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PhD THESIS

RESEARCH ON THE USE OF BUTANOL IN AUTOMOTIVE SPARK IGNITION ENGINES

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Contents

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Introduction

The PhD thesis entitled *RESEARCHES REGARDING THE BUTANOL USE AT THE AUTOMOTIVE SPARK IGNITION ENGINE (RESEARCHES REGARDING THE BUTANOL USE AT THE AUTOMOTIVE SPARK IGNITION ENGINE)* is part of the concerns of the research team of the Department of Thermotechnics, Engines, Thermal and Refrigeration Equipment (TMETF), Faculty of Mechanical and Mechatronics Engineering (FIMM) of the National University of Science and Technology POLITEHNICA Bucharest. The thesis contains original contributions in the field of research on the use of butanol in automotive spark ignition engines. The paper also deals with thermo-geodynamic modelling and simulation of the spark ignition engine starting from existing libraries of the AMESim software tool to study the influence of the use of butanol-gasoline blends in engine fuelling.

Chapter 1 gives a general overview of the international context that has led to the need for the use of alternative fuels in spark ignition engines; some relevant studies on the results obtained by other authors when fuelling spark ignition engines with butanol-gasoline blends are also presented. The chapter also gives a brief description of the properties of butanol and the technologies for its production. A great advantage of butanol is that it can be transported using existing infrastructure.

Chapter 2 provides a detailed investigation of the current state of research into the use of butanol in automotive spark ignition engines. This chapter presents studies related to the influence of butanol on combustion, cylinder pressure, combustion stability, specific fuel consumption, volumetric efficiency, thermal efficiency and pollutant emission concentrations. The studies presented used butanol in percentages ranging from 2.5% to 100%.

Chapter 3 describes the test stand, equipment and procedure for carrying out the experimental investigations. The results obtained are presented starting with the 10% butanol-gasoline blend at rich blends and at lean blends respectively with the 15% butanol-gasoline blend. Reference results are established when the engine is fuelled with gasoline. Maximum pressure, mean indicated pressure, rate of pressure rise during combustion, coefficients of variability of maximum pressure and mean indicated pressure, angles at which 5%, 50% and 90% of the heat is released, variation of heat release rate, combustion laws, specific energy consumption, thermal efficiency and pollutant emission concentrations are given.

Chapter 4 deals with the description of the modelling and simulation process as well as the presentation of the model from which the thermo-geodynamic modelling of the spark ignition engine is derived. The modelling work is divided into two main parts 1) presentation of the main sub-models as well as the underlying equations and 2) parameterisation of the model. At the end of the chapter a brief comparison is made between experimental results and those obtained by numerical simulation using gasoline.

Chapter 5 gives a more detailed comparison between experimental and numerical simulation results for gasoline and butanol-blends. The successful completion of the fitting, parameterisation and validation steps of the mathematical model makes it possible to further analyse the use of butanol in automotive spark ignition engines. The chapter presents the indicated diagrams, the maximum pressure, the variation of the rate of pressure rise during combustion, the variation of the rate of heat release and the combustion laws at rich and lean mixtures.

Chapter 6 presents the final conclusions of the PhD thesis i.e. personal contributions and next directions.

The paper concludes with a list of published works in extenso and bibliographical references in the order of their citation in the text.

Thank you

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The author

CHAPTER 1
THE RELEVANCE OF THE RESEARCH TOPIC. OBJECTIVES OF THE WORK

1.1 Introduction relevance of the research topic

The global energy market provides consumers with about 370 exajoules of energy per year, which is equivalent to about 170 million barrels of oil per day or about 11.73 terawatts (TW) per hour, [1]. About 95% of this energy comes from fossil fuels. It is estimated that about 10% of this total energy is produced by biofuels. Biofuels are presented as a sustainable and renewable solution compared to fossil fuels, [2], [3], [4], [5], [6], [7], [8]. Biofuels can be successfully used in today's internal combustion engines without design modifications. The process of obtaining new biofuels is relatively simple and inexpensive, and consists of mixing these renewable sources (acetone, ethanol, butanol, etc.) with fossil fuels. The process is cheap and simple compared to obtaining biofuels through chemical treatments,[13].

In December 2018, Directive (EU) 2018/2001, known as "RED II", was adopted, requiring at least 32% of energy consumption to come from renewable sources by 2030. In the transport sector the share of renewable energy should be 14%, [14]. In 2005, a similar program was launched in the United States called the Renewable Fuel Standard (RFS), which requires that all transportation fuels contain a minimum amount of renewable fuel. This minimum amount must be increased annually, [15].

The use of butanol as a transport fuel can reduce fossil fuel consumption by 39-56% and greenhouse emissions (CO_2) by up to 48% over a vehicle life cycle. Butanol's main problem delaying its use in large-scale internal combustion engines is its relatively high production cost, [16], [17]. The amount of bio-butanol obtained by ABE (acetone-ethanol-butanol) fermentation is relatively low, about 12 - 18 g/L, [17], [18].

A study carried out by Alasfour on a single-cylinder engine using a 30% butanol-gasoline blend resulted in a 7% decrease in engine power compared to running it on gasoline,[23],[24]. In a study conducted by Dernotte, an optimal blend of 40% butanol-gasoline was determined to minimise unburned hydrocarbon emissions in an indirect injection spark ignition engine,[25]. At concentrations above 40% butanol an increase in hydrocarbon emissions was observed. Additivation of gasoline with butanol, even at low concentrations, resulted in stabilisation of combustion especially at lean blends by reducing the ignition delay of the fuel, but without affecting the main combustion duration. This results in a similar laminar flame speed for gasoline and butanol. But it was also observed that the addition of butanol, even in small percentages can reduce the coefficient of variability of the indicated mean pressure. Laminar flame speed is a fundamental property of the air-fuel mixture, necessary to validate the chemical reaction mechanism for a better understanding of the turbulent combustion process in spark ignition engines. Many studies have determined the laminar flame speed for air-ethanol mixture, both numerically and experimentally, few studies have focused on determining the laminar flame speed for air-butanol mixture, [26].

Butanol has certain advantages over methanol. Butanol or butyl alcohol can be used to fuel spark ignition engines with minor modifications. Butanol is less corrosive and can be transported using existing infrastructure, [29]. It is much less hygroscopic than methanol so it cannot contaminate from water. Butyl alcohol also has a higher energy density than methanol and is miscible with gasoline. Butanol also has a lower heating value much closer to that of gasoline than ethanol, a higher stoichiometric coefficient making it more compatible with gasoline and current lambda control strategies. An advantage would be the simplicity of using butanol as an additive for existing fleets of vehicles, [29]. In short, the physical properties of butanol are closer to those of gasoline than ethanol.

The disadvantage of butanol compared to gasoline is the much higher latent heat of vaporisation. For indirect valve gate injection systems, as the fuel vaporises, cooling of the fresh mass entering the engine also occurs. This increases the density of the fuel mixture and also the charge mass. The cost of producing butanol is higher than methanol. The physical properties of butanol lead to a weaker spray of the fuel jet when it is injected, [30]. For the use of methanol and butanol as alternative fuels in the transport sector, it is important to improve their characteristics. One possible method is to combine methanol and butanol because the disadvantages of methanol would be reduced by butanol and those of butanol by methanol. However, this mixture should be investigated before being recommended, as both methanol and butanol have different thermodynamic properties and combustion characteristics, [30].

A ternary fuel mixture concept was investigated in the Turner study. However, his work did not show any improvements in the performance/emissions of the new fuel mixture on the spark ignition engine,[31]. Nazzal investigated the effects of gasoline-ethanol-methanol blends on the engine,[32]. He made various measurements at different operating points of the engine. The fuel mixture was made of 6% - ethanol, 6% - methanol and 88% - gasoline. The results obtained were compared with those obtained when the engine was fuelled with gasoline. This study showed that the ternary fuel mixture resulted in improved spark ignition engine performance. A 2016 study by Rodriguez-Anton et al. concluded that izobutanol has advantages over ethanol in terms of energy density, air/fuel ratio, vapour pressure and in terms of renewable content, [33]. In a study carried out by Balaji he examined different iso-butanol-ethanol-gasoline combinations in different percentages, e.g. 10% ethanol, 2.5% iso-butanol, 10% ethanol and 5% iso-butanol, 10% ethanol and 7.5% iso-butanol, [34]. It has shown that ternary blending can improve engine performance by reducing pollutant emissions compared to gasoline. But fuel consumption increased significantly compared to gasoline. In another study, the effects of using an ethanol-methanol-gasoline blend were investigated, with results showing improved engine performance and reduced emissions compared to gasoline, [35]. In a study led by Elfasakhany, better performance and lower emissions were achieved using blends of ethanol and methanol (3-10% by volume) in gasoline, [36]. The same Elfasakhany compared the results, performance and emissions obtained between a ternary blend of bioethanol-iso-butanol-gasoline and a dual blend of iso-butanol-gasoline, [37]. These two different mixtures powered a motorcycle engine. The ternary bio-ethanol-iso-butanol-gasoline blend improved thermal engine performance (power, torque and volumetric efficiency) over the dual iso-butanol-gasoline blend. Engine performance was slightly lower than when using gasoline. On the emissions side, lower values were obtained for unburned HC and CO by 15% and 20% respectively compared to gasoline and by 9% and 14% compared to the iso-butanol-gasoline blend. Elfasakhany also conducted a study examining an n-butanol-iso-butanol-gasoline blend and its effects on engine performance and emissions, [38]. The results were compared with those obtained with izobutanol-gasoline. The conclusion of the study was to recommend the use of the ternary blend over the binary or gasoline blend. The effects of n-butanol and iso-butanol fuelling of a single-cylinder HCCI engine were studied by J. Hunter,[39]. Improvements in the combustion process were observed and n-butanol was much more stable than iso-butanol at lean blends. In another study by Siwale he investigated a methanol-n-butanol-gasoline blend (53% methanol, 17% n-butanol and 30% gasoline) fuelling a spark ignition engine and compared the results with a dual blend of methanol-gasoline in different percentages (70% vol. methanol -30% vol. gasoline, M70, and 20% vol. methanol with 80% vol. gasoline, M20) and gasoline, [40]. Emissions of unburned hydrocarbons (HC) are lowest for the ternary blend compared to the binary blend and gasoline. Also, the ternary blend obtained higher emissions of CO, NO_x and CO_2 than the binary blend.

A 2022 study by Tyler Lark and Nathan P. Handricks looked at the effect of carbon emissions in corn ethanol production, [41]. The study is similar to that of conventional fuel production (well-to-wheel). The study concluded that in terms of carbon emissions, corn-based ethanol production is 24% more polluting than gasoline production. In addition, this ethanol production process can impact water quality, soil quality and surrounding ecosystems. The authors conclude that the bio-ethanol production process has failed to meet its carbon reduction target. This study contradicts another 2022 study published by Jan Lewandrowski, [42]. The study concluded that ethanol production from corn could reduce carbon emissions by 39 to 43% compared to gasoline production. Alcohols can be considered a good alternative solution as numerous studies have observed improved engine efficiency, reduced energy consumption and lower pollutant emissions, [43], [44]. The use of butanol can reduce the pressure on parts manufacturers because it is less corrosive compared to ethanol and thus can be used without problems in current fuelling and ignition systems, [45].

Yacoub et al. quantified the energy performance and pollutant emissions for a spark ignition engine optimized to run on alcohol (C_1- C_5)/gasoline blends, [59]. The ethanol-gasoline blend had the greatest improvement in detonation resistance of all blends while the C_4- C_5chain alcohols had lower detonation resistance compared to gasoline. For example, in 2005, McEnally and Pfefferle investigated the flammability of the four butanol isomers by measuring their temperature and chemical composition,[60]. Among other findings, the authors found that all four butanol isomers produced much higher concentrations of aldehydes and ketones as intermediate species than butane, which is relevant to the formation of pollutants when butanol is used as a fuel. In studies by Oßwald et al. in 2011 and Frassoldati in 2012 they identified 57 intermediate compounds in the flames of the four butanol isomers, [61], [62]. There were also significant variations in the group of intermediate compounds between butanol isomers, suggesting that pollutant emissions may also vary with chemical properties and concentrations. Moss et al. investigated self-ignition and ignition delay of the four butanol isomers using shock tubes in 2008, [63]. Among their results, the authors found 1-butanol had the shortest ignition delay, followed by izobutanol while tert-butanol and 2-butanol were the least reactive.

1.2 Technologies for obtaining butanol

Researchers have focused on making biofuels from edible crops, [64]. This solution is not sustainable in the long term, the aim being to obtain high quality biofuels and other chemical compounds from cheap non-edible biomass, [65], [66], [67], [68], [69]. Thus, the researchers proposed the transformation of lignocellulosic biomass, most of which is agricultural waste. With the development of bio-refineries, the processing of lignocellulosic biomass has been industrialised, releasing monosaccharides that can be fermented into high-quality chemicals. Cellulosic izobutanol occupies an important place in the fuel sector, [70].

Oxygenated compounds such as butyl alcohol present themselves as an attractive alternative fuel solution in the transport sector. Butanol has certain blending advantages with gasoline such as low vapour pressure (reduced volatile emissions) higher energy content, high miscibility with gasoline blending without vehicle modification and reduced carbon monoxide (CO) and unburned hydrocarbon (HC) emissions, [39], [71], [72]. In addition, butanol has certain advantages over ethanol such as the ability to be blended in any percentage with gasoline and used in modern internal combustion engines,[73]while ethanol can be blended with gasoline in a proportion of about 10% without constructive changes,[74]. Butanol's auto-ignition temperature is 341ºC, close to gasoline's 280ºC which makes it easy to mix with gasoline, [75].

Izobutanol can also be obtained from fossil sources by hydroformylation (also called oxo synthesis) of propylene, followed by catalytic hydrogenation of the aldehydes formed in the corresponding alcohols, a method which simultaneously produces izobutanol as well as 1-butanol, [76]. A similar process would be the carbonylation of propylene, also known as the Reppe process, in which olefin, carbon monoxide and water react under pressure in the presence of a catalyst, a method by which izobutanol and 1-butanol are obtained directly from propylene, [76].

Butanol can also be obtained from renewable sources by anaerobic fermentation of sugar from biomass using solventogenic clostridia and is known as bio-butanol, [77]. Some researchers have started to develop an advanced technology to achieve cost-effective high production of bio-butanol. Currently, sucrose and starch-based feedstocks from agricultural crops such as maize, wheat, barley, rice straw, cassava, wood, etc. are extensively used as feedstock for bio-butanol because of the simple technology of the sugar fermentation process. Manioc has been used in the bio-butanol fermentation process resulting in 16.4-16.9 g/l butanol yielding a substrate of 0.26-0.35 g/g and a volumetric yield of 0.35-0.46 g/l/h, [78]. Increased use of bio-butanol will lead to a greater need for biomass feedstock from agricultural crops. Thus, food security will become the main issue for obtaining a sustainable source of bio-butanol.

1.3 Objectives of the PhD thesis

The general objective of the PhD thesis is to study the impact of fuelling an automotive spark ignition engine with a butanol-gasoline mixture in different percentages. The specific objectives of the thesis are:

- To update information on the current state of research in the thesis area.
- Study the impact of butanol on spark ignition engine energy performance.
- Study of the impact of butanol on spark ignition engine operating stability.
- Study of the impact of butanol on spark ignition engine emissions.
- Modelling the combustion processes in the cylinder of the spark ignition engine fuelled with gasoline and butanol-gasoline blends.
- Validation of the proposed physico-mathematical model.

CHAPTER 2
ANALYSIS OF THE CURRENT STATE OF RESEARCH IN THE FIELD

2.1 Influence of butanol on combustion

The maximum cylinder pressure for butanol is achieved slightly faster than for gasoline, indicating a higher combustion rate of butanol. Reducing the spark advance results in an overlap of both the maximum pressure value and the angle at which it is obtained similar to gasoline, [82].

The evolution of the pressure in the cylinder provides information on the cooling of the mass of fresh charge introduced into the cylinder. At a coolant temperature of 20°C, the gas pressure when using iso-octane is highest on the compression stroke of the piston between the closing of the inlet valve and the moment of electrical spark. The second highest pressure is that of gasoline followed by the two alcohols with very small differences. The higher latent heat of vaporisation of ethanol and butanol compared to iso-octane and gasoline can lead to much higher cooling of the fresh mixture compared to iso-octane, resulting in lower compression pressures, [82].

S. Szwaja et al. studied the ignition of butanol when used in a spark ignition engine. The authors used the following percentages of butanol blended with gasoline: 0%, 20%, 60% and 100%. The engine used is a CFR (cooperative fuels research) engine produced by Waukesha Motor Company, [83]. The engine has a variable compression ratio (CR). Burning is faster on 100% butanol. The explanation is that butanol burns much faster than gasoline. Measurements were made at a compression ratio (CR) of 10, constant advance of 10 CAD before TDC, average indicated pressure of 330 kPa and stoichiometric mixtures, [83]. It is observed that simple butanol releases 10% of its heat in about 16 CAD compared to gasoline in 17.6 CAD. A 10% reduction in duration is thus observed when fuelling with simple butanol.

A reduction in the angle at which 50% of the heat is released is also observed as the percentage of butanol in the mixture increases. The explanation would be the shorter ignition delay of the butanol. The initial phase of combustion is largely influenced by the laminar flame speed at fuel ignition. The other combustion phases are influenced by turbulent flame propagation. The difference between $\alpha_{50\%}$ of butanol and $\alpha_{50\%}$ of gasoline is 2°. The 50% angle represents half of the combustion process and engine momentum depends very much on this angle. In order to achieve maximum torque, the spark advance must be adjusted according to the percentage of gasoline replaced by butanol, [83].

A tendency for faster heat release is observed with butanol feed (shorter combustion time in the initial phase) but the maximum release rate remains similar in all cases investigated. The authors conclude that the addition of butanol does not significantly influence the combustion law nor the variation of the heat release rate. To optimise the combustion phases the authors recommend adjusting the spark advance according to the percentage of butanol in the mixture, [83].

2.2 Influence of butanol on cylinder pressure

The limited effect of the n-butanol-methanol-gasoline (nBM) mixture on the indicated mean pressure is observed. Using the 3% nBM3 blend, the maximum cylinder pressure is 8% lower than gasoline. Using a higher volume of alcohols in the gasoline blend does not produce a significant increase in pressure. The reduction in maximum effective pressure for nBM3 is due to the lower (lower compared to gasoline) heat of methanol and n-butanol. The increase in mean effective cylinder pressure when fuelled with nBM7 and nBM10 is due to improved volumetric efficiency and improved combustion, [30]. The 3% nBM3 mixture leads to a significant decrease in both engine torque and power, especially at higher engine speeds. This decrease is observed when fuelling with all mixtures investigated. The heat of vaporisation of alcohols is higher compared to that of gasoline resulting in a decrease in engine momentum and thus engine power, [30].

Tan Tien Huynh et al. studied the influence of butanol on cylinder pressure. The percentages of butanol used by the authors are 10%, 30% and 50%. Measurements were made at different loads and speeds. A gradual decrease of the maximum pressure is observed with increasing percentage of butanol in the mixture. For example, a decrease of 12% is observed, compared to 19% when fuelling the engine with Bu30 and Bu50. At high revs and 30% load there is a slight increase in maximum pressure at Bu10 and Bu30 between 6% and 8%. At butanol percentages higher than 30% there is a decrease in maximum pressure of 6% compared to Bu0, [88].

At high loads (70%) and speeds of 2250 rpm it is observed that the highest pressure is obtained at the gasoline feed. The maximum pressure obtained by butanol blends gradually decreases with increasing butanol concentration. For example, when feeding Bu10 the maximum pressure decreases by 11.5%, when feeding Bu30 the maximum pressure decreases by 13% and when feeding Bu50 it decreases by 17%. The explanation for these decreases would be the lower heating value of the gasoline-butanol blends. The same results are observed at speeds of 4250 rpm, with the maximum pressure decreasing with increasing butanol concentration. The flame propagation speed decreases with increasing percentage of butanol in the blend. The spark advance was the same for all engine speeds, [88].

Stanislaw Szwaja et al. studied the impact of butanol and glycerol (glycerol) on spark ignition motorcycle performance. The blends proposed by the authors are: gasoline 95 (used in European markets), n-butanol (99.9% purity) and n-butanol 75% - glycerol 25%, [89]. The authors concluded that the mixtures investigated did not result in deterioration of the mean effective pressure or the indicated efficiency. These parameters showed a slight improvement due to the lower exhaust gas temperature reducing heat losses.

In another study by Szwaja et al. the influence of butanol on cylinder pressure was investigated. The authors obtained the highest pressure by fuelling the engine with 100% butanol and the lowest pressure with straight gasoline. The maximum cylinder pressure increases with the percentage of butanol in the blend. Maximum pressure is achieved faster when fuelling with butanol because it burns faster than gasoline, [83].

2.3 Influence of butanol on combustion stability

It is observed that the addition of oxygen improves the combustion process by reducing the coefficient of variation (COV) of the indicated mean pressure. The improvement of the combustion process does not seem to depend too much on the concentration of butanol in the mixture formed, especially in stoichiometric mixtures, since a similar reduction of COV is observed both at 20% concentration and at 40%, 60% and 80%. The same cannot be said for gasoline-ethanol blends at higher percentages than 10%. A reduction in COV for the indicated average pressure can also be obtained when blending gasoline with 10% ethanol, but at ethanol concentrations higher than 20% a higher instability of the blend with gasoline is observed, [81].

The authors conclude that the addition of butanol, even in small percentages, can improve combustion stability by reducing the coefficient of variation of the mean indicated pressure, especially in lean mixtures. A reduction in ignition delay by 2 – 3 CAD was also observed. The main burn duration was similar for all mixtures used, [25].

For gasoline fuelling a COV of 1.24% and for butanol fuelling a COV of 0.91% was obtained. The authors stated that butanol can be considered as a good substitute for gasoline in terms of combustion stability, [83].

2.4 Influence of butanol on specific fuel consumption

One of the advantages of butanol as an alternative solution is the higher air-fuel ratio compared to ethanol. A higher enthalpy of combustion (CE) of butanol would translate into a lower increase in specific energy consumption compared to ethanol. As expected, specific fuel consumption increases with increasing butanol concentration (an increase of about 28% compared to gasoline when fuelled with B80). Relative specific consumption obtained at 2000 rpm and an average indicated pressure of 367 kPa. Specific fuel consumption is lower for butanol when compared to ethanol (E100) where the increase in consumption was 59%, [84]. In the case of 50% ethanol-gasoline blend (E50) specific fuel consumption can increase by 19%, [85]. Specific consumption can increase by about 27% when fuelled with E60, [86]. The relatively small increase in specific fuel consumption when fuelled with butanol can be attributed to its higher combustion enthalpy, [25]. It can be seen that for each mixture used the increase in specific fuel consumption is relatively constant with increasing engine load. A similar evolution was observed by He B.-Q. et al. when E30 was fed. At constant engine speeds and small increases in engine load the specific fuel consumption can increase by between 5% and 8%, [87].

Tan Tien Huynh et al. studied the influence of butanol-gasoline blends on specific fuel consumption. The authors performed tests on a Daewoo A16DMS spark ignition engine with a displacement of 1.6L and a compression ratio of 9.5. It is observed that at percentages lower than 30% the specific fuel consumption of butanol-gasoline blends is lower compared to gasoline. This lower specific fuel consumption can be explained by the lower engine torque and engine power when fuelled with butanol blends. For example, at 30% load and 2250 rpm there is a reduction in specific fuel consumption of about 3.6% when fuelled with Bu20. At 4250 rpm this reduction in specific fuel consumption is 8.4%. It can also be seen that at a higher butanol concentration of more than 30% the specific fuel consumption starts to increase very much. This means that more butanol-gasoline is consumed for the same operating conditions. This large increase in specific fuel consumption is due to the lower air-fuel ratio of butanol (11 VS 14.6 for gasoline) and its lower heating value than gasoline (33 MJ/kg VS 43 MJ/kg), [88].

Yousif et al. studied the impact of butanol-gasoline blends on the performance of spark ignition engines. They used blends with concentrations from 25% to 50% butanol. Measurements were made at constant loads and at different speeds with both gasoline and gasoline-butanol blends. The gasoline used has an octane number of 94, sourced from Iraq, [90]. The biggest disadvantage of alcohols is their lower calorific value compared to gasoline. The specific fuel consumption is higher for both blends (B75Bu25 and B50Bu50) compared to gasoline (G). At higher revs there is a lower fuel consumption compared to gasoline when fuelled with B75Bu25 but B50Bu50 will still have a higher fuel consumption compared to gasoline. For the investigated speed range, a high specific fuel consumption of about 2.38% was observed when fuelling with G50Bu50. The results were influenced by the heating value of butanol which is close to that of gasoline. For example, the impact of lower heat of heating is not very large at lower butanol concentrations such as 25%. In this case, butanol improves combustion and combustion stability through the addition of oxygen and thus results in lower specific fuel consumption compared to gasoline. The impact of the lower heating value of butanol is felt if used in higher concentrations such as 50%, the specific fuel consumption is higher. This increase in specific fuel consumption was however limited for B50Bu50, as observed by Yu Li et al, [94].

In another study by Manish Saraswat et al. a minimum specific fuel consumption was achieved using a blend of 5% butanol with gasoline. The authors used a Honda spark ignition engine, [95]. GB5 has the lowest specific consumption, very close to gasoline, while GB15 has the highest consumption. The higher fuel consumption of GB15 is explained by the lower calorific value of butanol which is felt at higher percentages in the blend. The lower specific fuel consumption can also be explained by a higher thermal efficiency when fuelled with GB5, [95].

2.5 Influence of butanol on volumetric yield

In a published study, Ashraf Elfasakhany and Abdel-Fattah Mahrous investigated the influence of methanol and butanol blends on spark ignition engine performance and emissions. The engine was operated at full load at speeds of 2600 to 3400 rpm. Intake manifold pressure and temperature, cylinder pressure, exhaust gas temperature, engine torque, power and volumetric efficiency were measured. It is observed that with increasing engine speed, volumetric efficiency also decreases due to gas and friction losses in the intake manifold. It is also observed that the n-butanol-methanol-gasoline blend has a lower volumetric efficiency than gasoline. The reason for these decreases is the low vaporization pressure of n-butanol (2.27 kPa) compared to gasoline (31 kPa). Thus, a large amount of fuel vaporises in the engine intake.

By increasing the percentage of alcohols in gasoline, volumetric efficiency is expected to increase, even exceeding the values obtained for gasoline, [30]. In a study by Feng, a higher volumetric yield than gasoline was obtained using a 35% butanol-gasoline blend, [104].

D. Balaji et al. studied the influence of izobutanol blends on the fuelling of spark ignition engines. The authors used the maximum power dosage at gasoline fuelling. The tests were performed at constant rpm and 75% engine load. The following parameters were measured: engine speed, engine torque, fuel consumption time of 100 cm^3, CO, HC, NO_x and exhaust gas temperature. Parameters such as specific fuel consumption, volumetric efficiency, power and thermal efficiency were calculated using the usual equations. The blends used are gasoline (82.5% to 87.5%), 10% ethanol and iso-butanol (2.5% to 7.5%), [34]. Volumetric efficiency is higher with iso-butanol-ethanol-gasoline blends than with gasoline. The efficiency also increases with increasing engine torque up to 0.85 Nm, from where it starts to decrease for all fuels. The improvement in volumetric efficiency can be explained by the cooling of the fresh charge at the end of the intake process. The temperature of the fresh charge is also reduced by vaporisation in the intake manifold. Volumetric efficiency starts to decrease after 0.85 Nm because the amount of air admitted to the engine can no longer increase (choking occurs in the intake system).

2.6 Influence of butanol on thermal efficiency

D. Balaji et al. calculated the thermal efficiency as a function of engine momentum when fuelled with iso-butanol-ethanol-gasoline blends (in different percentages). They observed that with the addition of 2.5% izobutanol the maximum thermal efficiency can be obtained, [34].

Yousif et al. also studied the impact of butanol on thermal efficiency. Improved thermal efficiency can lead to lower specific fuel consumption. A slight improvement in thermal efficiency is observed when fuelled with butanol blends. Thermal efficiency increased by 3.63% for B75Bu25 and 1.8% for B50Bu50 compared to gasoline. The presence of butanol in the combustion chamber can improve combustion speed, as oxidation is complete due to the additional oxygen in the combustion chamber, [90].

The presence of oxygen can improve combustion and thermal efficiency. The high miscibility of butanol compared to other alcohols makes it ideal for blending in higher percentages with gasoline, [95].

2.7 Influence of butanol on pollutant emissions

The presence of carbon monoxide in the exhaust gas is a sign of incomplete combustion and can be considered a measure of engine power loss. As engine speed increases, engine power increases and CO and HC emissions decrease. N-butanol-methanol-gasoline blends have lower carbon monoxide emissions than gasoline. The use of alcohols in gasoline shows no change in CO emissions until around 3100 rpm where a sharp decrease in CO is observed. The presence of oxygen from n-butanol and methanol leads to improved combustion in the cylinder and thus lower CO and HC emissions. The percentage of n-butanol-methanol in gasoline has a large impact on emissions. Thus, nBM3 has negative effects on both engine performance and emissions. Increasing the percentage from nBM3 to nBM10 leads to lower CO and HC emissions through the addition of oxygen but emissions are still on average higher than gasoline, [30].

The opposite trend is observed for carbon monoxide, where CO_2 concentrations depend mainly on the air-fuel mixture as well as CO and HC. A more saturated mixture implies a more complete combustion and thus higher CO_2 concentrations in the exhaust gas. As the concentration of n-butanol-methanol in gasoline increases, CO_2 concentrations increase while CO and HC decrease. The explanation is the additional oxygen content which leads to a more complete combustion in the cylinder, [30].

The high concentrations of unburned hydrocarbons HC and carbon monoxide CO of fuels obtained by blending with alcohols as well as the low CO_2 emissions compared to gasoline are also due to intermediate reactions occurring towards the end of the combustion process. These side reactions depend on factors such as combustion chamber temperature, oxygen supply, etc. In the case of oxygenated fuels, oxygen concentrations do not limit these reactions but temperature does. n-butanol and methanol alcohols have a lower calorific value and also a higher latent heat of vaporisation than gasoline, therefore the combustion chamber temperature is lower (confirmed by the lower maximum cylinder pressure and lower exhaust gas temperature) and thus the conversion of carbon monoxide and unburned hydrocarbons to carbon dioxide is slowed down. In some cases, these conversion reactions are 'frozen'. Thus, CO and HC concentrations tend to increase while CO_2 emissions are decreasing. An important factor influencing emissions is the excess air coefficient λ. By increasing the percentage of alcohols in the gasoline, the mixture becomes more and more saturated and thus CO and HC emissions tend to decrease while CO_2 will increase,[30].

D. Balaji et al. studied the influence of ethanol and iso-butanol on spark ignition engine emission concentrations. The authors obtained lower emission concentrations when fuelled with the investigated blends compared to gasoline. The lowest CO concentrations were obtained with the 5% izobutanol blend because the addition of izobutanol improved combustion in the cylinder. Similarly, the lowest HC concentrations were obtained with 5% izobutanol. Nitrogen oxide emission concentrations increased when fed with iso-butanol-ethanol blends. NO_x concentrations increase with increasing percentage of ethanol in gasoline but they increase even more with iso-butanol-ethanol blends. The authors conclude that the greatest reductions in pollutant emission concentrations can be achieved with the 5% iso-butanol and 10% ethanol blend, [34].

Huynh, T.T et al. investigated the impact of butanol (in different percentages) on the concentration of pollutant emissions. CO emissions decrease with increasing percentage of butanol in the blend. The additional oxygen in the combustion chamber as well as the reduced carbon content of the butanol blends improve combustion in the cylinder, contributing to lower CO concentrations.

Unburned hydrocarbons may be the result of failed ignitions and/or incomplete combustion in the cylinder. They also depend on engine operating conditions as well as the properties of the fuels used, [49]. At loads of 30% it is observed that HC emissions decrease with increasing butanol concentration. This decrease in HC concentrations only occurs up to percentages below 30% butanol. At percentages higher than 30% butanol the combustion temperature decreases and thus the vaporisation and quality of the air-fuel mixture worsens. Consequently, combustion is incomplete and HC concentrations increase. At 70% load and 2250 rpm, HC concentrations show a similar evolution as CO concentrations, being lower compared to gasoline. The explanation would be the low carbon content and the complete combustion of the fuel. At 4250 rpm and high loads the behaviour is similar as for low loads, after a certain percentage of butanol the combustion gets worse and HC concentrations will increase, [88].

The formation of nitrogen oxides depends on several factors such as engine load, cylinder temperature, combustion chamber content and mixture density, [49], [93]. Nitrogen oxide concentrations are higher when fuelled with gasoline-butanol blends. The highest concentrations are obtained at high loads of 70%. For example, when fuelled with Bu25 at 30% load and 2250 rpm, NO_x concentrations increase by 22% compared to gasoline (Bu0). Also, for Bu25, at 30% load and 4250 rpm NO_x increases by 47%. At loads of 70% the concentrations increase by 33% and 52% at 2250 rpm and 4250 rpm respectively, [88].

The highest CO_2 concentrations are obtained at speeds of 2250 rpm. Explain would be additional oxygen content and additional depletion of the mixture (the dosage is not changed during the tests). Higher percentages of butanol result in poorer mixtures and thus lower CO_2 concentrations. For example, when fuelled with Bu40 at 30% load and 4250 rpm the CO_2 concentration drops by about 9%. At the same rpm, but at 70% load the concentrations drop by 8%. The lowest CO_2 concentrations are obtained at high revs of 4250 rpm and high loads of 70%. This can be explained by the fact that at low engine loads combustion is more efficient, resulting in higher CO_2 concentrations, [88].

CHAPTER 3
RESULTS OF EXPERIMENTAL INVESTIGATIONS

3.1 Description of the test stand, equipment and procedure for carrying out experimental investigations

An experimental internal combustion engine test stand is very complex, composed of systems and subsystems that interact with each other to investigate the desired phenomena. It involves transducers and sensors strategically placed so that certain parameters related to engine operation can be measured. The signals from the transducers must be picked up by data acquisition systems and stored for further processing and interpretation. Actuators and execution elements are used to modify certain functional parameters of the engine.

The test stand used by the author consists of a power machine, a Daewoo Cielo A15MF spark ignition engine coupled to an AVL Dynoperform 160 electric eddy current brake via a coupling. The engine was supercharged with a turbocharger, the main design features are shown in Table 3.1. The braking torque is controlled from a control panel (PCF). During operation the brake heats up and is provided with a cooling system so that it operates in optimum parameters. The internal combustion engine is also equipped with an open liquid cooling system. The stand is fitted with a fan driven by an electric motor for additional engine cooling. The amount of air admitted to the motor is controlled by the throttle flap regulated by an actuator, the air flow is measured with a flow meter mounted on the air intake path. The fuel flow is measured with a mass flow meter mounted in the fuel supply path. Cylinder pressure is measured with a piezoelectric transducer, the signal from which is amplified by a charge amplifier and transmitted to a data acquisition system. Engine boost pressure is read using a pressure gauge. The exhaust gases are passed through a catalytic converter and then measured by a gas analyser. The gas analyser also measures the excess air coefficient. There are several electronic control units (ECUs) to manage the optimal operation of the engine and brake. The layout of the stand is shown in Fig. 3.1.

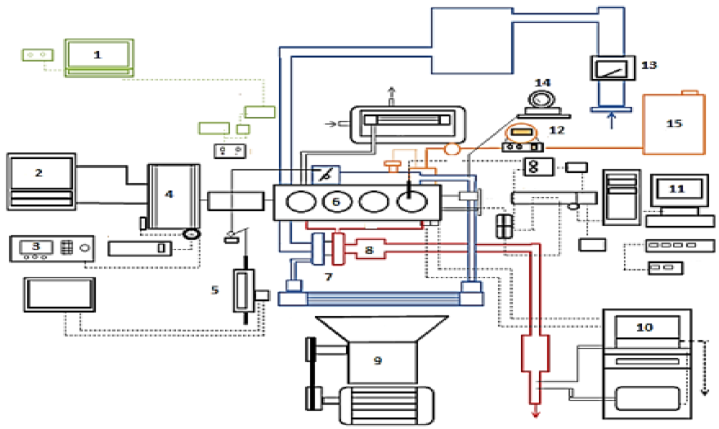

Fig. 3.1 Test bench, [111]

1 - ECU, 2 - brake cooling system, 3 - brake control panel, 4 - AVL brake, 5 - throttle valve actuator, 6 - A15MF engine, 7 - turbocharger, 8 - catalytic converter, 9 - fan, 10 - AVL gas analyser, 11 - desktop computer, 12 - fuel mass flow transducer, 13 - air flow transducer, 14 - boost pressure transducer, 15 - fuel tank.

Table 3.1 Main features of the A15MF engine

Tip motor	Spark ignition engine
Number of cylinders	4
Stroke	81.5 mm
Connecting rod length	120 mm
Bore	76.5 mm
Compression ratio	9.2

The principle of operation of piezoelectric transducers is as follows: when pressure is applied to a piezoelectric material (piezo is Greek for pressure), it deforms mechanically and electrical charges are generated. These electric charges will generate an electric field, so voltage values can be measured at the surface of the piezoelectric material. After the pressure force is no longer applied to the piezo material, the electrical charges disappear and the voltage value is zero. The pressure transducer used on the stand is a piezoelectric transducer of type AVL GU 13Z-24, mounted in the motor housing. The signal generated by it is amplified by an AVL 3067 A charge amplifier.

The motor moment, which is thus applied to the stator, is measured using strain gauge marks. A speed transducer is used to measure the speed. Using these measurements, the power of the engine coupled to the brake can be calculated. The brake generates heat that is dissipated by a water-cooling system, with water passing through the stator rings.

The gas analyser used is type AVL DiCom Analyzer 4000. The 4000 series models have a modular concept offering high flexibility, easy mounting and easy operation. The analyser offers high performance, accurate measurements and easy maintenance. It measures CO, CO_2 and HC emissions by infrared spectroscopy while O_2 and NO_x are measured by chemiluminescence. The infrared spectrum of the sample is collected by passing a beam of infrared light through the sample. Examination of the transmitted light indicates the amount of energy absorbed at each wavelength.

The chemiluminescence analysis method is based on the principle that nitrogen oxide (NO) reacts with ozone (O_3) resulting in nitrogen dioxide (NO_2), 10% electronically excited nitrogen dioxide (NO_2*) and oxygen. In the chemical reaction between nitrogen oxide and ozone NO - O_3, the excited NO_2* molecules return to NO_2. This process takes place with light emission directly proportional to the concentration of NO present in the gas to be analysed.

Hourly fuel consumption is measured with the Krohne Optimass 3050 C mass flow meter.

To determine the air flow the differential pressure in the inlet line must be determined. The pressure was measured by a liquid pressure gauge.

Working procedure: Engine operating speed used: 55% load, 2500 rpm. The following parameters were measured: engine torque, power and rpm, laboratory enclosure temperature and pressure, engine inlet air temperature, coolant temperature, cylinder pressure correlated to crankshaft rotation angle, air and fuel flow rates, λ and engine emission concentrations after catalyst. The quality of the air-fuel mixture is changed from lean to rich and vice versa by changing the amount of fuel injected. 250 consecutive engine cycles are acquired in order to determine the coefficient of variability for several parameters such as maximum cylinder pressure and average indicated pressure.

After the reference was made when the engine was fuelled with gasoline, the same measurements under the same engine operating conditions were made when fuelled with butanol-gasoline blends, (10% and 15% butanol in the mixture).

3.2 Results of experimental investigations of the engine fuelled with 10% butanol-gasoline blend

3.2.1 Rich mixtures

Diagrams shown record when fuelled with gasoline or 10% butanol-gasoline blend. From the diagrams the following aspects can be observed: 1) higher cyclic variability for gasoline, 2) higher maximum pressure for 10% butanol-gasoline, 3) de-ignition displacement for 10% butanol-gasoline blend. These results are recorded for the same engine tuning fuelled with the two fuels,[111]. Averaging the results gives a maximum pressure about 5% higher for GB10.

The coefficient of variability for the maximum pressure when fuelled with gasoline is about 13% while when fuelled with the 10% butanol-gasoline blend is 11%. Similar results were observed in [112], [113].

Better stability is observed for gasoline fuelling, with a 2.5% lower coefficient of variability. It appears that the addition of oxygen did not improve combustion stability at rich mixtures. The average mean pressure value shown is 9.94 bar for gasoline and 8.82 bar for GB10, a 12% decrease for the blend. This decrease in average indicated pressure can be explained by the lower calorific value of the GB10 blend (butanol has a lower calorific value than gasoline).

An increase in the rate of pressure rise is observed when feeding GB10 as well as a shift of the maximum value towards the TDC. This shift of the maximum pressure rise rate can be explained by the faster burning of butanol in the fast burn phase due to its higher burning rate, which also explains the increase of the maximum pressure.

In the case of 5% heat release a high dispersion is observed when feeding the 10% gasoline-butanol mixture. For gasoline COV_G=9.13% and COV_{GB10}=12.81%. It can be concluded that the addition of butanol oxygen does not help to improve the initial combustion phase in rich blends. It is also observed that the angle at which 5% of the heat is released is on average smaller compared to gasoline, indicating a faster initial combustion of the 10% gasoline-butanol blend. Similar results were obtained by Swaja where a linear decrease in the angle at which 10% of the total heat is released is observed with increasing percentage of butanol, [83].

In the case of the main combustion phase this is no longer the case. Cyclic dispersion is lower for the 10% butanol-gasoline blend, the $COV_{GB10} \cong 10.5\%$ and for gasoline $COV_G=16.5\%$. It can be seen in this case that gasoline burns faster than butanol, a phenomenon also observed by [89]. The average angle at which 50% of the total heat is released is lower with gasoline, a different result compared to [83]. Note that the authors used butanol blends in higher percentages, the minimum percentage being 20%.

In the case of final combustion, (90% of total heat) gasoline burns a little faster than the 10% butanol-gasoline blend. Cyclic dispersion values are low in both cases, COV=0.5%.

The conclusion is that the 10% butanol-gasoline blend has a small influence on the initial combustion phase, large influences on the main combustion phase and no major impact on the final combustion phase. The coefficient of variation for the main combustion phase was reduced by 6% by the addition of butanol. Comparing the results obtained with [56], the initial combustion phase is faster when fuelled with alcohols as in this case, and the coefficient of variability of the mean pressure indicated when fuelled with butanol-gasoline blends did not exceed 8%. This is also true in the results obtained in this paper (5.36%).

In the main combustion phase gasoline burns faster but the maximum value is similar in both cases of about 50 J/CAD. Szwaja et al. also observed that at higher percentages of butanol the rate of heat release can vary, but the maximum value is similar to that of gasoline,[83].

3.2.2 Lean mixtures

The maximum values are lower compared to those obtained with rich mixtures. In terms of the coefficient of variation of the maximum pressure, the addition of butanol oxygen does not bring a significant improvement ($COV_G=11.32\%$ and $COV_{GB10}=11.16\%$). The maximum pressure decreases by about 4.75% when fed with GB10. This decrease may be due to the lower heating value of butanol and the additional depletion of the mixture by oxygen addition of butanol. Similar results were observed in [88].

In contrast to rich blends, the average pressure indicated is similar when fuelled with both fuels and averages around 8.5 bar. The calculated coefficients of variation are $COV_G=3.75\%$ and $COV_{GB10}=3.5\%$. An improvement in combustion stability is observed with the addition of oxygen to lean mixtures. Similar results were observed in [83].

A similar evolution of pressure rise during combustion is observed for both fuels up to about 380 CAD, where at gasoline fuelling a sudden pressure rise occurs. A possible explanation would be the electrical spark advance optimised for gasoline fuelling.

The average value of this angle is 36.5 CAD for gasoline and 37.4 CAD for 10% butanol-gasoline. In rich blends the maximum pressure was obtained closer to the TDC when fuelled with GB10, but in lean blends it is observed that the addition of oxygen does not have a major influence on the angle at which the maximum pressure is obtained compared to gasoline.

It is observed that butanol burns faster than gasoline in the initial combustion phase, the average angle being 12 CAD for GB10 and 15 CAD for gasoline. Gasoline is more stable in this respect however, with COV_G=12.81% compared to COV_{GB10}=13.55%.

This time there is not a very big difference between the two fuels, the average angle of release of 50% of the heat being about 31 CAD in both cases. In terms of variability, butanol shows a significant reduction in the coefficient of variability by about 6% compared to gasoline. It can be stated that the main combustion phase is more stable when fuelled with GB10.

It can be observed that the combustion ends faster when fuelled with GB10, at an average angle of 48 CAD compared to 52 CAD for gasoline. At lean blends, butanol tends to burn faster in the initial and final phases of combustion. In terms of COV, at 90% no difference is observed between G and GB10.

A higher heat release rate is observed when feeding GB10 at the beginning of combustion. This phenomenon can be explained by the higher combustion rate of butanol. The maximum heat release rate is similar for both fuels (about 45 J/ CAD).

3.3 Results of experimental investigations of engine fuelled with 15% butanol-gasoline blend

3.3.1 Rich mixtures

No significant change is observed for the 15% butanol-gasoline blend, $(COV)_{PmaxG}$= 9.59% respectively 15% butanol-gasoline blend $(COV)_{PmaxGB15}$= 9.54%. The addition of butanol oxygen has no impact on the engine running stability at rich blends and on the indicated charts. The charts shown overlap almost perfectly in both cases. The maximum pressure in both cases is around 38 bar. Similar results were obtained in [114], [115].

The coefficient of variability is low as a percentage, the COV for gasoline being 3.74% and for the 15% butanol-gasoline blend 3.29%. The percentage difference is 12.8% in favour of the 15% butanol-gasoline blend. The average indicated pressure is similar in both cases, 9.81 bar for gasoline VS 9.92 bar for 15% butanol-gasoline.

A higher rate of pressure increase is observed for gasoline, but similar values are obtained on some engine cycles for the 15% butanol-gasoline blend. A possible explanation is the higher lower calorific value of gasoline as well as maintaining optimised engine settings when fuelled with gasoline.

The percentage of butanol does not influence the rate of heat release, the average maximum value being around 60 J per crankshaft revolution.

The coefficient of variability for 5% heat release is 9.13% for gasoline and 13.87% for the 15% butanol-gasoline blend (about 5% higher). The average value of the angle at which 5% of the heat is released is about 18 CAD for gasoline and 13 CAD for 15% butanol-gasoline, due to the higher combustion rate of butanol.

To release 50% of the heat, the average angle obtained when fuelled with gasoline is 23 CAD and 29 CAD for the 15% butanol-gasoline blend, indicating a trend reversal compared to 5% heat release. In this case, the coefficient of variation has a decrease of about 47% (from 16.52% to 10.24%) when fuelled with the 15% butanol-gasoline blend, indicating an improvement in the stability of the combustion process at 50% heat release. At 90% heat release, both fuels have a low coefficient of variation, with gasoline having an advantage of less than 0.5% in this case. The average angle is about 44 CAD in the case of gasoline and 50 CAD in the case of 15% butanol-gasoline, the combustion moving in the expansion. These values can be improved by optimising engine settings when fuelled with the 15% butanol-gasoline blend.

3.3.2 Lean mixtures

Values around 15% of the maximum pressure coefficient of variability are observed for both fuels. Higher values of variability coefficients are observed for both gasoline and 15% butanol-gasoline blends. There is a slight increase in COV (2.7%) for the 15% butanol-gasoline blend and a decrease of about 4.5% in maximum pressure. Similar results were observed in [114], [115].

A COV of about 6.75% was obtained for gasoline and 7% for the 15% butanol-gasoline blend for the indicated average pressure, an increase of about 7.4%. A general instability is observed at very lean blends for both gasoline and 15% butanol-gasoline. Improvements can be achieved by optimising engine settings when using butanol. The average pressure indicated is around 8.9 bar for gasoline and 8.5 bar for GB15. Engine performance is not significantly affected by the addition of 15% vol. butanol in gasoline even though it has a higher degree of running instability, at least at very lean blends.

Both fuels show a slower increase in the rate of pressure rise during combustion at lean blends having lower maximum values (1 respectively 0.9 bar/ CAD).

A small decrease of the maximum heat release rate is observed when fuelled with the 15% butanol-gasoline blend. This decrease can be explained by the lower heating value of butanol and the additional depletion of the blend caused by the additional oxygen addition of butanol.

The coefficient of variation has high values when fuelling with both fuels at lean blends. The COV at 5% heat release is higher when fuelling with the 15% butanol-blend compared to gasoline (COV_G= 25.23% VS COV_{GB15}= 25.89%). This results in an average angle $\alpha_{5\%}$ of 12 CAD for gasoline and 11 CAD for 15% butanol-gasoline. Faster initial combustion is observed for lean blends compared to rich blends where the average angle would be 18 CAD for gasoline and 13 CAD for 15% butanol-gasoline. These high values of the variability coefficients are also preserved at 50% and 90% heat release respectively for lean blends. The combustion process is not as stable as in rich mixtures. At 50% heat release, the COV for gasoline is 16.66% compared to 17.83% for 15% butanol-gasoline with an average angle of 54 CAD. At 90% heat release, the COV for gasoline is 0.77% and 0.81% for 15% butanol-gasoline.

3.4 Specific energy consumption and thermal efficiency

The highest specific fuel consumption is obtained with GB15 and the lowest with gasoline, the difference being 37%. For GB10, energy consumption is 5.7% higher than for gasoline. The higher energy consumption in cases GB10 and GB15 can be explained by the lower thermal efficiency at rich blends. Similar results were obtained in [25], [88] and [90].

Thermal efficiency is about 12.5% better on gasoline compared to GB15 on rich blends. However, it is observed that at lean and very lean blends the thermal efficiency is higher for GB10 and GB15, e.g. GB15 has 4.25% better efficiency than gasoline. In order to benefit from improved thermal efficiency, it is recommended to use butanol-gasoline blends at lean blends λ=1.3. At higher percentages of butanol in the blend the influence of its lower heating value is felt and thus a higher energy consumption is needed to generate the same power. The presence of butanol in the combustion chamber can improve the combustion rate as oxidation is complete due to the additional oxygen in the combustion chamber, [90],[95].

3.5 Pollutant emission concentrations

GB10 generates the lowest concentration of carbon monoxide at rich blends, 22% less than GB15 and 11% less than gasoline. At lean and very lean blends, no improvement is observed with the addition of oxygen. The additional oxygen in the combustion chamber as well as the reduced carbon content of butanol-butanol blends improve combustion in the cylinder, contributing to lower CO concentrations. Similar results have been obtained in [49], [91], [92], [114], [115].

Unburned hydrocarbons are also a product of incomplete, inefficient combustion caused by the lack of oxygen in the cylinder, of the flame extinguishing at the cylinder wall. Low cylinder wall temperature can lead to disruption of chemical reactions and the formation of unburned hydrocarbons in the boundary layer area of the wall. Other sources of unburned hydrocarbon formation can be engine cycles with failed ignitions (misfires), but some of these gases ignite in the exhaust system in the presence of oxygen. In rich blends, HC concentrations are 10% lower when fuelled with GB10 compared to gasoline and in lean blends when fuelled with GB10 and GB15 HC concentrations are 26% lower. At very lean blends the results are similar in all three cases. Similar results were also obtained in [114],[115]. It is possible that at high speeds and loads the combustion may worsen and thus HC concentrations increase again, [88].

The reaction rate depends on the temperature, so zones with different concentrations of nitrogen oxides appear in the cylinder following the flame front due to thermal inhomogeneity. In general, the highest concentration of nitrogen oxides is found in the spark plug area where the temperature is highest. A reduction of nitrogen oxides is observed when fuelling with butanol-gasoline blends at both rich and lean blends. At lean blends a reduction of nitrogen oxides concentrations of about 66% is observed and 52% at rich blends. At stoichiometric mixing GB10 has lower nitrogen oxide concentrations by about 72%. GB15 has approximately 42% lower concentrations. This phenomenon can be explained by the vaporisation of the butanol which produces a gas cooling effect and a lower flame temperature in the cylinder. Similar results were obtained in [34], [114], [115]. However, the results differ from those obtained by Huynh et al,[88]. They obtained higher concentrations of nitrogen oxides with increasing percentage of butanol. The authors conclude that more studies related to the reduction of NO_x concentration at butanol fuelling would be needed.

Unlike carbon monoxide and unburned hydrocarbons, carbon dioxide is the result of complete combustion. Burning hydrocarbons in the presence of sufficient oxygen will generate carbon dioxide and water as a reaction product. In rich mixtures GB10 is found to have 23% higher carbon dioxide concentrations compared to GB15 and 11% higher compared to gasoline. No major differences are observed between the three fuels at lean and very lean blends.

CHAPTER 4
MODELLING THE PROCESSES IN THE SPARK IGNITION ENGINE

4.1 Getting started

Simulating a phenomenon (either a simple one such as a force acting on a body or a complex system such as a gearbox) requires in the first phase the understanding of the phenomenon/system, the connection between the different subsystems, the determination of the mathematical formulas describing its functioning, then modelling followed by validation through physical experiments. In general, the first resulting models will not produce credible results as a system depends on many variables and states (some of which cannot be expressed concretely e.g. some yields and thus will have to be estimated); therefore, they have to be developed, numerically simulated, the results obtained analysed, corrected and adapted accordingly so that the results reflect reality as much as possible.

The physico-mathematical model is built around the energy balance equations of each thermodynamic process in the engine cylinder

Fortunately, by using numerical simulation applications such as AMESim we can significantly reduce the difficulties mentioned above because it already has predefined and validated models. For example, for the modelling of the physico-mathematical phenomena in this paper we started from a predefined model in the AMESim spark ignition engine libraries, AME/demo/Libraries/CFD1D/SI_SingleCylinder_WInj.ame,[117]. The model consists of a single cylinder and the displacement is calculated dynamically according to the main engine design parameters.

The aim of this chapter is to model the thermo-geodynamic processes in the cylinder of a standard spark ignition engine (SI engine) to determine its energy performance and the influence of the fuel used. In the first phase, the results obtained will be compared with those of experimental investigations of the Cielo Nubira engine with a displacement of 1.5 l fuelled with gasoline. A similar model was used in [116]. The main limitation of that model is that it can only generate results on stoichiometric mixtures.

4.2 Model description from AMESim

The starting model consists of the following sub-assemblies: throttle valve, injector, piston-rod assembly, crankshaft and engine cylinder. The model is shown in Fig. 4.1.

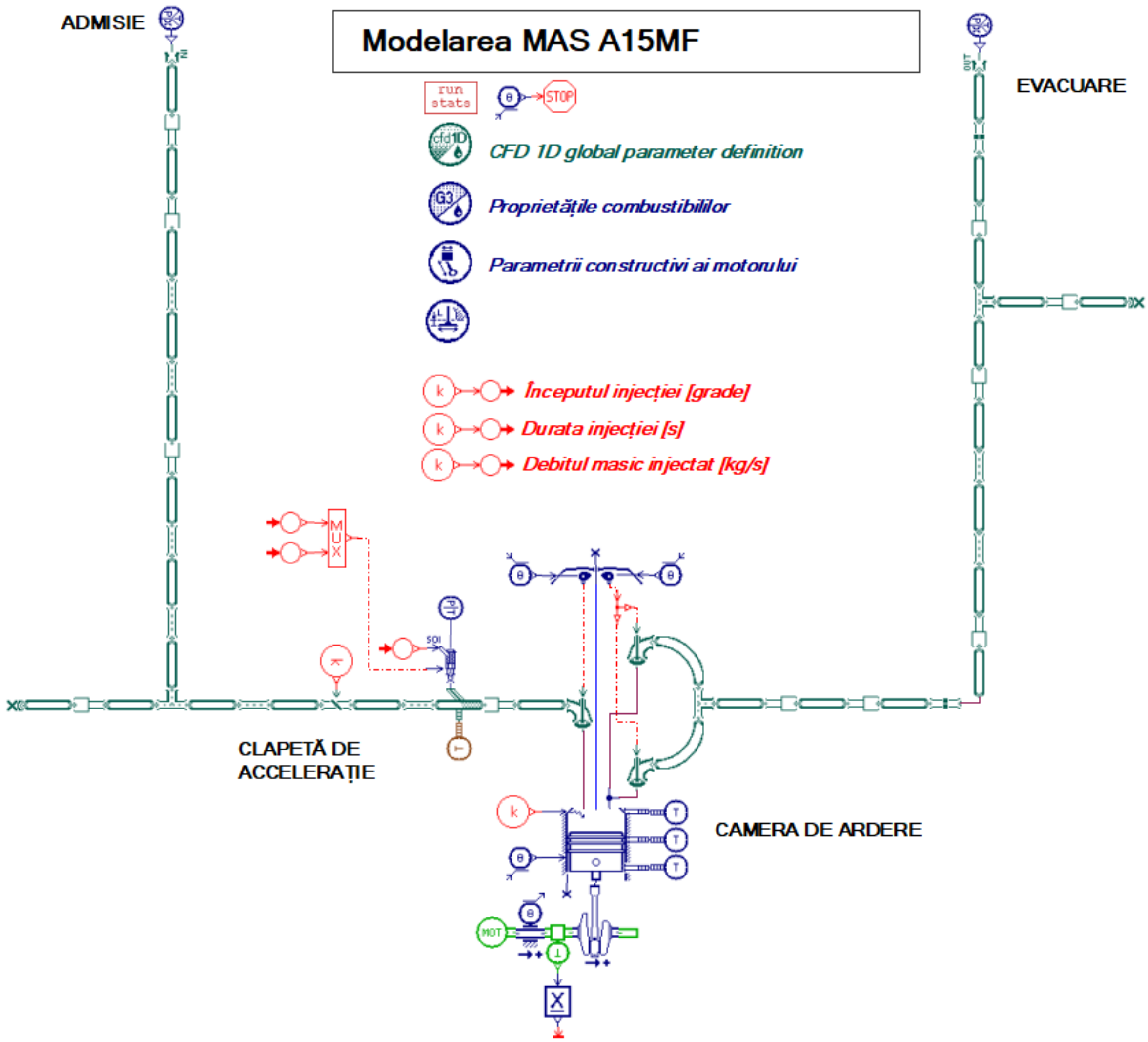

Fig. 4.1 Modelling the A15MF spark ignition engine in AMESim,[117]

The main sub-models of the system are:

- Throttle Valve Modelling (CFD1DJNCTHROTTLE00)
- Cylinder and combustion chamber modelling (ENGCCSI10)
- Modelling of connecting rod-crank mechanism (ENGCRKI00)
- Injector modelling (ENGINJ13)

Parameterization of the model is done by the following parameters:

- Definition of the fluids used - chemical composition, net calorific value, fluid density, latent heat of vaporization at reference temperature

- Engine definition - bore, stroke, connecting rod length, piston area
- Cylinder and combustion definition - spark advance, fuel ignition delay, combustion duration, Vibe kinematic coefficients
- Definition of engine load using the throttle body
- Definition of the engine speed

CHAPTER 5
COMPARISON BETWEEN EXPERIMENTAL AND NUMERICAL SIMULATION RESULTS

5.1 Rich mixtures

The following values were modified and optimized during the numerical simulation: net calorific value, liquid fuel density, liquid fuel specific heat, reference temperature for fuel latent heat of vaporization.

The chemical composition of the three fuels was determined from the chemical composition of gasoline used by the model C_8H_{18} and the n-butanol formula $C_4H_{10}O$. Depending on the degree of substitution of butanol in gasoline (percentage) the chemical composition of the 15% butanol-gasoline mixture is determined. Chemical formula of mixture GB15 as $C_{7.4}H_{16.8}O_{0.15}$.

The maximum pressure is around 37 - 38 bar. The maximum pressure as well as the angle at which the maximum pressure is obtained by numerical simulation overlap almost perfectly with the experimental results. The differences can be explained by the estimation of the fuel ignition timing, the estimation of the spark advance, the estimation of the Wiebe coefficients as well as the fuel injection duration. The enrichment coefficient of the mixture is 1.1 equivalent to an excess air coefficient of about 0.9.

The pressure rise variation during combustion obtained experimentally and by numerical simulation is similar but does not overlap perfectly. The differences can be explained by the estimation of the Wiebe coefficients used in the calculation of the heat release rate. This trend is observed both experimentally and by numerical simulation. At higher percentages of butanol in gasoline, a decrease in the maximum rate of pressure rise is observed.

A similar evolution of the variation of the heat release rate is observed, the maximum value being around 60J/ CAD when fuelled with G and GB15, both experimentally and by simulation. The angle at which the maximum heat release rate is obtained differs slightly from the experimental results, the possible explanation being the estimation of fuel ignition timing, advance to electrical spark, burn duration, injection duration, injection start time and Wiebe coefficients.

5.2 Lean mixtures

The maximum pressure is around 33 - 35 bar. The maximum pressure as well as the angle at which the maximum pressure is obtained by numerical simulation overlap almost perfectly with the experimental results. The differences can be explained by the estimation of the fuel ignition timing, the estimation of the spark advance, the estimation of the Wiebe coefficients as well as the fuel injection duration. The enrichment coefficient of the mixture is 0.78 equivalent to an excess air coefficient of about λ=1.28.

A similar evolution of the pressure rise rate during combustion is observed both experimentally and by numerical simulation results. A decreasing trend of the maximum pressure rise rate during combustion is observed with increasing butanol percentage.

The variation of the heat release rate is not identical for all fuels used but it is observed that the maximum rate is similar (between 47 - 50 J/CAD), also observed in [83]. The differences between the experimental and numerical simulation results could be explained by the estimation of fuel ignition timing, electrical spark advance, combustion duration, injection start time, injection duration and Wiebe coefficients.

Overall, the numerical simulation results are similar to [116]. The main difference between the two models used is that the model presented in this paper can work on lean and rich mixtures, being much closer to reality than the previous model which only worked on stoichiometric mixtures.

CHAPTER 6
CONCLUSIONS, PERSONAL CONTRIBUTIONS, FUTURE RESEARCH DIRECTIONS, DISSEMINATION OF RESEARCH RESULTS

6.1 Conclusions

From the analysis of the theoretical and experimental research results obtained, the following conclusions can be drawn:

I. Engine fuelled with 10% butanol-gasoline blend

1. 5% increase in maximum gas pressure compared to gasoline
2. 2% improvement in the coefficient of variability of maximum pressure due to the higher combustion speed of butanol
3. Approximation of the maximum pressure angle to the MIP (25 CAD versus 34 CAD for gasoline) due to reduction of the main combustion phase duration
4. Reduction of average indicated pressure by 12% when using rich blends due to lower heating value of butanol compared to gasoline
5. 2.5% worsening of the coefficient of variability of the average indicated pressure when using rich blends
6. Maximum heat release rate for 10% butanol-gasoline fuelling is similar to gasoline fuelling
7. Approximately 5% reduction in maximum pressure when using lean dosages due to the lower heating value of butanol and the additional depletion of the blend by the addition of butanol oxygen
8. Improved coefficient of variation of average indicated pressure when using lean dosages
9. Reduced influence on mean indicated pressure when using poor dosages
10. Increased maximum rate of pressure rise during combustion when using lean dosages
11. Increased stability of the main combustion phase, with a 6% lower coefficient of variation when using lean dosages

12. Reduced unburned CO and HC concentrations when using lean dosages due to additional oxygen content in the combustion chamber and reduced carbon content of the 10% butanol-gasoline blend
13. 10% lower nitrogen oxide emission concentrations compared to gasoline

II. Engine fuelled with 15% butanol-gasoline blend

14. Maintaining maximum gas pressure at the same level when using rich dosages
15. 12.8% reduction in the coefficient of variation of the average indicated pressure when using rich dosages
16. Reduction of the maximum rate of pressure rise during combustion when using rich mixtures
17. The average value of the angle at which 5% of the total heat is released is 13 CAD compared to 18 CAD for gasoline due to the higher combustion rate of butanol
18. Butanol shows a significant improvement in the coefficient of variability of the angle at which 50% of the total heat is released with a value of 10.24% compared to 16.52% for petrol when using rich blends
19. Decrease of about 4.5% in maximum pressure when using lean dosages
20. 2.7% increase in the coefficient of variability of maximum pressure when using poor dosages
21. Increase in the coefficient of variation of the average indicated pressure to 7.4% for GB15 from 6.75% for petrol
22. Reducing the maximum rate of pressure rise during combustion when using lean dosages
23. Reduction of the maximum rate of heat release explained by the lower heating value of butanol and the additional depletion of the mixture caused by the additional oxygen addition of butanol
24. 4.25% reduction in specific energy consumption when using lean dosages due to higher butanol burn rate
25. Reduction of NO_x concentrations by 42% due to the sharp reduction caused by butanol vaporisation
26. Reduction of CO_2 concentrations when using lean dosages due to higher butanol combustion rate
27. The model used to simulate thermo-geodynamic processes in the cylinder of the A15MF spark ignition engine produced results close to those obtained experimentally. It is a useful and efficient tool for investigating the thermo-gas-dynamic processes in the cylinder of the spark ignition engine fuelled with butanol mixed with gasoline in order to evaluate its performance.

Butanol can be considered a clean and viable alternative fuel for fuelling spark ignition engines in a gasoline blend. At higher percentages of butanol blended with gasoline, the energy and pollution performance of the engine is improved. Further reductions in pollutant emissions and improved engine running stability can be achieved by optimising engine settings. The use of butanol to fuel spark ignition engines does not require major engine design changes.

6.2 Personal contributions

The author has made the following personal contributions to the conception and writing of the PhD thesis:

1. Documenting and updating the state of the art of research into the use of butanol to fuel spark ignition engines
2. Upgrading the experimental investigation stand of the butanol fuelled engine
3. Establishment of procedures for conducting experimental investigations
4. Conducting experimental investigations on the engine test stand
5. Processing and analysing the results of experimental investigations
6. Adaptation and use of an existing numerical model in AMESim for modelling and simulation of thermo-geodynamic processes in the cylinder of a spark ignition engine fuelled with butanol mixed with gasoline
7. Comparative analysis of experimental results and numerical simulation results and validation of the numerical model used

6.3 Future research directions

1. Conduct experimental investigations of the combustion process at all load and speed regimes used in operation
2. Firing the spark ignition engine with higher percentages of butanol
3. Conducting experimental investigations of the spark ignition engine fuelled with butanol blended with petrol under track conditions
4. Optimization of spark ignition engine tuning fuelled with butanol in gasoline mixture

6.4 Dissemination of research results

The results of the research carried out by the author during the elaboration of the PhD thesis have been published in journals and volumes of prestigious international congresses/conferences.

Papers published in journals:

1. **Cristian Sandu**, Constantin Pana, Niculae Negurescu, Gheorghe Lăzăroiu, Alexandru Cernat, Rareș Georgescu and Cristian Nuțu, The Influence of N-Butanol Addition in Gasoline on the Combustion in the Spark Ignition Engine, Volume 15, Issue 18, 2023, https://doi.org/10.3390/su151814009, Sustainability 2023

Papers published in volumes of international congresses/conferences:

2. **Cristian Sandu**, Constantin Pană, Niculae Negurescu, Alexandru Cernat, Cristian Nuțu and Rareș Georgescu, The study of the spark ignition engine operation at fuelling with n-butanol-gasoline blends, Volume 180, 2020, e-ISSN: 2267-1242, https://doi.org/10.1051/e3sconf/202018001010, E3S Web of Conferences 180, 01010 (2020), TE-RE-RD 2020

3. **Cr Sandu**, C Pana, N Negurescu, Al Cernat, D Fuiorescu, R Georgescu and C Nutu, The study of cyclic variability at a n-butanol spark ignition engine fuelled, Volume 997, 2020,

https://doi.org/10.1088/1757-899X/997/1/012132, IOP Conference Series: Materials Science and Engineering 997, ACME 2020

4. **C Sandu**, C Pana, N Negurescu, A Cernat, D Fuiorescu, R Georgescu and C Nuţu, Theoretical and experimental research of an automotive gasoline engine fuelled with butanol, Volume 1235, 2021, https://doi.org/10.1088/1757-899X/1235/1/012036, IOP Conference Series: Materials Science and Engineering 1235, IMANEE 2021

5. **Cristian Sandu**, Constantin Pană, Niculae Negurescu, Alexandru Cernat, Cristian Nuţu and Rareş Georgescu, The study on the influence of utilizing n-butanol at fuelling spark ignition engines, Volume 1220, 2021, https://doi.org/10.1088/1757-899X/1220/1/012004, IOP Conference Series: Materials Science and Engineering 1220, AITS 2021

6. **Cr Sandu**, C Pana, N Negurescu, Al Cernat, D Fuiorescu, R Georgescu and C Nutu, The study of the spark ignition engine performance at fuelling with n-butanol-gasoline mixture, Volume 1262, 2022, https://doi.org/10.1088/1757-899X/1262/1/012073, OP Conference Series: Materials Science and Engineering 1262, ACME 2022

7. Rareș Georgescu, Constantin Pană, Niculae Negurescu, Alexandru Cernat, Cristian Nuțu, **Cristian Sandu**, A study on the influence of utilizing hydrogen at fuelling spark ignition engines, International Congress of SIAR of Automotive and Transport Engineering, EAEC-MVT 2022 26-28 October 2022, Timișoara, Romania

8. Rareș Georgescu, Constantin Pană, Niculae Negurescu, Alexandru Cernat, Cristian Nuțu, **Cristian Sandu**, Theoretical and experimental research on the use of hydrogen in the automotive spark ignition engine, XXIIIrd National Conference on Thermodynamics with International Participation, NACOT 2023 11-13 may 2023, Galați, Romania, IOP Publishing IOP Conference Series: Materials, Science and Engineering 1290 (2023)012010 doi:10.1088/1757-899X/1290/1/012010

9. Rareș Georgescu, Constantin Pană, Niculae Negurescu, Alexandru Cernat, Cristian Nuțu, **Cristian Sandu**, Aspects of the combustion variability analysis at an automotive engine fuelled with hydrogen, The 33rd SIAR International Automotive and Transport Engineering Congress, ESFA 2023, 2-4 November 2023 Bucharest, Romania

SCIENTIFIC REPORTS (P, F)

P - Competitive research-development-innovation projects obtained on a contact/grant basis in the country/abroad (Pn - national, Pi - international):

Pn - national:

Pn.1Member of the working team in the research project: Experimental Demonstrative Project PED 721/2022 - „The use of hydrogen - a viable solution for the greening of the internal

combustion engine of the car", acronym DIESELH2, Programme 2 - Increasing the competitiveness of the Romanian economy through research, development and innovation - Subprogramme 2.1- Competitiveness through research, development and innovation, grant of the Ministry of Research, Innovation and Digitization, CCCDI-UEFISCDI, project number PN-III-P2-2.1-PED-2021-0427, within PNCDI III, nr UEFISCDI 1078/30.06.2022, nr UPB 14837/23.06,2022, code CRESCDI 220223723, period 2022-2024.

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