle. The average $C_{NO,in}$ and $C_{NO2,in}$ are 212 ppm and 60 ppm, for fourth order model for US06 cycle.

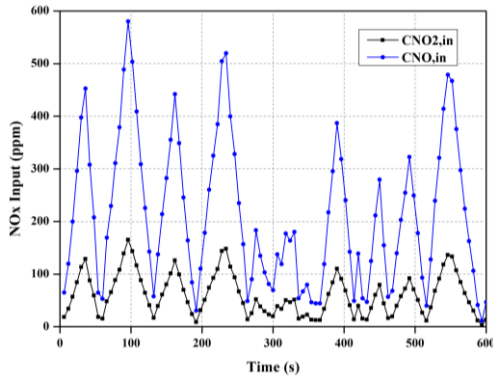


Fig. 1. Engine NO_x using US06

3. Methodology of the SCR system

The connection between the CI engine and the SCR system is shown in Fig. 2. A dynamometer is connected to a medium-duty V-8 automotive diesel engine at the inlet and the engine outlet is connected to the emission control system, which has a diesel oxidation catalyst (DOC), a diesel particulate filter (DPF) and a dual SCR catalyst [9, 11]. NO_x, NH₃, and thermocouple sensors are used to monitor the system. The SCR system injects aqueous NH₃ at the inlet. The dosing control unit determines the NH₃ injection quantity according to the sensors input [14]. A dual cell SCR is assumed to be a lumped single-cell SCR in this study.

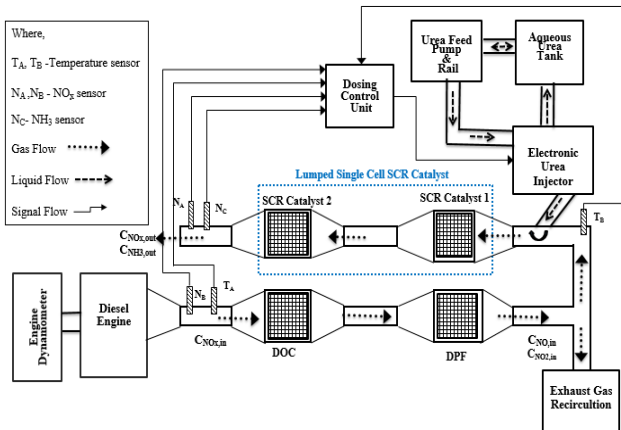


Fig. 2. Exhaust gas aftertreatment method using SCR

4. Third order model

4.1. Nonlinear model

The SCR model [24] and [14] is given by:

$$\begin{bmatrix} \dot{C}_{NO} \\ \dot{\theta}_{NH_3} \\ \dot{C}_{NH_3} \end{bmatrix} = \begin{bmatrix} -(r_{red}\phi_1\theta_{NH_3}C_{O_2}V + \frac{F}{V}) \\ \frac{1}{\theta_{NH_3}}(r_{ads}C_{NH_3}V) - r_{ads}C_{NH_3}V - (r_{des} + r_{red}C_{NO}C_{O_2}V^2) \\ -[\phi_1r_{ads}(1 - \theta_{NH_3}) + \frac{F}{V}] + \frac{1}{C_{NH_3} * V}(r_{des}\phi_1\theta_{NH_3}) \end{bmatrix} \begin{bmatrix} C_{NO} \\ \theta_{NH_3} \\ C_{NH_3} \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \frac{F}{V} \end{bmatrix} C_{NH_3,in} + \begin{bmatrix} \frac{F}{V} \\ 0 \\ 0 \end{bmatrix} C_{NO,in} \quad (1)$$

$$y = \begin{bmatrix} 1 & 0 & 0 \end{bmatrix} \begin{bmatrix} C_{NO} \\ \theta_{NH_3} \\ C_{NH_3} \end{bmatrix} \quad (2)$$

The rate of SCR system (r_z) is modelled using the Arrhenius equation, which is given by:

$$r_z = K_z \exp \left[\frac{E_z}{RT} \right], \text{ where } z \in \{r_{red}, r_{ads}, r_{des}\} \quad (3)$$

where: r_{ads} , r_{des} , r_{red} are NH₃ adsorption, NH₃ desorption, and SCR reduction reaction rates, respectively. K_z is frequency constant and E_z is activation energy constant. V is the catalyst volume, T is the exhaust gas temperature, F is the exhaust flow rate, ϕ_1 is ammonia storage capacity. C_{NH_3} , $C_{NH_3,in}$ are the concentration of ammonia at outlet and inlet, respectively. $C_{NO,in}$ and C_{NO} is the concentration of nitric oxide at inlet and outlet, respectively. C_{O_2} refers the oxygen concentration, θ_{NH_3} is the surface coverage fraction.

4.2 Linearized model

The nonlinear SCR model given in Eq. (1) and Eq. (2) are linearized [19, 24] as given in Table 2, and the resulting is:

$$\begin{bmatrix} \dot{C}_{NO} \\ \dot{\theta}_{NH_3} \\ \dot{C}_{NH_3} \end{bmatrix} = \begin{bmatrix} A_{11} & A_{12} & 0 \\ A_{21} & A_{22} & A_{23} \\ 0 & A_{32} & A_{33} \end{bmatrix} \begin{bmatrix} C_{NO} \\ \theta_{NH_3} \\ C_{NH_3} \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \frac{F}{V} \end{bmatrix} U + \begin{bmatrix} \frac{F}{V} \\ 0 \\ 0 \end{bmatrix} C_{NO,in} \quad (4)$$

$$y = \begin{bmatrix} 1 & 0 & 0 \end{bmatrix} \begin{bmatrix} C_{NO} \\ \theta_{NH_3} \\ C_{NH_3} \end{bmatrix} \quad (5)$$

where:

$$A_{11} = -\phi_1 r_{red} C_{O_2} V \theta_{NH_3} - \frac{F}{V}$$

$$A_{12} = -\phi_1 r_{red} C_{O_2} V C_{NO}$$

$$A_{21} = -r_{red} C_{O_2} V^2 \theta_{NH_3}$$

$$A_{22} = -r_{ads} V C_{NH_3} - (r_{des} + r_{red} C_{O_2} V^2 C_{NO})$$

$$A_{23} = (1 - \theta_{NH_3}) r_{ads} V$$

$$A_{32} = (\phi_1 r_{ads} C_{NH_3}) + \frac{(\phi_1 r_{des})}{V}$$

$$A_{33} = -(\phi_1 r_{ads}(1 - \theta_{NH_3}) + \frac{F}{V})$$

Table 2. Engine exhaust characteristics

Operating Conditions	Values
Flow rate (F)	0.167 m ³ /s
Temperature (T)	595 K
Oxygen concentration ($C_{O_2,in}$)	2 moles/m ³
Nitric oxide concentration ($C_{NO,in}$)	12 ppm
Nitrogen dioxide levels ($C_{NO_2,in}$)	4 ppm

Using the catalyst specifications and kinetic parameters [23] and [15] listed in Table 3 and Table 4 respectively, the result is:

$$\dot{X}(t) = A X(t) + B U(t) + E_1 d_{CNO}(t) \quad (6)$$

$$Y(t) = H X(t) \quad (7)$$

where: $X(t) = \begin{bmatrix} X_1(t) \\ X_2(t) \\ X_3(t) \end{bmatrix} = \begin{bmatrix} C_{NO} \\ \theta_{NH_3} \\ C_{NH_3} \end{bmatrix}$, $U(t)$ is the input vector, $d_{CNO}(t)$ is NO inlet disturbance

$$A = \begin{bmatrix} -619.29 & -3.1 \times 10^1 & 0 \\ -5.48 \times 10^{-2} & -5.20 \times 10^2 & 6.73 \times 10^{-2} \\ 0 & 5.72 \times 10^6 & -7.57 \times 10^2 \end{bmatrix} B = \begin{bmatrix} 0 \\ 0 \\ 16.7 \end{bmatrix} E_1 = \begin{bmatrix} 16.7 \\ 0 \\ 0 \end{bmatrix}$$

Table 3. SCR catalyst specification

Specification	Values
Type	Cu-Zeolite
Volume (V)	0.005 m ³
Space velocity (F/V)	33 s ⁻¹
C _{NH3} storage capacity (ϕ ₁)	120 moles/m ³
C _{NH3} desorption efficiency (δ ₁ , δ ₂)	0.5 and 0.5
C _{NH3} coverage fraction (θ _{NH3})	0.25

Table 4. SCR catalyst kinetic rate

Reaction	Frequency factor (K _z)	Activation energy (E _z)
Standard SCR (r _{red})	2.74 × 10 ⁸ m ³ /mol·s	76 kJ/mol
NH ₃ adsorption (r _{ads})	6.8 m ³ /mol·s	0 kJ/mol
NH ₃ desorption (r _{des})	5.202 × 10 ² 1/s	48 kJ/mol

4.3. Discrete model

The linear model given in Eq. (6) and Eq. (7) is discretized for 20 μs, the resulting is:

$$X(k+1) = A_d X(k) + B_d U(k) + E_{d1} d_{CNO}(k) \quad (8)$$

$$Y(k) = H_d X(k) \quad (9)$$

where:

$$A_d = \begin{bmatrix} 0.9877 & -0.0006 & -0.0000 \\ -0.0000 & 0.9847 & 0.0000 \\ -0.0001 & 112.9514 & 0.9850 \end{bmatrix} B_d = \begin{bmatrix} 0 \\ 0 \\ 3.31 \times 10^{-4} \end{bmatrix} E_{d1} = \begin{bmatrix} 3.18 \times 10^{-4} \\ 0 \\ 0 \end{bmatrix} H_d = [1 \quad 0 \quad 0]$$

5. Fourth order model

5.1. Nonlinear model

The SCR model [2] proposed by is given in Annex 1 – Eq. (10) and Eq. (11).

where ϕ is NH₃ storage capacity, with S₁ and S₂ as constants given by:

$$\phi = S_1 \exp(S_2 T) \quad (12)$$

The Arrhenius equations are used for the reaction rates (r_m), and given by:

$$r_m = K_m \exp \left[\frac{E_m}{RT} \right], \text{ where } m \in \{r_a, r_b, r_c, r_{d1}, r_{d2}, r_e\} \quad (13)$$

$$r_a = K_1 e^{-\left(\frac{E_1}{RT}\right)} C_{NO} C_{O_2} \theta_{NH_3} \phi V^2 \quad (14)$$

$$r_b = K_2 e^{-\left(\frac{E_2}{RT}\right)} C_{NO} C_{NO_2} \theta_{NH_3} \phi V^2 \quad (15)$$

$$r_c = K_3 e^{-\left(\frac{E_3}{RT}\right)} C_{O_2} \theta_{NH_3} \phi V \delta_1 \quad (16)$$

$$r_{d1} = K_{4F} e^{-\left(\frac{E_{4F}}{RT}\right)} C_{NH_3} (1 - \theta_{NH_3}) \phi V \quad (17)$$

$$r_{d2} = K_{4R} e^{-\left(\frac{E_{4R}}{RT}\right)} \theta_{NH_3} \phi \delta_2 \quad (18)$$

$$r_e = K_5 e^{-\left(\frac{E_5}{RT}\right)} C_{NO} C_{O_2} V^2 \quad (19)$$

where: E_m is activation energy constant, K_m is frequency factor. The representation of reaction rates r_a is SCR, r_b is fast SCR, r_c is NH₃ oxidation, r_{d1} is ammonia adsorption, r_{d2} is ammonia desorption and r_e is NO oxidation, C_{NO₂} is outlet concentration of NO₂. δ₁, δ₂ are NH₃ absorption and desorption efficiency, respectively. C_{NO₂,in} is inlet concentration of NO₂, C_{NO} is the outlet concentration of NO and θ_{NH₃} is the surface coverage fraction.

5.2. Linearized model

The nonlinear SCR model given in Eq. (10) and Eq. (11) are linearized [2] and [19] given in Table 2, and resulting is:

$$\begin{bmatrix} \dot{C}_{NO} \\ \dot{C}_{NO_2} \\ \dot{\theta}_{NH_3} \\ \dot{C}_{NH_3} \end{bmatrix} = \begin{bmatrix} A_{11} & A_{12} & A_{13} & 0 \\ A_{21} & A_{22} & A_{23} & 0 \\ A_{31} & A_{32} & A_{33} & A_{34} \\ 0 & 0 & A_{43} & A_{44} \end{bmatrix} \begin{bmatrix} C_{NO} \\ C_{NO_2} \\ \theta_{NH_3} \\ C_{NH_3} \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ \frac{F}{V} \end{bmatrix} U + \begin{bmatrix} \frac{F}{V} \\ 0 \\ 0 \\ 0 \end{bmatrix} C_{NO,in} + \begin{bmatrix} 0 \\ \frac{F}{V} \\ 0 \\ 0 \end{bmatrix} C_{NO_2,in} \quad (20)$$

$$y = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \end{bmatrix} \begin{bmatrix} C_{NO} \\ C_{NO_2} \\ \theta_{NH_3} \\ C_{NH_3} \end{bmatrix} \quad (21)$$

where:

$$A_{11} = -r_a C_{O_2} \phi V \theta_{NH_3} - \frac{1}{2} r_b \phi V C_{NO_2} \theta_{NH_3} - r_c C_{O_2} V - \frac{F}{V}$$

$$A_{12} = -\frac{1}{2} r_b \phi V C_{NO} \theta_{NH_3}$$

$$A_{13} = -r_a C_{O_2} \phi V C_{NO} - \frac{1}{2} r_b \phi V C_{NO} C_{NO_2}$$

$$A_{21} = -\frac{1}{2} r_b \phi V C_{NO_2} \theta_{NH_3} + r_c C_{O_2} V$$

$$A_{22} = -\frac{1}{2}r_b \emptyset VC_{NO} \theta_{NH_3} - \frac{F}{V}$$

$$A_{23} = -\frac{1}{2}r_b \emptyset VC_{NO} C_{NO_2}$$

$$A_{31} = -(r_a C_{O_2} V^2 + r_b V^2 C_{NO_2}) \theta_{NH_3}$$

$$A_{32} = -(r_b V^2 C_{NO}) \theta_{NH_3}$$

$$A_{33} = -(r_{d1} V C_{NH_3} + r_c C_{O_2} V \delta_1 + r_{d2} \delta_2 + r_a C_{O_2} V^2 C_{NO} + r_b V^2 C_{NO} C_{NO_2})$$

$$A_{34} = -\theta_{NH_3} (r_{d1} V) + r_{d1} V$$

$$A_{43} = -\emptyset r_{d1} C_{NH_3} + \frac{1}{V} r_{d2} \emptyset \delta_2$$

$$A_{44} = -(\emptyset r_{d1} (1 - \theta_{NH_3}) + \frac{F}{V})$$

Using the SCR catalyst specifications and, kinetic parameters [23] listed in Table 3 and Table 5 respectively, the result is:

$$\dot{X}(t) = A X(t) + B U(t) + E_1 d_{CNO}(t) + E_2 d_{CNO_2}(t) \quad (22)$$

$$Y(t) = H X(t) \quad (23)$$

where: $X(t) = \begin{bmatrix} X_1(t) \\ X_2(t) \\ X_3(t) \\ X_4(t) \end{bmatrix} = \begin{bmatrix} C_{NO} \\ C_{NO_2} \\ \theta_{NH_3} \\ C_{NH_3} \end{bmatrix}$,

$d_{CNO}(t)$ and $d_{CNO_2}(t)$ is NO and NO₂ inlet disturbances, respectively.

$$A = \begin{bmatrix} -42709.7 & -2.95 \times 10^4 & -2.22 \times 10^2 & 0 \\ -1.86 \times 10^4 & -2.95 \times 10^4 & -9.66 \times 10^1 & 0 \\ -5.57 & -5.36 & -7.35 \times 10^3 & -6.69 \times 10^{-2} \\ 0 & 0 & 67321.24916 & -129.12 \end{bmatrix}$$

$$B = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 16.7 \end{bmatrix}$$

$$E_1 = \begin{bmatrix} 16.7 \\ 0 \\ 0 \\ 0 \end{bmatrix} \quad E_2 = \begin{bmatrix} 0 \\ 16.7 \\ 0 \\ 0 \end{bmatrix} \quad H = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \end{bmatrix}$$

Table 5. SCR catalyst kinetic rate

Reaction	Frequency factor (K _m)	Activation energy (E _m)
Standard SCR (r _a)	2.74 × 10 ⁸ m ³ /mol·s	75 kJ/mol
Fast SCR (r _b)	1.03 × 10 ⁹ m ⁶ /mol ² ·s	40 kJ/mol
NH ₃ oxidation (r _c)	1.02 × 10 ⁵ 1/s	87 kJ/mol
NH ₃ Adsorption (r _{d1})	6.8 m ³ /mol·s	0 kJ/mol
NH ₃ Desorption (r _{d2})	5.2 × 10 ² 1/s	47 kJ/mol
NO oxidation (r _e)	3.99 × 10 ⁴ 1/s	57 kJ/mol

5.3. Discrete model

The linear model given in Eq. (22) and Eq. (23) is discretized for 1 μs, the result is:

$$X(k+1) = A_d X(k) + B_d U(k) + E_{d1} d_{CNO}(k) + E_{d2} d_{CNO_2}(k) \quad (24)$$

$$Y(k) = H_d X(k) \quad (25)$$

where

$$A_d = \begin{bmatrix} 0.9585 & -0.0285 & -0.0002 & -0.0000 \\ -0.0179 & 0.9712 & -0.0001 & -0.0000 \\ -0.0000 & -0.0000 & 0.9927 & -0.0000 \\ -0.0000 & -0.0000 & 0.0671 & 0.9999 \end{bmatrix} \quad B_d = \begin{bmatrix} 0.0000 \\ 0.0000 \\ -0.0000 \\ 1.67 \times 10^{-5} \end{bmatrix}$$

$$E_{d1} = \begin{bmatrix} 1.63 \times 10^{-5} \\ -0.0000 \\ -0.0000 \\ 0.0000 \end{bmatrix} \quad E_{d2} = \begin{bmatrix} -0.0000 \\ 1.64 \times 10^{-5} \\ -0.0000 \\ -0.0000 \end{bmatrix} \quad H = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \end{bmatrix}$$

6. Controller design

The first step is to identify a switching function [5], which results in

$$S = C^T X_d(k) \quad (26)$$

The switching function represents

$$S = C_1 X_1(k) + C_2 X_2(k) + \dots + C_n X_n(k) = C^T X_d(k) = 0 \quad (27)$$

Quasi-sliding mode conditions [6] is given by

$$S(k+1) = S(k) = 0, \text{ where } k = 0, 1, 2, 3, 4, \dots \quad (28)$$

Sliding surface results in

$$C^T = [C_1 \quad C_2 \quad C_3 \quad C_4 \dots C_n]$$

where:

$$C_1 = (-1)^{n-1} (\lambda_1 \lambda_2 \dots \lambda_{n-1}) \quad (29)$$

$$C_2 = (-1)^{n-2} (\lambda_2 \lambda_3 \dots \lambda_{n-1} + \lambda_1 \lambda_3 \dots \lambda_{n-1} + \lambda_1 \lambda_2 \dots \lambda_{n-1})$$

$$C_{n-1} = (-1) (\lambda_1 + \lambda_2 + \lambda_3 + \dots + \lambda_{n-1})$$

$$C_n = 1$$

with $\lambda_1, \lambda_2, \dots, \lambda_{n-1}$, are the desired eigen values. Choosing the appropriate eigenvalues yields the switching vector [6].

The second step is to find an appropriate controller that leads the system states to migrate on a switching surface. Super-Twisting Sliding Mode Controller (STSMC) is one of the best higher-order SMC [16] and [12], represented by:

$$U(k) = \lambda_{k1} |S(k)|^{\frac{1}{2}} \text{sign}(S(k)) + U_1(k) \quad (30)$$

$$U_1(k+1) = U_1(k) + \lambda_{k2} \text{sign}(S(k))$$

$$\text{sign}(S(k)) = \begin{cases} 1, & S(k) > 0 \\ \in [-1, 1], & S(k) = 0 \\ -1, & S(k) < 0 \end{cases} \quad (31)$$

where $S(k)$ is the sliding variable. The tuning parameters of the controllers are denoted by λ_{k1} and λ_{k2} .

6.1. DSTSMC design for third order model

The open loop eigenvalues are $\lambda_{a1} = 0.9877$, $\lambda_{a2} = 0.999$, $\lambda_{a3} = 0.9749$. For the desired closed loop eigenvalues of $\lambda_{b1} = -0.3$, $\lambda_{b2} = -0.4$, the sliding surface resulting is:

$$C^T = [0.1200 \quad 0.7000 \quad 1.0000]$$

The controller tuning parameters resulted in $\lambda_{k1} = 17$ and $\lambda_{k2} = 4 \times 10^{-4}$.

6.2. DSTSMC design for fourth order model

The open loop eigen values are $\lambda_{c1} = 0.9413$, $\lambda_{c2} = 0.9883$, $\lambda_{c3} = 0.9927$, $\lambda_{c4} = 0.999$. For the desired closed loop eigenvalues of $\lambda_{d1} = -0.4$, $\lambda_{d2} = -0.3$, $\lambda_{d3} = -0.2$, the sliding surface resulting is

$$C^T = [0.0240 \quad 0.2600 \quad 0.9000 \quad 1.0000]$$

The tuning parameters of the controller resulted in $\lambda_{k1} = 0.3$ and $\lambda_{k2} = 1 \times 10^{-7}$.

7. Results and discussion

The performance of the SCR system with STSMC is simulated. The De-NO_x conversion efficiency, NH₃ reacted, and NH₃ slip of SCR system [22] are determined by:

$$\text{De-NO}_x \text{ efficiency} = 1 - \left(\frac{C_{\text{NO},\text{out}}}{C_{\text{NO},\text{in}}} \right) \quad (32)$$

$$\text{NH}_3 \text{ Slip} = \text{NH}_3 \text{ injected} - \text{NH}_3 \text{ reacted} \quad (33)$$

$$\text{NH}_3 \text{ reacted} = (C_{\text{NO}_x,\text{in}}) \left(1 - \left(\frac{C_{\text{NO},\text{out}}}{C_{\text{NO},\text{in}}} \right) \right) \quad (34)$$

7.1. Outlet concentration of NO

The DSTSMC modifies the NH₃ injection in response to the inlet NO, and subsequently, the SCR system reacts to reduce NO emissions. From the results, third and fourth order SCR models have an average $C_{\text{NO},\text{out}}$ of 4.9 ppm and 3.6 ppm, respectively, over 600 s. The outlet concentration of nitric oxide is shown in Fig. 3.

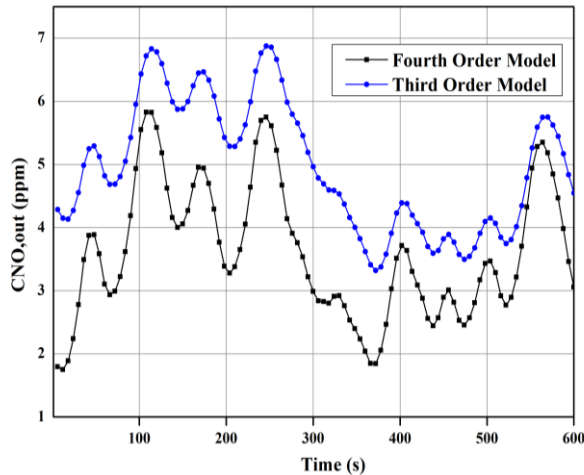


Fig. 3. Outlet concentration of nitric oxide

7.2. Outlet concentration of NO₂

The outlet concentration of Nitrogen di-oxide is displayed in Fig. 4. The SCR system reduces the average $C_{\text{NO}_2,\text{out}}$ to 1.4 ppm over 600 s for fourth order model.

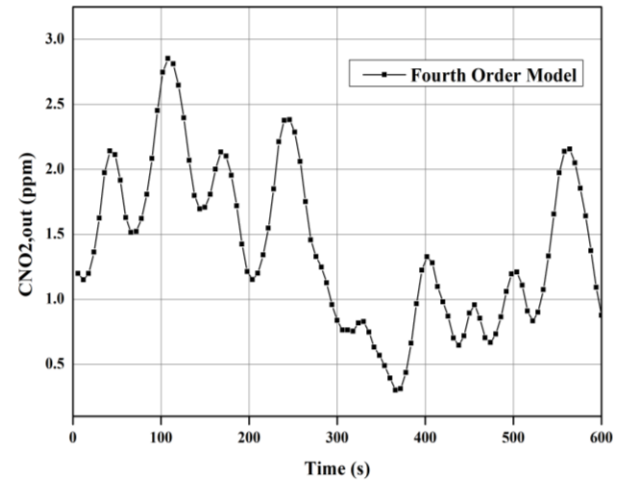


Fig. 4. Outlet concentration of nitrogen di-oxide

7.3. Ammonia reacted ratio

The SCR with STSMC maintains the C_{NH_3} reacted ratio as 0.98:1 and 0.99:1 for third and fourth order models, respectively. The reacted ratio levels are equivalent to outlet concentration of oxides of nitrogen decreased by the SCR system. The ammonia reacted ratio is shown in Fig. 5. According to the results, the high catalytic activation rate of fast-SCR causes the fourth order model to react more quickly which produces less slip of exhaust gases.

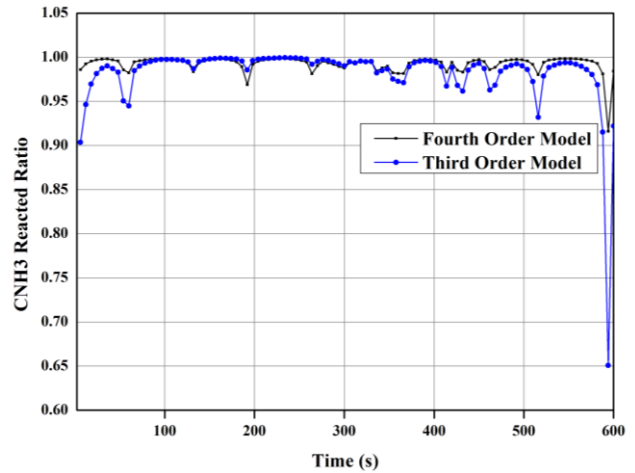


Fig. 5. Ammonia reacted ratio

7.4. De-NO_x efficiency

The C_{NO} conversion efficiencies are 95.7 % and 97% for the third and fourth order models, respectively. In addition, the fourth order model has C_{NO_2} conversion efficiency of 96.2%. Therefore, the fourth order model has a total de-NO_x efficiency of 96.6%. Fig. 6 displays oxides of Nitrogen conversion efficiency. Hence, the SCR system meets the stringent emission standards.

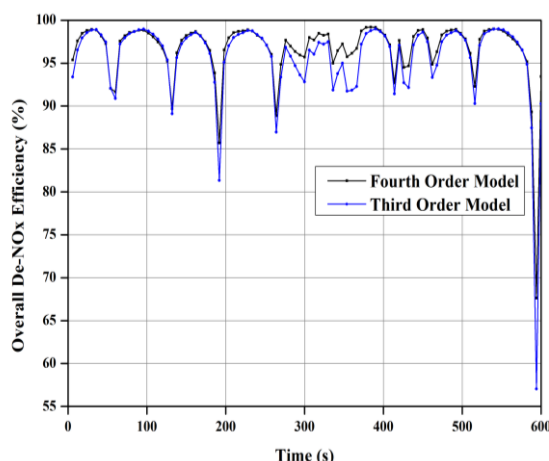


Fig. 6. Oxides of nitrogen conversion efficiency

7.5. Ammonia slip

The average C_{NH_3} slip of third and fourth order models are 1.77 ppm and 1.04 ppm, respectively over 600 s. The initial rise in NH_3 slip is caused by the controller overdosing, and the gain in de- NO_x conversion is negligible. The C_{NH_3} slip is presented in Fig. 7.

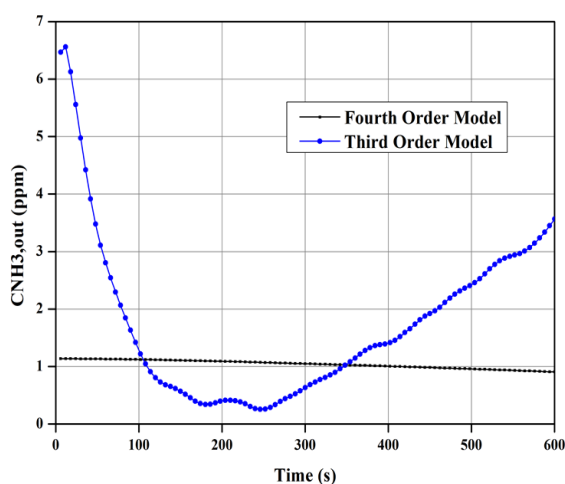


Fig. 7. Outlet concentration of ammonia

7.6. Ammonia injection ratio

The average C_{NH_3} injection ratio and the average usage of urea for third order model are 0.96:1 and 208 ppm,

respectively, over 600 s. The average C_{NH_3} injection ratio and the average usage of urea for fourth order model are 0.97:1 and 268 ppm, respectively, over 600 s. The results show that the fourth order model utilizes more NH_3 , but it exhibits a greater conversion rate with negligible NH_3 slip because to the fast-SCR kinetics. The ammonia injection ratio is shown in Fig. 8.

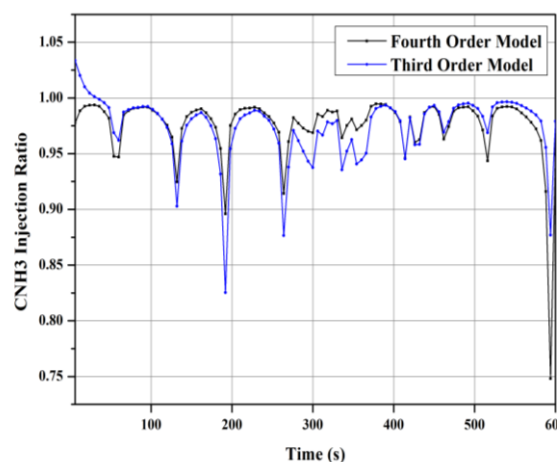


Fig. 8. Ammonia injection ratio

Conclusions

This work presents exhaust NO_x control for SCR system for automotive CI engine utilizing DSTSMC and the results are simulated. The third and fourth order SCR models exhibits de- NO_x conversion rate of 95.7% and 96.6%, respectively over 600 s. The average C_{NH_3} slip for the third order model is 1.77 ppm over 600 seconds. The fourth order SCR model has an average C_{NH_3} slip of 1.04 ppm over 600 seconds. The injection ratio and reacted ratio of ammonia to NO_x are enhanced for fourth order model due to fast-SCR kinetics. Hence, STSMC for the SCR system attains the targeted pollution norms for CI engines. In the future, an experimental validation of the SCR system in real-time using sensors and hardware for an automotive CI engine will be implemented. An on-board diagnostics-based SCR system with catalyst multi-functions such as catalyst ageing, thermal de-activation, and catalyst poisoning will be employed.

Nomenclature

BP	brake power
DOC	diesel oxidation catalyst
DPF	diesel particulate filter
IC	internal combustion

PM	particulate matter
SCR	selective catalytic reduction
SMC	sliding mode controller
STSMC	super-twisting sliding mode controller

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Annex

$$\begin{bmatrix} \dot{C}_{NO} \\ \dot{C}_{NO_2} \\ \dot{\theta}_{NH_3} \\ \dot{C}_{NH_3} \end{bmatrix} = \begin{bmatrix} -r_a C_{O_2} \theta_{NH_3} \phi V - \frac{1}{2} r_b C_{NO_2} \theta_{NH_3} \phi V - r_c C_{O_2} - \frac{F}{V} + \frac{F}{V} C_{NO,in} \\ -\frac{1}{2} r_b C_{NO} \theta_{NH_3} \phi V + \frac{1}{C_{NO_2}} r_c C_{NO} C_{O_2} V - \frac{F}{V} + \frac{F}{V} C_{NO_2,in} \\ -(r_{d1} C_{NH_3} V + r_c C_{O_2} V \delta_1 + r_{d2} \delta_2 + r_a C_{NO} C_{O_2} V^2 + r_b C_{NO} C_{NO_2} V^2) + \frac{1}{\theta_{NH_3}} r_{d1} C_{NH_3} V \\ -\left[\phi r_{d1} (1 - \theta_{NH_3}) + \frac{F}{V}\right] + \frac{1}{V * C_{NH_3}} r_{d2} \phi \theta_{NH_3} \delta_2 \end{bmatrix} \begin{bmatrix} C_{NO} \\ C_{NO_2} \\ \theta_{NH_3} \\ C_{NH_3} \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ 0 \\ \frac{F}{V} \end{bmatrix} C_{NH_3,in} \quad (10)$$

$$y = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \end{bmatrix} \begin{bmatrix} C_{NO} \\ C_{NO_2} \\ \theta_{NH_3} \\ C_{NH_3} \end{bmatrix} \quad (11)$$

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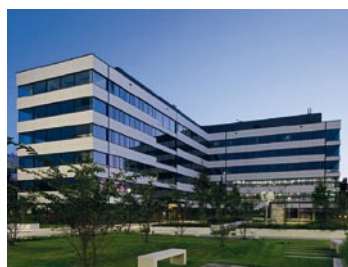
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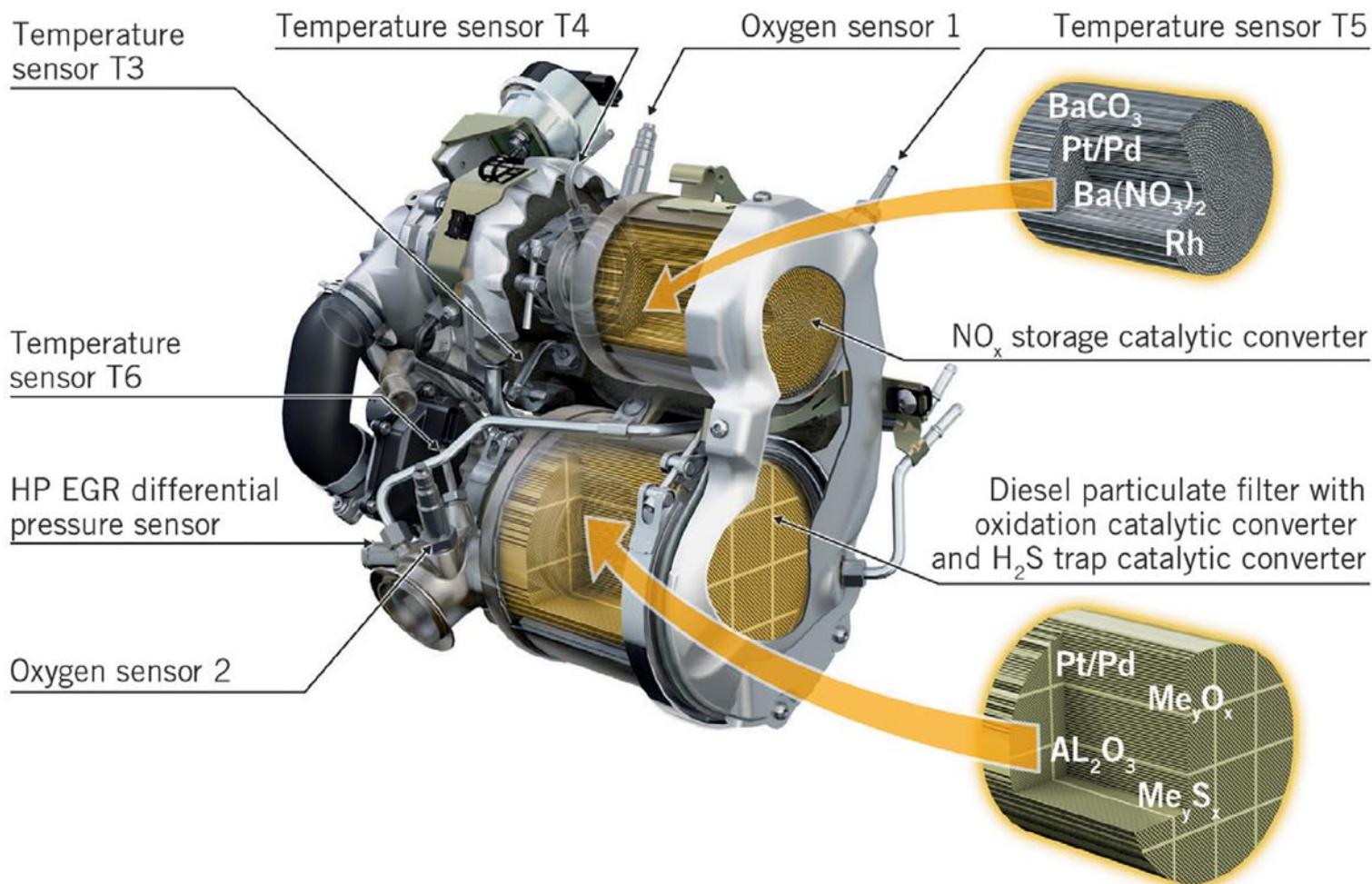
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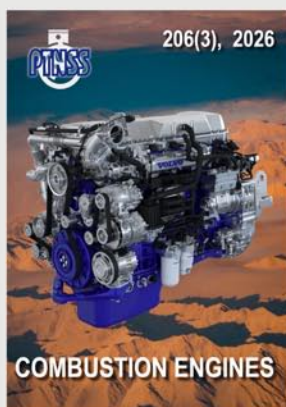
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