IOSUD – "DUNĂREA DE JOS" UNIVERSITY OF GALATI
Doctoral School of Mechanical and Industrial Engineering

ABSTRACT OF THE DOCTORAL THESIS

ALTERNATIVE FUELS IN INTERNAL COMBUSTION ENGINES

Phd student
Chivu Robert-Mădălin

Scientific Supervisor,
Prof. Dr. Ing. Florin Popescu
Prof univ.dr.ing. Jorge Martins

Series I 6 No. 79
GALAȚI
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With gratitude, eng. *Robert-Mădălin Chivu*
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Content

Introduction

a. Background and justification

In the contemporary era, pressures on traditional energy systems and growing concerns about climate change require an essential reassessment of the energy paradigm. The field of transport, especially that of internal combustion vehicles, is at the heart of this imminent revolution, with the imperative need to adopt sustainable and energy-efficient solutions. This thesis, dedicated to the study "Alternative Fuels in Internal Combustion Engines," represents a firm commitment to exploring and understanding the complexity of this crucial transition. Historically, fossil fuels have been the backbone of vehicle propulsion, but the recognition of the ecological consequences of this dependence has led to an intensified search for alternative and sustainable energy sources. The present thesis engages in a systematic and detailed effort to evaluate and evaluate the use of alternative fuels in internal combustion engines. This choice of focus stems from the belief that internal combustion engines remain key elements in the transport and powertrain landscape, and the adoption of alternative fuels in these contexts involves a complex range of technological, economic and environmental considerations. The primary objective of the thesis is to investigate how alternative fuels influence the performance of internal combustion engines, encompassing aspects such as energy efficiency, pollution emissions and adaptability to existing infrastructures.

Research objectives:

Technical Performance Analysis: Detailed investigation of how alternative fuels affect the technical performance of internal combustion engines, including aspects such as energy efficiency, engine power and rolling characteristics.

Emissions Assessment: With a focus on environmental sustainability, we will assess the impact of the use of alternative fuels on emissions from the combustion process in internal combustion engines, exploring options that minimise air pollution.

Through a rigorous and interdisciplinary approach to these objectives, this research aims to provide not only a deeper understanding of the use of alternative fuels in the context of internal combustion engines, but also to provide practical guidance for the automotive industry and for the development of public policies towards sustainable mobility. By achieving these goals, we want to contribute to a comprehensive and enforceable framework that encourages the efficient and sustainable transition to alternative fuels in the transport sector.

c. Relevance and potential impact of the study

The introduction of this doctoral thesis dedicated to the use of alternative fuels in internal combustion engines is outlined in the light of an imperative need in the current context of climate change and persistent dependence on fossil resources. This research aims to make a significant contribution to sustainable development by renewing and redefining the energy paradigm within automotive powertrains. Through exhaustive investigations and interdisciplinary analyses, it aims to shed light on the complexity involved in the efficient integration of alternative fuels and to outline the path towards future mobility with low environmental impact.

Relevance of the study:

In a global landscape characterized by increasing environmental pressures and an imperative need to limit greenhouse gas emissions, the investigation into the use of alternative fuels in internal combustion engines becomes a fundamental pillar in addressing climate change and in moving towards a sustainable future. Indeed, as society faces the dire consequences of massive dependence on fossil fuels, from global warming to the depletion of natural resources, imposing an urgent shift towards sustainable and innovative energy sources becomes not only necessary, but imperative.

Potential Impact of the Study:

Through the ambitious approach to address the use of alternative fuels in internal combustion engines, this research aims to exert a significant and tangible influence in multiple critical areas.

Ecological Perspective: First of all, from an environmental perspective, it is anticipated that the results obtained will be a central pillar in efforts to reduce the carbon footprint associated with car transport. Identifying and promoting the most efficient and sustainable fuel sources has the potential to fundamentally reshape the way cars contribute to climate change.

Social Impact: The social impact of the transition to alternative fuels is a dimension with particularly profound implications. Understanding how consumers react to these technological innovations and their process of adapting to new mobility paradigms are crucial.

In conclusion, the relevance and potential impact of this research transcend academic boundaries, presenting itself as a key pillar in steering towards sustainable mobility and facilitating a responsible use of energy resources. This thesis not only theoretically explores alternative fuels, but is essentially a catalyst for tangible change in the direction of a sustainable transformation of the automotive industry and our mobility model.

Chapter 1. Literature review

1.1 Internal combustion engines

The piston internal combustion engine is a thermal engineering device that converts thermal energy from the fuel combustion process into mechanical energy.

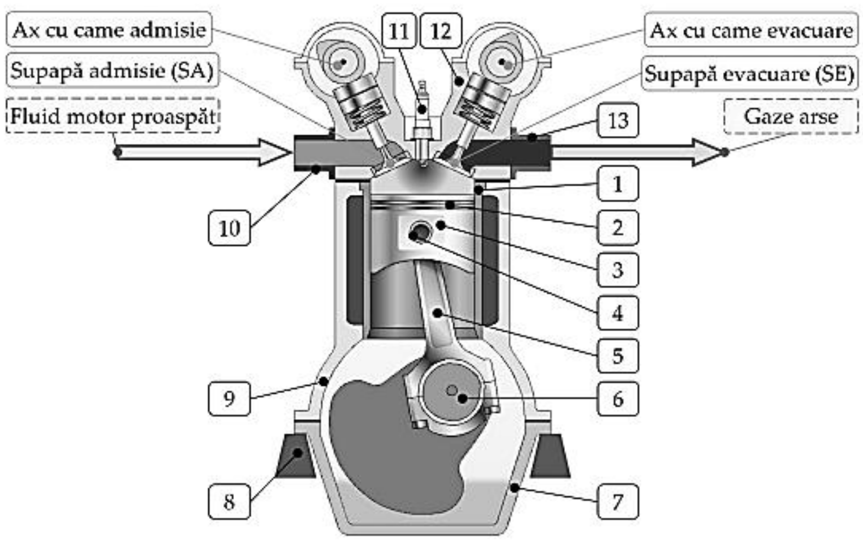


Figure 1 Diagram of an internal combustion engine, adapted from [2]

1.2 Fuels used in internal combustion engines

In internal combustion engines, a variety of conventional fuels are used to generate the heat energy needed for propulsion. These fuels are selected based on their combustion characteristics, availability, costs and environmental impact. In general, conventional fuels can be classified into the following main categories:

- Petrol
- Diesel (diesel
- Natural gas: Natural gas is mainly made up of methane and other gaseous hydrocarbons and is used as a fuel in certain types of engines, such as compressed natural gas (CNG) or liquefied petroleum gas (LPG) engines. It is considered a cleaner fuel than gasoline or diesel because it produces fewer carbon dioxide and particulate emissions during combustion.
- Propane and Butane

1.3 Approaches to the use of biofuels in internal combustion engines

The next chapter examines approaches to the use of biofuels in internal combustion engines, focusing on the evolution and beginnings of the use of these sustainable fuels. Biofuels are a key component in efforts to transition to more sustainable and environmentally friendly energy sources. Throughout history, the use of biofuels in internal combustion engines has been influenced by numerous factors, including technological developments, political and economic changes, and environmental concerns. The beginnings of the use of biofuels in internal combustion engines can be traced back to the context of the first Industrial Revolution, when the use of oil and natural gas in steam engines had a significant impact on the economy and society [1], [2].

1.4 Cost of Biofuels

The cost of biofuel production, compared to diesel, is a topic of interest and debate in the energy field. Biofuels, which come from organic sources such as agricultural crops, algae or waste, are a renewable alternative to fossil fuels such as diesel. However, the complex processes involved in their extraction, conversion and refinement often bring higher production costs than those associated with traditional diesel . Several factors influence the cost of production of biofuels, including the types of feedstocks used, the technologies applied for conversion and the scale of production. For example, biofuels from food crops, such as maize or soybeans, can be affected by price fluctuations in food markets, which can lead to significant variations in total production costs. In contrast, advanced biofuels, produced from non-food sources such as algae or agricultural residues, offer more sustainable alternatives, but may involve higher processing costs due to their less mature technologies and smaller economies of scale. Diesel production, on the other hand, mainly involves extraction and refining processes applied to crude oil, which benefit from a well-established infrastructure and economies of scale. However, the cost of production of diesel is susceptible to fluctuations in crude oil prices and geopolitical factors, making it vulnerable to market volatility [3], [4] [5], [6] ,[7] ,[8], [9].

1.5 Introduction of biofuels into internal combustion engines

Blending biofuel with conventional fuel: This method involves blending biofuel, such as biodiesel (for compression-ignition engines) or bioethanol (for spark-ignition engines), with conventional fuel, such as diesel or gasoline. Mixing can be done in varying proportions, depending on specific engine requirements and fuel compatibility. By using this technique, biofuel can be gradually integrated into existing fuel systems without requiring significant changes to the engine or fuel distribution infrastructure. However, it is important to consider the effects on engine performance and emissions, as well as the compatibility of materials with the fuel mixture [10], [11], [12], [13], [14], [15].

There are several types of biofuels, and they can be classified according to the raw material from which they are obtained and the technological process used to produce them. In general, biofuels can be divided into three main categories:

- Liquid biofuels
- Solid biofuels
- Gaseous biofuels

1.6 Fuel atomization

Fuel fragmentation in the combustion chamber is a crucial aspect of the combustion process in a variety of applications, including thermal power plants, internal combustion engines, industrial furnaces, and more. Atomization converts fuel, whether liquid or solid, into fine droplets or small particles, thus increasing the total surface area of the fuel exposed to the oxidizing medium, usually air.

1.7 Impact of biofuel and diesel blends on compression-ignition engine performance.

The researchers examined the engine's performance with mixtures of jatropha oil and mineral turpentine, combined with diesel. Results Figure *2* shows that the JMT50 mixture has an efficiency close to diesel, but the lower calorific value of jatropha oil leads to lower thermal efficiency and an increase in specific fuel consumption. Although oxygen improves burning, the viscosity and volatility of jotropha oil contribute to weaker combustion characteristics [16].

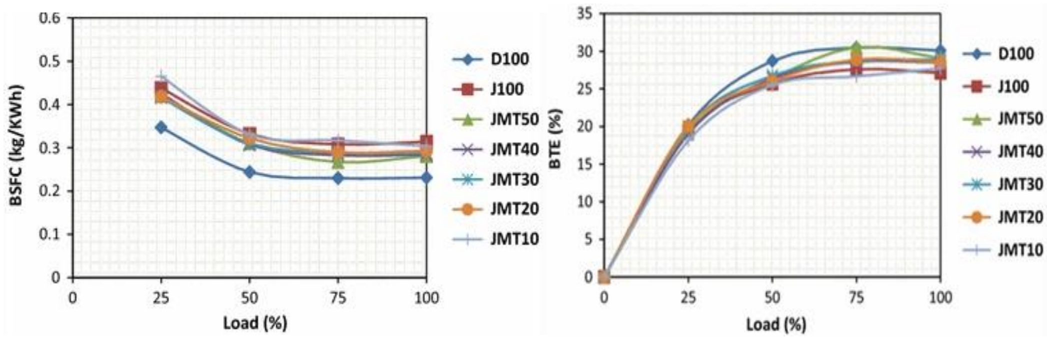


Figure 2 Performance parameters, adapted according to [16]

In another experiment, researchers evaluated the performance of a four-stroke single-cylinder engine using a mixture of jatropha biodiesel and turpentine oil. They tested five different blends, ranging from 10% to 50% turpentine oil, with the rest being jatropha biodiesel. The engine ran at a constant speed of 1500 RPM, and the mixtures were injected at a pressure of 200 bar, the injection pressure being relatively low for a new generation engine. The results showed that BSFC increased with the increase in the proportion of jatropha oil in the mixture, due to the lower calorific value compared to the reference fuel [17]. Performance parameters such as BSFC, BTE, cylinder pressure, and ignition delay were measured and analyzed. According to the authors' conclusions, it is found that BSFC recorded lower values, up to 75% of the engine load Figure *3* a), and BTE has been improved in the case of the use of fuel mixtures compared to standard diesel, in the same load range. However, the thermal efficiency decreased after the engine load exceeded 75%, due to the detonation phenomenon. Also, the volumetric efficiency was lower than the reference fuel because turpentine, a volatile compound, was introduced into the intake manifold

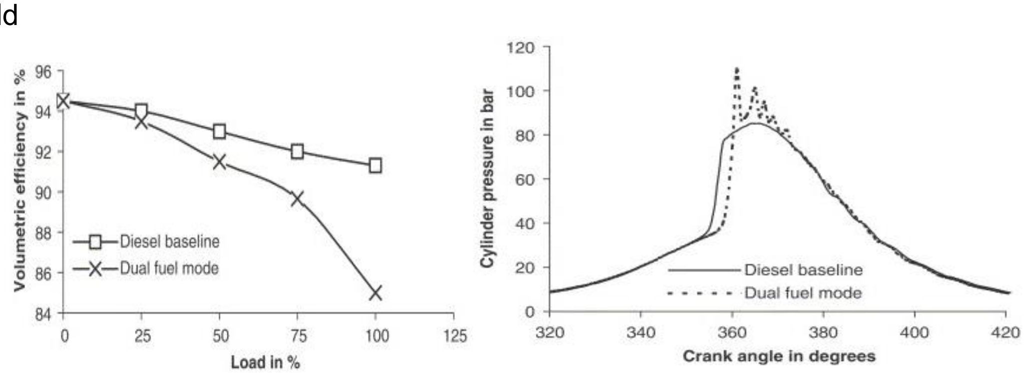
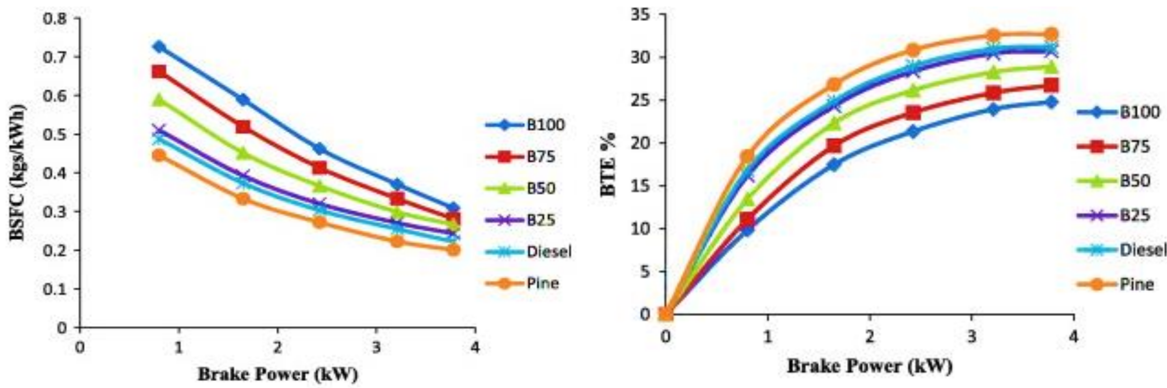


Figure 3 Volumetric efficiency and pressure in the cylinder, adapted after [62]

a) b)

In another study, experiments were conducted to evaluate mixtures of turpentine and Chlorella Vulgaris as biofuels. Various fuel combinations have been worked out, but the C50T50 blend has been found to have the best performance. According to the authors' findings, the C50T50 mixture demonstrated a faster acceleration in the heat release rate (HRR) compared to conventional fuel [18], [19], [20], [21].



(a) (b)

Figure 4 Specific fuel consumption and thermal efficiency, adapted according to [22]

Compared to diesel and other fuels tested by the authors, pine oil has a more complex molecular structure and different volatility, which allows for more complete and efficient burning. Thus, during combustion, the heat energy released by pine oil Figure *5* It is converted into a larger volume of gases that exert pressure on the piston in the cylinder.

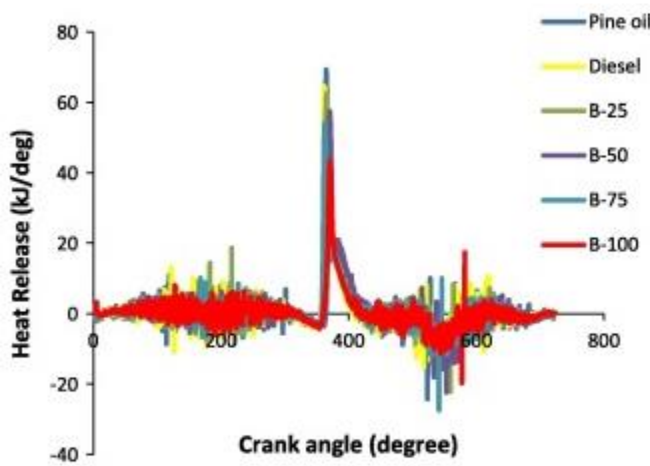


Figure 5 Heat release rate, adapted according to [22]

This increase in pressure in the cylinder during firing with pine oil can be observed in the various stages of the engine's operating cycle. From the initial ignition point to the maximum compression point, the pressure in the cylinder is monitored to assess combustion efficiency. In the case of pine oil, this pressure is maintained at optimal levels, indicating an even and complete combustion of the fuel.

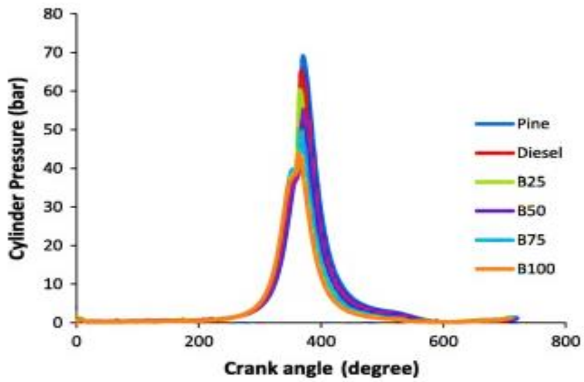


Figure 6 Cylinder pressure, adapted to [22]

Through the transesterification process, biodiesel was produced from eucalyptus oil, and the resulting mixture, together with diesel, was subjected to an experimental study using a single-cylinder direct injection diesel engine. The observations demonstrated that under all loading conditions, the specific fuel consumption (BSFC) and brake thermal efficiency (BTE) for the eucalyptus biodiesel blend exceeded the performance of conventional diesel [23].

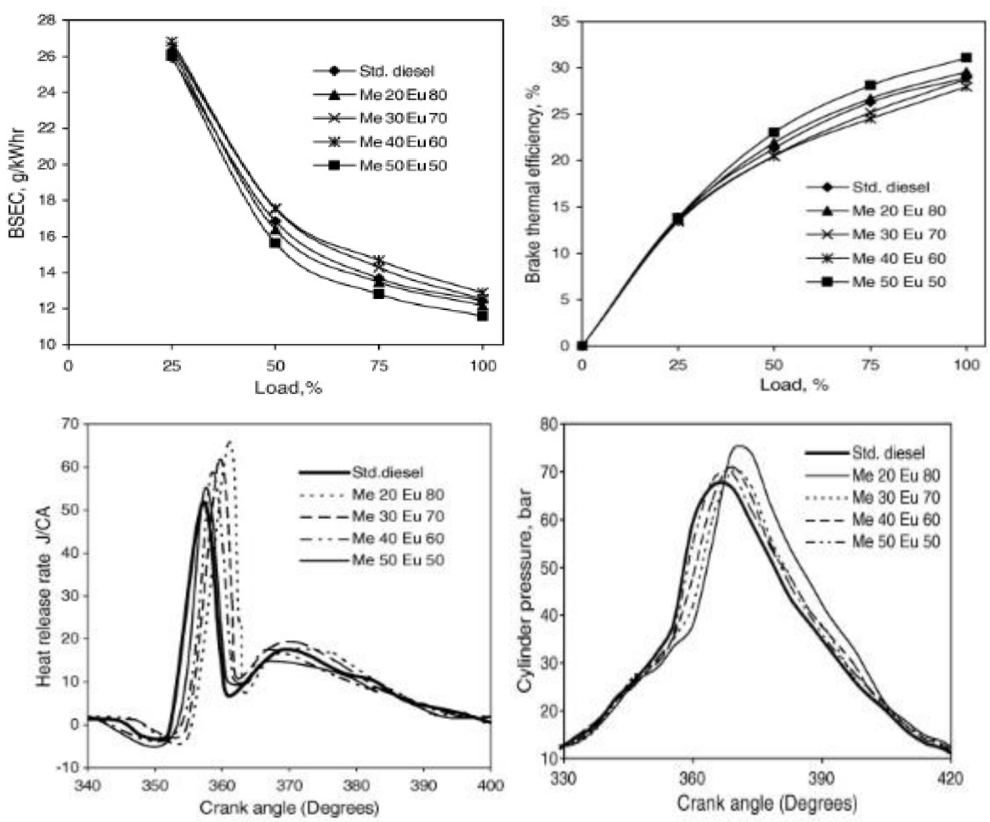


Figure 7 Performance of the engine powered by eucalyptus oil mixtures, adapted according to [24]

Using various proportions of essential oil blends, including lemon peel oil, and adding diethyl ether (DEE), the researchers investigated the variation in brake thermal efficiency (BTE) as a function of engine load. They found that BTE [25] Figure *8* is improved by 3.7% for LPO20 DEE10 blend (20% LPO, 70% diesel and 10% diethyl ether) compared to LPO20 (20% LPO and 70% diesel) at full load but nevertheless does not exceed thermal efficiency when using diesel.

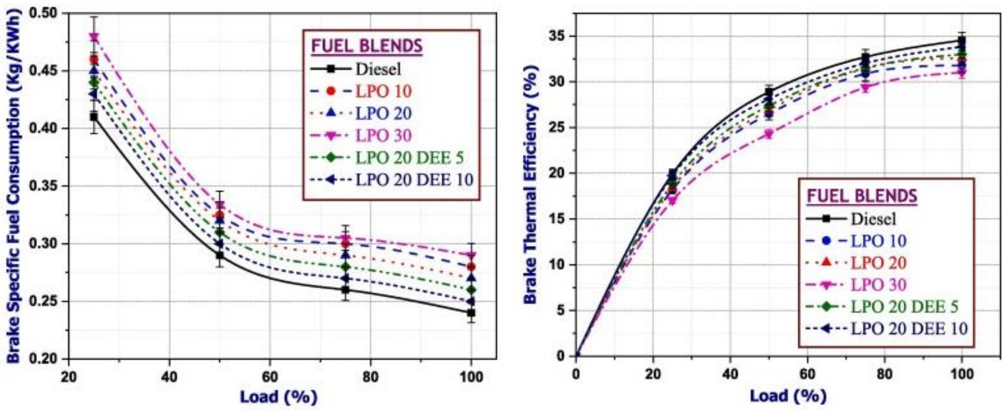


Figure 8 Performance indicators for essential oil, adapted according to [25]

Pressure in the cylinder Figure *9* increases significantly for fuels when additived with DDE, and reduces ignition delay period Figure *9* compared to non-DDE-additive fuel, due to the low viscosity of lemon peel oil, which promotes evaporation and atomization of the fuel, thus contributing to improved combustion.

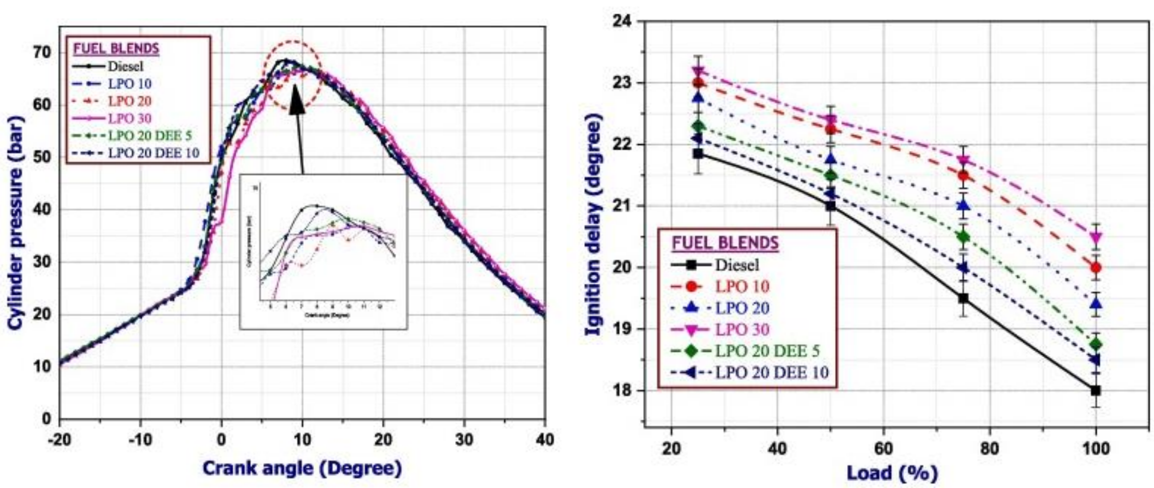


Figure 9 Ignition delay and cylinder pressure, adapted after [25]

1.8 Impact of biofuels on pollutant emissions

The study carried out on mixtures of turpentine and jatropha oil, in particular JMT20 and J100, provides valuable information on their impact on emissions of pollutants, in particular nitrogen oxides (NOx) [16] Figure *10*. The research presents a detailed analysis of these emissions according to different engine load conditions and mixture compositions. A significant finding of the study is the considerable reduction in NOx emissions. This reduction is attributed to the additional oxygen present in the mixtures, which leads to improved combustion efficiency. However, it is worth mentioning that NOx emissions tend to increase with a higher proportion of jatropha oil in the mixture, highlighting the complexity of the relationship between the composition of the mixture and emissions.

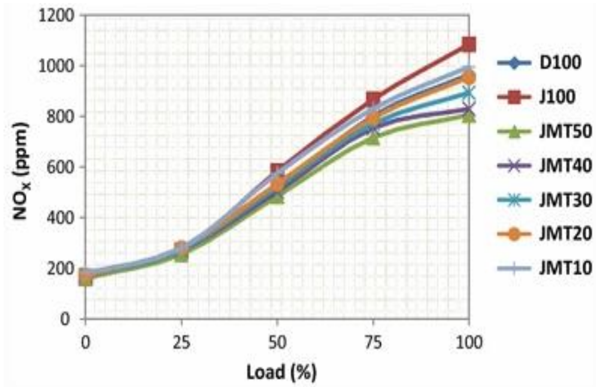


Figure 10 Emissions of nitrogen oxides [16]

The study investigates the emissions of a dual-powered engine, using a mixture of turpentine and diesel, under various charging conditions. The results reveal a gradual increase in CO emissions from 0% to 75% load [26] Figure *11*, attributed to a higher rate of fumigation and a lack of oxygen. At higher loads, the extinguishing of the flame and the formation of cooled layers in the vicinity of the combustion chamber walls lead to a 35% increase in CO emissions at maximum engine load.

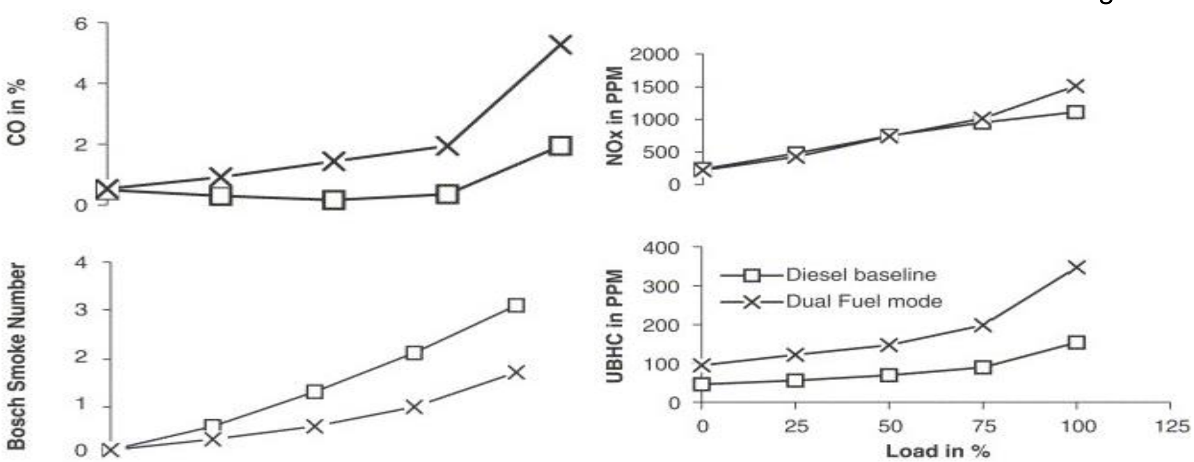


Figure 11 Pollutant emissions emitted by the dual-fuel engine [26]

In another study, researchers measure the pollutant emissions of a diesel engine that is powered by pine oil. The authors report that NOx emissions [27] Figure *12* increase with the increase in the proportion of pine oil in the fuel mixture, due to the prolongation of the ignition delay, which allows a more efficient mixing of fuel and air, thus increasing the combustion temperature.

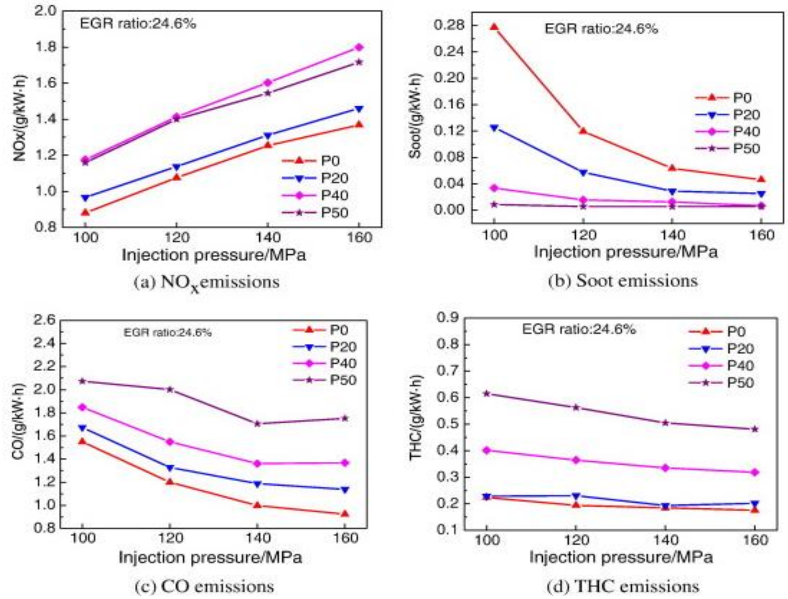
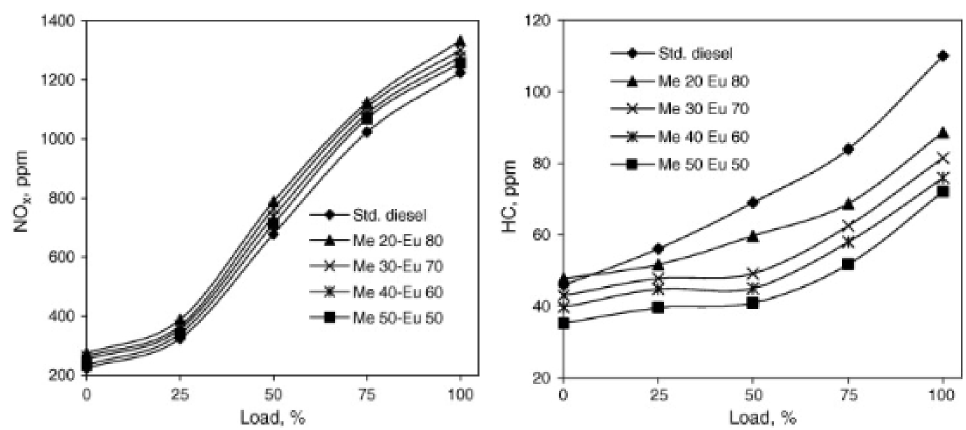


Figure 12 Pollutant emissions according to injection pressure [27]

Regarding pollutant emissions when using essential oils as fuel, the authors present the experimental data clearly and provide relevant interpretations of the results. A notable aspect of the study is the investigation of NOx emissions in the case of Me-Eu (methyl ester-eucalyptus oil) mixtures. The authors report an increase in NOx emissions [28] Figure *13*, probably due to the presence of oxygen in both components of the mixtures. This observation is consistent with previous research suggesting that oxygenated mixtures can lead to an increase in NOx emissions. The authors explain this tendency by complete combustion, which generates higher combustion temperatures, thus favoring the formation of NOx.



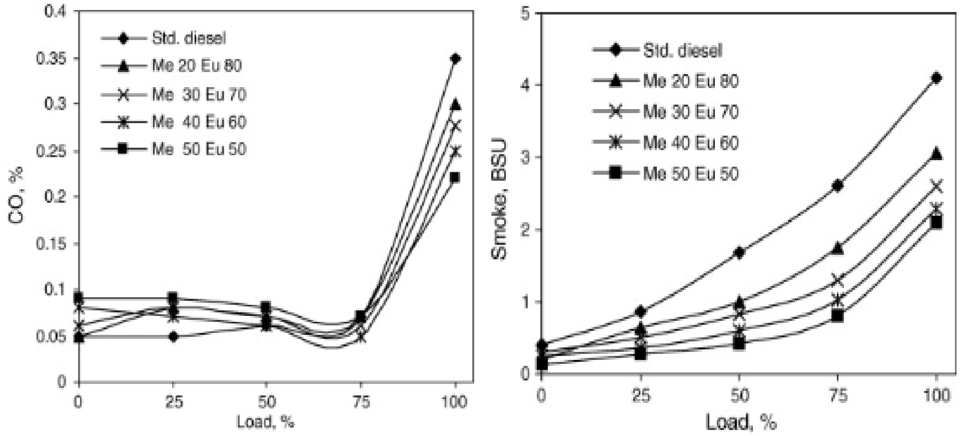


Figure 13 Pollutant emissions of the engine powered by eucalyptus essential oil and methyl ester [28]

1.9 Two-stroke naval engines

Two-stroke marine engines are a crucial element in the field of marine propulsion, being recognized for their superior efficiency and reliability. These engines operate on the basis of a simplified combustion cycle, completing a complete thermodynamic cycle in just two piston strokes: one for compressing the air-fuel mixture and another for exhausting the combustion gases Figure *14*.

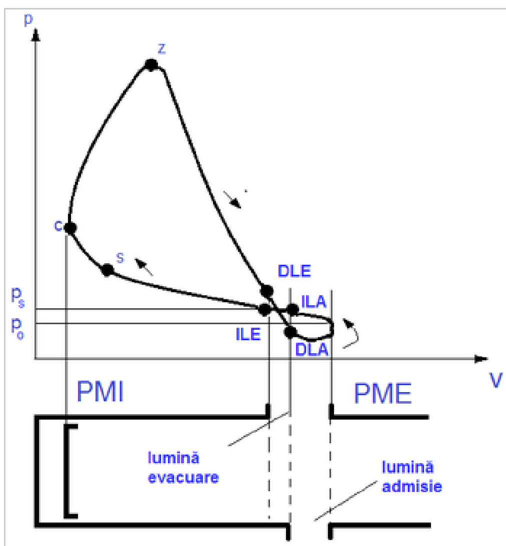


Figure 14 Two-stroke naval engine thermodynamic cycle [29]

This differs from four-stroke engines, which require four strokes to complete a full cycle Figure *15*.

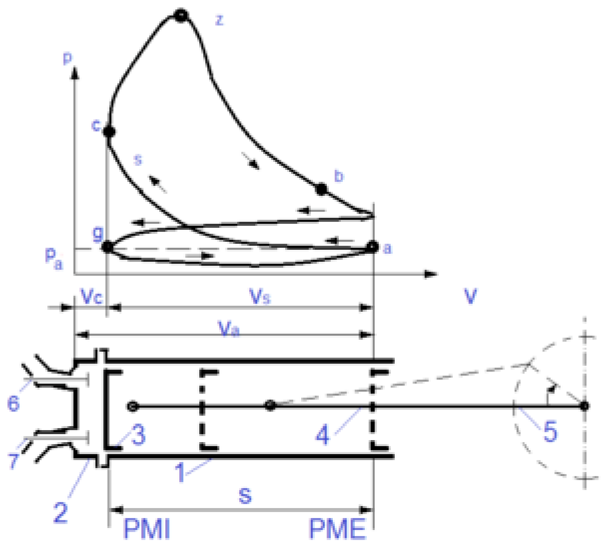


Figure 15 Four-stroke engine thermodynamic cycle [30]

1.10 Long-term use of biofuels

The long-term use of biofuels in diesel engines raises numerous questions and challenges, which require detailed analysis and a deep understanding of the technical, economic and environmental implications. In this chapter, we aim to examine several aspects, exploring the advantages and limitations of the long-term use of biofuels in the context of diesel engines, as well as their prospects for sustainability and energy efficiency. The literature illustrates a wide range of research relevant to the integration of biofuels into compression-ignition engines. These investigations raise crucial issues that need to be taken into account when assessing the long-term use of biofuels in these engines. The studies highlight both the potential advantages of biofuels and the limitations and challenges associated with their integration into diesel technology. Oxygenated biofuels achieve better characteristics in terms of engine performance and pollutant emissions compared to fossil fuels, according to experimental tests. However, in long-term use, it has been observed that they can cause carbon deposits on the metal surfaces of the engine. The authors conducted extensive tests on small (automobile) and large (agricultural machinery) engines in different operating regimes [31] Figure *16*.

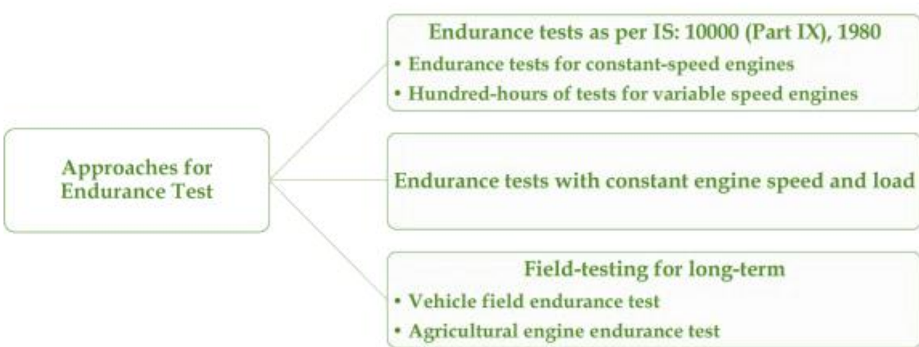


Figure 16 Biofuel testing conditions [31]

Chapter 2. Research Methodology. Experimental part

Experimental research into the use of biofuels in internal combustion engines is a key aspect in the development of sustainable and efficient solutions for the energy sector. This method of investigation provides a detailed and practical understanding of the behavior of biofuels under real operating conditions, allowing rigorous analysis of the effects on engine performance, combustion efficiency and pollutant emissions. First, experimental research is crucial for the theoretical validation and simulation of the models predicting the behavior of biofuels that is studied in Chapter 3.

2.1 Experimental stand

The engine used for fuel testing was a 1.6L PSA HDI diesel Figure 17, four-cylinder, four-stroke, water-cooled, turbocharged and intercooled. Changes to the engine included adjusting the turbocharger pressure and deactivating the EGR system, carried out to minimise parameter variability and allow for accurate analysis of the effects of fuel mixtures on engine performance.

Backgammon 1 Technical specifications of the internal combustion engine

Engine	1.6 HDi (75hp)
Cylindrical capacity	1560 *cm3*
Number of cylinders	4
Power	55 kW (75 hp) @ 4000
Couple	185 N.m @ 1750 rpm
Fuel	Diesel
Type of injection	Common-Rail
Bore diameter x piston stroke	75.0 x 88.3 mm
Compression ratio	18:1
Overeating	Turbo (Mitsubishi 49173 – 07503)
Number of valves	16
Engine layout	Transverse
Traction	Front Axle (FWD)

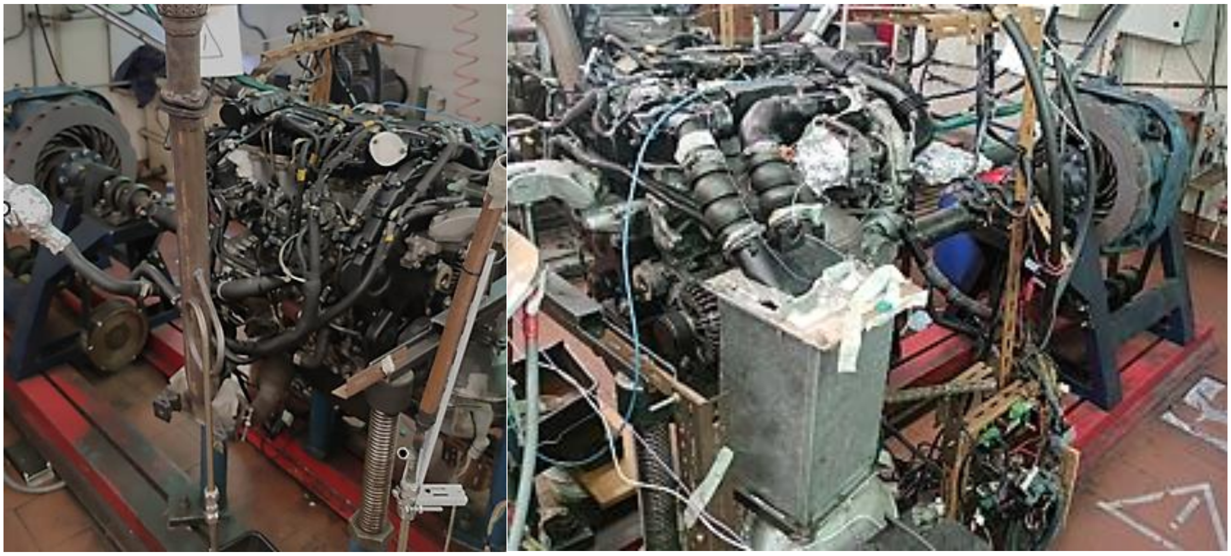


Figure 17 Experimental stand

2.2 Sensors and actuators

Sensors are devices or modules that detect and measure a variety of physical or chemical conditions in the environment and convert them into electronic signals that can be analyzed, recorded, or used to control other devices Figure *18*. They play a crucial role in many applications, from industrial systems and medical equipment to vehicles.

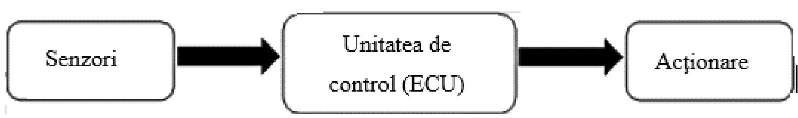


Figure 18 Operation of a control unit

There are several types of sensors used to measure crankshaft speed, each with its own specific operating principle:

1. Inductive sensors (Figure *19*)

   Advantages: Inductive sensors are durable, have a relatively low cost, and operate without an external power supply. Disadvantages: Their sensitivity can be influenced by the speed of rotation of the crankshaft and temperature variations.

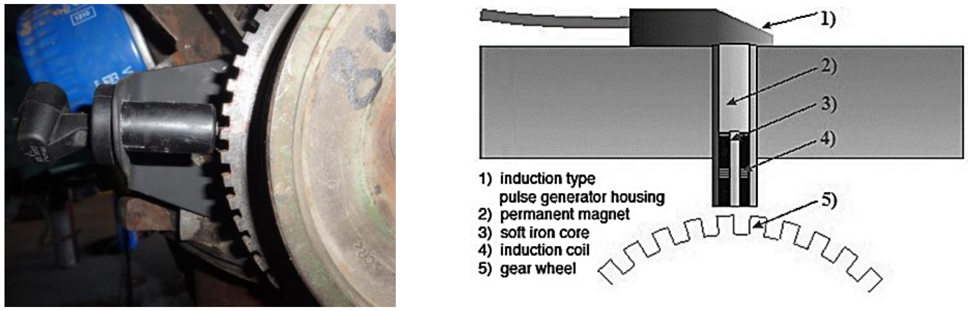


Figure 19 Inductive sensor [32]

2. Hall sensors (Hall effect)

Advantages: Hall sensors offer high accuracy and are able to operate at low speeds and under varying temperature conditions. Disadvantages: They require an external power supply and can be more expensive compared to inductive sensors.

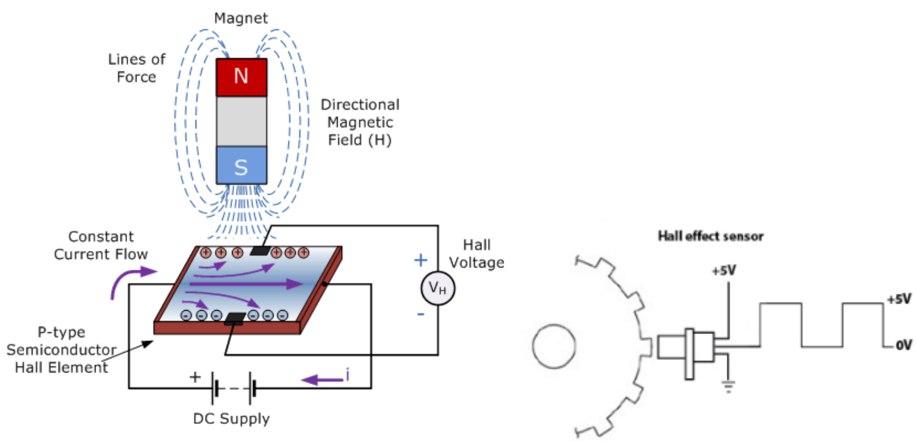


Figure 20 Hall Sensor Working Principle [33]

3. Optical sensors Advantages: Optical sensors offer very high resolution and fast response. Disadvantages: They can be sensitive to dirt and dust, requiring clean operating conditions and regular maintenance. For channels A, B, and Z, the output signals provide the following information: Channels A and B provide information about the position of the crankshaft. Depending on the desired resolution, the number of opaque and transparent sectors that can vary depending on the desired precision . The Z channel includes a single transparent sector, which emits a pulse with each complete rotation of the encoder, indicating the position of the upper dead center (TDC).

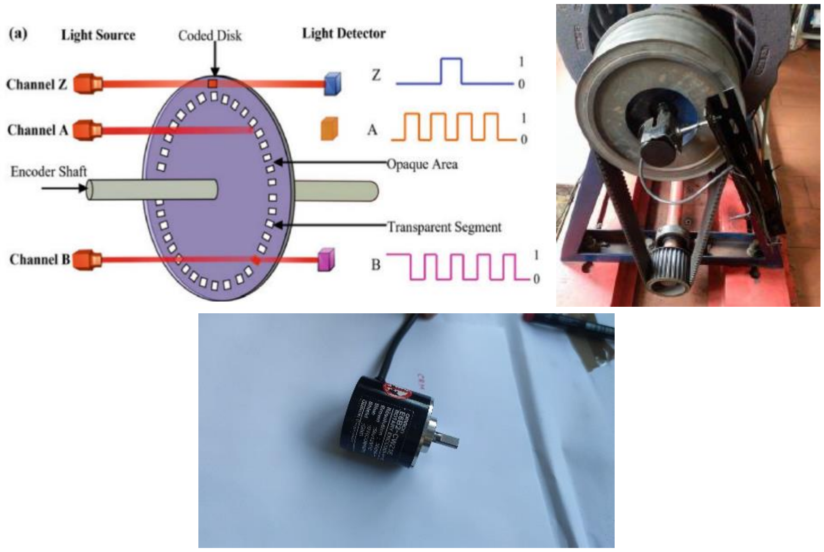


Figure 21 Principle of operation of the position encoder [34]

2.2.2 Fluid flow measurement sensor located on the intake manifold

Advantages: It provides an accurate and fast measurement of airflow, being sensitive to rapid changes in airflow.

Disadvantages: Sensitive to contamination and requires periodic cleaning to maintain the accuracy of measurements [35].

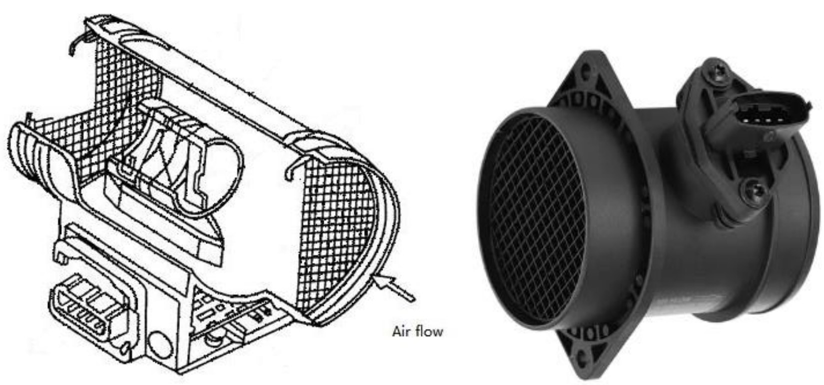


Figure 22 Working Principle of MAF Sensor

In modern diesel engines, the MAF sensor is essential for optimal and efficient engine operation.

2.2.3 Absolute pressure measurement sensor in the intake manifold

Advantages: High accuracy and stability, fast response time. Disadvantages: Temperature sensitivity and the need for thermal compensation. Capacitive element Principle: This type of sensor uses a variable capacitor consisting of two parallel plates, one of which is fixed and the other is mounted on a flexible diaphragm. When the pressure varies, the distance between the plates changes, changing the electrical capacitance. Changes in capacitance are converted into an electrical signal proportional to the pressure. Advantages: Long-term durability and stability, low sensitivity to temperature variations. Disadvantages: Complexity and higher manufacturing costs.

Figure 23 Intake manifold pressure measurement sensor

2.2.4 Temperature sensors

Sensor for measuring coolant temperature for engine management.

Engine coolant temperature sensor Figure *24* It is an essential device for monitoring and thermal management of the internal combustion engine. This sensor, ECT (Engine Coolant Temperature), measures the temperature of the coolant circulating in the engine's cooling system and provides this data to the electronic control unit (ECU).

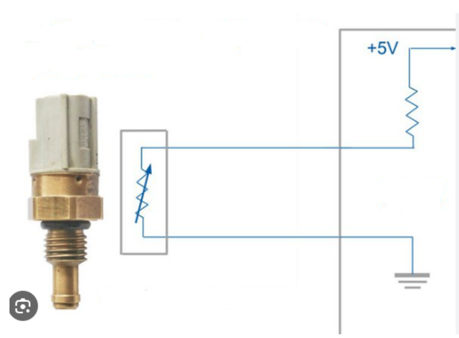


Figure 24 Sensor for temperature monitoring

Temperature sensor for data acquisition board (K-type thermocouples)

K-type thermocouples Figure *26* It is a temperature measuring device, recognized for its versatility and robustness in various industrial and research applications. These thermocouples are made of two different materials, chromel (an alloy of nickel and chromium) and alumel (an alloy of nickel and aluminum), which are joined at one end to form the measuring junction Figure *25*.

Figure 25 Type K Thermocouple Junction [36]

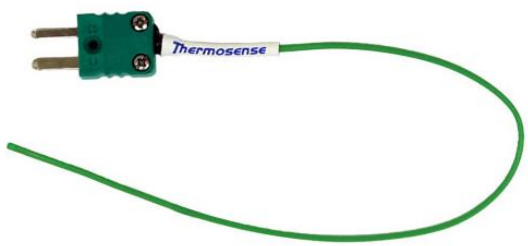


Figure 26 K-type thermocouple [36]

- The temperature monitoring in the laboratory was measured and recorded with a device called a thermohygrobarometer Figure *27* , capable of measuring temperature, humidity, and atmospheric pressure [37].

Figure 27 Thermohygrobarometer

2.2.5 Sensor for measuring the engine torque type S-cell

S-cell sensor Figure *28* It is a device used to measure engine torque. This name comes from its characteristic shape, which resembles the letter 'S'. S-load cells are frequently used for the measurement of tensile and compressive forces in various industrial and test applications.

Figure 28 S-cell sensor

2.2.6 Pressure sensor in the common fuel rail

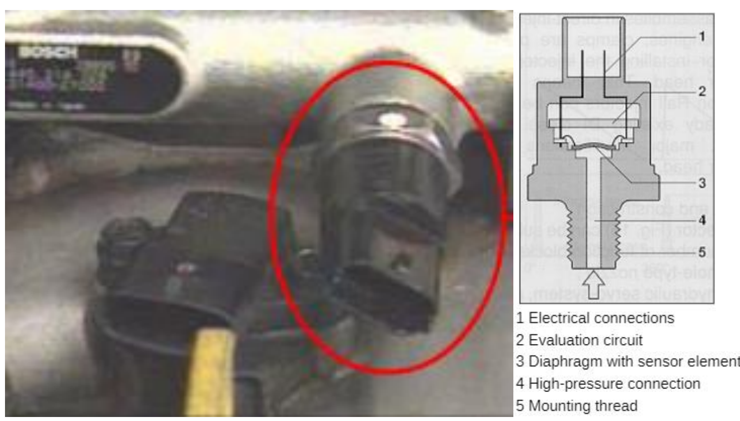


Figure 29 High Fuel Pressure Sensor [32]

2.2.7 Accelerator Pedal Sensor

Electronic accelerator pedal Figure *30*, is a crucial component in modern diesel engine load management. Unlike traditional wired systems, the electronic accelerator pedal works via electrical signals, transmitted from the pedal to the engine's Electronic Control Unit (ECU). The electronic accelerator pedal is equipped with sensors that monitor its position.

Figure 30 Accelerator Pedal

The accelerator pedal signal was monitored using the data acquisition system to determine the engine load. For example, when the accelerator pedal voltage is 1.39V Figure *31*, the engine has a load of 37.3%. If the accelerator pedal voltage signal is 1.01V Figure *31*, corresponds to an engine load of 27.1 %.

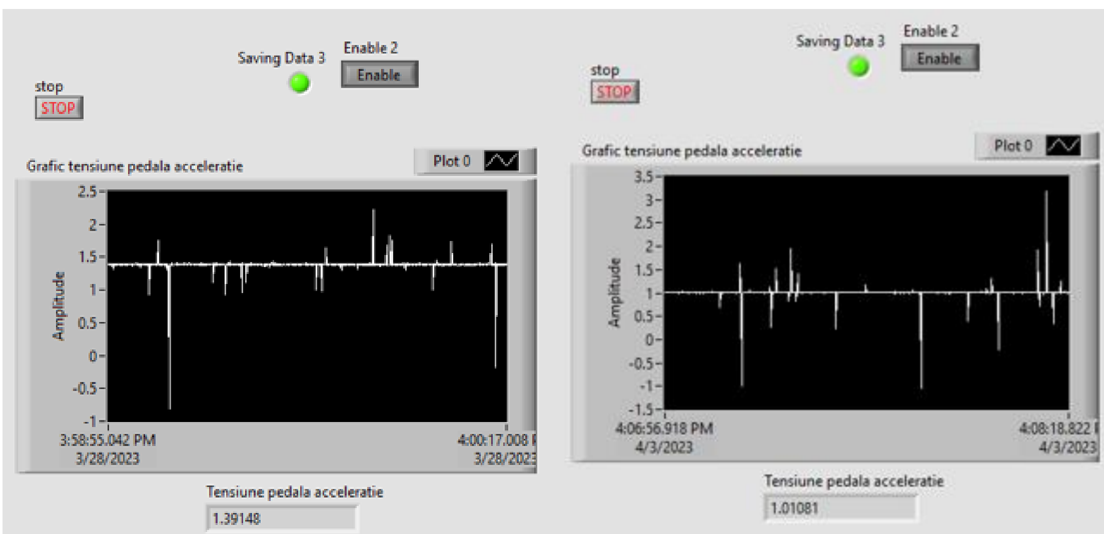


Figure 31 Accelerator Pedal Signal

2.2.8 Actuators (fuel injector)

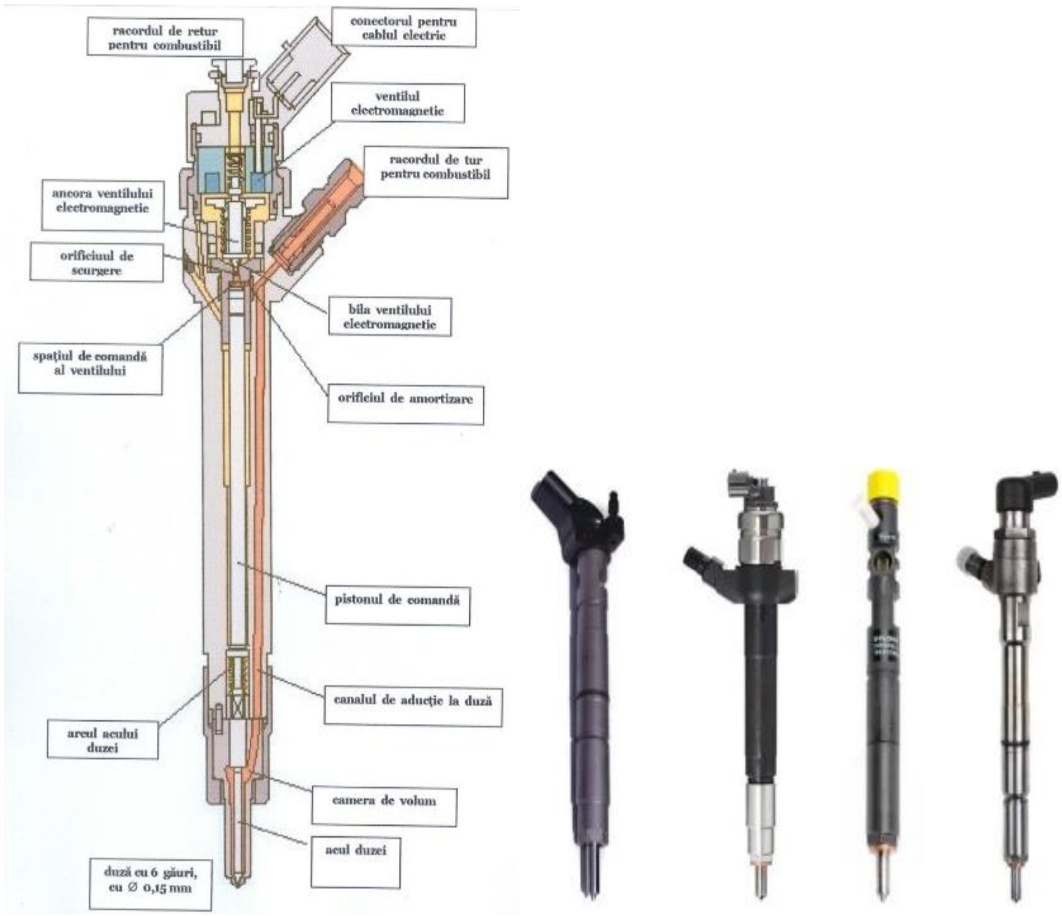


Figure 32 Fuel injector [38]

Converting electrical signal into motion: The fuel injector is controlled by the ECU, which sends electrical signals to the injector Figure *33*. These signals cause the injector needle to open and close, allowing fuel to be injected into the combustion chamber.

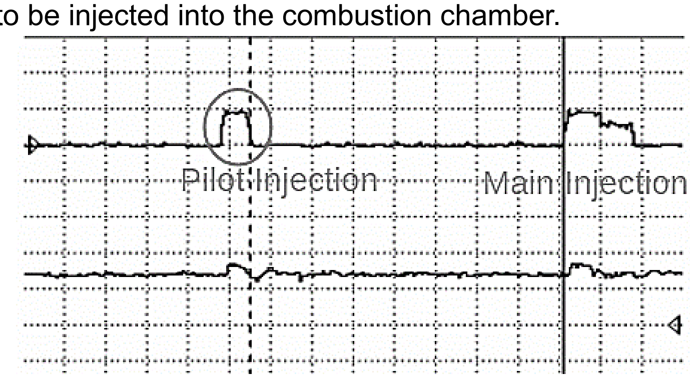
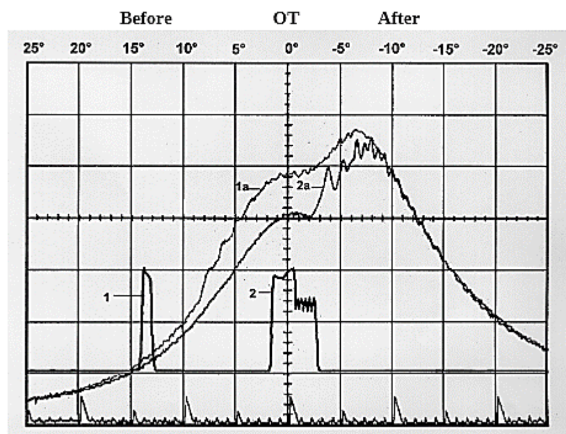


Figure 33 Fuel injector electrical signal

Fuel flow control: The signals from the ECU regulate the duration and timing of fuel injection as well as the injection strategy, thus influencing the flow of fuel injected. In Figure *34* A fuel injection strategy is presented, namely pre-injection and main fuel injection, as well as the evolution of the pressure in the combustion chamber when using pre-injection. It is observed that when pre-injection is used, the pressure in the combustion chamber is increased due to the atomization of the fuel.



1 = Pre-injection 1a = Combustion pressure with pre-injection
2 = Main injection 2a = Combustion pressure without pre-injection

Figure 34 Electrical Injector Signal with Fuel Pre-Injection Strategy [32]

This is essential for optimising engine performance and reducing emissions.

2.3 Supercharging system

Supercharging is a system used to increase the performance of a compression-ignition (diesel) engine by increasing the density of air admitted to the cylinders. This process is usually carried out with the help of a compressor, called a turbocharger or mechanical compressor (supercharger), which compresses the air before it enters the cylinders, thus allowing a greater amount of air to be introduced and, consequently, a greater amount of fuel. This results in more efficient combustion and more engine power.

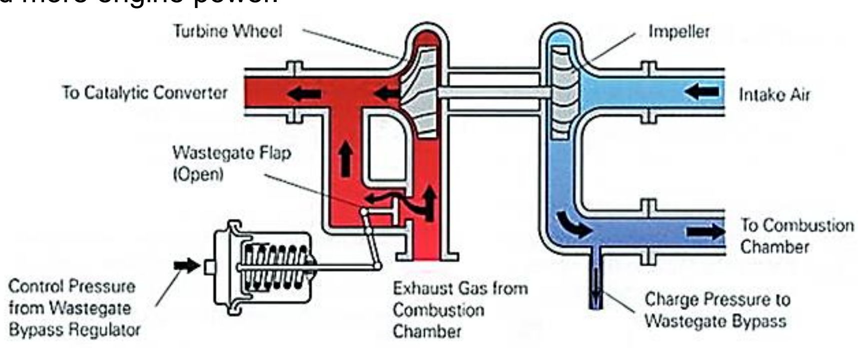


Figure 35 Turbine and compressor operation diagram [39]

- The intercooler Figure *36*: It is a heat exchanger used to cool compressed air before it enters the cylinders. Cooling the air increases its density, contributing to more efficient combustion and increased engine power.

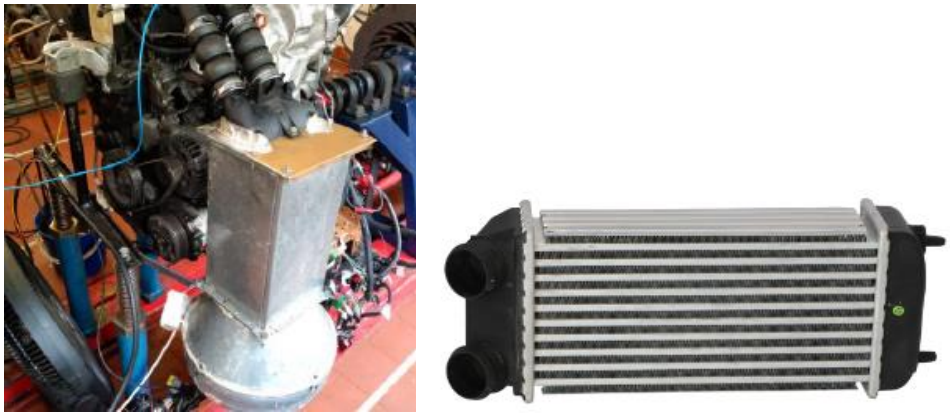


Figure 36 Heat exchanger (intercooler)

- Bypass and decompression valves (waste gate) Figure *37*: Allow exhaust gases to bypass the turbine to prevent it from over-revving Figure *35*, thus controlling the boost pressure. Decompression valves: Releases excess pressure from the intake system to prevent overloading and damage to the engine.

Figure 37 Turbine Decompression Valve

2.4 Common Rail High Pressure Fuel System

High Pump Figure *38* Pressure is an essential component in the injection system of modern diesel engines, playing a crucial role in ensuring their performance and efficiency. It has the role of pressurizing the fuel at extremely high pressures, necessary for precise injection and efficient combustion in the cylinders. The pressure generated by the high-pressure pump can reach up to 2000 bar or even more in some advanced applications, however the experimental stand is equipped with a pump capable of generating a maximum of 1600 bar.

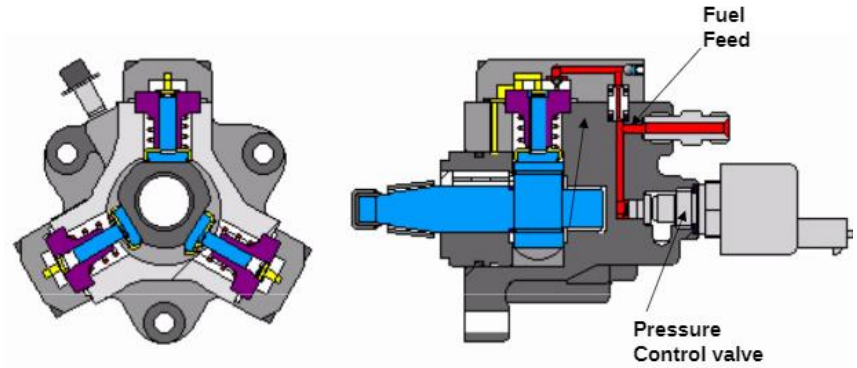


Figure 38 High Pressure Pump [32]

2.5 Control Unit Diagnostics and Programming System (ECU)

Kess v2 System Figure *39* is an advanced electronic control unit (ECU) diagnostic and reprogramming tool used exclusively in the automotive industry. It allows data to be read and written from the ECU, thus facilitating the optimization of vehicle performance, adjustment of engine parameters and diagnosis of errors.

Figure 39 Kess reprogramming system

2.6 Electromechanical brake

Telma electromagnetic brake Figure *40* It is an essential device used in evaluating diesel engine performance, especially for determining engine torque. This technology uses the principles of electromagnetic induction to create a controllable braking force, allowing for accurate and repeatable measurements of the engine performance parameters.

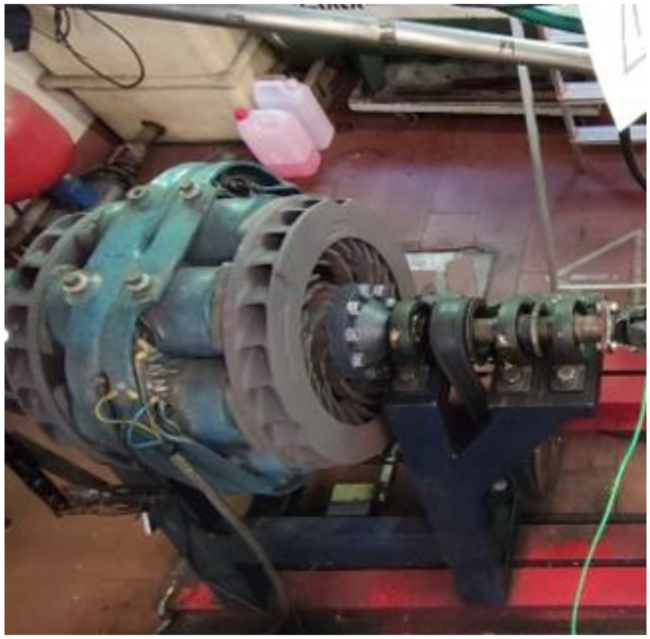


Figure 40 Electromagnetic brake

The device essentially consists of a rotor and a stator. The rotor is attached to the engine shaft Figure *17*, and the stator contains electromagnets.

Figure 41 Voltage Inverter

2.7 Measurement of pollutant emissions using gas analyzer and opacimeter

AVL offers advanced solutions for exhaust gas analysis and smoke opacity measurement, using high-precision equipment such as the gas analyzer used for this research (AVL DiGas 4000 Light) and the opacimeter (AVL DiSmoke) Figure *42*.

Figure 42 AVL Gas Analyzer & Opacimeter

2.8 Data acquisition system

2.8.1 National Instruments Data Acquisition Board

In this research, data acquisition boards from National Instruments were used Figure *43*, recognized for their superior performance and reliability in capturing and processing experimental data.

Figure 43 Procurement Data Board

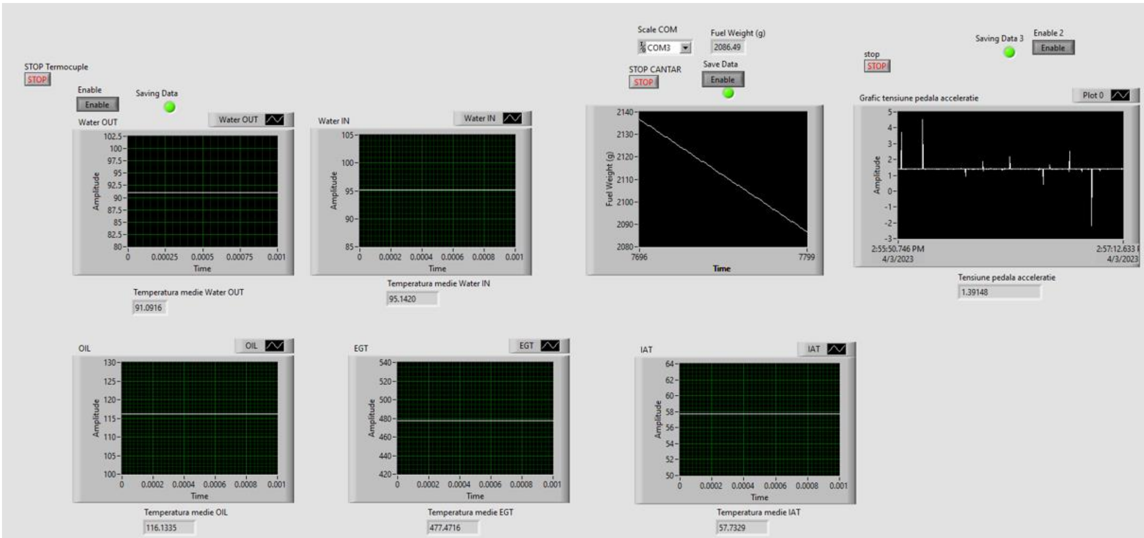


Figure 44 Graphical interface for National Instruments Data Acquisition Boards

2.8.2 Fuel consumption measurement

The experimental stand provides a controlled environment in which all variables influencing fuel consumption can be manipulated and measured accurately. This includes the temperature, pressure, humidity and load conditions of the engine. By isolating and controlling these factors, accurate and replicable data can be obtained, essential for the correct evaluation of engine performance. This level of precision cannot be achieved in real operating conditions, where variables are much more difficult to control.

Accurate measurement of engine fuel consumption at the experimental stand is essential for evaluating the performance and energy efficiency of engines. The use of a Kern precision scale with an accuracy of 0.01g, coupled with a data acquisition system, provides a high level of accuracy and detail necessary for advanced research in the field of propulsion technologies, the technical specifications of the scale can be found in [40].

Figure 45 Fuel consumption measurement

As part of the experiments on the engine stand, a Kern precision scale Figure *45*, capable of measuring masses with an accuracy of 0.01g, has been integrated into the fuel system. This scale, known for its reliability and accuracy, has been coupled to a data acquisition system, allowing continuous monitoring and accurate recording of fuel consumption during engine operation.

This system is configured to record data at very short intervals of time, providing a high temporal resolution. After collecting and storing fuel consumption data using the Kern precision scale and the data acquisition system, it is analysed in detail

2.9 Preparation of fuel mixtures

Mixing proportions between 5% and 30% Figure *46* are sufficient to be compatible with the internal combustion engine without the need for major modifications. This allows for the immediate use of biofuels, facilitating widespread adoption.

Figure 46 Biofuel blends with conventional fuel

Different blend ratios allow the use of biofuels to be adjusted according to their availability and specific engine requirements. This can optimise the balance between engine performance and environmental benefits. Experimental studies (Chapter 1) have shown that blending biofuels with conventional fuels in proportions of up to 30% does not adversely affect engine performance, except in some cases. In this research, 10 fuel mixtures were prepared according to the table Backgammon 2 and were tested under the same pregnancy conditions.

Backgammon 2 Biofuel blends with diesel

5T95D	5% Turpentine + 95% Diesel
10T90D	10% Turpentine + 90% Diesel
15T85D	15% Turpentine + 85% Diesel
20T80D	20% Turpentine + 80% Diesel
30T70D	30% Turpentine + 70% Diesel
5EU95D	5% Eucalyptus oil + 95% Diesel
10EU90D	10% Eucalyptus oil + 90% Diesel
15EU85D	15% Eucalyptus oil + 85% Diesel
20EU80D	20% Eucalyptus oil + 80% Diesel
30EU70D	30% Eucalyptus oil + 70% Diesel

2.10 Experimental results

This subchapter will present the experimental results obtained in the study on the performance and pollutant emissions of the diesel engine using alternative fuels (turpentine and eucalyptus oil).

2.10.1 Engine Performance

Torque variability depending on turpentine incorporation can be observed in the four distinct conditions tested Figure *47*. At low load and an engine speed of 1700 RPM, the 15T85D compound recorded the most significant increase in torque, 7.9% higher than that of diesel. At an engine speed of 2250 RPM and low load, the 30T70D compound showed the highest performance, with 5.2% more torque than diesel. At high load and an engine speed of 1700 RPM, the 15T85D compound recorded the largest increase in torque over the reference fuel, marking an increase of 6.8%. Under high load and high engine speed, the 30T70D mixture demonstrated a torque increase of 4.7%.

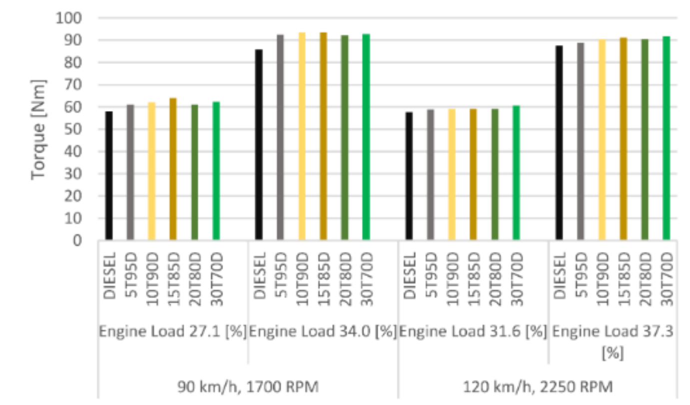


Figure 47 Torque variation, engine powered by turpentine-diesel mixtures

Figure 48 It shows the engine's braking power obtained with various fuel mixtures. At low revs, the increases were around 9% for fuel with a 15% turpentine incorporation, in both low rpm and low engine load settings.

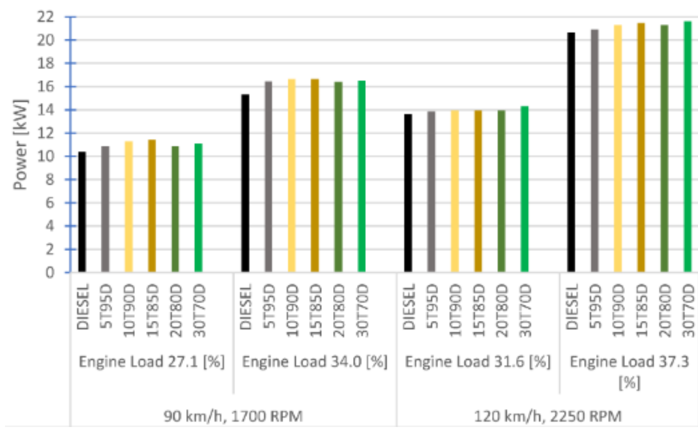


Figure 48 Measured power, engine powered by turpentine mixtures

Variations in specific fuel consumption for braking are illustrated in Figure *49*.

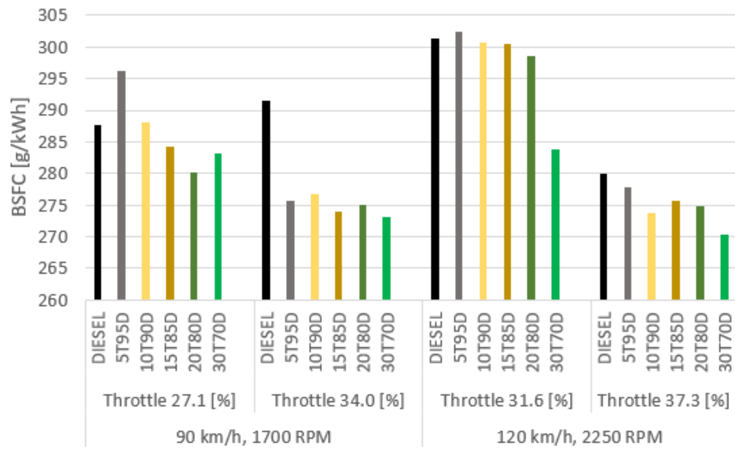


Figure 49 Specific fuel consumption, turpentine-diesel

The thermal brake efficiency (BTE) of the engine, expressed as percentages for various concentrations of biofuel and diesel blends, is illustrated in Figure *50*.

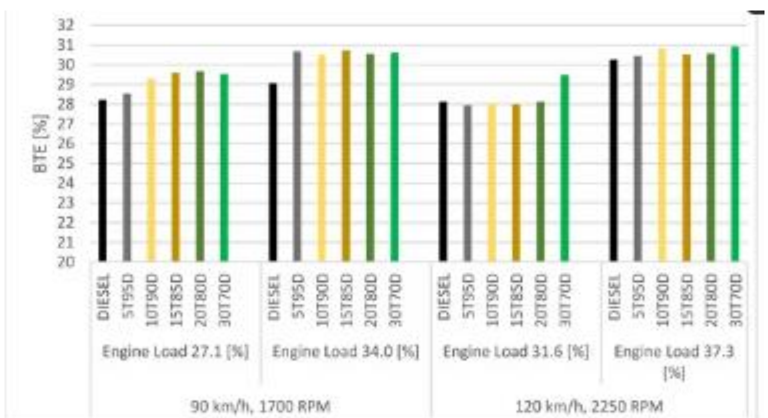


Figure 50 Thermal efficiency when braking, turpentine-diesel

The mixtures of eucalyptus essential oil and diesel followed the same test conditions.

The torque of the engine was evaluated based on the measurements made, and the results showed a significant correlation between the torque variation and the composition of the fuel mixtures (eucalyptus essential oil and diesel) Figure 51.

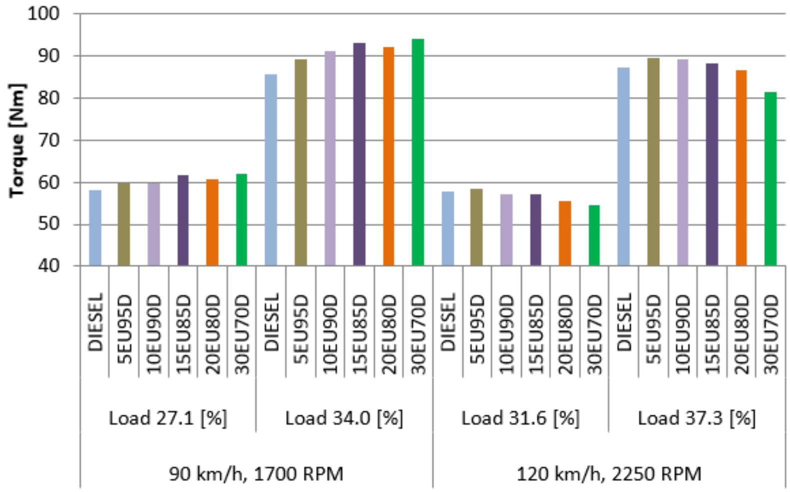


Figure 51 Torque variation, eucalyptus-diesel essential oil

For most engine revs and loads, incorporating eucalyptus oil in any percentage resulted in an increase in fuel consumption Figure *52*.

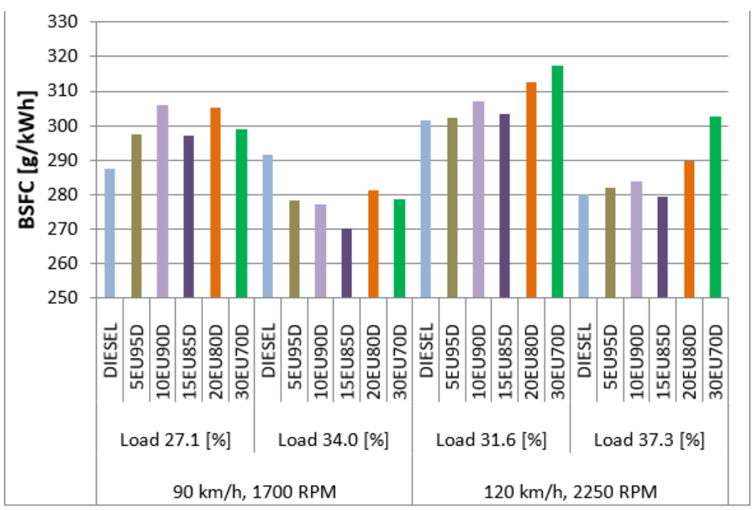


Figure 52 Specific fuel consumption, eucalyptus-diesel essential oil

Figure *53* shows the thermal efficiency of the brake. There are only minor oscillations between the different fuel mixtures at low engine speed (1700 rpm) and low engine load (27.1%).

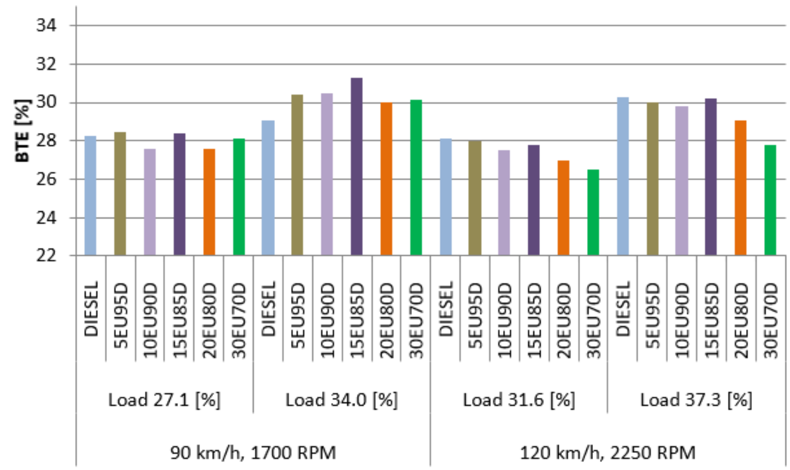


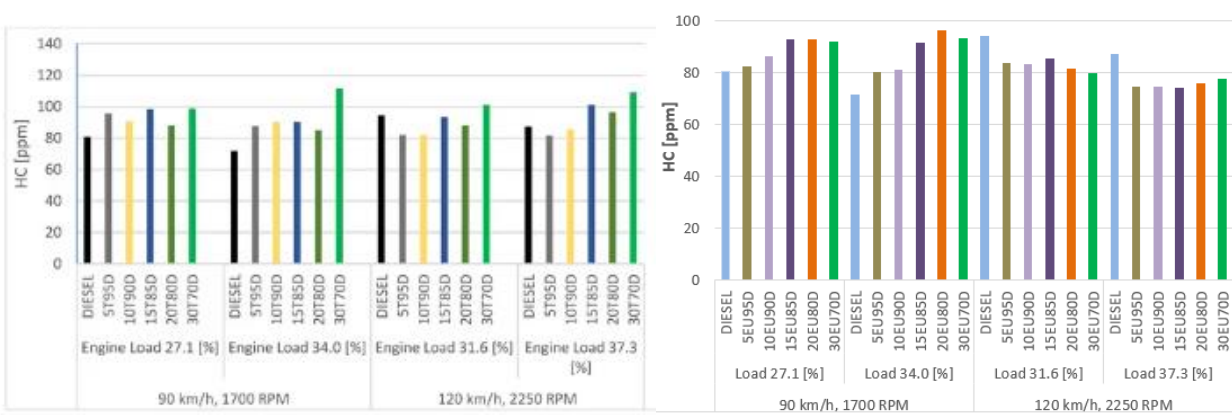
Figure 53 Thermal efficiency at the brake, eucalyptus-diesel essential oil

2.10.2 Pollutant emissions from the engine

Figure *54* a) shows the emissions of unburned hydrocarbons (HC). The mixture containing the highest percentage of turpentine in diesel generated the highest HC emissions for all engine loads (30T70D). In terms of engine load, the HC emissions of this mixture were 24.6% and 37.1% higher than those of the reference fuel at 1700 RPM. The HC emissions of the 30T70D were 25.3% and 17.3% higher than those of diesel at 2250 RPM. However, it should be noted that the timing (torque) improved with the incorporation of turpentine so the specific emissions did not increase as much.

Figure 54 b) present the concentrations of HC in ppm for each condition tested. Any incorporation of eucalyptus oil increases HC emissions, for low revs, but decreases when the engine is at high rpm regardless of the load.

Hydrocarbon emissions show relatively similar values, but turpentine mixtures demonstrate a slightly high and unstable trend compared to eucalyptus oil mixtures, where the emission trend is decreasing at high engine speeds.

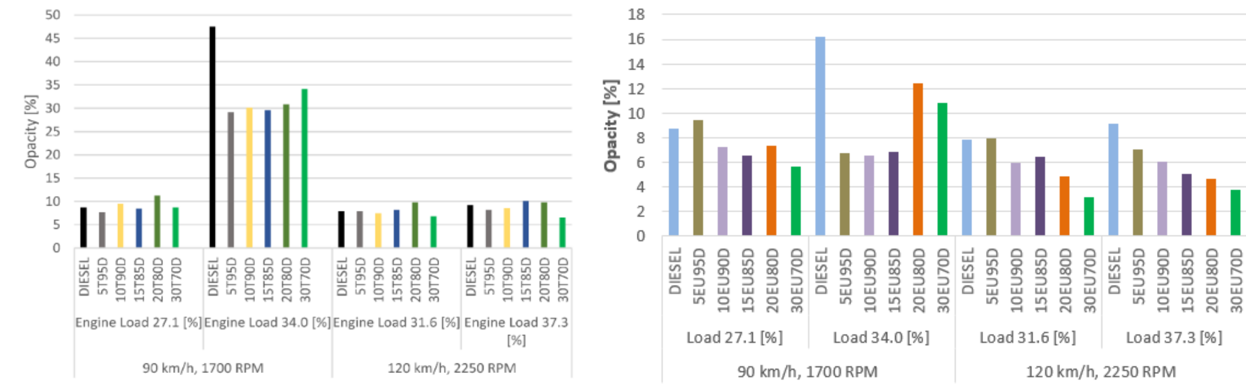


a) *b)*

Figure 54 Unburned hydrocarbon emissions

a) The smoke emissions of different fuel mixtures are presented in
b)

Figure *55*. Clearly, the high-load, low-speed test conditions produced much higher smoke levels than any other test conditions. This was true for all fuel mixtures.



b) *b)*

Figure 55 Smoke opacity

Carbon dioxide (CO_2) emission trends for turpentine and diesel blends Figure *56* remain constant, showing a slight increase, but not exceeding 3% under all test conditions. This behavior can be attributed to the combined characteristics of the fuels, which influence the combustion process and, implicitly, the production of CO_2. Maintaining the increase in CO_2 emissions within a narrow range indicates a relative stability of the mixtures in terms of combustion efficiency and impact on greenhouse gas emissions. These results suggest that although turpentine and diesel mixtures slightly alter the CO_2 emissions profile, the variations are small enough to be considered environmentally insignificant in the context of the conditions tested.

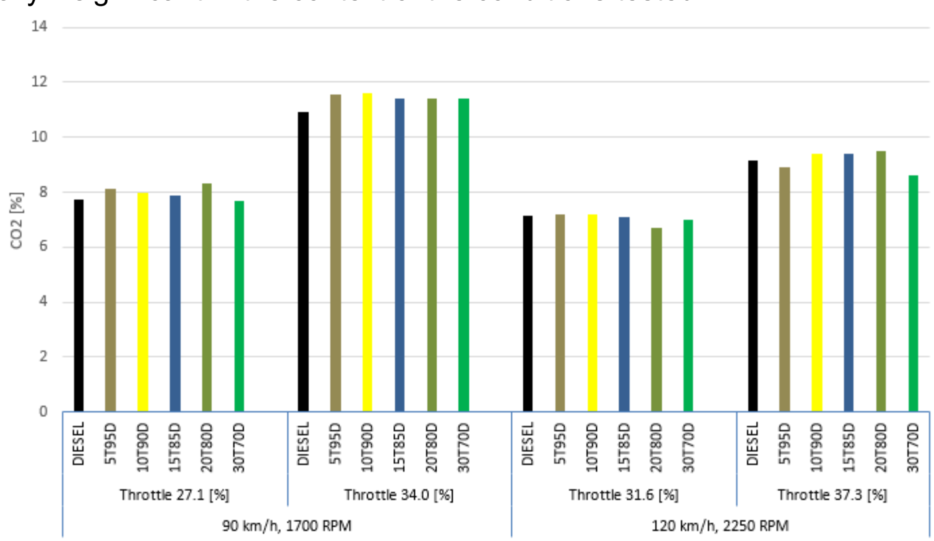


Figure 56 Carbon dioxide emissions, turpentine-diesel

Figure 57 shows that the measured carbon dioxide concentration reaches the lowest value in each case of engine speed and load. Because there is a higher concentration of eucalyptus oil in the fuel mixture, the CO2 concentration tends to decrease.

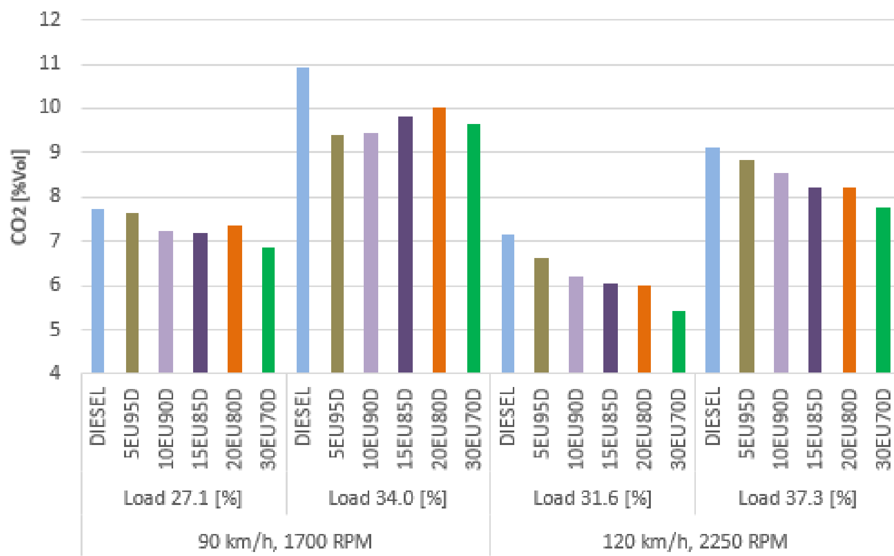


Figure 57 Emissions carbon dioxide, eucalyptus-diesel essential oil

Regarding nitrogen oxide emissions, it has been observed that when the concentration of turpentine increases, it leads to higher emissions of Nox Figure *58*. The presence of oxygen in biofuel has increased NOx production. In addition, it is known that, due to the conditions that favor the formation of nitrogen oxides, an increase in combustion performance tends to increase the combustion temperature and, indirectly, the emission of NOx. The 30T70D mixture recorded the highest values. Again, the incorporation of turpentine helped increase torque, so the increase in specific emissions is not so great.

Combustion temperature, excess air, and the presence of oxygen in the fuel molecule are all related to nitrogen oxide emissions. The exhaust gas temperature of the eucalyptus oil and diesel mixture was generally lower at all engine speeds and loads. Figure *59* shows emissions of nitrogen oxides.

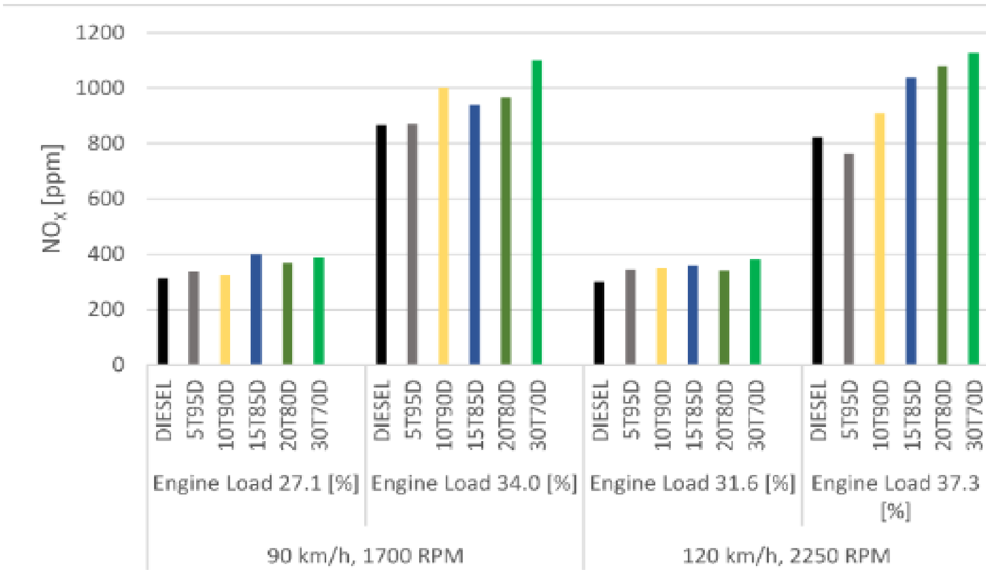


Figure 58 Emissions of nitrogen oxides, turpentine-diesel

However, this research found that nitrogen oxide emissions from the reference fuel were at least 17% higher than those of fuel blends even when engine torque and brake thermal efficiency were increased. This is an excellent result compared to other biofuels such as turpentine, which, due to the presence of oxygen in its molecule, tends to increase NOx emissions Figure *58*.

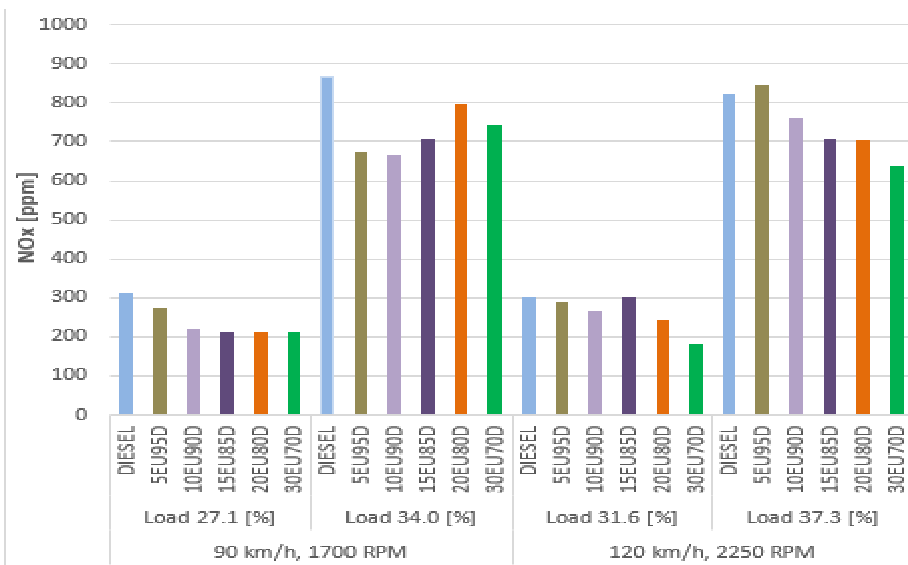


Figure 59 Emissions of nitrogen oxides, eucalyptus-diesel essential oil

Chapter 3. Numerical modeling. Zero-dimensional internal combustion engine simulation

3.1 Classification of simulation models

Zero-dimensional (0D) models, also known as thermodynamic models, are a simplified and efficient approach to simulating internal combustion engines

Advantages of zero-dimensional models. Due to the simplifying assumptions, 0D models are relatively quick and easy to implement. This makes them ideal for preliminary evaluations and quick optimizations of engine parameters. 0D models require reduced computational resources compared to more complex models (1D, 2D, 3D), allowing for fast simulations and multiple iterations. These models are frequently used in the initial phases of engine development to evaluate different design concepts and select the optimal operating parameters.

The limits of zero-dimensional models. 0D models cannot capture spatial variations in thermodynamic parameters inside the combustion chamber, which limits accuracy in predicting the local distribution of temperature and composition. By their nature, 0D models are rough approximations of the actual processes that take place in the engine. This can lead to errors in detailed predictions, such as emission formation and combustion behavior. The accuracy of 0D models depends largely on the quality of the experimental data used for calibration. Inaccuracies in the input data can lead to errors in the simulated results.

3.2 Unizonal model

The internal processes of unizonal model simulation can be described using a set of thermodynamic and transport equations. These equations include mass balances, energy balances, and the equation of state of gases.

In Figure *60* The unizonal model for this research is presented in which the limit of the system and the terms for conservation of mass and energy are observed.

Direct injection (engines with air intake) was considered for the terms presented. In the version with mixture inlet, the fuel mass is already included in the mass entering the system, m_{in}. According to the continuity equation, the change in mass m from volume, is equal to the sum of the inlet and outlet mass flows, $\dot{m}_i$. The law of conservation of mass for the system under consideration (combustion chamber) has the following differential form with respect to the angle of rotation of the crankshaft φ.

Equation 1 The law of conservation of mass

$$\frac{dm}{d\varphi} = \frac{dm_{in}}{d\varphi} + \frac{dm_{ex}}{d\varphi} + \frac{d_{m_{BB}}}{d\varphi} + \frac{d_{m_f}}{d\varphi}$$

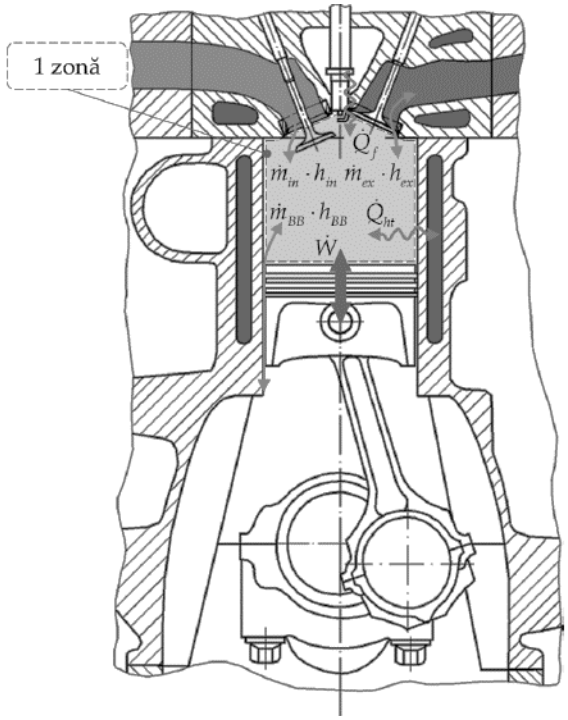


Figure 60 Unizonal model, adapted from [41]

The conservation of energy for the combustion chamber is expressed in differential form with the help of Equation 2.

Equation 2 General form of energy conservation

$$\frac{dU}{d\varphi} = \frac{dQ_f}{d\varphi} + \frac{dQ_{ht}}{d\varphi} + \frac{dW_v}{d\varphi} + \frac{dm_{in}}{d\varphi} h_{in} + \frac{dm_{ex}}{d\varphi} h_{ex} + \frac{dm_{BB}}{d\varphi} h_{BB}$$

Where, see Equation *24*

$$\boldsymbol{\frac{dW_V}{d\varphi} = -p\frac{dV}{d\varphi}}$$

Determining the state of a homogeneous gas involves the use of physical parameters such as pressure (P), volume (V) and temperature (T).

Ideal gas state equation:

Equation 3 Thermal equation of state

$$pV = nRT$$

The equation of state of gas can also be written in differential form

Equation 4 Equation of state of gas in differential form

$$p\frac{dv}{d\varphi}+V\frac{dp}{d\varphi}=nR\frac{dT}{d\varphi}+mT\frac{dR}{d\varphi}+RT+\frac{dm}{d\varphi}$$

The total internal energy U of the combustion chamber can be expressed as the sum of the products between the individual masses of the chemical species present and their specific internal energies. The equation for internal energy U is:

Equation 5 Internal Energy Equation

$$U=\sum_{i}^{n}m_i u_i$$

The diesel engine forms the fuel mixture in the cylinder, therefore the mass of fuel introduced turns into burned mass.

Equation 6 Fuel mass inserted into the cylinder

$$m=m_a+m_b$$

Where the mass of air that is required m_a can be determined from the total mass of fuel m_f, the air requirement *AFR* ($m_{a\,min}$) and the ratio of fuel to air denoted with λ.

When the engine is running, the internal energy changes, therefore in the air intake engine the internal energy varies depending on the angle of rotation of the crankshaft φ is presented in Equation 7.

Equation 7 Variation of internal energy during a motor cycle

$$\frac{dU}{d\varphi}=\frac{d\lambda}{d\varphi}\cdot AFR\cdot m_f\cdot u_a+\frac{dm_f}{d\varphi}\cdot\lambda\cdot AFR\cdot u_a+\frac{du_a}{d\varphi}\cdot\lambda\cdot AFR\cdot m_f$$
$$+\left\{\frac{dm}{d\varphi}-\left[\frac{d_{m_f}}{d\varphi}(1+\lambda\cdot AFR)+\frac{d\lambda}{d\varphi}m_f\cdot AFR\right]\right\}u_b+\frac{du_b}{d\varphi}\left[m-m_f(1+\lambda\cdot AFR)\right]$$

3.3 Modeling of internal energy

According to Heywood, the equation for calculating molar enthalpy is described by Equation *8*:

Equation 8 Equation for calculating molar enthalpy

$$\bar{h}_i=\bar{R}\cdot T\left(a_{i,1}+\frac{a_{i,2}}{2}\mathrm{T}+\frac{a_{i,3}}{3}T^2+\frac{a_{i,4}}{4}T^3+\frac{a_{i,5}}{5}T^4+\frac{a_{i,6}}{T}\right)$$

The internal energy u_i of a component is obtained from the specific molar internal energy $\bar{u}_i$ and molar mass $\bar{M}_i$. The average specific molar internal energy is calculated from the molar participations X_i and the specific molar internal energies of the components according to the equation **Error! Reference source not found.**.

3.4 Combustion modeling

Equation 9 Shaping the general shape of the combustion

$$\frac{dQ_f}{\mathrm{d}\varphi} = LHV \cdot \frac{dm_f}{\mathrm{d}\varphi}$$

$$Q_{f\,total} = LHV \cdot m_f$$

The most used combustion models are listed the Vibe, Dubl-Vibe, Vibe-hyperbole model.

The Vibe model.

The Vibe combustion model, developed by Ivan Vibe, is a simple and effective mathematical model used to describe the release of heat during the combustion process in internal combustion engines. This model is widely used due to its ability to adjust to represent various combustion scenarios by changing a small number of parameters,[42].

Equation 10 The Combustion Characteristic Equation of the Vibe Model

$$MFB = \frac{Q_f(\varphi)}{Q_{f\,total}} = 1 - e^{-a\left(\frac{\varphi - \varphi_{SOC}}{\Delta\varphi_{comb}}\right)^{m+1}}$$

$$\varphi_{SOC} \le \varphi \le \varphi_{SOC} + \Delta\varphi_{comb}$$

Vibe Parameter *a* is described by Equation *11*.

Equation 11 Equation of the Vibe parameter "a"

$$a = -\ln(1 - \eta_{c\,total})$$

ROHR (Rate of Heat Release) stands for the rate of heat release in an internal combustion engine. It is a measure of the energy released per unit time, or per unit angle of rotation of the crankshaft, during the combustion process.

Equation 12 Heat Release Rate Equation

$$ROHR = \frac{dQ_f}{d\varphi} = \frac{Q_{f\,total}}{d\varphi} a(m+1) \cdot \left(\frac{\varphi - \varphi_{SOC}}{\Delta\varphi_{comb}}\right)^m \cdot e^{-a\left(\frac{\varphi - \varphi_{SOC}}{\Delta\varphi_{comb}}\right)^{m+1}}$$

The Double Vibe Model

The Double Vibe is an extension of the classic Vibe model, used to more accurately describe the complex combustion process in diesel engines.

Equation 13 The first equation for the Vibe double model

$$\frac{dQ_{f\,1}(\varphi)}{d\varphi} = Q_{f\,1} \cdot a \cdot (m_1 + 1) \cdot \left(\frac{\varphi - \varphi_{SOC\,1}}{\Delta\varphi_{comb\,1}}\right)^{m_1} \cdot e^{-a\left(\frac{\varphi - \varphi_{SOC\,1}}{\Delta\varphi_{comb\,1}}\right)^{m_1+1}}$$

Equation 14 The second equation for the Vibe double model

$$\frac{dQ_{f\,2}(\varphi)}{d\varphi} = Q_{f\,2} \cdot a \cdot (m_2 + 1) \cdot \left(\frac{\varphi - \varphi_{SOC\,2}}{\Delta\varphi_{comb\,2}}\right)^{m_2} \cdot e^{-a\left(\frac{\varphi - \varphi_{SOC\,2}}{\Delta\varphi_{comb\,2}}\right)^{m_2+1}}$$

Cumulative fractions of heat released for the premix and diffusion phases $Q_{f\,1}, Q_{f\,2}$ is divided by the factor x and results Equation *15*.

Equation 15 Amount of heat released relative to X

$$Q_{f\,1} = x \cdot Q_{f\,total}$$

$$Q_{f\,2} = (1 - x) \cdot Q_{f\,total}$$

$$Q_{f\,total} = Q_{f\,pre} + Q_{f\,diff}$$

The Double Vibe function cannot reproduce the continuous combustion during the exhaust of gases from the cylinder.

3.5 Applicability of combustion patterns

Combustion in zero-dimensional models is used in various situations to make predictions about the evolution of the combustion process at an operating point that we do not know. This method involves determining the basic parameters through initial measurements and calculations, followed by adjusting the combustion curve according to the new operating conditions [43].

Equation 16 The combustion equation according to Woschini [41], [43]

$$\Delta\varphi_{comb} = \Delta\varphi_{comb\,ref} \left(\frac{\lambda_{ref}}{\lambda}\right)^{0,6} \left(\frac{n}{n_{ref}}\right)^{0,5}$$

Determining the parameters of the Vibe model according to ignition delay can be achieved by a methodical approach, taking into account ignition delay (φ_{IGD}), pressure (p), temperature (T) and speed (n). The essential parameters of the Vibe model are the form factor (m) and the form coefficient (a). These parameters can be determined using empirical relationships and specific adjustment methods. The parameter m can be determined according to Equation 24.

Equation 17 Determination of the parameter m according to Woschini

$$m = m_{ref}\left(\frac{\varphi_{IGD}}{\varphi_{IGD}}\right)^{0,5}\left(\frac{p_{IVC}}{p_{IVC\,ref}}\right)\left(\frac{T_{IVC\,ref}}{T_{IVC}}\right)\left(\frac{n_{ref}}{n}\right)^{0,3}$$

In the context of thermotechnics and heat engines, the term 'net heat' refers to the total thermal energy generated in a combustion or energy conversion process, minus the energy lost in the form of exhaust heat and other losses. It is a measure of the useful energy available after all thermal losses have been taken into account. The equation that describes this process in differential form is Equation *18*.

Equation 18 Equation in differential form of net heat

$$\frac{dQ_{f\,n}}{d\varphi} = \frac{dQ_f}{d\varphi} - \frac{dQ_{ht}}{d\varphi}$$

3.6 Heat loss through walls

Conduction heat exchange

Thermal conduction is based on the transfer of kinetic energy from the more energetic (hotter) molecules to the less energetic (colder) ones, in the absence of a macroscopic transport of matter. Fourier's law mathematically describes this phenomenon Equation *19*.

Equation 19 Calculation of conduction through the walls of the combustion chamber

$$Q(x,t) = -k \cdot \mathrm{A} \cdot gradT(x,t)$$

Equation 19 one can write for one-dimensional steady-state conduction as Equation 20

Equation 20 Equation for steady-state conduction

$$\dot{Q}_{cond} = -kA\frac{dT}{d_X}$$

Convection heat exchange

Newton's law of cooling describes the phenomenon of heat transfer by convection and is expressed mathematically by **Error! Reference source not found.**:

Equation *21* is a correlation for the convection heat transfer coefficient (h_C). This relationship is an empirical correlation and is used to calculate the convection heat transfer coefficient in pipes, taking into account the physical properties of the fluid and the flow conditions.

Equation 21 Convection heat transfer coefficient in pipes

$$h_C = C' \cdot D \cdot k \cdot \left(\frac{w_{avg}\rho}{\mu}\right)^{0,8}$$

Modeling Heat Exchange With Wall After Woschini

The Woschni model is one of the most used models to estimate this coefficient. The model is based on an empirical correlation that takes into account the operating conditions of the engine and the properties of the gas Equation *22*.

Equation 22 The equation of heat exchange with walls according to Woschini

$$h_c = 127,93 \cdot D \cdot p^{0,8} \cdot T^{-0,53} \cdot \left[C_1 S_{pm} + C_2 \frac{V_d \cdot T_{IVC}}{p_{IVC} V_{IVC}} (p - p_m)\right]^{0,8}$$

Modeling the heat exchange with the wall of the AVL company

In the context of the AVL model, the heat transfer coefficient will be the maximum between the Woschni method and the second term of the equation Equation *23*.

Equation 23 Heat Transfer Equation of the AVL Model

$$h_c = \max\left\{h_{c\,Woschini}; 0,013D \cdot p^{0,8} \cdot T^{-0,53}\left[14\left(\frac{d_{in}}{D}\right)^2 |V_{in}|\right]^{0,8}\right\}$$

3.7 Mechanical work

Therefore, mechanical work can be described under the differential equation with the help of Equation *24*.

Equation 24 Mechanical work according to the angle of the crankshaft

$$\frac{dW_V}{d\varphi} = -p\frac{dV}{d\varphi}$$

$$s(\varphi) = r + l - l \cdot \left(\lambda_s cos\varphi + \sqrt{1 - {\lambda_s}^2 \sin^2 \varphi}\right)$$

In particular, this is a differential expression that describes the change of mechanical work dW_V as a function of the change in volume dV, given the pressure p. This is a differential form of mechanical work in thermodynamics, where mechanical work W is given by:

Equation 25 Mechanical work, the general equation

$$dW_V = -p\, dV$$

The pressure p acts on a volume V to produce the mechanical work W. The negative sign indicates that when the volume increases (dV > 0), the system performs mechanical work on the external environment (dW_V< 0), and vice versa, when the volume decreases (dV < 0), the external environment performs mechanical work on the system (dW_V> 0).

3.8 Enthalpy flow

Specific enthalpy is determined by Equation 26.

Equation 26 Specific enthalpy

$$h = u + RT$$

Equation *27* describes the mass flow $\dot{m}_{in}$ rate of a gas passing through a hole or nozzle depending on the inlet conditions and pressure.

Equation 27 Equations of Inlet-Outlet Mass Flows through Valves

$$\dot{m}_{in} = \mu_{in}A_{in}\frac{p_1}{\sqrt{RT_{in}}}\sqrt{\frac{2k}{k-1}\left[\left(\frac{p}{p_{in}}\right)^{\frac{2}{k}} - \left(\frac{p}{p_{in}}\right)^{\frac{k+1}{k}}\right]}$$

$$\dot{m}_{ex} = \mu_{ex}A_{ex}\frac{p_1}{\sqrt{RT}}\sqrt{\frac{2k}{k-1}\left[\left(\frac{p_{ex}}{p}\right)^{\frac{2}{k}} - \left(\frac{p_{ex}}{p}\right)^{\frac{k+1}{k}}\right]}$$

The cross-sectional area can be calculated according to Equation 28.

Equation 28 Cross-sectional area

$$A_{in} = z_{in} x_{in} \cos\beta_{in} d_{m\,in} \cdot \pi$$

$$d_{m\,in} = d_{in} + 2 \cdot a$$

$$a = b \cdot cos\beta_{in}$$

$$b = \frac{1}{2} x_{in} \sin\beta_{in}$$

$$d_{m\,in} = d_{i\,in} + x_{in} \sin\beta_{in} \cdot cos\beta_{in}$$

$$A_{in} = z_{IV} \cdot x_{in} \cdot cos\beta_{in}(d_{i\,in} + x_{in} \sin\beta_{in} \cdot cos\beta_{in}) \cdot \pi$$

3.9 Computer simulation, model validation and error analysis

1. For the reference fuel:
   - Measured value: 57.7 *Nm* Figure 47, Figure 51
   - Simulated Value: 56.1 *Nm* (according to the simulation) **Error! Reference source not found.**

$$\text{Relative deviation} = \left|\frac{57{,}7 - 56{,}1}{57{,}7}\right| \cdot 100\ [\%] \approx 2{,}7\%$$

2. For the mixture of 30% turpentine and 70% diesel:
   - Measured value: 60.7 *Nm* Figure 47
   - Simulated Value: 58.5 *Nm* (according to the simulation) **Error! Reference source not found.**

$$\text{Relative deviation} = \left|\frac{60{,}7 - 58{,}5}{60{,}7}\right| \cdot 100\ [\%] \approx 3.6\%$$

3. For the mixture of 30% eucalyptus oil and 70% diesel:
   - Measured value: 54.6 *Nm* Figure 51
   - Simulated Value: 54.7 *Nm* (according to the simulation) **Error! Reference source not found.**

$$\text{Relative deviation} = \left|\frac{54{,}6 - 54{,}7}{54{,}6}\right| \cdot 100\ [\%] \approx 0{,}18\%$$

1. Reference fuel:

- Measured value: 13.63 *Kw* Figure *48*
- Simulated value: 13.25 *kW*

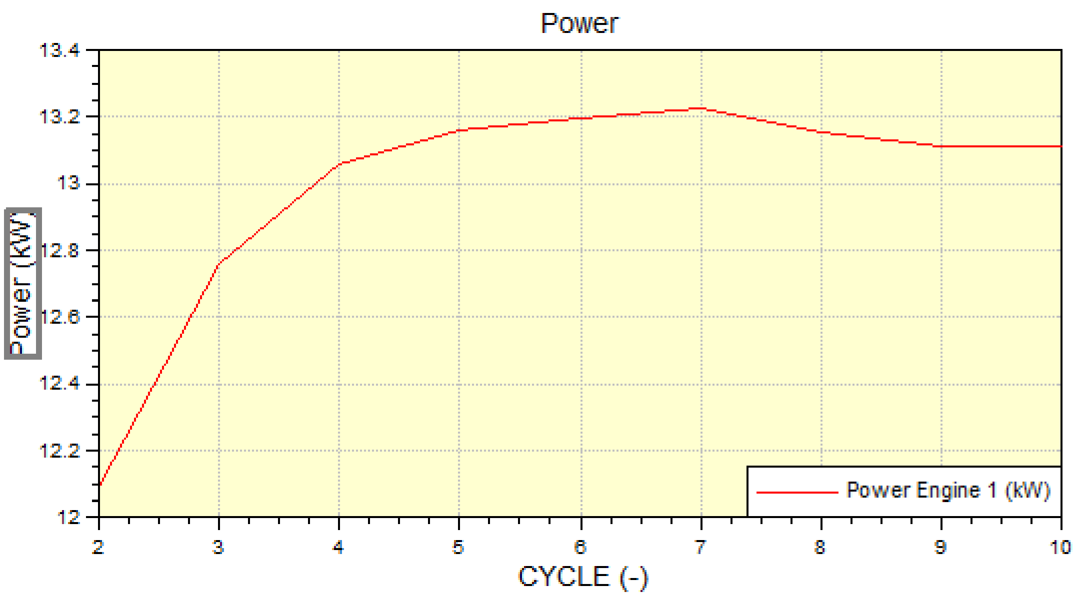
$$\text{Relative deviation} = \left|\frac{13{,}63 - 13{,}25}{13{,}63}\right| \cdot 100\ [\%] \approx 2{,}7\%$$

2. Blend of 30% turpentine and 70% diesel (30T 70D):
   - Measured value: 14.31 *Kw* Figure *48*
   - Simulated value: 13.8 *kW*

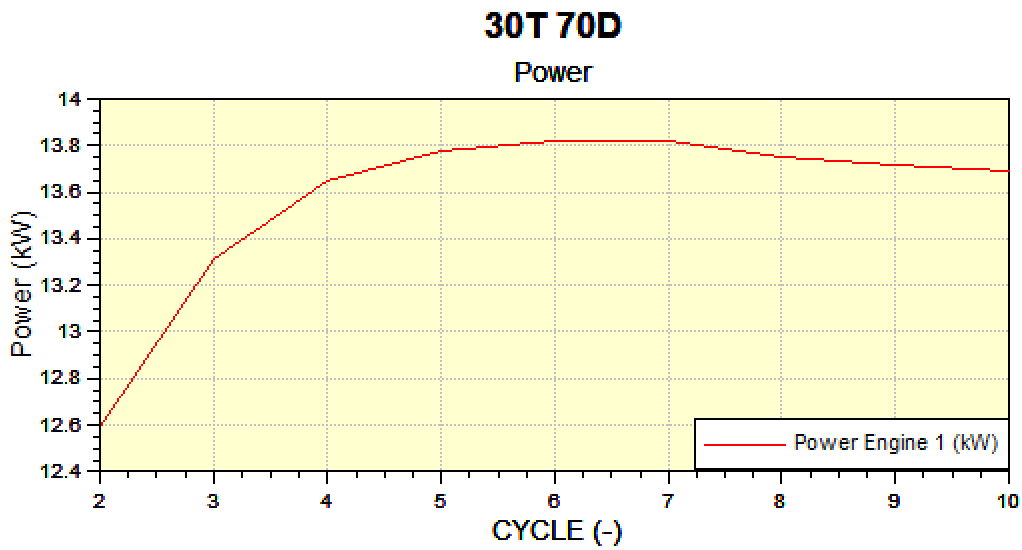
$$\text{Relative deviation} = \left|\frac{14{,}31 - 13{,}8}{14{,}31}\right| \cdot 100\ [\%] \approx 3{,}56\%$$

3. Mixture of 30% eucalyptus oil and 70% diesel (30EU 70D):
   - Measured value: 12.86 *kW*
   - Simulated value: 12.8 *kW*

$$\text{Relative deviation} = \left|\frac{12{,}86 - 12{,}8}{12{,}86}\right| \cdot 100\ [\%] \approx 0{,}46\%$$



a)



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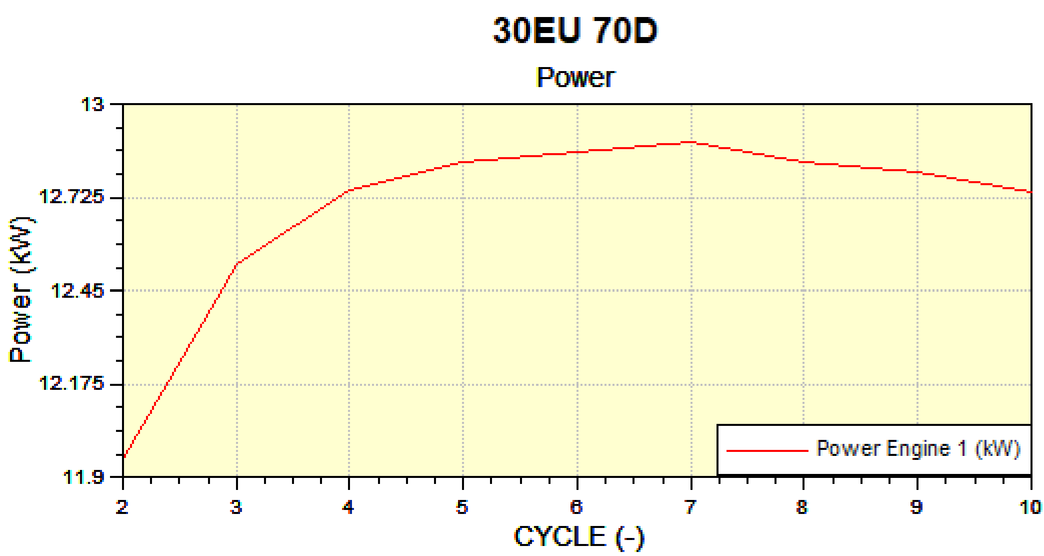


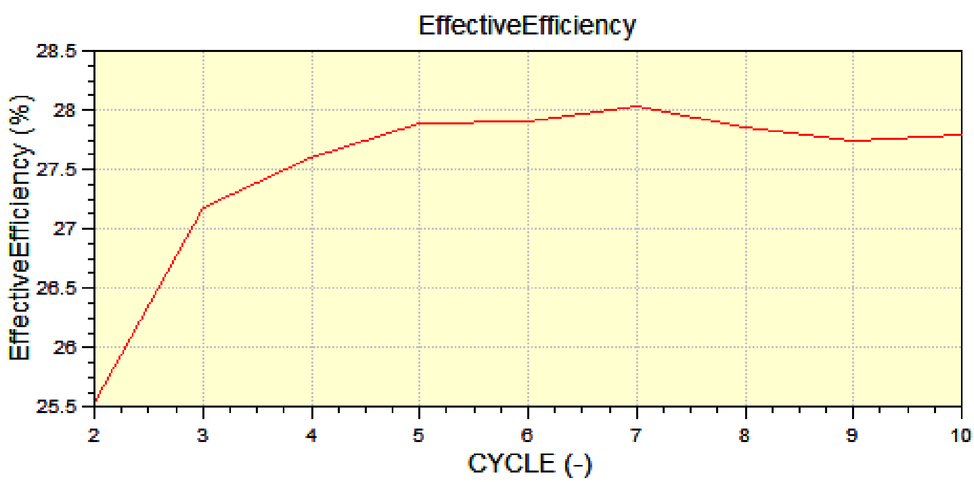
c)

Figure 61 Power variation resulting from computer simulation

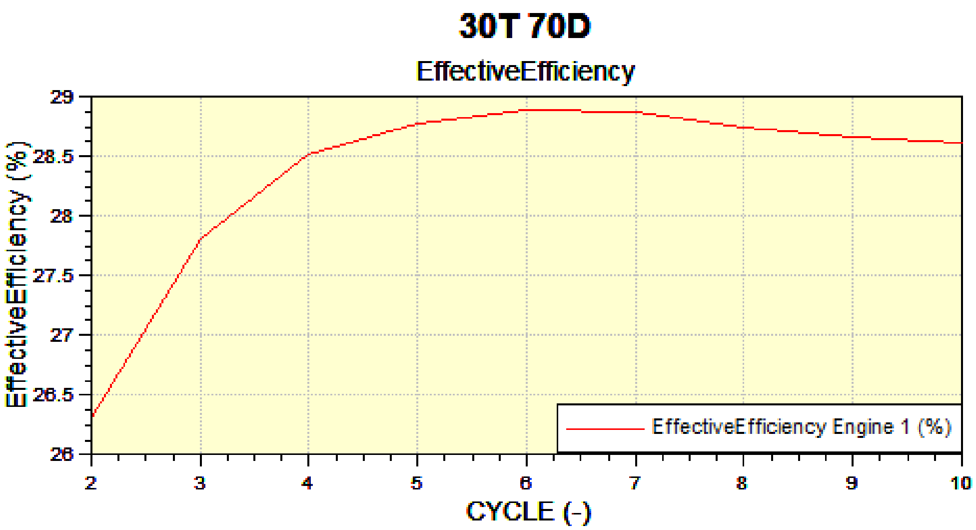
Numerical model validation: With relative deviations of 2.7%, 3.56%, and 0.46%, the numerical model appears to be an accurate representation of experimental reality. These small deviations indicate that the numerical model can be considered valid for further predictions and analyses in terms of the power produced by the engine.

Brake Thermal Efficiency (BTE) and Brake Specific Fuel Consumption (BSFC) are two essential parameters in evaluating the performance of an internal combustion engine. Essentially, these two parameters are interconnected and provide complementary information about the energy efficiency of the engine.

Figure *62* a) illustrate the thermal efficiency at the brake (BTE) for the reference fuel, diesel. It is observed that the efficiency starts at 25.5% and quickly increases to about 27.7% between cycles 2 and 4. This rapid increase can be attributed to the optimization of initial combustion processes, where fuel is used more efficiently as the engine reaches an optimal operating temperature and pressure. Subsequently, the efficiency continues to increase slightly to 28% in cycle 6, after which it stabilizes around this value until cycle 10, suggesting that the engine has reached a break-even point where combustion efficiency is maximum and constant. Figure *62* b) shows the thermal efficiency at the brake for a mixture of 30% turpentine and 70% diesel. Efficiency rapidly increases to 28.5% by cycle 4. This initial rapid increase is similar to that seen for diesel, indicating good compatibility of turpentine in the mixture. After cycle 7, efficiency decreases slightly and stabilizes around 28.5%, reflecting a break-even point similar to pure diesel, but at a slightly higher efficiency. Figure *62* c) shows the thermal efficiency at the brake for a mixture of 30% eucalyptus oil and 70% diesel. Efficiency starts at 25% and quickly increases to 27% by cycle 4. The rapid initial growth is similar to that of the other fuels, indicating that eucalyptus oil can be used effectively in the mixture. Efficiency continues to increase, reaching a maximum of 27.1% in cycle 7. This maximum value is lower than that obtained with the mixture of turpentine and diesel, suggesting that although eucalyptus oil does not improve efficiency over pure diesel and is not as effective as turpentine.



a)



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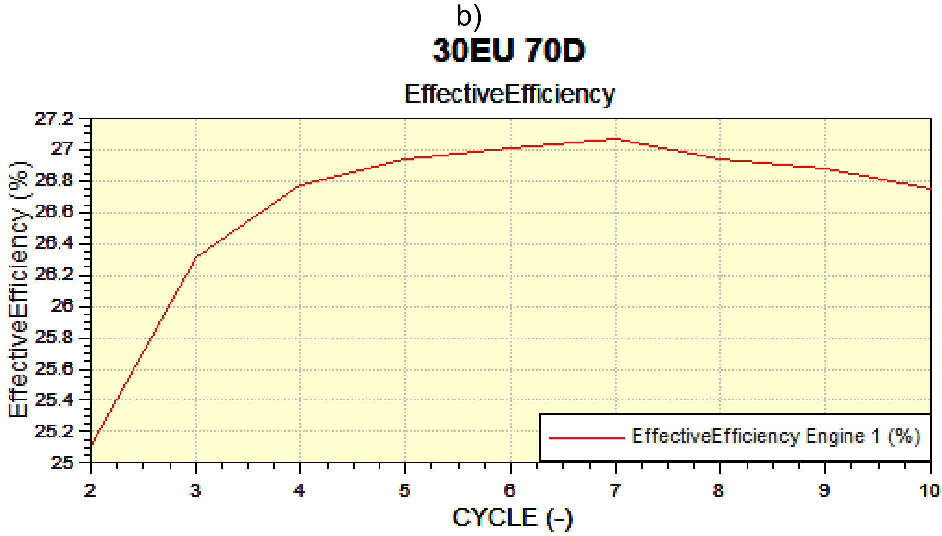


c)

Figure 62 Effective efficiency variation, simulated results

The measured values for thermal efficiency at the brake are:

- Diesel (reference fuel): 28.1 % Figure *50* Figure *50*Figure *53*
- Turpentine mixture: 29.5 % Figure *50*
- Eucalyptus oil mixture: 26.5 % Figure *53*

The simulated values for the peak BTE are:

- Diesel (reference fuel): 28 % Figure *62* a)
- Turpentine Blend: 29 % Figure *62* b)
- Eucalyptus oil mixture: 28 % Figure *62* c)

Comparing the three types of fuels, we notice that:

- **Pure Diesel**: the lowest value is 302 *g/kWh* after cycle 4 indicates a decent and constant fuel efficiency.
- **Turpentine Blend**: With a value of 289 *g/kWh* after cycle 4, this blend shows the highest fuel efficiency of the three, demonstrating the benefits of turpentine in optimizing engine performance.
- **Eucalyptus Oil Blend**: Stabilization at 311 *g/kWh* after cycle 4 suggests lower fuel efficiency, reflecting the limitations of this blend compared to pure diesel and turpentine blend.

Fuel efficiency, represented by BSFC, depends on several factors, including combustion quality, air-fuel ratio, and the chemical properties of the fuel. A fuel with better combustion properties will have a lower BSFC because it will produce more mechanical energy for the same amount of fuel.

Turpentine: Contains compounds that can improve combustion, resulting in more efficient fuel use and thus lower BSFC.

Eucalyptus oil: It may have combustion characteristics that are not as efficient, leading to less optimal fuel usage and a higher BSFC.

The measured values for the specific fuel consumption at the brake are:

- Diesel (Reference Fuel): 301.4 *g/kWh* Figure *49* Figure *52*
- Turpentine Mixture: 283.7 *g/kWh* Figure *49*
- Eucalyptus Oil Blend: 317.2 *g/kWh* Figure *52*

The simulated values for the BSFC peak are:

- Diesel (reference fuel): 302 *g/kWh* ***Error! Reference source not found.*** a)
- Turpentine mixture: 289 *g/kWh* **Error! Reference source not found.** b)
- Eucalyptus oil mixture: 311 *g/kWh* **Error! Reference source not found.** c)

Calculation of relative deviation

For Diesel (reference fuel):

$$\text{Relative deviation} = \left|\frac{301.4 - 302}{301.4}\right| \cdot 100\ [\%] \approx 0{,}2\%$$

For the turpentine mixture:

$$\text{Relative deviation} = \left|\frac{283.7 - 289}{283.7}\right| \cdot 100\ [\%] \approx 1{,}86\%$$

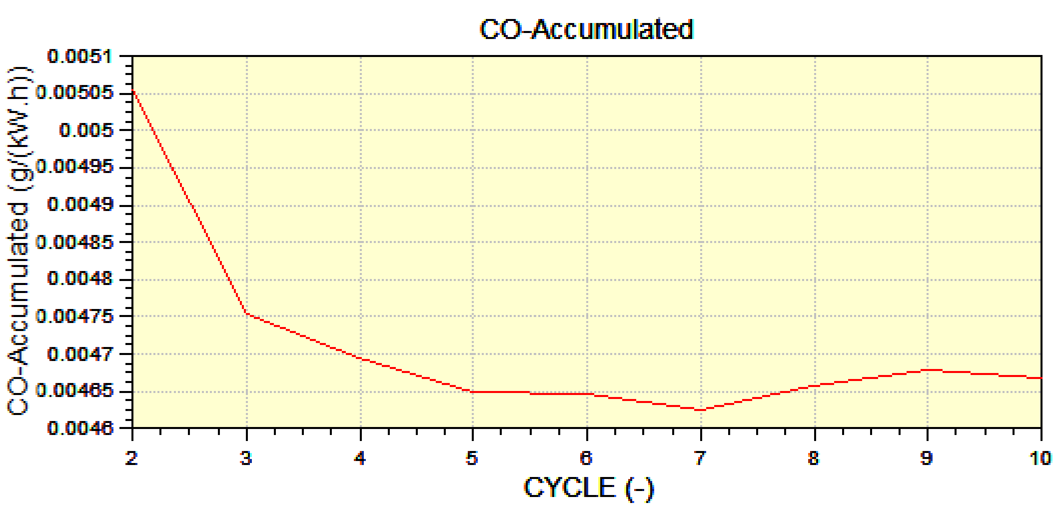
For the eucalyptus oil mixture:

$$\text{Relative deviation} = \left|\frac{317.2 - 311}{317.2}\right| \cdot 100\ [\%] \approx 1{,}95\%$$

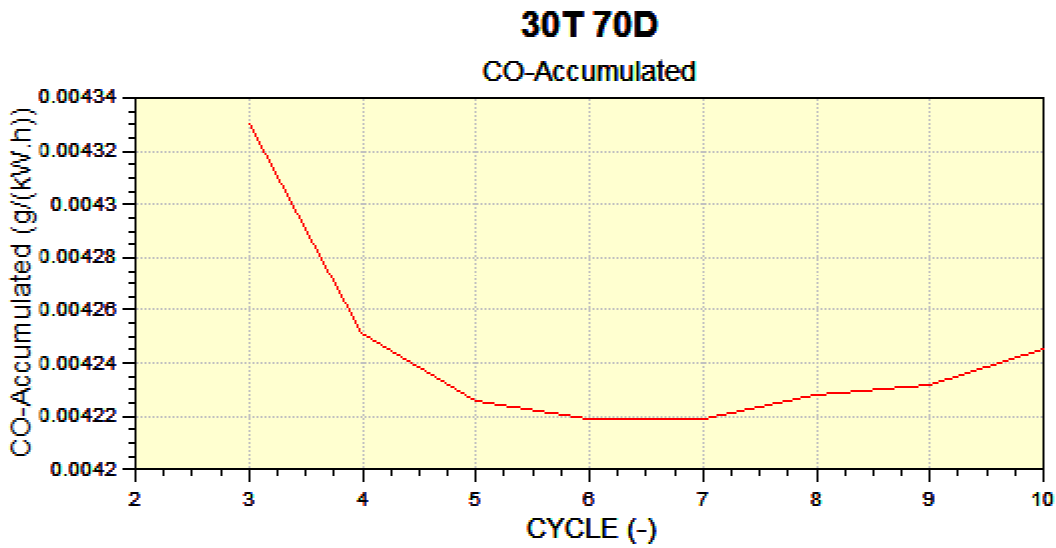
Based on the calculated relative deviations, all values are below 5%, which suggests a high accuracy of the numerical model. Therefore, I can conclude that the numerical model is valid and can be used to predict the BSFC for the analyzed fuels (pure diesel, mixture of 30% turpentine and 70% diesel, mixture of 30% eucalyptus oil and 70% diesel).

Analyzing the graphs provided in the Figure 63, we can see how the values of accumulated CO emissions vary depending on the type of fuel used. Figure 63 a), representing the reference fuel, shows initial accumulated CO values of approximately 0.00505 *g/kWh*, which fall rapidly and stabilize around 0.00465 *g/kWh*. In the case of Figure 63 b), which represents the mixture of 30% turpentine and 70% diesel, the accumulated CO values start from about 0.00433 *g/kWh* and fall to about 0.00422 *g/kWh*, indicating a more complete or efficient combustion compared to the reference fuel. In the third graph in Figure 63 c), for the mixture of 30% eucalyptus oil and 70% diesel, the accumulated CO values are intermediate, starting from approximately 0.00495 *g/kWh* and decreasing to about 0.00455 *g/kWh*.

These observations suggest that the turpentine mixture produces the lowest CO emissions, while the reference fuel has the highest initial values. The absence of CO data in the AVL can be explained by the very low emissions, below the detection limits of the device, resulting from the combustion efficiency of diesel engines, which operate with excess air. Diesel engines, due to their high air/fuel ratio and high combustion temperatures and pressures, favor complete oxidation of the fuel, which leads to very low CO emissions. In addition, the quality of alternative fuels, such as turpentine and eucalyptus oil, can contribute to a more complete combustion, further reducing CO emissions. Thus, the use of alternative fuels seems to positively influence CO emissions, while the meter has limitations in detecting very low concentrations of CO.



a)



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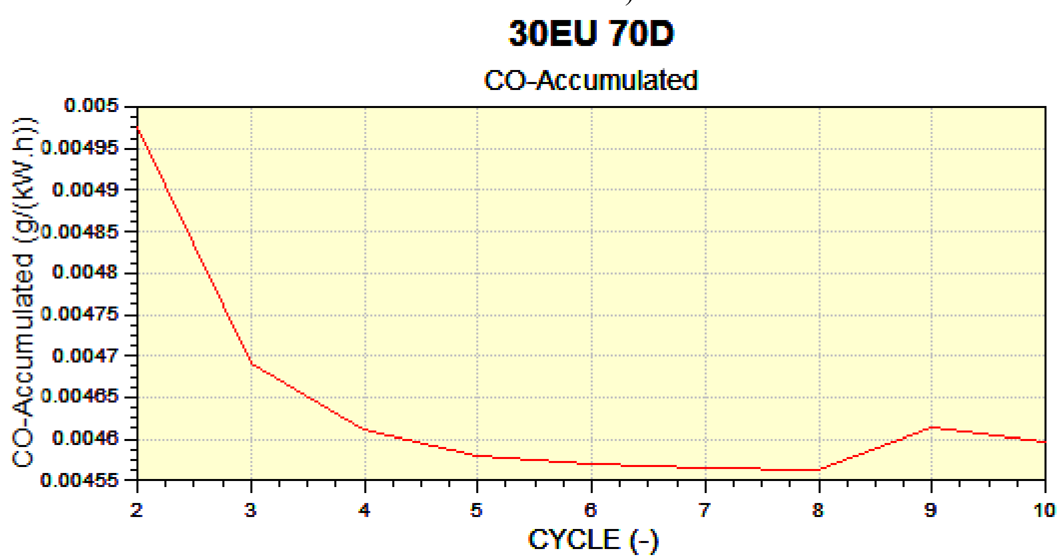

c)

Figure 63 Carbon monoxide emissions, simulated results

Chapter 4. Conclusions and personal contributions

4.1 Concluding Conclusions

The final conclusions represent the essential synthesis of our investigative approach, bringing together the main discoveries and reflections that resulted from the in-depth analysis of the approached theme. This chapter is intended to strengthen the reader's understanding of the importance and impact of the topic studied, highlighting both the achievements and limitations of the research, as well as the future directions of exploration. Through the final conclusions, we aim to reaffirm the relevance of the discussed issue and to emphasize the contribution made to the respective field, while offering an integrated perspective on the entire study.

Superior performance and properties: Experimental studies have shown that biofuels, especially oxygenated ones, ensure a more complete combustion due to the oxygen content, offering comparable or even superior performance to diesel in terms of power and engine torque. In addition, biofuels exhibit superior lubrication properties, resulting in reduced wear and tear on the engine's injection components, thus extending their lifespan.

Thermal efficiency and specific fuel consumption: Turpentine-diesel blends have seen an increase in thermal efficiency and a decrease in specific fuel consumption (BSFC), indicating increased engine efficiency. In contrast, the use of eucalyptus-diesel blends led to a decrease in thermal efficiency and an increase in BSFC, suggesting less efficient combustion compared to pure diesel.

Pollutant emissions: The use of biofuels has had a significant impact on pollutant emissions. Eucalyptus-diesel oil and turpentine-diesel mixtures significantly reduced smoke emissions, and CO_2 emissions were lower for eucalyptus-diesel mixtures and relatively constant for turpentine-diesel mixtures. Emissions of nitrogen oxides (NOx) also decreased in the case of eucalyptus-diesel mixtures, due to lower combustion temperatures and the presence of specific compounds in eucalyptus oil that inhibit the formation of NOx.

Unburned hydrocarbons (HC): Unburned hydrocarbon emissions increased at low engine speeds for biofuel blends, due to incomplete combustion at these revs. However, at higher revs, HC emissions have decreased, especially for eucalyptus-diesel blends, due to improved turbulence and the air-fuel mixture in the combustion chamber, which favours more complete combustion.

4.2 Personal contributions

During the first year of doctoral study, I focused on conducting an exhaustive review of the literature. This stage had as its main objective the identification of relevant research methodologies, the evaluation of previous works in the field and the synthesis of the results obtained.

Research methodology

In order to effectively structure and organize the literature review, we have adopted a systematic methodology, based on the following essential steps:

1. **Identifying relevant sources**: I used academic databases such as PubMed, Scopus and Google Scholar, Science Direct to find articles, conference papers and books relevant to my field of research. We have defined specific keywords and inclusion and exclusion criteria to select the most relevant works.

2. **Critical literature review**: Each identified paper has been critically evaluated, emphasizing the research methodologies used, the validity and reliability of the data, as well as the relevance of the results obtained for my research topic.
3. **Synergy of results**: We synthesized and integrated the results of previous studies, identifying both convergences and divergences between them. We have outlined the strengths and limitations of each paper in order to better understand the current landscape of research in the field.

Previous work in the field

The literature review revealed a significant number of relevant studies, which can be classified into several distinct categories:

- **Theoretical studies**: Works that provide a solid theoretical foundation for the research topic, including conceptual models and working hypotheses.
- **Empirical studies**: Research that used experimental or observational data to test hypotheses and validate theoretical models.
- **Reviews and meta-analyses**: Studies that have analyzed and synthesized the results of a large number of individual research to provide an overview of progress and trends in the field.

Results achieved

Through this review I was able to identify several promising research directions and significant gaps in the literature:

- **Innovative methodologies**: We have noticed an increased interest in the development of new methodologies, capable of addressing the complexity and dynamics of the studied phenomena.
- **Conflicting results**: In certain subdomains, the results of studies are contradictory, suggesting the need for further research to clarify these inconsistencies.
- **The need for interdisciplinarity**: Many studies have highlighted the benefits of interdisciplinary approaches, which combine perspectives and methods from various fields to gain a more comprehensive understanding of research problems.

Research Methodology: Experimental Part

In the second year of my doctoral study, I carried out the experimental activities at the University of Minho in Portugal, under the guidance of Professors Jorge Martins and Francisco Brito. This period was essential for establishing the research methodology and for implementing the practical stages of the experiments, as follows:

1. **Establishing the Research Methodology**
   Under the guidance of Professors Jorge Martins and Francisco Brito, I developed a rigorous research methodology, adapted to the specifics of my experiment. I have defined in detail the experimental parameters and calibration procedures, ensuring that every aspect of the methodology is in line with international academic and scientific standards.
2. **Sensor calibration**
   We performed precise calibration of the sensors, an essential process to ensure the accuracy of the data collected. We used calibration methods for each type of sensor, adapting them to the specific conditions of our experiment. This stage included repeated adjustment and testing of the sensors to guarantee minimal error in data recording.

3. **Operation of the experimental engine**
   I was able to get the experimental engine up and running, a crucial step in conducting the research. This involved checking and adjusting all mechanical and electronic components, ensuring the optimal operation of the engine in safe and efficient conditions.

Numerical Modeling: Simulation of the Zero-Dimensional Internal Combustion Engine

As part of the activities carried out in Romania, at the Lower Danube University, we carried out the numerical modeling of the zero-dimensional internal combustion engine. This stage included the development and implementation of the numerical model, the validation of empirical data and the writing of the scientific paper. His personal contributions in this direction are detailed below:

1. **Numerical modeling**
   We developed a zero-dimensional numerical model of the internal combustion engine, using advanced simulation and programming techniques. Zero-dimensional numerical modeling involves reducing the engine to a system of differential equations that describe the thermodynamic and kinetic behavior of processes. This involved:
   - **Formulation of the fundamental equations**: We have derived the energy balance and mass equations that govern the operation of the engine, integrating the relevant operating parameters, such as pressure, temperature and volume.
   - **Implementation of solving algorithms**: We used efficient numerical algorithms to solve the differential equations, ensuring the convergence and stability of the solutions obtained.
2. **Empirical data validation**
   The validation of the numerical model was an essential step to ensure the accuracy and reliability of the simulations. We compared the results obtained from the numerical model with the empirical data collected in previous experiments. This validation included:
   - **Comparison of performance parameters**: We compared the performance parameters, such as thermal efficiency, power developed and specific fuel consumption, obtained by simulation with those measured experimentally.

4.3 Limitations of the study

Types of fuels studied: The study will focus only on a few specific types of alternative fuels (turpentine and eucalyptus oil mixed with diesel). Other alternative fuels, such as compressed natural gas (CNG) or synthetic fuels, will not be included in this research.

Experimental scale: Experiments are carried out on small and medium-sized engines, used in passenger vehicles and light vans. Heavy-duty vehicle engines and industrial applications are only discussed theoretically and will not be tested experimentally.

Technical aspects addressed: The research will focus on the combustion and emissions performance of alternative fuels. Other aspects, such as production costs, logistics, large-scale availability or the impact on existing infrastructure, will only be mentioned briefly and will not be analysed in detail.

Chapter 5. Future research directions

The field of internal combustion engines and biofuels continues to be a dynamic and highly relevant one, given the need to reduce environmental impact and dependence on fossil fuels. Future research directions focus on technological and methodological innovations that improve engine performance, optimize the use of biofuels and promote sustainability. These directions include:

1. Optimization of Combustion Processes

- **Improving thermal efficiency:** Future research focuses on optimizing combustion processes to increase the thermal efficiency of internal combustion engines. This can be achieved by implementing advanced combustion process control systems.

2. Development and Characterization of Biofuels

- **Second and third generation biofuels:** Research should be directed towards second generation biofuels (obtained from lignocellulosic biomass, straw, paper, etc.) and third generation (biofuels obtained from algae). These biofuel sources have considerable potential for reducing greenhouse gas emissions and avoiding competition with food resources [44].

3. Hybrid Systems Integration and Partial Electrification

- **Hybrid engines:** Future research explores the integration of internal combustion engines into hybrid systems, which combine the internal combustion engine with an electrical system.
- **Partial electrification:** Studies on the partial electrification of vehicles, such as start-stop systems and brake energy recovery, are essential to reduce fuel consumption and pollutant emissions.

4. Advanced Modeling and Simulation

- **3D multi-dimensional modeling:** The development and use of advanced multi-dimensional models for simulating combustion processes and complex interactions in internal combustion engines is a promising research direction. These models can help optimize engine design and operating parameters, providing a deep understanding of thermodynamic and kinetic phenomena.

5. Investigation of advanced materials

- **High-temperature resistant materials (ceramic coatings):** Research into the development and use of advanced materials that can withstand high temperatures and pressures to increase the durability and efficiency of engines. These materials can include advanced metal alloys, ceramics, and composites.

6. Performance evaluation and optimization of real-time operating parameters

- Development of experimental stands for the evaluation of the performance of a spark-ignition/compression-ignition engine, as well as the evaluation of their pollutant emissions (Annex 2).

Eng. Chivu Robert-Mădălin
Alternative fuels in internal combustion engines

List of publications

Web of Science

1. **Chivu, R. M**., Martins, J., Popescu, F., Gonçalves, M., Uzuneanu, K., Frătița, M., & Brito, F. P. (2024). Assessment of Engine Performance and Emissions with Eucalyptus Oil and Diesel Blends. *Energies*, *17*(14), 3528. https://doi.org/10.3390/EN17143528
2. **Chivu, R. M.**, Martins, J., Popescu, F., Uzuneanu, K., Ion, I. V., Gonçalves, M., Codău, T.-C., Onofrei, E., & Brito, F. P. (2023). Turpentine as an Additive for Diesel Engines: Experimental Study on Pollutant Emissions and Engine Performance. *Energy 2023, Vol. 16, Page 5150*, *16*(13), 5150. https://doi.org/10.3390/EN16135150
3. Mocanu, G., Iosifescu, C., Ion, I. V., Popescu, F., Frătița, M., & **Chivu, R. M.** (2024). Energy Analysis of Waste Heat Recovery Using Supercritical CO2 Brayton Cycle for Series Hybrid Electric Vehicles. *Energies* , *17*(11). https://doi.org/10.3390/EN17112494

Scopus/BDI

4. Francisco P. Brito **Chivu Robert** Joaquim da Costa Maria Margarida Gonçalves Jorge Martins. (n.d.). *Sustainable Fuels Workshop Testing Alternative Fuels and Internal Combustion Engines at MEtRICs LaMoTA Test facilities*. https://doi.org/DOI: 10.13140/RG.2.2.20499.86567
5**. Chivu Robert**, Jorge Martins, Florin Popescu, Ion V. Ion and Fratita Michael. (n.d.). *MIXTURES OF ESSENTIAL OILS AND ALCOHOLS WITH DIESEL OIL USE AND IMPACT ON ENGINE PERFORMANCE AND POLLUTANT EMISSIONS. A REVIEW*. https://doi.org/DOI: 10.36910/conf_avto.v1i1.1395
6. Popescu, F., **Chivu, R. M.**, Uzuneanu, K., & Ion, I. (2022a). Design and finite element analysis of a circular steel profile frame using Autodesk Inventor. *IOP Conference Series: Materials Science and Engineering*, *1262*(1), 012069. https://doi.org/10.1088/1757-899X/1262/1/012069
7. Popescu, F., **Chivu, R. M.**, Uzuneanu, K., & Ion, I. (2022b). Design and finite element analysis of a lower arm using CATIA V5. *IOP Conference Series: Materials Science and Engineering*, *1262*(1), 012070. https://doi.org/10.1088/1757-899X/1262/1/012070
8. Ion V. Ion, Răzvan Mahu, Florin Popescu, Gabriel Mocanu, **Robert Chivu** PREDICTION OF TAR FORMATION IN A BIOMASS HEATING BOILER // Вестник КГЭУ. 2023. No1 (57). URL: https://cyberleninka.ru/article/n/prediction-of-tar-formation-in-a-biomass-heating-boiler (дата обращения: 29.07.2024).

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Annexes

Annex 1

pressure (displayed in [hPa])(average engine speed,injection quantity for desired value calculation)/hPa

mg/hub \ rpm	750,00	1000,00	1250,00	1750,00	2000,00	2100,00	2200,00	2300,00	2400,00	2500,00	2600,00	2700,00	2800,00	2900,00	0,00	300,00
0,0000	1020,00	1200,00	1200,00	1108,00	1245,00	1202,00	1159,00	1108,00	1155,00	1499,00	1540,00	1544,00	1443,00	1364,00	1463,00	1386,00
3,0000	1020,00	1200,00	1200,00	1167,00	1294,00	1268,00	1192,00	1103,00	1155,00	1509,00	1538,00	1540,00	1539,00	1414,00	1463,00	1396,00
6,0000	1020,00	1020,00	1200,00	1250,00	1331,00	1314,00	1216,00	1250,00	1158,00	1190,00	1507,00	1547,00	1589,00	1456,00	1374,00	1503,00
8,0000	1020,00	1020,00	1200,00	1303,00	1383,00	1358,00	1265,00	1294,00	1156,00	1190,00	1516,00	1546,00	1581,00	1549,00	1424,00	1513,00
10,0000	1020,00	1020,00	1035,00	1341,00	1479,00	1392,00	1307,00	1295,00	1290,00	1208,00	1225,00	1522,00	1581,00	1600,00	1462,00	1385,00
12,0000	1030,00	1020,00	1035,00	1334,00	1656,00	1419,00	1338,00	1345,00	1311,00	1200,00	1225,00	1524,00	1579,00	1594,00	1555,00	1431,00
14,0000	1040,00	1020,00	1024,00	1070,00	1757,00	1468,00	1357,00	1388,00	1313,00	1328,00	1261,00	1255,00	1575,00	1602,00	1610,00	1466,00
16,0000	1050,00	1030,00	1019,00	1070,00	1876,00	1648,00	1386,00	1456,00	1357,00	1343,00	1252,00	1255,00	1553,00	1611,00	1604,00	1558,00
18,0000	1060,00	1040,00	1014,00	1058,00	1090,00	1731,00	1411,00	1503,00	1394,00	1326,00	1356,00	1273,00	1280,00	1604,00	1613,00	1614,00
20,0000	1080,00	1050,00	1007,00	1053,00	1090,00	1751,00	1576,00	1502,00	1478,00	1371,00	1355,00	1279,00	1280,00	1577,00	1625,00	1615,00
22,0000	1150,00	1060,00	1026,00	1047,00	1069,00	1090,00	1611,00	1513,00	1529,00	1404,00	1337,00	1384,00	1305,00	1335,00	1616,00	1617,00
24,0000	1200,00	1080,00	1040,00	1080,00	1073,00	1090,00	1611,00	1519,00	1526,00	1489,00	1384,00	1370,00	1301,00	1335,00	1587,00	1622,00
26,0000	1200,00	1150,00	1052,00	1137,00	1105,00	1078,00	1120,00	1497,00	1530,00	1538,00	1414,00	1350,00	1393,00	1345,00	1350,00	1612,00
28,0000	1200,00	1200,00	1076,00	1203,00	1139,00	1082,00	1120,00	1502,00	1535,00	1534,00	1498,00	1405,00	1386,00	1368,00	1350,00	1582,00

injection mass per cycle (mg/cycle)(average engine speed,inner torque set value)/mg/cyc

Nm \ rpm	250,00	500,00	750,00	1000,00	1250,00	1500,00	1750,00	2000,00	2250,00	2500,00	2750,00	3000,00	3500,00	4000,00	4500,00
0,000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0000	0,0
10,000	2,4000	2,3000	2,2000	2,1000	2,0000	1,9000	1,9000	1,9000	1,9000	2,0000	2,1000	2,1000	2,6000	3,0000	3,3
20,000	4,6000	4,4000	4,2000	4,0000	3,6000	3,4000	3,2000	3,2000	3,2000	3,2000	3,3000	3,7000	4,7000	5,3000	5,8
35,000	7,2000	6,7000	6,1000	7,0000	7,3000	6,4000	6,0000	7,0000	6,5000	7,0000	6,3000	7,4000	7,5000	8,4000	8,8
50,000	10,6000	10,1000	9,5000	9,5000	8,8000	8,7000	7,8000	8,8000	8,2000	9,3000	9,0000	9,8000	9,5000	10,4000	11,3
75,000	16,1000	15,3000	14,4000	14,5000	13,1000	13,0000	11,9000	12,6000	12,3000	14,7000	12,3000	14,2000	13,9000	13,9000	14,4
100,000	22,2000	21,3000	20,1000	19,6000	17,9000	17,5000	15,9000	16,6000	16,0000	18,5000	16,8000	17,9500	19,3000	18,2000	18,0
125,000	28,2000	27,0000	25,5000	22,6000	23,3000	22,4000	20,9000	20,6000	20,3000	22,4000	19,9000	22,5000	22,7000	22,9000	22,7
150,000	34,3000	32,9000	30,9000	28,3000	28,8000	27,3000	27,2000	26,6000	25,4000	26,3000	24,4000	27,1000	27,2000	27,9000	27,5
175,000	40,5000	38,9000	36,5000	33,4000	30,4000	31,3000	30,3000	30,0000	30,6000	30,1000	30,1000	31,0000	31,1000	32,5000	32,5
200,000	46,5000	45,0000	42,6000	39,6000	36,0000	36,3000	34,3000	34,3000	34,8000	34,8000	33,7500	33,5000	34,3000	36,3000	39,1
225,000	52,6000	50,9000	48,2000	44,5000	40,8000	38,6000	37,4000	36,9000	36,5000	36,9000	37,5000	38,1000	39,9000	42,8000	45,5
240,000	58,6000	56,9000	53,0000	48,0000	44,1000	41,7000	40,3000	40,0000	40,3000	40,7000	41,7000	42,4000	44,5000	47,6000	49,7
275,000	65,0000	62,9000	59,6000	55,7000	53,0000	50,6000	49,0000	48,5000	48,5000	49,0000	50,0000	51,6000	54,0000	55,7000	57,9
300,000	70,2000	68,1000	64,6000	61,8000	58,8000	55,8000	54,7000	55,0000	55,8000	56,8000	58,2000	59,3000	61,9000	63,7000	64,6
450,000	105,4000	102,2000	96,9000	91,7000	87,4000	84,9000	84,0000	84,5000	86,4000	88,6000	90,9000	93,3000	98,9000	102,3000	105,0

crankshaft angle(average engine speed,current injection quantity)/deg Cr

mg/cyc \ rpm	200,00	400,00	800,00	1000,00	1250,00	1500,00	1750,00	2000,00	2250,00	2500,00	2750,00
0,0000	0,0000	0,0000	1,9219	3,3281	4,2891	5,1797	5,6719	5,8125	5,6484	5,4609	5,7891
1,7000	0,0000	0,0000	1,8750	3,3750	3,8437	4,2891	4,6406	5,0859	5,2266	5,3203	5,6953
3,4000	0,0000	0,0000	1,8750	3,5156	3,5859	3,7969	4,1484	4,4297	4,8047	5,3203	6,0937
5,1000	0,0000	0,0000	-0,1406	-0,7031	-2,8828	-3,5391	-2,7422	-1,2891	0,7734	2,2969	3,6094
6,8000	0,0000	0,0000	-0,0937	-0,4687	-2,5547	-3,3281	-2,6250	-0,8672	1,1953	2,8125	4,1250
8,5000	0,0000	0,0000	-0,0469	-0,1875	-1,8516	-2,5547	-1,3594	0,2344	1,8750	3,4219	4,8516
9,5000	0,0000	0,0000	0,0000	0,0234	0,0937	0,0937	0,3047	0,8906	2,4141	3,9609	5,3906
11,0000	0,0000	0,0000	0,0234	0,0703	0,5625	1,0547	1,6172	2,3203	3,3047	4,7344	6,3281
12,9000	0,0000	0,0000	0,0234	0,0937	0,8437	1,4531	2,0156	2,7422	3,9375	5,4844	7,0781
14,6000	0,0000	0,0000	0,0234	0,0937	0,9141	1,6875	2,2734	3,1172	4,3828	6,1172	7,6875
18,0000	0,0000	0,0000	0,0000	0,6094	1,2656	1,9687	2,6016	3,3750	4,6875	6,4922	8,3437
21,4000	0,0000	0,0000	0,0469	0,7266	1,6875	2,2500	2,9062	3,7500	5,5547	7,5469	9,4922
24,8000	0,0000	0,0000	0,3047	1,4766	2,3672	3,2109	4,3359	5,8594	7,4766	9,3047	11,2969
28,5000	0,0000	0,0000	0,4219	1,9922	4,6641	4,9922	5,6016	6,6328	8,1328	9,8906	12,5391
30,5000	0,0000	0,0000	0,4922	1,9922	5,8828	4,9922	6,0000	7,0547	9,2578	11,5078	12,7500
32,5000	0,0000	0,0000	0,4219	1,9922	6,4922	4,9922	6,0000	7,9922	10,0078	12,0000	13,0078

rail pressure in hPa(average engine speed,current injection mass)/hPa

mg/cyc \ rpm	600,00	750,00	850,00	1000,00	1250,00	1500,00	1750,00	2000,00	2250,00	2500,00	2750,00	3000,00	3250,00	3500,00	4000,00	4500,00
0,0000	220000	253000	275000	333000	368100	428100	462400	450000	435300	438300	402800	394600	394800	395800	396400	411600
1,7000	220000	253000	275000	342000	388100	441100	459000	397500	355500	341300	336000	382500	404800	405800	411400	426600
3,4000	220000	253000	275000	313000	420000	485000	487000	423000	369300	331500	321000	394500	414800	415800	416400	436600
5,1000	220000	253000	275000	353000	450000	500000	504500	504500	393500	360000	340500	409500	429800	426100	421400	446600
6,8000	253100	253100	275200	415800	535000	598000	608500	608500	508000	433500	411000	433500	444000	433100	428300	452900
8,5000	309200	309200	342000	476600	620000	668000	674000	674000	580000	538500	522500	493500	466000	443800	434400	455500
10,2000	367800	367800	413000	496000	645000	770700	805000	805000	780300	758000	676000	540100	477500	454000	442500	462900
11,9000	384900	384900	429000	522300	666700	845300	906000	906000	886000	842000	765000	605000	543300	515000	514300	524000
13,6000	395500	395500	439000	543000	610000	830000	936700	947000	926000	890000	833000	728000	680000	656700	627300	625000
15,5000	475000	475000	505800	542000	676000	830000	943300	975000	963000	947000	898000	837000	792000	777000	763300	755000
18,4000	572600	572600	572600	572600	655000	850000	975000	1036700	1048300	1033300	996000	940500	937500	955000	970000	980000
22,8000	588000	588000	588000	588000	720000	950000	1075000	1136700	1170000	1181700	1165500	1073000	1013000	1013000	1033000	1070000
30,0000	588000	588000	588000	588000	820000	1050000	1175000	1236700	1266700	1323300	1295000	1138000	1110000	1099000	1125000	1180000
33,0000	588000	588000	588000	588000	820000	1050000	1175000	1236700	1270000	1333300	1338000	1220000	1195000	1190000	1260000	1380000
37,0000	588000	588000	588000	588000	820000	1050000	1175000	1236700	1270000	1333300	1350000	1360000	1385000	1420000	1600000	1600000
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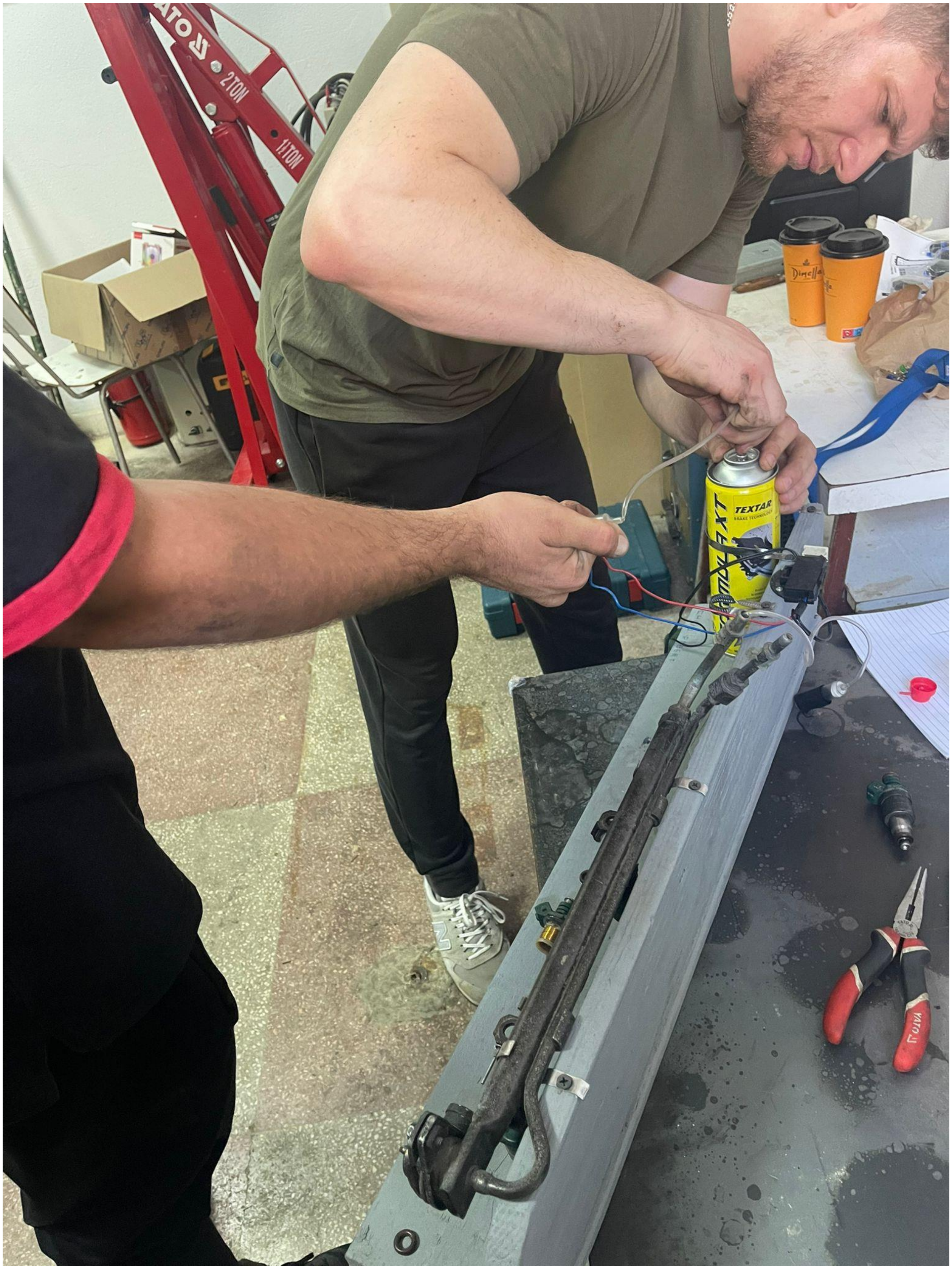
Annex 2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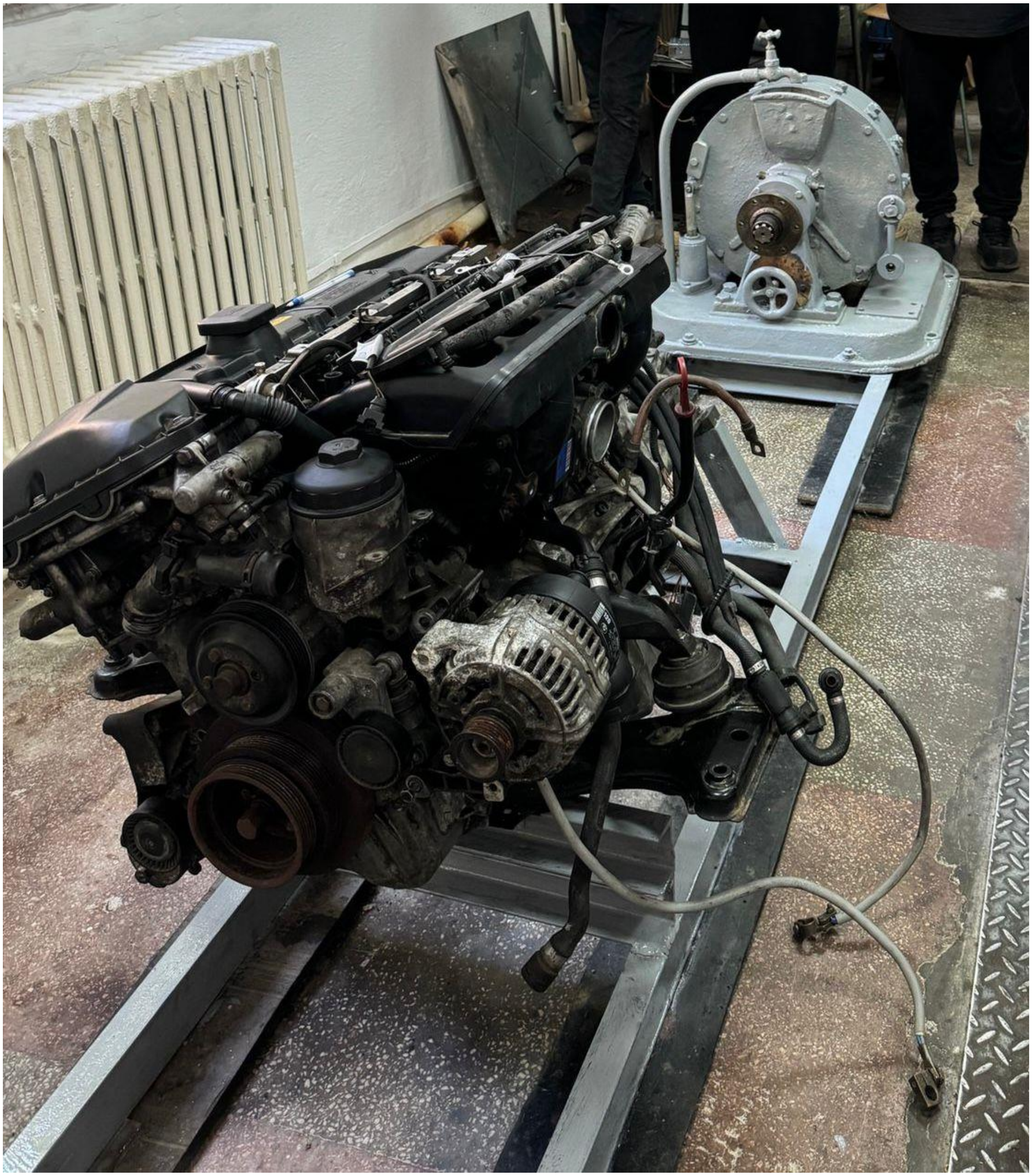
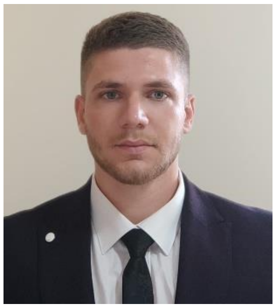


Eng. Chivu Robert-Mădălin
Alternative fuels in internal combustion engines

PERSONAL INFORMATION

Chivu Robert-Mădălin



0724874968

robert.chivu@ugal.ro

Sex M | Date of birth 15/03/1997 | Romanian Nationality

EXPERIENCE

PROFESSIONAL

01.10.2023-present Assistant Professor

01.11.2022- present Member of MEtRICs research center: Uminho, Guimarães, Portugal

01.09.2022-01.09.2023 Research activity at the University of Minho in Portugal, Guimaraes, The laboratory dies with internal combustion.

01.10.2021-present PhD student at the Faculty of Engineering, form of financing, scholarship budget, with teaching activity, "Dunărea de Jos" University, Galati

Doctoral School of Mechanical and Industrial Engineering

PhD supervisor: Prof. Dr. Ing. Florin Popescu, Prof. Dr. Eng. Jorge Martins

Theme: Alternative fuels in internal combustion engines.

Thermal Systems and Road Vehicles Department.

Domain- Mechanical Engineering.

EDUCATION AND TRAINING

2021-present – PhD

"Dunărea de Jos" University of Galati,

Doctoral School of Mechanical and Industrial Engineering

PhD supervisor: Prof. Dr. Eng. Florin Popescu,

Prof. Dr. Ing. Jorge Martins

2019-2021 – Master's degree.

"Dunărea de Jos" University of Galati

Faculty of Engineering - Modeling and Simulation Specialization in Engineering

2015 – 2019 – Bachelor's studies.

"Dunărea de Jos" University of Galati

Faculty of Engineering - Road Vehicles Specialization

2011-2015

"Railway Transport" High School Galati

Mother tongue | Romanian

Other Known Foreign Languages	THE CHOICE		SPEECH		WRITING
	Obedience	Read	Participation in Conversation	Oral Discourse	
English Language B2	B2	B2	B2	B2	

Digital skills

SELF ASSESSMENT				
Processing Information	Communication	Creation of Content	Security	Resolution of Problems
User experienced	User experienced	User experienced	User experienced	User experienced

Competent staff

Skills

Knowledge of using digital tools in teaching/learning.

Experience working in multicultural teams.

Good communication skills following national competitions and scientific sessions, conferences where I participated.

Teamwork.

Good adaptability to difficult working conditions.

Computer skills

AVL
Microsoft Teams, Zoom, Skype platform
Diesel RK
National Instruments
LabView
Microsoft Office suite
Autodesk Inventor
Autodesk AutoCAD
CATIA V5
SolidWorks
Ansys

Driver's license

- Categories AM, B1, B, C1, C, BE, C1E, CE

CR