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Special Issue Reprint

Internal Combustion Engine Performance

Edited by
Georgios Mavropoulos, Eleutherios Christos Andritsakis
and Roussos Papagiannakis

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Guest Editors

Georgios Mavropoulos
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About the Editors

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Georgios Mavropoulos is Professor of Internal Combustion Engines at the Department of Mechanical Engineering Educators of the School of Pedagogical and Technological Education (ASPETE) in Athens, Greece. He is the head of the Laboratory of Internal Combustion Engines and Automotive Powertrain. He is also a Senior Research Associate at the Internal Combustion Engines Laboratory (ICEL) at the School of Mechanical Engineering of the National Technical University of Athens (NTUA). His main research interests cover the areas of theoretical modeling and experimental techniques related to internal combustion engine performance, heat transfer, pollutant emissions, and exhaust heat recovery. He is the author of 43 research papers published in international journals and conference proceedings with referees. He has been teaching a wide variety of Mechanical Engineering courses in Greek universities and professional institutions. He has participated in numerous national and international research projects related to the energy sector. In parallel, he has served the manufacturing industry for over 20 years in the areas of product design engineering and quality improvement. As a team leader, he has developed several innovative product designs and manufacturing processes. He is a member of several international societies including SAE, ASME, and ACS. His work has been repeatedly awarded from various national and international business and research organizations.

Eleutherios Christos Andritsakis

Eleutherios Christos Andritsakis, currently affiliated with CERTH, has been, since 1983, studying, practicing and researching mechanical and marine engineering in Hellas (Greece), USA, Spain, Portugal, Austria, Poland, U.K., Germany, Italy, Croatia, Denmark, Cyprus, Belgium, France, Monaco, Switzerland, Singapore, Hong Kong, Brazil, Argentina, Uruguay, Chile, Peru, Ecuador, Russian Federation, Albania, Turkey, and “at sea” as well. In April 1988, while attending the last (10th) semester of the NTUA Diploma of Mechanical Engineering 5-year academic program (graduated from, promptly, by October 1988), took the offered at the time in English, Educational Testing Service (ETS) Graduate Record Examination (GRE) Subject Test on Engineering (Code 37), and ranked in the 95th percentile of the total scores, in the 91st percentile of the Engineering sub-scores, and in the 95th percentile of the Mathematics Usage sub-scores as well, of all global test examinees between 1988 and 1992. In July 1993, he earned an NTUA Research Doctorate in Mechanical Engineering after 3 years of doctoral research which sufficed for attaining original and significant research results in the areas of analytical, computational and experimental thermodynamics, aerothermochemistry, heat transfer and gas dynamics of reciprocating internal combustion engines, cited continuously to date by his co-authors and by independent international researchers as well.

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fuels in reciprocating engines. He has participated in several research projects undertaken by the ICEL/NTUA, and he has authored/co-authored over 50 refereed papers in international journals and conferences.

Preface

The internal combustion (I.C.) engine is one of the most important and successful technological developments of the last century. Its application throughout the years to the present has essentially influenced almost every aspect of human life, used either as the prime mover in land, sea, and air transportation, as the source of electrical power production, or as an emergency safety installation in hospitals and in factories.

The main reasons for this enormous distribution and success are the high energy density of liquid hydrocarbon fuels combined with the ability of the I.C. engine to efficiently cover a whole area of energy demand from a fraction of a W up to several dozen MW.

The world energy crisis and environmental impacts have played major roles in the development of the internal combustion engine over the last few decades. At this time, it is clear that a closer understanding of the thermodynamic processes occurring within the engine is necessary. As a result, research on I.C. engines has expanded enormously, both on simulation and experimental bases. Today, the main objectives are the improvement in engine performance, the minimization of fuel consumption/CO₂ emissions, and the reduction in the level of exhaust pollutants. To this aim, various alternative combustion techniques have been developed or are under development (e.g., direct injection SI engines and HCCI operation), and in parallel, various internal and after-treatment exhaust measures are also being examined.

The present Reprint of *Energies* presents results of innovative research and includes some of the latest developments on internal combustion engines. Among the topics included are the following:

Combustion mechanisms in spark and compression ignition engines;

Fuel injection and spray formation;

Pollutant formation (particulate matter, NO_x, CO, HC, noise);

Exhaust after-treatment systems (three-way catalysts, oxidation catalysts, diesel and gasoline particulate filters, SCR, NO_x adsorbers);

Internal measures for emission control (EGR, water injection, etc.);

Alternative fuel and biofuel effects on engine performance and emissions.

We sincerely hope that this effort contributes to an increase in interest in further IC engine development in the coming years.

Georgios Mavropoulos, Eleutherios Christos Andritsakis, and Roussos Papagiannakis

Guest Editors

Article

Comparative Evaluation of the Effect of Vehicle Parameters on Fuel Consumption under NEDC and WLTP

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Abstract: Higher speeds, faster acceleration and longer duration need a more realistic driving cycle. As a result, a new test procedure that reflects real-world driving conditions has been applied since 2017, and the previous development environment optimized for NEDC has also changed. In this study, several factors and technologies relating to fuel consumption, such as vehicle weight, tire rolling resistance, drag of aerodynamic, stop–start, and 48 V mild hybrid system, are evaluated as per the new worldwide harmonized light vehicles test procedure (WLTP) and compared with that of the previous European driving cycle (NEDC). The impact of the vehicle weight is increased in case of the WLTP due to faster acceleration compared to that under NEDC. The influence of aerodynamic force is very important as the average and maximum speed are increased. Meanwhile, the impact of idle stop–start technology is lower compared to that under NEDC due to the reduction in idle operation time. The 48-V mild hybrid system is still expected to play a role as a powerful fuel consumption reduction technology under new WLTP by applying energy regeneration, minor torque assist, and extended idle stop–start.

Keywords: WLTP; RDE; real-world driving conditions; test cycle; drag coefficient; tire rolling resistance; cycle energy; fuel consumption; idle stop–start; 48-V mild hybrid system; MHSG (mild hybrid starter and generator)

1. Introduction

It has been observed for a long time that exhaust gases emitted under real-world driving conditions are excessive when compared with those measured in a laboratory. Additionally, the worldwide harmonized light vehicles test procedure (WLTP) [1] legislation, which reflects the actual driving factors compared to NEDC [2], was introduced and has been applied since September 2017, owing to the increase in demand for fuel economy and exhaust gas regulations recently triggered by the exhaust gas manipulation incident [3]. Furthermore, the real driving emission (RDE) test, which measures exhaust emissions under actual driving conditions, was enacted to regulate NO_x and particle number (PN) emissions.

Based on a report [4] published in 2013 by the International Council on Clean Transportation (ICCT), which first considered excessive NO_x emission and requested investigation, the official CO₂ emissions by car manufacturers from 2001 to 2015 decreased by approximately 30% (from 170 g to 120 g). However, the actual CO₂ emissions on the road were reduced by approximately 10%, from 183 g to 167 g. Hence, this was publicized as a difference of approximately 42% between the officially announced figures and the measured values under actual driving conditions in 2015.

This difference can be due to the exploitation of a loophole by the vehicle manufacturers during the fuel consumption test for certification. Furthermore, actual road conditions vary considerably

and are worse than lab conditions. This, in turn, results in high exhaust gas and fuel consumption. Therefore, the Economic Commission for Europe enacted and implemented the WLTP, which is more representative of real-world driving conditions compared to NEDC used for measuring exhaust gas and greenhouse gas, for newly manufactured vehicles effective September 2017. Additionally, the necessity of the test for exhaust gas and fuel consumption under actual road conditions was established. The Technical Committee on Motor Vehicles determined the test timing and applicable standards for the RDE test.

When WLTP mode was applied, it was determined that CO₂ emission unavoidably increased when compared to the values measured under NEDC to driving cycle, gear transmission (MT cars), air temperature, vehicle load, and other factors as shown in Table 1 [5]. Specifically, in case of diesel vehicles, the fuel consumption was further worsened because the EGR operation range was expanded until the load on the engine was high enough to reduce NO_x and cope with the RDE mode that examines exhaust gas on actual roads.

Table 1. Parameters with potential impact on CO₂ emissions with different definitions in NEDC and WLTP [5].

Factors	NEDC	WLTP	CO ₂ Effect
Test cycle	NEDC Fixed gear positions	WLTC Vehicle specific gear positions	(-)
Road load determination	Driving cycle Gear shifting strategy for MT		
	Tire size and type	Worst tire (2nd worst if >3 tires with different rolling resistances)	(+) slightly
	Tire tread depth	>3000 km running-in or 50–90%	(+) slightly
	Aerodynamics	Worst bodywork, no definitions in movable parts	(+) slightly
Test temperature	Soak area	20–30 °C	(+)
	Test cell	20–30 °C	(+)
Vehicle masses	Test mass	Curb weight + 100 kg	(+)
	Inertia	Discrete classes	
	Rotating masses	Simulation of total inertia of the vehicle as driven on the road	
Others	Running in Pre-conditioning cycle	>3000 km Diesel: 3 × NEDC Petrol: 1 × UDC	
	Battery state of charge	Not defined	(+)
	4WD vehicles	1-axle dynamometer possible	(+)

Yu et al. [6] analyzed driving characteristics in major cities in China by using traffic information and compared their results with those obtained from the WLTC mode. They conducted tests with NEDC and WLTC modes for 30 different car models to verify the effect of the cycles on the fuel consumption. Their results are similar to those in Table 2 [5], which are provided by ICCT. The results consider the effect of different test loads and ambient temperature on fuel consumption.

Table 2. Total estimated impact of switching from NEDC to WLTP [5].

Regulatory Issue	Driving Cycle	Vehicle Mass	Temperature	Total Impact 23 °C	Total Impact 14 °C
Impact on CO ₂	+2.1%	+3.5%	+1.9%	+5.7%	+7.7%

Piotr et al. [7] compared the emission levels in case of the NEDC, FTP-75, and WLTP modes for gasoline and diesel cars. A switch from NEDC to WLTC revealed changes in NO_x, PM, and PN, which were higher than those in THC, CO, and CO₂. However, their results were limited to Euro5 and Euro6 compliant cars. In another study by Piotr et al. [8], they measured the fuel consumption as per each mode for five different car models. The results revealed that the fuel consumption was the lowest under the NEDC mode for cars with manual transmission. However, the results were still insufficient

to verify the fuel consumption based on cycle difference because of the different vehicle types (weight, tractive resistance) and engine displacements.

Melo et al. [9] compared the fuel consumptions for three types of driving cycles for internal combustion engines, hybrid electric vehicles (HEVs), and electric vehicle (EVs). They described the difference in the fuel consumption between NEDC and WLTC as approximately 9.5–13.5% for internal combustion engines. However, this result was not obtained via experiment. They utilized the CO₂ data that were available online at JATO Dynamics. Therefore, the results could include various deviations and could not explain the effect of test cycle sufficiently.

Sarp et al. [10] examined the effect of seven types of driving cycles, namely, NYCC, FTP75, NEDC, WLTC, US06, HWFET, and CADC, for cars with eight power train technologies. Hence, some improvements were expected for all cycles in the Miller cycle and downsizing engines. A prominent improvement was obtained under low-speed zones, such as downtown operation of HEVs and EVs.

The aforementioned studies [11] analyzed the effects of various power train types and driving cycles. However, the results were not validated via testing. Furthermore, many researchers proposed that experimental results are only for cars that comply with the existing emission regulations, such as Euro5 or Euro6. Therefore, there were very rare cases, which is the result for the specification satisfied the latest regulations, such as Euro6d-temp.

Pavlovic et al. [11] carried on tests and simulations on several cases to assess the potential total impact on the final reported type-approval CO₂ emissions. The analysis showed the biggest impact on CO₂ is coming from the changes in the road load determination procedure (~10% increase) and procedural changes concerning the test in the lab will bring another 8% and post-processing and declaration of results will result in difference of approximately 5%. Overall, the WLTP is likely to increase to the NEDC CO₂ emissions by approximately 25%. However, they did not provide quantitative input values, so it was difficult to grasp the CO₂ impact for each factor.

Pavlovic et al. [12] also tested the NEDC and WLTP cycle with vehicles that comply with Euro 6 emission standard. The worst case (TMH, test mass high) showed CO₂ emission and energy demands to be on average 11% and 44% higher than NEDC, respectively, and best scenario (TML, test mass low) has on average 1% higher CO₂ emissions. Their results also showed that higher vehicle inertia and road load along with the higher speed are the key parameters of the new procedure that contribute to the increased CO₂ emissions. However, their study was the result of testing vehicles that met Euro 5 and 6 regulations.

The objective of this paper was to investigate the different effect of factors and technologies related to fuel consumption between NEDC and WLTP modes. In this study, 1.6 L gasoline and diesel engine vehicles that meet the latest Euro-6d temp regulations were applied to clarify the contribution of fuel consumption parameters such as vehicle weight, tire rolling resistance, air resistance and idle stop-start under NEDC and WLTP. In order to analyze the qualitative and quantitative impact of these factors, a commercial AVL simulation tool was used, and vehicle tests were performed simultaneously to complement the previous research relied on simulation method. In addition, the recently spotlighted 48 V system was applied to diesel vehicle, and tests were conducted to investigate the quantitative fuel consumption effects on energy recovery, torque assist and extended idle stop-start technology by 48 V MHSG (mild hybrid starter and generator) machine. From this research, we clarified the effective factors which has good effect on better fuel economy.

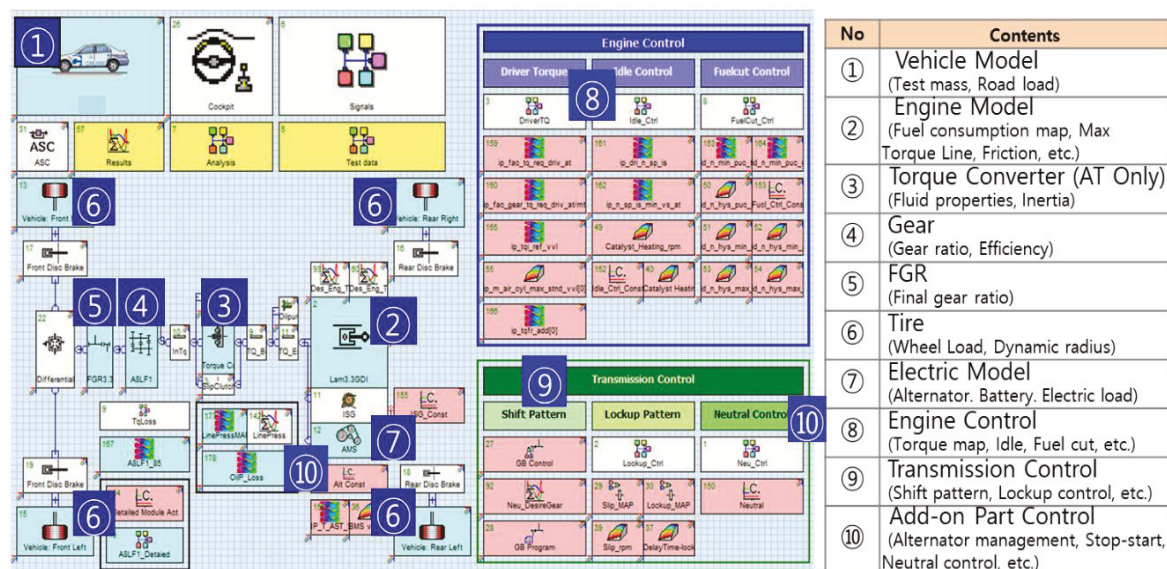
2. Simulation and Experimental Method

The test vehicles used in this study were compact SUV with 1.6-L gasoline turbo and diesel engine (Table 3), dry type 7-speed dual clutch transmission with idle stop–start technology. The catalyst selection and ECU calibration to comply with the Euro 6d-temp exhaust regulation were completed.

Table 3. Engine specifications in the test car.

Fuel Type	Gasoline	Diesel
Displacement (cc)	1591	1598
Compression Ratio	10.0	15.9
Max Power (kW/rpm)	130/5500	100/4000
Max Torque (Nm/rpm)	265/1500	320/2000
Cylinder layout	In-line 4-cylinder	In-line 4-cylinder
Emission Control System	WCC + UCC	cc_LNT + DPF + UF SCR
Emission	E6d temp	E6d temp

The fuel consumption simulation was conducted using CRUISETM from AVL, as shown in Figure 1. Furthermore, actual measurement data obtained from the gasoline/diesel engine and transmission efficiency at each gear were used for fuel consumption simulation. Additionally, real ECU/TCU mapping data such as desired torque map, gear shift pattern, and target fuel cut-in speed were applied to perform sophisticated analysis.

**Figure 1.** Vehicle simulation model by AVL CRUISETM.

Prior to vehicle test on chassis dynamometer, the full load performance and BSFC of gasoline and diesel engine were measured through engine dynamometer. Transmission efficiency was also measured for various speeds and input torques.

Figure 3 shows the average transmission efficiency at each speed with respect to the transmission fluid temperature. Specifically, the 7th speed has 5% higher transmission efficiency at 70 °C than at 40 °C.

To analyze the attribution of the test condition factors, the effect of the road load with respect to the changes in the vehicle weight, drag coefficient, and tire rolling resistance were estimated. The validation test was also conducted with actual vehicles. The calculation method of the road load is as follows.

Figure 2 shows the maximum torque and power curve of gasoline and diesel engines and the fuel consumption rate applied in this study. Specifically, the diesel engine exhibited a lower fuel consumption rate by approximately 8–18% than the gasoline at BMEP 4bar (50~51 Nm).

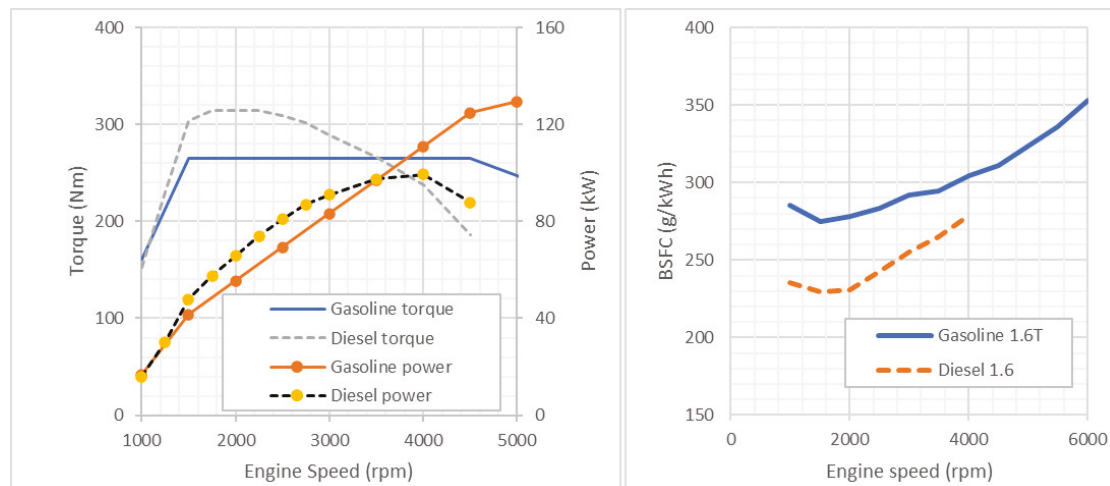


Figure 2. Full load performance and brake specific fuel consumption at TQ = 50.6 and 50.9 Nm (Gasoline/Diesel).

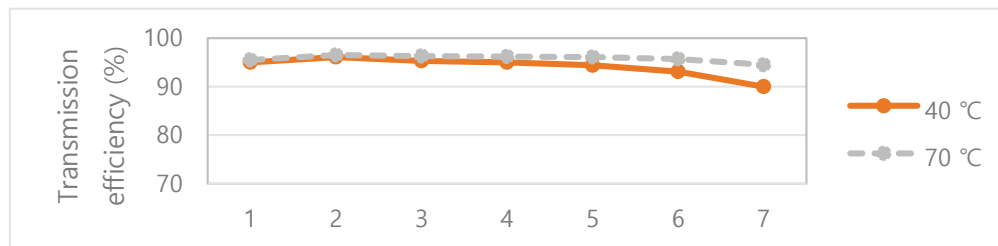


Figure 3. Average transmission efficiency at each gear speed with respect to fluid temperature.

The road load of the cars consists of mechanical drag and aerodynamic drag as shown in SAE J2263. Furthermore, the mechanical drag is composed of the tire rolling resistance, which is not associated with the vehicle speed and factors that are proportional to vehicle speed. Thus, air resistance was configured to be proportional to the square of the vehicle speed. However, the drag coefficient was modeled as a fifth order function of the yaw angle of the crosswind to the vehicle. Thus, in our case, temperature compensation was applied only for coefficients A and C, which differs from that prescribed by SAE. Thus, the reference air pressure value applied for coefficient C was also different.

Additionally, the actual measurement value was used for the projected front area of the vehicle. Table 4 shows the coefficient of drag for different tire types and options for the vehicle used in this study. Each Cd value was measured in a wind tunnel. In this study, coastdown tests were conducted on the 16-inch base option with the lowest Cd and 19-inch with the highest Cd.

Table 4. Coefficients of drag for each tire type and option.

Tire and Wheel	Bumper	Brake Type	Mud Guard	Cd	Diff	Remarks
215/70R16	A	A'	X	0.339	Base	The lowest Cd
245/45R19	A	A	X	0.352	+0.013	The highest Cd

We used VMS3200 to measure coastdown time on track. An on-board anemometer was specifically used to measure wind direction and velocity in real driving conditions. Figure 4 shows the coastdown measurement equipment and the requirements and accuracy of the measurement system.

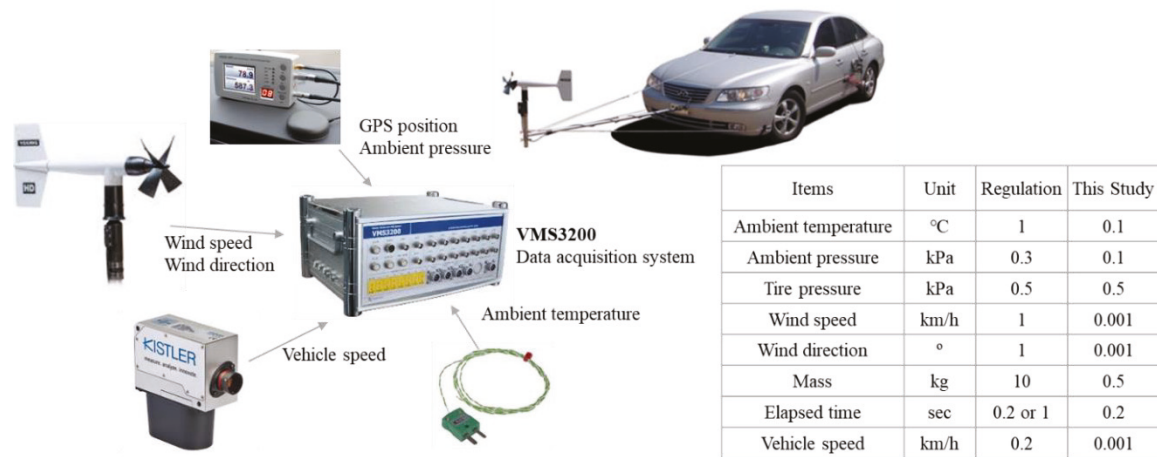


Figure 4. Coastdown test measurement system and its accuracy.

Subsequently, the neutral coasting time from 135 to 15 km/h, vehicle speed, wind velocity (speed, direction), and ambient temperature were measured five times in both directions of the straight road. Thus, they were measured a total of ten times. The measurements were used to calculate a total of eight unknown coefficients (A_m , B_m , C_m , a_0 , a_1 , a_2 , a_3 , and a_4) via a linear regression method. These coefficients were calibrated at standard temperature and atmospheric pressure conditions before the final road load coefficient was determined. The final road load coefficient was proportional to the second order of the vehicle speed (Figure 5 and Equation (1)).

$$-M_e \frac{dV}{dt} = A_m + B_m V + C_m V^2 + \frac{1}{2} \rho A V_r^2 C_d(Y), \quad C_d(Y) = a_0 + a_1 Y + a_2 Y^2 + a_3 Y^3 + a_4 Y^4 \quad (1)$$

$$F = f_0 + f_1 V + f_2 V^2$$

where M_e : Effective Vehicle Mass, dV/dt : Acceleration, V : Vehicle Velocity, V_r : Apparent Wind Relative to Vehicle, $C_d(Y)$: Coefficient of Aerodynamic drag at Yaw angle Y .

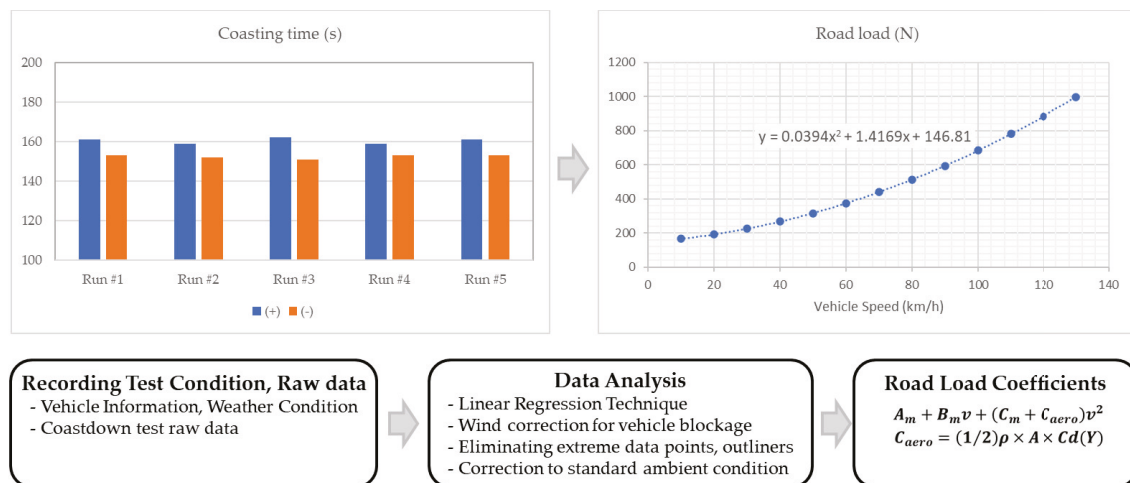


Figure 5. Worldwide harmonized light vehicles test procedure (WLTP) road load determination via coastdown method.

The equation to determine the road load for the NEDC test, as per (EU) 2018/1002 regulation using the WLTP tractive resistances (calculated above) is presented in Table 5. From NEDC to WLTP, the cycle energy increased by approximately 2.6–2.7 times based on the weight, average speed, and increases in the speed variation.

Table 5. Factors for fuel consumption contribution analysis in NEDC and WLTP.

Factors	Mass	Tire RRc	Aero	Idle Stop-Start	48 V System
Level	−10% Base (TML) +10%	−10% Base (TML) +10%	−10% Base (TML) +10%	Disable Enable -	Disable Enable -
Method	8 Simulations (TML, TMH, ±10% 6 cases) 2 Tests (TML, TMH)			2 Simulations 2 Tests	2 Tests

In this study, the fuel consumption in NEDC and WLTP was investigated with respect to vehicle weight, drag of coefficient, tire RRc(rolling resistance coefficient), Idle stop–start and 48-V system. Specifically, the vehicle weight, Cd, and tire RRc increased and decreased by 10% relative to the base TML (Test mass low) (Table 5).

Figure 6 shows the relation between vehicle road load and tire size used in this test and simulation. Additionally, the driving energy per distance of the test cycle is shown in this figure. The driving energy is calculated by integrating the road load with respect to the speed of the driving cycle, i.e., NEDC or WLTP.

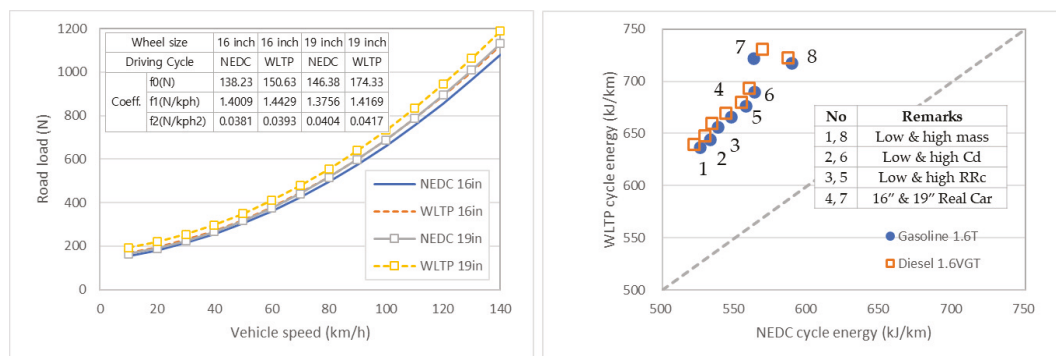
**Figure 6.** Road load curve and driving energy per distance according to the test cycle.

Figure 7 shows the overall schematic diagram of the chassis dynamometer and exhaust gas analyzer used for exhaust emissions and fuel consumption in this study. Measurement of vehicle emission gas was performed with a CVS (constant volume sampler) tunnel and emission gas analyzer. In the chassis dynamometer, real-time emission gas and vehicle data were collected for each driving speed by driving in the NEDC and WLTP modes. Furthermore, the final exhaust gas concentration was analyzed via the analysis of the sampling bag containing the exhaust gas. Gasoline and diesel fuel consumption were calculated by using carbon balanced Equations (2) and (3).

$$FC_{gasoline}[l/100\text{ km}] = \left(\frac{0.1206}{\rho_{fuel}} \right) \times \{ (0.829 \times HC) + (0.429 \times CO) + (0.273 \times CO_2) \} \quad (2)$$

$$FC_{diesel}[l/100\text{ km}] = \left(\frac{0.1165}{\rho_{fuel}} \right) \times \{ (0.858 \times HC) + (0.429 \times CO) + (0.273 \times CO_2) \} \quad (3)$$

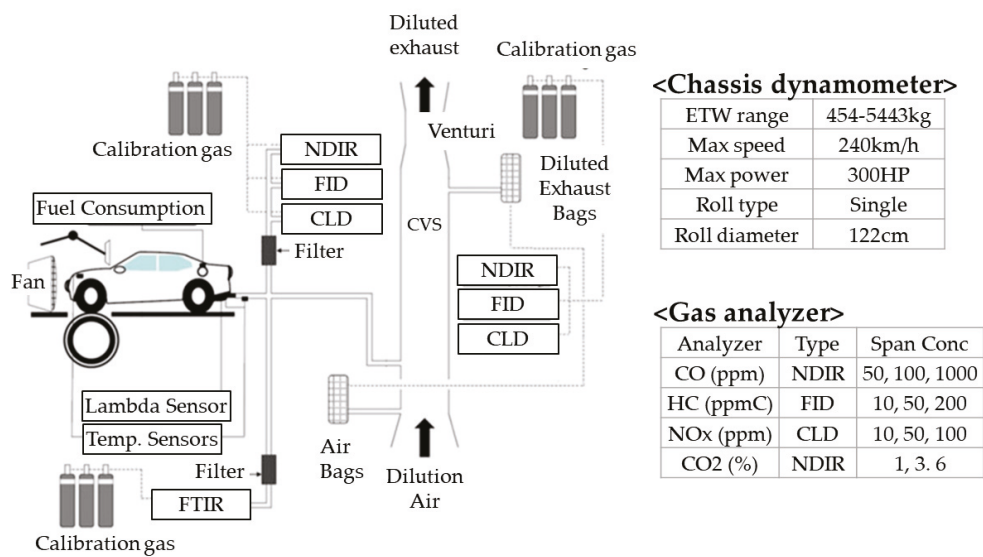


Figure 7. Schematic diagram of chassis dynamometer and emission measurement system.

3. Results and Discussion

Prior to this study, repeated tests were conducted to confirm the deviation of fuel consumption test in the chassis dynamometer. Given that this study only tested in test mass low (TML) and Test mass high (TMH) conditions for gasoline and diesel engines, it was necessary to check the errors in the test results. Based on the results of five repeated tests, under NEDC and WLTP modes, for 1.6 T gasoline engine, the standard deviation of NEDC was approximately 0.009(0.11%), and the standard deviation of WLTP was approximately 0.016(0.21%)

Figures 8 and 9 present all the simulation and experimental results conducted for NEDC and WLTP modes for the gasoline and diesel engine vehicles in this study. From these simulation and experimental results, we found that the test results can be used to determine the reliability and appropriateness of the simulation results.

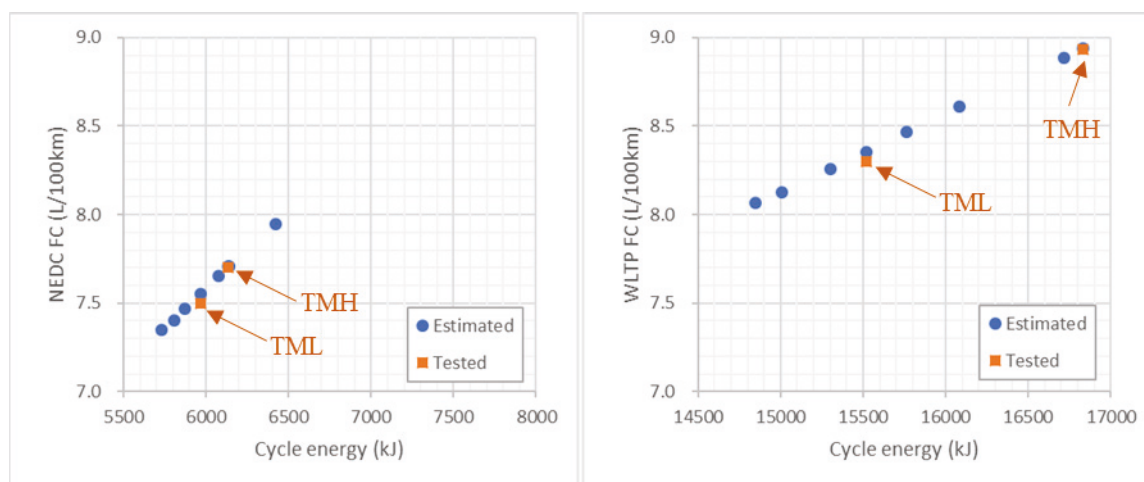


Figure 8. Fuel consumption under NEDC and WLTP according to the cycle energy of 1.6-L gasoline T-GDI.

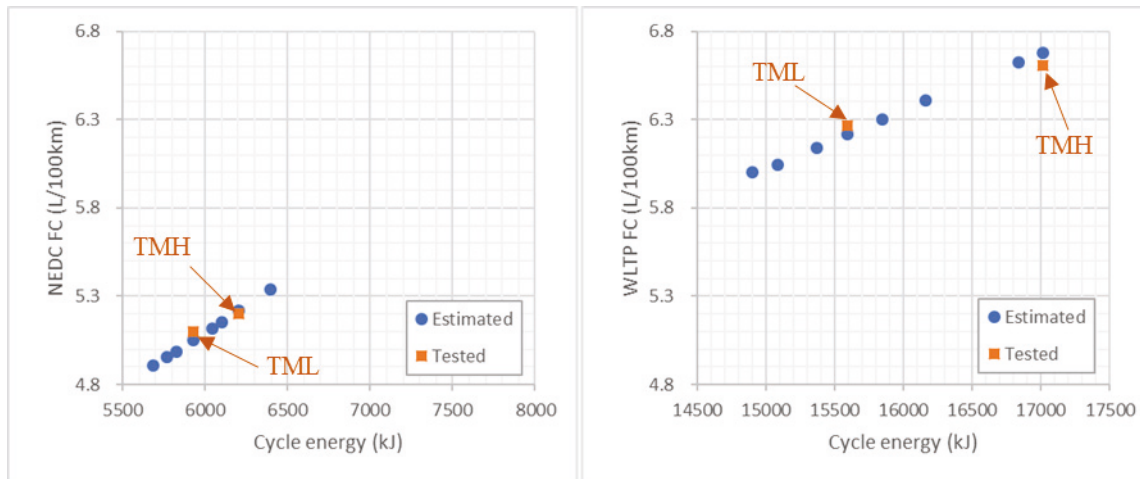


Figure 9. Fuel consumption under NEDC and WLTP according to the cycle energy of the 1.6-L diesel engine.

The effects of NEDC and WLTP mode on the fuel consumption with respect to road load factors, idle stop–start, and 48-V system technology are explained below.

3.1. Driving Cycle Effect

To determine the cycle energy according to the drive mode, the road load and test vehicle weight were considered. A margin of 3% was provided on the test vehicle weight by considering the rotational inertia of the wheel and tire.

$$E = \sum_{i=1}^n \left((f_0 + f_1 v_i + f_2 v_i^2) v_i \Delta t + 1.03 \times Mass \times \frac{v_i - v_{i-1}}{\Delta t} \right) \quad (4)$$

Figure 10 shows the speed and acceleration between WLTP and NEDC modes. The maximum acceleration under the WLTP condition tended to increase according to drive cycle difference. Table 6 indicates the cycle energy and CO₂ effect by simulation under two modes. The cycle energy of WLTP increased by 18% than that of NEDC mode. From the simulation result of WLTP mode, CO₂ effect increased by 2.7–3.0% for the diesel vehicles and by 4.8–4.9% for the gasoline turbo vehicle when compared with the results under the NEDC condition. According to Pavlovic et al. [11], the effect of the cycle difference was 5.88% for gasoline and 5.24% for diesel vehicle. In case of diesel vehicle, the effect may vary depending on the engine displacement.

Figures 11 and 12 show the distribution of the operating time and fuel amount for gasoline and diesel vehicle under NEDC and WLTP cycles. In this figure, the darkness in shade means high contribution. The engine operation zone significantly increased under WLTP compared to that under NEDC, due to increases in engine load and speed. In particular, the dotted line is an additionally expanded area compared to NEDC. In the case of the diesel engine, due to the increase in high load operating time, the EGR operation zone and operating time for reducing NO_x emissions were increased. Thus, this resulted in the deterioration of fuel consumption.

However, under the WLTP mode, the fuel amount did not degrade because the running time under warmed engine condition was increased compared to NEDC. Figure 13 shows the behavior of the coolant and engine oil temperatures per test cycle mode for the diesel vehicle. Figure 14 shows the distribution of the coolant and oil temperatures. The operating time was increased above 80 °C and engine operates under better engine efficiency in the WTLTP mode compared to NEDC. This result does not imply that WLTP mode itself always increases fuel consumption rate [L/100 km].

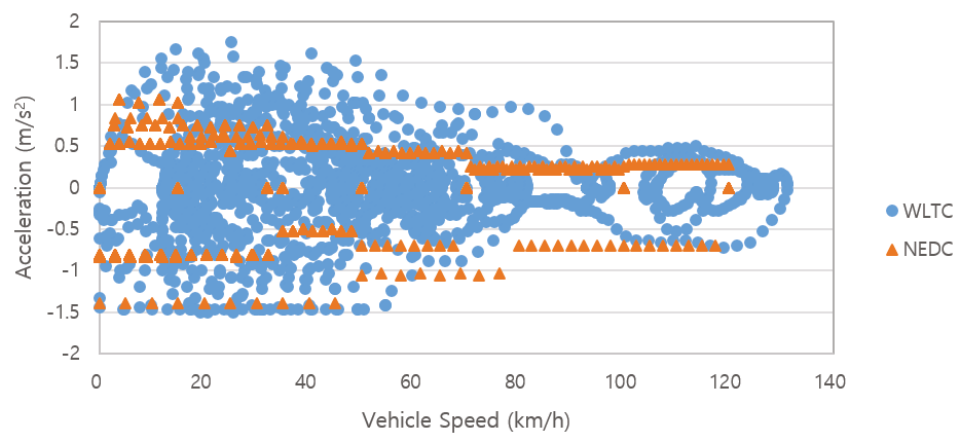


Figure 10. Speed and acceleration of WLTC and NEDC.

Table 6. Cycle energy and CO₂ effect by simulation according to the test mode (per test mass of 1718 kg, $f_0 = 150.632$ N, $f_1 = 1.44292$ N/kph, $f_2 = 0.039255$ N/kph²).

Category	Unit	NEDC	WLTP	Remarks
Distance	km	10.9	23.3	
Max Acceleration	m/s/s	1.06	1.75	
Min Acceleration	m/s/s	−1.39	−1.50	
Cycle Energy	kJ	6179	15,488	Including rotating inertia effect
	kJ/km	565	666	
	%	Reference	18%	
CO ₂ effect	Gasoline	Reference	+4.8–4.9%	By CRUISE Simulation
	Diesel	Reference	+2.7–3.0%	

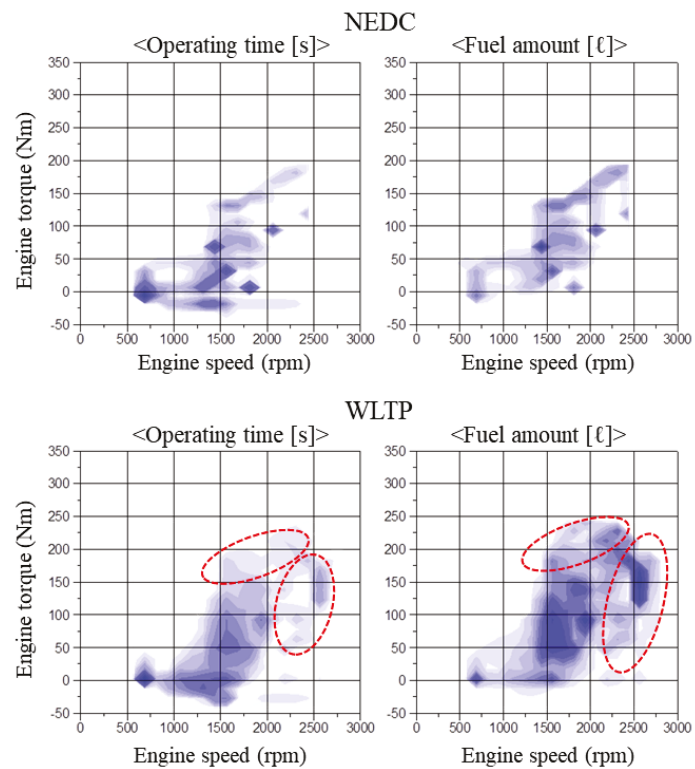


Figure 11. Operation time and fuel amount according to the driving cycle of the gasoline vehicle.

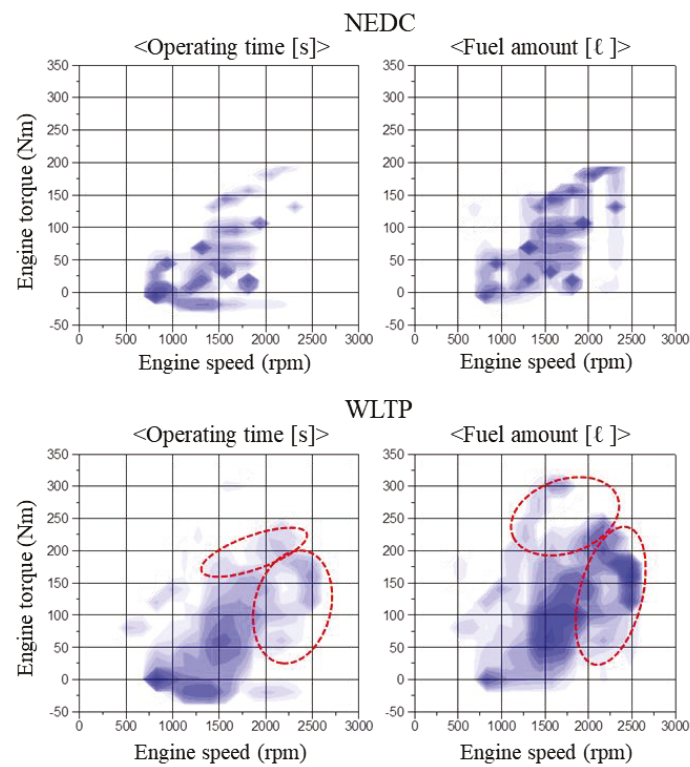


Figure 12. Operation time and fuel amount according to the driving cycle of the diesel vehicle.

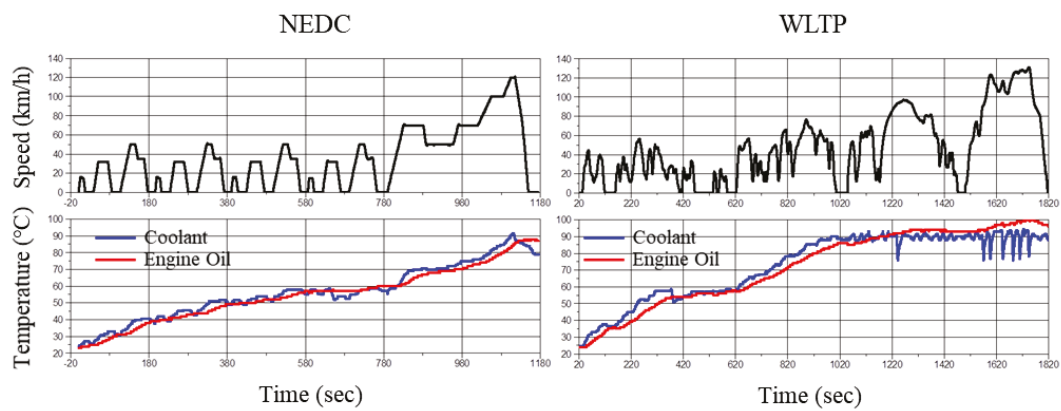


Figure 13. Coolant and engine oil temperature distribution according to the test cycle.

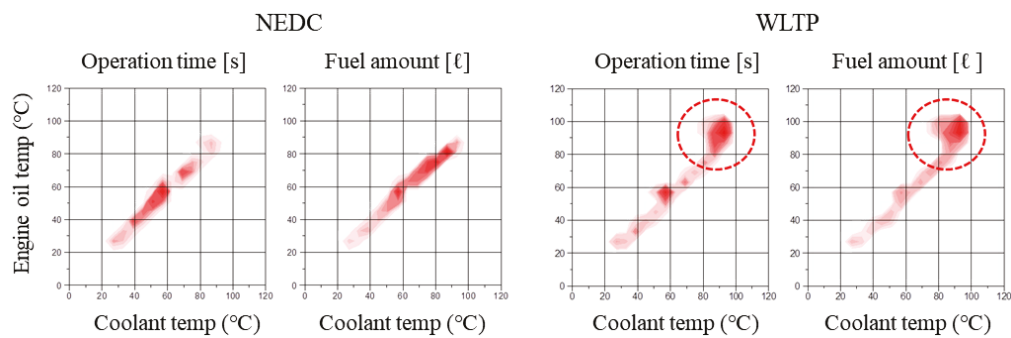


Figure 14. Operation time and fuel amount according to coolant and engine oil temperature (Diesel 1.6 VGT engine).

3.2. Test Mass Effect

In the case of the new regulations, the test vehicle weight was changed to that of cars that are currently available for sale. Hence, based on the existing NEDC, the test vehicle weight was the equivalent test weight (ETW) according to the reference mass (RM). Meanwhile, with the new WLTP regulation, the test mass was determined considering the RM, gross vehicle weight (GVW), and full option weight.

In the case of the vehicles used in this study, the test mass was increased by 18–74 kg for the minimum option and by 213–227 kg for the maximum option with respect to the test mass in the existing regulation, as shown in Table 7.

Table 7. Determination of the test mass.

Items		NEDC	WLTP
Test mass	Lowest Trim	1590 kg	1718–1774 kg
	Highest Trim	1810 kg	1913–1927 kg
Determination of test mass		ETW	RM + 15%* (GVW-(RM + option))

As the test mass and maximum acceleration were significantly increased in WLTP when compared with those of NEDC, the vehicles consumed more fuel in the acceleration zone. Specifically, increases in test mass affected the tire rolling resistance additionally. This was attributed to the increases in fuel consumption at constant speed. Figures 15 and 16 show the contribution of the fuel consumption to the driving conditions with respect to the drive modes for gasoline and diesel vehicle. The fuel consumption increased during acceleration, and the contribution in the overall fuel consumption was also increased by 5–6% in both types of vehicles.

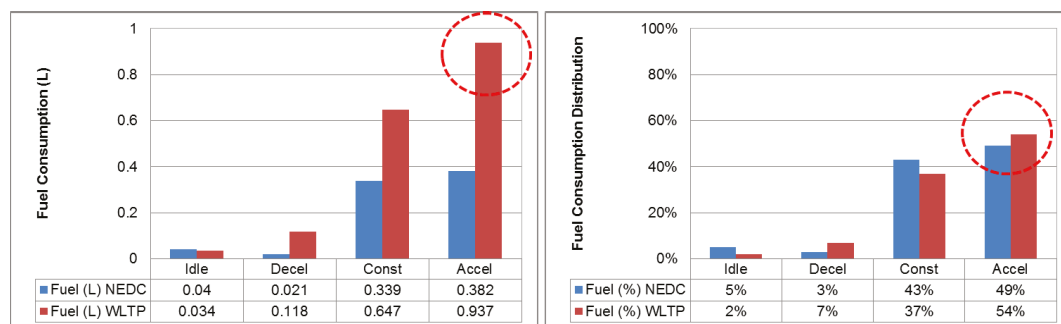


Figure 15. Contribution of the fuel consumption to the driving conditions (1.6 L turbo gasoline).

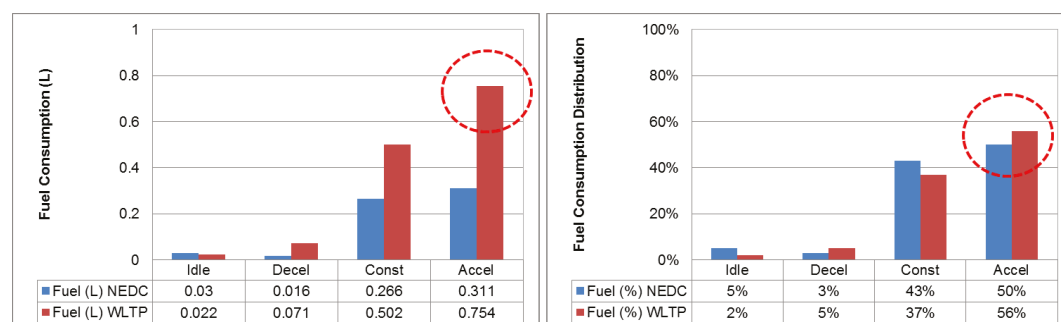


Figure 16. Contribution of the fuel consumption to the driving conditions (1.6 L diesel).

The effects of the test mass on the cycle energy and fuel consumption are presented in Figures 17 and 18. Once road load is determined from coastdown testing, the road coefficients for NEDC are determined according to the regulation. Therefore, the energy can differ under the

NEDC cycle even with the same NEDC ETW. Based on the NEDC test weight, the cycle energy values differed by 4.0–7.8%, and the fuel consumption differed by 2.7–5.7% with NEDC. For WLTP, the cycle energy increased by approximately 4% when the road load increased by 10%. Consequently, the fuel consumption increased by approximately 6.4–6.6%.

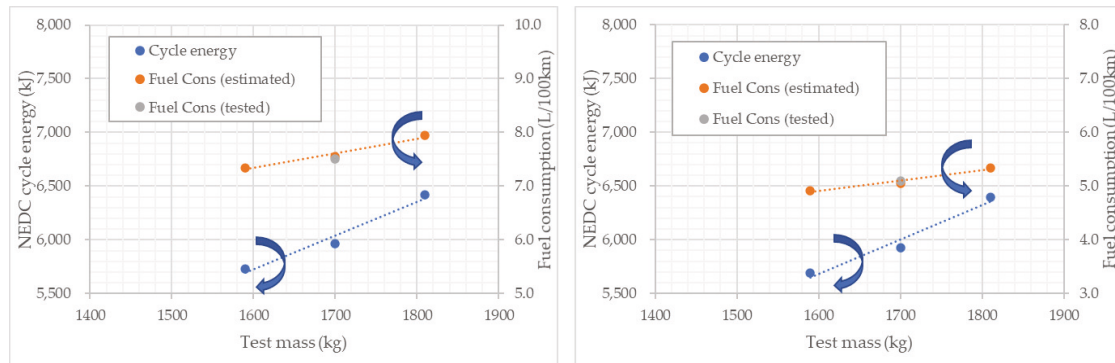


Figure 17. Effect of the test mass on the energy and fuel consumption in NEDC (left/right: gasoline/diesel).

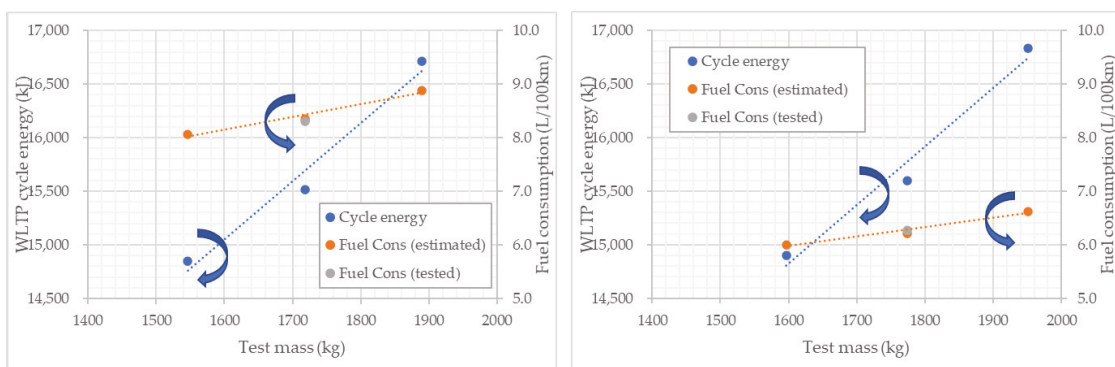


Figure 18. Effect of the test mass on the energy and fuel consumption in WLTP (left/right: gasoline/diesel).

Test mass is the most influential factor in fuel consumption in each test cycle, and Pavlovic et al. [12] and Moon et al. [13] also showed similar trends in their simulation and test.

3.3. Tire Rolling Resistance Effect

The tire rolling resistance is usually less affected by vehicle speed. The effects of the tire rolling resistance on the cycle energy and fuel consumption are shown in Figures 19 and 20. When the tire rolling resistance was changed by 10%, the cycle energy was changed by approximately 1.7–1.9% under NEDC. On the other hand, it was changed by approximately 2% under WLTP. However, the fuel consumption effect in both cycles were similar as 1.1–1.4%. Dimaratos et al. [14] also showed that limitation of rolling resistance by 10% offers a fuel consumption benefit of 1.2% and 2.2% in NEDC and WLTP, respectively, in diesel vehicle simulation results.

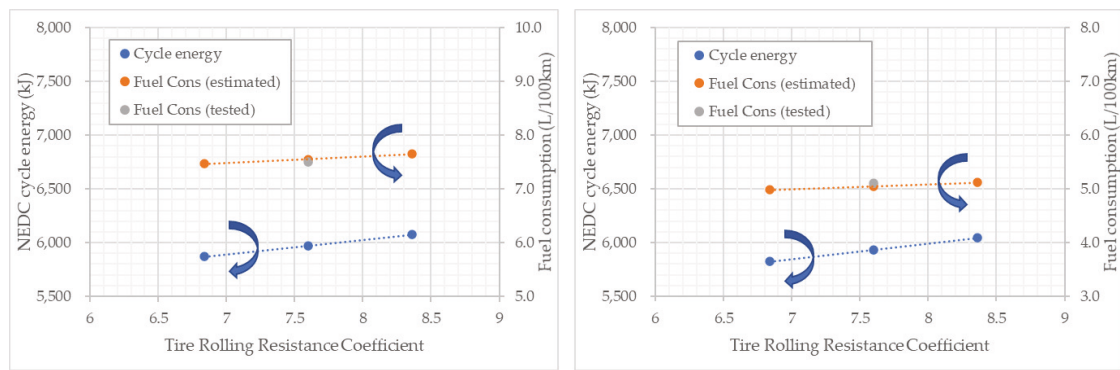


Figure 19. Effect of the tire rolling resistance on the cycle energy and fuel consumption in NEDC (left/right: gasoline/diesel).

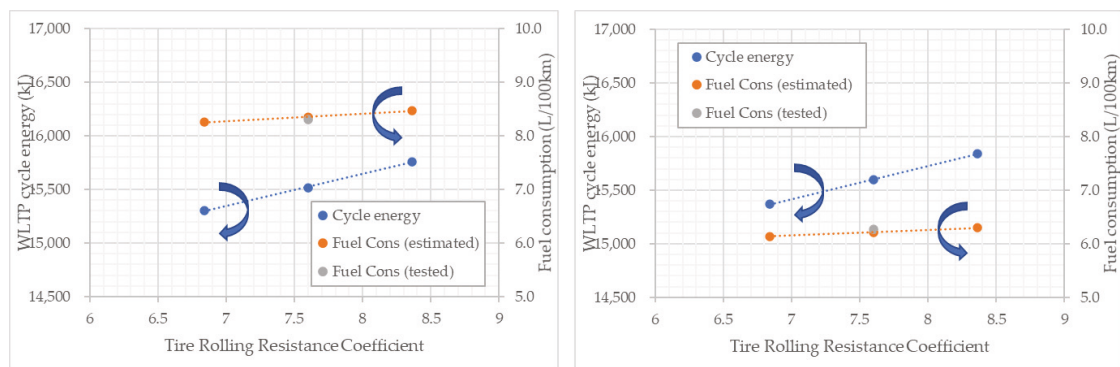


Figure 20. Effect of the tire rolling resistance on the cycle energy and fuel consumption in WLTP (left/right: gasoline/diesel).

3.4. Coefficient of Drag Effect

Aerodynamic drag typically increases as a secondary function of vehicle speed. Therefore, it affects fuel consumption during a high-speed driving condition, such as on a highway.

The effects of the aerodynamic drag (drag coefficient) on the cycle energy and fuel consumption are presented in Figures 21 and 22. As the drag coefficient was modified by 10%, the cycle energy changed by 2.7–3.0% in the NEDC, while it changed by 3.3–3.6% in WLTP. Accordingly, the fuel consumptions in NEDC were changed by 1.9–2.1% and 2.7–3.1% in WLTP. The increased effect of aerodynamics might be due to increases in the average speed and maximum speed in WLTP when compared with those in NEDC. Dimaratos et al. [14] presented that 10% reduction in aerodynamic force decreases fuel consumption by 2% and 2.8% in NEDC and WLTP, respectively.

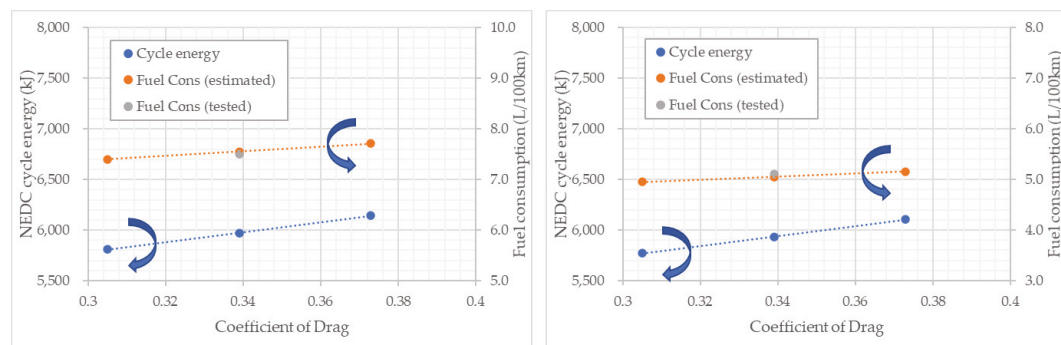


Figure 21. Effect of the aerodynamic drag (drag coefficient) on the cycle energy and fuel consumption in the NEDC mode (left/right: gasoline/diesel).

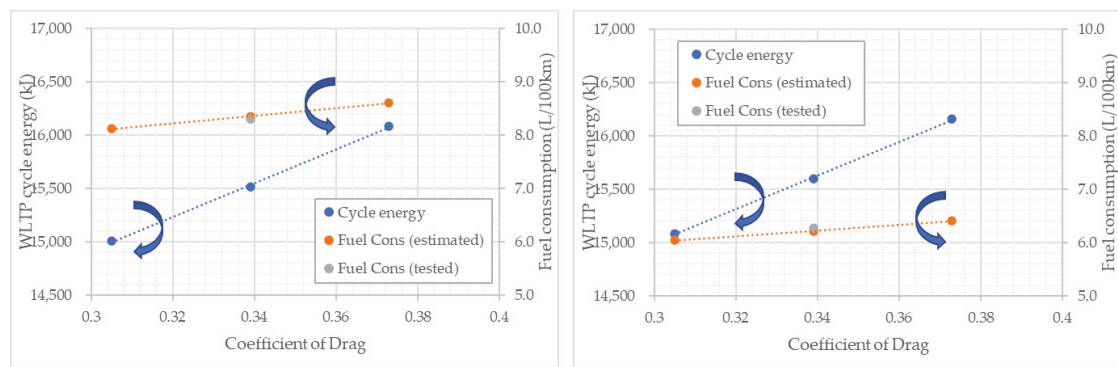


Figure 22. Effect of the aerodynamic drag (drag coefficient) on the cycle energy and fuel consumption in WLTP (left/right: gasoline/diesel).

The influence of the vehicle's drag coefficient on fuel consumption went up in WLTP regardless of the type of engine, and Pavlovic et al. [12] also showed an increase in the impact of aerodynamics on fuel consumption from NEDC to WLTP. Moon et al. [13] and Dimaratos et al. [14] also showed similar results through their simulation analysis.

3.5. Effect of Idle Stop-Start Technology

The idle stop–start technology, which is common in the European market but not in the North American market, stops engine operation when a vehicle is stopped, to reduce emissions and save fuel consumption. Given that the idle stop frequency is higher by approximately 24% in the NEDC mode, the fuel consumption effect of the idle stop–start technology is known to be high. Meanwhile, under the WLTP mode, the idle stop frequency is approximately 13% of the total modes, the vehicle weight and drive speed are increased, and the fuel consumption required for driving is increased. Therefore, the effect of idle stop–start technology is reduced when compared to that under the NEDC mode. The effects of the idle stop–start on the fuel consumptions of 1.6-L gasoline and diesel engines for each test cycle are presented in Table 8. By changing the test cycle, the effect of idle stop–start technology was reduced by approximately 2.0–2.5%. It is still a useful tool to ensure compliance with the CAFE regulation and lower fuel consumption in urban driving condition. Dimaratos et al. [14] also showed the effect of stop–start on CO₂ emissions is 2.5–4.8% in NEDC and 1.2–2.6% in WLTP. Stop–start technology definitely has a stronger effect on NEDC, owing to the longer idling period compared to WLTP.

Table 8. Comparison of fuel consumption for idle stop–start by test cycle.

Remarks		NEDC	WLTP	Difference
Total drive time		1180 s	1800 s	620 s↑
Idle stop–start time		280 s (23.7%)	226 s (12.6%)	54 s↓
The effect of Stop/start on fuel consumption	1.6 L Gasoline	3.9%	1.5%	2.4–2.5%p↓
	1.6 L Diesel	3.0–3.1%	1.0–1.1%	2.0%p↓

3.6. Effect of the 48-V System

A P0 type of 48-V system was implemented on the diesel 1.6-L engine. For the 10-kW grade MHSG (mild hybrid starter and generator), the electric energy with the regenerative braking was stored during deceleration to assist the torque in starting-and-stopping and driving and to reduce fuel consumption by replacing the electric energy required for driving the vehicle. The idle stop–start technology was applied in the zone with speeds lower than 20 kph during deceleration. This is to ensure that the fuel consumption in the rolling idle zone before stopping the vehicle is not incurred. Figure 23 shows the control strategy of the 48-V motor/generator with respect to the vehicle speed.

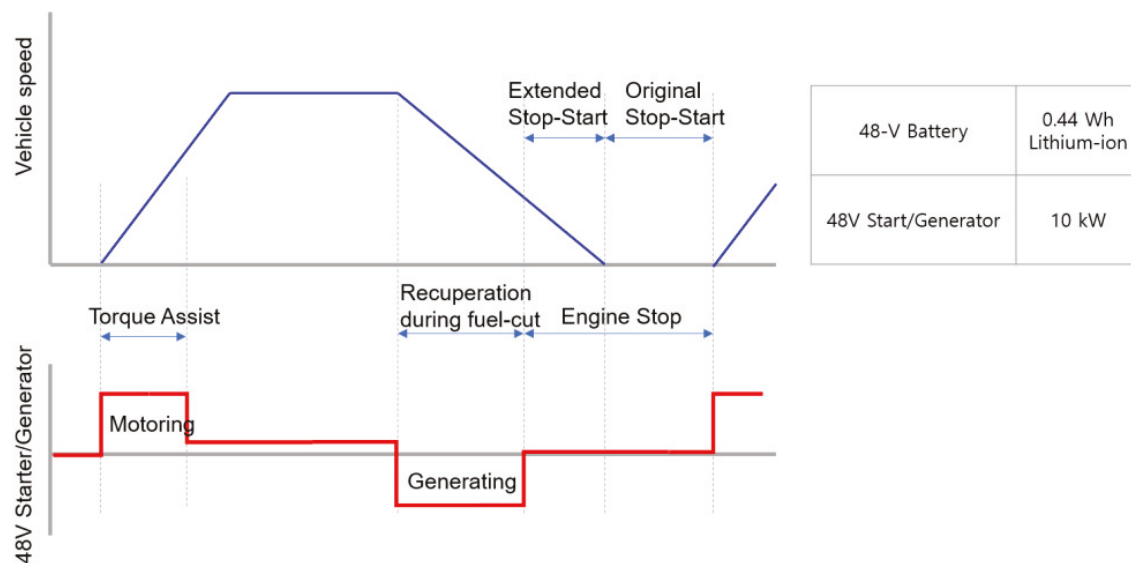


Figure 23. Strategy for capacity and control of 48-V starter/generator for improving fuel consumption.

Under the WLTP, the fuel consumption of the 48-V system improved by approximately 1.5% when compared with that under the NEDC, as shown in Table 9. Hence, along with the idle stop–start technology, the 48-V system is a cost-effective technology for lowering the fuel consumption in WLTP.

Table 9. Fuel-consumption improvement of the 48-V system according to drive cycle (1.6 L diesel engine).

Category		NEDC	WLTP	Remark
Fuel-consumption improvement	Total	4.7%	6.2%	
	Recuperation	3.2%	4.0%	Assist torque, coping with the electric load of the vehicles
	Extended stop–start	1.5%	2.2%	Operated in $0 < \text{Speed} < 20 \text{ km/h}$

Figure 24 presents measured actual torque and behavior of the battery charging in the 48-V motor under NEDC and WLTP. In the WLTP, there are several deceleration events when compared with that of the NEDC mode. Therefore, the regenerative energy of the 48 MHSG was maximized during deceleration, and MHSG also has a minor torque assistance during launching. The electric energy recovered by MHSG is stored in 12 V battery through a DC-DC converter, and is in charge of various electric loads of the vehicle. This ensured an improvement in fuel consumption under WLTP by approximately 1.5% compared with that in NEDC.

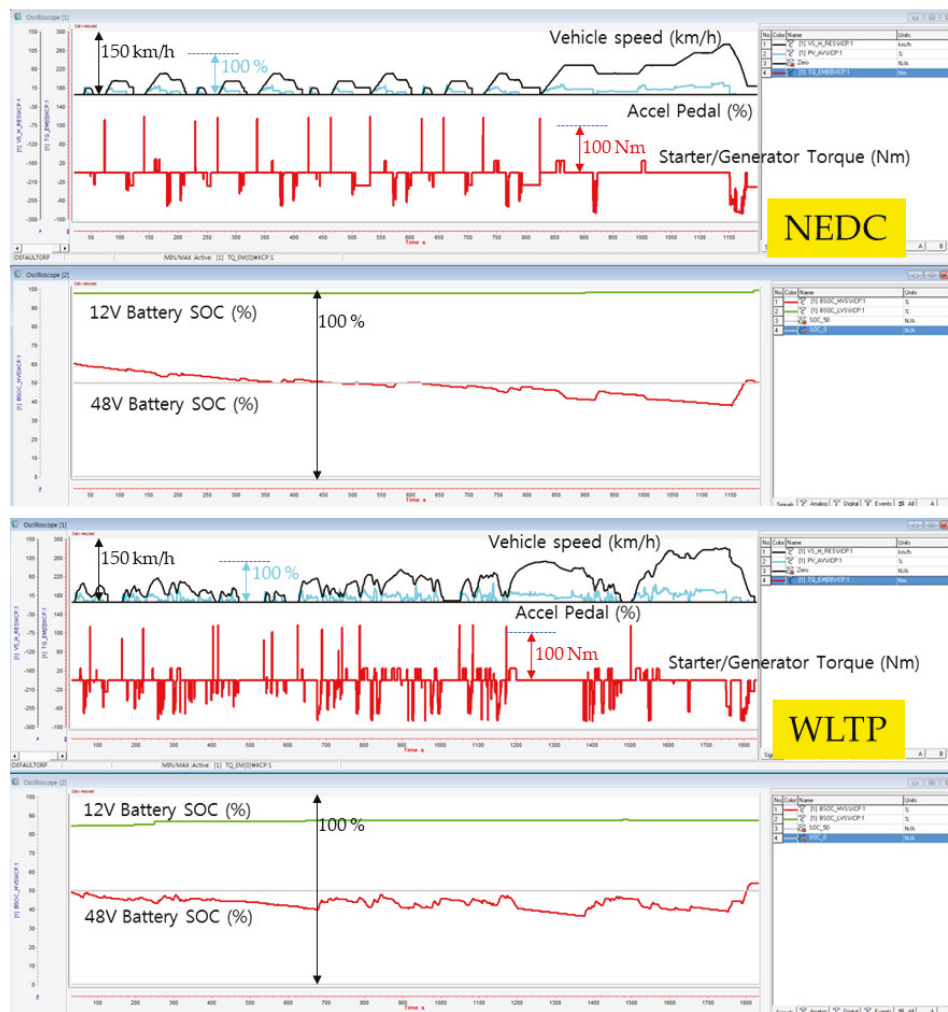


Figure 24. Torque control of the 48-V motor/generator in NEDC and WLTP modes (Top: NEDC, Bottom: WLTP).

4. Conclusions

When the test cycle was changed from NEDC to WLTP, the fuel consumption was worsened as the cycle energy was increased, due to increase in the acceleration and maximum speed. Conversely, the fuel consumption was also improved due to the increase in operating time after the vehicle was warmed up.

The engine load under the WLTP increased compared with that under the NEDC after considering the weight of the vehicles available for sale. Test weight affects the fuel consumption to the highest extent when compared to the other factors. Specifically, the effect of the vehicle weight increased as the acceleration increased in the WLTP when compared with that in the NEDC. This study showed that when the test mass was changed by 10%, the fuel consumption was affected by approximately 4% in the WLTP.

Under the WLTP, the average and maximum speed of the vehicles increased. This resulted in a higher effect on aerodynamic factors. This study showed that the fuel consumption in WLTP was affected by approximately 3.1% when the aerodynamic drag coefficient was changed by 10%. Thus, this was the sensitive factor that affected the fuel consumption together with vehicle weight.

The tire rolling resistance significantly affects the fuel consumption in low-speed driving conditions, such as urban cycle. Therefore, the effect of the tire rolling resistance on the fuel consumption was reduced because the average vehicle speed was increased in the WLTP mode compared with that in

the NEDC. The rolling resistance improved the fuel consumption by approximately 1.4% when the tire rolling resistance was changed by 10% in the WLTP. This indicated that the tire rolling resistance did not affect the fuel consumption as much as the other factors in WLTP.

The effectiveness of the idle stop–start technology on the fuel consumption decreased due to the reduction in idling interval in WLTP compared to NEDC. 48 V MHSG made it possible to realize about 6% fuel improvement by minor torque assist during launching, increased energy recovery during deceleration and extended stop–start technologies in WLTP.

Future Work

Further research is required on vehicles with manual transmission. The manual shift pattern prescribed in the regulation is applied in the NEDC mode. Furthermore, the shift pattern generated by the Steven Tool should be applied under the WLTP mode (692/2008/EC, Annex XXI, Sb-Annex 2). It is quite complex to calculate the WLTP gear selection. Therefore, future work will involve developing a tool which calculates optimum n-min drive for balancing exhaust gas and CO₂.

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Conflicts of Interest: The authors declare no conflict of interest.

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Article

Nonlinear Active Disturbance Rejection Control of VGT-EGR System in Diesel Engines

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Abstract: In this paper, a nonlinear active disturbance rejection control (NLADRC) strategy based on nonlinear extended state observer (NLESO) is proposed to solve the unmodeled dynamics, coupling and disturbance due to change of working point in the variable geometry turbine (VGT) and exhaust gas recirculation (EGR) system, so as to achieve accurate control of intake manifold pressure and mass air flow in a diesel engine. To achieve decoupling, the double-input double-output (DIDO) VGT-EGR system is decomposed into two single-input single-output (SISO) subsystems, and each subsystem has a separate nonlinear active disturbance rejection controller. At the same time, the convergence proof of the designed NLESO is also given theoretically. Finally, the NLADRC controller is compared with linear active disturbance rejection controller and proportional–integral–derivative (PID) controller. Through simulation, it is indicated that the proposed NLADRC controller has better transient response performance, resistance to external disturbance and robustness to the change of engine operating point.

Keywords: nonlinear active disturbance rejection controller; variable geometry turbine; exhaust gas recirculation; diesel engine

1. Introduction

As one of the main power sources of trucks and ships, diesel engines have the characteristics of strong power performance, good fuel economy and long service life. However, the increasing number of diesel engines makes energy shortage more serious. In addition, the exhaust emissions of diesel engines are mainly nitrogen oxide (NO_x) and particulate matter (PM), which are one of the main causes of haze formation. With the increasingly stringent emission regulations and the continuous improvement of power and economy requirements, variable geometry turbine (VGT) and exhaust gas recirculation (EGR) technology has become widely used in diesel engines. Part of the exhaust gas generated in the cylinder is delivered to the intake manifold through the EGR valve to reduce NO_x emissions. However, excessive exhaust gas entering the intake manifold can cause insufficient fuel combustion. The other part of the exhaust gas through the VGT valve drives the turbine, which drives the compressor to absorb more fresh air and improve the combustion efficiency. However, excessive exhaust gas emissions will reduce the exhaust gas back to the intake manifold, thus increasing NO_x emissions. In addition, complex system dynamics phenomena such as sign inversion, nonminimum phase and overshoot also exist in VGT-EGR system [1–3]. Therefore, the precise control of VGT-EGR system in diesel engines is very important to

reduce emissions. At the same time, cross coupling and complex system dynamics make it difficult to control VGT-EGR system accurately.

For the control of VGT-EGR system in diesel engines, many control strategies have been proposed by scholars. To achieve the required emission level and safe operation of the engine and turbocharger, Wahlström et al. [4] proposed a proportional–integral–derivative (PID) structure that controls the air/fuel ratio λ and the intake manifold EGR fraction x_{egr} . The air/fuel ratio λ is controlled by the EGR valve and the EGR fraction x_{egr} is controlled by the VGT position. In addition, Wahlström and Eriksson [5] proposed a control structure composed of PID controller and nonlinear compensator. Compared with the control structure without nonlinear compensator, this control structure deals with nonlinear effects and reduces EGR error. Beside PID control method, other commonly used control methods are also applied in the control of VGT-EGR system in diesel engines, such as sliding mode control [6,7], model predictive control (MPC) [8,9] and control Lyapunov function (CLF) [10,11]. From the perspective of system theory, Shen, Wu and Zhang applied the optimal control strategy based on multi-valued logic to the control of internal combustion (IC) engines [12,13] and hybrid electric vehicles (HEVs) [14]. In addition, for some related works on optimization, and advanced control strategies for IC engines, please refer to [15–19].

However, the VGT-EGR system is a complex and uncertain system, which contains the unmodeled dynamics and the changes of different working points of the engine also have coupling effects on the system. Therefore, the control strategy of VGT-EGR system needs to be independent of the accurate model and has strong robustness to disturbance. Active disturbance rejection control (ADRC) is a method proposed by researcher Han Jingqing [20], which inherits the advantages of PID and is less dependent on the accurate model. The coupling and external disturbance of the system are regarded as the total disturbance and extended into a new state, which is observed by the extended state observer. Then, the system is transformed into an integral series control system by compensating the disturbance into the control input. In addition, to solve the contradiction between speediness and overshoot, a tracking differentiator is designed. Finally, a more efficient nonlinear state error feedback control law is used to replace the traditional PID control law. Therefore, ADRC plays an increasingly important role in the control of nonlinear system and multi-input multi-output (MIMO) system such as quadrotor [21,22], aero engine [23], and diesel engines [24–26]. In addition, Wang et al. [27] introduced the ADRC method into the control of the differential drive assist steering system of electric vehicles, and selected the standard working conditions for simulation and experimental verification. The results showed that compared with the PID controller, the ADRC controller can not only reduce the steering effort of the driver obviously, but also have better control performance in tracking accuracy and smooth road feeling of the driver. Shi et al. [28] proposed a hybrid ADRC control for the control of superheated steam temperature in coal-fired power plants. Numerical simulation results showed that the hybrid ADRC can improve the performance of tracking and disturbance rejection under the condition of good robustness. At the same time, experiments were carried out on a 150 MW power plant simulator. The experimental results showed that the hybrid ADRC can improve the control performance of superheated steam temperature and its structure is simpler than cascade control. Zhou et al. [29] proposed the ADRC technology as a speed loop controller for permanent magnet synchronous motor. The ADRC is used to measure and compensate unknown disturbances such as rotational inertia and stator resistance, so that the system has strong against the system parameter change and external disturbance, and the decoupling control of permanent magnet synchronous motor is realized. Simulation and experimental results show that the controller has strong robustness, stability and accurate dynamic tracking performance.

However, the ADRC was originally proposed by Han in a nonlinear form, but nonlinear active disturbance rejection control (NLADRC) requires too many parameters to be adjusted. Gao proposed a bandwidth-based parameter tuning method for the first time, which simplified the ADRC from the original nonlinear form to the linear form, which greatly reduced the parameters of the controller and greatly

promoted the application of ADRC. However, linear active disturbance rejection control (LADRC) cannot meet the requirements of the system which needs high control precision and fast response speed. In the decoupled control of diesel engines, Song and Xie et al. [24,25] decoupled the double-input double-output (DIDO) system into two independent single-input single-output (SISO) systems, each of which can be designed as an ideal monolithic object for control. The nonlinear, uncertain and time-varying process dynamics and the external disturbance are uniformly treated as a total disturbance, which is estimated and compensated by extended state observer (ESO) and eliminated in the control input, thus achieving the purpose of controlling the intake manifold pressure and air quality flow rate of diesel engines by LADRC controller. Based on the above work, our goal is to design a control strategy for diesel engines with satisfactory transient response performance, strong resistance to external disturbances and robust performance for engine operating conditions.

The contributions made in the paper can be summarized as follows.

1. Based on the work of Song and Xie [24,25] on the LADRC controller in diesel engines, we designed the NLADRC controller for VGT-EGR system in order to improve the control accuracy.
2. Based on the theoretical result of Guo [30] on ADRC, we give the convergence proof of the designed NLSEO, which theoretically ensures the rationality of the designed controller.
3. In addition, the NLADRC controller designed by us is compared with PID controller [24] and LADRC controller. First, the NLADRC controller we designed can quickly track the given signal without overshoot. Secondly, the control effect of the controller is not affected by the external square wave and sinusoidal control disturbance. Finally, the controller can track the given signal well after changing the diesel engine speed and fuel injection without resetting the parameters of the controller.

The rest of this paper is organized as follows: The next section describes the dynamics of diesel engines. Combined with the dynamics of Section 2, the nonlinear extended state observer is designed for intake manifold pressure and air mass flow in Section 3. Section 4 gives the convergence proof of the designed NLSEO. Section 5 presents the design of NLADRC controller and the comparative simulation results of NLADRC controller, LADRC controller and PID controller. Finally, the conclusion is given in the last section.

2. System Dynamics

Figure 1 shows the diesel engine model, which is mainly composed of seven parts: cylinder, compressor, turbine, intake manifold, exhaust manifold, VGT valve and EGR valve.

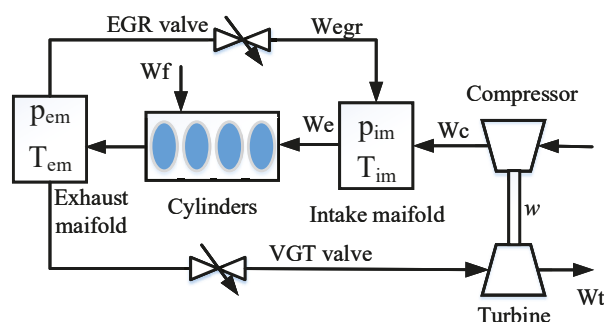


Figure 1. A model structure of the diesel engine.

According to the law of ideal gas and the law of conservation of mass, the dynamics of manifold pressure can be modeled [1] as

$$\dot{p}_{im} = \frac{RT_{im}}{V_{im}}(W_c + W_{egr} - W_e), \quad (1)$$

$$\dot{p}_{em} = \frac{RT_{em}}{V_{em}}(W_e + W_f - W_{egr} - W_t), \quad (2)$$

where p_{im} , T_{im} and V_{im} respectively represent the intake manifold pressure, temperature and volume. In addition, p_{em} , T_{em} and V_{em} respectively represent the exhaust manifold pressure, temperature and volume. W_c , W_e , W_f , W_{egr} and W_t represent the mass air flow, the total mass flow from the intake manifold into the cylinder, the fuel mass flow into the cylinders, the mass flow through the EGR valve, and the turbine mass flow, respectively.

The fuel mass flow W_f into the cylinder can be expressed [1] as

$$W_f = \frac{10^{-6}}{120} u_\delta n_e n_{cyl},$$

where u_δ represents the injected fuel, n_e represents the engine speed and n_{cyl} is the number of cylinders.

The mass flow through the EGR valve W_{egr} and the turbine mass flow W_t are expressed [25] as

$$W_{egr} = A_{egr}(u_{egr}) \frac{\sqrt{2p_{im}(p_{em} - p_{im})}}{\sqrt{RT_{em}}}, \quad (3)$$

$$W_t = A_{vgt}(u_{vgt}) \frac{\sqrt{2p_{amb}(p_{em} - p_{amb})}}{\sqrt{RT_{em}}}, \quad (4)$$

where u_{egr} and u_{vgt} are input signals for diesel engines and represent the open ratio of VGT and EGR valve, respectively. It is 0% when the valves are closed and 100% when the valves are fully open. $A_{egr}(u_{egr})$ is a polynomial function of u_{egr} , so is $A_{vgt}(u_{vgt})$.

The compressor power P_c and turbine power P_t can be calculated [25] as

$$P_c = \frac{W_c c_p T_{amb}}{\eta_c} \left(\left(\frac{p_{im}}{p_{amb}} \right)^\mu - 1 \right), \quad (5)$$

$$P_t = W_t c_p T_{em} \eta_t \left(1 - \left(\frac{p_{amb}}{p_{em}} \right)^\mu \right), \quad (6)$$

where T_{amb} and p_{amb} represent ambient temperature and pressure. η_m , η_c and η_t represent the mechanical efficiency in the turbocharger, the isentropic efficiency of compressor and the isentropic efficiency of turbine, respectively. c_p is the constant pressure specific heat capacity, and τ_t is time constant. The power μ can be expressed as $\mu = 1 - \frac{1}{\gamma}$ by the specific heat capacity ratio γ .

The relationship between compressor power P_c and turbine power P_t can be expressed as

$$\dot{P}_c = \frac{1}{\tau_t} (-P_c + \eta_m P_t). \quad (7)$$

The dynamic process of the exhaust process is very fast, so it is considered a stable process, then $\dot{p}_{em}$ is 0, Equation (2) can be simplified [25] as

$$W_e + W_f - W_{egr} - W_t = 0. \quad (8)$$

Define $g(u_{vgt}) = A_{vgt}(u_{vgt})$ and $f(u_{egr}) = A_{egr}(u_{egr})$, and their Taylor series at VGT valve opening $u_{vgt} = u_0$ and EGR valve opening $u_{egr} = u_1$ are

$$g(u_{vgt}) = g(u_0) + g'(u_0)(u_{vgt} - u_0) + \dots + \frac{g^{(n)}(u_0)}{n!}(u_{vgt} - u_0)^n + R_n(u_{vgt}), \quad (9)$$

$$f(u_{egr}) = f(u_1) + f'(u_1)(u_{egr} - u_1) + \dots + \frac{f^n(u_1)}{n!}(u_{egr} - u_1)^n + R_n(u_{egr}), \quad (10)$$

where $R_n(u_{vgt}) = o((u_{vgt} - u_0)^n)$, $R_n(u_{egr}) = o((u_{egr} - u_1)^n)$, and they represent polynomials of higher order than n .

Substituting Equations (4), (8) and (9) into Equation (1), the dynamics of intake manifold pressure p_{im} becomes

$$\dot{p}_{im} = \frac{RT_{im}}{V_{im}}(W_c + W_f) + K_0K_1 + K_0g'(u_0)u_{vgt}, \quad (11)$$

where

$$K_0 = -\frac{\sqrt{2RT_{im}}}{\sqrt{T_{em}}V_{im}}\sqrt{p_{amb}(p_{em} - p_{amb})},$$

$$K_1 = g(u_0) - g'(u_0)u_0 + \dots + \frac{g^n(u_0)}{n!}(u_{vgt} - u_0)^n + R_n(u_{vgt}).$$

Differentiate Equation (6) and convert the form, the representation of $\dot{W}_c$ can be expressed as

$$\dot{W}_c = \frac{\dot{p}_c - \frac{W_c c_p T_{amb} \mu}{\eta_c p_{amb}} \left(\frac{p_{im}}{p_{amb}}\right)^{\mu-1} \dot{p}_{im}}{\frac{c_p T_{amb}}{\eta_c} \left(\left(\frac{p_{im}}{p_{amb}}\right)^{\mu} - 1\right)}. \quad (12)$$

Substituting Equations (3), (5)–(8) and (10) into Equation (12), we obtain the dynamics of mass air flow W_c as

$$\dot{W}_c = K_2(W_e + W_f) - K_4W_c - K_2K_3K_5 - K_2K_3f'(u_1)u_{egr}, \quad (13)$$

where

$$\Pi_1 = \frac{p_{im}}{p_{amb}}, \Pi_2 = \frac{p_{amb}}{p_{em}}, K_2 = \frac{\eta_m \eta_c \eta_t T_{em} (1 - \Pi_2^\mu)}{T_{amb} \tau_t (\Pi_1^\mu - 1)},$$

$$K_3 = \frac{\sqrt{2p_{im}(p_{em} - p_{im})}}{\sqrt{RT_{em}}}, K_4 = \frac{1}{\tau_t} + \frac{\mu \Pi_1^{\mu-1} \dot{p}_{im}}{p_{amb} (\Pi_1^\mu - 1)},$$

$$K_5 = f(u_1) - f'(u_1)u_1 + \dots + \frac{f^n(u_1)}{n!}(u_{egr} - u_1)^n + R_n(u_{egr}).$$

3. NLESO Design

In this part, we will design the NLESO for VGT-EGR system according to the dynamics in Section 2. The decoupling control scheme is shown in Figure 2. As shown in the figure, the DDO VGT-EGR system with u_{vgt} and u_{egr} as inputs and intake manifold pressure p_{im} and mass air flow W_c as outputs is decomposed into two subsystems. One of the SISO systems takes u_{vgt} as input and intake manifold pressure p_{im} as output, while the other takes u_{egr} as input and mass air flow W_c as output.

Figure 3 presents the block diagram of NLADRC controller designed for p_{im} loop of VGT-EGR system in diesel engines. NLADRC controller mainly consists of three parts: tracking differentiator (TD), nonlinear states error feedback control laws (NLSEF) and NLESO. The desired output signal p_{im-d} is arranged by TD, and then the transient process signal p_{im-v1} is obtained. p_{im-v1} subtracts the state signal z_1 observed by NLESO, and its deviation is used as the input of NLSEF to produce control action u_0 . Then u_0 subtracts the total disturbance z_2 estimated by NLESO and divides it by the compensation coefficient b to obtain the control output u_{nladrc} of NLADRC controller. Considering that the output signal of the controller may be

affected by the external disturbance, the output u_{nladrc} of the controller plus the external disturbance d is the input signal u_{vgt} of the diesel engine.

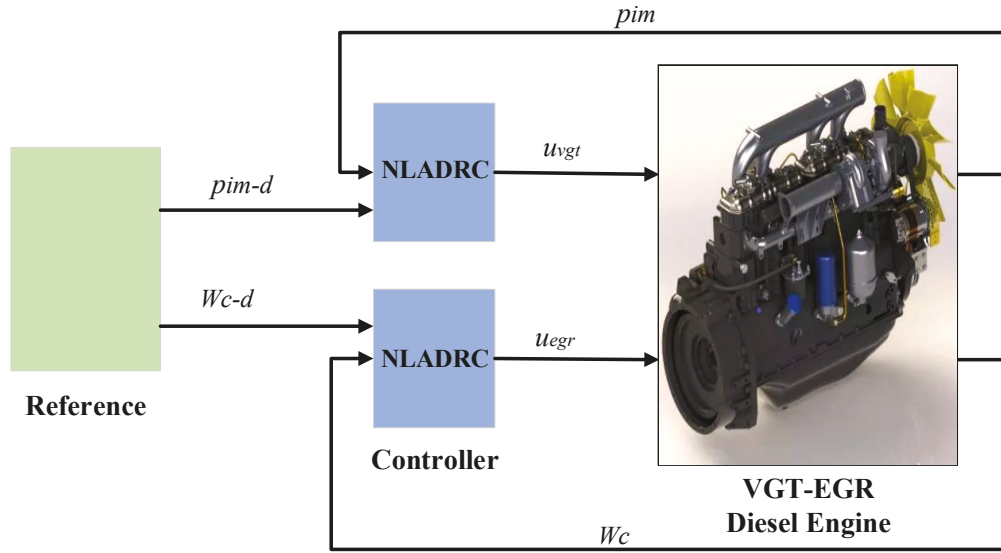


Figure 2. NLADRC controller diagram.

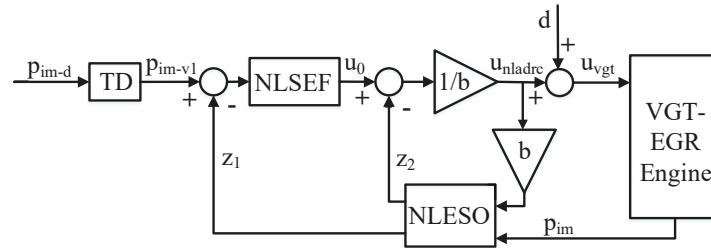


Figure 3. The structure of NLADRC controller for p_{im} loop in VGT-EGR diesel engines.

Considering that the system may be affected by external interference, we add external interference w to the system (11). In addition, the system (11) is converted to the total disturbance and input form as follows

$$\begin{aligned}\dot{p}_{im} &= \frac{RT_{im}}{V_{im}} (W_c + W_f) + K_0 K_1 + K_0 g'(u_0) u_{vgt} + w \\ &= \frac{RT_{im}}{V_{im}} (W_c + W_f) + K_0 K_1 + (K_0 g'(u_0) - b_{vgt}) u_{vgt} + b_{vgt} u_{vgt} + w \\ &= f_{p_{im}} + w + b_{vgt} u_{vgt} = F_{p_{im}} + b_{vgt} u_{vgt},\end{aligned}\quad (14)$$

where $F_{p_{im}}$ represents the total disturbance of p_{im} loop. The system function $f_{p_{im}}$ contains coupling and disturbance caused by the change of working point. The coefficient b_{vgt} is an approximate estimate of $K_0 g'(u_0)$, and the estimated deviation $K_0 g'(u_0) - b_{vgt}$ between b_{vgt} and $K_0 g'(u_0)$ is also included in $f_{p_{im}}$, which is timely estimated and compensated. For the diesel engine physical system, we can notice that there are some positive constants c_0, c_1, c_2 and the positive integer k that satisfy $|b_{vgt} u_{vgt}| + |f_{p_{im}}| + |w| + \left| \frac{\partial f_{p_{im}}}{\partial t} \right| + \left| \frac{\partial f_{p_{im}}}{\partial p_{im}} \right| \leq c_0 + c_1 |p_{im}|^k$ and $|w| + |p_{im}| \leq c_2$.

Let the state variable $x_1(t) = p_{im}(t)$, $x_2(t) = F_{p_{im}}(t) = f_{p_{im}}(t, x_1(t)) + w(t)$. Then, system (14) is expressed in the form of extended state space

$$\begin{bmatrix} \dot{x}_1(t) \\ \dot{x}_2(t) \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ 0 & 0 \end{bmatrix} \begin{bmatrix} x_1(t) \\ x_2(t) \end{bmatrix} + \begin{bmatrix} b_{vgt} \\ 0 \end{bmatrix} u_{vgt}(t) + \begin{bmatrix} 0 \\ 1 \end{bmatrix} \dot{F}_{p_{im}}(t). \quad (15)$$

According to system (15), the NLESO for the intake manifold pressure p_{im} is designed as

$$\begin{cases} \dot{z}_1(t) = z_2(t) + b_{vgt} u_{vgt}(t) + g_1\left(\frac{x_1(t) - z_1(t)}{\varepsilon}\right), \\ \dot{z}_2(t) = \frac{1}{\varepsilon} g_2\left(\frac{x_1(t) - z_1(t)}{\varepsilon}\right), \end{cases} \quad (16)$$

where $z_1(t)$ and $z_2(t)$ are estimates of the intake manifold pressure and the total disturbance, respectively. ε is constant gain. $g_1\left(\frac{x_1(t) - z_1(t)}{\varepsilon}\right)$ and $g_2\left(\frac{x_1(t) - z_1(t)}{\varepsilon}\right)$ are functions of the state deviation expressed as $g_1\left(\frac{x_1(t) - z_1(t)}{\varepsilon}\right) = a_1\left(\frac{x_1(t) - z_1(t)}{\varepsilon}\right) + fal\left(\frac{x_1(t) - z_1(t)}{\varepsilon}, 0.5, 1\right)$, and $g_2\left(\frac{x_1(t) - z_1(t)}{\varepsilon}\right) = a_2\left(\frac{x_1(t) - z_1(t)}{\varepsilon}\right)$. Parameters a_1 and a_2 are positive, and nonlinear function $fal(e, \alpha, \delta)$ is specifically expressed as

$$fal(e, \alpha, \delta) = \begin{cases} |e|^\alpha \text{sign}(e), & |e| > \delta, \\ \frac{e}{\delta^{1-\alpha}}, & |e| \leq \delta, \end{cases}$$

where α and δ are the parameters to be adjusted, and δ represents the width of the linear region.

Similar to the simplification process of Equation (11), Equation (13) can be simplified in the form of input u_{egr} and disturbance similar to (14)

$$\dot{W}_c = f_{W_c} + w + b_{egr} u_{egr}, \quad (17)$$

according to the similar design method of p_{im} loop, NLESO can be designed for mass air flow W_c .

4. Convergence Analysis of NLESO for the Intake Manifold Pressure

In the above section, we designed NLESO using the intake manifold pressure p_{im} loop as an example. In this section, we will provide the convergence analysis of the proposed NLESO.

Theorem 1. *The proposed NLESO given in (16) for the intake manifold pressure loop dynamics (15) have the following two properties:*

1. For every positive constant a , $\lim_{\varepsilon \rightarrow 0} |x_i(t) - z_i(t)| = 0$ uniformly in $t \in [a, \infty)$.
2. $\overline{\lim}_{t \rightarrow \infty} |x_i(t) - z_i(t)| \leq O(\varepsilon^{3-i}), i = 1, 2$,

where $\overline{\lim}_{t \rightarrow \infty} |x_i(t) - z_i(t)| = \limsup_{t \rightarrow \infty} |x_i(t) - z_i(t)|$, it means that the upper limit is the upper bound of the limit of the series of convergents. The state variables $x_1(t)$ and $x_2(t)$ respectively represent the intake manifold pressure p_{im} and the total disturbance $F_{p_{im}}$. In addition, the states of observer $z_1(t)$ and $z_2(t)$ are approximations of $x_1(t)$ and $x_2(t)$, respectively.

Proof of Theorem 1. Set

$$e_i(t) = x_i(t) - z_i(t), \eta_i(t) = \frac{e_i(\varepsilon t)}{\varepsilon^{2-i}}, i = 1, 2. \quad (18)$$

Then the derivative of $\eta_1(t)$ can be calculated as

$$\begin{aligned}\dot{\eta}_1(t) &= \dot{\epsilon}_1(\epsilon t) = \dot{x}_1(\epsilon t) - \dot{z}_1(\epsilon t) \\ &= x_2(\epsilon t) + b_{vgt} u_{vgt}(\epsilon t) \\ &\quad - z_2(\epsilon t) - b_{vgt} u_{vgt}(\epsilon t) - g_1 \left(\frac{x_1(\epsilon t) - z_1(\epsilon t)}{\epsilon} \right) \\ &= x_2(\epsilon t) - z_2(\epsilon t) - g_1(\eta_1(t)) \\ &= \eta_2(t) - g_1(\eta_1(t)).\end{aligned}$$

Similarly, the derivative of $\eta_2(t)$ can be expressed as

$$\begin{aligned}\dot{\eta}_2(t) &= \epsilon \dot{\epsilon}_2(\epsilon t) = \epsilon (\dot{x}_2(\epsilon t) - \dot{z}_2(\epsilon t)) \\ &= \epsilon \left(\dot{F}_{p_{im}}(\epsilon t) - \frac{1}{\epsilon} g_2 \left(\frac{x_1(\epsilon t) - z_1(\epsilon t)}{\epsilon} \right) \right) \\ &= -g_2(\eta_1(t)) + \epsilon \dot{F}_{p_{im}}(\epsilon t).\end{aligned}$$

We construct positive definite function as

$$V(\eta(t)) = \eta(t)^T P \eta(t) + \int_0^{\eta_1(t)} f_{al}(s, 0.5, 1) ds, \quad (19)$$

where

$$P = \begin{bmatrix} \frac{a_2^2 b_2 + a_2 b_1}{2a_1 a_2} & -\frac{b_2}{2} \\ -\frac{b_2}{2} & \frac{a_1^2 b_2 + a_2 b_2 + b_1}{2a_1 a_2} \end{bmatrix}, \begin{cases} b_2 > 1 \\ 2b_1 - b_2 > 1 \\ b_1 \neq b_2 \end{cases}, \eta(t) = \begin{bmatrix} \eta_1(t) \\ \eta_2(t) \end{bmatrix}.$$

The derivative of $V(\eta(t))$ with respect to t along the $\eta(t)$ can be calculated as

$$\begin{aligned}\frac{d}{dt} V(\eta(t)) &= \frac{\partial V}{\partial \eta_1(t)} \dot{\eta}_1(t) + \frac{\partial V}{\partial \eta_2(t)} \dot{\eta}_2(t) \\ &= \frac{\partial V}{\partial \eta_1(t)} (\eta_2(t) - g_1(\eta_1(t))) - \frac{\partial V}{\partial \eta_2(t)} g_2(\eta_1(t)) + \frac{\partial V}{\partial \eta_2(t)} \epsilon \dot{F}_{p_{im}}(\epsilon t).\end{aligned} \quad (20)$$

Next, we will calculate $\dot{F}_{p_{im}}(\epsilon t)$, $\frac{\partial V}{\partial \eta_1(t)} (\eta_2(t) - g_1(\eta_1(t))) - \frac{\partial V}{\partial \eta_2(t)} g_2(\eta_1(t))$, and $\frac{\partial V}{\partial \eta_2(t)}$ respectively. First, we have

$$\begin{aligned}\dot{F}_{p_{im}}(\epsilon t) &= \left. \frac{d}{ds} f(s, x_1(s)) \right|_{s=\epsilon t} + \dot{w}(\epsilon t) \\ &= \epsilon \frac{\partial}{\partial t} f(\epsilon t, x_1(\epsilon t)) + \frac{\partial}{\partial x_1} f(\epsilon t, x_1(\epsilon t)) (x_2(\epsilon t) + b_{vgt} u_{vgt}(t)) + \dot{w}(\epsilon t),\end{aligned} \quad (21)$$

there exists a constant $M > 0$ that satisfies $M \geq |\dot{F}_{p_{im}}(\epsilon t)|$.

Notice that $g_1(\eta_1(t)) = a_1\eta_1(t) + fal(\eta_1(t), 0.5, 1)$ and $g_2(\eta_1(t)) = a_2\eta_1(t)$. Then, we have the following calculation

$$\begin{aligned}
 & \frac{\partial V}{\partial \eta_1(t)} (\eta_2(t) - g_1(\eta_1(t))) - \frac{\partial V}{\partial \eta_2(t)} g_2(\eta_1(t)) \\
 &= \left(\frac{a_2^2 b_2 + a_2 b_1}{a_1 a_2} \eta_1(t) - b_2 \eta_2(t) + fal(\eta_1(t), 0.5, 1) \right) (\eta_2(t) - a_1 \eta_1(t) - fal(\eta_1(t), 0.5, 1)) \\
 &\quad - \left(\frac{a_1^2 b_2 + a_2 b_2 + b_1}{2 a_1 a_2} \eta_2(t) - b_2 \eta_1(t) \right) a_2 \eta_1(t) \\
 &= -b_1 \eta_1(t)^2 - b_2 \eta_2(t)^2 + (\eta_2(t) - a_1 \eta_1(t)) fal(\eta_1(t), 0.5, 1) \\
 &\quad - \left(\frac{a_2^2 b_2 + a_2 b_1}{a_1 a_2} \eta_1(t) - b_2 \eta_2(t) + fal(\eta_1(t), 0.5, 1) \right) fal(\eta_1(t), 0.5, 1) \\
 &= -b_1 \eta_1(t)^2 - b_2 \eta_2(t)^2 + (1 + b_2) \eta_2(t) fal(\eta_1(t), 0.5, 1) \\
 &\quad - fal(\eta_1(t), 0.5, 1)^2 - \left(a_1 + \frac{a_2^2 b_2 + a_2 b_1}{a_1 a_2} \right) \eta_1(t) fal(\eta_1(t), 0.5, 1) \\
 &\leq -b_1 \eta_1(t)^2 - b_2 \eta_2(t)^2 + (1 + b_2) \eta_2(t) fal(\eta_1(t), 0.5, 1) \\
 &\leq -b_1 \eta_1(t)^2 - b_2 \eta_2(t)^2 + \frac{1 + b_2}{2} (\eta_2(t)^2 + fal(\eta_1(t), 0.5, 1)^2).
 \end{aligned} \tag{22}$$

Due to $|fal(\eta_1(t), 0.5, 1)| \leq |\eta_1(t)|$, we can further calculate as

$$\begin{aligned}
 & \frac{\partial V}{\partial \eta_1(t)} (\eta_2(t) - g_1(\eta_1(t))) - \frac{\partial V}{\partial \eta_2(t)} g_2(\eta_1(t)) \\
 &\leq -b_1 \eta_1(t)^2 - b_2 \eta_2(t)^2 + \frac{1 + b_2}{2} (\eta_2(t)^2 + fal(\eta_1(t), 0.5, 1)^2) \\
 &\leq -b_1 \eta_1(t)^2 - b_2 \eta_2(t)^2 + \frac{1 + b_2}{2} (\eta_2(t)^2 + \eta_1(t)^2) \\
 &= - \left(\frac{2b_1 - b_2 - 1}{2} \eta_1(t)^2 + \frac{b_2 - 1}{2} \eta_2(t)^2 \right) = \eta(t)^T P_1 \eta(t) \triangleq -W(\eta(t)),
 \end{aligned} \tag{23}$$

where

$$P_1 = \begin{bmatrix} \frac{2b_1 - b_2 - 1}{2} & 0 \\ 0 & \frac{b_2 - 1}{2} \end{bmatrix}.$$

Obviously, $V(\eta(t))$ satisfies

$$\begin{aligned}
 V(\eta(t)) &= \eta(t)^T P \eta(t) + \int_0^{\eta_1(t)} fal(s, 0.5, 1) ds \\
 &\geq \eta(t)^T P \eta(t) \triangleq V_1(\eta(t)).
 \end{aligned}$$

We notice that P is a symmetric positive definite matrix, which means that P has two positive eigen values denoted by $\lambda_{1\min}$ and $\lambda_{2\max}$. In addition, set $\beta_1 = \lambda_{1\min}$, we have

$$V(\eta(t)) \geq V_1(\eta(t)) \geq \beta_1 \|\eta(t)\|^2. \tag{24}$$

Let us calculate $V(\eta(t))$ as

$$\begin{aligned} V(\eta(t)) &= \eta(t)^T P \eta(t) + \int_0^{\eta_1(t)} f(s, 0.5, 1) ds \\ &\leq \eta(t)^T P \eta(t) + \int_0^{\eta_1(t)} s ds \\ &= \eta(t)^T P \eta(t) + \frac{\eta_1(t)^2}{2} \\ &= \eta(t)^T P_2 \eta(t) \triangleq V_2(\eta(t)), \end{aligned}$$

where

$$P_2 = \begin{bmatrix} \frac{a_2^2 b_2 + a_2 b_1}{2a_1 a_2} + \frac{1}{2} & -\frac{b_2}{2} \\ -\frac{b_2}{2} & \frac{a_1^2 b_2 + a_2 b_2 + b_1}{2a_1 a_2} \end{bmatrix}.$$

Similarly, we can get the eigenvalues of symmetric positive definite matrix P_2 are $\lambda_{3\min}$ and $\lambda_{4\max}$, respectively. In addition, set $\beta_2 = \lambda_{4\max}$, we have

$$V(\eta(t)) \leq V_2(\eta(t)) \leq \beta_2 \|\eta(t)\|^2. \quad (25)$$

And for the positive definite function $W(\eta(t))$, we also have the expression as

$$\beta_3 \|\eta(t)\|^2 \leq W(\eta(t)) \leq \beta_4 \|\eta(t)\|^2, \quad (26)$$

where β_3 is the minimum eigenvalue of symmetric positive definite matrix P_1 , and β_4 is the maximum eigenvalue of symmetric positive definite matrix P_1 .

We have an expression of $\frac{\partial V}{\partial \eta_2(t)}$ as

$$\frac{\partial V}{\partial \eta_2(t)} = \frac{a_1^2 b_2 + a_2 b_2 + b_1}{2a_1 a_2} \eta_2(t) - b_2 \eta_1(t) = \frac{\partial V_1}{\partial \eta_2(t)},$$

so we can get

$$\left| \frac{\partial V}{\partial \eta_2(t)} \right| = \left| \frac{\partial V_1}{\partial \eta_2(t)} \right| \leq \left\| \frac{\partial V_1}{\partial \eta(t)} \right\| = \|2P\eta(t)\| \leq 2\|P\| \|\eta(t)\| = 2\sqrt{\lambda_{\max}(P^T P)} \|\eta(t)\|.$$

Because P is a symmetric positive definite matrix, we have

$$2\sqrt{\lambda_{\max}(P^T P)} \|\eta(t)\| = 2\lambda_{\max}(P) \|\eta(t)\|,$$

and set $\beta = 2\lambda_{\max}(P) = 2\lambda_{2\max}$, we can know that $\left| \frac{\partial V}{\partial \eta_2(t)} \right|$ satisfies

$$\left| \frac{\partial V}{\partial \eta_2(t)} \right| \leq \beta \|\eta(t)\|. \quad (27)$$

So, Equation (20) combined with Equations (21) and (23)–(27), we have

$$\begin{aligned}\frac{d}{dt}V(\eta(t)) &= \frac{\partial V}{\partial \eta_1(t)}(\eta_2(t) - g_1(\eta_1(t))) - \frac{\partial V}{\partial \eta_2(t)}g_2(\eta_1(t)) + \frac{\partial V}{\partial \eta_2(t)}\varepsilon \dot{p}_{im}(\varepsilon t) \\ &\leq -W(\eta(t)) + \varepsilon M\beta \|\eta(t)\| \\ &\leq -\frac{\beta_3}{\beta_2}V(\eta(t)) + \frac{\sqrt{\beta_1}}{\beta_1}\varepsilon M\beta \sqrt{V(\eta(t))}.\end{aligned}\quad (28)$$

Because

$$\frac{d}{dt}\sqrt{V(\eta(t))} = \frac{d\sqrt{V(\eta(t))}}{dV(\eta(t))} \frac{d}{dt}V(\eta(t)) = \frac{1}{2\sqrt{V(\eta(t))}} \frac{d}{dt}V(\eta(t)),$$

from Equation (28) we can get

$$\frac{d}{dt}\sqrt{V(\eta(t))} \leq -\frac{\beta_3}{2\beta_2}\sqrt{V(\eta(t))} + \frac{\sqrt{\beta_1}\varepsilon M\beta}{2\beta_1}.\quad (29)$$

Combining Equation (24) with Equation (29), we know that $\|\eta(t)\|$ satisfies

$$\|\eta(t)\| \leq \sqrt{\frac{V(\eta(t))}{\beta_1}} \leq \frac{\sqrt{\beta_1 V(\eta(0))}}{\beta_1} e^{-\frac{\beta_3}{2\beta_2}t} + \frac{\varepsilon M\beta}{2\beta_1} \int_0^t e^{-\frac{\beta_3}{2\beta_2}(t-\tau)} d\tau,\quad (30)$$

which together with Equation (18) satisfies

$$\begin{aligned}|e_i(t)| &= \varepsilon^{2-i} \left| \eta_i\left(\frac{t}{\varepsilon}\right) \right| \leq \varepsilon^{2-i} \left\| \eta\left(\frac{t}{\varepsilon}\right) \right\| \\ &\leq \varepsilon^{2-i} \left[\frac{\sqrt{\beta_1 V(\eta(0))}}{\beta_1} e^{-\frac{\beta_3 t}{2\beta_2 \varepsilon}} + \frac{\varepsilon M\beta}{2\beta_1} \int_0^{\frac{t}{\varepsilon}} e^{-\frac{\beta_3}{2\beta_2}(t/\varepsilon-\tau)} d\tau \right] \\ &\rightarrow 0, i = 1, 2\end{aligned}\quad (31)$$

uniformly in $t \in [a, \infty)$ as $\varepsilon \rightarrow 0$. $\square$

5. NLADRC Controller Design and Simulation Results

In this part, we will design the discrete NLADRC controller for VGT-EGR system and give the simulation results. Due to the discrete controller adopted in the subsequent simulation, we discretize (16), and then discrete NLESO for the intake manifold pressure p_{im} is expressed as

$$\begin{cases} z_1(k+1) = z_1(k) + h \left(z_2(k) + b_{vgt} u_{vgt}(k) + g_1 \left(\frac{x_1(k) - z_1(k)}{\varepsilon} \right) \right), \\ z_2(k+1) = z_2(k) + h \left(\frac{1}{\varepsilon} g_2 \left(\frac{x_1(k) - z_1(k)}{\varepsilon} \right) \right), \end{cases}\quad (32)$$

where h is the sampling time.

The discrete form of TD is expressed [20] as

$$\begin{cases} v_1(k+1) = v_1(k) + h v_2(k), \\ v_2(k+1) = v_2(k) + h f_{han}(v_1(k) - v(k), v_2(k), r, h), \end{cases}\quad (33)$$

where r is the parameter indicating the speed of the transition process, $v(k)$ is the desired intake manifold pressure, $v_1(k)$ is the transition process of the desired intake manifold pressure, $v_2(k)$ is the derivative of the transition process, and $fhan(x_1, x_2, r, h)$ is the synthesis function of the fastest control expressed [20] as

$$\begin{cases} d = rh, \\ d_0 = hd, \\ y = x_1 + hx_2, \\ a_0 = \sqrt{d^2 + 8r|y|}, \\ a = \begin{cases} x_2 + \frac{(a_0-d)}{2}\text{sign}(y), & |y| > d_0, \\ x_2 + \frac{y}{h}, & |y| \leq d_0, \end{cases} \\ fhan = \begin{cases} -r\text{sign}(a), & |a| > d, \\ -r\frac{a}{d}, & |a| \leq d. \end{cases} \end{cases}$$

The NLSEF is designed as

$$\begin{cases} e(k) = v_1(k) - z_1(k), \\ u(k) = k_p fal(e(k), 3/4, 0.01). \end{cases} \quad (34)$$

where k_p is the proportional gain.

The control effect after compensation by disturbance estimate $z_2(k)$ is as follows

$$u_{vgt} = \frac{(u(k) - z_2(k))}{b_{vgt}}. \quad (35)$$

Similar to the controller design method of p_{im} loop, we can also design NLADRC controller for W_c . Therefore, we do not need to introduce the controller design process of W_c loop in detail.

Next, we will verify the control effect of the NLADRC controller for VGT-EGR system from the transient response performance, disturbance resistance performance and robustness on the diesel engine model created by Wahlström [1], and compare it with the LADRC controller and the PID controller. And the parameters adjustment of PID controller are described by Ziegler-Nichols method in [24]. In the simulation experiment, the diesel engine working point: engine speed n_e : 1900 r/min, fuel injection u_δ step from 110 mg/cycle to 160 mg/cycle. The parameters of LADRC controller and NLADRC controller are summarized in Tables 1–3.

Table 1. NLADRC controller parameters.

Index	h	b	a_1	a_2	ϵ	k_p
VGT loop	0.01	−454	3	1	0.3	30
EGR loop	0.01	−0.015	3	1	0.4	20

Table 2. LADRC controller parameters.

Index	h	b	β_1	β_2	k_p
VGT loop	0.01	−100000	130	4225	1.3
EGR loop	0.01	−0.015	30	225	2.2

Table 3. Nonlinear function $fal(e, \alpha, \delta)$ parameters.

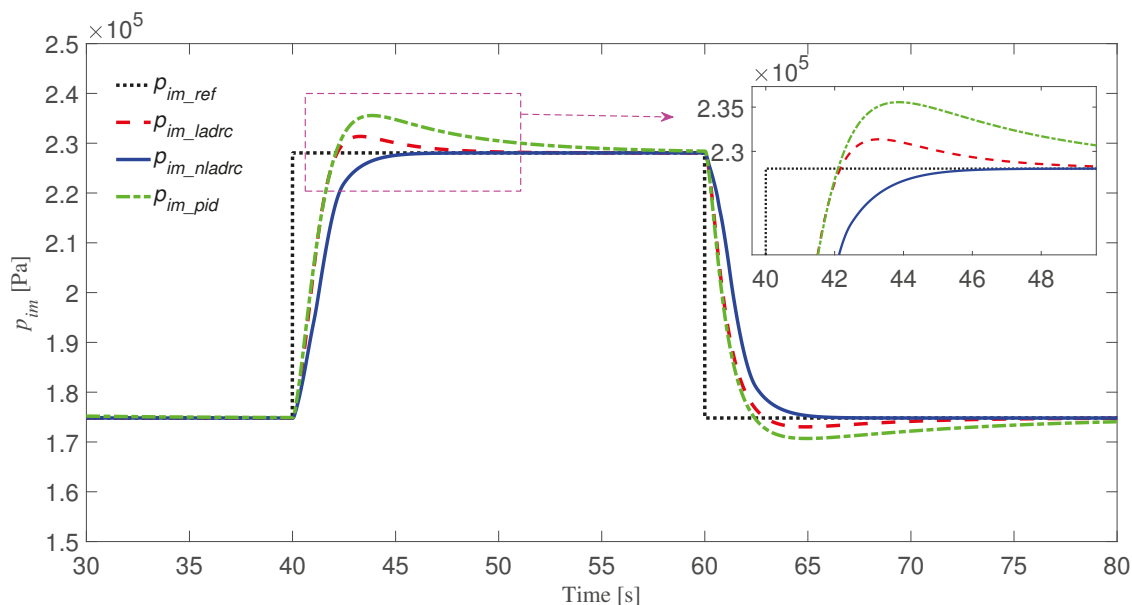
Index	α_{vgt}	δ_{vgt}	α_{egr}	δ_{egr}
NLESO	0.5	1	0.5	1
NLSEF	0.75	0.01	0.75	0.01

5.1. Transient Performance

Figure 4 and 5 present the tracking response curves of intake manifold pressure p_{im} and mass air flow W_c to square wave signal. In the figure, the black curve represents the expected intake manifold pressure, the red curve represents the response curve with NLADRC controller, the blue curve represents the response curve with LADRC controller, and the green curve represents the response curve with PID controller, and other figures in this paper are similar. Table 4 summarizes the setting time and overshoot of the response curves of NLADRC controller, LADRC controller and PID controller.

Table 4. Setting time and overshoot of three controllers.

Index	NLADRC	LADRC	PID
T_s of p_{im} loop	6 s	12 s	20 s
T_s of W_c loop	6 s	15 s	20 s
σ of p_{im} loop	0%	1.5%	3.3%
σ of W_c loop	0%	1.5%	1.1%

**Figure 4.** Response curve of intake manifold pressure p_{im} .

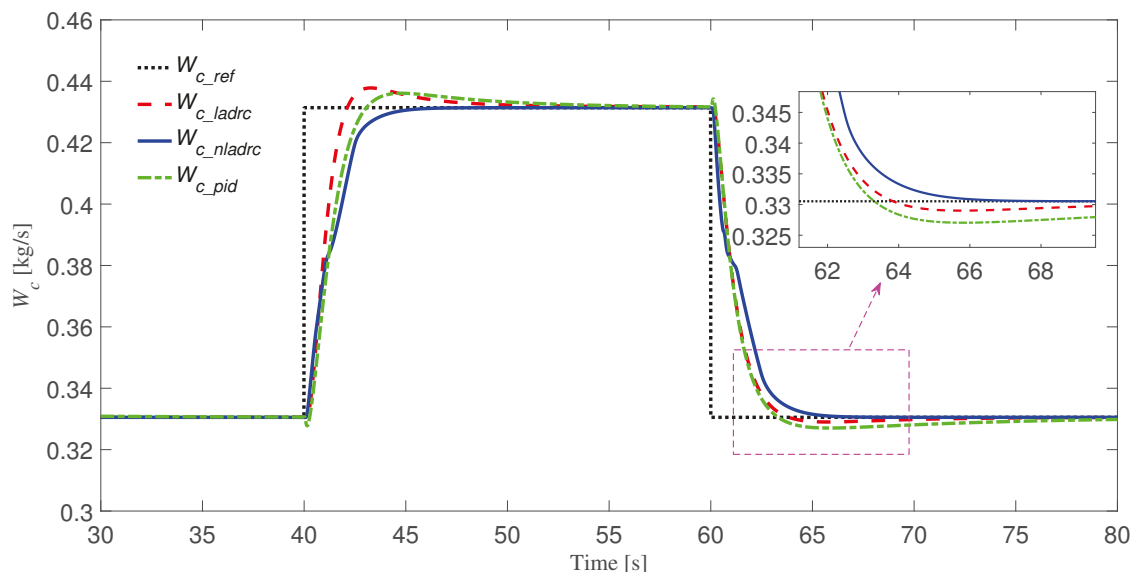


Figure 5. Response curve of mass air flow W_c .

Compare the tracking effect of intake manifold pressure p_{im} in Figure 4, or the tracking effect of mass air flow W_c in Figure 5. Obviously, system with NLADRC controller tracks square wave signals better. In the figure, the NLADRC controller tracks the desired intake manifold pressure p_{im_ref} or mass air flow W_{c_ref} without overshoot, while the LADRC controller and PID controller have relatively large overshoot. The intake manifold pressure p_{im} and mass air flow W_c of diesel engines are the outputs of dynamic system, which have certain inertia and cannot change suddenly. However, the set values p_{im_ref} and W_{c_ref} are given outside the system and can be changed instantaneously. If we use their deviation to produce control effect directly, it will lead to the initial error is too large, and the controller will have a great control effect, which will have a great impact on the system and easy to produce overshoot. TD was mentioned earlier when we introduced the structure of the NLADRC controller designed for the p_{im} loop. In the p_{im} loop, TD converts transient intake manifold pressure p_{im_d} into a slow-varying transition process signal p_{im_v1} . In this way, the slowly changing transition process signal p_{im_v1} minus the output signal p_{im} of diesel engines will not produce excessive deviation, and NLADRC controller will not produce excessive control effect, so that the intake manifold pressure p_{im} of the diesel engine does not generate overshoot. In the previous NLESO design, we introduced the nonlinear function $fal(e, \alpha, \delta)$, whose image is shown in Figure 6. When the parameter α of the nonlinear function is equal to 1, it becomes a linear form of the error. As can be seen from the figure, the linear form and the nonlinear form intersect at (1, 1). When the error is greater than 1, the value of the nonlinear function is less than that of the linear form, so the control effect generated by the NLADRC controller is less than that of the linear form, so the system response with the NLADRC controller will not produce overshoot.

In addition, the setting time of NLADRC controller is about 6 s, while that of LADRC controller and PID controller is 2 times and 3 times of that of NLADRC controller, respectively.

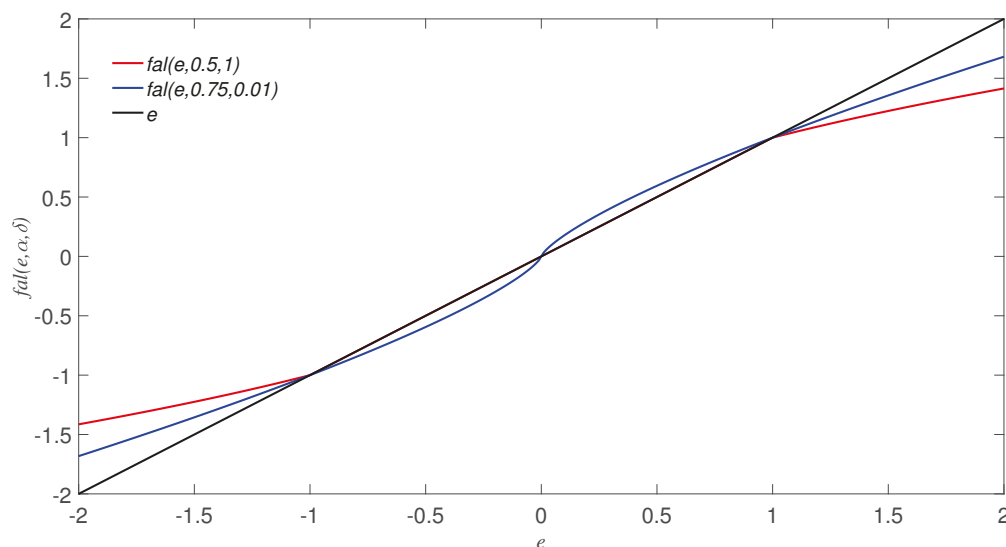


Figure 6. The nonlinear function $fal(e, \alpha, \delta)$ and the linear function e .

5.2. Disturbance Rejection

In the previous section, we compared the effect of three controllers on tracking square wave signals. In this part, we will verify the resistance of the three controllers to control disturbances. The control disturbance is d in the NLADRC controller structure described in Section 3. In this part, we take the p_{im} loop as an example to compare the resistance of the three controllers to external control disturbances. At 40 s, we set the desired intake manifold pressure, and over time the diesel engine's intake manifold pressure reaches the desired value. In addition, then at 60 s, we add a square wave control interference with an amplitude of 10 and a width of 10 s. Figure 7 shows the resistance of the three controllers to square wave disturbances. It is obvious that the response curve of the NLADRC controller has a slight amplitude change after adding the disturbance, and quickly returns to the expected value after 2 s. However, the amplitude variation of the LADRC controller and PID controller is much larger than that of the NLADRC controller, and the recovery time is relatively long. Next, we analyze why NLADRC controller is better than LADRC controller and PID controller in resisting square wave disturbance. First, let us look at the first figure in Figure 8, which shows the output signals u_c of the three controllers. The NLESO in NLADRC controller accurately estimates the amplitude and period of the square wave form disturbance and compensates to the control effect u_c . We can also see from the image of u_c that at 60 s the compensated disturbance in the NLADRC controller is closer to the added control disturbance. The output signal u_c of the controller plus the external control disturbance d is the input signal u_{vgt} of the diesel engine, which is shown in the second figure of Figure 8. As can be seen from the figure, the u_{vgt_nladrc} is basically unchanged after disturbance is added. This is because the compensated disturbance in u_c and the external disturbance d cancel each other. Therefore, compared with LADRC controller and PID controller, NLADRC controller can resist external interference better.

In addition to square wave form disturbance, sinusoidal form control disturbances is also considered to be added. Sinusoidal interference with an amplitude of 5 and a period of 20 s was added at 60 s, and Figures 9 and 10 present the resistance effect of the three controllers against sinusoidal disturbance. For sinusoidal control interference, we have the same conclusion as the square wave form, which will not be repeated here.

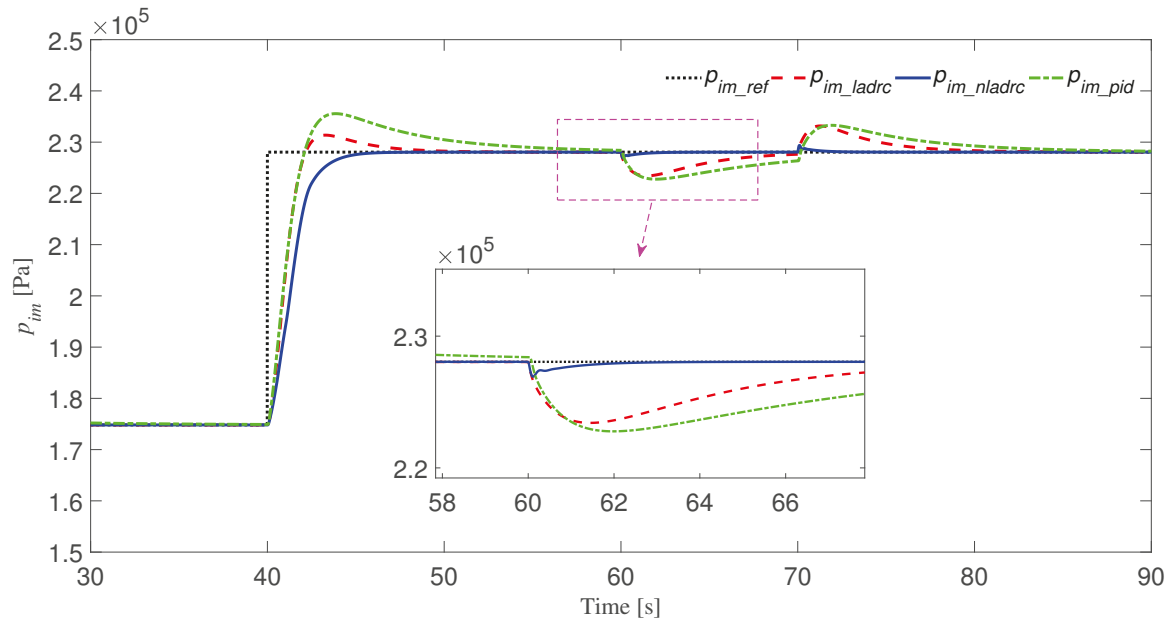


Figure 7. Response curve of intake manifold pressure p_{im} after adding square wave form control disturbance.

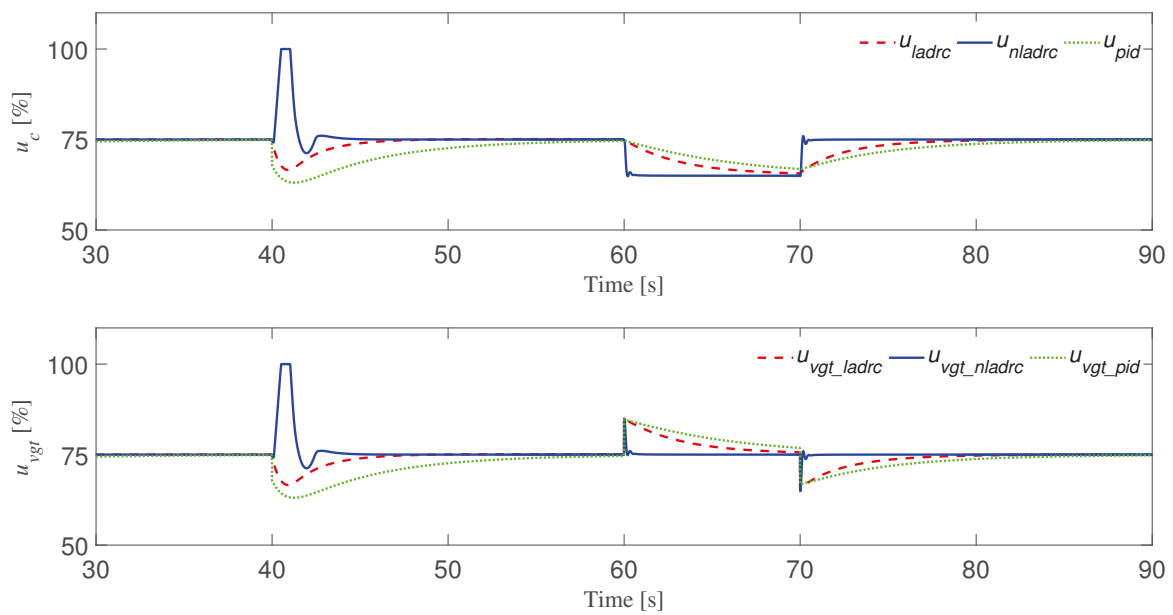


Figure 8. Output signal u_c of controller and input signal u_{vgt} of VGT valve.

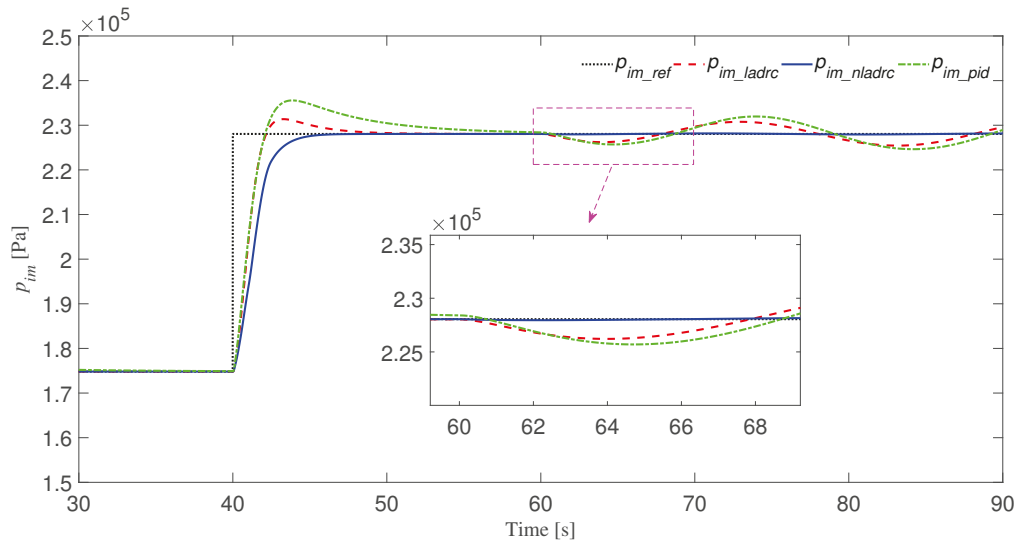


Figure 9. Response curve of intake manifold pressure p_{im} after adding sinusoidal control disturbance.

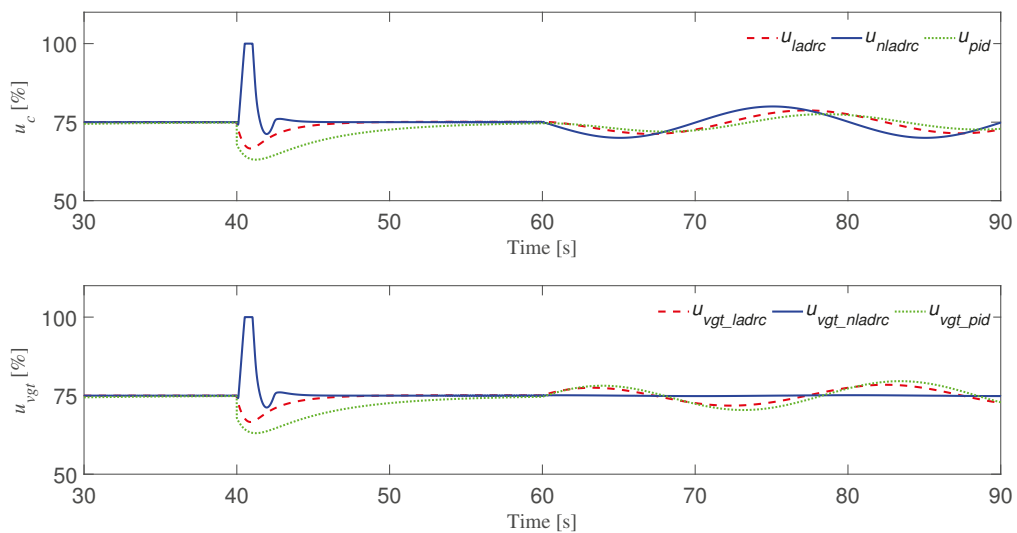


Figure 10. Output signal u_c of controller and input signal u_{vgt} of VGT valve.

5.3. Robustness

Because the diesel engine needs to change the working condition frequently, the controller designed must be able to have strong robust performance to the changing working condition. In this part, we mainly verify whether the NLADRC controller has strong robust performance to the change of working conditions from the two aspects of speed n_e and fuel injection u_δ . First, we verify that the controller is robust to changes in speed. Before 80 s, the speed n_e of diesel engines is 1900 r/min. The engine speed is reduced to 1700 r/min and 1500 r/min at the time of 80 s and 140 s, respectively. Figures 11 and 12 respectively show the response curves of intake manifold pressure p_{im} and mass air flow W_c before and after changing the speed. In addition, the fuel injection u_δ of the engine is increased from 50 mg/cycle to 200 mg/cycle with steps of 50 mg/cycle, and the simulation images are obtained without adjusting the control parameters, as shown in Figures 13 and 14. In the p_{im} loop and the W_c loop, the NLADRC controller can track the given signal faster and better without recalibrating parameters after changing the

engine speed and fuel injection. Therefore, the NLADRC controller has strong robustness to the change of diesel engine operating conditions.

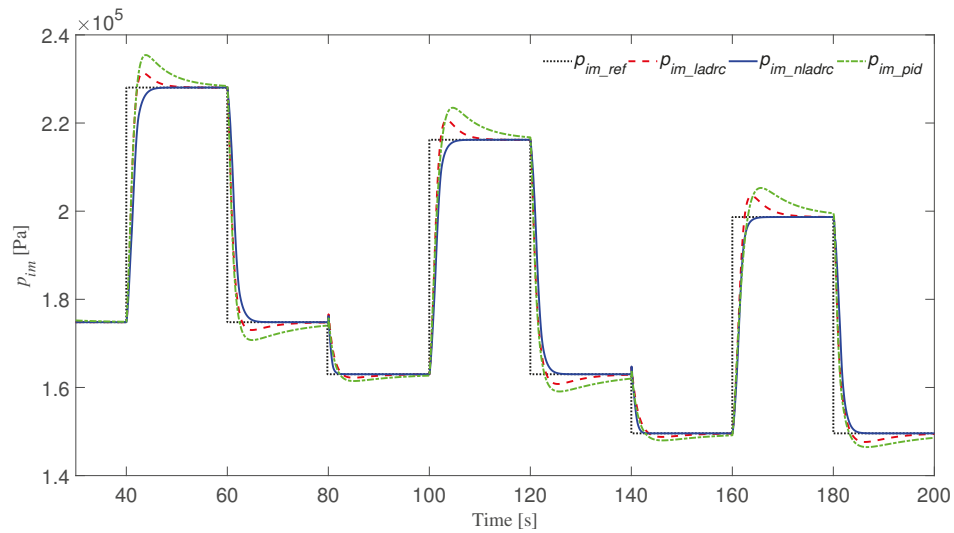


Figure 11. The response curve of intake manifold pressure at different engine speeds.

Although NLADRC controller has good tracking performance, resistance to external interference and robust performance to the change of working conditions for VGT-EGR system in diesel engines, too many parameters of NLADRC controller are too troublesome to adjust, which may limit the wide application of NLADRC controller.

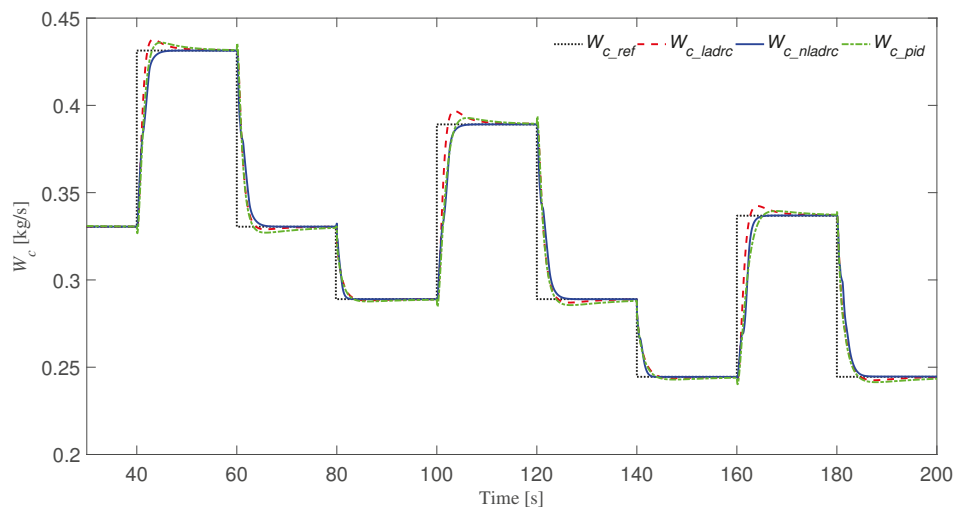


Figure 12. The response curve of mass air flow at different engine speeds.

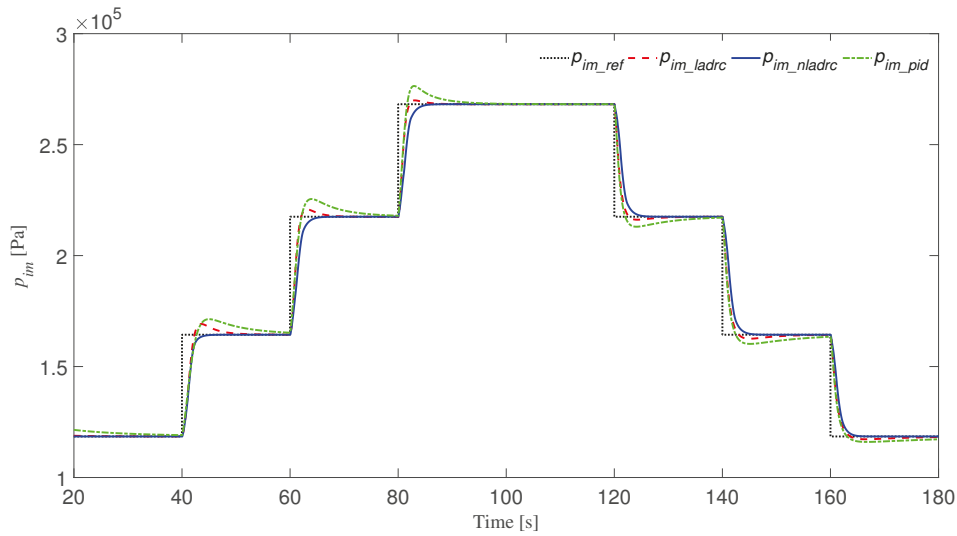


Figure 13. The response curve of intake manifold pressure under different fuel injection conditions.

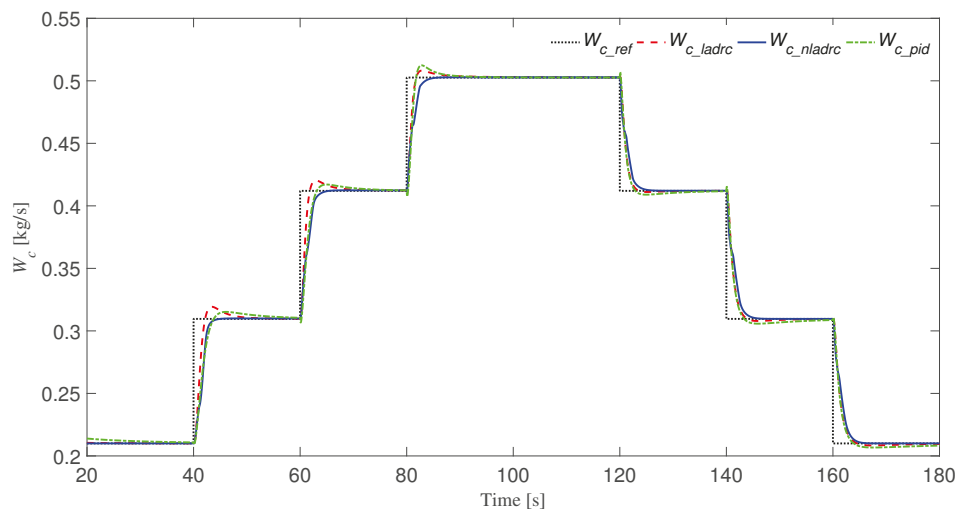


Figure 14. The response curve of mass air flow under different fuel injection conditions.

6. Conclusions

In this paper, considering that the VGT-EGR system of diesel engines is a complex nonlinear system with unknown dynamics, the operating conditions of the system often change in actual operation. Therefore, the control strategy of intake manifold pressure and mass air flow of diesel engines based on NLADRC control theory is proposed. To ensure the rationality of the designed controller, the convergence proof of the designed NLESO is also given. Finally, the performance of the designed NLADRC controller is verified from the following three aspects: the tracking effect of the set signal, the resistance to external disturbance and the robust performance to the change of fuel injection and speed. Compared with the LADRC controller and PID controller, the designed NLADRC controller is effective for the VGT-EGR system in diesel engines.

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and editing, P.Z., J.Z., Y.W. and Y.L.; project administration, Y.W. and Y.L.; funding acquisition, Y.W., Y.L. and Y.W. All authors have read and agreed to the published version of the manuscript.

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Nomenclature

VGT	variable geometry turbine
EGR	exhaust gas recirculation
MIMO	multi-input multi-output
DIDO	double-input double-output
SISO	single-input single-output
PID	proportional–integral–derivative
MPC	model predictive control
CLF	control Lyapunov function
ADRC	Active disturbance rejection control
ESO	extended state observer
LADRC	linear active disturbance rejection control
NLADRC	nonlinear active disturbance rejection control
TD	tracking differentiator
NLESO	nonlinear extended state observer
NLSEF	nonlinear states error feedback control laws
p_{im}	intake manifold pressure [Pa]
p_{em}	exhaust manifold pressure [Pa]
T_{im}	intake manifold temperature [K]
T_{em}	exhaust manifold temperature [K]
V_{im}	intake manifold volume [m ³]
V_{em}	exhaust manifold volume [m ³]
p_{amb}	ambient pressure [Pa]
T_{amb}	ambient temperature [K]
P_c	compressor power [W]
P_t	turbine power [W]
u_{vgt}	VGT valve opening ratio [%]
u_{egr}	EGR valve opening ratio [%]
W_c	mass air flow [kg/s]
W_f	fuel mass flow [kg/s]
W_{egr}	EGR gas mass flow [kg/s]
W_t	exhaust gas mass [kg/s]
W_e	mass flow into the cylinder [kg/s]
R	gas constant [J/kgK]
η_m	turbocharger mechanical efficiency
c_p	constant pressure specific heat capacity [J/kgK]
η_c	compressor isentropic efficiency
η_t	turbine isentropic efficiency
A_{egr}	effective cross-sectional areas of the EGR valve [m ²]
A_{vgt}	effective cross-sectional areas of the VGT valve [m ²]

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Article

Emulsifier-Free Water-in-Biodiesel Emulsion Fuel via Steam Emulsification: Its Physical Properties, Combustion Performance, and Exhaust Emission

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Abstract: The focus of this work is to investigate the effect of emulsifier-free emulsion fuel via steam emulsification (SD) to the diesel engine through physical properties, combustion performance, and exhaust analysis, and compare with conventional emulsion fuel with water percentages of 5% and 10% (E5 and E10) and biodiesel blend (B5). The SD was prepared using a custom 200 mL glass mixing column. The B5 fuel quantitatively was filled in the column, and then the steam was injected from the bottom of the mixing column through the porous frit glass with the pores ranging from 40 to 100 μm . The average water droplet size of SD is 0.375 μm with the average water percentage of 6.18%. The brake specific fuel consumption (BSFC) and brake thermal efficiency (BTE) of SD improved 4.19% and 3.92%, respectively, as compared to B5. The in-cylinder pressure (ICP) was lower than B5, however, yielding close to the B5 at 4 kW engine load. As for the exhaust emission test, NO_x and PM for SD were reduced significantly with a percentage reduction of 25.22% and 10.68%, respectively, as compared to neat B5. The steam emulsification method offers a huge potential to be explored further as the concept offers the alternative method of making emulsion fuel without the use of conventional mechanical mixers.

Keywords: steam condensation; diesel engine; exhaust gas emission; water-in-biodiesel emulsion fuel; combustion characteristic

1. Introduction

Research interest in water-in-diesel/biodiesel emulsion fuel has grown exponentially worldwide. This is due to its significant impact on the exhaust emission, especially on the simultaneous reduction of nitrogen oxides (NO_x) and particulate matter (PM). Recent studies focused on improving the quality of the emulsion fuel by the addition of nano-sized particles, which include metallic, nonmetallic, organic, oxygenated, and their combination. The results reported that the nanoadditive is able to improve the thermal physical properties, enhance the heat transfer rate, and stabilize the fuel mixtures [1]. Hasanuddin et al. [2] tested various nanometal additives blended into water-in-diesel emulsion fuel (WD) consisting of aluminum oxide (Al_2O_3), copper(II) oxide (CuO), magnesium oxide (MgO), manganese(IV) oxide (MnO), and zinc oxide (ZnO). The authors reported that WD blended with Al_2O_3 yielded the highest reduction of brake specific fuel consumption, up to 5.5%, as compared to neat WD. Sajith et al. [3] carried out an extensive investigation on a diesel engine fuelled with and without cerium oxide additives (CeO_2), to study the performance and emission characteristics. They found that the

viscosity and volatility hold direct relations with the doping level of 20–80 ppm. The emission levels of HC and NO_x are appreciably reduced with the inclusion of CeO₂. Another study tested nanofluid emulsion, which consists of 50 and 100 ppm of TiO₂ nanoparticles blended in the orange peel oil biodiesel that are denoted as OOME-T50 and OOME-T100, respectively. The engine test results were compared with the neat biodiesel. The result reported that brake thermal efficiency (BTE) increased by 1.4% and 3.0% for OOME-T50 and OOME-T100, respectively, as compared to the biodiesel. In addition, a significant reduction of about 9.7%, 24.2%, 18.4%, 16.0% in smoke, CO, NO_x, and HC was observed for OOME-T100 nanoemulsion fuel [4]. Nevertheless, the drawbacks of inclusion of nanoadditives in WD are high cost and engines prone to technical issues after long-term usage, such as injector chocking and piston ring sticking [5].

Apart from that, the water-in-diesel/biodiesel emulsion has a major weakness that hinders the commercialization of the fuel, which is a stability issue. The stability of WD is strongly dependent on the surfactant that functions as a stabilizing agent by reducing the interfacial tension between water and oil [6]. However, once exceeding the stability period, the immiscible liquids will separate and will harm the engine system. Most of the studies conducted are towards finding the best stabilizing agent that is able to preserve its stability for a very long time [7]. Preetika et al. [8] conducted a detailed analysis of the stability of water-in-diesel emulsion fuel stabilized by a commercially available surfactant, Span[®]80 and Tween[®]80. The authors found that 10% of the surfactant was able to enhance time stability with no macroscopic separation, although the droplet size increases from 159 to 330 nm in 50 days.

Apart from fuel quality enhancement through nanoadditives and surfactant studies, there are a few studies heading toward a different research direction, which is producing WD without the presence of surfactant and/or additives, the so-called emulsifier-free WD or nonsurfactant emulsion fuel (NWD) [6,9,10]. This fuel is considered as unstable emulsion fuel when the fuel will reach to sedimentation within less than 60 s. The NWD was made by continuously mixing water and diesel through an in-line mixing system and directly supplied into the engine, thus eliminating the need for high-stability emulsion. Ithnin et al. [6] tested the NWD in a single-cylinder diesel engine through an in-line mixing system that consists of a combination of high-shear mixer and ultrasonicator. The result revealed that the NWD does give substantial improvement to the engine, with a 3.89% reduction in BSFC and 3.59% increase in BTE as compared to diesel fuel. PM and NO_x decreased with a significant reduction of 16.33% and 31.66%, respectively. Another method of producing NWD was reported by Aizam et al. [9], by using a concept of rapid mixing through an internal mixing injector. However, this concept is applied to the burner system.

The drawback of producing NWD through a continuous rapid mixing method is the setup employed many components, among others, a pump, high-shear mixer, and ultrasonic transducer thus requiring additional maintenance cost to the engine. Sugeng et al. [11] have tested a new concept of making NWD by employing thermal energy carried by steam and found out that the emulsion possibly can be made without the use of a mechanical stirrer and/or ultrasonic transducer, yielding potential savings in costs. However, the constraint of the concept is that it is difficult to control the same condition of the fuel every time the fuel supplied to the engine, especially the water content. Apart from that, no detailed studies have been conducted to evaluate the physical properties and combustion characteristic of NWD via steam emulsification. From that motivation, this study aims to further investigate the effect of emulsifier-free emulsion fuel via steam emulsification (SD) to the diesel engine through the physical properties, combustion performance, and exhaust analysis, and compare it with conventional WD and biodiesel–diesel blend. The biodiesel blend used in the study is B5, where 5% palm oil methyl ester is blended with the neat diesel fuel (B5). A new prototype of steam emulsification has been developed with control parameters, such as flow rate and temperature for ensuring the consistent condition of SD supplied to the engine; it is installed close to the engine fuel feed system. In the present experimental work, the surfactant-added emulsions with 5% and 10% water content, denoted as E5 and E10, respectively, and are tested for comparison with SD. A single-cylinder,

direct injection diesel engine is tested under four different load conditions (1, 2, 3, and 4 kW) and with a constant engine speed of 2000 rpm. As for the engine combustion performance analysis, the brake specific fuel consumption (BSFC), brake thermal efficiency (BTE), in-cylinder pressure (ICP) and rate of heat release (ROHR) are analyzed in detail. The exhaust emission analysis, on the other hand, covers the PM, NO_x, CO, and CO₂.

2. Experimental Setup and Procedure

2.1. Fuel Preparation

In the present study, SD was compared with surfactant-added emulsion fuel and biodiesel–diesel blend fuel. The surfactant-added WD was prepared using the conventional method by adding 5% and 10% water percentage to B5 and fuels denoted as E5 and E10, respectively. The Span[®]80 with a HLB (hydrophilic–lipophilic balance) of 4.3 was added at a concentration of 2% to stabilize the emulsions. The emulsion fuel was prepared by using a mechanical homogenizer at a propeller speed of 2500 rpm for 5 min. As for the biodiesel–diesel blend fuel, 5% of palm oil methyl ester blended with the neat diesel fuel (B5) was used.

SD was prepared using a custom 200 mL glass column with the schematics of the mixing column as illustrated in Figure 1. The experimental setup for the steam emulsification system is shown in Figure 2. As shown in Figure 1, the biodiesel was filled halfway up the column through the biodiesel inlet (2) to allow the fuel to exit through the side nozzle, and then the engine was run for 10 min. The steam was supplied at the bottom nozzle of the column through the steam inlet (4). The steam inlet (4) nozzle was attached with porous frit glass orifices with a P2 (P100) European standard mesh size (ISO 4793), as shown in Figure 3 P2 has pores ranging from 40 to 100 µm. A 400 W steam generator was used for the production of steam. Thermocouples were installed at the column's body and the fuel feed system to monitor the emulsion temperature in the column and at the point of injection, respectively. After the 10 min warm-up, the steam was supplied to the column and bubbles of steam rose from the steam nozzle creating a turbulent mixing. Some steam leaves the bulk liquid uncondensed as vapor through the steam outlet channel (1), as shown in Figure 1. Accumulated water condensate at the bottom of the device was discharged from the mixer manually through the water discharge channel (5). As the temperature in the column stabilized at 80 °C, the measurements began. The fuel eventually cooled down during transport and reached the injector pump of the engine at 70 °C before the fuel was injected into the combustion chamber through the emulsion outlet channel (3).

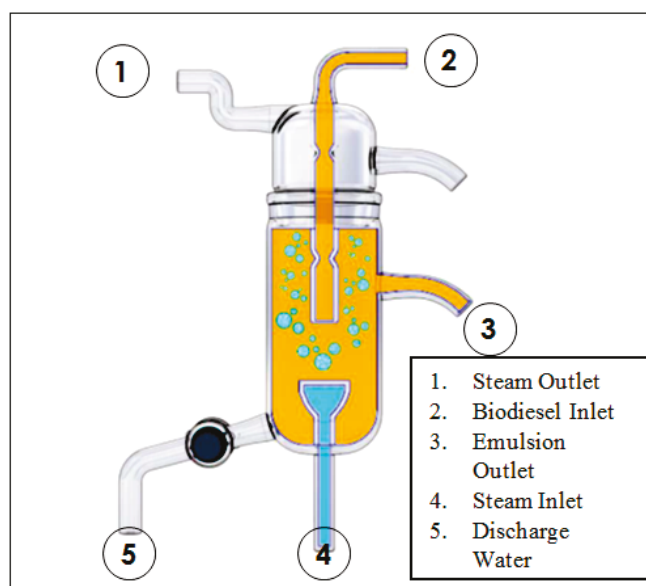


Figure 1. Schematic diagram of the steam emulsification mixing device.

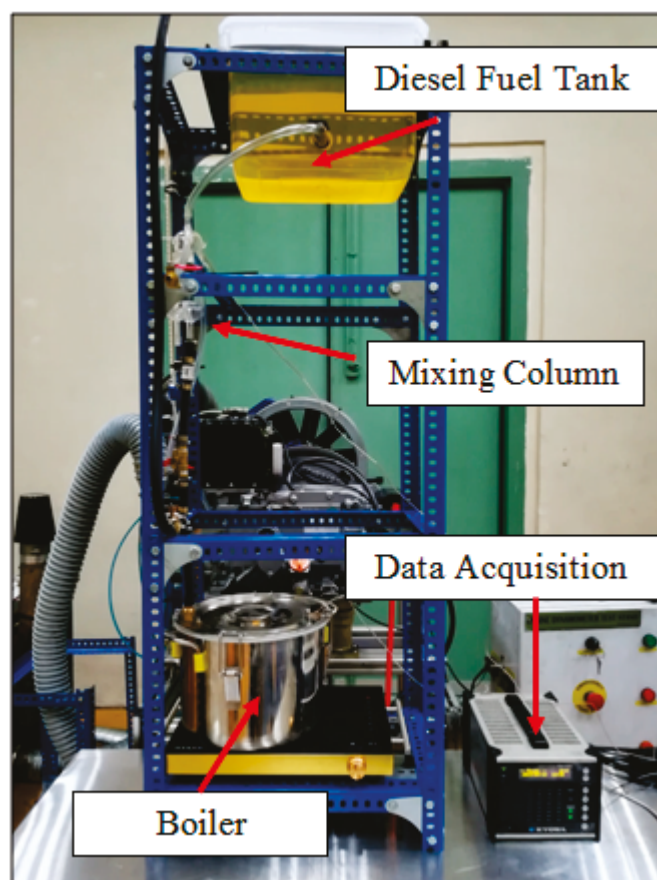


Figure 2. Apparatus setup for steam-generated emulsion fuel.

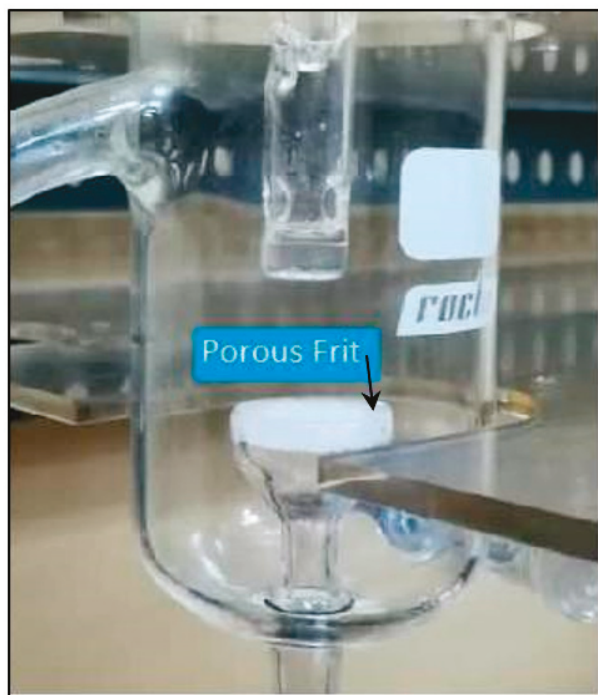


Figure 3. Porous frit P2 in the mixing column.

2.2. Fuel Properties Test

The fuel properties test of SD includes water droplet size and distribution, water percentage, calorific value, and viscosity. The water droplet size and distribution measurements were conducted using a laser diffraction particle size analyzer SALD-2300. Calorific value and viscosity were measured using a CAL2k ECO Bomb Calorimeter and rotary viscometer. As for the water percentage measurement of SD, the distillation method was utilized. SD samples were taken during the engine tests at each incremental load of 1 kW. Conforming to the ASTM D95 standards, the samples were then distilled at 130 °C, just enough to remove the water from the emulsion. The water percentage is determined by calculating the weight of the fuel before and after the distillation process.

2.3. Engine Test Setup

The type of engine used in the test is a Yanmar TF90-M, 0.493 L single-cylinder, four-stroke, water-cooled, direct injection diesel engine. Other basic specifications of the engine are shown in Table 1. The dynamometer used in the experiment is a 10 kW eddy current KLAM RETARDER T10 with maximum torque up to 25 Nm at 3700 rpm. The accuracy for torque and speed sensor is $\pm 0.25\%$ and ± 1 rpm, respectively. The BSFC of B5 and emulsion fuels were measured by using a 50 mL burette. The fuel consumption was then measured by measuring the time elapsed for the fuel to reach the 5 mL mark. The tests were repeated at least three times. As for the combustion characteristic analysis, a Kistler in-cylinder pressure sensor with Dewesoft Data acquisition system was utilized. The data logger is capable of recording a pressure measurement for every 0.1 degree of crank rotation increment. For each test, a continuous 50 cycle combustion pressure was recorded and reported after averaging them. The injection timing of the engine for the test was set constant to 18° (BTDC) for all engine loads (1–4 kW). As for the emission measurements, an ECOM J2KN gas analyzer was used for the measurement of NO_x, HC, CO, and CO₂. As for the measurement of the PM, a custom minidilution tunnel with a dilution ratio of 10 was used. The filter was used for trapping the PM was a Teflon filter (MILLIPORE FHLP04700) with a diameter and orifice of 47 mm and 0.45 μ m, respectively. The PM concentration was determined by weighing the PM filter before and after the experiment. A schematic diagram of the minidilution tunnel is shown in Figure 4.

Table 1. Specifications of the diesel engine.

No. of Cylinders	1
Brand	Yanmar TF90-M
Cylinder bore $\times$ stroke, mm	85 $\times$ 87
Displacement, L	0.493
Cooling system	Water-cooled
Maximum rated power output, kW	7.0
Maximum torque, kgf.m/rpm	3.4/1600
Rated engine speed, rpm	2400
Fuel injection pump, kg/cm ²	Borsch type
Injection pressure	200
Injection timing.	18 BTDC
Compression ratio	18

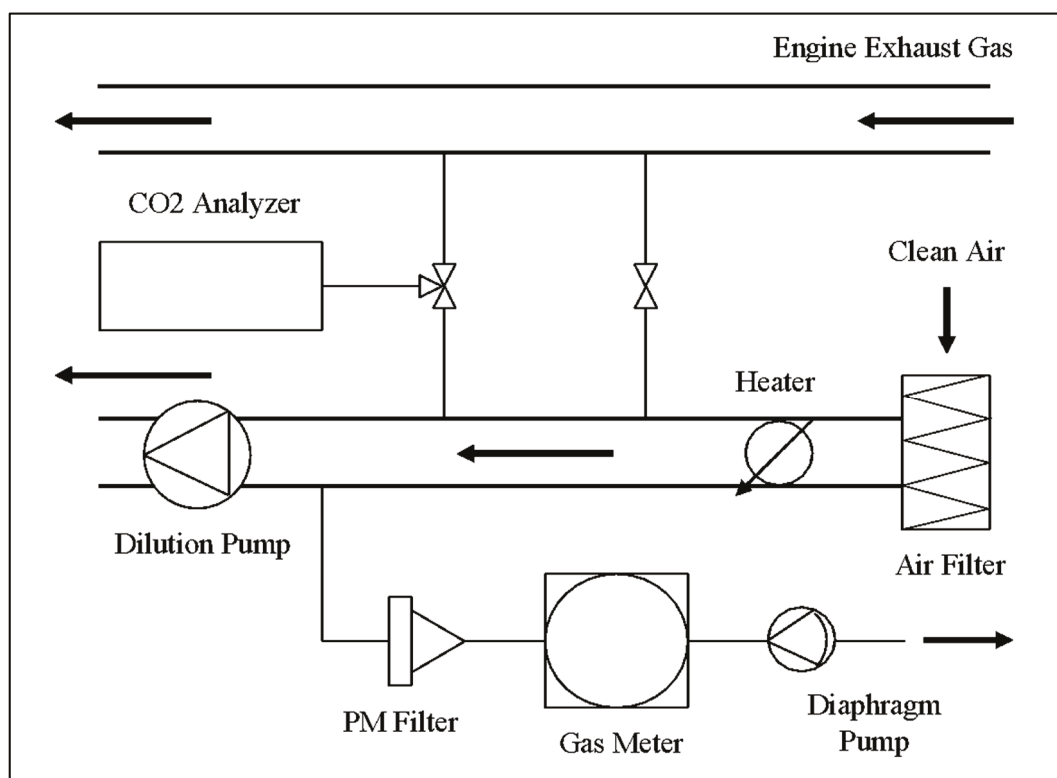


Figure 4. Schematic diagram of a minidilution tunnel.

3. Result and Discussion

3.1. SD Formation Observation

The sequence formation of SD via steam emulsification from 7 to 40 s duration is shown in Figure 5. As shown in the figure, by using the porous frit membrane for steam distribution, an interesting mechanism was observed. As the steam enters the mixing column homogeneously by the help of the porous frit membrane, the steam turned directly into cloudy water droplets due to the high-temperature difference between steam and biodiesel at 20 s duration time. During the contact, some steam will condense and change its phase to liquid water droplets suspended in the dispersion. The phase change is called the warm condensation effect, a condensation process where the vapor temperature is higher than the sink. Later in the steady-state phase, as the temperature of the bulk emulsion approaches the steam temperature, a fraction of the steam bubbles reached the surface and created boiling effects, which agitated the emulsion bulk at 25 s duration time. At 40 s mixing time, a whitish milky color emulsion was formed in the mixing column indicating that emulsion had started to form homogeneously. The agitation created by the steam ensures that the emulsion stays homogenous before it enters the engine fuel feed system. From the observation experiment, it is clearly shown that the emulsion fuel possibly can be formed without using any surfactant and mechanical mixers through steam emulsification.

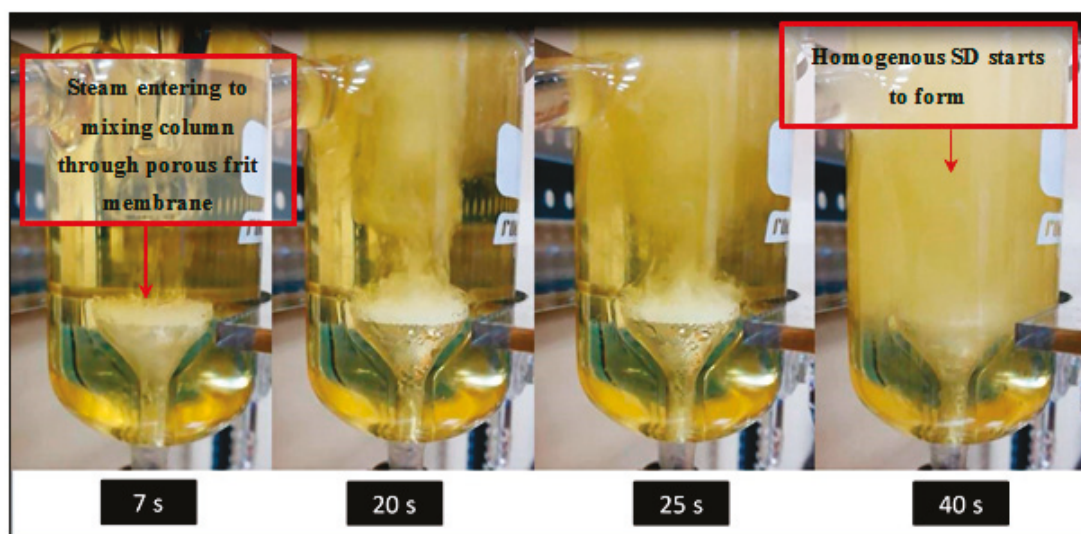


Figure 5. Sequence formation of SD via steam emulsification.

3.2. Fuel Properties

The water content of SD that was measured by the distillation method is listed in Table 2. As shown in the table, the water content in SD was obtained with an average of 6.18% using the distillation method. The sample of SD was taken from every load condition (1 kW to 4 kW). The value of the water content of SD showed slight differences between each load; this is due to the difficulty in controlling precisely the rate of condensation of steam injected into biodiesel. The density, viscosity, and calorific value for all types of fuel are tabulated in Table 3. The viscosity of E10 was the highest compared to other fuels due to its highest water content. The viscosity of SD was measured at 70 °C since the fuel reached the engine fuel feed system was at that particular temperature caused by the high temperature of the steam. The viscosity of SD was the lowest as compared to other fuel, which was 2.2 mm²/s at 70 °C. However, as the temperature decreased to 40 °C, the viscosity equaled to that of E5, which was 4.6 mm²/s. This was due to the almost similar water content for both emulsions. As for the calorific value, all types of WD were lower than that of B5. However, when comparing calorific value between the emulsions, E5 was the highest, followed by SD and E10. It was clearly shown that the calorific value of the WD was reduced when the amount of water in the WD was increased. This was due to the presence of water that lowers the heating value of WD.

Table 2. Water percentage of SD.

Load (kW)	1	2	3	4	Average
Water Percentage (%)	5.99	6.34	6.27	6.13	6.18

Table 3. Properties of fuels.

Properties	B5	E10	E5	SD
Density (g/cm ³)	0.86	0.88	0.87	0.87
Calorific Value (MJ/kg)	45.28	40.78	43.02	42.54
Viscosity (mm ² /s) @40 °C	4.3	4.7	4.6	4.6
				2.2 @70 °C

3.3. Droplet Size Distribution

The presence of the submicron droplets was confirmed by the droplet size distribution measurement using the Shimadzu SALD 2300, as shown in Figure 6. The measurements were

started 45 min after the preparation process. As shown in the figure, the distribution curves show a lognormal distribution with a mean droplet size of $0.375\ \mu\text{m}$ for SD. The lognormal distribution in fine particles follows the Smoluchowsky model, where fine particles meet and coalesce [12]. As shown in the figure, a series of average droplet size is plotted at an interval of 15 min. The mean droplet size was increased, from $0.375\ \mu\text{m}$ to $0.476\ \mu\text{m}$. The SD is considered as unstable emulsion fuel. Due to the absence of the surfactant, the dispersed water droplets tend to gather with each other. The droplets initially are separated by a thin film. Due to the attraction of the van der Waals forces, the thin film thickness rapidly decreased over a short period of time. Once the droplet film reached a critical value, it broke and lead to newly formed droplets to move to each other, forming a larger droplet [6]. However, the system employed the concept of continuously mixing biodiesel fuel with steam through an in-line mixing column before the produced fuel is directly supplied into the engine. Thus, the concept does not require high stability of emulsion for fuelling the engine.

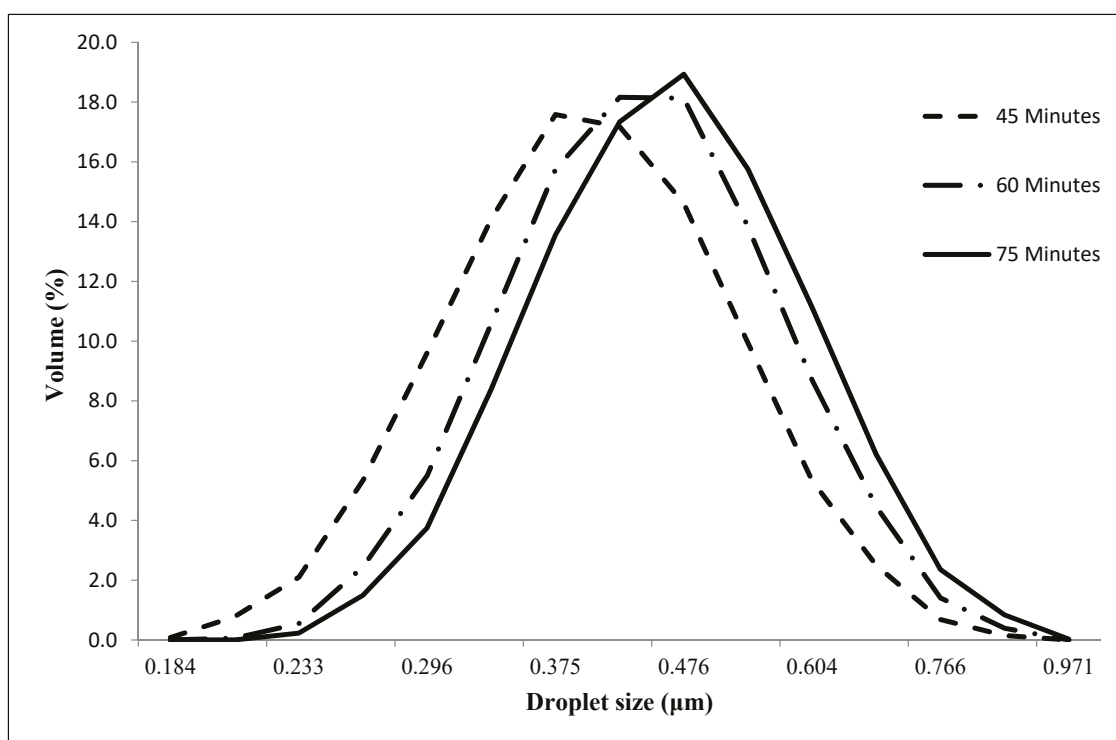


Figure 6. Droplet size distributions of SD.

3.4. Engine Performance

For the engine performance test, all types of fuels were tested at an engine speed of 2000 rpm with four different loads ranging from 1 to 4 kW applied on separate runs. Brake specific fuel consumption (BSFC), brake thermal efficiency (BTE), in-cylinder pressure (ICP), and rate of heat release (ROHR) for each fuel were analyzed and evaluated.

3.4.1. Brake Specific Fuel Consumption

The brake specific fuel consumption (BSFC) of the engine, fuelled with B5, SD, E5, and E10 under varying load from 1 to 4 kW with a constant speed of 2000 rpm, is illustrated in Figure 7. As shown in the graph, the fuel consumption of all types of fuel shows a similar trend where the BSFC reduced as the load increased, indicating that the engine combusts more efficiently at a higher load. When comparing the emulsions with B5, all WD consumed lower fuel usage with the average reduction of 8.14%, 4.19%, and 2.08% for E10, SD, and E5, respectively. The BSFC of WD was considered by taking only biodiesel as total fuel for all types of emulsion fuel, including E5, E10, and SD. By means, the amount of water

inside WD was not considered as fuel consumption. Although the energy content for all the WD is significantly lower compared to the B5, the WD consumed the fuel much less than the B5. This is due to the help of microexplosion phenomena that improve combustion efficiency [13]. Microexplosion is secondary atomization of the primary spray as a result of the rapid evaporation process of water that is initially contained in the oil drop, resulting in more surface area of the fine droplets that can be exposed to the air, leading to an improvement in the fuel and air mixing process [7]. E10 consumed the least amount of fuel, followed by SD and E5. This is because of the higher water content in the fuel as compared to the other WD. According to Jeong and Lee [14], the higher water content in the emulsion will create a longer and more intense microexplosion. Nevertheless, SD was still able to combust better compared to E5 and B5 due to higher water content. Other than that, the viscosity of SD is much lower compared to the other emulsions due to the higher temperature caused by the steam, resulting in improved spray atomization and broader spray angle. Consequently, this enhances the air–fuel mixing, thus improving the combustion.

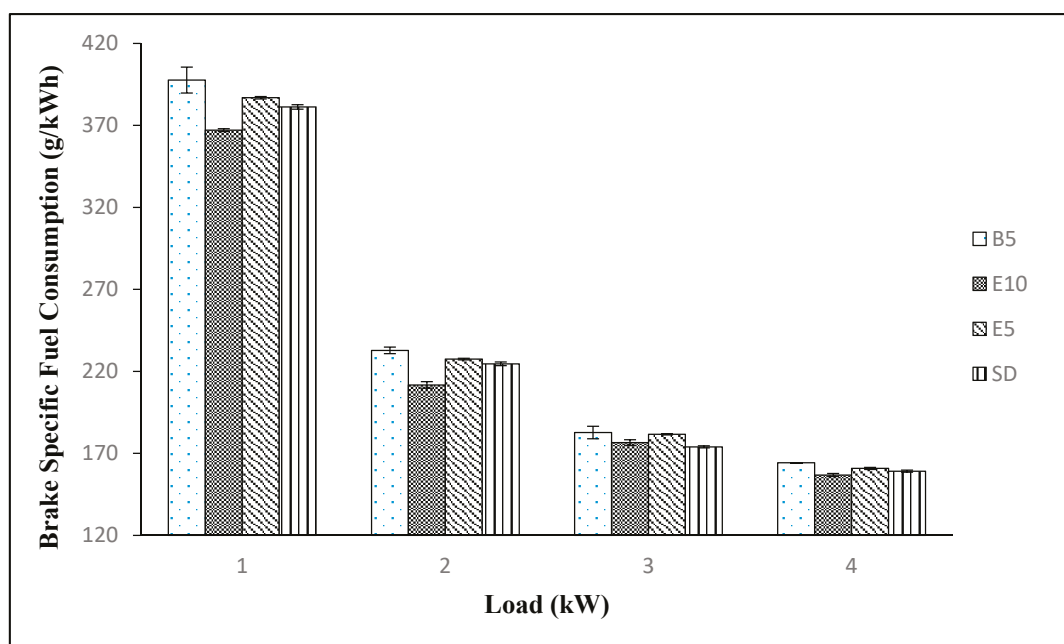


Figure 7. Brake specific fuel consumption versus load.

3.4.2. Brake Thermal Efficiency

The brake thermal efficiency (BTE) of the engine fuelled with B5 and all types of WD with different loads ranging from 1 to 4 kW is shown in Figure 8. As shown in the graph, the brake thermal efficiency of all types of fuel increases as the load increases. Again, this is due to the improvement of engine combustion at high loads, which is in agreement with the finding from the BSFC result. The BTE of all types of WD is higher than that of B5 in all load conditions. Similar to BSFC, the calculation of BTE only considers the biodiesel as the total fuel. The BTE of E10 is reported to be the highest compared to B5 with 5.67% improvement, followed by SD (3.92%), and E5 (1.77%), though all the emulsions have lower calorific value than B5. A similar trend as in BSFC result, BTE of SD stands between E10 and E5, which is mainly due to the amount of water content in the emulsion that affects the onset and strength of the microexplosion. Other than that, the low viscosity of SD as compared to E5 and E10 also contribute to the improvement of the combustion efficiency through better fuel atomization and wider spray angle. In addition, the improvements of BTE for all WD also come from the ignition delay of the fuels. The detailed discussion on the combustion analysis is discussed in the subsection on combustion characteristics. The longer ignition delay during the combustion, the more fuel that can be physically prepared for the chemical reaction, leading to an increase in the rate of heat release.

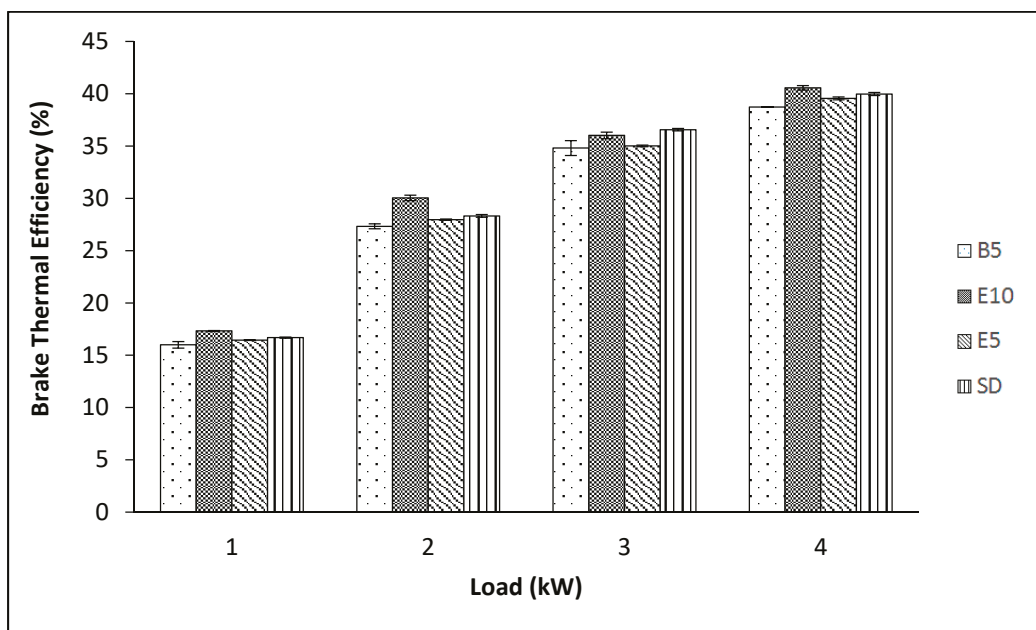


Figure 8. Brake thermal efficiency versus load.

3.4.3. In-Cylinder Pressure

Figures 9–12 show the in-cylinder pressure (ICP) for B5 and all types of WD with different loads ranging from 1 to 4 kW under constant speed. As shown in Figures 9–12, it is clear that at a low load condition, the peak pressure of all types of fuel rises a little late after TDC. Nonetheless, at a high load condition, the peak pressure starts to rise earlier. This is because, at a low load condition, less fuel is injected into the combustion chamber, thus producing a lower residual gas temperature and wall temperature, leading to a lower charge temperature at injection timing. As a result, the ignition delay increased and retards the start of combustion (SOC). At high engine load, the gas temperature inside the cylinder is higher due to the higher amount of fuel injected into the engine, promoting to shorten the ignition delay period and advancing the SOC [15]. Compared to B5, the peak of in-cylinder pressure of all types of WD is lower and rises later in all load conditions. The lower ICP for all WD is due to the lower energy content of the emulsions due to the small amount of fuel that is replaced with water. When comparing between the emulsions, E5 has the highest ICP, followed by SD and E10 at load condition of 1–3 kW. At 1 kW, the trace of ICP for SD is close to E10 while at 2 kW and 3 kW, the peak in-cylinder pressure of SD is slightly lower than that of E5. These may be due to unstable and inconsistent water percentage inside SD that formed through the steam condensation method. However, at 4 kW, SD combusts with the highest ICP, yielding closest to B5. This is due to the lower viscosity of SD that enhanced the fuel atomization at high temperature and load condition, thus improving the air–fuel mixing process, which results in higher peak cylinder pressure. In addition to that, the occurrence of microexplosion of the emulsion is advanced at a higher engine load condition. If the temperature is relatively high, the heat flux is larger, leading to the transference of a higher rate of heat transfer, and, hence, leading to a quicker onset of microexplosion [16].

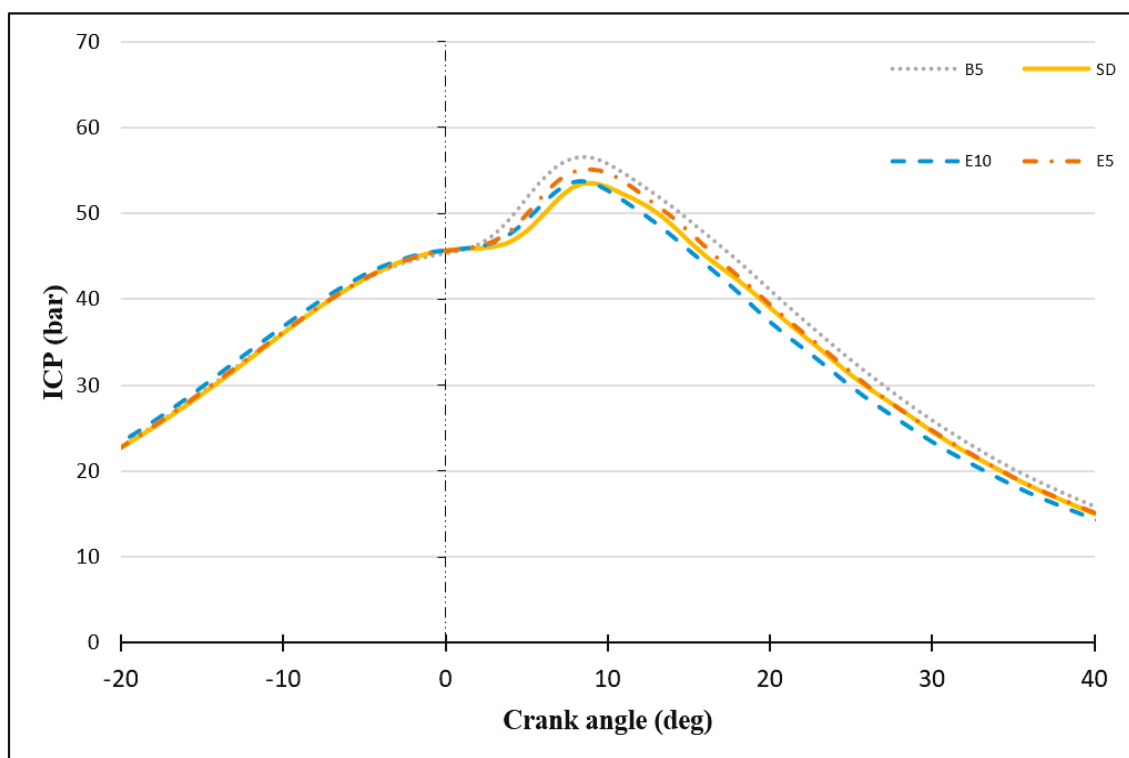


Figure 9. In-cylinder pressure (ICP) traces for B5, E10, E5, and SD at 1 kW.

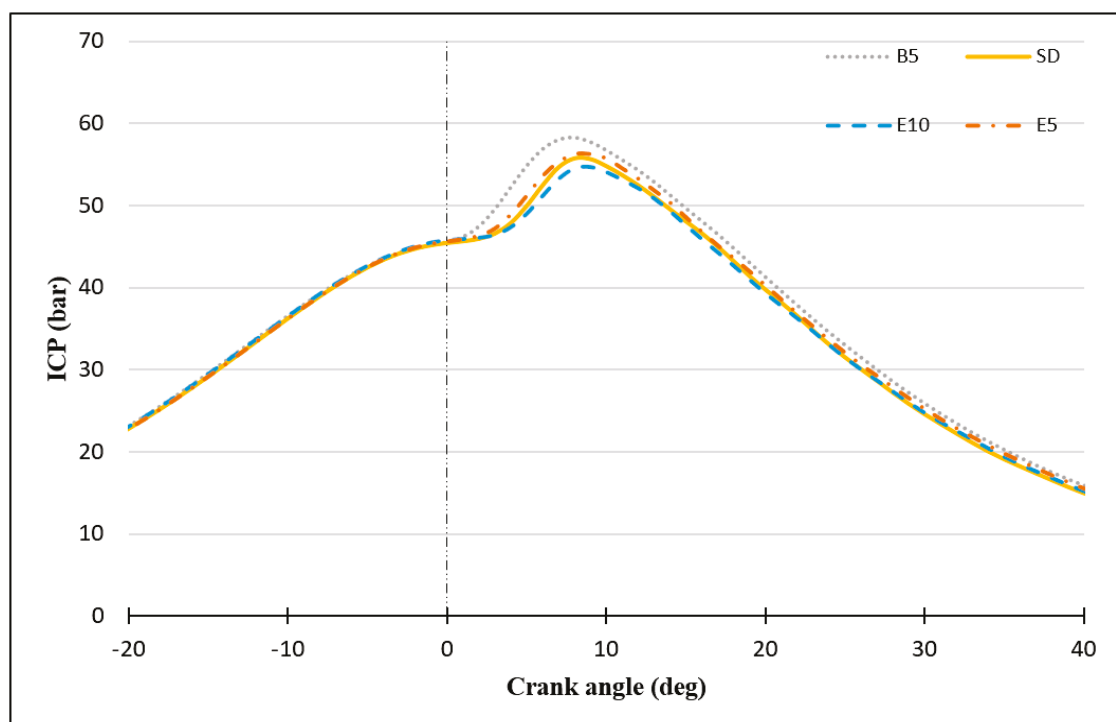


Figure 10. In-cylinder pressure traces for B5, E10, E5, and SD at 2 kW.

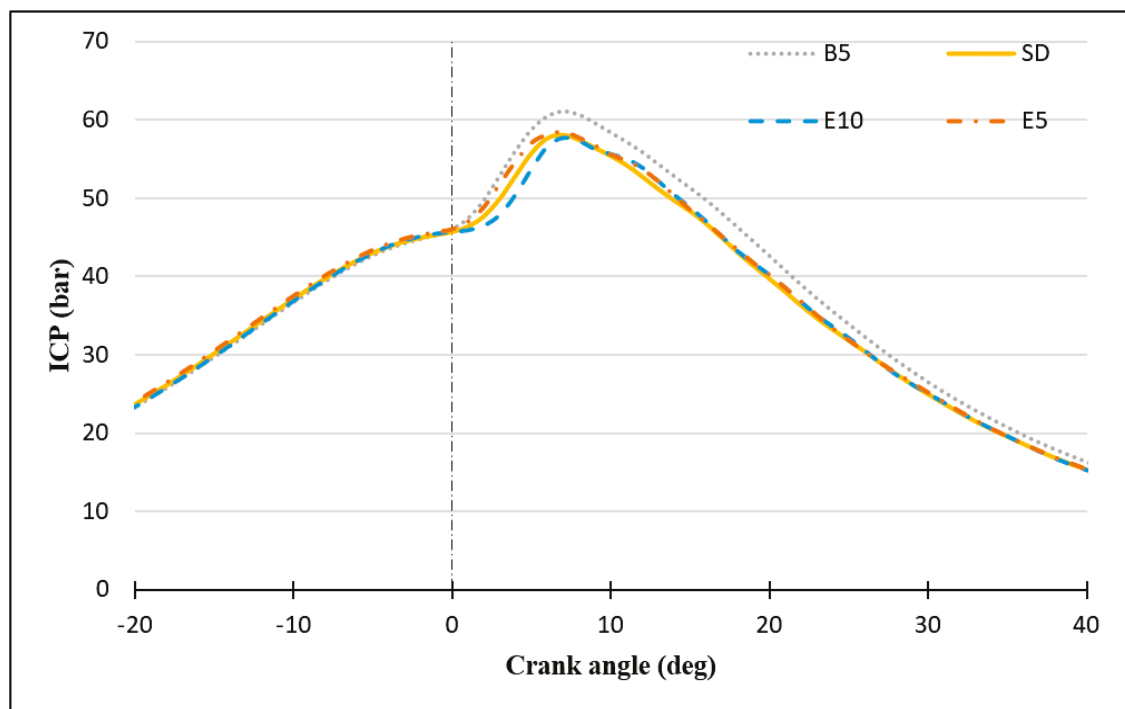


Figure 11. In-cylinder pressure traces for B5, E10, E5, and SD at 3 kW.

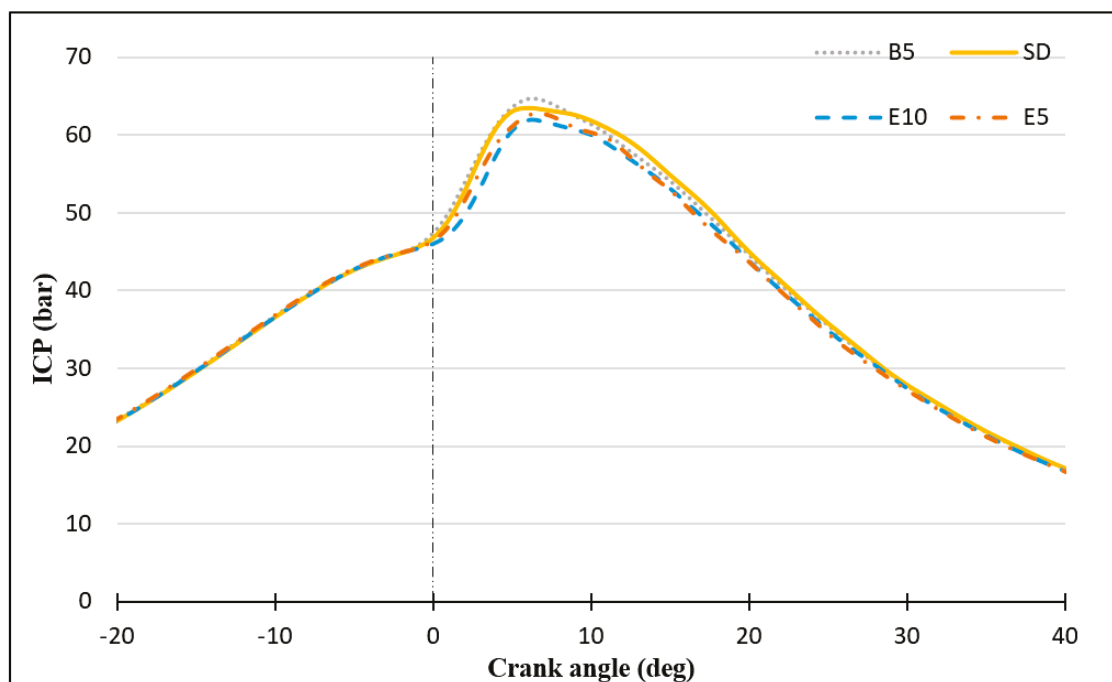


Figure 12. In-cylinder pressure traces for B5, E10, E5, and SD at 4 kW.

3.4.4. Rate of Heat Release

Figures 13–16 show the rate of heat release ROHR with traces for B5 and WD (E5, E10, and SD) at varied engine load conditions from 1 to 4 kW and constant speed of 2000 rpm. Rate of heat release is

calculated through the following equation, which is based on the pressure data of the engine cylinder and the ratio of specific heat, γ , is set constant.

$$Q = \frac{\gamma}{\gamma + 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta}$$

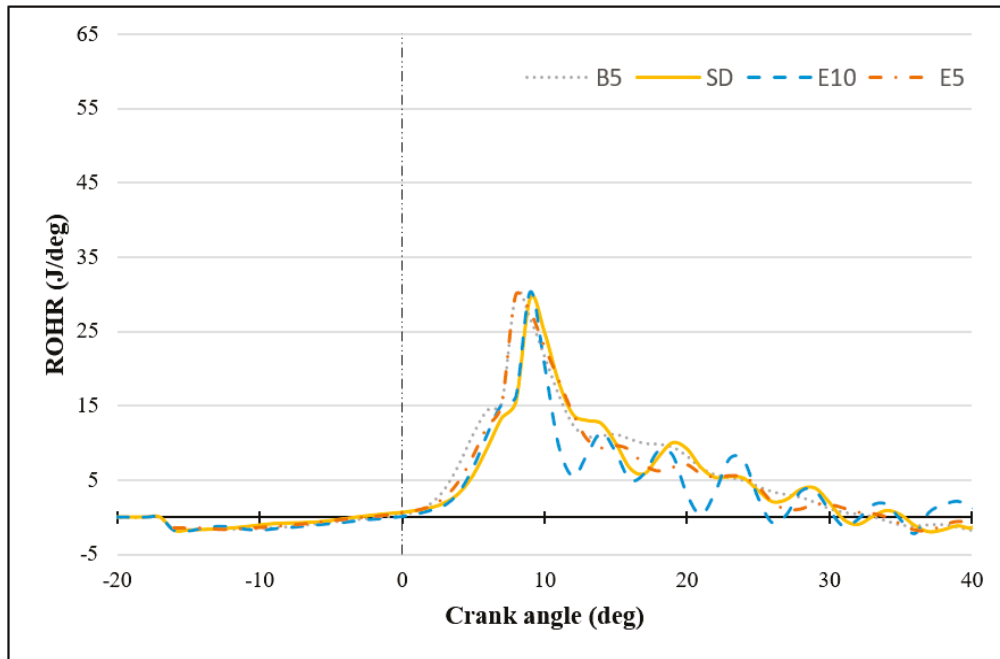


Figure 13. Comparison of the ROHR at 1 kW for various fuels.

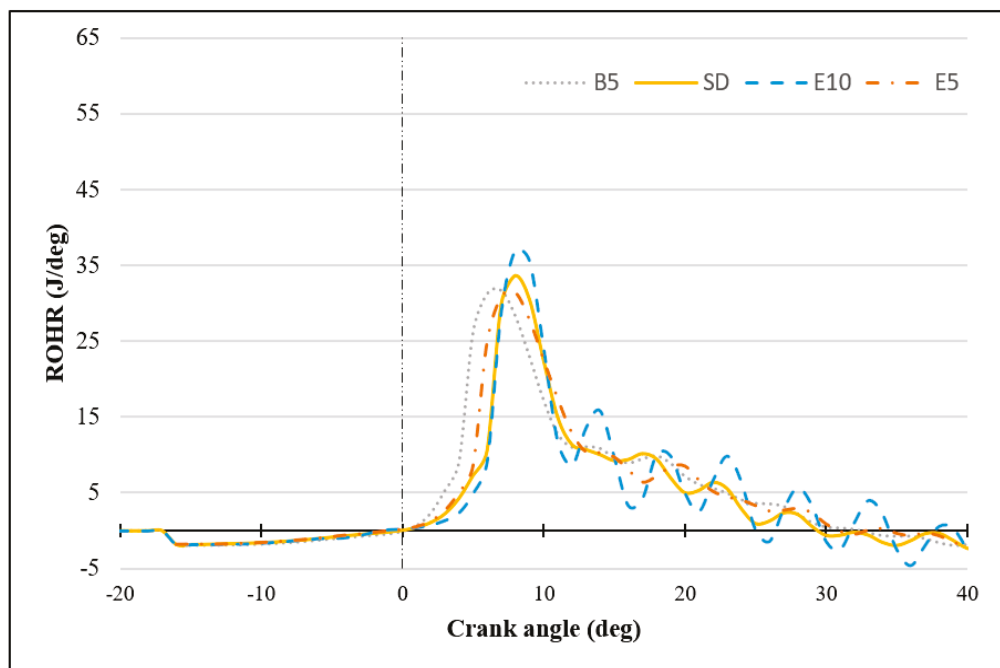


Figure 14. Comparison of the ROHR at 2 kW for various fuels.

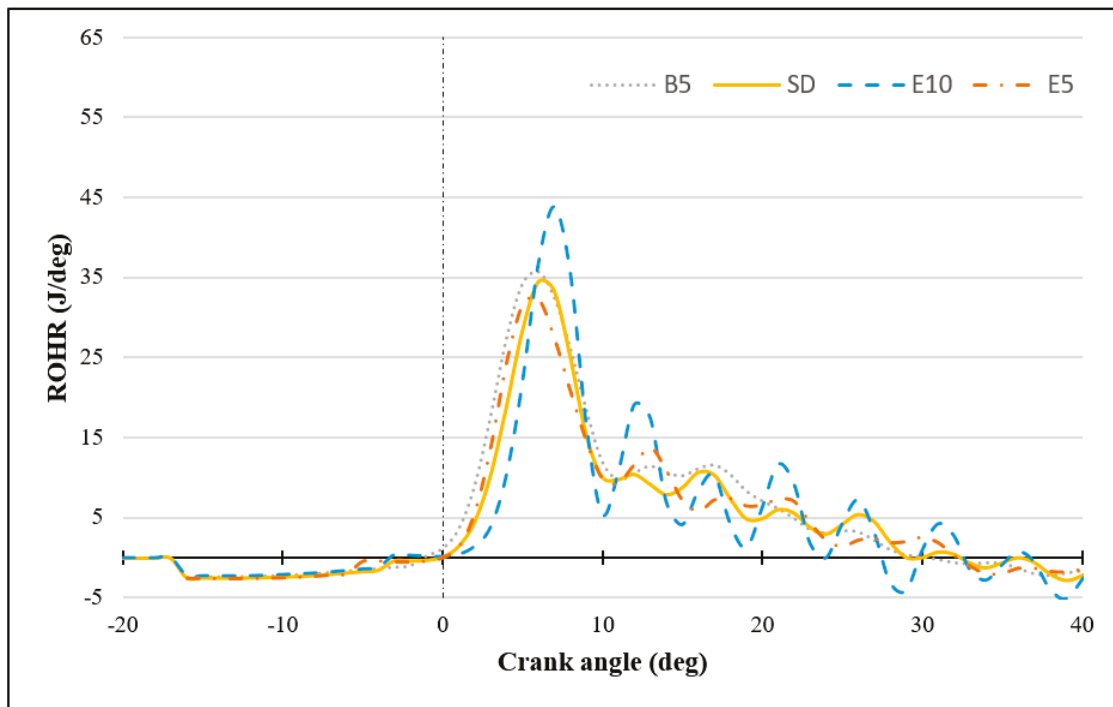


Figure 15. Comparison of the ROHR at 3 kW for various fuels.

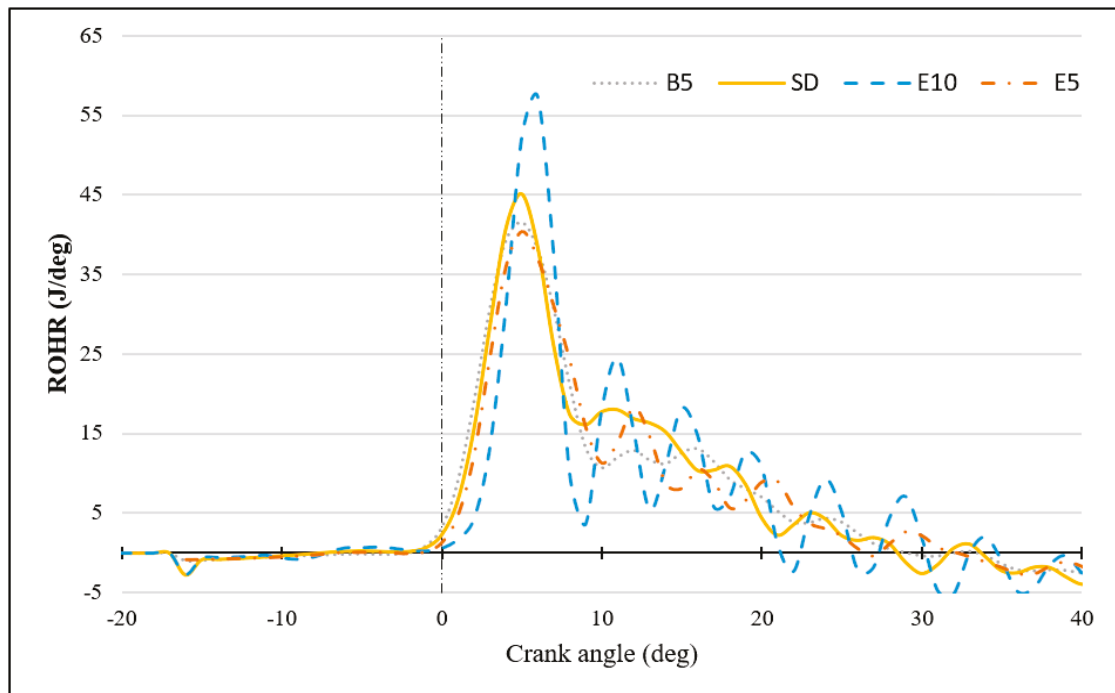


Figure 16. Comparison of the ROHR at 4 kW for various fuels.

As shown in the figures, all fuels show the trend of its respective ROHR peak rises steadily as the load increases, confirming that the engine combusts more efficiently as the load is increased. Comparing the emulsions with B5, it is clearly shown that the SOC of all emulsions was delayed. The increase in ignition delay is due to the presence of the water in the emulsion, where the water forms latent heat during the vaporization process. Subsequently, it slows the rate of the temperature increment of the droplet. As a result, the physical delay increases. Simultaneously, the presence of water vapor reduces

the fuel concentration, resulting in an increase in the chemical delay. Both the chemical and physical delay increase in the overall ignition delay [17]. However, as the load increases, the ignition delay for all WD is shortened and almost similar to biodiesel, especially at 4 kW. Once again, this is due to the onset and strength of the microexplosion that is perceptible at a high load condition. Both SD and E10 produced higher ROHR as compared to B5, especially at 3 and 4 kW load. The microexplosion most likely caused this. The same effect did not show as clearly in the lower loads. This might be that at lower loads the effect of water quenching is more perceptible than the effect of the microexplosion.

3.5. Exhaust Emission

3.5.1. Nitrogen Oxides

Nitrogen oxides (NO_x) emissions are among the major issues of diesel engines. Figure 17 shows the concentration of NO_x of diesel engine fuelled with B5, E10, E5, and SD under different load conditions with constant speed. As shown in the graph, the concentration of NO_x in all types of WD is greatly reduced compared to that of B5. E10 records the lowest concentration of NO_x compared to B5, with an average reduction of 28.16% in every load condition, followed by SD (25.22%) and E5 (21.67%). It is clearly shown that NO_x formation is reduced when the amount of water is increased. This is due to the high latent heat from the evaporation of water in the emulsion when it absorbs the heat during the combustion [18]. According to Farfaletti et al. [19], the combustion temperature is reduced due to the heat sink effect where the water content in the inner phase absorbs the calorific heat value of the emulsion, thus reducing the burning gas temperature inside the combustion. As a result, the generation of NO_x was restricted. However, when comparing the fuels in each load, at 4 kW the NO_x reduction for all the emulsions was lower compared to other loads. There was only 5.51%, 6.16%, and 7.09% NO_x reduction for E5, SD, and E10, respectively, as compared to B5. This phenomenon is another proof showing that microexplosion helps the fuel to combust better at high load, causing the higher combustion temperature. Nevertheless, with the presence of water, WD still emits lower NO_x as compared to B5.

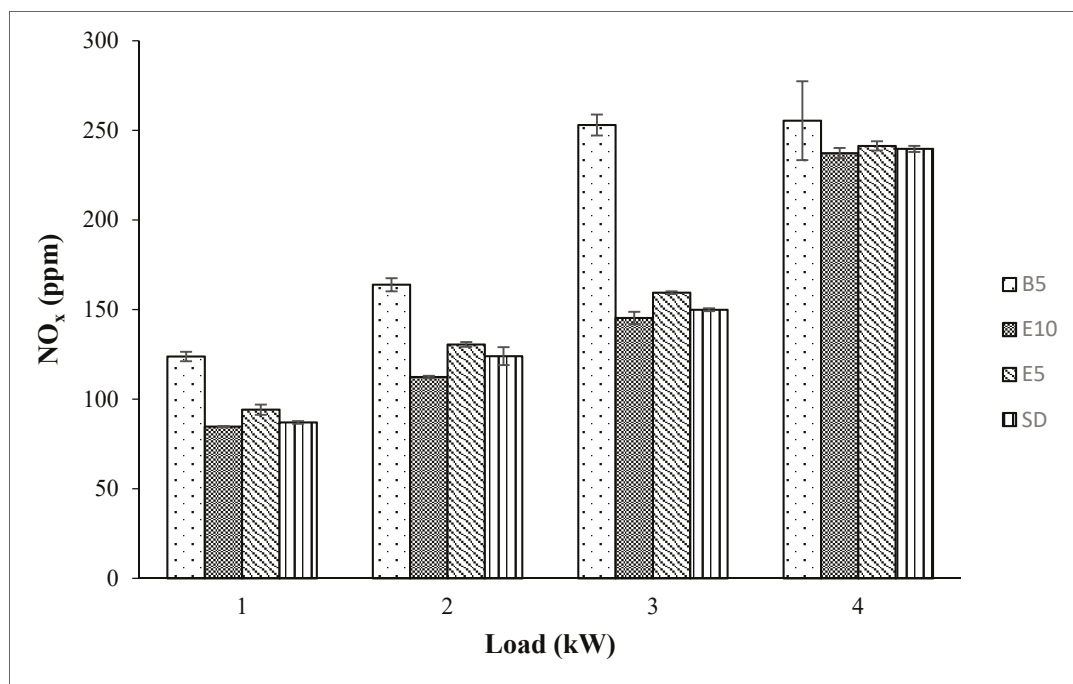


Figure 17. Concentration of NO_x for B5, E10, E5, and SD under different load conditions.

3.5.2. Carbon Monoxide

Figure 18 shows the concentration of CO for B5, E10, E5, and SD with different loads ranging from 1 to 4 kW under the constant speed of 2000 rpm. As shown in the graph, all fuels consistently show a similar trend, where the CO formation is significantly reduced as the load increased, evidence indicating the engine combusts more efficiently at a higher load condition. This is because, in the low load condition, the combustion is usually in the quenching and lean flame zones, resulting in the formation of more CO. All WD produced higher CO in all load conditions as compared to B5. The highest formation of CO is recorded at 1 kW where E10 emits the highest value, with 52.66% of increment followed by SD (41.84%) and E5 (37.7%). The main reason for the increment of CO in WD is due to the presence of water, which lowers the calorific value of the fuels. As a result, the combustion temperature decreased, resulting in shortened oxygenating time for the CO gas. In addition, the oxidation process of CO would freeze when the combustion temperature is lower than 1400 K, thus, retarding the conversion process of CO to carbon dioxide (CO₂) [20]. Furthermore, the longer ignition delay of the emulsions also contributed to higher CO. However, as the load increases, the increment of CO of emulsions as compared to B5 are reduced; at 4 kW, the E5, SD, and E10 produce 19.12%, 23.12%, and 34.77% higher CO, respectively, as compared to B5. At the higher engine load condition, the temperature is high enough to enhance the benefit of microexplosions, which affect its onset and the strength of the explosion, thus showing significant improvement in CO formation. As compared with the emulsions, SD emits a fair amount of CO, standing between E5 and E10. Other than the effect from the amount of water content in the SD, the lower viscosity of SD plays a significant role in reducing the formation of CO, which helps to improve the fuel atomization and evaporation during the combustion.

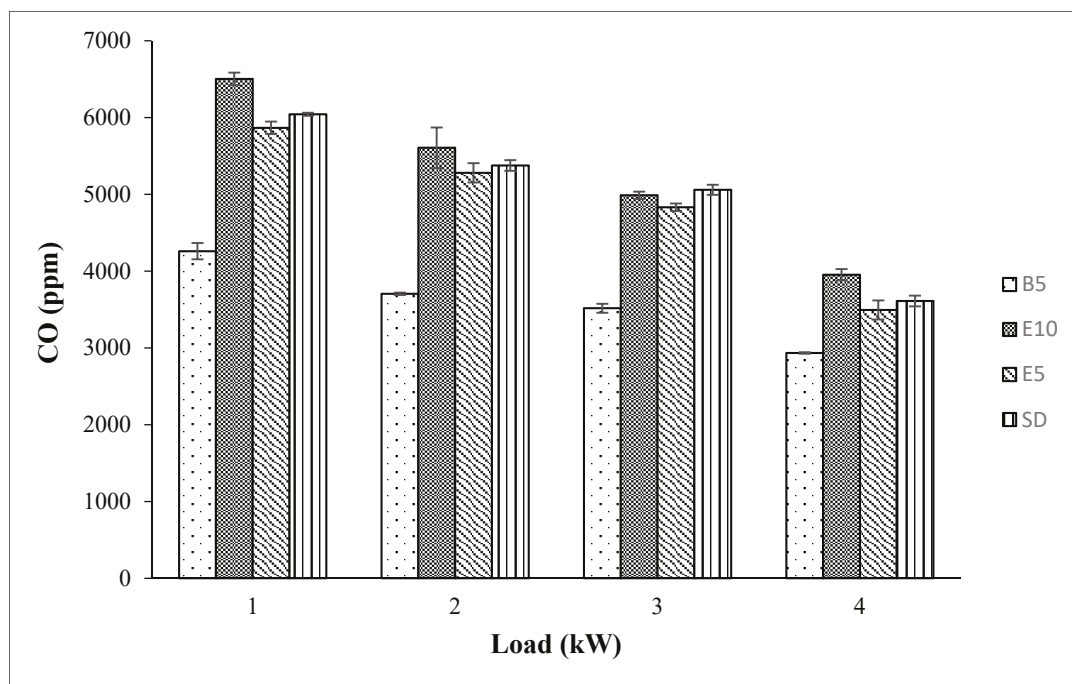


Figure 18. Concentration of CO for B5, E10, E5, and SD under different load conditions.

3.5.3. Particulate Matter

Figure 19 shows the concentration of particulate matter (PM) emissions for B5 and all types of WD at different engine loads ranging from 1 to 4 kW under the constant speed of 2000 rpm. A similar trend is shown in the graph where PM formation for all fuels is reduced as the load is increased, indicating better combustion at higher load. When comparing the emulsions with B5, it is clearly shown that the PM for all types of emulsions is lower. The PM concentration in E10 is recorded to be the

lowest compared to B5 with the average reduction of 24.45%, followed by SD (10.68%), and E5 (8.79%). Again, it is clearly shown that higher water content in the emulsion promotes better combustion and emits lower PM. The reduction of PM in WD can be explained by the occurrence of microexplosions. The secondary atomization promotes the combination of fuel and air, which improves combustion efficiency and suppresses the formation of soot. The reduction of PM can also be attributed to the increase in the amount of hydroxyl (OH) radicals in the presence of water, which leads to the oxidation of soot precursor [21]. The increased concentration of OH radical weakens the formation of soot in the gas phase and restricts the soot growth rates [22]. However, SD still produced considerably lower PM as compared to B5 and E5. This is due to the lower viscosity of the fuel that helps to improve the spray atomization and evaporation. Other than that, the small water droplet size and distribution in SD promotes and enhances the onset and the strength of microexplosions.

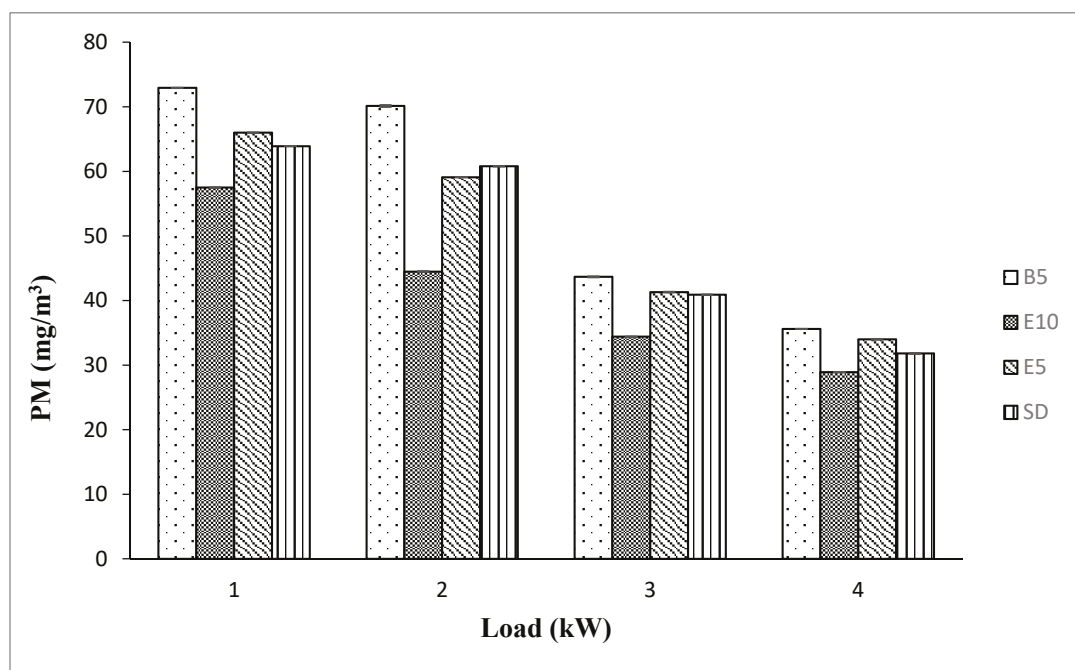


Figure 19. Concentration of PM for B5, E10, E5, and SD under different load conditions.

3.5.4. Carbon Dioxide

The main product of the exhaust emission of a diesel engine is carbon dioxide (CO_2). The formation of CO_2 indicates complete combustion within the diesel engine. In general, incomplete combustion reduces the formation of CO_2 but increases CO concentration [23]. Figure 20 shows the concentration of CO_2 for B5, E10, E5, and SD under different load conditions. At low loads, the concentration of CO_2 for all types of WD is slightly lower than that of B5. Nevertheless, at a higher engine load condition, the formation of CO_2 in WD is slightly higher than that of B5. This indicates that, at higher load, the emulsions combust more efficiently compared to the low load condition. This is supported by the results shown in Figure 18, where the CO formation is reduced significantly as the engine operates at a higher load condition.

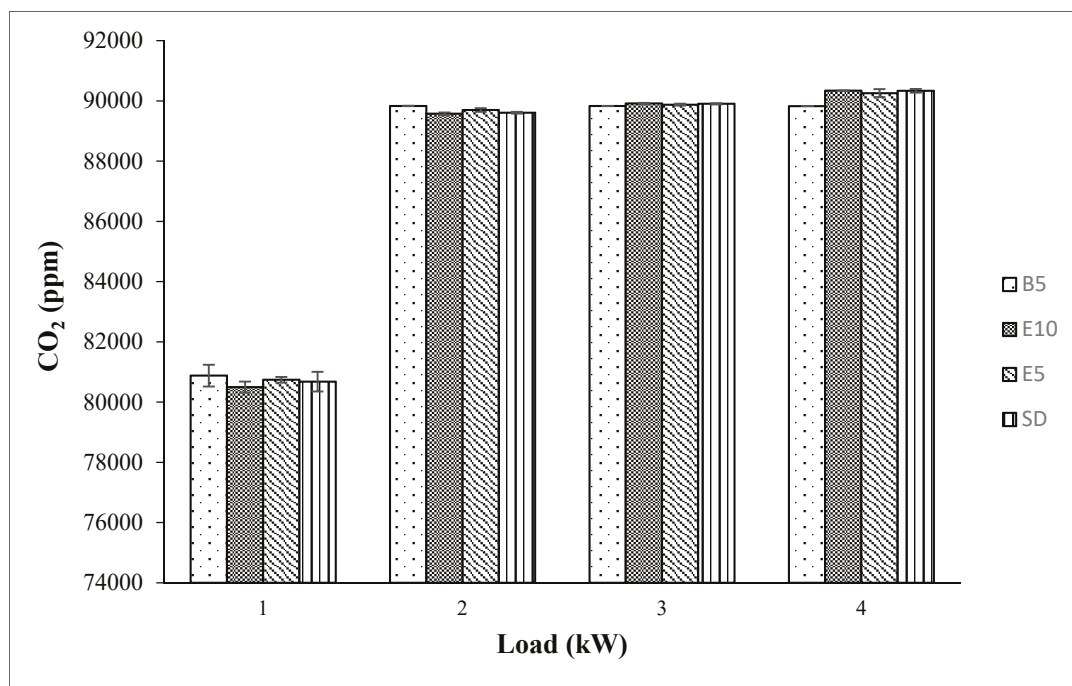


Figure 20. Concentration of CO₂ for B5, E10, E5, and SD under different load conditions.

4. Conclusions and Recommendations

The concept of producing the emulsifier-free water-in-biodiesel emulsion fuel (SD) through the steam emulsification method was proven to work, as demonstrated by this research and elimination of the conventional mixing method. SD was successfully produced by injecting steam through the porous frit glass into the mixing column, which initially was filled with biodiesel. The steam emulsification process took 40 s before the homogenous SD started to form and directly supplied the engine fuel feed system. The average water droplet size of SD is around 0.375 μm with an average water percentage of 6.18%. From the engine performance results, it can be concluded that SD maintained the emulsion fuel benefits towards the environment and energy usage, where BSFC and BTE improved 4.19% and 3.92%, respectively, as compared to neat B5. The ICP was lower than B5, however, yielding close to the B5 at 4 kW engine load. The ROHR was slightly higher than B5 and the SOC shifted closest to B5 at higher engine load. As for the exhaust emission test, NO_x and PM for SD were reduced significantly, with a percentage reduction of 25.22% and 10.68%, respectively, as compared to neat B5. However, the CO emission rises at a low load condition, and is significantly reduced closest to B5 as the load is increased.

Although in this study the emulsion can be produced via the steam emulsification method and comparable with conventional emulsion, the system requires high energy to produce the steam. In the present study, the energy to produce the steam was not considered as part of the load. However, if this energy is considered as the load, the SD would consume more fuel compared to the conventional emulsion. The research on SD is relatively new in the study of emulsion fuel, yet offers a potential to be explored further as the concept offers the alternative method of making emulsifier-free emulsion fuel without the use of the conventional mixers. Therefore, as for future research, the method of producing the steam taken from the engine heat waste can be explored further as it can significantly reduce the energy usage for the heating system.

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Article

Potential for Shock-Wave Generation at Diesel Engine Conditions and Its Influence on Spray Characteristics

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Abstract: Increasing the fuel injection pressure is currently the most effective way to achieve a better fuel–air mixing quality in modern engines. Systems capable of delivering fuels at a pressure of over 250 MPa have been widely adopted in diesel engines. At such high injection pressures, the shock-wave generation during fuel injection has been noticed. Investigations can be found widely discussing on how the shock-wave generation during fuel injection would affect the spray dynamics. However, the argument remains whether the shock wave can occur at diesel engine conditions since the diesel engine is operated at very high ambient temperature and density. Even if it could occur, how significantly the spray-induced shock wave affects the spray characteristics is rarely known. To address these concerns, this study was proposed. First, experiments were conducted to obtain the detailed spray dynamics from the nozzle exit to spray downstream field by taking advantage of the X-ray phase-contrast imaging (XPCI) and schlieren imaging techniques. It is found that supersonic and subsonic ligaments coexist in one spray. Increasing the injection pressure or reducing the ambient density would extend the supersonic part in the spray. Multiple shock waves occur subsequently from the nozzle exit, where the spray has the highest local velocity. Shock-wave generation during fuel injection could enhance spray penetration, whereas this effect depends on the length of the supersonic part in the spray. Finally, a diagram was proposed to predict the potential for the shock-wave generation and discuss the possible effect on spray characteristics at diesel engine conditions.

Keywords: diesel spray; shock-wave generation; nozzle-exit velocity; X-ray phase-contrast imaging; schlieren imaging; diesel engine condition

1. Introduction

Fuel injection systems in modern engines have evolved towards higher injection pressures to achieve a better quality of mixture combustion. With the elevated fuel injection pressures, sprays possess a high level of turbulence, which can enhance the air entrainment and fuel breakup [1–3]. On the other hand, the higher injection pressure results in faster spray penetration that can improve air utilization and combustion speed [4]. Systems that are capable of delivering the fuel at an injection pressure of over 250 MPa have been widely adopted in diesel engines, while the ones in development are being tested to endure an injection pressure of 300 MPa or even higher [5–7].

The continuous increase in fuel injection pressure results in the generation of supersonic spray, which can induce the shock wave during fuel injection. Nakahira et al. [8] first noted the possible

existence of the spray-generated shock waves in their experiment using an optical diesel engine. In the subsequent studies, the characteristics of the spray-generated shock waves have been examined in many aspects. By using the X-ray radiograph measuring technique, MacPhee et al. [9] found that there exists an average of a 15% variance in the gas density across the leading shock front. Afterward, a numerical model was proposed by Im et al. [10] to interpret the underlying mechanism reported by MacPhee et al. Based on an experiment under the schlieren imaging setup, Song et al. [11] reported that the spray-generated shock waves have a positive effect on the development of the spray tip penetration and mass of the entrained air. Huang et al. [12,13] submitted similar studies to Song et al.'s. Additionally, using the schlieren imaging setup, Li et al. [14] studied the effect of the reflected shock waves on spray characteristics and claimed that the reflected shock waves significantly suppress the development of the spray in the radial direction, while it barely affects the development of the spray tip penetration. Jia et al. [15,16] reported that there exists a gradually decreasing tendency in the peak velocity of the sequentially-generated shock waves, which is thought to be because of the weakened wave intensity caused by energy loss. Most recently, Salvador et al. [17] proposed a model to scale the spray tip penetration at supersonic conditions. It is noted that the studies aforementioned were mostly focusing on the characteristics of the leading shock wave and its effect on the spray. However, it is clearly observed in all the relevant investigations that multiple shock waves are generated sequentially during the fuel injection. Discussions are still missing at present to understand how the multiple shock waves are generated and how they affect the spray as a whole. This shortage is likely because detailed spray-dynamics measurements are still missing, which encountered a significant challenge, especially at the near-nozzle spray field, due to the incapability of the conventional measurement techniques.

Another important subject that deserves careful consideration is the potential for the shock-wave generation during fuel injection at diesel engine conditions. As it is well known, the ambient temperatures and pressures in the combustion chamber are critical. High ambient temperature increases the sound speed of ambient mixture, while high ambient density and pressure result in a fast spray deceleration. Both effects suppress the shock-wave generation during fuel injection. To the best of the authors' knowledge, the most relevant investigation of this subject was reported by Kook et al. [18], which suggested that spray-generated shock waves may not be expected at injection timings typical of a diesel engine. However, their discussions did not take full account of the spray dynamics in the near-nozzle region due to a lack of experimental results, which is found to vary significantly from those in the downstream spray field [19,20].

Regarding the increasingly powerful fuel-injection systems in development, to predict the shock-wave generation at diesel engine conditions has become of great significance. Additionally, it deserves further classification as to how significantly the spray-induced shock wave can affect spray characteristics once it occurs. To address this purpose, first, experiments were conducted to obtain the detailed spray dynamics from the nozzle exit to spray downstream field by taking advantage of the X-ray phase-contrast imaging (XPCI) and schlieren imaging techniques. The XPCI is an incomparable technique that ensures a direct measurement of the near-nozzle spray dynamics. This method has been proved valid through many previous investigations [21–24], but for the first time, it was used to examine the spray dynamics in a supersonic-spray point of view. The shock-wave generation during fuel injection was observed using a schlieren imaging setup, which is most typically used to examine the shock-wave characteristics. Afterward, the results from the two separate experiments were linked to unveil the relationship between near-nozzle spray dynamics and multiple shock-wave generations. Then, discussions were extended to evaluate the potential for the shock-wave generation and the possible influence at diesel engine conditions.

2. Description of Experiments

2.1. Experimental Setup

2.1.1. X-ray Phase-Contrast Imaging (XPCI) Technique

In this study, the spray velocities in the near-nozzle region were examined using the XPCI technique at the 7ID-B station at Advanced Photon Source (APS). First, a brief introduction to the mechanism of XPCI will be presented. In the X-ray regime, the refractive indices are lower than but very close to 1. This feature is critical in terms of the fuel spray imaging because it prevents the multiple scattering that happens using the conventional light sources. Owing to the nature of X-ray, X-ray imaging can observe directly the liquid-phase core veiled by the dense spray droplets cloud, especially in the near-nozzle region. When X-rays propagate through an object, the energy absorption and phase shift of X-rays occur. Regarding the soft-material object, like fuel spray, the X-ray phase-shift after propagating through the object is generally three orders of magnitude larger than its energy attenuation. In terms of this fact, the XPCI technique is particularly effective to image the primary spray breakup in the near-nozzle field with highly temporal resolution.

The XPCI experimental setup is illustrated in Figure 1. The storage ring is a circular tube where the electrons circle for hours whose speed is close to the speed of light. As the electrons travel around the ring, they pass through different types of magnets, and in the process, they produce X-rays. A mechanical shutter is settled in the beam path to ensure X-rays can enter the imaging system only at the imaging instant (8 ms opening duration). This is necessary to protect the imaging system from the heat-load of X-rays. The X-ray phase-shift cannot be imaged directly by conventional optical lenses. Instead, a scintillator crystal (LuAg:Ce) is used to convert the X-ray phase-shift into the intensity variations of visible light. The visible-light image is reflected by a 45° mirror and captured using a high-speed camera (Model SA-Z, Photron Limited., Tokyo, Japan). The imaging specifications were fixed at a frame rate of 67,889 Hz (one-fourth of synchrotron revolution frequency), an image resolution of 512×512 , and a spatial resolution of $2.54 \mu\text{m}/\text{pixel}$.

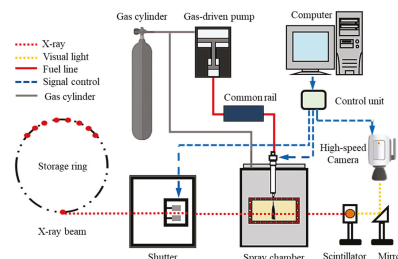


Figure 1. Experimental setup for the X-ray phase-contrast imaging.

XPCI technique can extract the spray velocity in the near-nozzle field. The calculation method is described below. In the Hybrid-Singlet beam mode of APS, an irregular pulse pattern is generated with a periodicity of $3.682 \mu\text{s}$. Figure 2a shows the details of this pulse pattern, which consists of a single-bunch with a 150 ps duration and a 16 mA current isolated from the remaining eight groups of seven consecutive bunches (eight septets) with 11 mA current per group and periodicity of 68 ns. By appropriately adjusting the timings of the camera shutter gate and the fuel injection relative to the X-ray signal, two septets of X-rays can be imaged in one single frame. In other words, an X-ray double-exposed image is obtained, as shown in Figure 2b. Then, the spray velocities are extracted from the images under an analysis resembling the particle imaging velocimetry (PIV). To be specific, the auto-correlation calculation is performed in the region of interest (ROI), which detects the displacement vector of the imaged features of spray during the 68 ns time interval, as shown in Figure 2c. Different from the PIV technique that uses the droplets as tracers, the XPCI uses the instability-wave features in the spray as tracers. Due to the line-of-sight nature of XPCI, all features of spray along the beam path are recorded in one frame. The velocity result obtained from XPCI is

believed to present the velocity of the largest features in the beam path located in the local spray center. Detailed explanations can be found in our previous publication [21–24].

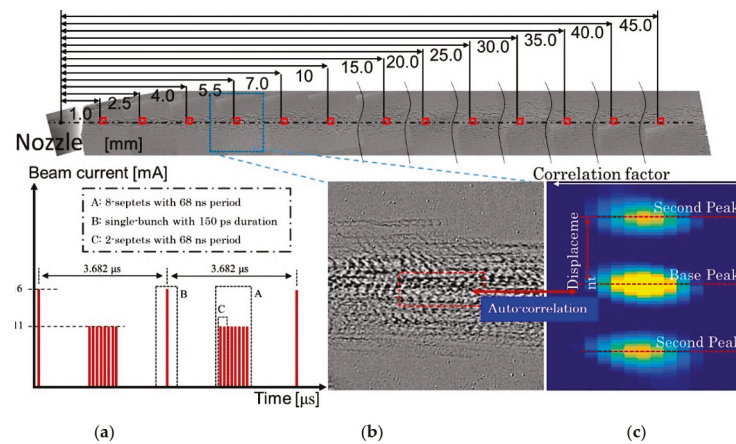


Figure 2. Spray velocity calculation based on a double-exposed image of X-ray imaging. (a) X-ray bunch mode; (b) double-exposed image (c) auto-correlation calculation to derive spray velocity.

2.1.2. Schlieren Imaging Technique

Shock waves are a type of propagating disturbance that can be characterized by an abrupt, nearly discontinuous change in pressure, temperature, and density of the medium. According to its physical nature, shock waves are observed typically using a schlieren imaging setup, which appears like the interference patterns in images. Figure 3 shows schematically the schlieren imaging setup used in this study. The entire test rig consists of a common rail injection system, a control unit, a constant volume vessel, and a LED-based imaging system. The schlieren images of shock waves were captured using the same high-speed camera (Model SA-Z, Photron Limited, Tokyo, Japan) as that in the XPCI experiment, and the image specifications were decided as a frame rate of 200,000 Hz, an image resolution of 384×176 pixels, and a pixel size of $130 \mu\text{m}$. To the best of the authors' knowledge, these extremely high imaging specifications cannot be found in relevant investigations. By making good use of the high-performance of the high-speed camera, observing the detailed characteristics of the multiple shock waves during fuel injection was expected.

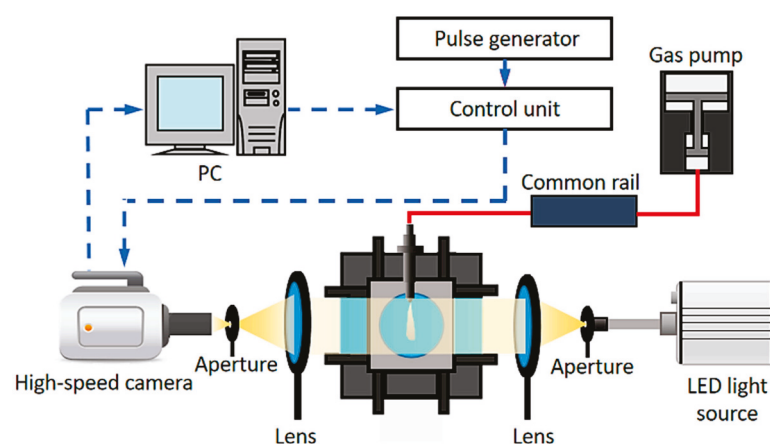


Figure 3. The schlieren imaging test rig.

A set of MATLAB codes was created to enhance the contrast of the shock-wave images according to [18]. Figure 4 shows the sampling images before and after image processing with different post-processing methods. In method A, the spray image was divided by an image before the injection event, whereas in method B, the spray image was divided by the previous-frame image. It is

considered that Method B works better to distinguish shock waves from the nonuniform background. This nonuniform background created by refractive-index gradients would become even more severe at the higher gas temperature and density conditions. To address this issue, method C is proposed that combines the post-processed images by methods A and B to illustrate the spray and shock waves both clearly. As a result, method B and method C were mainly used in this study to observe the shock-wave generation during fuel injection.

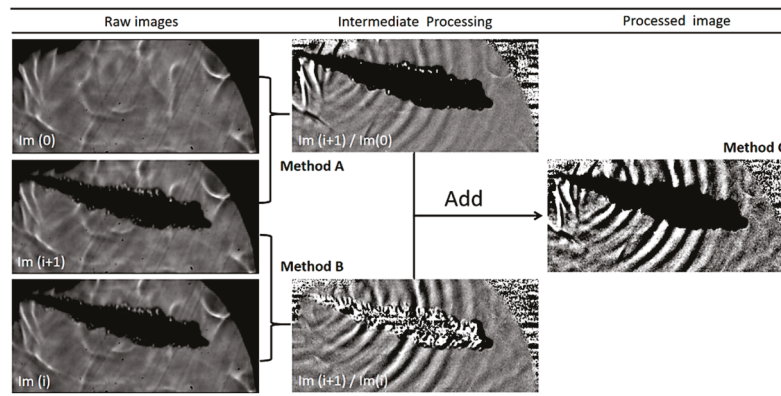


Figure 4. Images processing method. From the left, the raw spray images of $Im(0)$, $Im(i)$, and $Im(i + 1)$, the intermediate-processing images, and the processed image are shown. The raw images were obtained under an ambient-gas temperature of 500 K and a density of 6.3 kg/m^3 .

The spray tip penetration (S) was used to characterize the spray characteristics in the schlieren imaging experiments. A set of MATLAB codes shared from Engine Combustion Network (ECN) was employed for the parameter calculations. Detailed introductions to these codes can be found on its official website [25] and a previous publication [26]. In the calculation codes, the penetration is defined as the distance along the spray axis to a location where $1/2$ of the pixels on an arc of $\theta/2$ centered on the spray axis are dark. The spray angle, θ , is defined by the following relationship:

$$\frac{\theta}{2} = \tan^{-1} \left(\frac{A_{p, S/2}}{S/2^2} \right) \quad (1)$$

where $A_{p, S/2}$ is the projected spray area of the upstream half of the spray in an image. The definition of the spray parameters is shown schematically in Figure 5. It is found that using the current code can extract the spray boundary well and ensures to obtain the spray parameters in high accuracy.

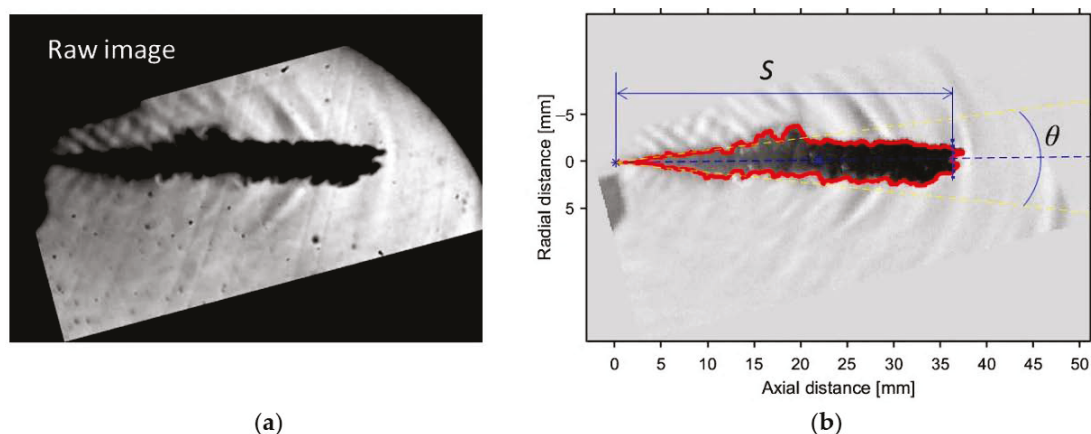


Figure 5. Definition of spray tip penetration (S) and spray angle (θ): (a) Raw image of spray; (b) definition of spray tip penetration and spray angle.

2.2. Experimental Conditions

In previous investigations, single-hole nozzles are extensively used to investigate the spray-generated shock waves in favor of the simplified optical diagnostics. However, in practice, the single-hole nozzles are rarely adopted in engines because they are incapable of injecting the required fuel mass within a limited duration. To achieve consistency with the usage in engines, a commercial multiple-hole nozzle was used in this study, which was capped with a spray blocker. Using the spray blocker can assure only the targeting spray of interest passing through it and prevent the multiple spray plumes from obstructing the imaging. In the XPCI experiments, the ambient gas was fixed using N_2 . It is known that the spray dynamics are sensitive to the ambient gas density rather than the type of ambient gases, whereas N_2 can provide a high-level phase contrast of X-ray due to its nature of a small-sized molecular. At every condition, the measurement was repeated 10 times and the averaged data were used for the analysis. In the schlieren imaging experiments, N_2 , CO_2 , and SF_6 were used as the ambient gas to obtain three distinct sound-speed conditions. The sound speed in an ideal gas is attainable using the following relationship [18].

$$C = \sqrt{\frac{\gamma \cdot R \cdot T}{M}} \quad (2)$$

where R is the universal gas constant, and T is the temperature of the gas. The gas properties are described by γ the specific heat ratio, M the molecular mass. The sound speed of a mixture gas depends on chemical constituents and gas temperature. Then, Equation (2) will contain the sum of each gas constituent, γ_i , and M_i , and x_i is the fraction of the i constituent [27].

$$\gamma = \sum_i^N x_i \cdot \gamma_i, \quad M = \sum_i^N x_i \cdot M_i \quad (3)$$

It is noted that the spray-generated shock-wave has been rarely examined at high ambient-temperature and the mixture-gas conditions, in which engines work typically. To address this shortage, a high-temperature with the gas-mixture condition was included in this study. The experiment firstly used a premixed gas mixture to generate high temperature and pressure inside the spray chamber. The reactant constituents of the premixed gas mixture have 5.9% C_2H_4 , 35.5% O_2 , and 58.6% CO_2 in mole fractions, while the equilibrium combustion products consisted of 12% H_2O , 18% O_2 , and 70% CO_2 in mole fractions. The gas temperature (T_c) after a premixed burn was estimated based on a relationship proposed by Naber et al. [28], which involves the chamber wall temperature (T_w), and the mass averaged bulk temperature (T_b).

$$\frac{T_c}{T_b} = 1 + a \left(1 - \frac{T_w}{T_b} \right) + b \left(\frac{T_b}{T_w} - 1 \right) \quad (4)$$

$$a = 0.108 \left(\frac{\rho}{\rho_{ref}} \right)^{-0.295}, \quad b = \frac{V_c}{V} \quad (5)$$

$$T_{b,n} = \frac{P_n}{P_{n-1}} T_{b,n-1} \quad (6)$$

where T_w was fixed at 378 K in the current experiment. The term a is an empirical parameter and was found to be dependent on density (ρ), varying with density to the 0.295 power ($\rho_{ref} = 20.4 \text{ kg/m}^3$). The term b is a constant equal to the ratio of the chamber crevice volume (V_c) to the total chamber volume (V). A value of $b = 0.05$ was used in the current experiment according to an estimation of the spray chamber. The chamber pressure, P_n , was monitored using a pressure sensor (Kistler 6125B) coupled with a Kistler model 5011B charge amplifier at a sampling rate (n) of 100 kHz. Before the formal experiment, several trying-out experiments were necessary to obtain the history of average

chamber pressure. Then, chamber-pressure results (P_n) were imported to calculate the bulk temperature (T_b) and the gas temperature (T_c). Afterward, in the formal experiment, fuel was injected at a targeting ambient-gas condition, which was at 500 K and 6.3 kg/m³ in this study, and the results were used further for discussions. At every condition, the measurement was repeated five times and the averaged data were used for the analysis. Experimental conditions are summarized in Table 1.

Table 1. Experimental conditions.

Injector Specifications				
Type	Diesel Solenoid Injector			
Nozzle	Eight holes			
Nozzle diameter	0.11 mm			
XPCI Experiments				
Ambient gas	N ₂			
Ambient condition	1.1, 9.0, 17.7 kg/m ³ at 297 K			
Injection pressure	65, 135, 160 MPa			
Injection duration	2 ms			
Schlieren Imaging Experiments				
Ambient gas	N ₂	CO ₂	SF ₆	Mixture: 12% H ₂ O; 18% O ₂ ; 70% CO ₂
Ambient condition	9.0 kg/m ³ at 297 K			6.3 kg/m ³ at 500 K
Injection pressure	30, 40, 50, 65, 135, 160 MPa			65, 80, 100, 135 MPa
Injection duration	1 ms			

3. Results and Discussion

3.1. Spray Velocity Characteristics

Figure 6 shows the spray velocities at the nozzle exit. These results were measured in the XPCI experiments. M_a denotes the sound speed of N₂, equaling to 352 m/s at the experimental conditions. It is found that the nozzle-exit spray velocities at the injection beginning are subsonic regardless of the injection pressure. Afterward, spray velocities rise quickly with time and become supersonic in case of the 135 and 160 MPa injection pressures. The injection pressure of 65 MPa seems to be a transition condition that turns the spray velocities from subsonic to supersonic.

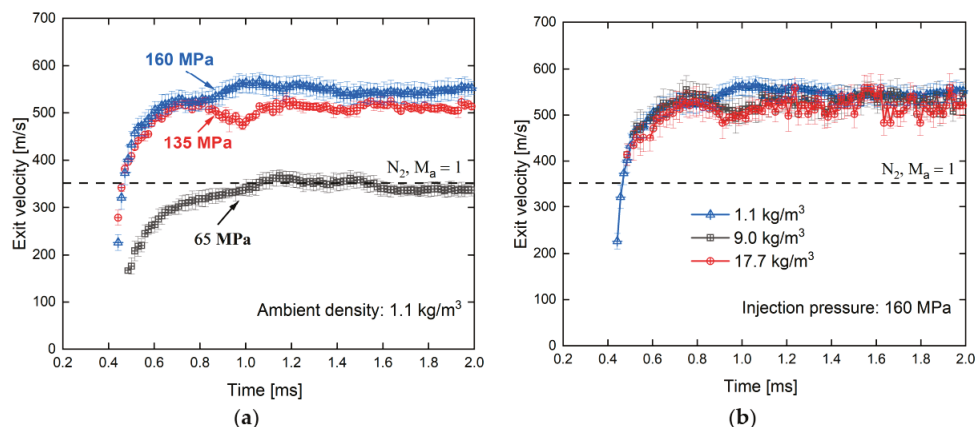


Figure 6. Spray velocities at 2 mm from the nozzle exit. (a) Injection pressure effect; (b) ambient density effect. M_a denotes the sound speed of gas. (Ambient gas condition: N₂ and 297 K; Local sound speed: 354 m/s).

Spray velocities at the nozzle exit remain constant when the ambient density increases from 1.1 to 17.7 kg/m³, as shown in Figure 6b. According to the Bernoulli principle [29], the spray velocity at the nozzle exit is proportional to the root of the pressure drop across the nozzle hole, i.e., the differential pressure between the sac and ambient gas. In the diesel-injection scenario, the ambient-gas pressure is a hundred times smaller compared to the sac pressure. Thus, although the ambient pressure was increased to obtain high ambient densities, the pressure drop across the nozzle hole was almost unchanged. As a result, the phenomenon that spray velocities at the nozzle exit are insensitive to the ambient density can be understood.

Figure 7 shows that the spray axial velocities in the center of spray. Results were derived from the XPCI experiments, and they were the time-averages within the steady-state (1.0–2.0 ms). It is found that spray velocities decrease gradually in the center of spray with distances. The higher ambient density causes a faster deceleration of spray velocities. When using an injection pressure of 65 MPa, spray velocities are subsonic except those in the near-nozzle region, whereas they are mostly supersonic at 160 MPa except in the downstream spray field. These results indicate that sprays can have a supersonic core at the high-injection pressure conditions regardless of the ambient density. Moreover, it is of particular interest that there is a state that supersonic and subsonic ligaments coexist in one spray. Increasing the injection pressure or reducing the ambient density would extend the supersonic part in the spray.

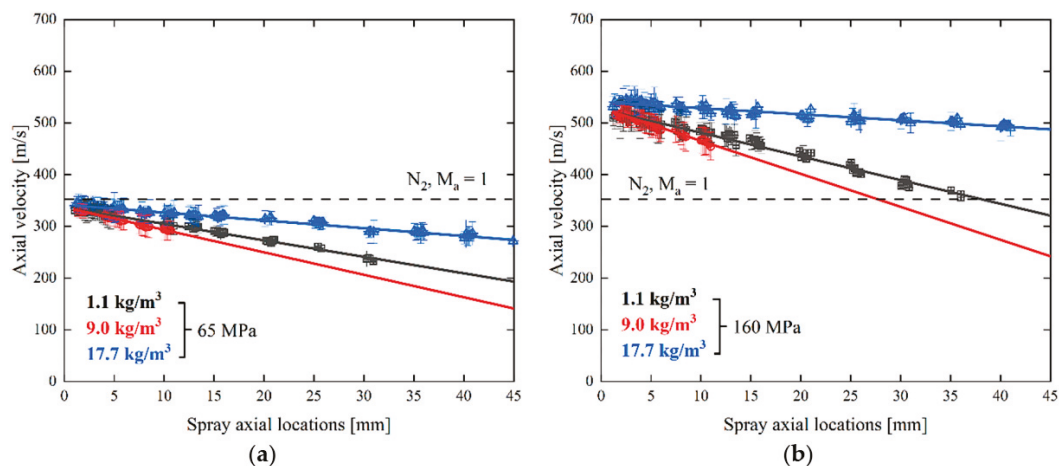


Figure 7. Spray axial velocities under various ambient densities. (a): 65 MPa; (b): 160 MPa. (Ambient gas condition: N₂ and 297 K; Local sound speed: 352 m/s).

Next, the spray tip velocities derived from the schlieren imaging experiments are examined. The spray tip velocity is one essential parameter that has been extensively used to characterize the spray-generated shock waves. As shown in Figure 8, the maximum of the spray tip velocity is found to be within 100–250 m/s under the injection pressures ranging from 30 to 160 MPa. It is worth noting that the experiments using CO₂ and SF₆ have a lowered charging pressure of the spray chamber due to their larger molecular mass compared to N₂. This was to keep identical ambient densities. However, no apparent difference in spray tip velocity is found among using various ambient gases, which indicates that the spray tip velocities are insensitive to the types of ambient gases and the charging pressures. Furthermore, it is noted that the spray tip velocities are subsonic except the ones in the SF₆ experiments.

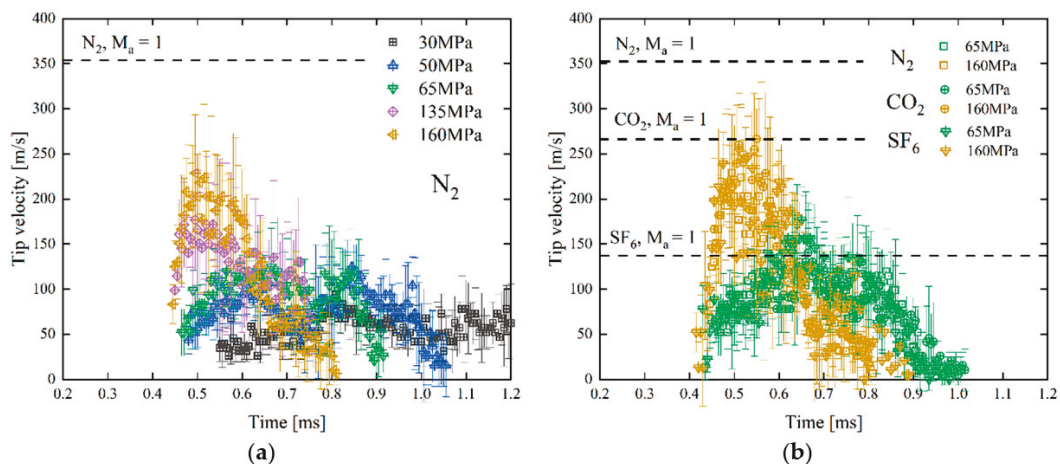


Figure 8. Spray tip velocities under different injection pressures: (a) spray tip velocities at various injection pressures; (b) spray tip velocities against various ambient gases. (Ambient gas condition: 297 K and 9.0 kg/m³; Local sound speed: 352 m/s at N₂; 267 m/s at CO₂; 135 m/s at SF₆).

3.2. Shock-Wave Generation during Fuel Injection

The spray images from the experiments at an injection pressure of 65 MPa are selected to reveal the generation of spray-induced shock waves, as shown in Figure 9. Spray velocities at the nozzle exit (from the XPCI experiments) and spray tip (from the schlieren imaging experiments) are presented as well. In the experiment of SF₆, the nozzle-exit spray velocity is supersonic at the injection beginning. Accordingly, the multiple-shock-wave generation appears. The spray tip velocity rises with time and becomes supersonic from timing 2 when the shock wave attaches to the spray frontier. At timing 3, the spray tip velocity drops to be subsonic again. Simultaneously, the shock wave is found to detach the spray frontier. Subsequent shock waves keep generating from the nozzle exit. The one that appears in a later timing seems to have a sharper angle. This can be explained by the fact that spray velocities at the nozzle exit are increasing from timing 1 to timing 4. Larger spray velocity results in a sharper shock-wave angle. The shock-wave angle is otherwise known as the Mach angle, which is proportional to the speed of a supersonic object [30].

In contrast to that of SF₆, the spray tip velocities in the experiment of N₂ and CO₂ are subsonic during the entire injection, whereas the nozzle-exit spray velocities do not reach supersonic until timing 5 and timing 2, respectively. Interference patterns near the nozzle exit are observed on timing 5 and timing 2's images, which represent the generation of shock waves. When comparing the spray characteristics at various ambient gases, it is noted that the spray in the experiment of SF₆ has a faster spray penetration and narrower spray angle compared to those in N₂ and CO₂. However, this variance of spray characteristics seems not to appear until sprays have penetrated downstream, i.e., until timing 4. Apparent changes in spray characteristics are not observed before and after the shock-wave generation. More discussions on the shock-wave effect on spray characteristics are presented later.

It might be notable that some weak waveforms are appearing in the timing 2 and 3's images of the N₂ experiments. Those waveforms are acoustic pressure waves rather than shock waves. The reason lies in the fact that shock waves represent a sudden and violent change in stress, density, and temperature. Owing to its nature, shock waves are illustrated as bright interference patterns in schlieren images. However, the waveforms in timing 2 and 3 have a quite similar intensity with the image background. The difference can be easily distinguished in images. As a result, it is believed that those waveforms are acoustic pressure waves, which also occur even if the spray is subsonic. Similar discussions can be seen in [16].

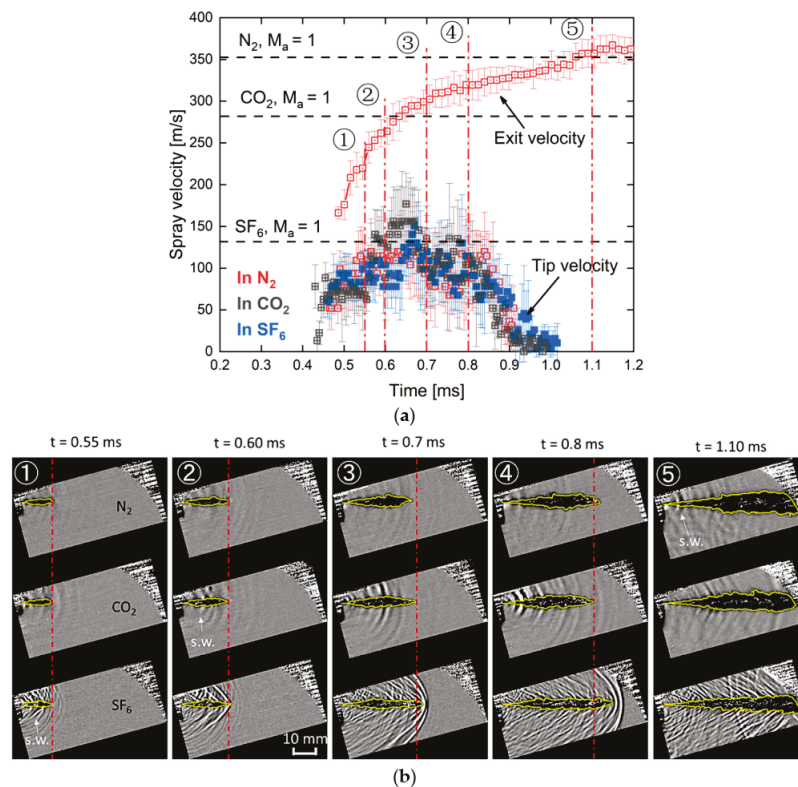


Figure 9. Spray velocity (a) and the shock-wave generation vs. time (b); ambient gas conditions: 297 K and 9.0 kg/m^3 ; injection pressure: 65 MPa. ‘s.w.’ stands for the shock wave. The red dash-dot line represents the spray leading edge of CO_2 .

It is worth noting that in the experiment of N_2 , shock waves appear so late that the spray tip penetration has exceeded 45 mm. Besides, shock waves in this condition are observed only in the near-nozzle region, and their intensity, i.e., the gray level in images, attenuates fast as it propagates along the spray axial direction. Figure 10 shows how the spray axial velocity and shock waves change in the center of spray. It is found that the shock-wave intensities increase slightly first from the nozzle exit and then sharply drop after certain distances from the nozzle exit. The locations where the strong shock waves disappear are close to the positions that spray velocities change from supersonic to subsonic. In other words, these results reveal that once shock wave detaches spray, its intensity reduces quickly.

Figure 11a shows the spray tip penetration and spray angle at various conditions. These results are the averaged results from five repetitions. It is noted that the spray tip penetration in SF_6 (160 MPa) is approximately 23.5% higher than those in N_2 near the observing limit ($T_2 = 0.65$ ms). The spray characteristics in CO_2 are between those in N_2 and SF_6 . This resulting trend is consistent with previous reports [11,17], which have discussed that this behavior is related to the density gradient appearing across the shock-wave. However, it is worth noting that the previous reports mainly discussed the spray characteristics and its link with the spray-frontier shock wave. Owing to an extraordinary high frame rate applied in the current experiment (four times of that in [11] and five times of that in [17]), this study noted that the variance in spray characteristics further relates to the supersonic-part length in the spray. Evidence can be found in the images to support this argument. Figure 11b shows a sequence of spray images at 160 MPa. The spray penetrations of three ambient gases remain identical at $T_1 = 0.55$ ms. T_1 represents a time that the shock wave is detaching from the spray frontier in the N_2 condition. Shortly after T_1 , the spray penetration in SF_6 and CO_2 exceeds that in N_2 . Then, at $T_2 = 0.60$ ms, the shock wave is detaching from the spray frontier in the CO_2 condition. Simultaneously, the spray

penetration in SF_6 starts to exceed that in the CO_2 condition, and the variance keeps enlarging until $T_4 = 0.65$ ms.

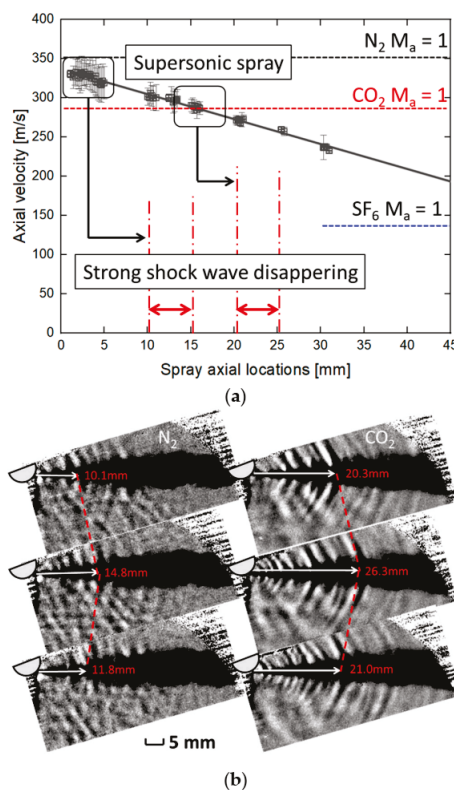


Figure 10. Spray velocity in axial axis (a); the shock-wave generation at steady state (b) and images timings from top to bottom: 1.100, 1.125, 1.150 ms; ambient gas conditions: 297 K and 9.0 kg/m^3 ; injection pressure: 65 MPa.

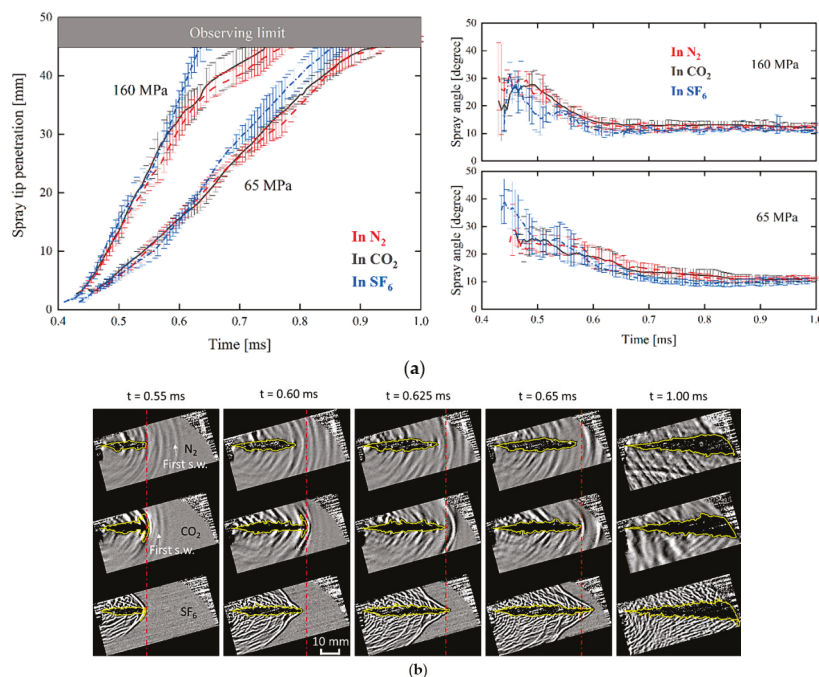


Figure 11. Shock wave generation effect on the spray penetration and spray angle (a), and a sequence of spray images at 160 MPa (b). Ambient gas conditions: 297 K and 9.0 kg/m^3 . The red dash-dot line represents the spray leading edge of CO_2 .

During the fuel injection process, fuel ligaments are injected sequentially from the nozzle, while the supersonic ones may induce shock waves. As the ligament velocities in the spray are low at the injection beginning, the first-appearing shock wave is emitted from the spray frontier shortly, as seen the 'First s.w. (shock wave)' indicated in the image. As the needle further opens, the injected ligaments have an increasingly higher velocity until reaching a steady state. These ligaments propagate downstream and substitute the frontal ligaments. Thus, the velocity of the spray frontier keeps increasing, and the supersonic-part length of ligaments in the spray extends. Once the ligaments at the spray frontier obtain a supersonic velocity, the shock wave remains attached to the spray frontier.

On the other hand, the shock wave is induced continuously from the nozzle exit, and these subsequent shock waves are observed to merge to the spray frontier, which is more evident by watching the animation of the spray images. It is worth noting that while there exists a shock wave remaining at the spray frontier, the spray has a sharp tip shape, which behaves like the ligaments at spray frontier are dragged forward by the attached shock wave. This is probably explainable in terms of the fact that there exists a low-density zone behind the shock wave. Multiple shock waves are generated during the fuel injection, whereas a similar process should repeat for every supersonic ligament. Thus, the longer the supersonic-part length, the further the ligaments have been boosted to penetrate. As a result, the fastest spray tip penetration was observed in SF₆, where the entire spray is supersonic, and the supersonic-part length exceeds 45 mm.

3.3. Potential for Shock-Wave Generation at Diesel Engine Conditions

In this section, the potential for shock wave generation and its possible influence at diesel engine conditions are being considered. First of all, the results above indicate clearly that the multiple shock waves during fuel injection are induced from the nozzle exit. The spray velocities at the nozzle exit (V_{exit}) in various injection pressures are derived according to Bernoulli's principle (Equation (7)).

$$V_{exit} = C_v \times \sqrt{\frac{P_{in} - P_b}{\rho_f}} \quad (7)$$

where P_{in} , P_b , and ρ_f are the injection pressure, ambient gas pressure, and fuel density, respectively. C_v is the velocity coefficient, which stands for the velocity decrease from the theoretical value due to the energy loss when the fuel is passing through the nozzle hole. Based on the experimental results, it is found that the nozzle used in this study has a C_v of 0.89 approximately. The effect of gas density has not been considered in this calculation because it barely affects the spray velocities at the nozzle exit. Finally, the predictions of the spray velocities at the nozzle exit can be obtained, as shown in Figure 12a. It is known that the supersonic sprays occur at the nozzle exit when the injection pressures approximately exceed 65, 40, and 10 MPa in the experiments of N₂, CO₂, and SF₆, respectively.

Figure 12b shows the images showing the shock-wave generation in various injection pressures. It is found that shock waves appear in the experiments of N₂ and CO₂ from the injection pressures of 65 and 40 MPa, respectively. This result matches well with the predictions shown in Figure 12a. In the experiments of SF₆, shock waves appear from an injection pressure of 30 MPa, which is the lowest injection pressure enabled by our fuel injection system. As predicted in Figure 12a, a 10 MPa injection pressure would be enough to induce the shock wave during fuel injection. Therefore, the shock-wave generation at 30 MPa of the SF₆ experiments is reasonable.

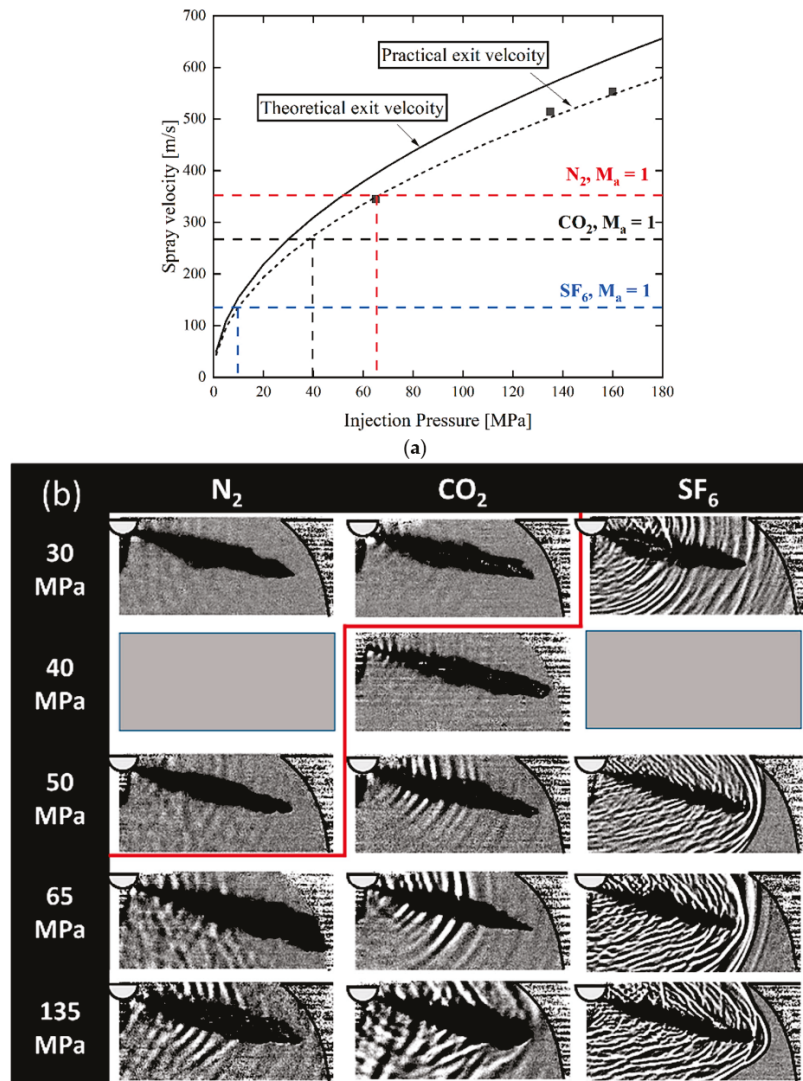


Figure 12. Spray velocity at the nozzle exit (predictions) and the maximum of spray tip velocity (experiments) in various injection pressures (a); shock-wave generation vs. injection pressure; ambient gas condition: 297 K and 9.0 kg/m^3 (b).

Moreover, the shock-wave generation during fuel injection was examined in a high-temperature and gas-mixture condition. First, a premixture was burned to increase the ambient temperature inside the spray chamber. Then, fuel was injected into the after-combustion mixture at a targeting ambient temperature, which was 500 K in this study. The equilibrium combustion products consisted of 12% H_2O , 18% O_2 , and 70% CO_2 in mole fractions. The sound speed of the after-combustion mixture can be calculated using Equations (1)–(3). It equals to 371 m/s at the experimental condition. Meanwhile, the sound speed of the after-combustion mixture is also attainable according to the shock-wave propagating speed in the medium. As seen in Figure 13a, these two shock-wave velocity results are very close, which indicates excellent reliability of current validations.

On the other hand, it is predictable using Equation (7) that a minimum injection pressure of approximately 80 MPa is needed to generate shock waves at the current combustion condition. This prediction has been validated, the results of which are shown in Figure 13b. It is confirmed that shock waves can be observed using an injection pressure of 80 MPa. Thus, it is still feasible to predict the shock-wave generation during fuel injection by examining whether the spray velocities at the nozzle exit can reach supersonic. It might be of concern that at diesel engine conditions, the ambient temperatures of typical injection timings are higher than 500 K. However, the schlieren-image background is so

much disturbed by the high-temperature gas that the shock waves can barely be distinguished. More discussions are presented in a later section based on an analytic calculation.

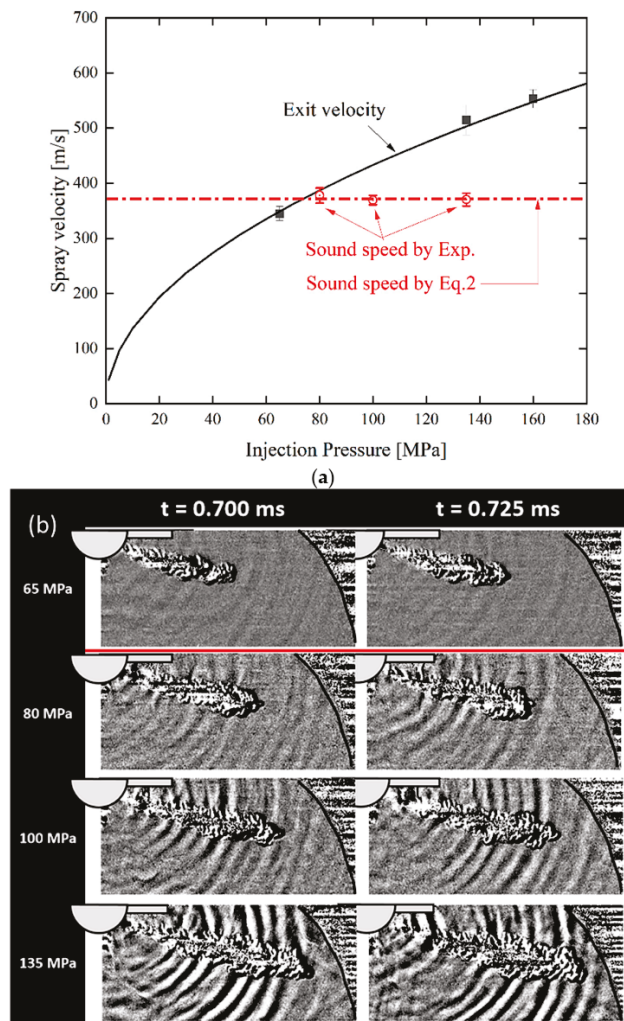


Figure 13. Shockwave generation in the high ambient temperature condition (a); ambient gas condition: mixture (12% H₂O; 18% O₂; 70% CO₂), 500 K and 6.3 kg/m³ (b).

The possible supersonic-part length in the spray deserves careful consideration because it has been proposed that the longer the supersonic part in the spray, the larger the variance exists in spray characteristics. Based on the XPCI results, the spray-center velocity at different axial locations $V(s)$ can be estimated using Equation (8)

$$V(s)/V_{exit} = k \cdot S/d_{eq} \quad (8)$$

where $d_{eq} = d \cdot \sqrt{\rho_f/\rho_a}$, and d is the nozzle hole diameter. ρ_f and ρ_a are fuel density and air density, respectively. This relationship originates from the gas jet theory, which has been modified to suit the scenario of the near-nozzle spray dynamics [31,32]. Equation (8) can be converted to an expression linking the local spray velocity (V) with the Bernoulli injection velocity (V_b), i.e., the theoretical velocity under an injection pressure, by combining Equations (7) and (8).

$$V(s)/V_b = C_v \cdot k \cdot S/d_{eq} \quad (9)$$

Based on the results in Figure 6, it is known that the nozzle currently used has a C_v of 0.89 approximately. Then, the spray velocity in different axial locations can be replotted against d_{eq} , as shown in Figure 14. It is clear that the spray velocity decelerates along the axis at an identical slope, k , regardless of the conditions. Moreover, $V(s)/V_b$ at $S = 15, 25$, and 45 mm can be known in terms of various ambient densities, as shown in Figure 14b. The further the downstream location, and the higher the ambient density, the lower the local spray velocity exists.

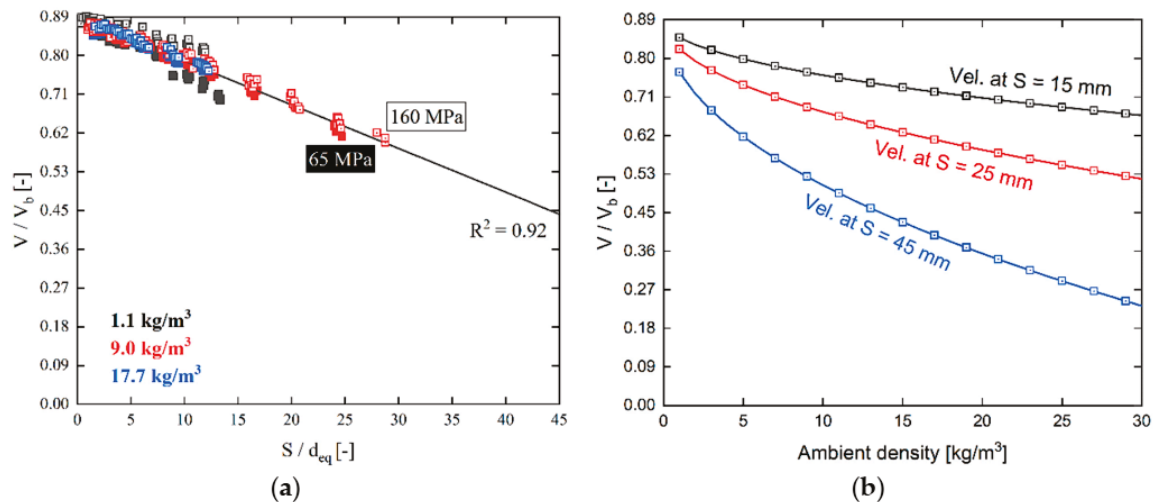


Figure 14. Spray velocity in different axial locations: (a) spray velocity vs. axial location; (b) spray velocity vs. ambient density.

Next, an operating scenario of a diesel engine is proposed that has the ambient conditions of 350 K, 1 bar, and 1.1 kg/m³ at the Bottom Dead Center (BDC), and a compression ratio of 16. Following an ideal gas and adiabatic compression assumption, it is easy to calculate the ambient conditions, i.e., T_{CA} , P_{CA} , ρ_{CA} (CA means the crank angle), in the compression stroke of this diesel engine. It is worth noting that the ambient temperature inside the engine chamber decides the sound speed of the mixture (according to Equations (2)–(4)), whereas the ambient density and pressure vary the spray propagation, i.e., the velocity of the entire spray (according to Equation (9)).

The critical injection pressure is proposed, which represents the minimum injection pressure to induce shock waves at a given condition. The critical injection pressure is attainable through an inverse calculation of the Bernoulli equation with the sound speed as input. Furthermore, the calculation can be extended to consider the supersonic-part length of the spray. Based on Equation (9), it is feasible to predict the local velocity within the spray against the injection pressure and the ambient density. In contrast, the injection pressure to reach supersonic spray at a given location is known by setting the sound speed as the local velocity at a specific location.

The calculation results are presented in Figure 15, and the result at 900 K is used for detailed explanations here, which represents a typical ambient temperature at the injection timing of diesel engines. First, regarding an injector having a theoretical injection velocity, an injection pressure of approximately 150 MPa is needed to generate shock waves during the fuel injection. The theoretical injection velocity can be otherwise known as that the injector has a C_v equaling to 1, whereas, typically, the C_v of diesel injectors is within 0.85–0.95. C_v gets higher as the injection pressure increases, or the nozzle has a high needle-lift and improved nozzle-geometry designs [33–35]. The lower C_v means the more energy lost during the fuel passing through the nozzle hole, which results in a lower injection velocity. In practice, engineers are devoted to increasing the nozzle C_v , which is beneficial to reduce the injection duration and pumping-energy loss while delivering the same amount of fuel. The shorter the injection duration, the faster the combustion occurs, which can improve the combustion quality and engine efficiency.

Regarding an injector with a C_v of 0.85, an injection pressure of approximately 210 MPa is required to induce shock waves during fuel injection (indicated as the top edge of the gray zone in Figure 15). Despite such high injection pressure needed for the shock-wave generation, it has been well established by major common rail suppliers that their products can provide an injection pressure up to 250 MPa [5–7]. Simultaneously, the powerful systems that can deliver fuels at even higher injection pressure than current products are still in development. As a result, it should not be surprising that the shock waves appear during fuel injection at diesel engine conditions. On the other hand, to obtain a supersonic-part length in the spray of $S_{supersonic} = 15, 30$, and 45 mm, the injection pressure should exceed 313, 525, and 1040 MPa, respectively. Given the fact that these injection pressures are incredibly high, the shock wave would appear mainly in the near-nozzle region, and the supersonic part in the spray is short. Thus, it is not expected that the shock-wave existence could significantly affect the spray characteristics, regarding an injection timing that is close to the Top Dead Center (TDC) when the ambient temperature and ambient density are both high.

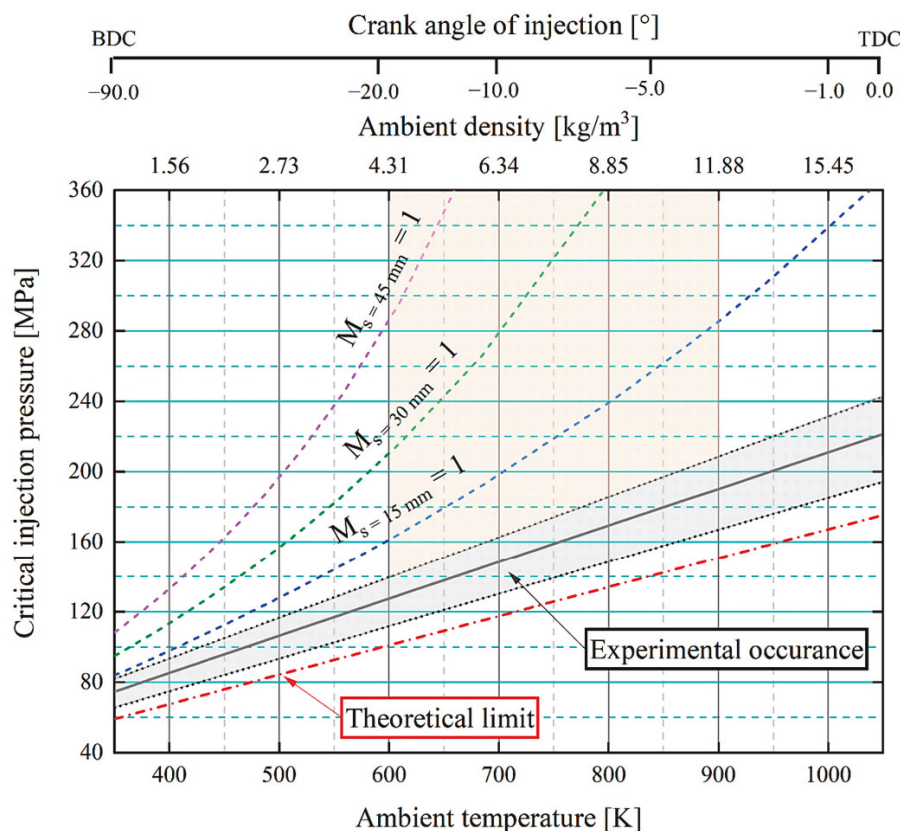


Figure 15. Critical injection pressure to induce a shock wave in a diesel engine. Ambient gas condition: mixture (21% O_2 ; 79% N_2); engine conditions in reference: 350 K BDC temperature, 16 compression ratio, and 1061 K TDC temperature. $M_{s=15\text{ mm}}$ means the Mach number at $S = 15$ mm.

However, it is also worth noting that the sophisticated combustion modes in modern engines have been investigated for long to achieve a further improved combustion quality, for instance, HCCI (homogeneous charge compression ignition) or RCCI (reactivity controlled compression ignition). These combustion modes require advanced injection timings when the ambient temperatures and densities are relatively low, where shock waves could more easily appear during fuel injection. For instance, at an advanced injection condition that was -20° before the Top Dead Center (TDC) when the ambient conditions are 590 K and 4.1 kg/m^3 , to obtain a local supersonic velocity at $S = 15, 30$, and 45 mm, the critical injection pressure could be as low as 173, 224, and 302 MPa, respectively. In that scenario, the supersonic part in the spray would largely extend, and the shock-wave effect on the spray characteristics might be no longer ignorable.

4. Conclusions

In this study, the potential for the shock-wave generation, and the possible influence at diesel engine conditions was examined. First, by taking advantage of the X-ray phase-contrast imaging (XPCI), experiments were conducted to obtain the spray-velocity characteristics from the nozzle exit to spray downstream field. Then, the shock-wave generation during fuel injection was observed using the schlieren imaging technique. Based on the results from two separate experiments, discussions were presented to understand the characteristics of the multiple shock waves. Accordingly, a diagram was proposed to predict possible shock-wave generation at diesel engine conditions. The key findings of the current study are summarized below.

1. Spray velocity has the maximum value at the nozzle exit then decreases gradually with distances in the center of spray. Increasing the ambient density results in the faster deceleration of spray velocity, but it barely affects the spray velocity at the nozzle exit. This can be understood by the fact that the spray velocity at the nozzle exit mainly depends on the pressure drop across the nozzle hole, which is insensitive to the change in ambient pressures since they are relatively small compared to the fuel pressures inside the nozzle. It is much more difficult for the supersonic spray generation in spray frontier than that at the nozzle exit.
2. Supersonic and subsonic ligaments coexist in one spray. Increasing the injection pressure or reducing the ambient density would extend the supersonic part in the spray. Multiple shock waves are generated during the fuel injection, and they most likely occur from the nozzle exit where the spray has the highest local velocity. Shock-wave generation during fuel injection could increase the spray penetration and reduce the spray angle. This effect gets enhanced as there is a longer supersonic part in the spray.
3. A diagram was proposed to predict possible shock-wave generation at diesel engine conditions. Based on that, it is noted that the shock wave can likely be induced during the fuel injection at diesel engine conditions. However, under a late injection timing, the supersonic part in the spray is short, and thus, significant changes in spray characteristics by the shock-wave generation is not expected. In contrast, the supersonic part in the spray extends largely under an advanced injection timing, when the shock-wave effect on the spray characteristics might be no longer ignorable.

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Article

Creating a Virtual Test Bed Using a Dynamic Engine Model with Integrated Controls to Support in-the-Loop Hardware and Software Optimization and Calibration

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Abstract: In the current study, a 0D/1D engine model built in the commercial code GT-Suite was coupled with the Electronic Control Unit (ECU) model created in the Simulink environment, aiming to more accurately predict the interaction of the engine and aftertreatment system (ATS) operating parameters, both during steady-state and transient maneuvers. After a detailed validation based on extensive experimental data from a heavy-duty commercial diesel Internal Combustion Engine (ICE), the engine model was fine-tuned and the 0D predictive diesel combustion model, DIPulse, was calibrated to best predict the combustion process, including engine-out NO_x emissions. For correct prediction of the engine's behavior in transient operations, the complete control strategy of the air path, including boost, exhaust gas recirculation (EGR), main and pilot Start of Injection (SOI), injection pressure, and exhaust flap, was implemented in the Simulink environment. To demonstrate the predictive capability of the model, a hot World Harmonized Transient Cycle (WHTC) was simulated, obtaining good agreement with the experimental data both in terms of emissions and performance parameters, confirming the reliability of the proposed approach. Finally, a case study on possible fuel consumption improvement through thermal insulation of the exhaust manifold, exhaust ports, and turbocharger was carried out.

Keywords: predictive combustion; 0D/1D engine model; ECU model; WHTC; coupled simulation

1. Introduction

The need to reduce man-made greenhouse gas emissions has led to new and challenging targets for future CO₂ emissions [1]. The European Union (EU) Commission proposes a reduction of the average CO₂ emissions of new heavy-duty vehicles by 15% and 30% in 2025 and 2030, respectively, compared with the 2019 baseline.

Moreover, EU regulation limits for emissions are becoming more strict, and type-approval procedures will also change radically with the introduction of World Harmonized Transient Cycle (WHTC) and Real Driving Emission (RDE) tests. The main challenge in the automotive industry is to find an optimal balance between fuel consumption and drivability within the boundaries set by emissions legislation, prompting Original Equipment Manufacturers (OEMs) to utilize different technologies such as exhaust gas recirculation (EGR) [2], Variable Valve Actuation (VVA) [3,4], exhaust line and turbocharger insulation [5], cylinder deactivation [6], and high-injection pressure systems [7] on the engine side together with complex aftertreatment layouts [8] with catalytic heating [9].

Since new complex engine and aftertreatment systems demonstrate considerable interactions and interdependencies [10], stand-alone aftertreatment system considerations might not lead to the optimal solution for the whole system. As total system complexity increases, using robust and reliable engine simulation models will bring aftertreatment

system (ATS) boundary conditions into the loop during ATS development. The co-simulation activity presented here is the first step to a dedicated full-system simulation tool that will eventually evaluate the cost and benefit interactions of all the technologies described above.

Detailed simulation of charge motion, in-cylinder flow, and combustion [11–13] are usually carried out using 3D Computational Fluid Dynamics (CFD) codes. However, the relatively high computational costs of these methods have restricted their application to some extent, particularly when considering complex systems. In this context, 0/1D models, which are traditionally less computationally intensive, can be used as reliable simulation tools.

Different types of 0/1D diesel combustion models have been proposed in the literature [14,15]. In this study, the commercially available predictive combustion model DIPulse [16,17], developed by Gamma Technologies (Westmont, United States), was used to accurately predict fuel burn rate and engine performance parameters. Moreover, engine-out NO_x emissions were predicted by the extended Zeldovich method [18].

Diesel engine system controls [19] are another important consideration for modern engines to further improve fuel consumption and reduce NO_x and particulate matter; modeling of real control strategies for air and injection systems is necessary to reliably predict engine behavior, especially under transient conditions.

In order to create a virtual test bed, a physical engine model should be coupled with the correct control strategy. Martin et al. [20] created a virtual test bed using a gas dynamics software called VEMOD. A virtual Electronic Control Unit (ECU) model was used in this study, which might control the engine differently from the real ECU model, especially under transient conditions. For instance, the Proportional, Integral, and Derivative (PID) parameters (Proportional, Integral, and Derivative gains), transient correction at the start of injection, transient correction of injection pressure, and flap actuation during transient engine operations will affect the results both in terms of performance and emissions.

In this study, the complete airpath model of the ECU was developed in the Simulink environment to control the engine model and, together with the detailed combustion and emissions models, engine performance parameters both during steady state and, more importantly, during transient operation were predicted accurately.

The airpath control part of the real ECU model offers the possibility to run the engine with the real control strategy in the Software-in-the-Loop (SiL) ambient, replicating exactly the engine control behavior in real transient driving conditions. In previous studies [20], more simplified approaches have been suggested.

The model is further utilized to simulate a hot WHTC and compare the measurements. One of the most important parameters for ATS performance is the temperature after the turbine, and, as will be shown in the results (Section 4.3), using the ECU control strategy and suggested combustion model, temperature prediction was improved compared with prior research such as the paper published by J. Martin et al. [20], which shows more than a 100 °C error in turbine outlet temperature. The error rate is reduced to 30 °C in the current study, which resulted in more precise prediction of ATS performance. Finally, after validation of the model, the benefits of thermal insulation of the exhaust manifold, exhaust ports, and turbocharger on fuel consumption were evaluated. This case study indicates how a reliable and computationally efficient 0D/1D simulation model can be used to not only decrease the test bench activity and chassis dyno experimental campaign, but also to better target any testing activity, once at the test bench. Furthermore, considering that GT-Suite is utilized extensively in the Internal Combustion Engine (ICE) analysis, the current study shows the capability to create a virtual test bed using GT-Suite coupled with ECU control models.

2. Engine Characteristics and Operating Points

The engine used for the study is a heavy-duty six-cylinder diesel engine, equipped with a common rail, high-pressure EGR system, and Variable Geometry Turbocharger (VGT), the main characteristics of which are described in Table 1.

Table 1. Main characteristics of the test engine.

Engine Characteristics	
Bore	128
Stroke	144
N° cylinders	6
Displacement	11 L
Peak Power	370 kW @1900 rpm
Peak Torque	2360 Nm @1000 rpm
Valvetrain	4 valves per cylinder

The experimental tests were performed over a complete engine operating map with the speed varying from 550 to 2200 rpm, as shown in Figure 1.

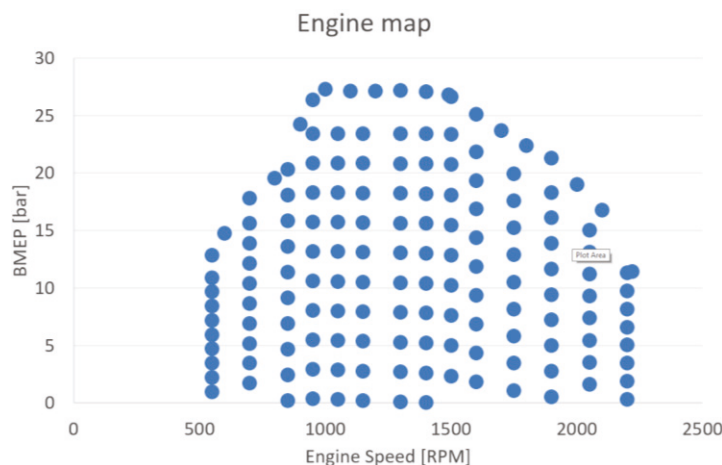


Figure 1. Experimental test matrix.

3. Simulation Model Description

The simulation activity begins with the careful validation of the detailed model. As shown in Figure 2, the detailed model is divided into three main parts, namely: engine, turbocharger, and aftertreatment. The engine model was created by reducing the 3D model of the engine airpath to 1D, and by converting it into simple flowsplits and pipes using the GTSuite tool GEM3D.

The turbocharger is traditionally modeled using maps of flow characteristics provided by the supplier. Particularly for turbine maps, heat transfer is most often neglected. This does not significantly affect general engine performance, but does affect temperature prediction at the turbine outlet and, thus, aftertreatment inlet temperature [21–24]. Therefore, we decided to include a sub-model that takes thermal effects into account. The heat transfer path from turbocharger to coolant and environment is shown in Figure 3. The nodal model of heat transfer in the turbocharger utilized in this work was proposed by Serrano et al. [25–27].

As shown in Figure 4, three thermal masses are considered in order to account for the shaft/bearings, turbine, and compressor wheels. The turbine and compressor volutes are simplified [28,29] to a pipe (Figure 5) with the diameter and length calculated by Equations (1) and (2). Due to a lack of experimental data regarding heat transfer in turbochargers, the limits and procedure recommended in [16] were used.

$$l_{vol} = l_l + \frac{r_1 + r_2}{2} \pi \quad (1)$$

$$d_{vol} = \sqrt{\frac{4V_{vol}}{\pi l_{vol}}} \quad (2)$$

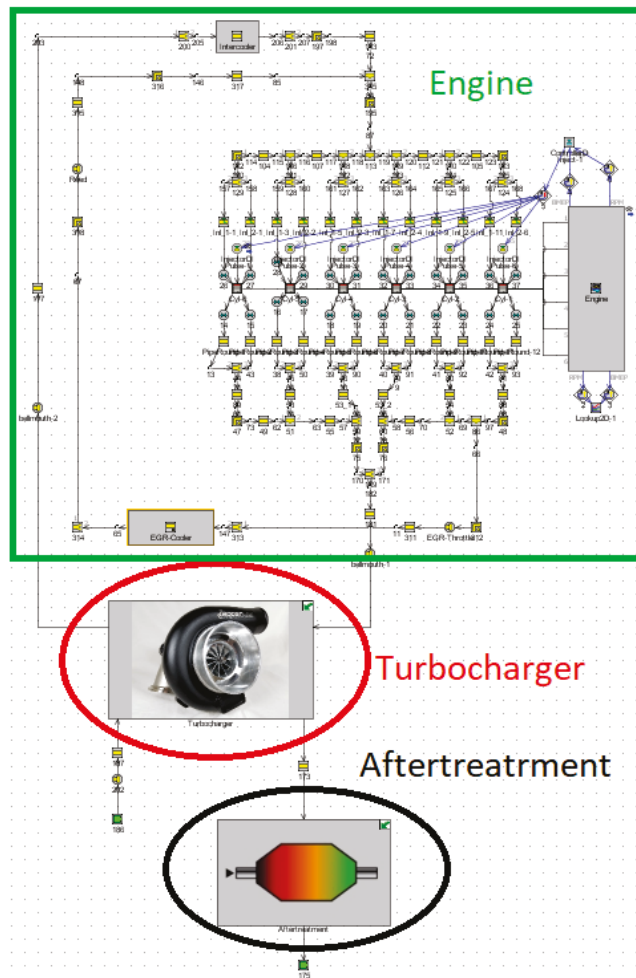


Figure 2. Model layout with the engine, turbocharger, and aftertreatment.

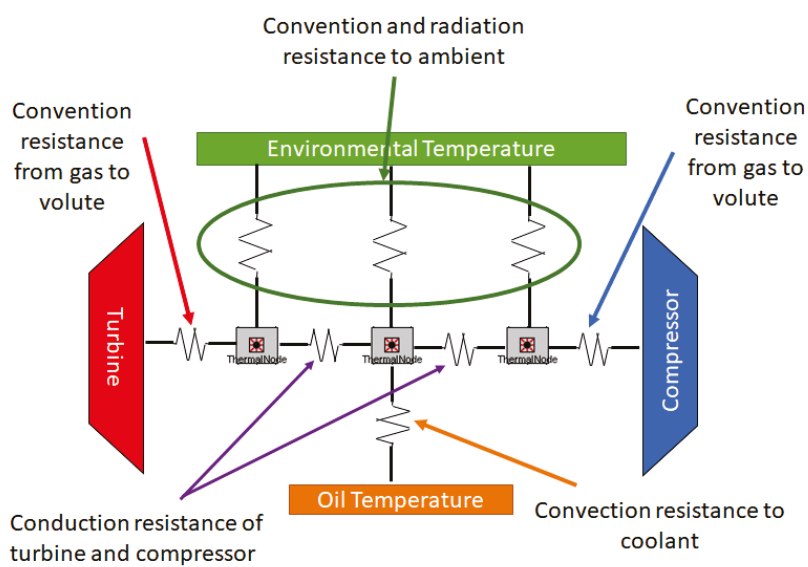


Figure 3. Heat transfer path from turbocharger to coolant and environment.

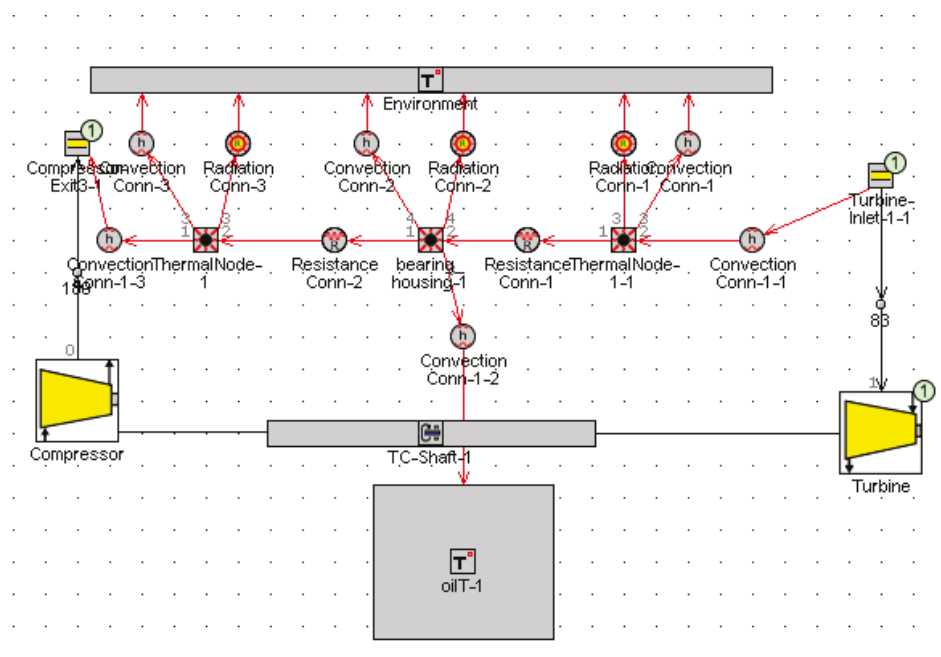


Figure 4. The 0D heat transfer model from turbocharger to coolant and environment.

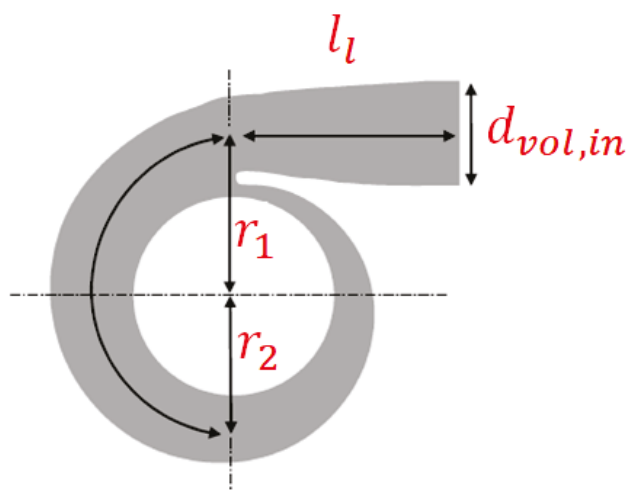


Figure 5. Simplified representation of the turbine volute geometry [16].

In 0D simulations, the aftertreatment line (Figure 6) is usually modeled by the calibration of a series of reactions in order to predict tailpipe emissions. In this case, since the thermal behavior and pressure drop of the ATS are of interest, and not the tailpipe NOx, and to improve the simulation time, the aftertreatment is modeled considering the actual thermal inertia of the system, such as the material properties and size of each component, without solving any reactions; hence, the heat generated or absorbed by the reactions is neglected. In a further activity, not presented here, the model was also coupled with a full chemical ATS 1D model.

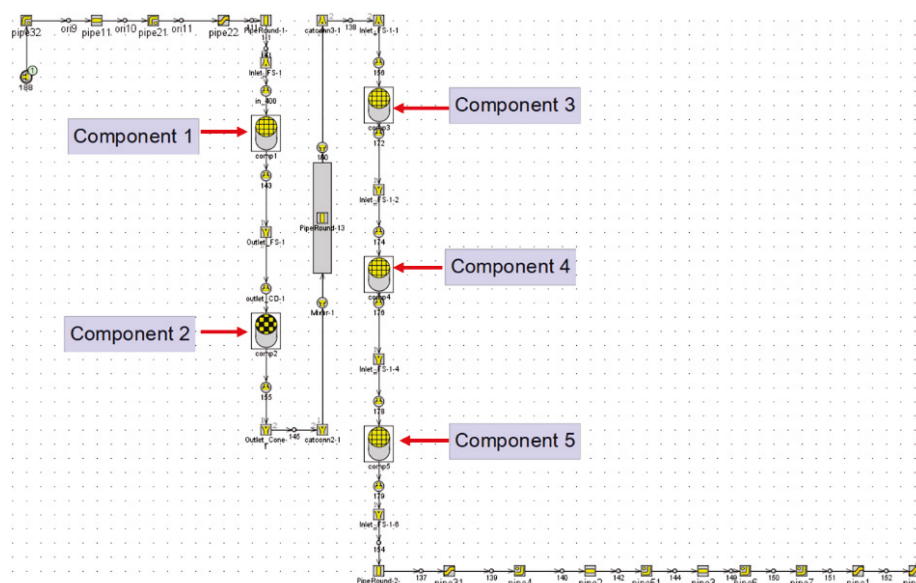


Figure 6. Aftertreatment line layout.

3.1. Predictive Combustion Model

In order to take into account the effect of different EGR rates, Start of Injection (SOI), rail pressure, pilot injection, and operating conditions on combustion, the DIPulse predictive combustion model [16,17], was used. The DIPulse model separates the in-cylinder content into three thermodynamic zones, each with their own composition and temperature [16].

All trapped mass at intake valve closure is considered the main unburned zone. The spray is divided into two basic parts, namely unburned and burned zones. The former includes the fuel and entrained gas, and the latter contains the product of combustion. The model takes into account the fuel evaporation, mixing, and burning process [16]. The combustion sub-model contains four calibration multipliers:

1. Entrainment Rate (ERM);
2. Ignition Delay (IDM);
3. Premixed Combustion Rate (PCRM); and
4. Diffusion Combustion Rate (DCR).

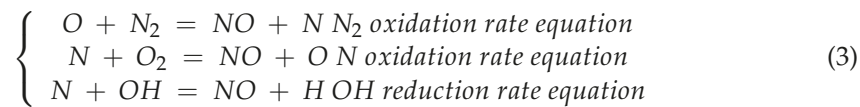
A Design of Experiment (DoE) [30] technique, combined with a Genetic Algorithm (GA), was used to find the optimum calibration parameters with the aim of minimizing the Root Mean Squared (RMS) error between experimental and simulated burn rates. The calibration approach has been discussed in more detail in [17,31]. Optimized values of the calibration parameters are shown in Table 2.

Table 2. Optimized values of the combustion model's calibration parameters.

Calibration Parameters	Value
ERM	1.86
IDM	0.35
PCRM	0.85
DCRM	0.65

3.2. Engine-Out NO_x Model

The NO_x calculation is based on the extended Zeldovich mechanism [18]. The reactions are shown in Equation (3).



Two calibration multipliers are used:

- a multiplier for the predicted net rate of NO_x formation; and
- a multiplier for the activation energy of the N₂ oxidation rate equation.

A DoE analysis was carried out with the aim of minimizing the error function that was defined based on the difference between the simulated and measured NO_x engine-out concentration.

3.3. Airpath Control Model

The ECU airpath control model includes boost pressure, EGR, injection pressure, pilot and main SOI, and exhaust flap control. Depending on the engine and ATS conditions, the ECU combustion model can vary between different modes, including normal mode, cold start mode, and regeneration mode.

The engine model passes the physical operating parameters to the ECU model, and the control parameters then adjust to the requested engine operation. It is important to note that the details of the ECU model could not be published for confidentiality reasons (Figure 7).

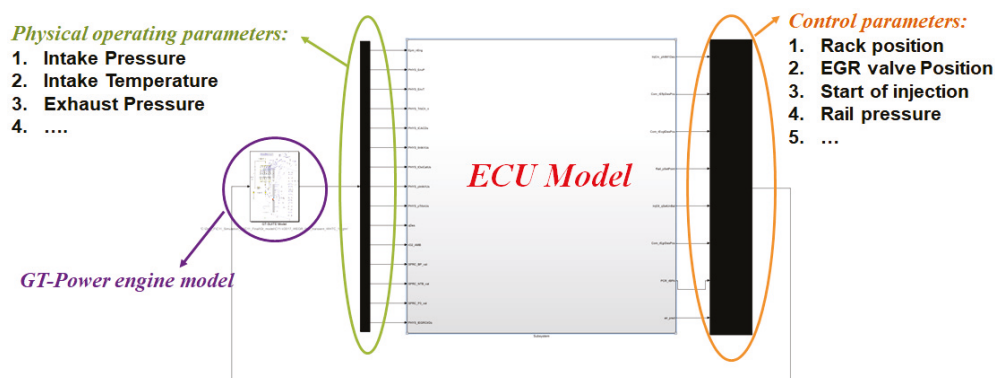


Figure 7. Schematic presentation of the Electronic Control Unit (ECU) model coupled to the GT-Suite power engine model.

The boost pressure is closed-loop and contains four main governors:

- a boost pressure governor;
- an exhaust pressure—steady governor;
- an exhaust pressure—transient governor; and
- a turbine speed governor.

Using the engine mode and physical parameters of the engine model, such as turbocharger speed, boost pressure, exhaust pressure, and the corresponding setpoints and gradients, a state machine decides which governor should be active. For instance, if the turbocharger speed exceeds its maximum limit, the turbocharger speed governor will activate to protect the turbocharger.

The EGR control is open-loop over the complete engine map. The desired EGR rate is a function of injected fuel, speed, and environmental conditions, which are part of the ECU model in the form of calibration maps. In order to correctly recirculate the desired

EGR quantity, the EGR valve model calculates the desired valve opening position based on the engine mode and the following input signals:

- desired mass flow through the EGR valve;
- pressure before EGR valve;
- pressure after EGR valve; and
- temperature before EGR valve.

The SOI and injection pressure controllers are of the open-loop type. The SOI and injection pressure are calculated based on the maps which are the function of operating mode, EGR rate, injected fuel, and engine speed. The base value will be corrected under transient conditions.

The exhaust flap control is also of the open-loop type. The exhaust flap is inactive in the normal operation mode of the engine. The exhaust flap's base position is calculated as a function of injected fuel and engine speed, which is further scaled based on the distance of the Selective Catalytic Reduction (SCR) temperature from a threshold level such that, as the difference between the SCR temperature and the threshold level reduces, the exhaust flap scaling acts to increase the flap opening.

3.4. ECU and GT-Suite Coupling

After the creation of the GT-Suite model, the 'SimulinkHarness' template should be included in the model. In this template, all input (from GT-Suite to Simulink) and output (from Simulink to GT-Suite) parameters should be defined and linked to the relevant part of the model. The input parameters required by the control strategy are the physical parameters, including engine speed, requested fuel, air flow rate, boost pressure, and turbocharger speed. The output parameters are the start of injection, injection pressure, VGT position, main and pilot injection, flap position, and EGR valve position.

In the next step, the GT-Suite Model's S-Function block should be inserted into the ECU model. This block can be found in the Simulink Library Browser. As can be observed in Figure 7, all input and output parameters are linked in the S-function GT-Suite block and the ECU model. Afterwards, within the GT-Suite Model's S-Function block, it is necessary to specify the number of inputs and outputs, the time step, and the GT-Suite model's name.

For example, regarding the combustion control, the injection pressure, SOI, and injection quantities of all injection events are transferred from Simulink (the ECU model) to GT-Suite, where the DIPulse predictive combustion model will evaluate the burn rate and consequently the in-cylinder pressure.

Further information about the requirements for, and descriptions of, the GT-Suite and Simulink coupling can be found in [16].

4. Results and Discussion

4.1. Steady State Simulation

After the DIPulse combustion model was calibrated as described in Section 3.1, the optimized DIPulse multipliers were fixed for all engine operating points, and the accuracy of the model was confirmed for all operating points on the map. The pressure and burn rate profiles are presented in Figure 8 for three speeds at full load. Good agreement was obtained between the predicted burn rates and the in-cylinder pressure with measured values. Similarly good agreement was achieved across the entire engine operating map.

To demonstrate the predictive capability of the model, the errors between experimental and simulated values of Brake Specific Fuel Consumption (BSFC) (Figure 9a), air mass flow rate (Figure 9b), temperature before turbine (Figure 10a), temperature after turbine (Figure 10b), peak cylinder pressure (Figure 11a), and turbocharger speed (Figure 11b) are shown as contour plots in the engine map. The absolute average errors are 1.77%, 1.46%, 9.3 °C, 10.8 °C, 2.9 bar, and 1.2%, respectively.

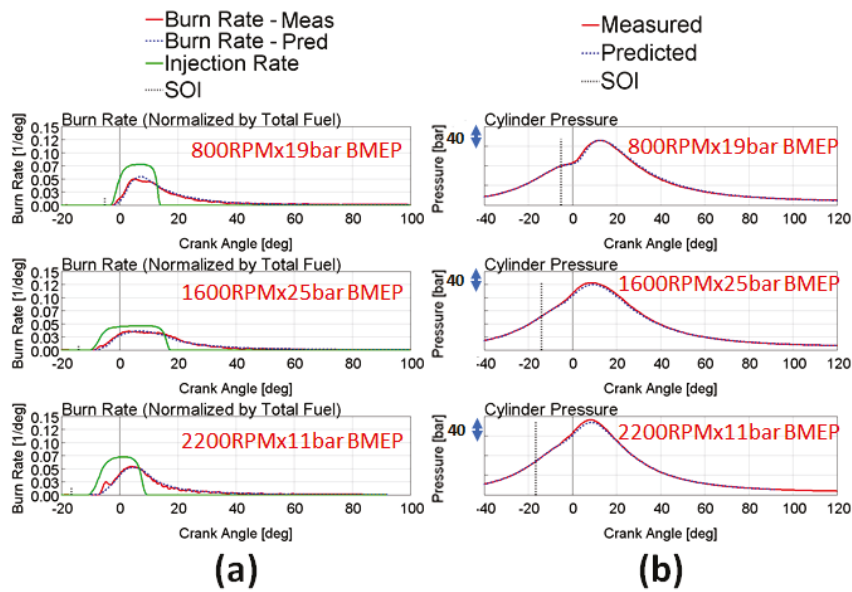


Figure 8. Comparison between measured and simulated in-cylinder pressure (a) and burn rate (b).

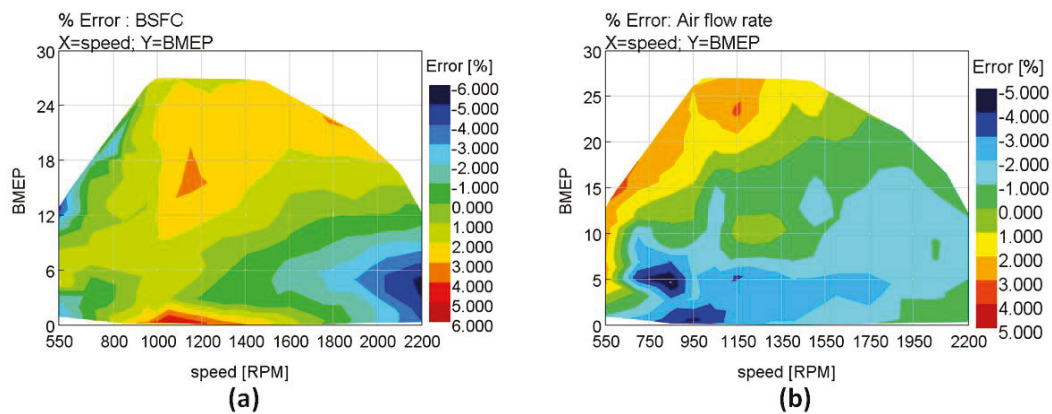


Figure 9. Percentage error between experimental and simulated values of Brake Specific Fuel Consumption (BSFC) (a) and air flow rate (b).

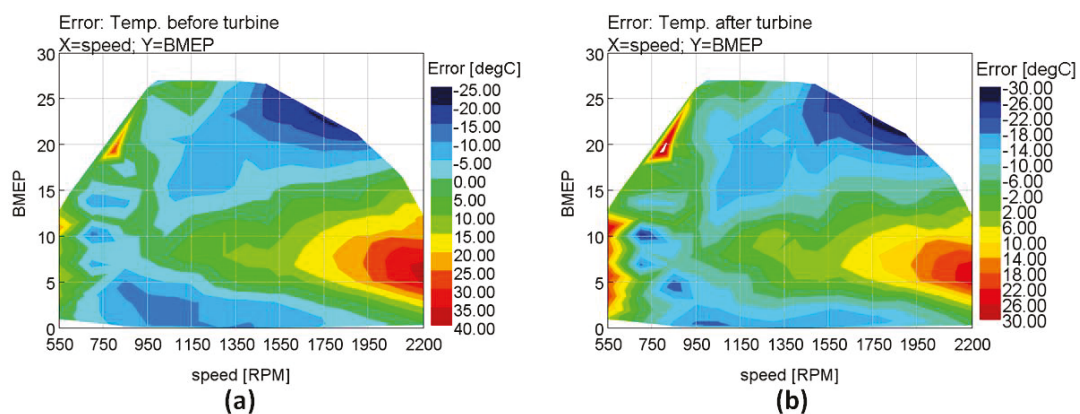


Figure 10. Error between experimental and simulated values of temperature before turbine operation (a) and temperature after turbine operation (b).

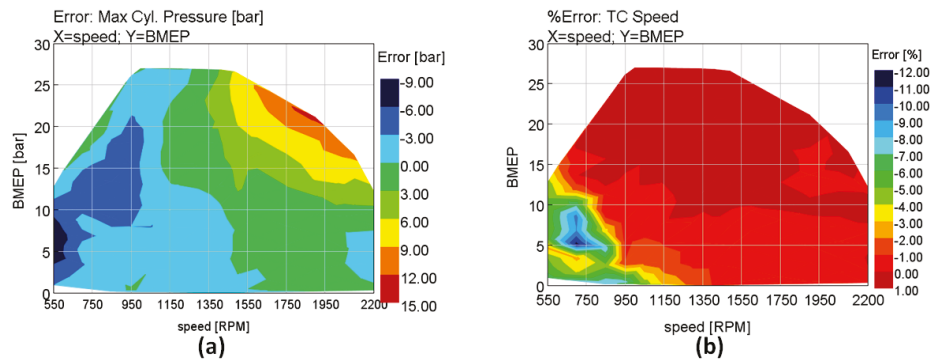


Figure 11. Error between experimental and simulated values of peak cylinder pressure (a) and turbocharger speed (b).

As shown in Figure 12, the simulated engine-out NO_x values versus the measured values have an R-squared coefficient of 0.81 and an error band of 20% at most of the points on the engine map.

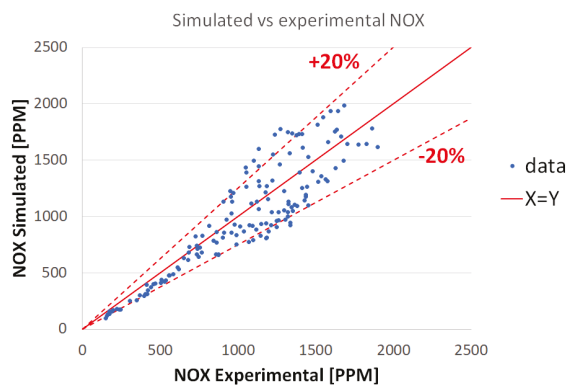


Figure 12. Comparison between experimental and simulated engine-out NO_x.

4.2. Transient Operation Simulation

The engine's behavior was numerically and experimentally analyzed during transient operation. An abrupt change in pedal position (from 0 to 100%) was specified to occur at a constant engine speed. The objective was to define the response time of the turbocharger. The tests were carried out at two different speeds (900 rpm and 1900 rpm). The results are shown in Figures 13 and 14.

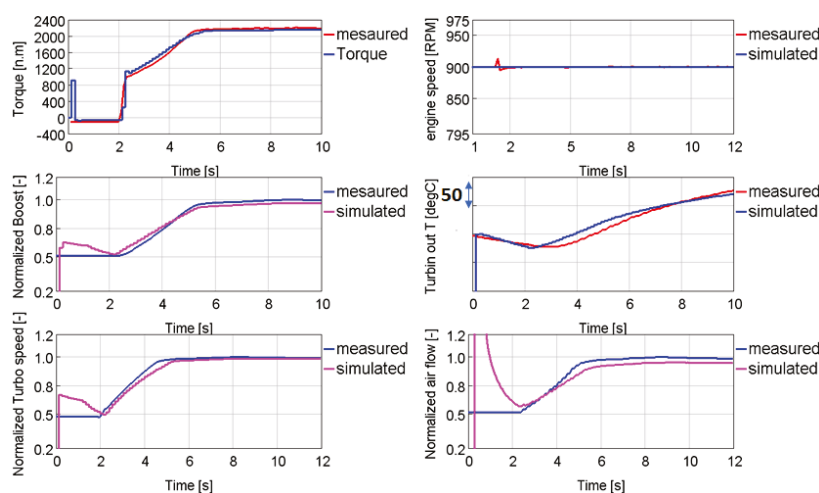


Figure 13. Operating parameters of the engine during transient operation at 900 rpm.

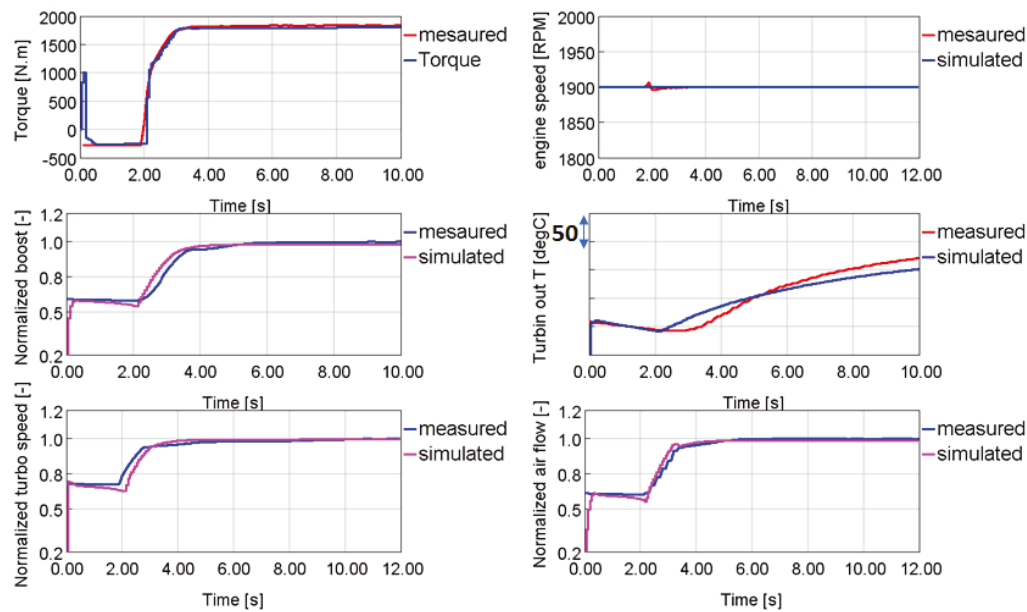


Figure 14. Operating parameter of the engine during transient operation at 1900 rpm.

4.3. WHTC Simulation

We used the simulation to analyze the engine's behavior during a hot World Harmonized Transient (WHT) homologation cycle with the aim of assessing the predictive capability of the model. The engine speed and torque traces (typical WHTCs) are the main inputs of the model. The engine speed was imposed, while the torque was controlled by the fueling within the limits defined in the ECU (smoke map and maximum fuel injection).

As shown in Figure 15, there is satisfactory agreement between the experimental and simulated temperature before diesel catalyst oxidation and SCR temperature. A comparison of the other operating parameters is shown in Figure 16.

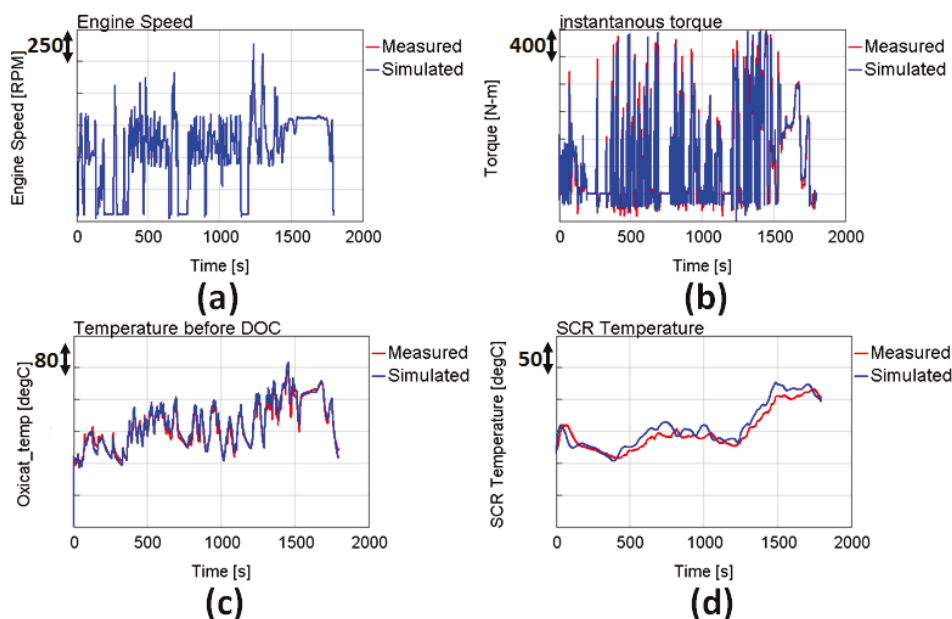


Figure 15. Comparison of simulated and experimental engine speed (a), torque (b), temperature before diesel catalyst oxidation (c), and selective catalytic reduction (SCR) temperature (d).

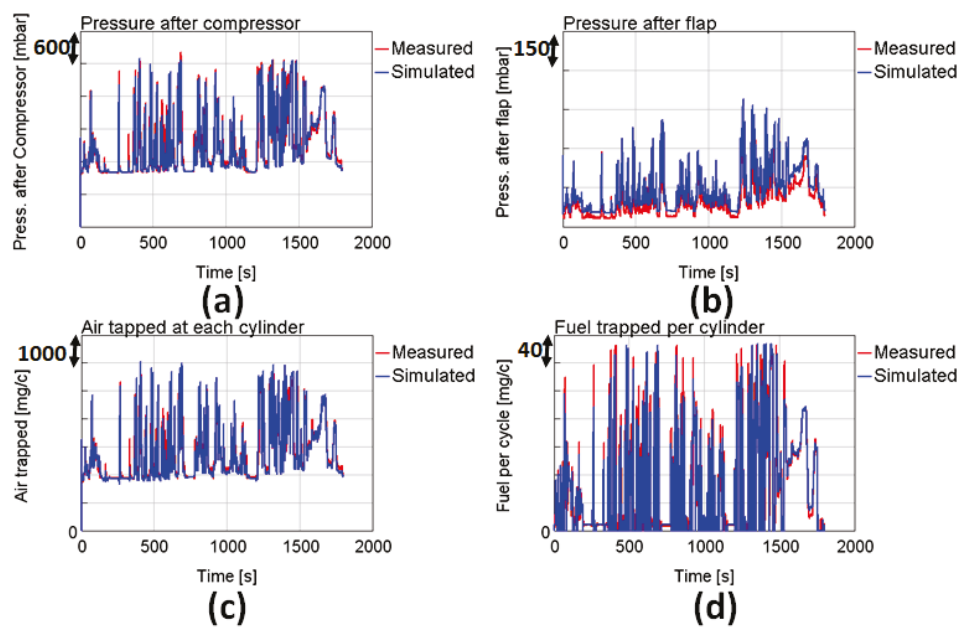


Figure 16. Comparison of experimental and simulated pressure after compression (a), pressure after flap opening, (b) air trapped at each cylinder (c), and fuel trapped per cylinder (d).

The exhaust flap and combustion are controlled by the SCR temperature. Two combustion modes were used in the WHTC. In the first part (from 0 s to about 1350 s), the ‘late combustion’ mode was used, in which the exhaust flap was active (Figure 17d) since the SCR temperature was below the specified threshold temperature in the calibration (Figure 17b). In the second part (from 1350 s to 1800 s), the ‘normal combustion mode’ was active and the exhaust flap was fully open (Figure 17d). The difference in SOI between the two combustion modes is shown in Figure 18. It can be observed that the difference in SOI is larger at a low speed and in the low load region.

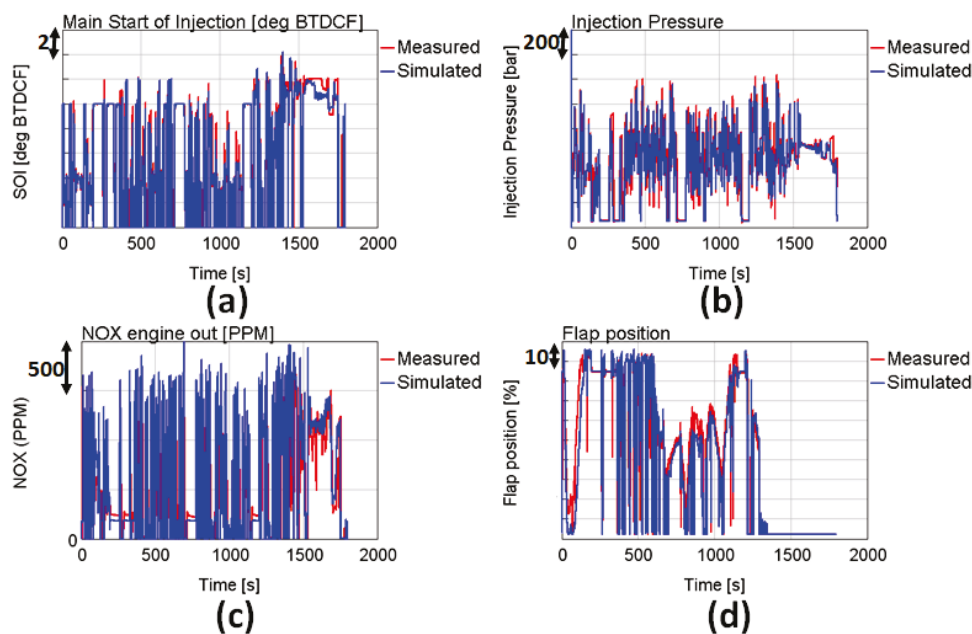


Figure 17. Comparison of experimental and simulated main Start of Injection (SOI) (a), injection pressure (b), engine-out NOx (c), and flap position (d).

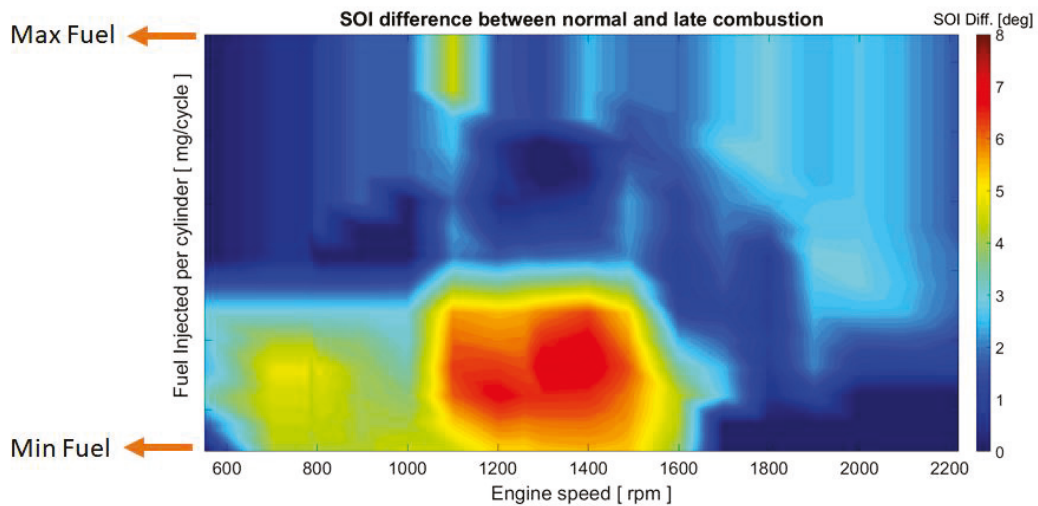


Figure 18. Difference in SOI between normal and late combustion.

A comparison of the simulated and experimental cumulative injected fuel, power, and engine-out NO_x emissions is shown in Figures 19–21, respectively. The error in BSFC (g/kWh) and engine-out NO_x (g/kWh) emissions is 0.3% and 11.5%, respectively, over the entire cycle.

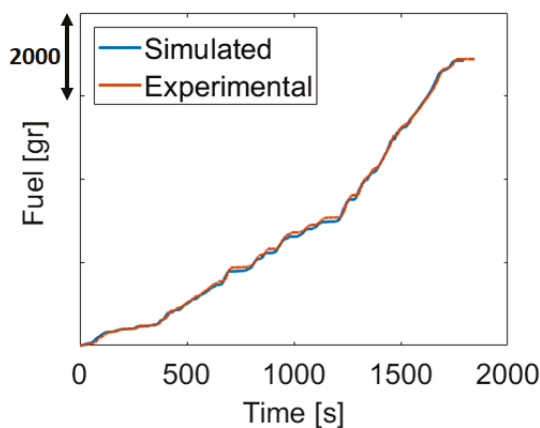


Figure 19. Comparison of simulated and experimental cumulative injected fuel during the World Harmonized Transient Cycle (WHTC).

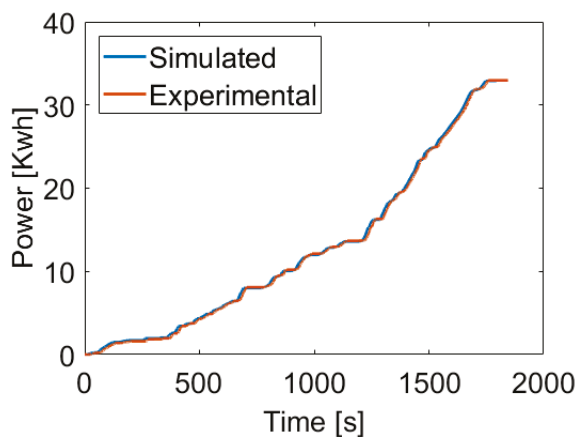


Figure 20. Comparison of simulated and experimental cumulative power during the WHTC.

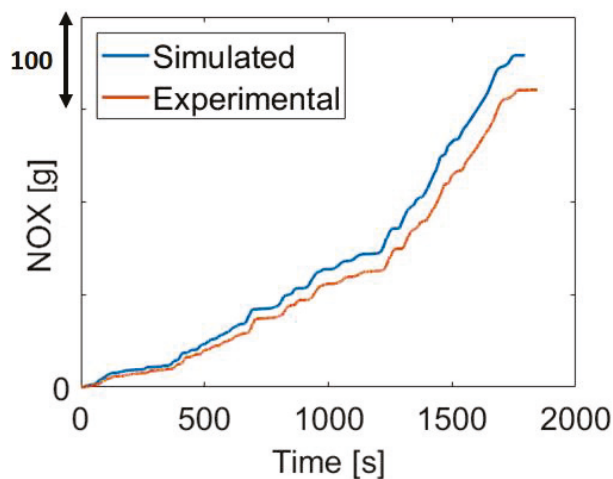


Figure 21. Comparison of simulated and experimental cumulative engine-out NO_x during the WHTC.

4.4. Thermal Insulation Analysis

As the SCR temperature approaches the target operating temperature, the flap actuation will be reduced and, once the target temperature is reached, combustion will be switched to the optimal mode. Preserving the available exhaust gas enthalpy through insulating the exhaust manifold, exhaust ports, and turbocharger (Figures 22 and 23) should make it possible to reduce flap activation and, thus, improve fuel consumption.

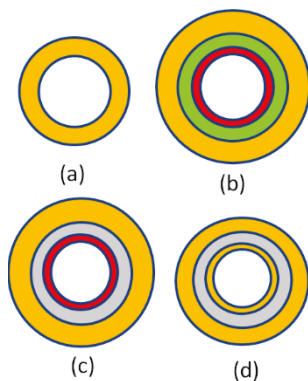


Figure 22. (a) Baseline, (b) exhaust port insulation, (c) exhaust manifold insulation, and (d) turbocharger insulation.

Since there were no experimental data available regarding the insulation material, we used the data available in [32,33].

Different thermal insulation layouts can be found in [32]. The actual insulation of the baseline, exhaust manifold, exhaust ports, and turbocharger is shown in Figure 22. The baseline exhaust manifold was made from cast iron (Figure 22a). We made the following assumptions about the insulation:

- for the exhaust port, a Ytria-stabilized zirconia [34] is sandwiched between the inner stainless-steel liner and the external iron structure (Figure 22b);
- for the exhaust manifold, an air gap exists between the inner stainless-steel liner and the external iron manifold (Figure 22c); and
- for the turbocharger, an external iron shield creates one air gap layer on the cast-iron turbine housing (Figure 22d).

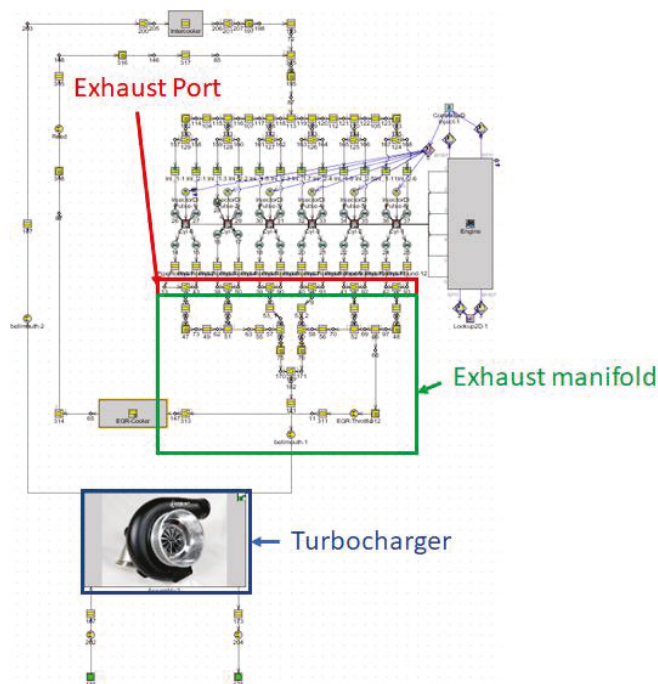


Figure 23. Insulated parts in the model: exhaust port, exhaust manifold, and turbocharger.

The material properties that we used are shown in Table 3 [32]. It is worth noting that the values for thermal conductivities in the table are averages and the temperature-dependent thermal conductivity was used in the simulation.

Table 3. Material properties [32].

Material	Color	Density ($\frac{kg}{m^3}$)	Thermal Conductivity ($\frac{W}{m \cdot K}$)	Surface Emissivity (–)
Stainless-steel	Red	7900	25.4	0.85
Air	Gray	1.16	0.0667	–
YSZ	Green	4950 [1]	1.0 [1]	0.729
Iron	Yellow	7870	69.5	0.8

In the final step of the analysis, the WHTC was simulated considering the insulation of different parts. The BSFC benefits are shown in Figure 24. Thermal insulation of the exhaust ports, exhaust manifold, and turbocharger leads to BSFC improvements of 0.5%, 0.24%, and 0.25%, respectively. In the ideal situation, if the SCR could be maintained at a temperature higher than the minimum threshold over the entire cycle, and thus always be in the optimal combustion mode, a maximum fuel saving of 3.2% over the baseline is possible. This is the maximum possible benefit using various technologies, such as catalytic heating.

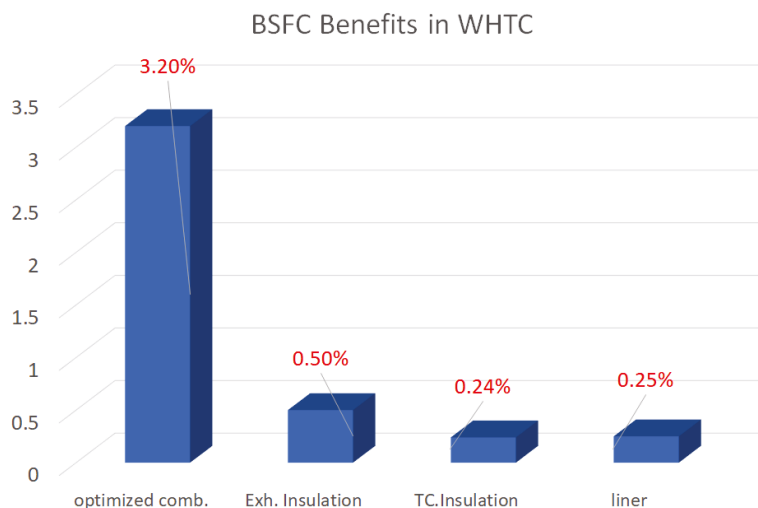


Figure 24. BSFC benefits obtained by insulating the exhaust manifold, turbocharger, and exhaust ports.

5. Conclusions and Future Work

The aim of this study is to couple a 0D combustion model and an ECU model in order to correctly predict, under steady-state and transient conditions, the combustion trend, emissions, and other performance-related parameters of a heavy-duty diesel engine. Optimization of individual subsystems, such as the ICE, ATS, and control strategy, may not lead to the global optimal solution due to their complex interaction within the entire system. However, the holistic approach of physical models coupled with the real airpath ECU controls presented in this work increases the reliability of the predicted system performance and interactions. Such a tool could be utilized as a virtual test bench to better identify a system's behavior in terms of emissions and engine performance. It could also be used to prepare the engineer before they go to the test bench, where an efficient use of time is critical to reducing development costs.

After careful validation of the engine model, the DIPulse predictive combustion model was calibrated using a DoE technique with the objective of minimizing the RMS error between the predicted and experimental burn rates. Additionally, the engine-out NO_x emissions were predicted using the extended Zeldovich mechanism.

Subsequently, the ECU model, developed in the Simulink environment and validated during both transient and steady-state operation, was coupled with the engine model. The results show good agreement with measurements over a WHTC.

The coupled model was then used to predict the benefits of thermal insulation on the exhaust line in terms of fuel consumption during a hot WHTC. It was shown that insulation of the exhaust ports, exhaust manifold, and turbocharger leads to BSFC improvements of 0.5%, 0.24%, and 0.25%, respectively. A further simulation was carried out considering constant operation in the optimal combustion mode with a fully open exhaust flap. This analysis predicted a maximum achievable BSFC benefit using innovative technologies, such as catalytic heating, of about 3.2% over the baseline hot WHTC.

Future development of the work presented here will include an evaluation of the effectiveness of different thermal management technologies, such as VVA, catalytic heating, thermal insulation, intake flaps, and cylinder deactivation during a cold WHTC. Moreover, a fully calibrated ATS model, including chemical reactions, will be included in order to predict tailpipe NO_x emissions in a fully integrated engine, controls, and ATS analysis.

Finally, due to its reduced computational and calibration requirements, the 0D/1D model developed in this study could be used as a “virtual test rig”, allowing for the evaluation of changes in hardware or software and their impact on the system from an early stage of engine development.

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Abbreviations

Aftertreatment System	ATS
Brake Mean Effective Pressure	BMEP
Brake Specific Fuel Consumption	BSFC
Computational Fluid Dynamics	CFD
Design of Experiment	DoE
Diesel Oxidation Catalyst	DOC
Diffusion Combustion Rate Multiplier	DCRM
Electronic Control Unit	ECU
Entrainment Rate Multiplier	ERM
European Union	EU
Exhaust Gas Recirculation	EGR
Genetic Algorithm	GA
Ignition Delay Multiplier	IDM
Internal Combustion Engine	ICE
Original Equipment Manufacturers	OEMs
Premixed Combustion Rate Multiplier	PCRM
Proportional, Integral and Derivative	PID
Real Driving Emission	RDE
Root Mean Squared	RMS
Selective Catalytic Reduction	SCR
Software in the Loop	SiL
Start of Injection	SoI
Variable Geometry Turbocharger	VGT
Variable Valve Actuation	VVA
World Harmonized Transient Cycle	WHTC

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Article

Monte Carlo Simulation Methodology to Assess the Impact of Ambient Wind on Emissions from a Light-Commercial Vehicle Running on the Worldwide-Harmonized Light-Duty Vehicles Test Cycle (WLTC)

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Abstract: A Monte Carlo simulation methodology is suggested in order to assess the impact of ambient wind on a vehicle's performance and emissions. A large number of random wind profiles is generated by implementing the Weibull and uniform statistical distributions for wind speed and direction, respectively. Wind speed data are drawn from eight cities across Europe. The vehicle considered is a diesel-powered, turbocharged, light-commercial vehicle and the baseline trip is the worldwide harmonized light-duty vehicles WLTC cycle. A detailed engine-mapping approach is used as the basis for the results, complemented with experimentally derived correction coefficients to account for engine transients. The properties of interest are (engine-out) NO and soot emissions, as well as fuel and energy consumption and CO₂ emissions. Results from this study show that there is an aggregate increase in all properties, vis-à-vis the reference case (i.e., zero wind), if ambient wind is to be accounted for in road load calculation. Mean wind speeds for the different sites examined range from 14.6 km/h to 24.2 km/h. The average increase in the properties studied, across all sites, ranges from 0.22% up to 2.52% depending on the trip and the property (CO₂, soot, NO, energy consumption) examined. Based on individual trip assessment, it was found that especially at high vehicle speeds where wind drag becomes the major road load force, CO₂ emissions may increase by 28%, NO emissions by 22%, and soot emissions by 13% in the presence of strong headwinds. Moreover, it is demonstrated that the adverse effect of headwinds far exceeds the positive effect of tailwinds, thus explaining the overall increase in fuel/energy consumption as well as emissions, while also highlighting the shortcomings of the current certification procedure, which neglects ambient wind effects.

Keywords: driving cycle; fuel consumption; emissions; wind speed; Monte Carlo simulation; light commercial vehicles

1. Introduction

Much attention has been paid lately to real-world driving emissions from passenger cars and commercial vehicles. The difference between real-world and type-approval values for fuel consumption and emissions peaked at almost 42%, on average, in 2016 [1]. Since then, there has been some decline, although the gap is far from closed [2], and recent studies show that the newly adopted Worldwide-harmonized Light-duty Test Procedure (WLTP) alone will not be able to bridge that gap [3]. When it comes to light commercial vehicles, their contribution to aggregate transport emissions is expected to come under the spotlight, as home-delivery practices become all the more widespread [4–6], with the COVID-19 crisis further accelerating this trend.

One significant contributor to the gap between real-world and type-approval emissions, accounting for more than one-third of it, is the under-estimation of real-world road load forces, namely rolling resistance, aerodynamic drag, and gradient resistance [7–9].

To make matters worse, the latter is entirely excluded, mostly for practical reasons, from road load determination during type-approval testing, although its effect on emissions is far from negligible even for a zero net elevation trip [10–15].

Another hugely important variable of the real-world driving environment is natural wind and its fluctuations in terms of velocity as well as direction with respect to the vehicle movement. During a typical trip, both wind velocity and wind direction can vary significantly depending on the surroundings, the presence of other vehicles on the road, the appearance of wind gusts, etc. [16]. On the contrary, the aerodynamic development of vehicles typically takes place in wind tunnels or with the use of CFD techniques assuming ideal conditions (e.g., uniform flow), thus introducing discrepancies vis-à-vis the actual aerodynamic performance of a vehicle driven on the road [17–19].

Such discrepancies are amplified by the exclusive use of the zero-yaw drag coefficient to determine the vehicle's aerodynamic performance. The zero-yaw drag coefficient represents the minimum drag condition for a vehicle, with the actual drag experienced by a vehicle expected to rise considerably as a function of the yaw angle [20].

The WLTP provides alternatives with regards to the determination of aerodynamic drag as part of the total road load. On the one hand, the vehicle can be tested in a wind tunnel assuming minimum levels of turbulence and zero yaw [21,22]. On the other hand, the vehicle may undergo a coast-down test in order to determine the road load coefficients for use in dynamometer setting. During this procedure, the vehicle is driven on a flat and dry test track, and average wind speeds do not exceed 3 m/s. The test is conducted in both directions, and the net effect of wind is further eliminated by a relevant wind correction algorithm [23–25]. In both cases, natural wind and its effect on vehicle performance and emissions is neglected. The opposite is true for the RDE procedure, as there is no limit on the permissible value for ambient wind speed throughout a trip [23]. However, the RDE procedure focuses on NO_x and particle number, not yet accounting for fuel consumption/CO₂ emissions [26].

The concept of the wind-averaged drag was introduced as early as the 1970s, with the aim of providing a more accurate representation of the real-world aerodynamic performance of a vehicle [27]. Initial studies were limited to vehicle operation at constant speed but quickly the focus shifted to driving schedules [28]. Several methods have been suggested so far with regards to the derivation of the wind-averaged drag coefficient, all of them considering different distributions of vehicle speed, wind speed, and wind direction relative to the vehicle movement [20]. Recent studies [22,29] applied the wind-averaged drag concept in the context of homologation cycles, including the WLTC as well as EPA drive cycles (FTP, HWFET, US06, SC03), relying on meteorological data for a particular region (e.g., the UK or Western Europe, in general) to derive a reference wind speed for the calculations.

The present study aims to expand on the above methodologies by implementing a Monte Carlo simulation methodology to generate a large number of random wind speed and direction profiles, based on appropriate statistical distributions derived from wind speed data for different cities around Europe. What is more, the scope of study is also expanded to include engine-out pollutants and energy consumption. The driving schedule used to assess the impact of the aforementioned wind profiles on performance and emissions is the worldwide harmonized light-duty vehicles test cycle—WLTC. This allows for a direct comparison to the type-approval case, i.e., no wind. Calculations are performed with the use of a detailed vehicle and engine model that was detailed in previous publications by the present research group [30,31].

The distinctive advantage of the proposed methodology that sets it apart from previous studies is that instead of relying on a suitable correction to be applied on the vehicle's drag coefficient across the whole cycle (in order to account for real-world aerodynamic performance), here the drag coefficient is calculated on a second-by-second basis, based on the instantaneous values of wind speed and direction, as well as vehicle speed. By doing so, useful correlations between wind speed and direction, on the one hand, and

fuel/energy consumption and emissions on the other, can be derived, allowing for the effect of engine transients on emissions (not limited to CO₂ but covering pollutants, too, namely soot and NO) to be further highlighted. What is more, a comparison between the methodology presented herein and the cycle-averaged drag coefficient method, presented in Howell et al. [22], is also carried out.

2. Materials and Methods

2.1. Impact of Ambient Wind on Aerodynamic Drag

When a vehicle is driven on the road with a vehicle speed V_V , in the presence of a natural wind blowing with velocity V_W and direction θ , the relative air speed V_R (Figure 1) is as follows:

$$V_R(V_V, V_W, \theta) = \sqrt{V_V^2 + V_W^2 + 2V_V V_W \cos \theta} \quad (1)$$

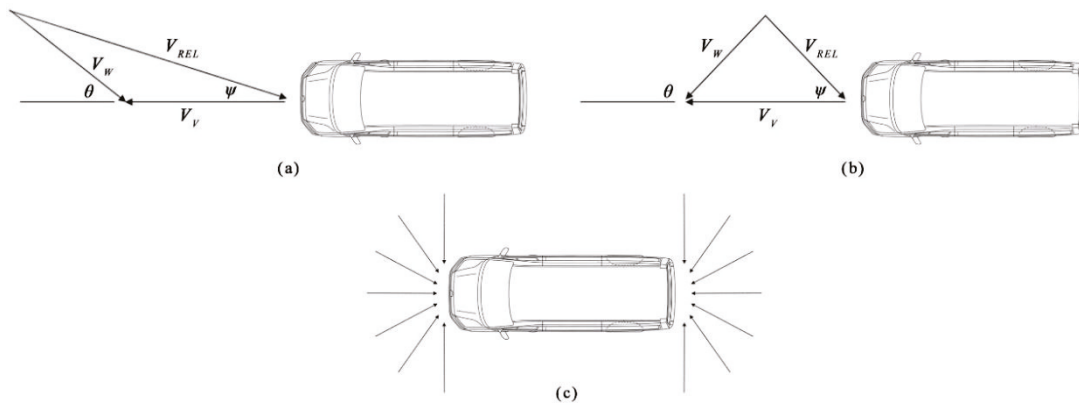


Figure 1. (a) Velocity triangle for headwind, (b) Velocity triangle for tailwind, (c) Headwinds and tailwinds approaching the vehicle equiprobably.

The relative air speed V_R creates an angle ψ with the vehicle direction of movement. This is called the yaw angle and is given as follows:

$$\psi(V_V, V_W, \theta) = \tan^{-1} \left(\frac{V_W \sin \theta}{V_V + V_W \cos \theta} \right) \quad (2)$$

Finally, the aerodynamic drag on the vehicle due to the presence of the ambient wind is

$$F_{aero}(V_V, V_W, \theta) = \frac{1}{2} \rho A C_d(\psi) V_R^2 \quad (3)$$

where ρ is the ambient air density, A is the vehicle frontal area, and $C_d(\psi)$ is the aerodynamic drag coefficient as a function of the yaw angle ψ .

The relative change in the vehicle drag coefficient vis-à-vis the yaw angle is shown in Figure 2, normalized with respect to the drag coefficient at zero yaw. The relevant data were drawn from Heisler [32]. It was assumed that the vehicle was symmetrical around the X-axis, and thus the $C_d(\psi)$ curve in Figure 2 was symmetrical around $\psi = 0^\circ$.

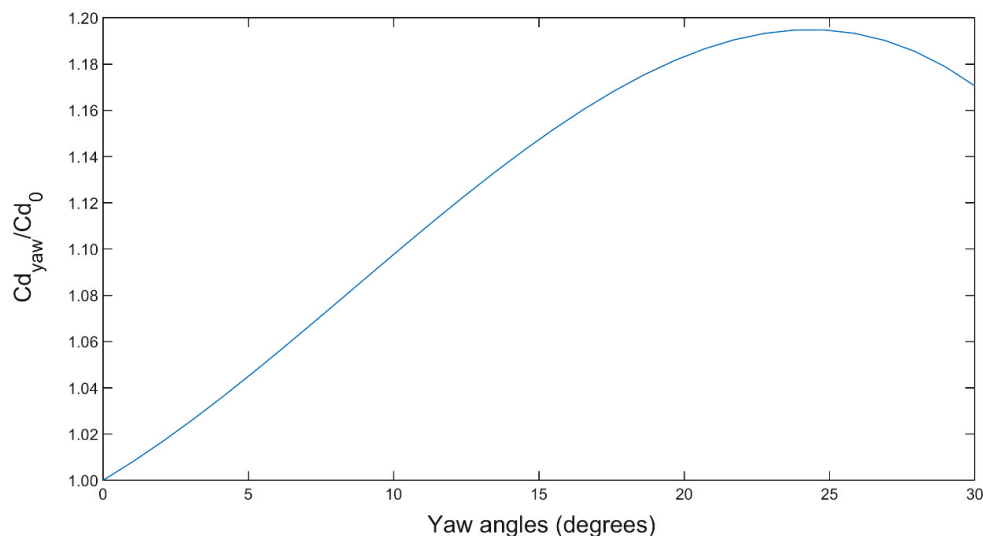


Figure 2. Increase in the drag coefficient with respect to the yaw angle.

It was noted that the increase of $C_{d_{\psi \neq 0^\circ}}$ over $C_{d_{\psi = 0^\circ}}$ reached a maximum value at $\psi \approx 24^\circ$, then started decreasing slowly. Following Sovran [28], we expected that as the yaw angle increased, the drag coefficient would eventually become equal to zero at some yaw angle $\psi_{Cd=0} < 90^\circ$. A further increase in yaw angle would then see the drag coefficient assuming negative values corresponding to an “aerodynamic push” on the vehicle. Unfortunately, most investigations on the change of drag coefficient with respect to yaw angle were limited to yaw angles of less than 30° , assuming that larger yaw angles were unlikely. In the context of the present study, a cap was imposed on instances of the WLTC where $\frac{V_W}{V_v} > 0.5$. This was to avoid errors from over-extrapolation, given the lack of data for large yaw angles.

2.2. Generation of Random Wind Profiles

Random wind profiles were generated using the Weibull distribution with respect to wind speed. The Weibull distribution is the most commonly used probability distribution function with respect to the assessment of the wind energy potential of a given area [33–35]. The Weibull probability density function is as follows:

$$f(V_W) = \frac{k}{c} \left(\frac{V_W}{c} \right)^{k-1} e^{-\left(\frac{V_W}{c} \right)^k} \quad (4)$$

where $f(V_W)$ is the probability of observing wind speed V_W , k is the dimensionless shape factor, and c (m/s) is the scale parameter. The mean wind speed can be calculated based on the k and c values as follows:

$$\overline{V_w} = c \Gamma \left(1 + \frac{1}{k} \right) \quad (5)$$

where Γ is the gamma function.

Data for the Weibull distributions used in this study were drawn from the European Wind Atlas [36]. The European Wind Atlas categorizes data for each site according to three different roughness classes over land. Those roughness classes correspond to different terrain surface characteristics and are defined by the relevant surface roughness length z_0 . Table 1 shows the roughness class categorization, including the relevant WLTC phase that can be assumed to exhibit similar terrain characteristics.

Table 1. Terrain surface characteristics, roughness classes, and roughness lengths for each WLTC phase (data from [35]).

Terrain Surface Characteristics	Roughness Class	z_0 (m)	WLTC Phase
Farmland with very few buildings, trees, etc., airport areas with building and trees	1	0.03	Extra high
Farmland with closed appearance	2	0.1	High
Suburbs, shelter belts	3	0.4	Medium
City center	4	1.0	Low

From the roughness classes presented in Table 1, the European Wind Atlas provided data for classes 1 to 3. Weibull distribution parameters for roughness class 4 (city center—low phase of the WLTC) were extrapolated from the tabulated values using Equations (6)–(8) [36], assuming a surface roughness length $z_0' = 1.0$:

$$c' = wc_a + (1 - w)c_b \quad (6)$$

$$k' = wk_a + (1 - w)k_b \quad (7)$$

$$w = \frac{\ln(z_{0b}/z_0')}{\ln(z_{0b}/z_{0a})} \quad (8)$$

In the above equations, subscripts a and b refer to roughness lengths and Weibull parameters for roughness classes 2 and 3, respectively.

Eight different sites were considered, all of them located near major urban centers that could be assumed to exhibit similar driving conditions to the WLTC phases. Those sites were picked from various regions across Europe, exhibiting different scale and shape parameters for the relevant Weibull probability distribution functions. It was generally observed that for sites in Northern Europe, the shape factor k was close to 2.00; therefore the one-parameter Rayleigh distribution could also be used instead of the two-parameter Weibull one [36].

Table 2 presents geographical data for the sites considered in this study, whereas Table 3 summarizes the scale and shape parameters applicable to each cycle phase and for each site. It is noted here that the altitude of each site was not considered in the calculations and is only listed here for reference. The same goes for other climatological data (e.g., temperature). Such a full-blown investigation of the wind drag components fell out of the scope of the present study.

Table 2. Geographical data for the eight sites considered (data from [35]).

Site	Latitude	Longitude	Altitude (m)
Site 1	52°18' N	04°46' E	−4
Site 2	37°54' N	23°44' E	28
Site 3	41°18' N	02°05' E	5
Site 4	52°28' N	13°24' E	48
Site 5	52°27' N	01°44' E	94
Site 6	55°38' N	12°40' E	5
Site 7	45°43' N	04°57' E	201
Site 8	45°26' N	09°17' E	103

Table 3. Weibull scale factor c and shape factor k for the eight sites considered (data from [35]).

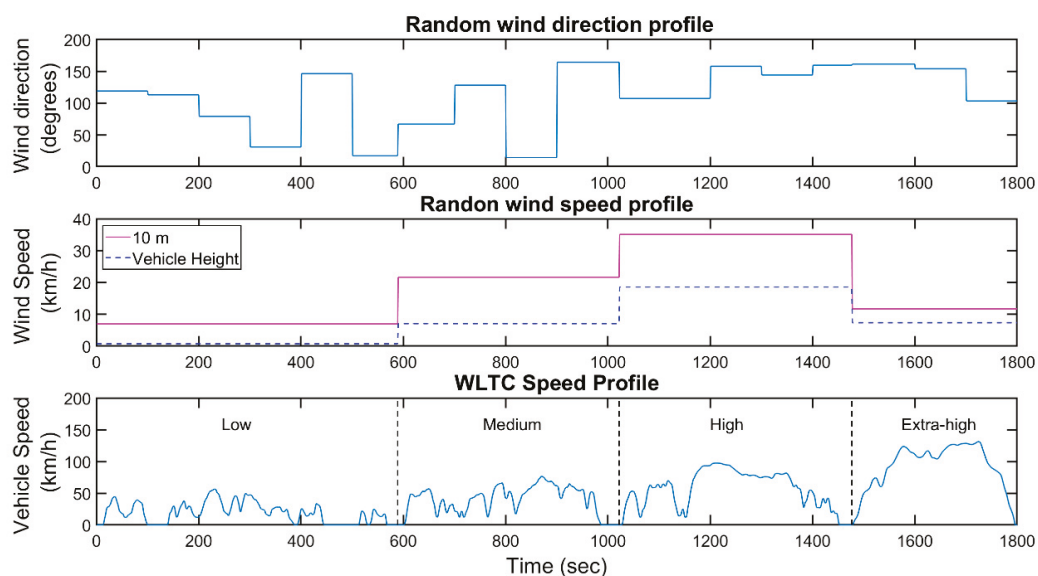
	Low		Medium		High		Extra High		CF
	c (m/s)	k	c (m/s)	k	c (m/s)	k	c (m/s)	k	
Site 1 (S1)	3.1	1.89	3.8	1.89	4.9	1.89	5.6	1.87	1.07
Site 2 (S2)	2.6	1.36	3.1	1.35	3.9	1.34	4.5	1.34	1.16
Site 3 (S3)	2.3	1.51	2.8	1.51	3.5	1.51	4.0	1.50	1.23
Site 4 (S4)	2.7	2.00	3.4	2.00	4.4	2.00	5.0	1.99	1.07
Site 5 (S5)	4.1	1.80	5.0	1.80	6.4	1.80	7.3	1.79	1.04
Site 6 (S6)	3.6	2.04	4.3	2.04	5.4	2.04	6.2	2.03	1.04
Site 7 (S7)	2.2	1.25	2.7	1.24	3.4	1.23	3.9	1.22	1.11
Site 8 (S8)	2.8	1.29	3.4	1.28	4.3	1.27	4.9	1.27	1.25

It is noted that these Weibull distribution parameters corresponded to a wind blowing at 10 m above ground. This wind speed was then corrected to the vehicle height using the logarithmic law [37,38]. Following Howell et al. [29], we considered z_{veh} to be equal to $0.4 \times VH$, where VH is the vehicle height.

$$\frac{V_{W_{veh}}}{V_{W_{10}}} = \frac{\ln(z_{veh}/z_0)}{\ln(z_{10}/z_0)} \quad (9)$$

With regards to wind direction, the uniform distribution was chosen with $\min = 0^\circ$ and $\max = 180^\circ$. This is consistent with most approaches regarding wind-averaged drag calculation that assume equiprobable wind direction in the range of 0° to 180° [20]. The same approach was taken by Khayyam [39] and validated via a Pearson chi-square test against empirical data.

Once the statistical distributions for wind speed and direction were defined, the generation of random profiles proceeded by picking a random wind speed V_W for each cycle phase from the relevant Weibull distribution for each of the sites examined. Then a random wind direction θ was picked for each of the phase segments. The low and medium phases were broken down to 10 and 8 segments, respectively, whereas for the high and extra-high phases, the number of segments was 5 and 3, respectively (see Figure 3). This was done in order to reflect the influence of traffic conditions, as well as road geometry for the different type of roads that each cycle phase represented (city center, extra-urban, rural, and highway).

**Figure 3.** Random WLTC wind profile.

Finally, the diurnal wind speed variation was also taken into account. The European Wind Atlas [36] presents data on the mean annual observed wind speeds at different hours of the day for each site, with a three-hour step. A light commercial vehicle is typically expected to be driven in the hours between 06:00 and 18:00; therefore a correction factor was applied to each random wind speed, following the below formula for each site:

$$CF_{diurnal} = \frac{(\bar{V}_W)_{06:00-18:00}}{(\bar{V}_W)_{24h}} \quad (10)$$

Correction factors for the wind diurnal variation are included in the rightmost column of Table 3. It was noted that sites in Southern European regions exhibited higher diurnal correction factors as opposed to Northern European ones.

2.3. Experimental Investigation and Engine Modeling

For each of the random wind profile WLTC trips, generated based on the methodology described above, instantaneous emissions from a diesel-powered, turbocharged, light-commercial vehicle were calculated with the use of an integrated engine and vehicle model. This model formed the basis of previous publications of the present research group [30,31] and is briefly discussed here for the sake of completeness.

The engine model was built based on a combined experimental/computational approach. The context of quasi-linear modeling was adopted for the most part, in a similar fashion to previous mapping-based studies [40,41]. The starting point for the model was the steady-state experimental investigation of the engine at hand; polynomial expressions were derived for each of the properties of interest vis-à-vis instantaneous engine load and speed [42]. With regards to engine power, as well as fueling and CO₂ emissions, this approach was expected to provide accurate results.

However, that was not the case for NO and soot emissions (also investigated in the present study), mostly due to the effect of turbocharger lag. The latter is strongly related to engine transient events (accelerations and load changes) and can result in significant overshoots for NO and soot emissions [43,44], which a mapping-based approach can easily overlook. Thus, on top of the steady-state mapping of the engine, a detailed transient investigation was also carried out, focusing on transient events similar to the ones encountered during a typical transient driving schedule. Fast-response analyzers were used, enabling the derivation of correction coefficients for both NO and soot based on the magnitude of the transient event [30].

NO was chosen for the present analysis since it forms the biggest part of NO_x emissions from diesel engines [45,46]. On the other hand, soot was studied as surrogate to particulate matter, which is very hard to measure instantaneously [47]. For all results presented herein, it was assumed that the engine was fully warmed up, i.e., no cold-start effects were taken into account. Lastly, it is noted that all emitted pollutants referred to engine-out conditions.

2.4. Integrated Engine and Vehicle Model

Coupled to the engine model was a detailed vehicle model capturing instantaneous vehicle longitudinal dynamics based on Equation (11) [32,48]:

$$(F_{tr})_i = (F_{roll})_i + (F_{aero})_i + (F_{grad})_i + (F_{in})_i = f_r m_V g \cos \alpha_i + \frac{1}{2} \rho A C_d(\psi_i) V_{R_i}^2 + m_V g \sin \alpha_i + MF m_V \left(\frac{dV_V}{dt} \right)_i \quad (11)$$

Tractive force F_{tr} needs to be exerted on the wheels by the engine in order for the vehicle to overcome the sum of forces applied to it, including rolling resistance F_{roll} , aerodynamic resistance, gradient resistance F_{grad} , and inertia resistance F_{in} . In the above equation, f_r is the rolling resistance coefficient, m_V is the vehicle mass, g is the gravitational acceleration, and α is the road slope angle. The rolling resistance coefficient f_r was calculated on a second-by-second basis using a detailed mass-spring and damper

model to account for tire hysteretic behavior, which is the biggest contributor to rolling resistance losses [49,50]. Tire micro-slippage, as well as tire windage losses, were also considered in the estimation of the rolling resistance coefficient [31].

The gradient angle a_i was set to zero for the whole duration of the driving schedule in order to isolate ambient wind effects. The term MF is the mass factor and was included in the calculation of the inertia resistance to represent the inertia of the driveline rotating components. It is calculated using Equation (12) [48]:

$$MF_i = \frac{m_V + m_r}{m_V} = 1 + 0.04 + 0.0025 (i_{g_i} i_b)^2 \quad (12)$$

With regards to instantaneous engine speed N_{e_i} , we have

$$\omega_{w_i} = \frac{V_{V_i}}{R_d} \rightarrow \frac{2\pi N_{e_i}}{60 i_b i_{g_i}} = \frac{V_{V_i}}{R_d} \rightarrow N_{e_i} = \frac{V_V i_b i_{g_i}}{2 \pi R_d} \times 60 \cdot 10^3 \quad (13)$$

where ω_{w_i} is the instantaneous wheel angular speed, R_d is the tire dynamic radius, i_b is the back-axle transmission ratio, and i_{g_i} is the gear transmission ratio.

The tire dynamic radius R_d is defined as the ratio between the vehicle forward speed and the wheel angular speed. Its value sits somewhere between the tire unloaded or geometric radius and the tire static radius. The latter is a function of tire dimensions as well as inflation pressure [51,52], whereas the former is a design feature of the tire. The three radii are related as per Equation (14) [53]:

$$R_d = \frac{V_{V_i}}{\omega_{w_i}} \approx R_g - \frac{R_g - R_{st}}{3} \quad (14)$$

The instantaneous engine load T_{e_i} is

$$T_{e_i} = \frac{F_{tr_i} R_d}{i_b i_{g_i} \eta_{tf_i}} + \frac{P_{aux}}{N_{e_i}} \quad (15)$$

P_{aux} is the auxiliaries power (lighting, A/C, radio, etc.), which was also taken into account.

Instantaneous drivetrain losses η_{tf_i} are estimated based on Equation (16) below [54]:

$$\eta_{tf_i} = \left[0.96 - 0.000707 V_{V_i} - 0.000029 V_{V_i}^2 \right] \times \left\{ 0.998 [1 - 0.007 (NG - I_i)] - 0.0001965 \left(2.08^{NG - I_i} V_{V_i} \right) \right\} \quad (16)$$

where NG is the total number of forward gears used and I_i is the gear used at each second of the cycle. The latter was defined following the procedure detailed in the WLTP technical specifications [21].

Once the instantaneous engine speed and torque were defined, fuel/energy consumption was calculated from the steady-state investigation of the engine, as were the steady-state values for NO and soot. Based on the load and speed change, transient correction coefficients were applied to the estimation of NO and soot transient overshoots that, together with the steady-state values, formed the overall NO and soot emissions [30]. Finally, instantaneous values for fuel/energy consumption, CO₂, NO, and soot emissions were translated into cumulative, distance-specific values for the WLTC, as well as for the different cycle phases.

The data of the engine and vehicle used for the analysis are summarized in Table 4.

Table 4. Engine and vehicle data.

Engine						
Engine type	Four-stroke, in-line, six-cylinder, turbocharged, aftercooled, direct-injection diesel engine					
Bore/Stroke	97.5 mm/133 mm					
Compression ratio	18:1					
Maximum power	177 kW @ 2600 rpm					
Moment of inertia	0.87 kg m ²					
Vehicle-LCV						
Gross vehicle mass/Kerb mass	3500 kg/2614 kg					
Frontal area	4.976 m ²					
Dimensions (W/H/L)	1740 mm/2860 mm/7012 mm					
Drag coefficient	0.28 @ zero yaw angle					
Gear ratios	1st	2nd	3rd	4th	5th	6th
	4.655	2.480	1.440	1.000	0.796	0.685
Back-axle ratio	3.50:1					
Auxiliary power	750 W					
Tire dimensions	225/65 R16					
Tire inflation pressure	260 kPa					
Tire efficiency class	C					

2.5. Monte Carlo Simulation Iterations

The purpose of this study was to examine the expected increase in fuel consumption and emissions, vis-à-vis the type-approval case (i.e., no wind), for a light-commercial vehicle driven in an ambient wind environment. Thus, the results are expressed as the percentage rise over the reference case. In order to do so with a certain level of confidence, the central limit theorem (CLT) was implemented [55] in order to define the necessary number of random-wind WLTC trips to be executed.

The CLT states that for a given sample of simulation runs n , the sample mean

$$\bar{Y} = \frac{1}{n} \sum_{j=1}^n Y_j \quad (17)$$

has a normal distribution with mean $\mu_{\bar{Y}} = \mu$ and variance $S_{\bar{Y}}^2 = \frac{S^2}{n}$, where μ and S^2 are the mean and variance of all possible random-wind WLTC trips, respectively. Typically, the variance S^2 is unknown, however it can be estimated with the relevant sample variance for a sufficiently large sample size n (i.e., number of random-wind WLTC trips):

$$S^2 = \frac{1}{n-1} \sum_{j=1}^n (Y_j - \bar{Y})^2 \quad (18)$$

Thus, the $(1-\alpha)100\%$ confidence interval can be defined as follows:

$$\bar{Y} \pm z_{1-\frac{\alpha}{2}} \frac{S}{\sqrt{n}} \quad (19)$$

where $z_{1-\frac{\alpha}{2}} = 1.96$ for a 95% confidence interval ($\alpha = 5$).

The confidence interval can also be expressed as a percentage of the sample mean $\bar{Y}$:

$$CI_{95\%}(\%) = 100 \times \frac{z_{1-\frac{\alpha}{2}} \frac{S}{\sqrt{n}}}{\bar{Y}} \quad (20)$$

With the aim of estimating $\bar{Y}$ with a $CI_{95\%}(\%) \leq 5\%$ (i.e., with an accuracy of at least $\pm 5\%$), an initial sample was taken, of sample size $n = 100$, to get a rough estimate of mean $\bar{Y}$ and variance S^2 . Then these values were imported into Equation (20), and we

solved for the number of iterations n required for the desired accuracy. For all of the sites considered, a total number of 5000 random-wind WLTC trips was adequate for delivering the desired accuracy.

It is noted that in the analysis above, the individual outcomes of each random-wind trip Y_j , $j = 1, \dots, n$ for all properties of interest are calculated as follows:

$$\text{Fueling} : Y_j^{FC} = 100 \times \left[\frac{(FC)_j}{(FC)_{ref}} - 1 \right] \quad (21)$$

$$\text{Specific Energy Consumption} : Y_j^{EC} = 100 \times \left[\frac{(EC)_j}{(EC)_{ref}} - 1 \right] \quad (22)$$

$$\text{NO Emissions} : Y_j^{NE} = 100 \times \left[\frac{(NE)_j}{(NE)_{ref}} - 1 \right] \quad (23)$$

$$\text{Soot Emissions} : Y_j^{SE} = 100 \times \left[\frac{(SE)_j}{(SE)_{ref}} - 1 \right] \quad (24)$$

Therefore, the final estimates for the expected change in emissions, compared to the reference case, are as follows:

$$\text{Fueling} : \bar{Y}^{FC} = \frac{1}{n} \sum_{j=1}^n Y_j^{FC} \quad (25)$$

$$\text{Specific Energy Consumption} : \bar{Y}^{EC} = \frac{1}{n} \sum_{j=1}^n Y_j^{EC} \quad (26)$$

$$\text{NO Emissions} : \bar{Y}^{NE} = \frac{1}{n} \sum_{j=1}^n Y_j^{NE} \quad (27)$$

$$\text{Soot Emissions} : \bar{Y}^{SE} = \frac{1}{n} \sum_{j=1}^n Y_j^{SE} \quad (28)$$

2.6. Cycle-Averaged Drag Coefficient Method

The cycle-averaged drag coefficient (CADC) method has also been discussed in the context of this study [22,29]. The method relies on the wind-averaged drag coefficient methodology, which considers certain values of vehicle speed, as well as wind speed and direction. In the case of a driving schedule, such as the WLTC 3-2 examined here, for each cycle trace the instantaneous wind-averaged drag coefficient is calculated following Equation (29), considering equiprobable wind direction angle θ :

$$C_{dwi} = \frac{1}{\pi} \int_0^\pi C_d(\psi_i) \left(\frac{V_{Ri}}{V_{Vi}} \right)^2 d\theta \quad (29)$$

The relative air speed V_{Ri} , as well as the yaw angle ψ_i , are calculated following Equation (1) and Equation (2), respectively. Having calculated the wind-averaged drag at each cycle trace, the cycle-averaged drag coefficient is then calculated for the whole duration T of the cycle:

$$C_{CADC} = \frac{\int_0^T C_{dwi} V_{Vi}^3 dt}{\int_0^T V_{Vi}^3 dt} \quad (30)$$

The vehicle aerodynamic drag to be used in Equation (11) is then:

$$F_{aero}(V_V, C_{CADC}) = \frac{1}{2} \rho A C_{CADC} V_V^2 \quad (31)$$

In order to get to the cycle-averaged drag coefficient C_{ADC} , we needed a reference wind speed, which we obtained by considering the mean wind for each site, based on the relevant Weibull distribution parameters (Table 3).

3. Results and Discussion

3.1. Simulation Results

First, the reference case was simulated assuming an unloaded vehicle (kerb mass plus driver) and zero wind (zero road gradient, too). The results for the reference case are listed in Table 5, including a phase-by-phase breakdown of the cycle.

Table 5. Distance-specific results for the type-approval case (zero wind).

Property	Fueling (L/100 km)	CO ₂ (g/km)	NO (g/km)	Soot (g/km)	Specific Energy Consumption (kWh/100 km)
WLTC 3-2	13.14	336.82	5.84	10.53	31.34
Low	19.89	509.71	9.57	24.59	33.74
Medium	13.08	335.04	6.96	18.30	30.04
High	11.15	285.83	4.69	8.37	27.30
Extra high	12.38	317.24	4.78	4.78	34.70

The estimated increase in fuel/energy consumption and emissions, vis-à-vis the zero-wind case, is tabulated in Table 6 for all sites, including the relevant accuracy of the estimate.

Table 6. Monte Carlo simulation method results (mean values and standard deviations are %; all other values correspond to rise % over the reference case).

Site	Fueling/CO ₂	NO	Soot	Specific Energy Consumption
Site 1 (S1)	1.09 ± 4.04%	0.83 ± 4.11%	0.29 ± 2.67%	1.77 ± 4.35%
Site 2 (S2)	1.02 ± 4.53%	0.77 ± 4.58%	0.27 ± 2.98%	1.67 ± 4.82%
Site 3 (S3)	0.93 ± 4.30%	0.71 ± 4.34%	0.24 ± 2.85%	1.49 ± 4.44%
Site 4 (S4)	0.97 ± 4.03%	0.74 ± 4.12%	0.25 ± 2.53%	1.56 ± 4.38%
Site 5 (S5)	1.50 ± 3.84%	1.14 ± 3.85%	0.38 ± 2.65%	2.52 ± 3.99%
Site 6 (S6)	1.20 ± 3.90%	0.91 ± 3.97%	0.32 ± 2.54%	1.97 ± 4.17%
Site 7 (S7)	0.84 ± 4.29%	0.64 ± 4.32%	0.22 ± 2.84%	1.36 ± 4.65%
Site 8 (S8)	1.29 ± 3.89%	0.97 ± 3.88%	0.32 ± 2.73%	2.15 ± 4.07%
Mean	1.10	0.83	0.28	1.80
Standard Deviation	0.22	0.16	0.05	0.38

Figure 4 plots the percentage increase against the mean wind for each site, with the aim of providing insight into the relation between the two. This mean wind for each site was taken to be the Weibull mean (see Equation (5)) for the extra-high phase of the cycle at 10 m above ground. The relevant diurnal wind speed correction factor CF (Equation (10)) was also taken into account.

The windiest site was site 5, with a mean wind speed of 24.2 km/h, whereas the least windy one was site 7, which exhibited a mean wind speed of 14.6 km/h. There was a strong correlation between the percentage rise in emissions and mean wind speed. Specific energy consumption was the property most influenced from the presence of ambient wind, with the percentage increase, vis-à-vis the zero-wind case, being equal to 1.80% on average (max = 2.52%, min = 1.36%, S = 0.38%). Fuel consumption (hence CO₂ emissions) followed, with the average increase being 1.10% (max = 1.50%, min = 0.84%, S = 0.22%), whereas NO emissions were modestly increased by 0.83% (max = 1.14%,

min = 0.64%, $S = 0.16\%$). Soot was the least influenced property, with its average rise being equal to 0.28%.

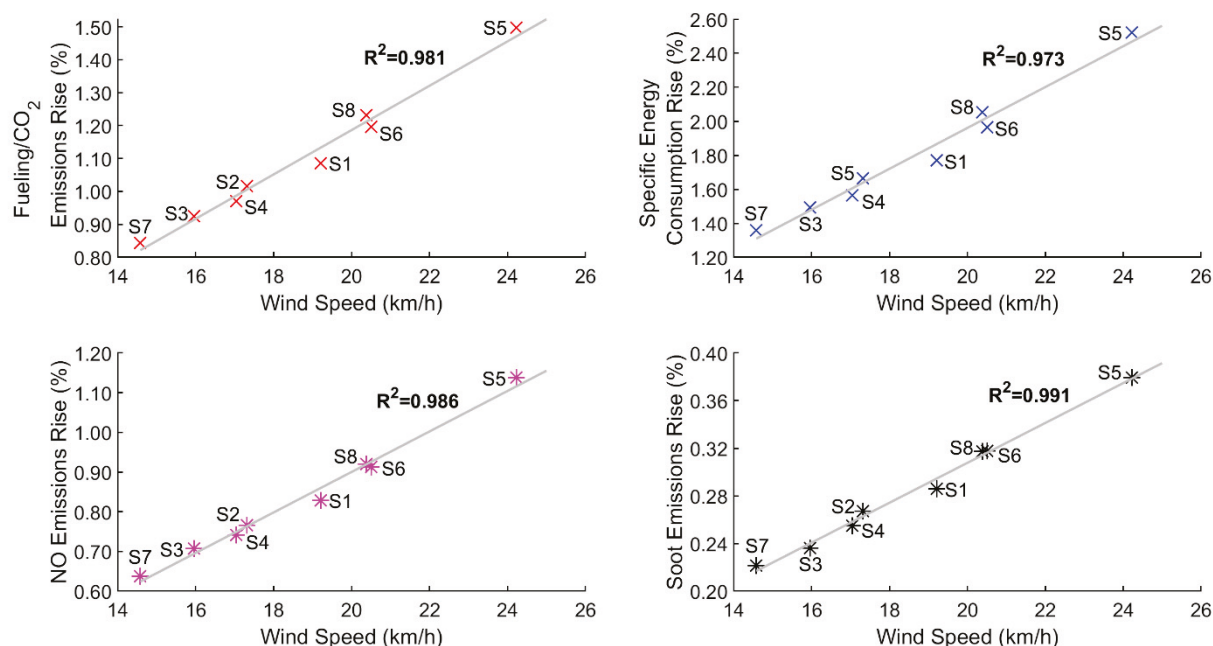


Figure 4. Expected rise in consumption and emissions vs. mean wind speed.

Figure 5 plots the increase in all properties of interest with respect to the Weibull scale parameter c (neglecting CF), and Figure 6 does the same, this time with regards to the Weibull shape parameter k . The increase in fuel/energy consumption, as well as NO and soot emissions, correlated well with the scale parameter c (Figure 5), whereas that was not the case at all when it came to the shape parameter k (Figure 6).

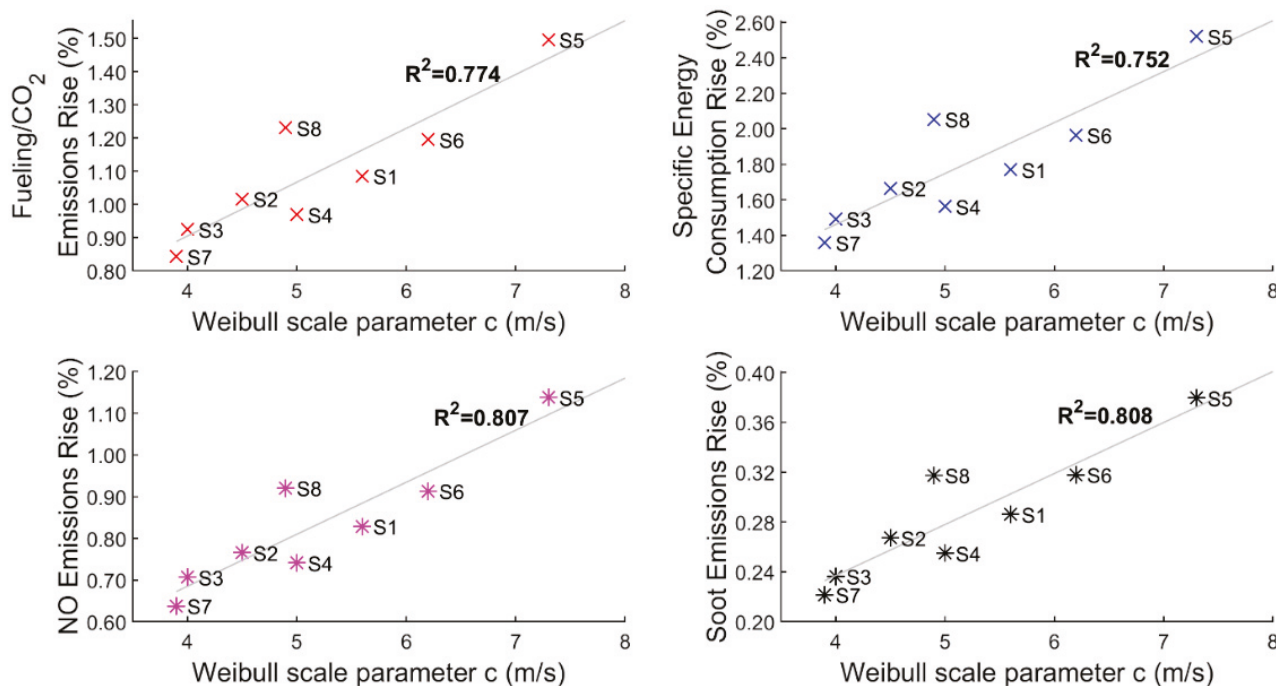


Figure 5. Expected rise in fuel/energy consumption and emissions vs. Weibull scale parameter c .

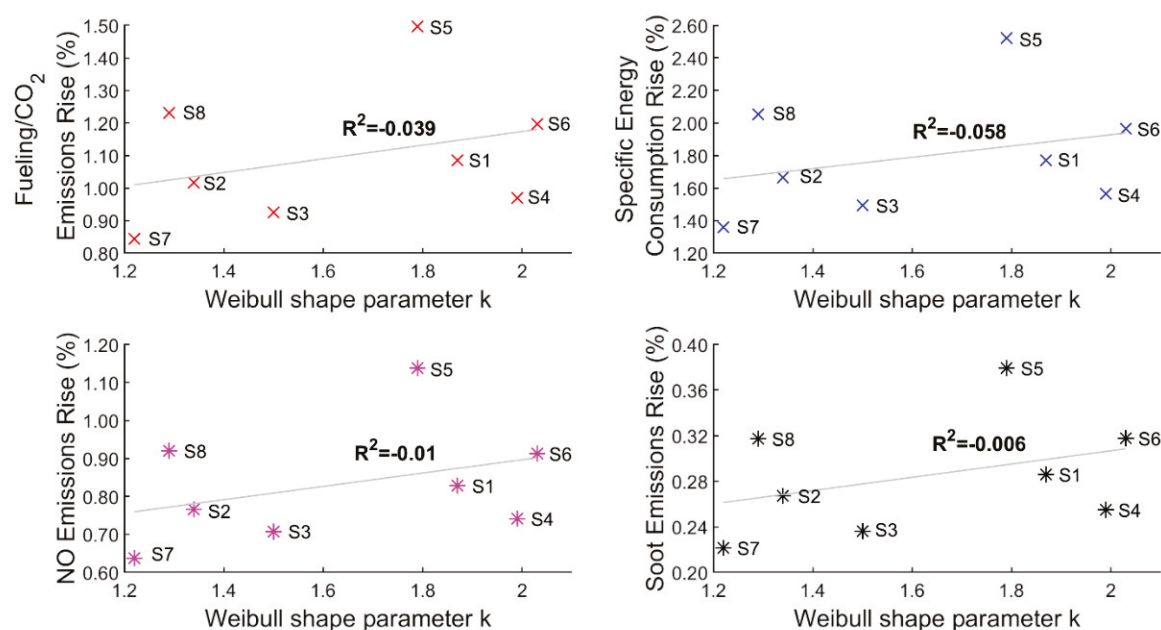


Figure 6. Expected rise in fuel/energy consumption and emissions vs. Weibull shape parameter k .

3.2. Correlation of Performance and Emissions with Wind Speed and Direction for Individual Trips

The above results give a solid estimate of how much the real-world values for fuel consumption and emissions are expected to vary from the type-approval ones, if the effect of ambient wind is accounted for. In other words, this is the increase that we expect to see from a vehicle being driven in said wind environment over a long period of time. This is not to be confused with the expected increase (or decrease) in consumption and emissions due to wind blowing at a certain speed and direction for an individual trip.

Figures 7–14 provide more insight into the latter case. Figures 7–10 plot the percentage change in all properties vis-à-vis wind speed, whereas Figures 11–14 plot the same change, this time with respect to mean wind direction. The results showcased in the above plots were derived from site 2 and were broken down in the different phases of the WLTC, with the aim of providing more insight into the effect of ambient wind on different vehicle speeds (city center as opposed to highway, etc.).

It is reminded here that the generation of random wind profiles proceeds by picking a single value for wind speed for each cycle phase, whereas for wind direction, each phase is broken down into several segments. Therefore, wind speed in Figures 7–10 is the wind speed blowing for the whole duration of each cycle phase, whereas wind direction in Figures 11–14 refers to the mean wind direction for the same cycle phase. Idling and deceleration instances of the cycle were discarded in the calculation of mean wind direction, given that wind drag was not contributing to the engine load for the above cases. Moreover, in Figures 7–10, “negative” winds correspond to a predominantly tailwind environment (i.e., $\theta > 90^\circ$), whereas “positive” winds reflect a predominantly headwind environment ($\theta < 90^\circ$). Finally, it is noted that the wind speed referenced in Figures 7–10 is the wind blowing at vehicle height and should not be confused with the wind speed at 10 m above ground, referenced in Figure 4.

It is no surprise that the extra-high phase of the WLTC (Figures 7 and 11) exhibited the biggest percentage increases for all properties. Not only were vehicle speeds higher, but the prevailing wind speeds were also higher, due to the different boundary layer characteristics (see Equation (9)). For a wind blowing at 30 km/h (approximately 50 km/h at 10 m reference height) the rise in fueling/ CO_2 emissions reached 25%, whereas for the specific energy consumption, the equivalent rise was almost 40%. NO emissions for the same case rose up to 21%. Finally, soot, although the most insensitive property with

respect to wind for the current engine, still showcased a considerable increase, almost 13%, for the same wind speed. A similar picture is drawn when looking at the high phase of the cycle (Figure 8). For wind speeds above 20 km/h at vehicle height (approximately 40 km/h at 10 m above ground), the increase in fueling/ CO_2 exceeded 10%, and the same applied to NO emissions.

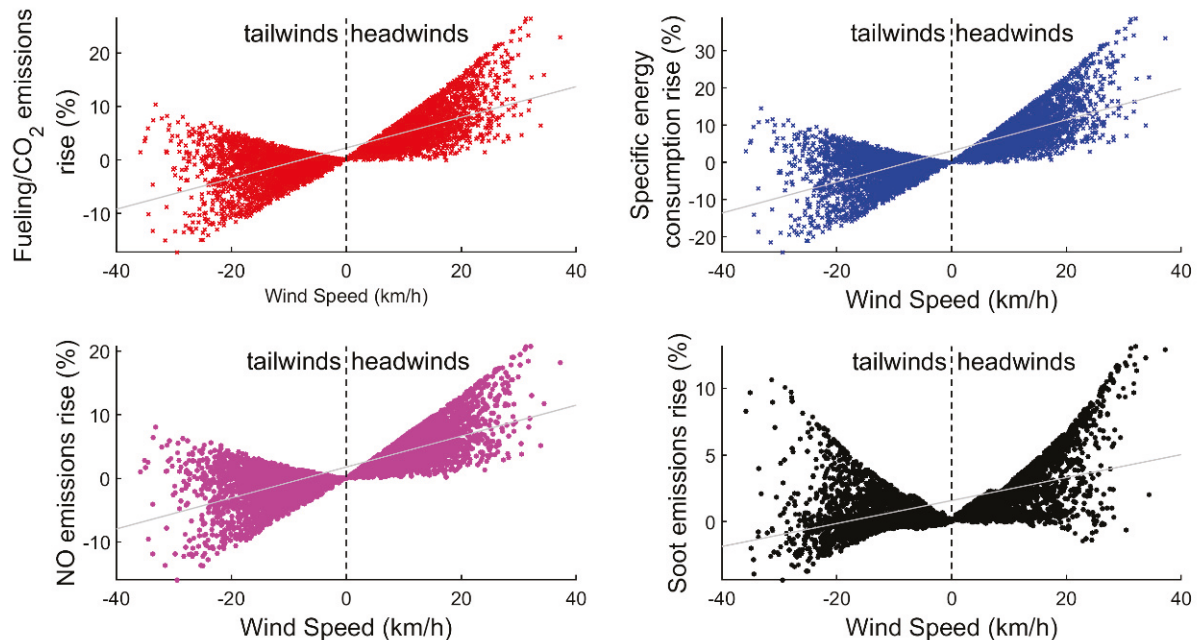


Figure 7. Percentage increase in consumption and emissions vs. wind speed for the extra-high phase of the WLTC.

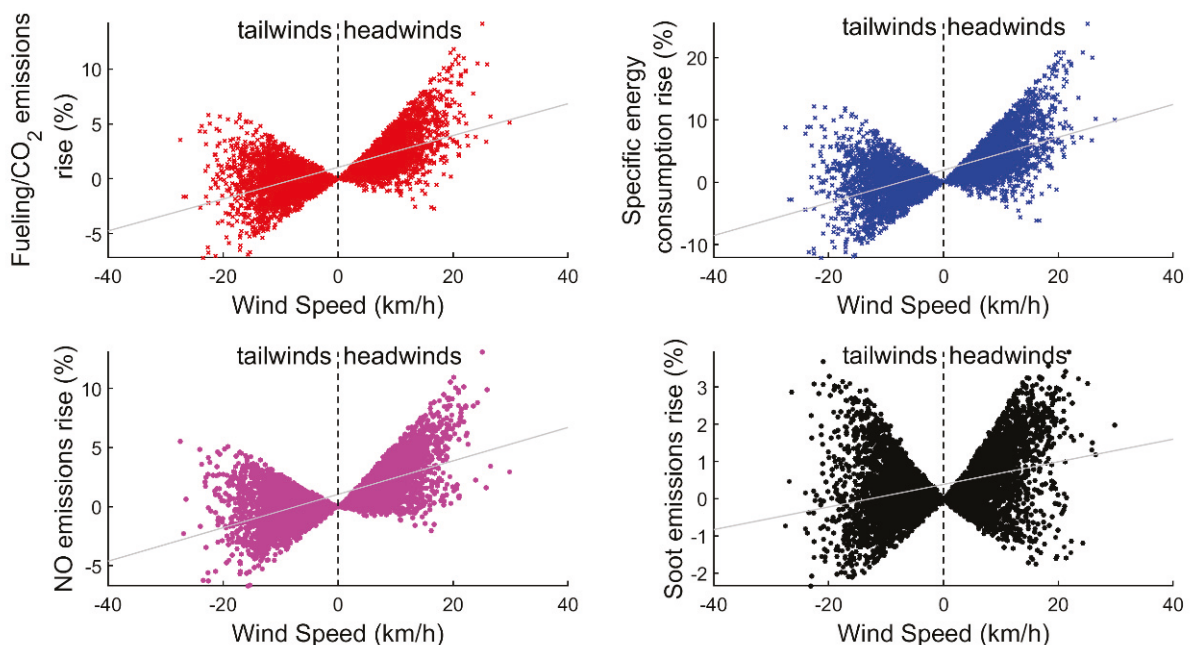


Figure 8. Percentage increase in consumption and emissions vs. wind speed for the high phase of the WLTC.

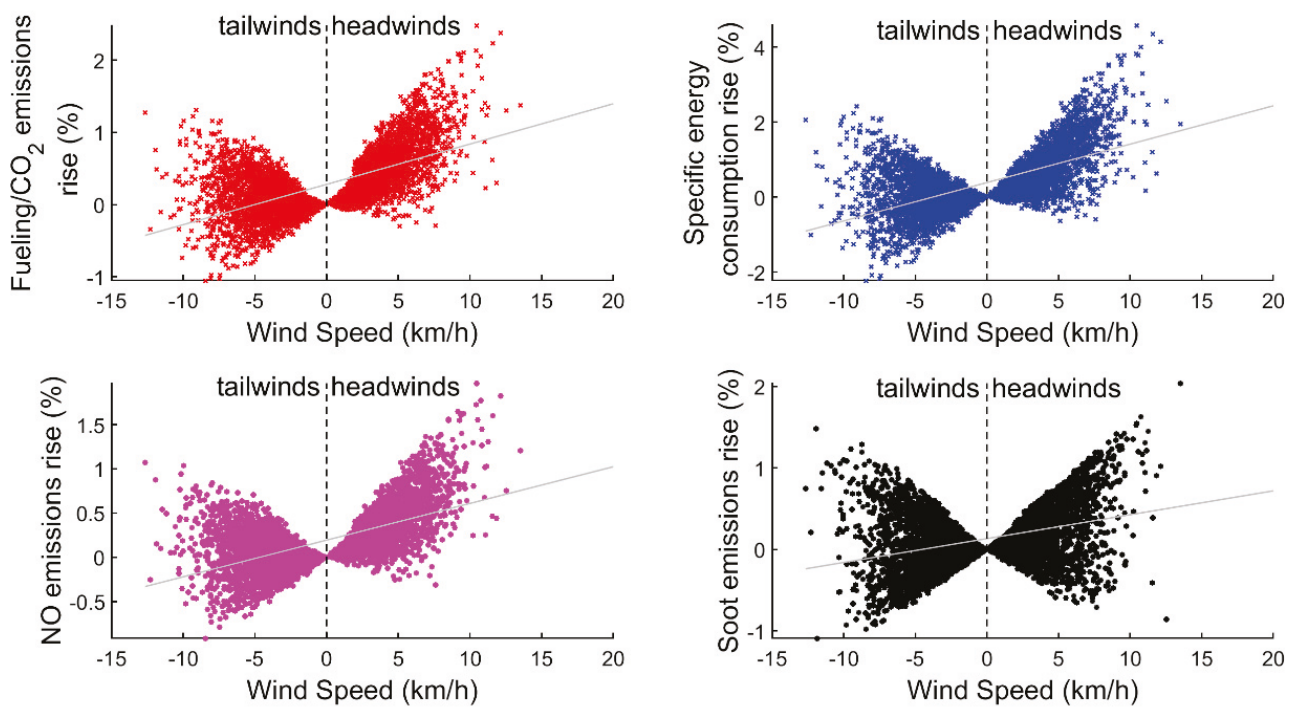


Figure 9. Percentage increase in consumption and emissions vs. wind speed for the medium phase of the WLTC.

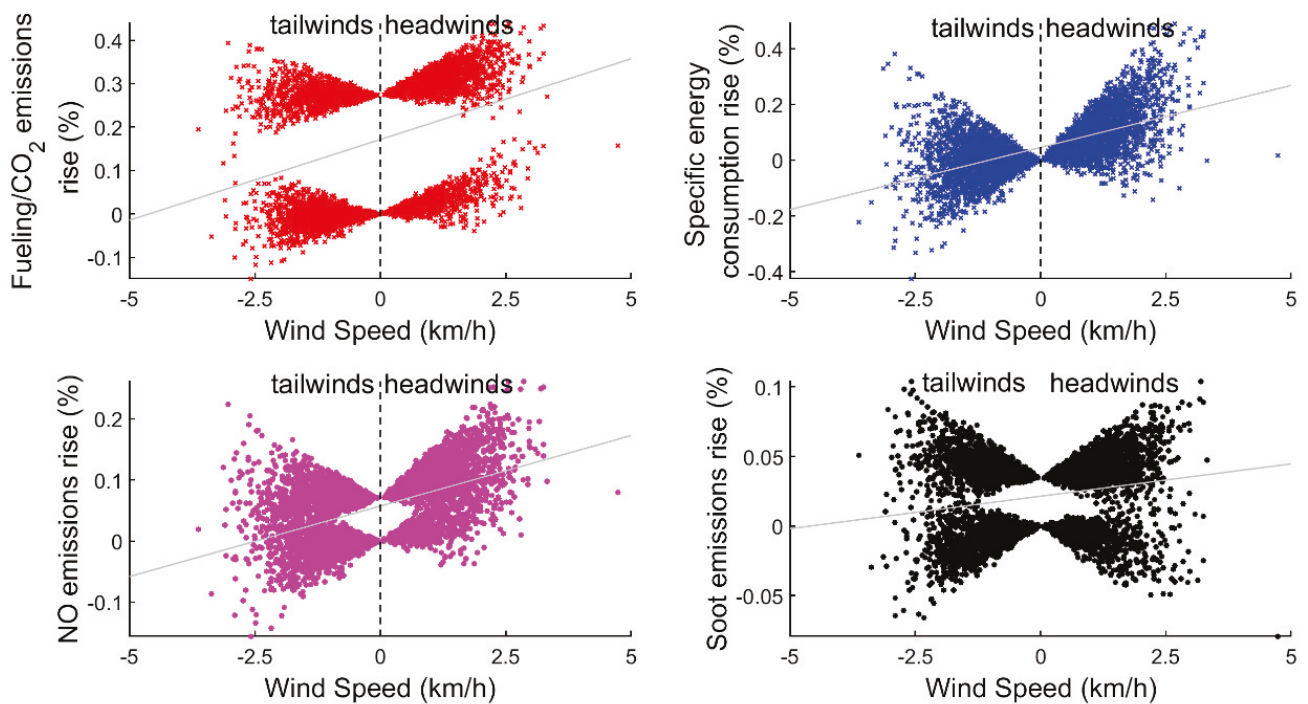


Figure 10. Percentage increase in consumption and emissions vs. wind speed for the low phase of the WLTC.

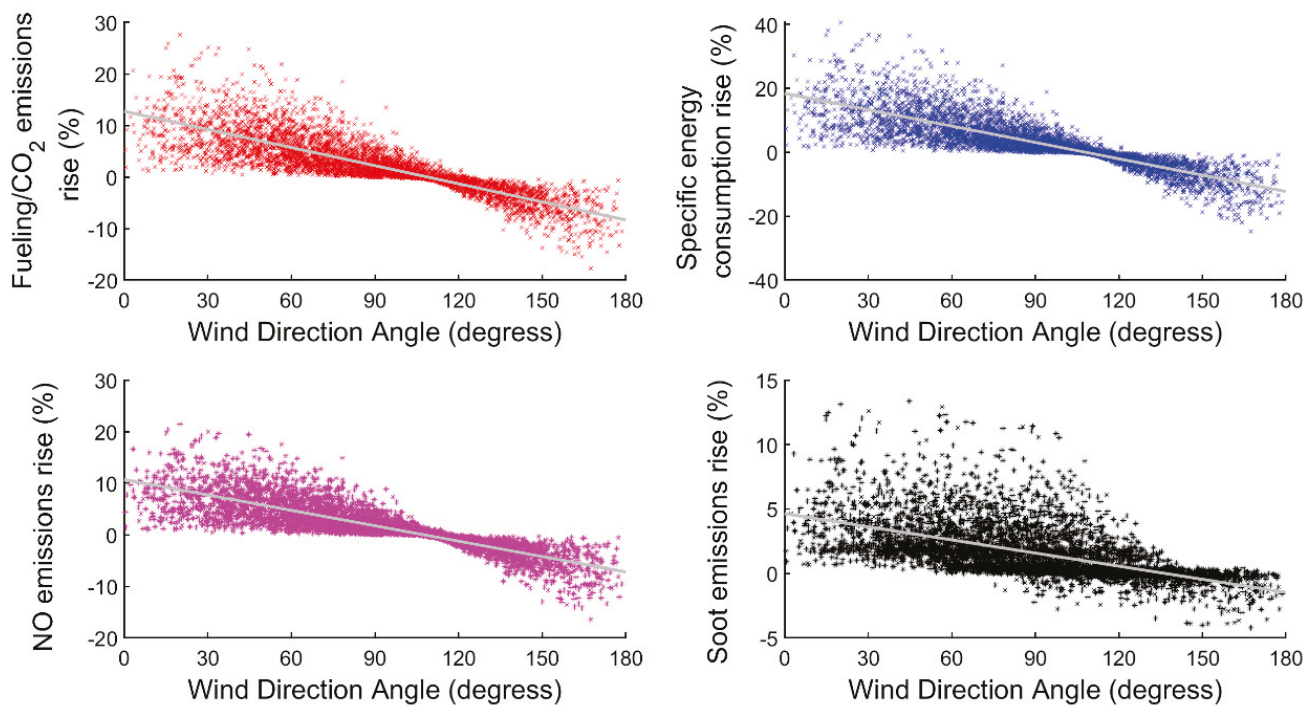


Figure 11. Percentage increase in fuel/energy consumption and emissions vs. mean wind direction for the extra-high phase of the WLTC.

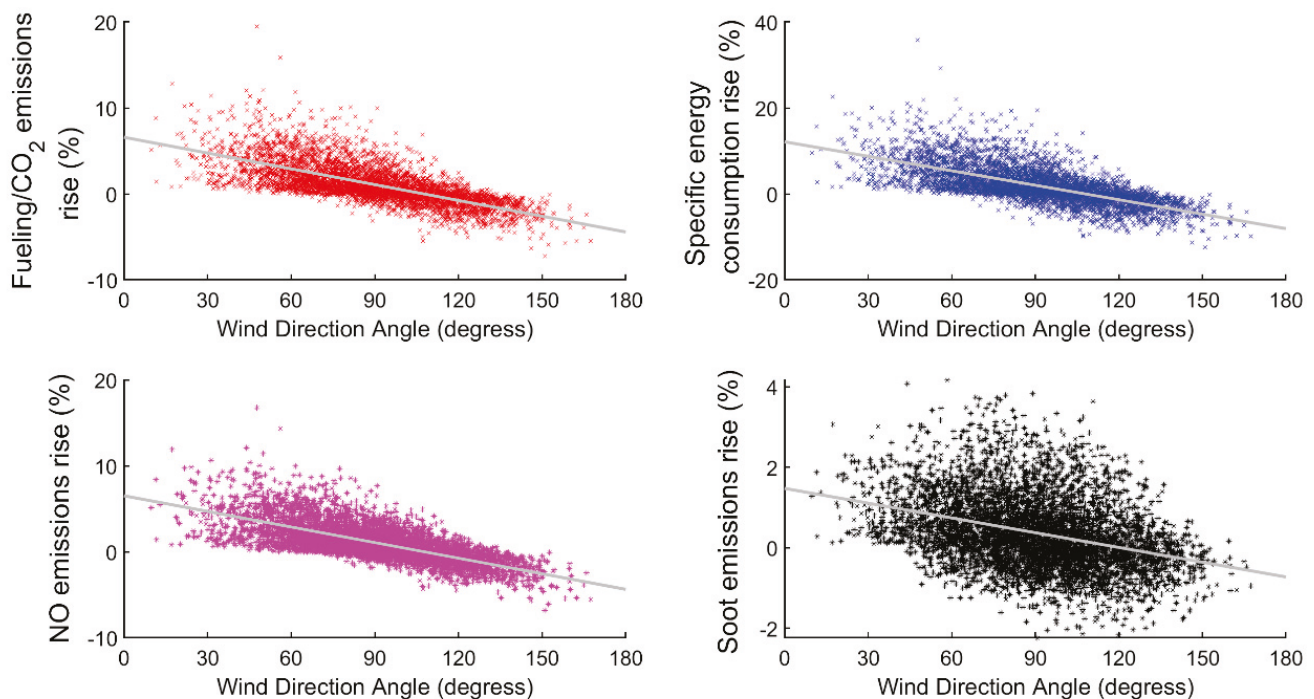


Figure 12. Percentage increase in fuel/energy consumption and emissions vs. mean wind direction for the high phase of the WLTC.

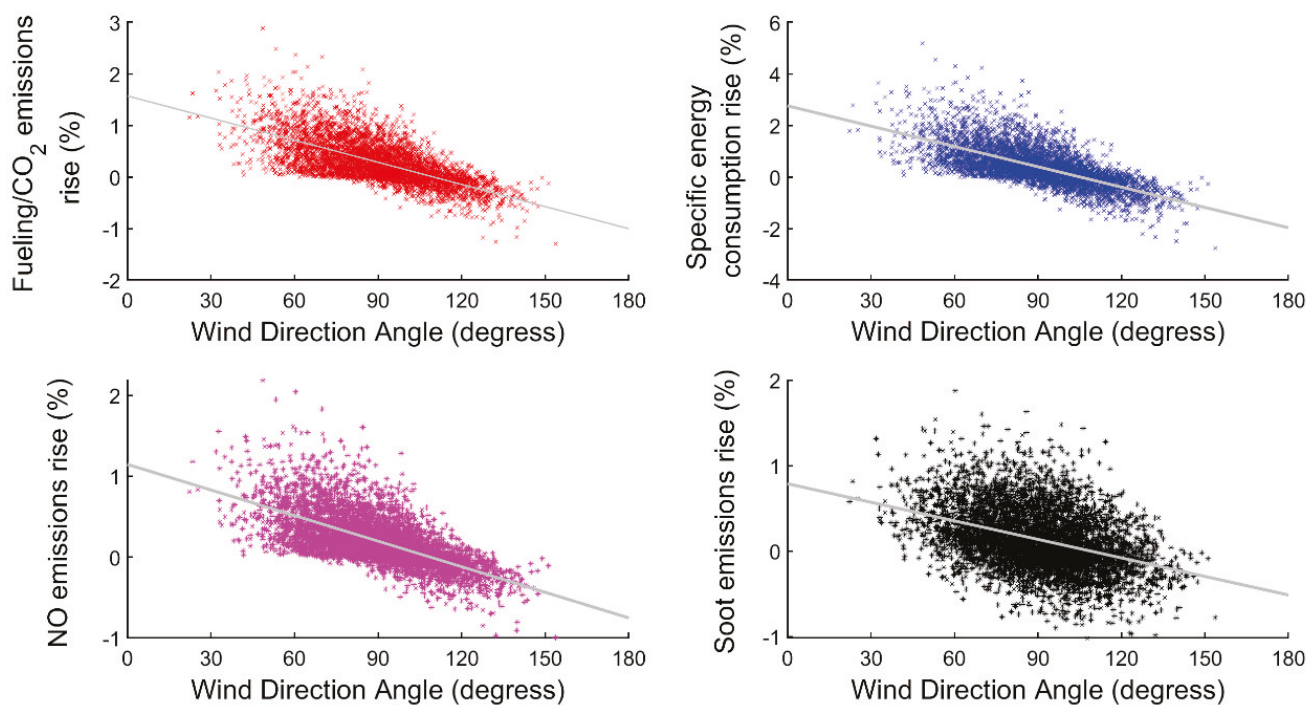


Figure 13. Percentage increase in fuel/energy consumption and emissions vs. mean wind direction for the medium phase of the WLTC.

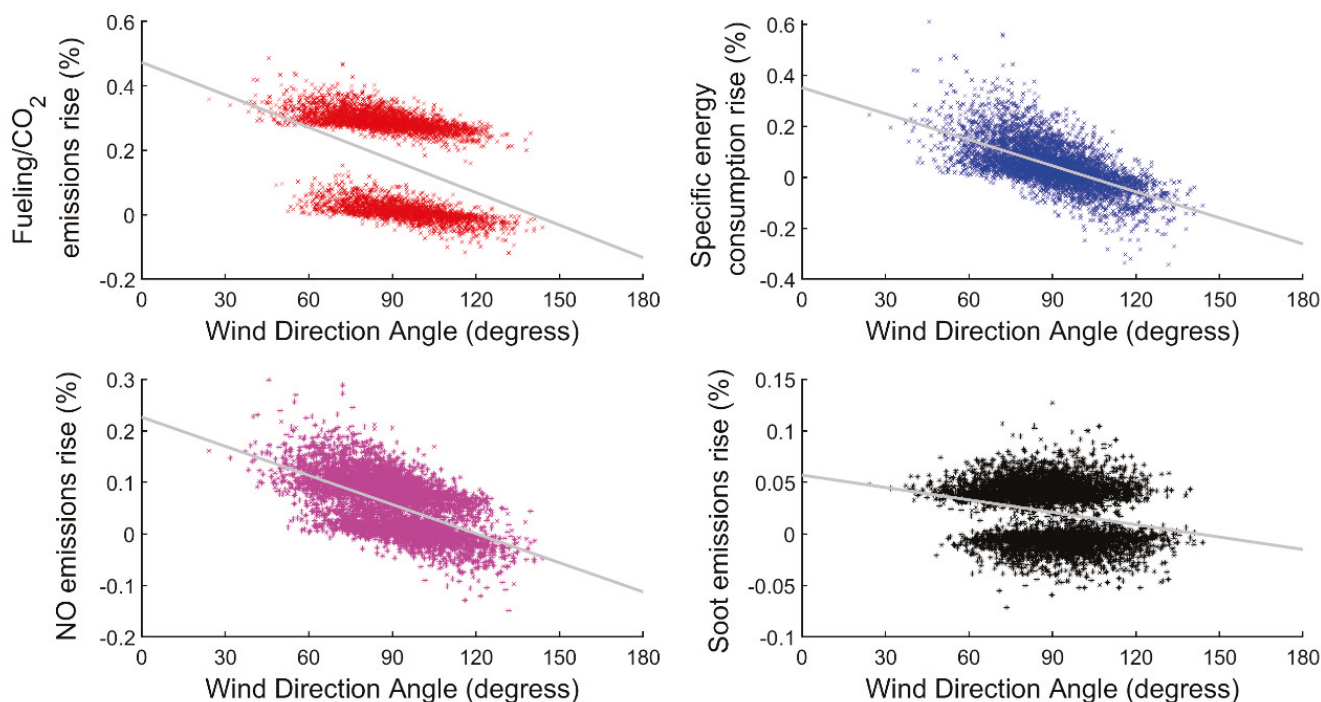


Figure 14. Percentage increase in fuel/energy consumption and emissions vs. mean wind direction for the low phase of the WLTC.

Focusing on NO emissions, it is noted that higher vehicle speeds, when coupled with strong headwinds, led to increased engine load and therefore higher in-cylinder temperatures, thus favoring the formation of NO [45,56]. The relation between NO emissions and engine load is further illustrated in Figure 15, where the increased NO

emissions in the extra-high phase were attributed to the higher operating points, vis-à-vis the reference case, for engine speeds above 2000 rpm.

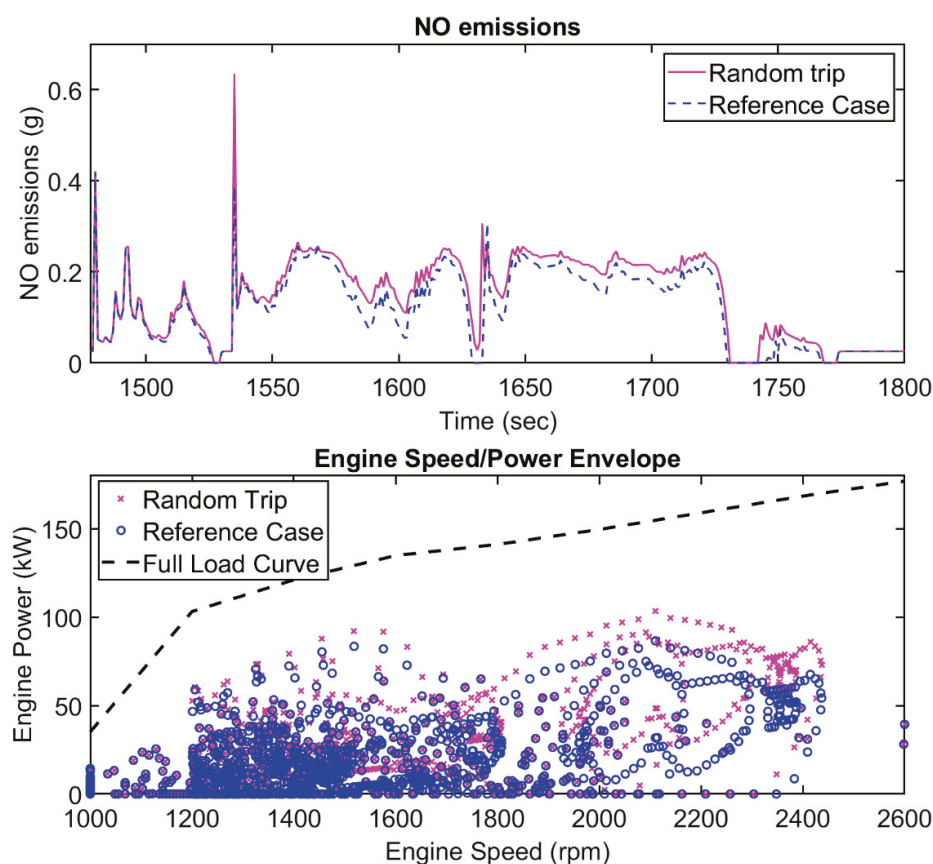


Figure 15. Engine speed /power envelope (WLTC) and corresponding NO emissions (extra-high phase).

On the other hand, when it comes to soot emissions, it must be clarified that the stated increase happened in a backdrop of relatively low soot emissions for the extra-high phase of the cycle (see Table 5). Therefore, transient events, although instantaneous, can have a relatively big impact on overall soot emissions [57]. It is noted that soot emissions are mostly linked to accelerations from standstill [43,44], typical during city-center driving, where both vehicle and wind speeds (thus aerodynamic drag) are low.

As we moved to the first phases of the cycle, wind speeds, as well as vehicle speeds, were lower and thus the relevant change in all properties of interest was more modest. The maximum rise over the reference case for NO emissions in the low phase of the cycle (Figure 10) was approximately 0.25%, whereas for the soot the maximum rise it was 0.1%. Parameters such as road slope, tire inflation pressure, or auxiliary usage were more prominent in terms of their effect on consumption and emissions for the specific vehicle speed range compared to aerodynamic drag [15,31].

On the other hand, Figures 11–14 showcase a generally declining trend for consumption and emissions vis-à-vis wind direction angle. As we gradually moved from a predominantly headwind environment to a predominantly tailwind environment, wind drag on the vehicle and thus engine load and associated fuel/energy consumption and emissions were lowered. However, the decrease in consumption and emissions as we moved towards tailwinds was not enough to offset the relevant increase due to headwinds. Considering the extra-high phase results (Figure 11), where the effect of wind drag was more pronounced, we noted that the maximum increase was 28% for fueling/CO₂ emissions, 22% for NO emissions, 13% for soot, and 41% for specific energy consumption.

The above numbers correspond to a mean wind direction angle equal to 20° . On the other hand, the minimum values for emissions were observed for a mean wind direction equal to 167° , manifesting an 18% drop in fueling/ CO_2 emissions, 16% in NO, 4% in soot, and 25% in energy consumption.

The same pattern held true for the rest of the cycle phases (Figures 12–14), although for the low and medium phases it was far less pronounced due to vehicle and wind speeds being lower. However, in Figure 12 (high phase) it is shown that there were multiple trips with mean wind direction below 90° , where fueling/ CO_2 emissions or even NO emissions exhibited a noteworthy increase: above 8% for fueling and NO, and almost 4% for soot. On the contrary, the maximum drop in a tailwind environment was -7.3% for fuel consumption, -6.8% for NO emissions, and -2% for soot.

This fundamental “asymmetry” between the negative effect of headwinds and the positive effect of tailwinds lies behind the findings presented in Figure 4. For a sufficiently large number of trips, the inadequacy of tailwinds to offset the effect of headwinds resulted in a net increase in fuel/energy consumption and emissions, thus highlighting the shortcomings of the current certification procedure, which ignores the effect of ambient wind.

Finally, going back to Figure 11, it is noted that the highest values for consumption and emissions increase did not appear at zero (or close to zero) wind angles, but rather at angles $\theta \approx 20^\circ$ – 30° . Going back to Equations (1) and (2) and Figures 1 and 2, we noticed that for such wind angles θ , the decrease in relative air speed V_R compared to the case where $\theta \approx 0^\circ$ (due to $\cos \theta < 1$) was offset from the rise in yaw angle and the subsequent rise in the drag coefficient. This observation challenges the widely accepted notion that vehicle aerodynamics should be optimized for the zero-yaw condition.

3.3. Comparison with the Cycle-Averaged Drag Coefficient Method

The results for the CADC (cycle-averaged drag coefficient) method presented in Section 2.6 are tabulated in Table 7. The difference in the estimated rise compared to the Monte Carlo methodology is included in brackets. In general, the CADC method slightly underestimated the effect of ambient wind on emissions compared to the Monte Carlo method. The discrepancy was more pronounced for soot emissions, although their relative increase vis-à-vis the reference case was rather small in both cases.

Table 7. Cycle-averaged drag coefficient method result (all values correspond to rise % over the reference case).

Site	Fueling/ CO_2	NO	Soot	Specific Energy Consumption
Site 1 (S1)	1.09 (+0.6%)	0.82 (−1.6%)	0.16 (−45.2%)	1.69 (−4.3%)
Site 2 (S2)	0.98 (−3.7%)	0.72 (−6.2%)	0.14 (−47.4%)	1.48 (−11.4%)
Site 3 (S3)	0.85 (−8.2%)	0.64 (−9.6%)	0.12 (−48.9%)	1.34 (−10.5%)
Site 4 (S4)	0.91 (−6.2%)	0.69 (−6.6%)	0.13 (−49.7%)	1.45 (−7.0%)
Site 5 (S5)	1.42 (−5.5%)	1.09 (−4.4%)	0.19 (−49.9%)	2.31 (−8.2%)
Site 6 (S6)	1.17 (−2.0%)	0.88 (−3.5%)	0.16 (−50.7%)	1.85 (−6.0%)
Site 7 (S7)	0.77 (−8.5%)	0.57 (−10.2%)	0.11 (−50.3%)	1.19 (−12.6%)
Site 8 (S8)	1.17 (−5.1%)	0.88 (−4.7%)	0.16 (−50.9%)	1.84 (−10.4%)
Average	1.04 (−5.5%)	0.79 (−4.8%)	0.14 (−50.0%)	1.64 (−8.9%)

Compared to the Monte Carlo method, the CADC method resulted in an estimated 1.04% increase in fueling/ CO_2 emissions (−5.5% compared to the average Monte Carlo rise), 0.79% in NO (−4.5%), 1.64% in energy consumption (−8.9%), and finally 0.14% soot emissions (−50.0%).

This discrepancy can be attributed to the fundamental difference between the two methods. The cycle-averaged drag coefficient method is a useful and quick tool to assess the impact of a given wind at a given driving schedule [22,29,31]. However, as the name suggests, it relies on replacing the zero-yaw drag coefficient with the cycle-averaged one,

which then remains constant throughout the cycle. Contrary to that, the Monte Carlo method suggested in this study calculates the drag coefficient at each second of the cycle (as a function of the yaw angle ψ). By doing so, the aggravation of engine transients due to the instantaneous increase in the wind drag coefficient is allowed to come through.

4. Conclusions

In the present study, fuel and energy consumption, as well as engine-out emissions, were calculated for a light commercial vehicle being driven in a simulated ambient wind environment. This ambient wind environment was constructed by picking random values from the Weibull probability distribution function for wind speed and from the uniform distribution for wind direction. The Weibull scale and shape parameters for the random wind profiles were drawn from eight different sites around Europe. Considering a sufficiently large number of random trips, we arrived at the estimate of the relative increase in fuel/energy consumption and emissions compared to the type-approval case (i.e., zero wind) due to ambient wind presence. The driving schedule employed was the WLTC 3-2.

There was an aggregate increase in all properties, vis-a-vis the reference case (i.e., zero wind), if ambient wind was accounted for in road load calculation. The mean winds for the sites examined ranged from 14.6 km/h to 24.2%. The average fueling/ CO_2 rise across all sites examined was 1.10% ($S = 0.22\%$), whereas the NO emission average rise was 0.83% ($S = 0.16\%$) and soot emissions exhibited an almost negligible overall increase equal to 0.28% ($S = 0.05\%$). Specific energy consumption was the property most influenced, displaying a 1.80% ($S = 0.38\%$) increase over the reference case.

However, the above numbers do not tell the whole story. Section 3.2 dug a little deeper by breaking down results from a large number of random trips into the separate phases of the WLTC cycle, i.e., correlations between the change in consumption and emissions on the one hand and wind speed and direction on the other were derived for different vehicle speed regimes. It was shown that for the high and extra-high phases of the cycle, where average vehicle speeds were significantly higher than the low and medium phases (and wind speeds were also expected to be higher as surface roughness decreased), the increase in emissions was much higher, reaching up to 28% for fueling/ CO_2 emissions, 22% for NO, and 13% for soot in the extra-high phase.

Furthermore, it was noted that although a “tailwind” environment can have a positive effect on the performance and emissions of the vehicle, that effect is not enough to cancel out the adverse effect of a “headwind” environment. Especially with regards to (engine-out) NO and soot emissions, it was found that the instantaneous coupling of a “headwind” with the specific operating point of the engine could greatly affect the overall emissions, even though the overall wind environment may be favorable (i.e., tailwind).

The results from the present study point to the shortcomings of the current homologation procedure, which excludes ambient wind from the calculation of total road load, among other parameters, such as road gradient. Given the technical challenges involved in trying to mechanically simulate the highly dynamic road load in the dynamometer, numerical simulation methodologies such as the one presented herein can prove themselves to be a valuable tool in the effort to accurately capture the real-world performance and emissions of a vehicle. Obviously, real driving emissions testing is expected to play a huge part in that goal, however, simulation is still indispensable, given that a huge number of scenarios can be studied in a fragment of the time and cost involved in RDE testing. What is more, simulation methodologies can further assist engineers in applications such as eco-routing, which, although not directly linked to the challenges of vehicle homologation, can still play a significant role in bringing transportation emissions down.

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Abbreviations

Scheme		
A	vehicle frontal area	m^2
c	Weibull scale parameter	ms^{-1}
C_{CADC}	cycle-averaged drag coefficient	-
C_d	drag coefficient	-
C_{DW}	Weighted drag coefficient	-
$CF_{diurnal}$	diurnal wind speed variation correction factor	-
$CI_{95\%}$	95% confidence interval	%
F_{aero}	aerodynamic resistance	N
F_{grad}	gradient resistance	N
F_{in}	inertia resistance	N
F_{roll}	rolling resistance	N
f_r	rolling resistance coefficient	-
F_{tr}	tractive force	N
g	acceleration of gravity	ms^{-2}
I	gear used	-
i_b	back-axle ratio	-
i_g	gear ratio	-
k	Weibull shape parameter	-
MF	mass factor	-
m_r	mass of driveline rotating components	kg
m_v	vehicle mass	kg
N_e	engine speed	rpm
NG	total number of gears used	-
R_d	tire dynamic radius	m
R_g	tire geometric radius	m
R_{st}	tire static radius	m
S	sample standard deviation	%
T_e	engine torque	Nm
V_R	vehicle/wind relative speed	kmh^{-1}
V_V	vehicle speed	kmh^{-1}
V_W	wind speed	kmh^{-1}
$V_{W_{10}}$	wind speed at 10 m above ground	kmh^{-1}
$V_{W_{veh}}$	wind speed at vehicle height	kmh^{-1}
$\bar{Y}$	random trip sample mean	%
Y_j	random trip result	%
Z_0	surface roughness length	m
Z_{veh}	height of wind drag application on vehicle	m
Z_{10}	height above ground level	m
α	grade angle	deg
Γ	gamma function	-
η_{tf}	transmission efficiency	-
θ	angle between vehicle and wind speeds	deg
ρ	ambient air density	kgm^{-3}
ψ	yaw angle	deg
ω_w	wheel angular velocity	$rads^{-1}$

Abbreviations

CADC	cycle-averaged drag coefficient
CFD	computational fluid dynamics
CLT	central limit theorem
EPA	Environmental Protection Agency
FTP	Federal Test Procedure
HWFET	Highway Fuel Economy Test
ICCT	International Council on Clean Transportation
LCV	Light commercial vehicle
RDE	Real driving emissions
US06 (SFTP)	Supplemental Federal Test Procedure
WLTC	Worldwide-harmonized Light-duty Vehicles Test cycle
WLTP	Worldwide-harmonized Light-duty Vehicles Test Procedure

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Article

Computational and Experimental Analysis of Axial Flow Cyclone Used for Intake Air Filtration in Internal Combustion Engines

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Abstract: The properties and advantages of axial flow cyclones are presented; several dozen of them are already widely used as the first stage of inlet air filtration in internal combustion motor vehicle engines, work machines and helicopters. The necessity to conduct research on cyclones to improve separation efficiency has been demonstrated. Using the commercial engineering software Ansys Fluent, at a constant inlet velocity of 10 m/s, an assessment was made on the effect of the separation length and inlet diameter of the outlet tube on changes in separation efficiency in axial flow cyclone. Each of the examined parameters was variable while maintaining other factors at a constant level. In the numerical calculations, test dust was used, which was the equivalent of AC fine dust, the particle size composition of which was taken into account using the Rosin–Rammner model. Increase in the separation efficiency was observed with an increase in the separation length and a decrease in the diameter of the cyclone inlet tube. For the cyclone model with an increased separation length and reduced diameter of the inlet pipe, numerical tests of separation efficiency and pressure drop were performed for various velocities at cyclone inlet in the range of 2.5–15 m/s. The obtained characteristics of modified axial flow cyclone were experimentally verified on a laboratory stand during cyclone prototype tests, the model of which was printed using the additive manufacturing technique.

Keywords: air filter; axial flow cyclone; separation efficiency; pressure drop; numerical and experimental research; combustion engines

1. Introduction

A cyclone, also known as a cyclone separator, is a device for separating solid particles from gas, and uses for this purpose the centrifugal force acting on a particle produced by the swirling gas. The principle of gas purification in a cyclone, which consists in rotating an aerosol stream, has not changed since 1885, when it was patented in the USA [1]. The cyclone patent was granted to John M. Finch from the United States in 1885 and transferred to the Knickerbocker company. Although the “dust collector,” as it was then called, contained the essence of modern cyclones, the dust could have emerged from the side of the cylindrical body rather than from the bottom of the conical portion as it does today. It was also quite a complex device and did not resemble today’s cyclones [1].

Due to the simple, solid structure, lack of moving parts, high operational safety and due to stable and low pressure drops, as well as the comparatively low investment and operating costs, cyclones are very efficient and popular devices for separating solids from the air stream, not only in industry, but also for filtering the intake air in vehicle internal combustion engines, agricultural and construction machinery. A cyclone works by making the gas stream rotate and the particles acquire centrifugal force. Gas stream rotation can be achieved by tangentially bringing it to the top of the cylindrical part or by flowing through a stationary swirler (located at the inlet of the cylindrical part), made of profiled blades with a helical outline, where the rectilinear movement changes into a swirling motion, which then changes into a helical movement.

Due to operation principle, cyclones are divided into reverse, when the flowing aerosol stream changes flow direction by 180° (Figure 1a,b); clean gas and collected particles leave the device on opposite sides and axial flow, when the flowing aerosol stream does not change flow direction, then the gas and particles “pass” through the cyclone in only one direction, leaving the device from the same end (Figure 1c).

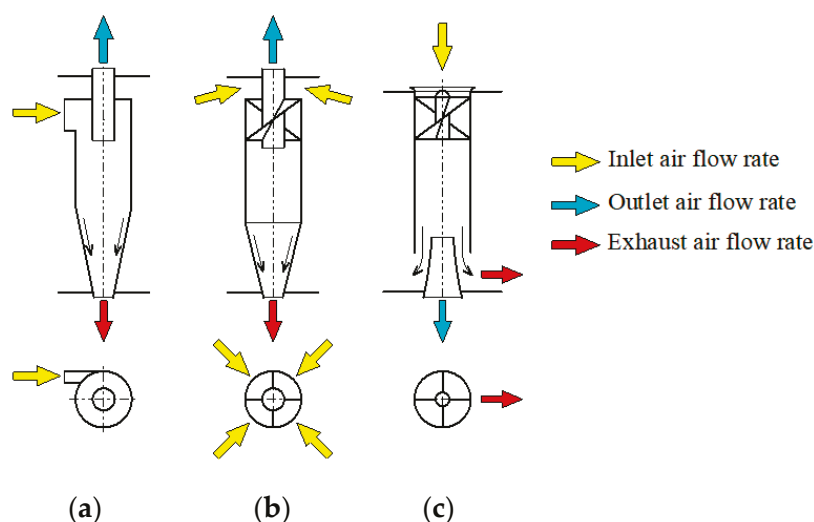


Figure 1. Types of cyclones used for filtering the intake air in motor vehicles: (a) reverse-flow cyclone (b) return with axial inlet, (c) axial flow (vortex tube separators).

In a reverse cyclone, the gas stream, after reaching the bottom of the cone, turns into a backward helical movement upwards and flows out through an outlet pipe made centrally in relation to the cylindrical part of the cyclone or in the center of the swirler, and then continues to move in a helical motion. In an axial-flow cyclone, the gas stream does not change direction and exits through a centrally located outlet tube where it continues to rotate in a helical motion.

Theoretical and experimental cyclone studies, the design of which has not changed much over the years, have been carried out since the 19th century, and therefore the available literature contains many studies presenting numerical and experimental studies with results of the process of aerosol separation in return cyclones with tangential inlet, e.g., [2–7]. Tests of a return cyclone with a tangential inlet carried out so far have focused on its four basic elements: the cyclone inlet [8–14], the cyclone body [15,16], the cyclone outlet pipe [17,18], the outlet and the dust container [19,20].

Compared to other purifying devices, cyclones have many advantages, for example [21–24]:

- simple and compact structure, no moving rotating parts and frictionally cooperating elements, which ensures high reliability;
- low operating costs: they do not require maintenance because they do not use surface filters that require regeneration;
- ability to work in difficult conditions, such as: high temperature and pressure, different gas humidity;
- relatively low and constant (not increasing for the same air stream flowing to the engine) pressure drop during operation, which ensures low energy consumption; pressure drop increases only in partition filters, where as a result of the retention and dust accumulation, the structure of the filter bed is densified, and thus the air flow is hindered;
- high throughput: the ability to separate large dust mass from large air streams;
- constant characteristics during operation, depending only on the air stream, the value of which changes insignificantly depending on the engine speed.

- Cyclones also have some disadvantages, for example:
- separation efficiency decrease in the absence of tightness;
- pressure drop increase with increasing flow velocity, which increases energy consumption;
- low separation efficiency of polluted air from particles $d_p < 10 \mu\text{m}$.

Return cyclone with tangential inlet, due to its significant advantages, is widely used in filtration processes in many industries. Large cyclones with a body diameter of up to several meters are used in sawmills [25], cement mills [26], oil refineries [27], coking plants, metallurgy, building materials plants and grain processing.

Cyclones are used to filter waste gases in many processes related to energy and heat production, both on a large scale, where energy generation technologies are based on burning coal [28], and where energy is produced on a smaller scale, e.g., heat recovered during the incineration of municipal waste. Smaller diameter cyclones are used for bioaerosol sampling, drug-delivery systems and also for vacuum cleaners. Cyclones with internal diameters of 40 mm and smaller ones are used in air intake filtration systems for combustion engines in tracked vehicles, working and agricultural machines, operated most often in conditions of high dust concentration in the air, the value of which often exceeds 1 g/m^3 [29–32].

Axial flow cyclones are characterized by several times lower pressure drop than reverse cyclones and, unfortunately, a slightly lower separation efficiency, but more favorable installation conditions in the engine intake system. The air stream then flows through the filter without turbulence, without changing its direction, which results in less energy loss of the motor [33]. Thus, axial flow cyclones are attracting increasing attention and are becoming future-oriented devices for filtering intake air in internal combustion engines [34].

Motor vehicle engines suck in from the environment pollution of maximum sizes up to $80\text{--}100 \mu\text{m}$ and different chemical composition, which characterizes greater mass of SiO_2 and Al_2O_3 67–91% and 5–14%, respectively, and with smaller mass share of other ingredients: Fe_2O_3 , MgO , CaO . Particles in the inlet gas stream vary so much in size, density, shape, viscosity, brittleness, erosion, surface charge, concentration, and other characteristics that no separation method or type of separator is suitable for capturing the entire spectrum.

Dust suspended in the air is the most common cause of accelerated wear of two frictionally cooperating machine elements, such as the P-PR-C coupling elements (piston-piston rings-cylinder walls) in engines. The dust sucked in with the air enters the combustion chamber (above the piston) and, therefore, the upper part of the cylinder and piston as well as the upper piston rings are subject to the greatest wear. Abrasive wear of engine components is caused mainly by particles of $1\text{--}40 \mu\text{m}$, with the most harmful particles in the range of $1\text{--}20 \mu\text{m}$ [35–37]. About 30% of the pollutants entering the engine escape unchanged along with the exhaust gases from the cylinders to the exhaust system. Only 10–20% of the dust that enters into the engine with the air through the intake system settles on the cylinder liner wall. Together with the oil, this part of the dust forms a kind of abrasive paste which, in contact with the mating surfaces of the engine, for example P-PR-C, causes abrasive wear. The most dangerous for two cooperating engine components are dust particles whose diameter d_p is equal to the oil film thickness $h_{\min}$ between this two surfaces at the moment. In typical combinations of internal combustion engine, oil thickness film depends on the conditions and parameters of the engine as well as the properties of the oil and, therefore, they assume different values in the range of $h_{\min} = 0\text{--}10 \mu\text{m}$ [38]. Moreover, the value of the minimum oil film thickness and the period of its occurrence in the engine cycle depends on sliding surfaces' shape asymmetry geometry.

Excessive wear of the P-P-C connection is the cause of increased blowing of fresh load into the crankcase, which causes a decrease in compression pressure and engine power, an increase in exhaust gas blow-by into the crankcase, which accelerates oil aging

and increased consumption of oil flowing over the piston, which increases solid particles emissions [39,40]. One of the ways to reduce the friction and wear of the P-P-C connection is to use surface asymmetry shape of the slip rings while maintaining the required thickness of the oil film necessary for the fluid friction conditions between the mating surfaces [41]. Another way to reduce friction between engine components reciprocating or rotating, increasing fuel economy along with emission and noise requirements, is to use a modern Common Rail fuel injection system capable of injection pressures above 2500 bars in combination with a newly developed combustion system [42,43].

Thus, a gas filtration device must be able to retain a wide variety of materials ranging from tens of micrometer particles to sub-micrometer particles, from hard minerals such as silica (SiO_2) to soft food products. Some of these materials are very free flowing, while others tend to coagulate. For this reason and due to high dust concentrations occurring during the operation of off-road and special vehicles, two-stage systems of engine intake air filtration are used: inertial filter and partition filter connected in series. The first stage of filtration is usually a multicyclone, where initial filtration takes place—separation of the largest particles of impurities (over 15–35 μm) and mass. Small-sized dust grains remaining in the air stream are directed to a partition filter, usually made of pleated paper with a suitable surface, where they are retained with an accuracy of more than $d_p = 2\text{--}5\ \mu\text{m}$.

As a result, required air purity is supplied to the engine cylinders, which reduces piston rings and cylinder surfaces wear, thus increasing engine and vehicle reliability. At the same time, the filtration system's operation time is extended, limited by the filter cartridge reaching a certain value of the permissible resistance Δp_{fdop} .

In modern trucks, working machines, farm tractors and helicopters, axial flow cyclone assemblies are already widely used as the first stage filtration instead of return cyclones with a tangential inlet [44]. A porous partition (filter cartridge) made of pleated filter paper placed in series behind the multicyclone is characterized by high separation efficiency (over 99.5%) of dust grains over 5 μm , limited and low (220–250 g/m^2) dust absorption, and continuous increase in pressure drop due to the retention and dust accumulation inside a thin layer of paper (0.6–0.8 mm). Air filter pressure drop, which increases during operation (resulting from an increase in the filter element resistance), causes additional energy losses in the engine and, therefore, should not exceed the permissible value Δp_{fdop} , set by the standards. The value of Δp_{fdop} is determined from the condition of a 3% decrease in engine power and is 2.5–4.0 kPa—passenger car engines, 4–7 kPa—truck engines and 9–12 kPa—special purpose vehicles [45]. After reaching the permissible value of Δp_{fdop} , the filter element, although it still maintains a high separation efficiency, should be replaced with a new one, so that it does not cause an additional increase in filter resistance and engine energy losses in the form of power decrease and an increase in specific fuel consumption. From a technical point of view, air filter service life is commonly defined as the restriction level that causes the pressure on the passenger car filter to drop by about 2.5 kPa above pressure drop of the new (clean) filter [46]. For trucks and special vehicles, the Δp_{fdop} values are assumed to be approximately 6.25–7.5 kPa above the pressure drop of clean air filter [47].

Depending on dust mass directed with air onto the filter cartridge, which in a two-stage filter is mainly determined by cyclone separation efficiency, increase in air pressure drop occurs with varying intensity. A group of cyclones (multicyclones), which is the first stage of filtration, retains a significant (87–95%) dust mass, which means that only a small part of it goes to the second stage of filtration.

Thus, the increase in the pressure drop of the filter element is less intense, which increases the service life of the air filter several times, and thus the vehicle life without the need to service the filter. Each increase in cyclone filtration efficiency means a smaller dust mass directed to the second filtration stage, which translates into a less intensive increase in filter resistance, lower energy consumption by the engine and an extension of the time until permissible value of Δp_{fdop} is reached. Therefore, it is justified to conduct works

aimed at testing axial-flow cyclones in terms of improving the efficiency and filtration accuracy while maintaining a low pressure drop.

However, it is not very common to find papers presenting the results of numerical studies and experimental cyclones. This can be explained by the difficulty of modeling and practical implementation with high precision of the swirler blades, which is the most technologically complicated part of the cyclone.

The axial-flow cyclone consists of three basic elements: a swirler, a cylindrical body and an outlet tube. The swirler, most often having four blades with a helical surface, is located in the front part of the cyclone body (Figure 2). Symmetrically arranged blades are inclined in relation to the cyclone axis (direction of air flow) at the angle α_k . The outer diameter of the swirler results from the inner diameter of the cylindrical body of the cyclone. The helical configuration of the blades presents serious manufacturing difficulties, especially for very small diameter turbulators. That is why handlebars with flat surfaces are often used.

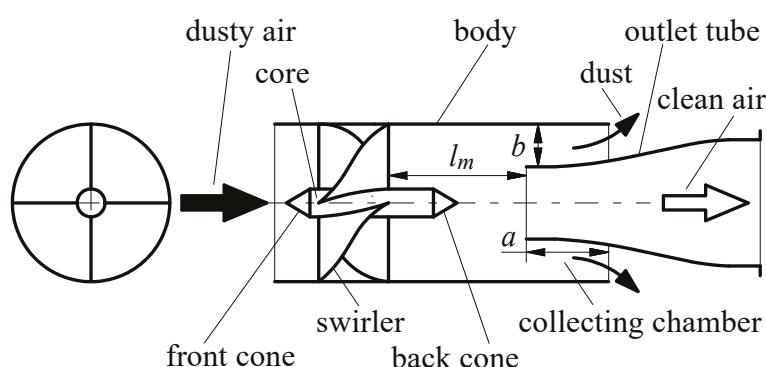


Figure 2. Main elements of axial flow cyclone: a —length of the collecting chamber, b —width of the collecting chamber, l_m —separation length of the cyclone (distance of the swirler from the separation chamber).

In the central part of the swirler there is a cylindrical core, which is integral with the blades. Most often, the core extends beyond the rim of the blades (extension part) into the cyclone. In the case of a core with a diameter greater than 5 mm, a streamlined shape in the form of a hemisphere or a cone is attached at the inlet. A similar cone is mounted to the core on the outlet side of the swirler.

The cones are an integral part of the turbulators and can significantly affect cyclone pressure drop and efficiency. The main function of the front cone is to direct the incoming stream away from the central swirl area. The front cone directs the incoming air uniformly radially outward around the swirler. This direction of the air causes, by impinging on the front cone, an initial radial momentum to the dust particles towards the wall of the cylindrical portion.

A conical outlet tube is placed concentrically to the cylindrical part of the cyclone at a predetermined distance l_m from the cyclone, the circular cross-sectional area of which increases its value starting from the inlet opening towards the outlet. The outer wall of the conical outlet tube and the inner wall of the cylindrical body overlap and form a ring with a width of b , called the dust collection chamber (Figure 2).

Pressure drop generated by the cyclone and the separation efficiency are the key parameters characterizing the performance of the cyclone separator. The filtration characteristics of axial-flow cyclones depend on many variables that can be divided into three groups. The first group consists of geometric parameters that are characteristic dimensions of the cyclone elements. These are the dimensions:

- (a) related to the cylindrical part of the cyclone: inner diameter of the cylindrical part D , height of the cylindrical part, separation length of the cyclone l_m , length of the circumferential gap a ;

- (b) swirler: swirler length l_z , steering wheel travel (helix pitch) S , number of vanes, diameter of swirl core d_r ;
- (c) separation chamber: width of the collecting chamber b , length of the collecting chamber a ;
- (d) outlet tube: outlet tube length c , outlet tube inlet inner diameter d_w , outlet tube outlet inner diameter D_w .

The parameters of the flowing aerosol are the second group in which the properties of the treated gas are distinguished: temperature, density, viscosity, dust concentration in the air and dust properties: density, particle size distribution, chemical composition, grain shape, tendency to coagulate. The third group is the gas flow conditions in the cyclone, including: gas flow velocity through the cyclone, degree of suction, type of gas flowing.

Due to the fact that separation efficiency and pressure drop in a cyclone, including axial flow cyclones, are determined by a large number of parameters, this prompted scientists to use Computational Fluid Dynamics (CFD) programs to evaluate them, partially eliminating the need for time-consuming and costly experimental research, and shortens design work.

Computer simulation packages, based on specially created codes (CFD) [48] are helpful tools that are successfully used during the design and optimization of devices operation used in energy production [49–51], chemical engineering [52,53], process engineering [54,55] and environmental protection processes [56]. The versatility of application, the possibility of quick modification of the size and geometry of the designed device, as well as the wide possibilities of simulating single and multiphase flows, make them useful for the analysis and design optimization of axial-flow cyclones.

However, in the available literature, there are not too many numerical test results showing the influence of cyclone and flowing aerosol parameters on the performance of the axial-flow cyclone. The subject of the research were the individual elements of the cyclone: the number of swirl guides, the angle of the steering wheel [57–63], the pitch of the swirler's spiral, the height and diameter of the swirler and the diameter of the core [57,61,64], the width of the collecting chamber gap [57,61,65], distance of the swirler from the inlet of the outlet pipe [57,64], diameter and height of the cylindrical part of the cyclone [64].

For example, Klujszo et al. [57] carried out a numerical study of the influence of different designs of the swirl blades, the diameter of the swirler and the diameter of the core, the width of the collecting chamber gap and the distance of the swirler from the outlet pipe inlet on the efficiency and flow resistance of axial-flow cyclone. Using the CFD model, three different constructions of the swirl flaps were simulated: straight, 3-plane (20° , 20° , 20°) and 3-plane (30° , 20° , 10°) vanes. In all cases, the total angle of the steering wheels was kept at 60° . Compared to straight blades, three-plane blades, which gradually change the flow, significantly reduce the pressure drop of the cyclone. It has been found that the distance of the swirler from the inlet of the outlet tube (separation distance 1 m) has a slight influence on the pressure drop and the separation efficiency. It has been shown that for a fixed diameter of the swirler, an increase in the diameter of the core and the cone face reduces the flow area between the guides, causing an increase in the flow velocity and an increase in the pressure drop.

The gap width of the collecting chamber b has a significant influence on the efficiency and flow resistance of the cyclone. Increasing the b -size reduces the inlet diameter of the outlet tube d_w , increasing the flow velocity of the stream and increasing the pressure drop while increasing the separation efficiency due to the larger dust mass being trapped by the larger aperture. At the same time, the width of the slit b should be as small as possible in order to reduce the pressure loss as it flows through the outlet pipe.

However, when the width b is reduced too much, dust is deposited on its walls, which deteriorates its permeability, as a result of which the separation efficiency deteriorates. Therefore, the dimensions of the slot should be selected depending on the diameter of the

swirler. for a swirler with a diameter of 50.8 mm, a gap of about 6.4 mm turned out to be the best solution.

The authors of [65] came to similar conclusions by examining the effect of the gap width of the collecting chamber in the range $b = 1.5\text{--}7.5$ mm on the cyclone performance. They determined that the chamber width for the swirler diameter $D = 45$ mm and the steering wheel travel (helix pitch) $S = 85.7$ mm cannot be less than 6.5 mm.

Gopalakrishnan et al. [58] investigated the efficiency and pressure drop of the axial-flow cyclone with an axial inlet depending on the number of swirl flaps (3, 4 and 5), its height and the steering angle. Each of these parameters was variable while keeping other factors constant. It has been observed that with a larger number of blades, the pressure drop increases with no increase in the separation efficiency. By increasing the tilt angle of the steering wheel in the range of $145\text{--}175^\circ$, the simultaneous increase in separation efficiency and pressure drop was achieved. It has been found that the height of the swirler is a sensitive parameter and has a significant effect on both the pressure drop and the separation efficiency. The flow resistance decreases with the height of the swirler, but at the cost of a decrease in separation efficiency.

In [59], the CFD method was used to investigate the effect of the number (6, 8 and 10) of guide vanes on the pressure drop of axial-flow cyclones. It was found that with an increase in the number of blades, the pressure drop of the axial-flow cyclones at the same inlet velocity becomes smaller and smaller.

The experimental research presented in [64] was aimed at determining the interaction of the cyclone diameter D , the active length of the cyclone l_m , and turbulators with different blade pitch on the performance of the axial-flow cyclone. It has been shown that regardless of the diameter of the cyclone D , there is a clear maximum of separation efficiency, depending on l_m , which moves towards higher values of l_m with the increase of the swirl guide pitch. It was found that the optimal diameter of the cyclone and the optimal active length should be 45.6 mm and 80 mm, respectively, at the steering wheel stroke $S = 72.5$ mm.

Mao et al. [60] presented a study of the influence of three parameters of axial-flow cyclone: the shape of the steering wheel, the angle of the blades' outlet and the diameter of the guide cone on separation efficiency and pressure drop. The shapes of the steering wheels are divided into five types. The flap angles had the following values: 20° , 25° , 30° , 35° , 40° . The diameter of the guide cone includes five sizes: $0.3D$, $0.4D$, $0.5D$, $0.6D$ and $0.7D$, where $D = 50$ mm—the diameter of the swirler.

The shape of the guide bar and the angle of the guide bar can affect the separation efficiency and pressure drop of the cyclone. The diameter of the guide cone affects the pressure drop of the cyclone, but does not affect its separation efficiency.

During numerical tests, the authors of [61] found that reducing the angle of deflection of the vanes α may increase the tangential velocity and improve the separation efficiency, but it is associated with an increase in the pressure drop. On the other hand, increasing the turning angle β of the blades causes an intensive increase in the pressure drop and a slight increase in the separation efficiency. In turn, increasing the core diameter D_c leads to an increase in the tangential and axial velocity in the cyclone, and also significantly increases the pressure drop. To obtain high separation efficiency, the diameter D_c should be 65–70 mm.

As a result of research, many cyclone designs have been developed for specific applications. Reference [66] optimized the shape of an axial flow cyclone for air purification in an underground station using the fluid dynamics (CFD) method. The 50% $2.5\text{ }\mu\text{M}$ solids separation efficiency was achieved with minimal pressure drop. On the other hand, the authors of [67] developed an axial flow cyclone for the separation of nanoparticles with a diameter of 272–448 nm. Hsu et al. found that an axial flow cyclone can collect ultrafine particles [68]. The authors of the work [69], as a result of the multifunctional optimization of the cyclone structure with axial flow, achieved a compromise between the

static pressure drop and the separation efficiency. The 8 μm particle removal efficiency increased by 100% and the static pressure drop increased by 69.3%.

The analysis shows that the numerical tests of the axial-flow cyclone focus mainly on the assessment of the influence of the swirler's parameters on the efficiency and resistance of the cyclone flow. The subject of the tests are: the height of the swirler and its diameter, the diameter of the core, various constructions of the swirler blades, the number of blades, and the angle of the steering wheel. There are no studies on the inlet diameter of the outlet tube, and little work has been done on the effect of the distance of the swirler from the outlet tube inlet on the cyclone performance.

The results of numerical tests obtained from many models are not experimentally verified with the use of real dust, which calls into question the reliability of the model. Therefore, in order to partially fill the gap in this respect, the authors developed a model of a axial-flow cyclone and performed its research on the impact of the separation length of the cyclone, the internal diameter of the outlet pipe and the length of the swirl core on the efficiency and pressure drop. Using the results of similar studies and the results of their own numerical research, they validated the constructed model, and then printed the cyclone prototype using the additive manufacturing technique. The cyclone obtained in this way was tested experimentally in terms of separation efficiency and pressure drop with the use of test dust for various inlet velocities. The measures taken aimed at improving the efficiency and filtration accuracy while maintaining a low pressure drop of the cyclone.

2. Theoretical Basis of Aerosol Filtration in Cyclones

The aerosol separation process in a cyclone, although its structure is not complicated, is very complex and difficult to describe mathematically, therefore little has been published on their design and calculation. This is mainly due to the complex motion of the gas, which is not only a rotational (helical) motion, but also a two-way reverse motion, and a number of side effects occurring at that time. For this reason, there is no precise mathematical description of dust grains' movement in the cyclone, which is relative both to the fixed walls of the device and to the gas flowing inside the cyclone. This movement takes place in a space-varying (and specific for each cyclone structure) field of gas velocity, and thus is under the influence of the system of forces acting on each grain changing in space. Additional difficulties are different grain sizes, their density and shape, and resulting various force relationships (inertia and aerodynamic), as well as the possibility of grain collisions with the cyclone walls and causing, among others, dust coagulation. Models of particle movement in cyclones, found in many studies, were developed using the main dimensions of the cyclone, parameters of the flowing aerosol and with many simplifying assumptions, such as [22,70–73]:

- dust particle moves in a laminar motion within the validity range of Stokes' law;
- dust particles are homogeneous spheres;
- coagulation of dust grains is not taken into account and there is no detachment from the cyclone walls;
- gas moves in the cyclone along a specific, adopted spiral equation;
- cyclone body is a cylinder with a constant diameter along its entire length.

In the cyclone, as a result of the supply and swirl of the air stream, the dust grains inside it, the density of which is more than 2500 times greater than the air density, remain, under the influence of the inertia force F_B and overcoming the resistance force of the F_R medium, directed to the inner wall of the cyclone, after reaching it, they slow down and stop, and then, as a result of gravity, they fall into the dust collector located below. An increase in the air flow velocity to the cyclone, and thus the increase in the velocity of dust particles, increases the inertia force, which increases cyclone separation efficiency.

The obtained radial velocity of the dust grains causes their collision with the cyclone wall [73–75]. The energy that the dust grains have at the moment of collision with the wall of the cyclone j may be significant, and then the grains will bounce and are carried away

by the gas stream rapidly moving towards the cyclone outlet tube. Increasing the inlet velocity of the cyclone increases the energy of the grains, which may result in a greater intensity of their reflection. As a result, grains reflected from the walls of the cyclone will be more susceptible to being entrained by the outlet stream, which will reduce the separation efficiency.

A characteristic parameter of the cyclone's operation is the inlet speed v_0 , which is defined as the quotient of the air stream flowing into the cyclone Q_0 and the cross-sectional area of the inlet stub pipe at its narrowest point A_0 and calculated from:

$$v_0 = \frac{Q_0}{A_0} [\text{m/s}] \quad (1)$$

where: Q_0 —aerosol air stream flowing into the cyclone, A_0 —cross-sectional area of the inlet stub pipe at its narrowest point.

The contaminated air is supplied to the axial-flow cyclone along the cyclone axis, directly onto the vanes that form the swirler. Upon contact with the surface of the blades, which are inclined to the direction of the incoming air at an angle γ , path of dust grains changes. During the grain movement on the surface of the swirler's blades, a peripheral velocity component appears. Due to its aerodynamic effect on dust grains, apart from the axial movement, they also perform a transverse movement, which causes grains to move to the channel wall. In this way, inhomogeneous dust concentration is created in the air stream along the radius of the cyclone, higher near the channel wall.

After lowering blades of the swirler, the dust grains' path is helicoidal. These dust grains are influenced by: centrifugal force F_B , aerodynamic force F_R and buoyancy force F_G . In the plane perpendicular to the cyclone axis, two forces influence on dust grain: the inertia force F_B and the aerodynamic drag force of the F_R medium [73,76,77]. Dust particles move in a helical motion, and the shape of this path will depend on the mutual relationship of two forces values F_B and F_R . The force of gravity has a negligible effect on the grain path and, therefore, is not taken into account in its motion. In turn, value of the F_B and F_R forces will depend on the diameter, shape and density of dust grain and the type of gas. The rotating aerosol causes the formation of a centrifugal force, the value of which for a dust grain of mass m_z located at a distance r from the axis of rotation can be determined from [77]:

$$F_B = m_z \frac{u_s^2}{r} = \frac{1}{6} \pi \rho_p d_p^3 \frac{u_s^2}{r} \quad (2)$$

where: ρ_p —particle density, d_p —particle diameter, u_s —tangential component of the particle velocity, equal approximately to the tangent gas velocity component v_s .

The centrifugal force causes a particle to move towards the cyclone wall at radial velocity u_r against drag force of the medium F_R . Resistance force of F_R center can be determined by [77,78]:

$$F_R = \lambda \cdot A_p \frac{u_r^2}{2} \rho_g \quad (3)$$

where: A_p —projection surface of the particle (the projected surface area of the particle onto the plane perpendicular to the direction of its motion), ρ_g —gas density, λ —coefficient of frictional resistance, u_r —component of radial motion of the particle [78].

$$\lambda = f(\psi_k, Re) \quad (4)$$

where: Ψ_k —grain shape coefficient, Re —Reynolds number.

Dust grains movement towards the cyclone wall is determined by the equation resulting from the mutual relationship of the centrifugal force and the resistance force of the medium [72]:

$$m_z \frac{du_r}{dt} = \frac{m_z \cdot u_s^2}{r} - F_R \quad (5)$$

The above equation shows that dust grains with diameters greater than a certain limiting dimension d_{zg} , for which the $F_B > F_R$ condition is met, will move in a helical motion, making larger and larger circles and approaching the wall of the cylindrical part of the cyclone. After reaching the cyclone wall, dust grains slow down and stop, and then fall by gravity into the settling tank, so they will be separated with 100% efficiency. On the other hand, dust particles with diameters greater than d_{zg} , for which the resistance force F_R is greater than the inertia force F_B , will be directed to the center of the cyclone and will be entrained by the internal vortex of air flowing out of the cyclone outlet tube. For these particles, the separation efficiency is assumed to be 0%.

Dust grains with a dimension equal to d_{zg} , which are in the cyclone and for which the centrifugal force F_B and the radial drag component of the F_R medium are equal, will (theoretically) circulate in a circle with a radius r . However, due to the very irregular shape of dust grains, which determines the value of the aerodynamic force and their different density, which affects grain mass and the centrifugal force, grains for which the $F_B = F_R$ condition applies, do not circulate around a circle with a certain radius. It is assumed in the literature that 50% of them are retained in the cyclone and 50% are entrained by the internal vortex towards the cyclone outlet. The size of dust grains, the efficiency of which is 50%, is called the cut-off diameter d_{zg} and is marked as d_{50} [44,72]

Authors [35,79] describe the boundary grain diameter d_{zg} in a axial flow cyclone (Figure 3) with:

$$d_{zg} = \sqrt{\frac{18\mu_g(\ln 2)r_w^2 S^2}{8\pi\rho_z Q_0 l_m}} \quad (6)$$

where: Q_0 —air stream flowing through the cyclone, r_w —radius of the cyclone outlet tube, l_m —separation length of the cyclone, S —steering wheel travel (helix pitch).

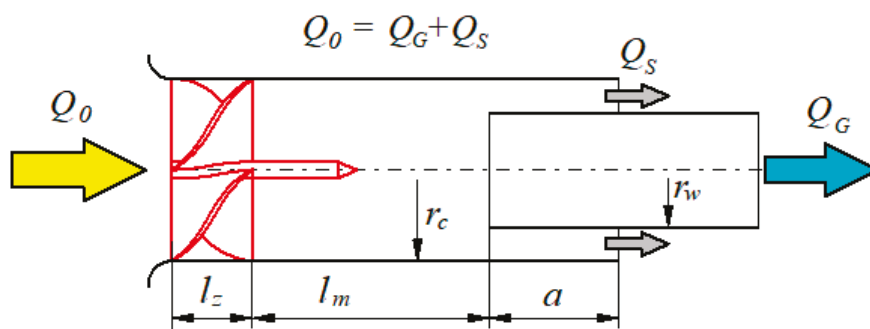


Figure 3. Characteristic dimensions and main elements of axial flow cyclone: Q_G —stream of purified air flowing from the cyclone through an outlet pipe, Q_s —air stream flowing from the cyclone together with separated solid particles, l_z —length of the swirler, a —separation chamber length, r_c —radius of the cylindrical part.

Above expression allows for determination of cyclone separation efficiency depending on the boundary grain diameter, as presented in [35,79]:

$$\varphi = 1 - \exp\left[-\ln 2\left(\frac{d_p}{d_{zg}}\right)\right] \quad (7)$$

Authors [80] derived analytically the dependence cyclone separation efficiency, which were used for filtering the intake air of helicopter engines in the following form:

$$\varphi_M = (1 - \beta)\left[1 - e^{\left(\frac{-8\pi Q_0 \tau_p l_s}{r_w^2 S^2}\right)}\right] + \beta \quad (8)$$

where: β —parameter determining the distribution of the air stream in the cyclone, defined by (Figure 3):

$$\beta = \frac{Q_s}{Q_G + Q_s} \quad (9)$$

where: Q_G —air stream sucked in by the engine, Q_s —air stream in the suction channel.

Taking the Stokes number for the helical flow in cyclones into account, the cyclone separation efficiency can be presented as follows [80]:

$$\varphi_M = (1 - \beta) \left[\left(1 - e^{\left(\frac{-8\pi^2}{(1-\beta)} St_c \right)} \right) \right] + \beta \quad (10)$$

According to the authors of [80], Stokes number in cyclones can be determined by:

$$St_c = \frac{\rho_g d_p^2 l_s u_g}{18 \mu_g S^2} \quad (11)$$

where: u_g —fluid velocity, μ_g —fluid viscosity, d_p —particle diameter, ρ_g —fluid density.

3. Axial Flow Cyclone Model

Analysis of aerosol flow through axial flow cyclone can be performed using the commercial engineering software Ansys Fluent. Using the equations of fluid mechanics, it is possible to determine the velocity distributions occurring inside the cyclone and to determine the trajectory of dust grains that are in the stream of flowing gas.

However, it is important to select an appropriate model that is suitable and will enable the solution of complex gas motion, which is not only a whirling (helical) motion, but a two-way reverse motion, with many side effects.

Authors [81] proved that only the turbulent flow model of the RSM reflects the experimental test well.

The work uses the discrete phase model, which is characterized by the fact that the trajectory of particles is determined using Lagrange's formulations. For the direction of flow "x", the balance of forces acting on the particle can be represented by the equation [76]:

$$\frac{du_p}{dt} = F_D(u - u_p) + \frac{g_x(\rho_p - \rho)}{\rho_p} + F_x \quad (12)$$

where: $F_D(u - u_p)$ —drag force acting on the particle determined from:

$$F_D = \frac{18\mu}{\rho_p d_p^2} \frac{C_D Re}{24} \quad (13)$$

where: u —gas flow rate in the cyclone, u_p —dust grains velocity, μ —air dynamic viscosity, ρ —air density, ρ_p —dust grains density, d_p —dust grain diameter, Re —relative Reynolds number, which is defined as:

$$Re = \frac{\rho d_p |u_p - u|}{\mu} \quad (14)$$

Drag coefficient C_D occurring in the dependence (13) depends on the shapes of dust grains and the Reynolds number [82]. There are many empirical dependencies in the available literature for determining the C_D resistance coefficients. However, the most popular is the relationship given by Morsi and Alexander, which determines the drag coefficient of a spherical particle moving in the air with the following relationship [83], and the expression is important for a wide range of Reynolds number ($Re = 0.1 \div 50,000$):

$$C_D = a_1 + \frac{a_2}{Re} + \frac{a_3}{Re^2} \quad (15)$$

where: a_1, a_2, a_3 —constants for spherical and smooth particles; the values of the constants a_1, a_2, a_3 for different ranges of the Reynolds number are given in [78].

The following assumptions were made for model construction:

- Solid phase flow was defined as being a set of spherical particles (spheres) suspended in a continuous phase.
- The trajectory of particle motion is determined using Lagrange's equations.
- The velocity and region of particle flow into the cyclone were defined.
- The grain size range and the particle size distribution of test dust were assumed.
- A homogeneous material of dust grains with a specific density corresponding to that of SiO_2 was adopted.
- A defined velocity profile was defined at the cyclone inlet and atmospheric pressure was assumed at the outlet.
- According to [76,84], the particle separation regions have been defined:
 - “reflect” condition—surfaces of swirler blades and internal walls of the cylindrical cyclone channel, where the particles are reflected;
 - “trap” condition—the surface of the cyclone outlet, where the number of dust grains retained by the cyclone is determined;
 - “escape” condition—an area where the number of dust grains that have not been stopped by the cyclone is defined.

For the “reflect” condition, the coefficient of restitution in the normal direction is determined by the relationship:

$$e_n = \frac{V_{2,n}}{V_{1,n}} \quad (16)$$

where: e_n —restitution coefficient in the normal direction, $V_{1,n}$, $V_{2,n}$ —particle velocity in the direction normal to the wall before and after the collision.

Taking into account the above assumptions, the following relationship was adopted to calculate separation efficiency of axial flow cyclone:

$$\varphi_{c(d_z)} = \frac{n_{p, \text{trapped}}}{n_{p, \text{injected}} - n_{p, \text{incomplete}}} \quad (17)$$

where: $n_{p, \text{trapped}}$ —number of dust grains retained in cyclone, $n_{p, \text{injected}}$ —number of dust grains injected into the cyclone with air, $n_{p, \text{incomplete}}$ —number of uncounted dust grains.

For numerical and experimental tests in this work, the PTC-D test dust was used, which is a substitute for AC fine in Poland [84]. The chemical and particle size composition of this dust is presented in Tables 1 and 2. Basic dust components are SiO_2 and Al_2O_3 , whose total share in the dust is less than 90%. They are also two hardest minerals, the hardness of which on the ten-point Mohs scale is 7 and 9, respectively. Moreover, the mass fraction of dust grains in the range of 0–5 μm is less than 40% of total dust mass. These are grains that are not retained in cyclones, but in partition filters made of filter paper with nanofibers addition.

Table 1. Granulometric composition of the test dust PTC-D [84].

Dust Particle Sizes d_p [μm]	Mass Fraction of Grains in the Dust F_m [%]
0 ÷ 5	38.55
5 ÷ 10	15.97
10 ÷ 20	16.48
20 ÷ 40	19.46
40 ÷ 80	9.54

Table 2. Chemical composition of the test dust PTC-D [84].

Dust Components	Mass Fraction of Ingredients in the Dust Z_m [%]
SiO ₂	67.15
Al ₂ O ₃	15.25
Fe ₂ O ₃	4.9
Na ₂ O	4.5
K ₂ O	4.5
CaO	2.35
MgO	1.15
Moisture	3.0

During numerical tests, the particle size distribution of the PTC-D test dust was taken into account by using the Rosin-Rammler model, for which two parameters are determined: n —spread parameter and $\bar{d}_p$ —average dust grain diameter. A characteristic feature of this model is that size ranges of the diameters into which the dust taken for testing is to be divided are defined. Mass distribution of dust grains with a diameter greater than d_p can be determined by the following equation [82]:

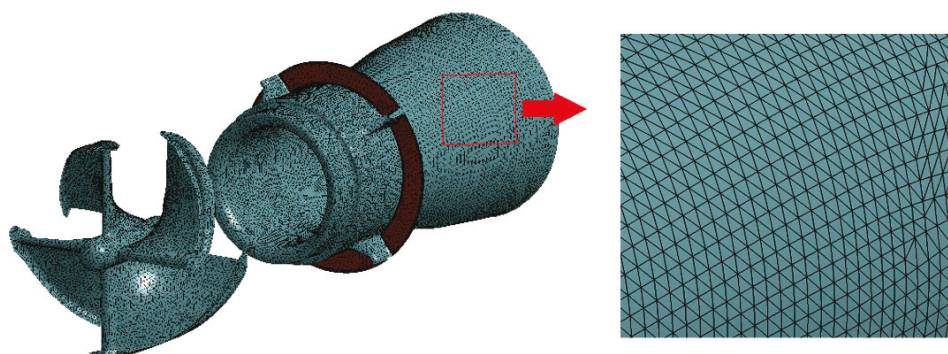
$$Y_d = e^{-(d_z/\bar{d}_z)^n} \quad (18)$$

where: n —spread parameter, d_p —particle diameter, $\bar{d}_p$ —average diameter.

Practical use of this relationship requires knowledge of n and $\bar{d}_p$ parameters. The exact and comprehensive methodology for determining and the values of n and $\bar{d}_p$ parameters for the particle size composition of the PTC-D test dust is presented in [29].

Cyclone geometry design, which is an element of the inlet air filter of the wheeled conveyor engine, was used to build axial-flow cyclone model. The cyclone geometry was obtained using the results of digital scanning with 3D ATOS II scanner produced by the GOM company. Information about shape and geometry of the cyclone was obtained, which was helpful in creating a digital copy of it.

Sets of points obtained as a result of the scanning, described by the x , y , z coordinates, reflect the surface of the scanned cyclone. These points, connected to form a mesh of triangles, reflect the shape of an object with a set of several million small triangles. On the basis of the obtained triangle mesh, a three-dimensional model of a cyclone is built, the geometry of which can be freely changed and numerical tests can be carried out. The advantage of the obtained model is also the fact that the modified model can be printed on a 3D printer and experimental tests can be performed, which was done in this work. Axial flow cyclone model is shown on Figure 4.

**Figure 4.** Axial flow cyclone model.

A very important aspect that should be taken into account when preparing a flow analysis is the correct behavior of the fluid on the interface “channel wall-fluid”. For this purpose, a transition from very narrow elements to increasingly larger so-called boundary layer zone. The parameter that helps determine the size of the first element as well as the boundary layer width is the $Y +$ parameter.

For the model used in this study, the boundary layer thickness was $g = 0.5$ mm, and height of the first element in this layer was $h = 0.01$ mm. To generate finite volumes in the proposed model, an interpolation algorithm was used, which causes the mesh size to increase incrementally with a given step as it approaches the channel axis. As a result of the activities performed, the flow domain in axial flow cyclone was obtained, consisting of 2,314,958 finite volumes (Figure 5). The initial boundary conditions used for the flow analysis are shown in Figure 6. Detailed assumptions of the model and its detailed description are discussed in [29].

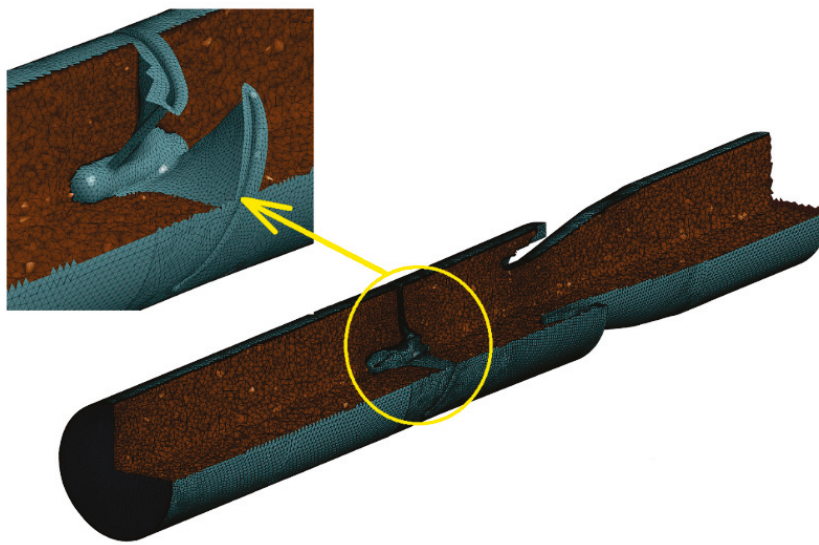


Figure 5. Flow domain in axial flow cyclone.

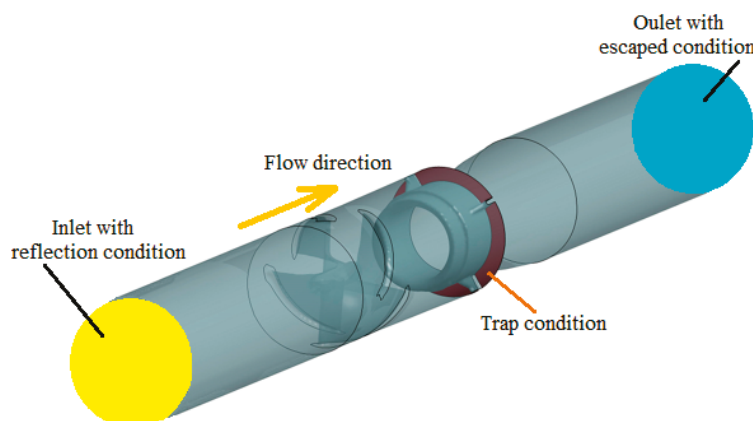


Figure 6. Initial boundary conditions used for flow analysis in axial flow cyclone model.

4. Scope, Subject and Conditions of Cyclone Numerical Tests

The aim of the research was to numerically evaluate the impact of axial cyclone changes in three dimensions:

- A. separation length of the cyclone l_m ,
1. internal diameter of the outlet pipe d_w ,
2. length of the swirl core l_r ,

on the separation efficiency and pressure drop.

The subject of the research was axial flow cyclone which is an element of an air filter multi-cyclone used in a special vehicle, operated in conditions of high dust concentration in the air. The main elements and characteristic dimensions of this axial flow cyclone are shown in Figure 7.

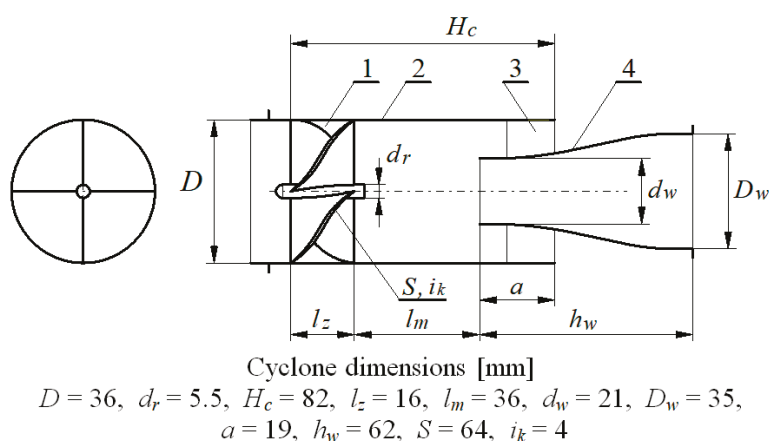


Figure 7. Main elements and characteristic dimensions of axial flow cyclone from special vehicle air filter: 1—swirler, 2—cylindrical part, 3—dust collecting chamber, 4—outlet pipe, H_c —cyclone cylindrical part length, l_m —separation length of the cyclone, D —swirler diameter, l_z —rotor length, d_w —outlet tube inlet diameter, D_w —outlet tube outlet diameter, a —separation chamber length, h_w —outlet tube length, S —steering wheel travel (helix pitch), i_k —number of blades, d_r —core diameter.

Numerical tests of the cyclone included preparation, research and results analysis. In the preparation part, research conditions were established. Duct type, the number of measuring points and values of inlet velocities were assumed. Research methodology has been developed. Dimensions of the cyclone were determined, which are given in Figure 7.

The research part was carried out using the following course of action:

- (1). The cyclone separation efficiency φ_c was determined successively depending on the three parameters l_m , d_w and l_r for one set value of the inlet velocity $v_0 = 10$ m/s. Each of these parameters was variable while keeping other factors constant.
- (2). For the maximum value of $l_m = 56$ mm and the minimum value of $d_w = 19$ mm, cyclone separation efficiency φ_c and the pressure profiles were determined depending on the inlet velocity in the range of $v_0 = 2.5$ –15 m/s. This range of inlet velocity v_0 into the cyclone results from the varying air demand of the engine when its rotation speed varies from minimum to maximum.

Results of numerical tests were analyzed as follows:

- (1). Efficiency characteristics of the cyclone $\varphi_{cA} = f(l_m)$, $\varphi_{cB} = f(d_w)$ and $\varphi_{cC} = f(l_r)$ for the inlet velocity $v_0 = 10$ m/s were developed and analyzed.
- (2). Cyclone efficiency characteristics $\varphi_c = f(v_0)$ for the value $l_m = 56$ mm and $d_w = 19$ mm were developed and analyzed, depending on the inlet velocity in the range $v_0 = 2.5$ –15 m/s, and then compared with the characteristics $\varphi_c = f(v_0)$ of the original cyclone from numerical and experimental tests.
- (3). Pressure profiles in the cyclone depending on the inlet velocity in the range of $v_0 = 2.5$ –15 m/s were analyzed.

Analysis of the results was completed with a summary in the form of conclusions.

Performed numerical tests included the assessment of three variants of changes in the cyclone structure:

- A Change in l_m length in relation to its original value in the range of (Figure 8):

- (a) increasing the length of l_m with the following values: $m = 5 \text{ mm}$, 10 mm , 15 mm , 20 mm ,
- (b) reducing the length l_m by the value: $m = 5 \text{ mm}$, 10 mm , 15 mm , 20 mm .

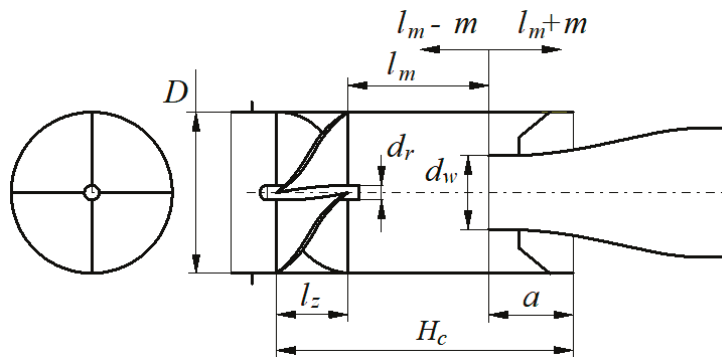


Figure 8. Length change l_m in relation to its original value $l_m = 36 \text{ mm}$.

Length change l_m by the value m changes the length of the cylindrical part of the cyclone H_c by the same amount.

- B Change of the inlet diameter d_w (Figure 9) of the cyclone outlet tube in relation to its original value (without changing the diameter D_w of the outlet tube), but with introducing outlet tube inlet streamlined shape within the range of:
 - (a) increasing the diameter d_w with the value: $w = 1 \text{ mm}$, 2 mm ,
 - (b) reduction of the diameter d_w by the value: $w = 1 \text{ mm}$, 2 mm .
- C Change in the core length by the value: $r = 5, 10, 15$ and 20 mm and ending it with a cone of angle α as shown in Figure 10.

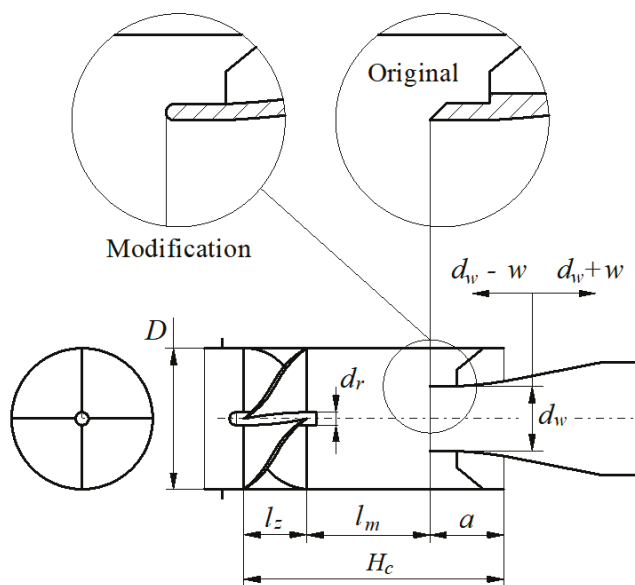


Figure 9. Diameter d_w change of the cyclone outlet tube from its original value and applying the streamlined inlet shape of the outlet tube.

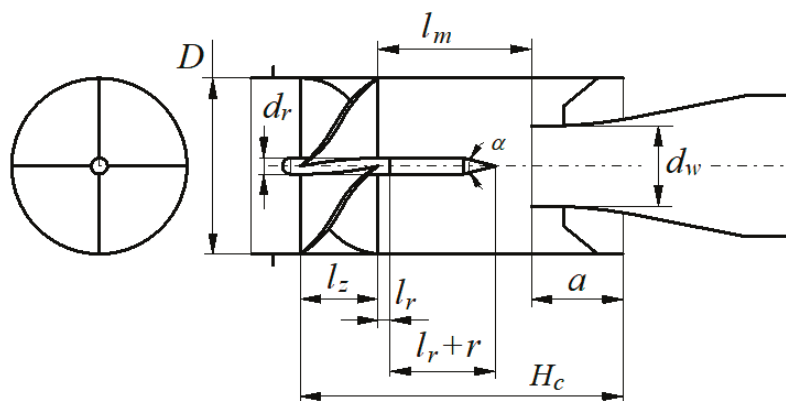


Figure 10. Cyclone modification C—change of the core length in relation to its original value.

Numerical tests were carried out in the range of inlet velocity of $v_0 = 2.5\text{--}12.5$ m/s, and the resulting air stream $Q_G = 10\text{--}45$ m³/h flowing through a single cyclone during engine's work in the range of rotational speed from minimum to maximum. Model tests of cyclone separation efficiency were determined with the use of single-component (silica) dust, which is the equivalent of AC fine polydisperse dust, the particle size distribution of which was taken into account using the Rosin–Rammner model (29).

During the simulation of air flow containing dust particles through the tested cyclone, the following input parameters were adopted:

- dust grain density $\rho_z = 2650$ kg/m³—SiO₂ density, being the main component of mineral dust (65–95%), was assumed,
- dust concentration $s = 0.5$ g/m³,
- air density $\rho = 1.225$ kg/m³,
- air dynamic viscosity coefficient $\mu = 17.85 \cdot 10^{-6}$ Ns/m²
- air kinematic viscosity coefficient $\nu = 14.61 \cdot 10^{-6}$ m²/s,
- ambient air temperature $t_H = 18$ °C,
- outside pressure $p_H = 1013$ hPa.

During the tests, the number of retained grains N_z for which $F_B > F_R$ was determined and the number of grains N_{50} , which meet the condition:

$$F_R = F_B \quad (19)$$

Grains for which the centrifugal force F_B and the radial component of medium resistance F_R are equal, theoretically circulate along a circle of radius r . In practice, it is assumed that 50% of them are separated from the cyclone, and 50% are carried away by the internal vortex towards the outlet cyclone. Dust grains size, which efficiency is 50%, is called the boundary grain diameter d_{zg} and it is marked as d_{50} .

Separation efficiency in the axial flow cyclone resulting from the model tests was determined as the quotient of the sum of the number of particles retained by the cyclone N_z and 50% of the number of particles meeting the d_{50} condition and the number of particles supplied to the N_D cyclone in the air stream, using the following relationship:

$$\varphi_{mc} = \frac{N_z}{N_D} = \frac{N_z + 0,5 N_{50}}{N_D} \quad (20)$$

where: N_z —number of particles retained by the cyclone, N_D —number of supplied particles, N_{50} —number of particles meeting the d_{50} condition.

5. Numerical and Experimental Test Results of Cyclone

Numerical tests results of three modification variants of axial flow cyclone structure are shown in Figures 11–16. Cyclone separation efficiency change φ depending on the

active length of the cyclone l_m (original value $l_m = 36$ mm) is shown in Figure 11. Increasing the length l_m with the values: $m = 5$ mm, 10 mm, 15 mm, 20 mm increases the cyclone separation efficiency in the range of 87–90%. Reducing the length of l_m by the same values of m causes an intense decrease in separation efficiency of the cyclone to 79.5%.

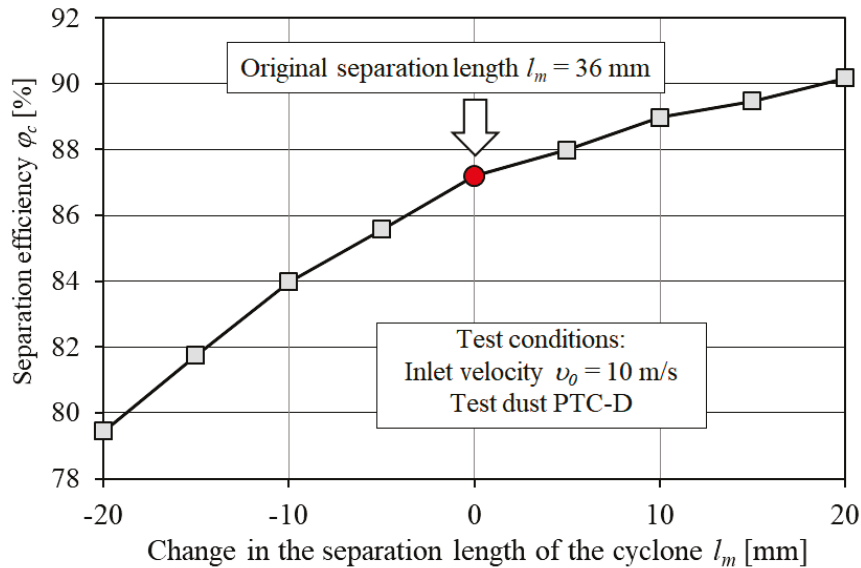


Figure 11. Cyclone separation efficiency φ_c depending on the change of the cyclone's separation length l_m .

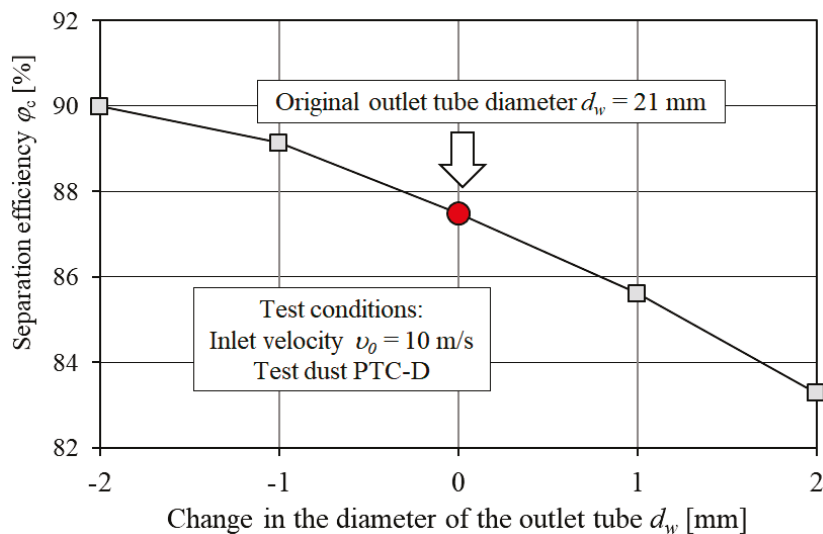


Figure 12. Cyclone separation efficiency φ_c depending on the change of the internal diameter of the cyclone outlet pipe d_w .

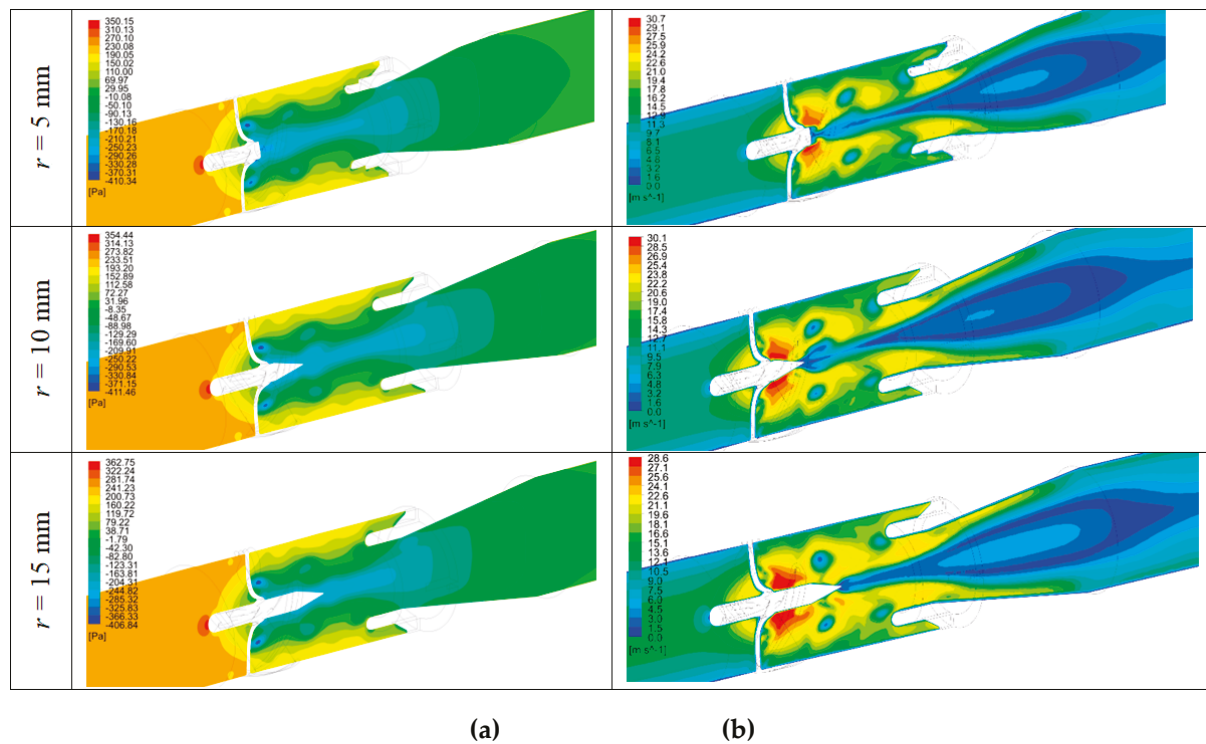


Figure 13. Pressure values (a) and velocity (b) in the cyclone after inserting the core with a cone at the outlet of the swirler-inlet velocity $v_0 = 10$ m/s.

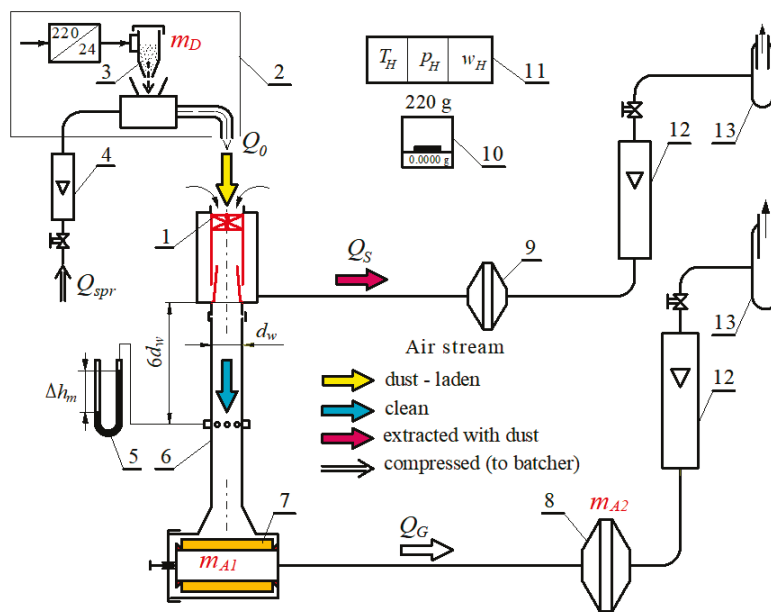


Figure 14. Functional diagram of the test stand for axial-flow cyclones: 1—tested axial flow cyclone, 2—dust dispenser, 3—dust container, 4—compressed air measuring rotameter, 5—U-tube liquid manometer, 6—outlet pipe from cyclone 7—measuring absolute filter A1 of the main stream, 8—measuring absolute filter A2 of the main stream, 9—filter securing the suction stream rotameter, 10—analytical balance, 11—set for measuring humidity, ambient air temperature and pressure, 12—rotameters for measurement main air stream Q_G and extraction Q_S , 13—fans forcing the air stream flow.

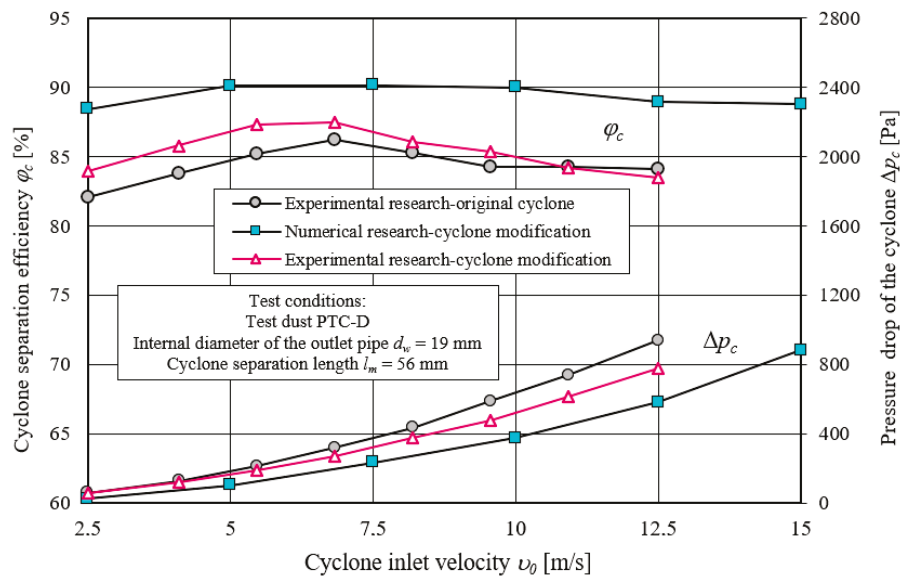


Figure 15. Numerical and experimental test results of cyclone separation efficiency depending on the inlet velocity v_0 .

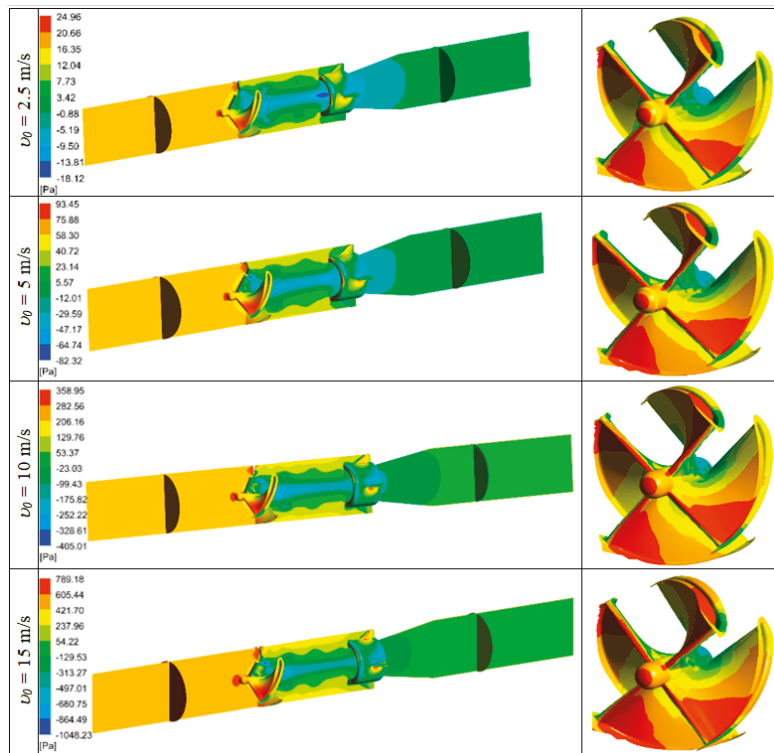


Figure 16. Pressure distribution p_c (in Pa) in axial flow cyclone depending on the different inlet velocities of the cyclone v_0 .

Increasing cyclone separation length l_m naturally extends the spiral grain path until reaching the collecting chamber gap. Each rotation of the dust grain brings it closer to the inner wall of the cylindrical portion until it comes into contact, decelerates due to friction, and falls by gravity into the collecting chamber. The greater the separation length of the cyclone l_m , the greater the possible number of grain rotations. This is especially important for the dust grains that leave the blade close to the swirl core. Their path to the wall is much longer. Thus, the number of particles that reaches the wall of the inner cylindrical

part and the collecting chamber is greater, which increases separation efficiency. As can be seen from Figure 11, the intensity of the increase in separation efficiency decreases at higher values of l_m length. This can be explained by the fact that as one moves away from the swirler, the intensity of the swirl of the jet decreases due to friction, and thus the peripheral component of the velocity, which moves the grains to the cyclone wall, decreases. Hence, a smaller number will get into the collecting chamber.

Cyclone filtration change efficiency depending on the diameter of the inlet of the outlet pipe d_w (original value $d_w = 21$ mm) without changing the diameter of the outlet port D_w and using the streamlined shape of the outlet tube is shown in Figure 12. Increasing the diameter d_w by the following values: $w = 1$ mm, 2 mm causes a decrease in and decreasing it by the same values increases the separation efficiency.

Reducing the diameter d_w has a direct effect on increasing the width b of the collecting chamber, which enables the collection of a larger dust mass, which by the force of inertia reached the channel wall and slides along its cylindrical surface, making a helical movement. This increases separation efficiency. In close proximity of the channel wall, the turbulence of the jet decreases rapidly due to friction. For the same reason, the velocity of the grains in the circumferential direction and the inertia force acting on them in the radial direction also decreases. This increases the possibility of the separated dust being entrained by the air stream swirling towards the outlet tube. Greater width of the collecting chamber b prevents this phenomenon. However, increasing the width b is directly related to reducing the inner diameter of the inlet opening of the outlet tube d_w , which causes an increase in speed at this point and an increase in pressure drop. For the inlet velocity $v_0 = 10$ m/s, the reduction (compared to the original) of the inner diameter of the tube d_w by 2 mm increases separation efficiency from 87.5% to 90% and the increase in the flow velocity at the inlet opening of the outlet tube d_w by over 20%, which significantly increases pressure drop of the cyclone.

Extending the core length by the value: $r = 5, 10, 15$ and 20 mm and ending it with a cone of angle α as shown in Figure 10 did not cause significant changes in cyclone separation efficiency. Figure 13 shows pressure contours in the median plane. A negative pressure area is visible in the area between the swirl generator and the outlet pipe opening. Extending the core towards the outlet tube and ending it with a cone changed the pressure profiles (Figure 13a) and increased velocity around the core (Figure 13b).

In order to verify the model, numerical tests were carried out on the characteristics of the cyclone filtration efficiency $\phi_c = f(v_0)$ depending on the inlet velocity in the range $v_0 = 2.5$ –15 m/s. Tests included a modified cyclone model, which had the separation length increased by 20 mm compared to the original ($l_m = 56$ mm) and cyclone inlet pipe diameter decreased by 2 mm ($d_w = 19$ mm). For these parameters, at inlet velocity of the cyclone $v_0 = 10$ m/s, the cyclone reached maximum values of separation efficiency. A model with such parameters was printed using the additive manufacturing technique, and then its experimental tests were carried out. Separation efficiency characteristics $\phi_c = f(v_0)$ and pressure drop characteristics $\Delta p_c = f(v_0)$ and depending on the inlet velocity in the range of $v_0 = 2.5$ –12.5 m/s were made. The same scope of experimental research covered the original cyclone before the modification.

Experimental tests of axial flow cyclone were carried out on a stand built in accordance with ISO 5011 (Figure 14). "Fine" polydisperse test dust with particle size $d_{zmax} = 80$ μ m was used (Table 1) and chemical composition (Table 2) which was supplied to the cyclone by compressed air dispenser to obtain a dust concentration of 0.5 g/m³ at the cyclone inlet.

To determine separation efficiency ϕ_c of the axial-flow cyclone, the mass method was used, consisting in measuring the dust mass retained by the cyclone m_{Zc} and the mass of dust m_{Dc} delivered to tested cyclone and dust mass retained by the absolute filters m_{A1} and m_{ZA2} at a specified time t_p . Dust mass supplied to the cyclone and the dust mass retained by the absolute filters were determined as the difference between the masses of the dust container and filters before and after the measurement. The measurements

were performed for a fixed value of the inlet air stream to the cyclone Q_0 . There were 8 values of air flows in the range $Q_0 = 10\text{--}55 \text{ m}^3/\text{h}$ at equal intervals, which correspond to the inlet velocity into the cyclone in the range $v_0 = 2.5\text{--}12.5 \text{ m/s}$. For each determined value of the air stream Q_0 , $j = 5$ measurement cycles were performed, during which dust masses necessary to determine the cyclone separation efficiency were determined. For this purpose, an analytical balance with a measuring range of 220 g and an accuracy of 0.1 mg was used. After each measurement cycle j , separation efficiency was determined using the following relationship, and then the average of the measurements was calculated:

$$\varphi_{cj} = \frac{m_{Zcj}}{m_{Dcj}} \cdot 100\% \quad (21)$$

Dust mass retained by the m_{Zcj} cyclone was calculated using the indirect method from the following formula:

$$m_{Zcj} = m_{Dcj} - (m_{A1j} + m_{A2j}) \text{ [g]} \quad (22)$$

The pressure drop in the tested axial flow cyclone Δp_{cj} was determined after each measurement cycle on the basis of the measured static pressure drop Δh_{mj} in the outlet pipe after the cyclone at a distance of $6d_w$ from the end of the cyclone outlet pipe, where d_w —internal diameter of the cyclone outlet pipe. Δh_{mj} was read in mm of H_2O on a U-tube liquid manometer. The value of the pressure drop was calculated using:

$$\Delta p_{cj} = \frac{\Delta h_{mj}}{1000} \cdot (\rho_m - \rho_H) \cdot g \text{ [Pa]} \quad (23)$$

where: ρ_m —manometric liquid density [kg/m^3], ρ_H —air density [kg/m^3], g —gravity acceleration [m/s^2].

Results of the numerical and experimental tests of separation efficiency $\varphi_c = f(v_0)$ and the pressure drop $\Delta p_c = f(v_0)$ depending on the inlet velocity v_0 of the modified cyclone and the results of the experimental tests of the cyclone in the original version are presented in Figure 15.

Cyclone separation efficiency characteristics obtained from numerical tests have a course similar to the characteristics resulting from experimental tests of the original and modified cyclone, while the efficiency from numerical tests obtain higher values in the entire range of the inlet velocity used.

Differences in cyclone separation efficiency characteristics, as to their course and value, should be explained by the adopted model of dust particle size composition. The actual dust consists of several components, each with a different density. Moreover, the shapes of the real dust grains are irregular. The adopted dust model has one fixed density (silica) and dust grains have the shape of a sphere. These factors have a decisive influence on the trajectory of dust particles and their reaching the collecting chamber.

Separation efficiency characteristics of original and modified cyclones follow a similar course, but differ in value. As the inlet velocity v_0 increases, the separation efficiency increases, and after reaching the maximum value for $v_0 = 6.8 \text{ m/s}$, then its slow decrease occurs. Such a course of cyclone characteristic $\varphi_c = f(v_0)$ is consistent with the results of other studies presented in the literature [79].

A cyclone with modifications achieves a higher separation efficiency than the original cyclone in the whole range of inlet velocity v_0 , which confirms the correctness of the design changes introduced in the cyclone.

At the inlet velocity of $v_0 = 6.8 \text{ m/s}$, the efficiency increased from $\varphi_c = 86.2\%$ to $\varphi_c = 87.5\%$, i.e., by 1.5%. Below the speed $v_0 = 6.8 \text{ m/s}$, the increase in separation efficiency of the cyclone caused by the introduced design changes is clearly greater. This enables

the cyclone to operate at lower air stream velocities, i.e., smaller pressure drops, the value of which depends on the speed, based on the equation:

$$\Delta p = \frac{\rho \cdot v^2}{2} \quad (24)$$

As the inlet velocity v_0 increases, pressure drop of tested cyclones increases parabolically, which results from the above equation. A cyclone with the introduced modifications achieved pressure drop lower by about 15% than the original cyclone, which will result in lower engine power losses. Obtaining lower pressure drop of the cyclone results from the streamlined shapes introduced at the inlet of the guide vanes and the outlet tube (Figure 8). Characteristics analysis of $\phi_c = f(v_0)$ and $\Delta p_c = f(v_0)$ (Figure 15) shows that the cyclone should not operate above the inlet velocity of $v_0 = 10$ m/s, because then its operation becomes ineffective. There is then a decrease in separation efficiency and an increase in the cyclone pressure drop.

Test results for the values and pressure profiles p_c for the cyclone model with the maximum value of the separation length $l_m = 56$ mm and the minimum diameter of the inlet pipe $d_w = 19$ mm depending on the inlet velocity in the range $v_0 = 2.5$ –15 m/s are shown in Figure 16.

The highest values of pressure p_c were recorded only on a small initial part of the surface of swirler blades. Field of highest pressure starts from the front of vanes towards the cylindrical part of the hull. Inclination of steering wheel at this point is much greater than near the spindle. For the inlet velocity $v_0 = 2.5$ m/s, the pressure in the front part of the steering wheel is $p_{c1} = 24$. With the increase of the inlet velocity v_0 , the pressure values on a part of the surface of the blades are increasing. For inlet velocity $v_0 = 5$ m/s, the pressure has the value $p_{c1} = 93$ Pa, for $v_0 = 10$ m/s, $p_{c1} = 207$ Pa, and for $v_0 = 15$ m/s, $p_{c1} = 789$ Pa. As the air stream moves to the swirler's outlet, the pressure decreases, but the intensity of decrease is greater in the center of the swirler (near the core) at the point where the blade contacts the cylindrical part of the cyclone hull. As a result, there is a negative pressure in the central part behind the swirler, the greater the greater the inlet velocity. For the inlet velocity $v_0 = 2.5$ m/s, the pressure downstream of the swirler (spindle end) has the value $p_{c2} = -5.19$ Pa, for $v_0 = 5$ m/s $p_{c2} = -29.6$, and for $v_0 = 15$ m/s, $p_{c2} = -497$ Pa (Figure 16).

Figure 15 shows characteristics $\Delta p_c = f(v_0)$ of pressure drop of the cyclone after modification, obtained on the basis of the numerical test results, as the pressure difference between the p_{c1} inlet and p_{c2} outlet of the cyclone, according to the following equation:

$$\Delta p_c = p_{c1} - p_{c2} \quad (25)$$

As the inlet velocity v_0 (air stream) increases, the flow resistance $\Delta p_c = f(v_0^2)$ increases in a parabolic manner, which is consistent with the literature. Characteristics of the cyclone $\Delta p_c = f(v_0)$ obtained from numerical tests have a course similar to the characteristics resulting from experimental tests of the original cyclone, with pressure drop values from the experiment slightly higher in the entire range of the inlet velocity $v_0 = 2.5$ –12.5 m/s.

6. Conclusions

This paper presents the possibilities of modifying the structure of axial flow cyclone in the direction of increasing separation efficiency understood as an increase in efficiency or a decrease in pressure drop. The cyclone, which is an element of the air filter used in a tracked vehicle, has been investigated numerically and experimentally.

Using the commercial engineering software Ansys Fluent, a numerical evaluation of influence of three parameters of this cyclone: separation length l_m , internal diameter of the outlet tube d_w and length of the vortex core l_r on separation efficiency and pressure drop was performed. Based on the results of numerical tests obtained at the inlet velocity $v_0 = 10$ m/s, it was found that:

- changing cyclone separation length l_m in the range of 16–56 mm increases separation efficiency of the cyclone in the range of 79.5–90%, which is consistent with the literature and may be helpful for the verification of the developed model,
- changing the internal diameter of the outlet pipe d_w in the range d_w 19–23 mm causes a decrease in separation efficiency of the cyclone in the range of 90–83.5%, which results from the reduction of the separation chamber space and is consistent with the cyclones operation theory,
- d_r core elongation, in the range of $d_r = 5$ –20 mm, does not have a significant impact on separation efficiency and therefore this change was not applied in the further part of the study.

The cyclone model, in which the separation length of the cyclone l_m and the internal diameter of the outlet pipe d_w were simultaneously changed to the value $l_m = 56$ mm and $d_w = 19$ mm (for these values and at $v_0 = 10$ m/s, cyclone obtained the highest separation efficiency in preliminary tests) shows an increase in the inlet velocity of $v_0 = 2.5$ –7.5 m/s, and after exceeding it, a decrease in separation efficiency. Maximum separation efficiency obtained at $v_0 = 7.5$ m/s has value of $\phi_{cmax} = 90.2\%$. Thus, it is a typical course of $\phi_c = f(v_0)$ of the cyclone separation efficiency characteristic, which proves the correctness of the model.

The obtained characteristic $\phi_c = f(v_0)$ of modified axial flow cyclone was experimentally verified on laboratory stand during experimental tests. Subject of the research was a numerically tested cyclone model, which was printed on a 3D printer using the additive manufacturing technique.

Separation efficiency $\phi_c = f(v_0)$ and the flow resistance $\Delta p_c = f(v_0)$ characteristics depending on the inlet velocity in the range of $v_0 = 2.5$ –12.5 m/s were made. The same scope of experimental research covered the original cyclone-before the modification. Experimental research shows that:

- separation efficiency characteristics of the original cyclone and after the modification follow a similar course, but differ slightly in values;
- characteristic $\phi_c = f(v_0)$ from the numerical tests is similar to the cyclone characteristics from the experimental tests, but takes higher values, which may result from the model assumptions, in particular the particle size composition of the dust used for the tests;
- the modified cyclone achieves at the inlet velocity range $v_0 = 2.5$ –11 m/s higher separation efficiency by 1.5% and lower pressure drop by approximately 15% than the original cyclone, which confirms the correctness of the design changes introduced in the cyclone;
- below speed $v_0 = 6.8$ m/s, increase in cyclone separation efficiency caused by the introduced design changes is clearly greater. This enables cyclone to operate at lower air stream velocities and, therefore, with lower pressure drops, which result in engine energy losses;
- from the course of characteristics $\phi_c = f(v_0)$ and $\Delta p_c = f(v_0)$ it follows that the cyclone should not operate above the inlet velocity $v_0 = 10$ m/s, because its operation becomes ineffective due to a decrease in filtration effectiveness and a simultaneous increase in cyclone pressure drop, which is the source of engine energy losses.

Obtained results of numerical and experimental cyclone tests give a basis to claims that the developed and used model reacts correctly to the introduced changes in the cyclone structure, it predicts well the course of the characteristics $\phi_c = f(v_0)$ and $\Delta p_c = f(v_0)$ and therefore it can be applied for testing subsequent modifications.

Cyclone modification methods demonstrated in this study are expected to be useful in improving the design of two-stage air filters to ensure longer life and more reliable engine operation in off-road environments with high air dust concentrations.

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Article

The Use of the Fourier Series to Analyze the Shaping of Thermodynamic Processes in Heat Engines

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Abstract: The article presents the application of the Fourier series to theoretical considerations on the method of maximum temperature control in thermodynamic cycles of internal combustion engines equipped with an additional independent kinematic system. The analysis assumes that the processes are zero-dimensional and the gases consumed in the engine cycles are perfect, simplifying the considerations for temperature control as a function of the two variables, pressure and volume, of which the volume as a geometric quantity can be completely controlled. In view of this fact, a predetermined temperature curve was assumed, ultimately reducing the considerations of specific volume changes, that is to say a kinematic system that could implement these changes. Moreover, in the analysis of volume changes, a cycle not used so far in the description of internal combustion engines was used. In the next step, the cycle was modified using the popular Vibe function, which was replaced in the theoretical cycle by two isochoric and isothermal transformations. Heat exchange was completely omitted in the considerations, in that it is of secondary importance, ultimately bringing the temperature function to the function of one variable, the angle of rotation of the crankshaft. Then, the kinematics was divided into the kinematics of the crank-piston system and the additional system, which was approximated with five words from the Fourier series, which in the technique correspond, for example, to the system of oscillators. At the end of the article we have explained one of the ways of actual technical implementation using a single nonlinear oscillator, the so-called ACC system equivalent to a few words from the mentioned Fourier series.

Keywords: Carnot; Otto; Atkinson; theoretical thermodynamic cycle; Fourier series; ACC engine; Vibe function

1. Introduction

In this paper Carnot cycle was modified using the popular Vibe function [1], which was replaced in the theoretical cycle by two isochoric and isothermal transformations. We know from the theory of thermal machines [2] that the Carnot cycle achieves the highest efficiency at a given temperature difference, while the Otto cycle achieves the highest efficiency for a given compression ratio. Therefore, in the combination of both circuits, it was decided to look for the best solution. Before we go in to that, however, it is necessary to present the already existing solutions that have reached the stage of technical implementation. In this regard, we can find a large amount of scientific information, which is why it was decided to choose those that contain a compendium of knowledge on the development of spark ignition engines or those that have very unusual solutions or have been published recently. We know from the theory of thermal machines [2] that the

Carnot cycle achieves the highest efficiency at a given temperature difference, while the Otto cycle achieves the highest efficiency for a given compression ratio. Therefore, in the combination of both circuits, it was decided to look for the best solution, but before we get to that, it is necessary to present the already existing solutions that have reached the stage of technical implementation. In this regard, we can find a large amount of scientific information, which is why it was decided to choose that which contained a compendium of knowledge on the development of spark ignition engines or that has very unusual solutions or has been published recently. The selection of specific items from the first group was limited to the basic book and review articles, while the second group is the individual choice of the authors. We found a lot of articles on similar topics, but a few of the recently published, in which development concepts in this field could be seen, were selected in spark ignition engines. The authors of the articles had focused on both the historical context and modern applications, giving them a detailed analysis. In this analysis, we found, among others, the aspect related to the timing system control, which currently seems to be the most common in engines, [3–9].

The methods mentioned can be applied simultaneously [6], which can further increase the efficiency [10–15]. Despite the benefits presented resulting from the implementation of the Atkinson Miller cycle and the very advanced technical solutions used, for which it would be difficult to find an alternative in the form of a similar value of efficiency, for example [16], it was decided to conduct a basic analysis of a different way of implementing the engine's working cycle. The analysis was made because empirical tests confirmed [17] the possibility of increasing the compression ratio in a spark-ignition internal combustion engine to values greater than 18:1, without harmful detonation as described in [18–22]. This, if optimized successfully, could lead to a significant increase in the efficiency of positive-ignition internal combustion engines covering a much wider range of speeds and loads [16], with a concomitant significant reduction in maximum temperature more than the Atkinson Miller-cycle engines mentioned. This other method, practically unused in internal combustion engines, of influencing the thermodynamic process as a system of two pistons, one of which independently regulates the change of compression ratio, has been conceptually known since 1904. The idea has gradually evolved, achieving full freedom in shaping the circuits and a real technical construction described in a patent from 2013. This method consists of an additional, very fast volume change independent of the crank-piston system caused by an additional piston, which allows, for example, some of the fuel to be burned at a constant temperature. To do this, the reaction time of the additional system should be determined on the basis of the combustion time in spark ignition engines. In this case, we can also find many scientific studies, in which HCCI engines will have the shortest combustion times [23,24], while typical combustion descriptions can be found in [4,25], and on this basis, we can determine burning rate or duration of burning.

According to [4], the combustion process of a previously prepared combustible mixture in spark ignition engines takes place spontaneously from the moment of ignition. Of course, this process is determined by the initial thermodynamic parameters, temperature, and pressure, as well as the composition and quality of the combustion mixture preparation. As a rule, the change in volume is neglected, as it is relatively small in the most commonly used crank-piston system, while the rotational speed is taken into account, which forces the duration of the combustion process to be adjusted to the engine's operating cycle. The authors propose a method consisting of a very quick additional, "compensating" volume change in a place where the reaction of typical kinematic systems is insufficient, that is to say, too slow or too fast, or simply not adjusted. Therefore, the analysis of the required kinematic changes in the range corresponding to the combustion process in a typical spark-ignition piston heat engine between 330 ca and 430 ca, which additionally uses a Fourier series in the final phase will be carried out further. The aim is to induce strictly defined volume changes, i.e., kinematic changes responsible and affecting other thermodynamic parameters, pressure, and temperature during the combustion process. For this analytical task, a new

thermodynamic cycle, hereinafter referred to as the proposed one, was selected, with an increased degree of Carnotization, i.e., as close as possible to the Carnot cycle. In practice, we obtain a combustion process with a lower maximum temperature, and thus lower NOx emissions (EURO 6, EURO 7), and at least we maintain the maximum efficiency of the currently used spark-ignition internal combustion engines.

According to the accepted compromise, the proposed thermodynamic cycle can be imagined in a simple way as the Carnot cycle, cut on both sides in the p-V system by isochoric transformations. The proposed cycle is characterized by a high compression ratio, many times lower than in the Carnot cycle, of course, but ranging from 20:1 to even 25:1 [26]. A detailed description of this cycle is included in [1] along with the derived equations and graphic illustrations. Nevertheless, in order to conduct a clear analysis, it should be cited together with a short description. In Figure 1, the proposed theoretical cycle, the green color is distinguished by an additional isothermal transformation, which is not taken into account at all in the theoretical analysis of the current internal combustion engines [2,10]. At this point it can be added that, for a more general case, the isothermal transformation can be replaced by any polytropic transformation, which has also been described [27]. For reference purposes, the same figure shows in red the Otto cycle with a typical compression ratio of 11:1 for spark ignition internal combustion engines.

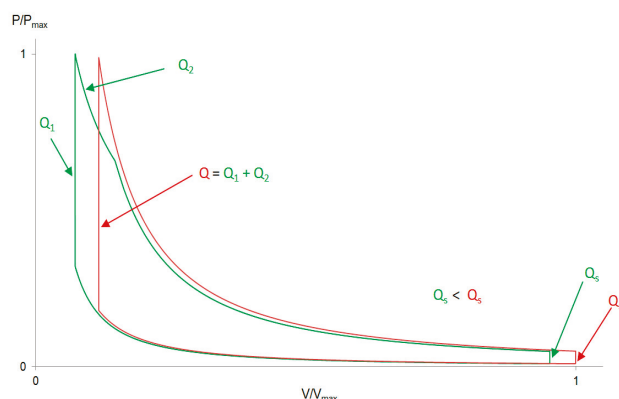


Figure 1. The energy concept of the new theoretical cycle. The proposed theoretical cycle, the green color is distinguished by an additional isothermal transformation, which is not taken into account at all in the theoretical analysis of the current internal combustion engines.

In the proposed cycle, i.e., describing an ideal motor operating at full load, the total energy is supplied to the cycle at a constant volume, the energies Q_1 as in the cycle (Otto) and at a constant temperature, the energy Q_2 as in the cycle (Carnot). The minimum value of the compression ratio for the cycle considered should not be lower than the value of 17:1, otherwise it will not have the acceptable theoretical efficiency value of the Otto cycle with a compression ratio of 11:1. The use of the proposed cycle with a compression ratio lower than 17:1 [27], which should be emphasized, may be appropriate as aimed at reducing the maximum temperature of the cycle, due to the quality of the fuels used or reducing the emission of unfavorable compounds, for example, such as NOx [28–30]. In such a case, it would be advantageous if it would be the first cycle of several successive thermodynamic cycles in a cascade, for example, the main purpose of which would be to improve the unfavorable energy balance of the first. To summarize these brief considerations on the theoretical cycles in Figure 1, the most expected effect for the proposed theoretical cycle should be the possibility to control the maximum temperature in the cycle while maintaining the highest possible efficiency, and also in a more general case, such as for any polytrope exponent while delivering the second energy quantity Q_2 . In short, this means obtaining a relationship between the efficiency of the cycle and its maximum temperature. In the case of temperature, it comes down to selecting one of the graphs presented in the one taken from [1] Figure 2.

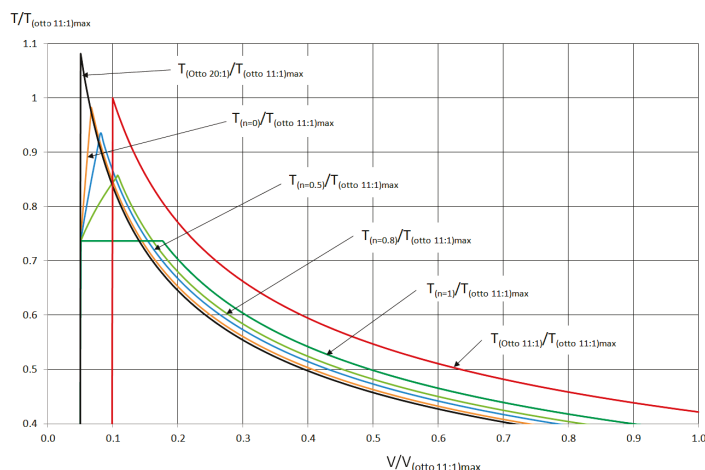


Figure 2. The curve of the maximum temperature in various variants of the proposed theoretical cycle.

The changes presented in Figure 2 are relative changes related to the maximum temperature obtained in the theoretical Otto cycle with a compression ratio of 11:1 and cover only the most important parts of the theoretical cycles. Thus, the value of 1 is taken by Otto in red, and the lowest value of 0.74 is in dark green, the proposed circuit. On the other hand, the highest temperature value of as much as 1.08 is in black, the Otto cycle with a compression ratio of 20:1, while variants with polytropic changes in other colors have values lower than 1. Despite having the same compression ratio of 20:1, the dark green graph is the most interesting because it is characterized by the lowest maximum temperature, the cycle in which Q_2 energy is supplied in an isothermal transition as in Figure 1. For this reason, a mathematical analysis of the problem related to the proposed theoretical cycle was started.

2. Materials and Methods

For the proposed theoretical cycle shown in green in Figure 1, an analysis was performed in which several necessary assumptions were made. The key assumption marked by No. 1 and dark green Figure 2 is that kinematic modifications are only sought for isochoric and isothermal transformations in which heat is supplied, which is equivalent to the combustion process in real engines. After adopting the first assumption—the question arose as to how to create a new kinematic system, since theoretical cycles do not take into account the influence of time, and yet they should correlate with real cycles subject to time. Therefore, at this point, the authors proposed adopting further assumptions, which included:

- (1) The method of heat release as described by the Vibe function with parameters chosen arbitrarily by the Authors, meeting specific requirements for spark ignition engines.
- (2) There is no additional heat transfer in the compression and expansion stroke other than its supply.
- (3) The working gas is perfect.
- (4) Each change in volume is related to the angle of rotation of the crankshaft—as the main kinematic system.

After adopting the above assumptions, the concept of such a transformation of the theoretical cycle may seem strange, but it allows the considerations related to the curve of the combustion and heat transfer process to be omitted [5]. The omission of these processes allowed for a simplification of the problem which led to the construction of mathematical models with analytical solutions. Since the vast majority of heat engines are internal combustion piston engines containing a typical crank-piston mechanism, it was decided to use this system in further analysis, mainly due to its unquestionable quantitative advantage over other systems and over a hundred years of development

history (assumption No. 5). However, at this point, it should be added that although the use of a typical crank-piston system was the most practical because it included the largest number of piston engines—it does not have to be an exception to the analysis. In fact, many other systems can be subjected to a similar analysis, significantly expanding the spectrum of considerations. According to the authors, the analysis of such systems as:

- Atkinson kinematic mechanism [6]
- Atkinson Miller cycle on a rotary internal combustion engine [13]
- MCE5 kinematic mechanism (innocent as to enter an internet source)
- Scuderi kinematic mechanism (innocent as to enter an internet source).

Ultimately, according to assembly No. 5, we thus remained with the first dominant crank-piston kinematic system, while the second kinematic system sought would be the system called “complementary”, a system that adjusted the total kinematics to the desired form. In order to facilitate the description of further mathematical transformations, the theoretical circuit with numbered characteristic points is shown in Figure 3 again.

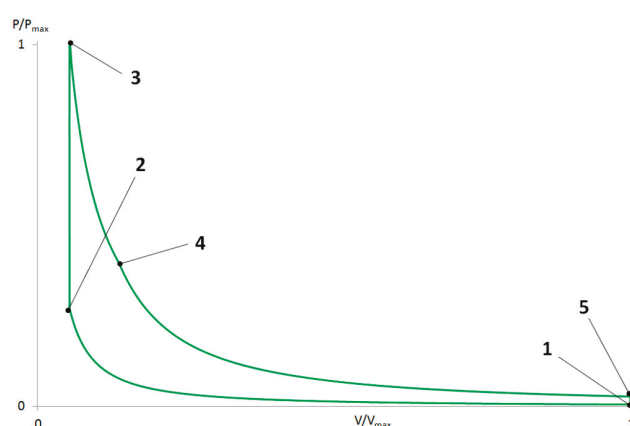


Figure 3. Proposed theoretical cycle with characteristic points, where: 1-2 suction; 2-3 isochoric transformation points; 3-4 the isothermal transformation points; 4-5 depressurization; 5-1 exhaust.

In the first step, the transformation consisted of determining the function describing the temperature in advance, which allowed for the elimination of pressure and the transition from the function of two points $T = \int (p, V)$ to the side of one variable $T = \int (p(V), V)$, i.e., $T = \int (V)$. In the second step, it was assumed that the isochoric transformation points 2-3 and the isothermal transformation points 3-4 from Figure 3 are replaced by the Vibe point, which realistically describes the heat supply to the tools. Adopting the Vibe function described, the ignition moment for the initiation is determined by the ignition advance angle: this caused a slight deformation of isochoric transformations by minimal changes at points 2-3.

The fourth step was to close the closed theoretical open circuit, which gave it the character of a function with the argument of crankshaft rotation. In the last, fifth step, the kinematics created in the fourth step was finally made, the kinematics of the complementary system. The use of the Vibe function to describe the curve of pressure changes in the cylinder as a function of the crankshaft rotation, that is to say to determine the manner in which the Q_1 energy portion and the remaining Q_2 energy portion are released, is an arbitrary decision of the authors. At this point, it should definitely be said that other functions can also be used, which, for example, describe combustion in internal combustion engines in a more accurate manner confirmed by tests taking into account chemical composition [31–34].

The Vibe function has many calculation variants [3,35,36] taking into account at least a few parameters, but despite everything, its characteristic shape seems to undergo only a specific stretching or narrowing along the axis of the argument, the crank angle (ca). However, because it was necessary to provide specific parameters of this function, for

example, to create graphic illustrations in the further part of the analysis, the influence of several most important parameters necessary to determine the Vibe function was taken into account, which included the following values:

- rotational speeds of 1200 rpm, 1500 rpm, 2400 rpm, 3600 rpm, 4800 rpm and 6000 rpm, respectively
- compression ratios 14:1, 17:1, 20:1
- ignition angle values in the range of 335 ca–350 ca.

The adoption of the aforementioned parameters included in Table 1 made it possible to obtain the numerical values necessary to calculate the curve of the Vibe function Equation (1).

$$m_{90\%burn} = 1 - e^{6908 \left(\frac{\phi_{c1} - \phi_1}{\phi_{c1}} \right)^{n_1}} \quad (1)$$

Table 1. Parameters for creating a Vibe function.

Constant Parameters	Cr	Rpm	Vibe Parameters		
			ϕ_1	f_2	n
K = 1.35	14:1	1200			
		1500	349.088	30.389	3.7572
		2400	346.225	35.300	3.6883
		3600	341.912	43.7488	3.4470
		4800	339.379	49.9006	3.2058
		6000	338.036	53.9323	3.0323
	17:1	1200	337.197	56.9553	2.8995
		1500	350.096	28.9820	3.8511
		2400	347.345	33.718	3.7856
		3600	343.215	41.8482	3.5417
		4800	340.811	47.7463	3.2947
		6000	339.547	51.6004	3.1165
	20:1	1200	338.765	54.4851	2.9799
		1500	350.911	27.8934	3.9062
		2400	348.249	32.4930	3.8445
		3600	344.264	40.3887	3.6006
		4800	341.958	46.1055	3.3505
		6000	340.755	49.8352	3.1694

The description of the kinematics of a specific complementary system requires the adoption of further assumptions containing specific numerical values of the additional volume change. The assumptions made were:

- Maximum temperature in all calculations is constant and 27.5% lower than the maximum temperature in the Otto cycle with a compression ratio of 11:1.
- Each of the back-ups will start a kinematic shift at the same crankshaft position of 368 ca (8 ° ca after TDC).
- The value of the cp/cv exponent will be 1.35 in all of them

After all the energy is supplied, which takes place at the end of the isothermal transformation, the energy from the complementary system is also returned in isothermal transformation, and the complementary system kinematics returns to the starting position. At this point, we can write general equation (2) describing the kinematics together with the complementary kinematics of the engine implementing the isothermal transformation from

368 ca, which after introducing CA as an argument of the function will assume the final form (3).

$$V_{T=\text{const}} = V_c + V_F \quad (2)$$

$$V_{T=\text{const}} = V_{\min} + \frac{V_s}{S} \left[\sqrt{\left(\left(1 + \frac{S}{2} \right)^2 - e^2 \right)} - \left(\frac{S}{2} \cos(\varphi) + l \cos(\gamma) \right) \right] + V_{F(\text{ca})} \quad (3)$$

where:

V_k —the kinematic volume of the engine,

V_c —the total volume of the piston,

$V_{\min}$ —the minimal volume of the combustion chamber,

V_s —the displacement volume of the engine,

s —the displacement of the piston,

l —the length of the connecting rod,

V_{sp} —the volume of the additional cylinder,

A_{sp} —the area of the surface of the additional piston in the engine's cylinder.

After introducing the Vibe function and the isothermal transformation conditions, the volume change in the complementary system can be finally written by:

$$V_{F(\text{ca})} = V_{T=\text{const}} - V_{\min} + \frac{V_s}{S} \left[\sqrt{\left(\left(1 + \frac{S}{2} \right)^2 - e^2 \right)} - \left(\frac{S}{2} \cos(\varphi) + l \cos(\gamma) \right) \right] \quad (4)$$

Substituting the numerical values from Table 1 into the above equation, subsequent charts from Figures were created. Seventy-four, Figures 4–6 illustrating the effect of the complement system. The charts in Figures 4b, 5b and 6b were intentionally scaled in the relative volume, i.e., related to the maximum volume, to show how small the required range of changes is. A characteristic feature of these graphs presented in Figures 7–9 is their smooth course and tendency to decrease changes with increasing rotational speed, which is important in Fourier analysis, which in turn is consistent with logic.

In turn, increasing the compression ratio does not require an increase in the reaction of the complementary system, but a slight acceleration of this reaction.

Performing the calculations according to the mentioned steps and adopting the parameters of the function Vibe from Table 1 for 1500 rpm, the final result was obtained as illustrated by the graphs in Figures 4a, 5a and 6a. Next to each of the index charts, the required changes to the complementary structure are provided. The dashed line on each Figures 4b, 5b and 6b indicates additional changes in the volume presented in figures with the index a, i.e., 41, Figures 5a and 6a, respectively. For the graphs in Figures 4b, 5b and 6b, it can be assessed that the predicted maximum volume change is achieved at a minimum speed of 1200 rpm and remains within 10 to 11% percent of the total volume, which will turn out to be favorable for technical solutions in that they must bring about these changes. In the complementary system, despite the clearly increasing compression ratio, no significant changes in volume can be observed, as shown in Figure 10. These differences become significant for another parameter determining the curve of the Vibe function, the rotational speed, which varies widely from 1200 rpm up to 6000 rpm, as shown by the colored runs of the complementary circuit in each of the figures. It should be mentioned that the biggest change in the complementary system corresponding to the rotational speed of 1200 rpm is due to the shortening of the Vibe function interval, and in this case, the complementary system is primarily responsible for the volume change. Analogies of this situation can be found in many charts with the maximum torque of real spark ignition engines, where the lack of a complementary system makes it impossible to use the full load, which would certainly result in detonation [37–39]. In these graphs, we will see how the load is deliberately reduced to compensate for the slow volume changes favoring the detonation phenomenon, which gradually disappears as the rotational speed increases.

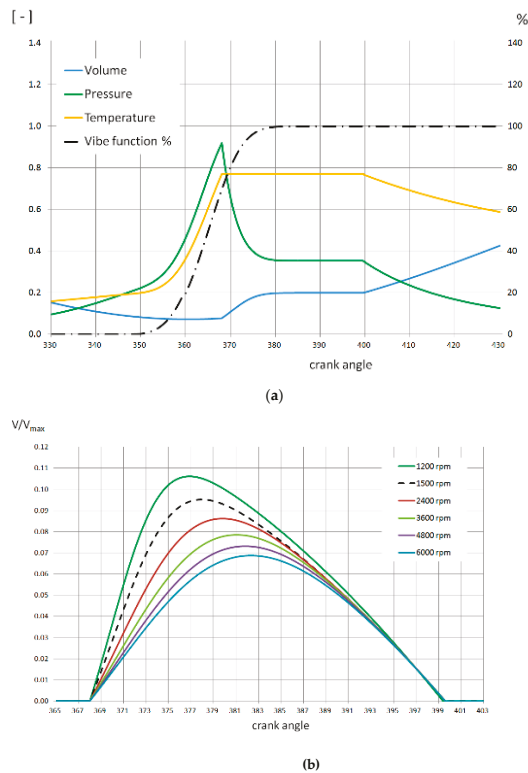


Figure 4. (a) A fragment of the parameters V , p , T , and the Vibe function for the proposed theoretical cycle in the form of an open diagram for the compression ratio cr 14:1. (b) required reaction of the complementing system cr 14:1 as a function of rotational speed (rpm).

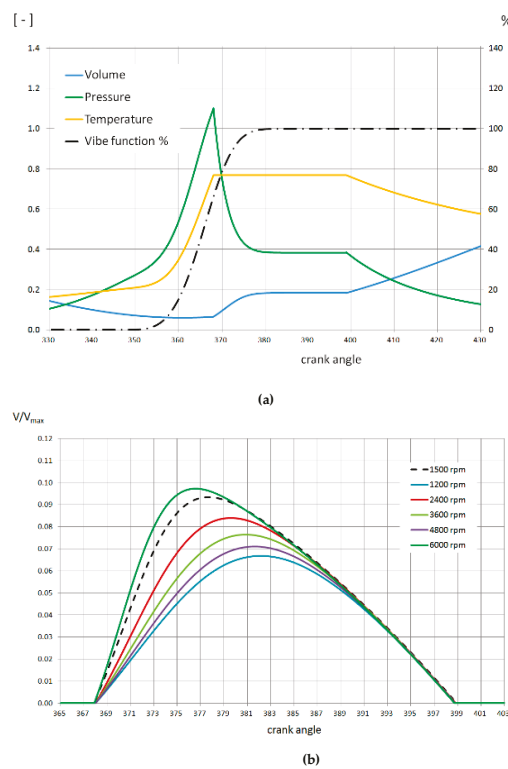


Figure 5. (a) A fragment of the parameters V , p , T , and the Vibe function for the proposed theoretical cycle in the form of an open diagram for $cr = 17:1$, (b) required reaction of the complementing system cr 17:1 as a function of rotational speed (rpm).

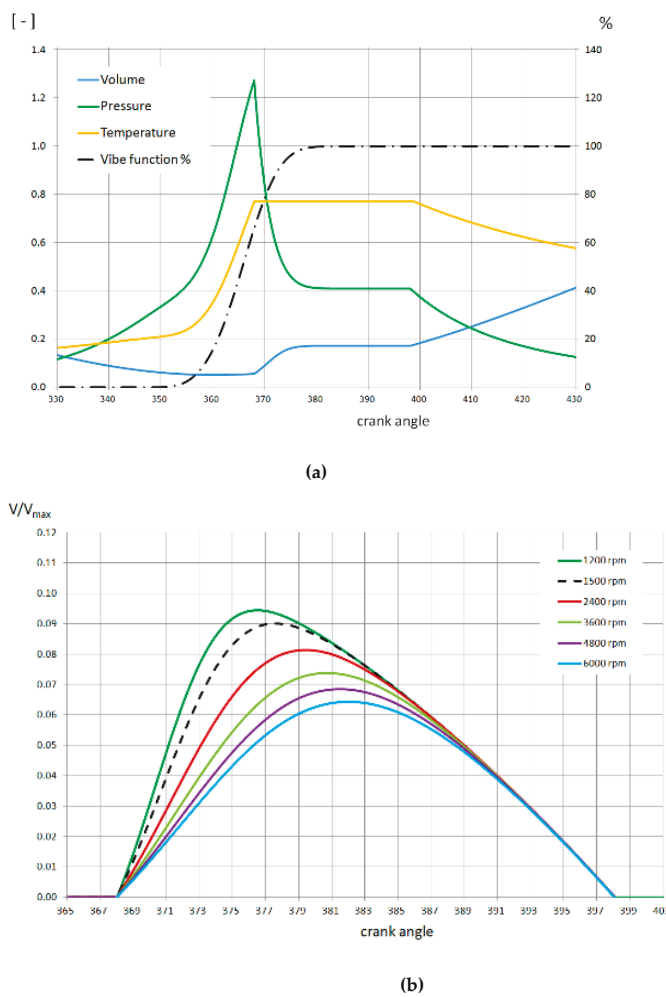


Figure 6. (a) A fragment of the parameters V , p , T , and the Vibe function for the proposed theoretical cycle in the form of an open diagram for $cr = 20:1$, (b) the required reaction of the complementing system $cr 20:1$ as a function of rotational speed (rpm).

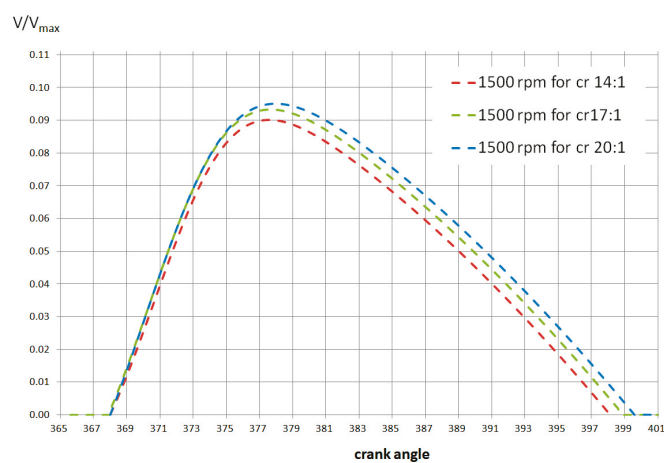


Figure 7. Comparison of volume changes in the complementary system for the rotational speed of 1500 rpm and different stages, respectively: $cr = 14:1$, $cr = 17:1$ and $cr = 20:1$.

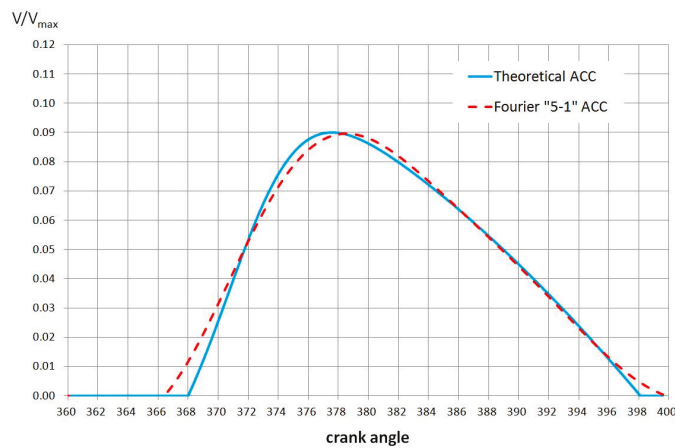


Figure 8. Comparison of volume changes in the complementary system, calculated analytically with those determined by five terms of the Fourier series for 1500 rpm, $cr = 14:1$.

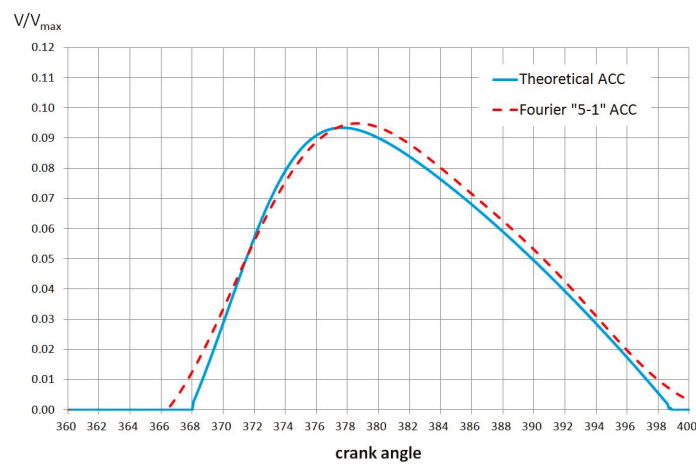


Figure 9. Comparison of volume changes in the complementary system calculated analytically with those determined by five terms of the Fourier series for 1500 rpm, $cr = 17:1$.

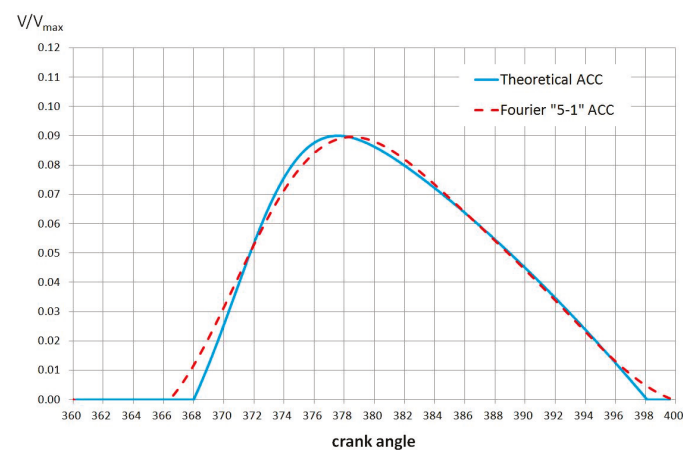


Figure 10. Comparison of the analytically calculated volume changes in the complementary system with those determined by five terms of the Fourier series for 1500 rpm, $cr = 20:1$.

It was possible to complete the analysis at this stage because the kinematics of the complementary system were obtained: it was possible to implement the proposed cycle. However, in the analysis cited there is no recipe for technical implementation. For this

purpose, it was decided to use Fourier's analysis, that is to say its series, the words of which are described by equations well known, for example, describing mechanical oscillators, crank systems, or electrical circuits. Besides this, the Fourier series is used to balance the internal combustion engines themselves [4], showing that only three words give excellent technical effects in the form of counter-rotating weights compensating for its unbalance. Among other things, further analysis was undertaken for these reasons, assuming that mechanical oscillators would be a specific basis for the search for complementary kinematic systems.

In order to perform a Fourier analysis certain conditions must be met for the function subject to expand into this series, in our case two functions. As a reminder, they are listed below.

We call any series of characters by a trigonometric series:

$$\frac{a_0}{2} + \sum_{n=1}^{\infty} a_n \cos\left(\frac{n\pi x}{T}\right) + b_n \sin\left(\frac{n\pi x}{T}\right) \quad (5)$$

where: T is a positive real number, each term in the series is a $2T$ periodic function.

The Weierstrass theorem states that any continuous and periodic function can be approximated with any precision by a trigonometric series. We suppose that we are dealing with a function f integrable on the interval $[-T, T]$. Here is a trigonometric series, in which the coefficients a_n and b_n are determined from the equations:

$$\begin{cases} a_n = \frac{1}{T} \int_{-T}^T f(x) \cos\left(\frac{n\pi x}{T}\right) dx \\ b_n = \frac{1}{T} \int_{-T}^T f(x) \sin\left(\frac{n\pi x}{T}\right) dx \end{cases} \quad \text{for } n = 1, 2, 3, \dots \quad (6)$$

we call this the Fourier series. To answer the question when a function f is the sum of its Fourier series, let us enter the Dirichlet conditions:

1. The function f has only a finite number of type I discontinuity points on the interval $[-T, T]$, and in each of these points x_* is equal

$$f(x_*) = \frac{1}{2} (f(x_*^-) + f(x_*^+)) \quad (7)$$

2. There is an equation:

$$f(-T) = f(T) = \frac{1}{2} (f(T^-) + f(T^+)) \quad (8)$$

The 3rd interval $[-T, T]$ can be divided by points x_0, x_1, x_N on N intervals such that on each interval $[x_n, x_{n+1}]$, $n = 0, \dots, N-1$ function is continuous and monotonic.

Theorem 1. *If the function f , integrable on the interval $[-T, T]$, satisfies the Dirichlet conditions, it is the sum of its Fourier series.*

Note that if a function is odd (i.e., satisfies $f(-x) = -f(x)$) then only the coefficients b_n (and coefficients are zeros) appear in its Fourier solution. Likewise, if the function is even (i.e., satisfies $f(x) = f(-x)$) then only the coefficients a_n can appear in its Fourier expansion (the coefficients b_n are zeros).

In the case of a crankset, the condition for continuity is obvious: the same applies to the Vibe function, which by its nature is continuous, so the above assumptions are met. On the other hand, the measurements of the actual averaged (smooth) pressure curves indicated in the engine almost always seem to be satisfied.

Having quoted the above conditions, it is possible to predict and even expect at this stage that the kinematics of the volume change the system with such a series or a specific number of its words, leading to the creation of an isothermal transformation or another very similar to it in a real engine.

3. Results

After determining the volume change function in the complementary system, its decomposition into trigonometric functions included in the Fourier series was started. In the analysis carried out below, five words from the Fourier series will be used, but of course, the number of these terms can be increased or decreased by reaching the assumed level of accuracy of the complementary system representation (or other assumed thermodynamic criterion). The numerical values of the coefficients to be developed for the five words as previously are summarized in Table 2 and the general equation (x) is given below.

$$V_{F(ca)} \approx a_1^{rpm} \sin(4x) + a_2^{rpm} \sin(8x) + a_3^{rpm} \sin(12x) + a_4^{rpm} \sin(16x) + a_5^{rpm} \sin(20x)$$

Table 2. Coefficients.

cr	rpm	a ₁	a ₂	a ₃	a ₄	a ₅
14:1	1200	0.078718	0.011653	−0.022814	−0.012141	−0.009962
	1500	0.071430	0.007490	−0.022002	−0.009800	−0.004288
	2400	0.065596	0.002333	−0.021861	−0.006159	−0.003153
	3600	0.060440	−0.001021	−0.020488	−0.003426	−0.002528
	4800	0.056627	−0.002797	−0.019079	−0.001933	−0.002822
	6000	0.132480	−0.004040	−0.001939	−0.007415	−0.002717
17:1	1200	0.070661	0.011940	−0.021313	−0.011261	−0.009106
	1500	0.068714	0.009289	−0.021850	−0.009806	−0.003878
	2400	0.062317	0.00384	−0.022160	−0.006123	−0.003878
	3600	0.057358	0.000457	0.004094	−0.003329	−0.002659
	4800	0.053703	−0.001382	−0.019560	−0.001745	−0.002312
	6000	0.050785	−0.002484	−0.018356	−0.000773	−0.002212
20:1	1200	0.066977	0.013058	−0.021146	−0.011142	−0.004241
	1500	0.064353	0.010148	−0.021995	−0.009661	−0.006483
	2400	0.058918	0.005096	−0.022141	−0.006137	−0.003368
	3600	0.054122	0.001726	−0.020987	−0.003329	−0.002088
	4800	0.050593	−0.000135	−0.019688	−0.001697	−0.001733
	6000	0.047767	−0.001263	−0.018502	−0.000697	−0.001639

After determining these coefficients, one could create graphs analogous to those in Figures 7–9, Figures 4b, 5b and 6b, but limited to three selected for 1500 rpm and cr 14:1, cr 17:1 and cr 20:1. On the other hand, the waveforms of the functions created using the Fourier series, which can be found in Figures 7–9, were compared with the calculated changes in the volume in the complementary system.

From the analysis performed, which resulted in the comparative graphs in Figures 7–9, it appears that there is a potential to replace the complementary system with a finite number of oscillators, allowing for a constant temperature to be obtained during the supply of the second energy portion (combustion in the second phase), for example. Again, at this stage, it would be possible to complete the determination of parameters necessary to build a function with an additional change of volume using the Fourier series, but the system excited at the cost of energy from the circulation, according to the assumptions adopted, should return it back to circulation in the most favorable way possible. In general, it was assumed that the energy absorbed by the make-up system is returned to the engine with an efficiency similar to the cycle realized [1], which takes place in an isochoric transformation at constant pressure, so it has no thermodynamic effects. This can be imagined in such a way that the increase in volume in the crank-piston system is 100% compensated for by the decreasing volume of the complementary system which returns energy to the circuit (for example in the ACC system). The curve of the relative energy related to the total energy supplied to the circuit in the complementary circuit is shown in Figure 10, along with the corresponding relative change in volume in this system.

For Figure 11 presented the change in the flow of relative mechanical energy in the complementary circuit, its efficiency was determined to be approximately 63%, which is a fully acceptable value that meets the criterion adopted in [1]. The rest of the energy was converted into kinetic energy, which also takes place in real complementary systems, in which, additionally, friction is generated, completely omitted in the considerations. In Figure 10, we can see how the mechanical energy in the complementary system during volume expansion was accumulated and then returned to circulation during volume reduction, which must take place at a precisely defined angular moment of the cycle and obviously is associated with the very short reaction time of the complementary system, especially for real engines. Summarizing the above considerations, it can be stated that kinematically it would be possible to replace the complementary system with a small number of oscillators, and preferably with one replacing the others, as shown in Figure 12a,b, which shows the graph with the waveforms of the five Fourier harmonics and the Figure 12b one-mass equivalent system.

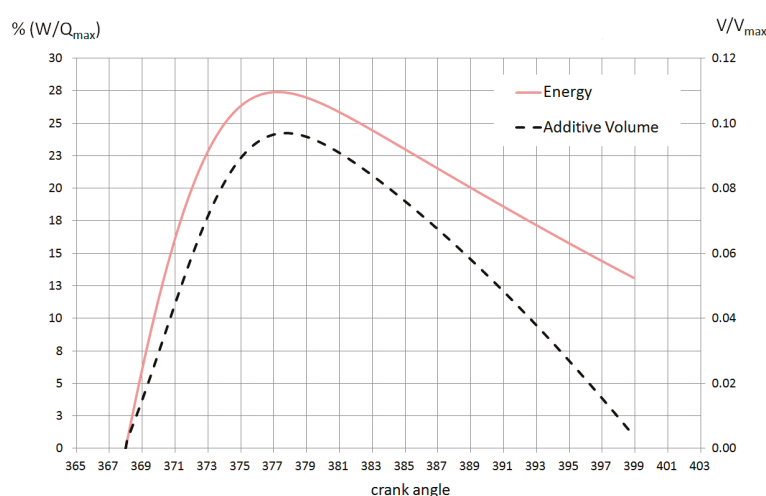


Figure 11. The flow of mechanical energy through the complementary system with respect to the relative volume in this system (1500 rpm, $cr = 17:1$).

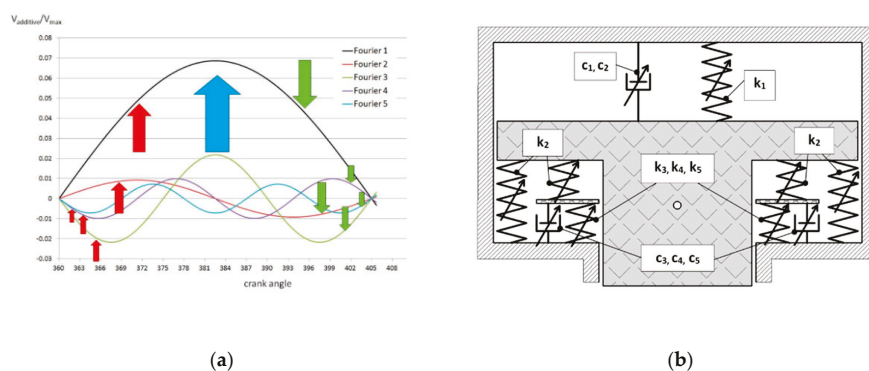


Figure 12. (a) Diagram of the five Fourier oscillations and the exciting and extinguishing forces. (b) model of the system realizing the oscillations in the system. All elements k_1 – k_5 and C_1 – C_5 from Figure 12b have been replaced with one symbolic spring, which should facilitate the analysis of the accumulation and recuperation process by the ACC system.

Figure 12a shows forces in multimass systems that would be inherently difficult to control, and how they were eliminated in favor of the more complex single-mass system in Figure 12b. In this simple way, a technically acceptable solution was obtained that could work in a very unfavorable environment of high pressure temperatures and great acceleration values.

Without going into details of this single-mass system equipped with nonlinear springs and damping the dynamics of the mentioned systems, and limiting only to changes in volume, the drawing with the next diagram is shown in Figure 13, i.e., the expected changes in the position of the single-mass system at its characteristic points with respect to the crankshaft. This time the single-mass circuit has been replaced with the real circuit with the so-called ACC system, which was drawn in the diagram from Figure 13 in the correct proportions and at characteristic points, and at its characteristic points of location in relation to the crankshaft. In Figure 13, the complexity of the support from the diagram in Figure 12b has been replaced by a single symbolic spring. Below the diagram there are typical parameter values for the characteristic positions of the parameter values, which were determined on the basis of empirical tests [27] obtained for various rotational speeds and loads.

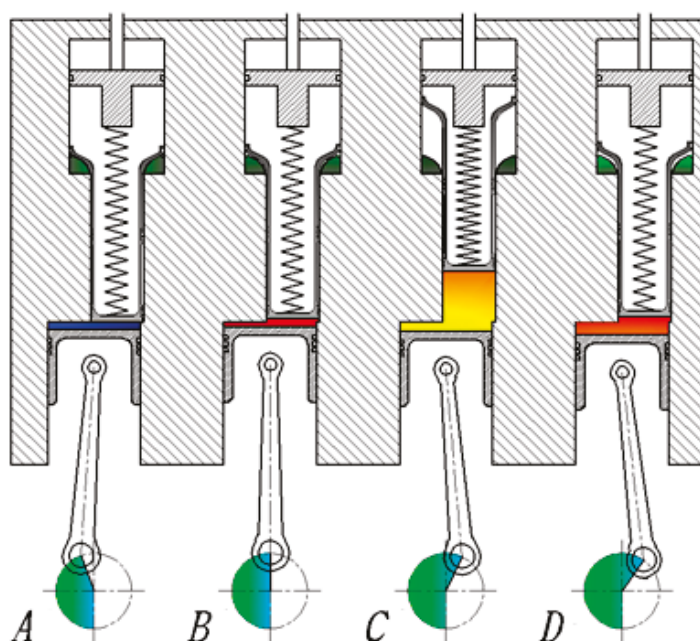


Figure 13. Real system ACC at key operating points.

A—ignition moment (30° – 10° ca before TDC),

B—TDC.

C—maximum deflection of the piston in the ACC system (20° – 40° ca after TDC).

D—first minimum deflection of the additional piston in the ACC system (30° – 60° ca after TDC).

It should be noted here that the technical implementation of the adopted concept in a diesel engine requires the necessary condition from the ACC system shown in Figure 13, which is “appropriate reaction time, i.e., the time in which an additional change in volume occurs, because it must be compatible with time of burning”. In the case under consideration, it was assumed that the time should be comparable to that resulting from the Vibe function, i.e., taking into account the angle of rotation of the crankshaft. Ignoring the detonation combustion situations occurring, for example, in HCCI engines [9,11], the combustion time of which is too short (0.05 ms to 0.2 ms), it can be approximated that in the current combustion systems of engines with GI the burning time is between 0.4ms and 3ms, which is mainly due to the load and the combustion system used, which is approximated by the Vibe function [5]. The time from the theoretical analysis is shorter, 50 to 60%, than the empirical results presented in Figures 14–16, because the analysis did not take into account the heat transfer and the effect of slowing down the combustion process for faster volumetric work by the ACC system.

The ACC system, which is a viable technical solution, performs an additional volume change using the so-called additional piston with a very low weight (approximately 130 g)

and pneumatic support. Pneumatic support, which is nonlinear by definition, because the Clapeyron equation described with a fair approximation is an advantage, fits better into the cycle modifications. In addition, the system is equipped with the possibility of adjusting the volume and pressure of the supply, allowing the system to be adjusted to various load and rotational speeds. In this respect, kinematically related volume change systems clearly differ from the ACC system with regulation, which allows at least a good kinematic approximation to an ideal complementary system cooperating with a crank-piston system, but there is no problem of energy recuperation, which may ultimately determine the advantage one of the solutions. It is of great importance for the traction motor that the ACC motor works with a very high compression ratio at partial loads at least 18:1.

4. Discussion

For a credible discussion, empirical measurements made in the engine with the ACC system turned out to be necessary, which eliminated all doubts regarding the technical implementation. The presented measurements were carried out in an engine with a maximum compression ratio of 17.5:1, in which only one parameter was changed in the ACC system (complementary), it was the value of the supply pressure, which was changed in steps from 13 bar, 11 bar and 10 bar. The value of the rotational speed was close to the value of $1500 \text{ rpm} \pm 15 \text{ rpm}$, which corresponds to the dashed lines in Figures 14–16. The results of empirical tests were presented in the form of open diagrams of the pressure course and the change in relative volume caused by the movement of the additional piston from the ACC system. To facilitate the interpretation, the same range of the values of the crankshaft rotation angle from 330° ca to 430° ca was kept, also in order to show the mutual correlation with the diagrams of theoretical cycles in Figures 4a, 5a and 6a. In the presentation of the results, the following sequence was observed from the highest (13 bar) to the lowest (10 bar) supply pressure of the ACC system. The graphs are presented for an incomplete load by placing on each of the three graphs the corresponding volume change in the theoretical complementary system calculated for a full load using the Vibe function. The graph in Figure 13 corresponds to the maximum cr 17.1:1 that was achieved at part load (approximately 65%) for a speed of 1500 rpm in the engine without the intervention of the ACC system. The graph from Figure 14 will be a direct reference for the other two from Figures 15 and 16, which were created for a similar load, but the effect of ACC operation is visible because the change was caused by reducing the supply pressure of the tank in the system, i.e., the parameters of the symbolic spring from Figure 13 to the values of 11 bar and 10 bar. In this way, the compression ratio was reduced to cr = 16.3:1 Figure 15 and cr = 15.8:1 Figure 16, by modifying the pressure course, and thus also the temperature course after TDC.

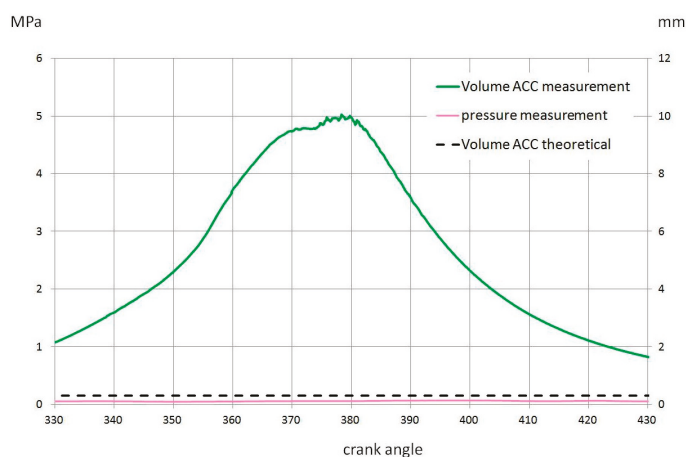


Figure 14. Curve of indicated pressure without volume change in the ACC system, (cr = 17.1: 1, 13 bar) without relative volume in the theoretical complementary system (cr = 17:1).

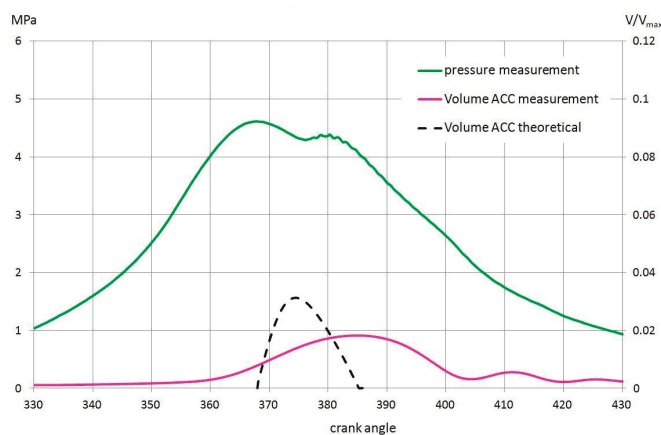


Figure 15. Curve of indicated pressure and relative change in volume in the ACC system ($cr = 16.3:1$, 11 bar) and relative volume in the theoretical complementary system ($cr = 17:1$).

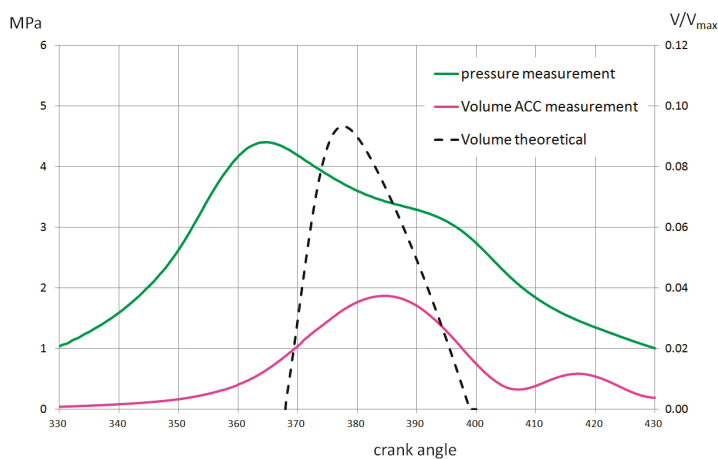


Figure 16. Curve of the indicated pressure and the relative change in volume in the ACC system ($cr = 15.8:1$, 10 bar) and the relative volume in the theoretical complementary system ($cr = 17:1$).

The presented graphs Figures 15 and 16 show some delay in the change of the maximum volume between the real ACC system (pink line) and the theoretically determined system (black dashed line), although it should be noted that the phase consistency is quite good for the arbitrary adoption of the values contained in Table 1. In a real ACC engine we deal with faster work and which slows down the combustion reaction and causes its extension, which, as it turns out, will not prevent the temperature from being kept at a constant level much lower than in theoretical considerations. On the basis of the averaged measurements of the indicated pressure and the total change in the volume of the crank-piston system with the ACC system presented in Figures 14–16, the temperature course in the ACC engine was reconstructed using the equations from [17]. In presenting the results of this simulation, the derivatives of these changes were used instead of changes in the total volume, which were relatively small. This treatment was aimed at sharpening the areas where the influence of the ACC system was most visible.

When assessing the values of the maximum temperature from the graphs in Figures 17–19 in the vicinity, we can see that the minimum is present in Figure 18. This is the most important evidence obtained from the empirical studies, from which directly follows that temperature regulation. It is also obvious that the make-up system must be tuned to a single cycle in the cylinder of the ACC engine in order to properly regulate the maximum temperature. In order to finally confirm the simulation results, the last photoelectric measurements were carried out using a quartz sight glass, behind which a phototransistor was placed to measure the radiation intensity in the combustion chamber

and confirm the quality of the simulation based on empirical results. The results of this measurement, combined with the course of the indicated pressure are presented in Figure 20 and can be considered as fully fledged empirical evidence. Above statement confirm the fact that for properly set parameters of the engine and the ACC system, it is possible to maintain the maximum temperature at a certain level, e.g., close to a constant level. This takes place in the second part of the combustion of the air-fuel mixture in a spark-ignition internal combustion engine and is consistent with the analysis performed.

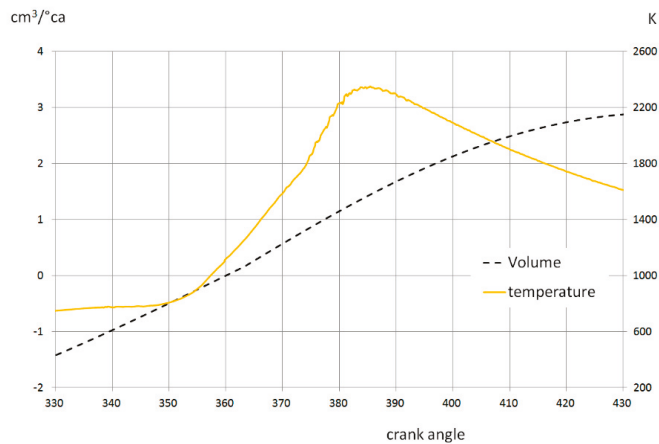


Figure 17. Temperature curve and volume changes (1500 rpm, 13 bar).

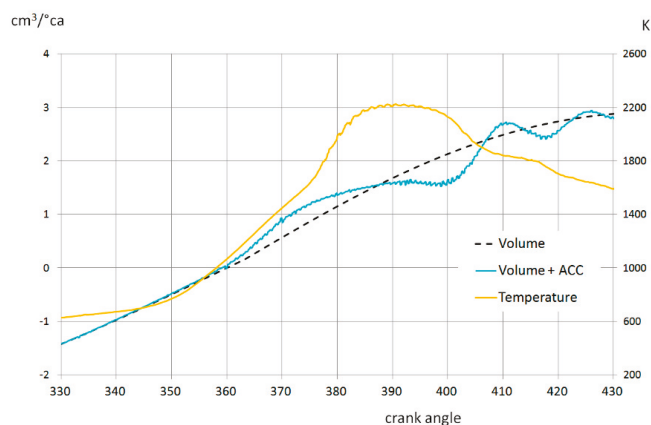


Figure 18. Temperature curve and volume changes (1500 rpm, 11 bar).

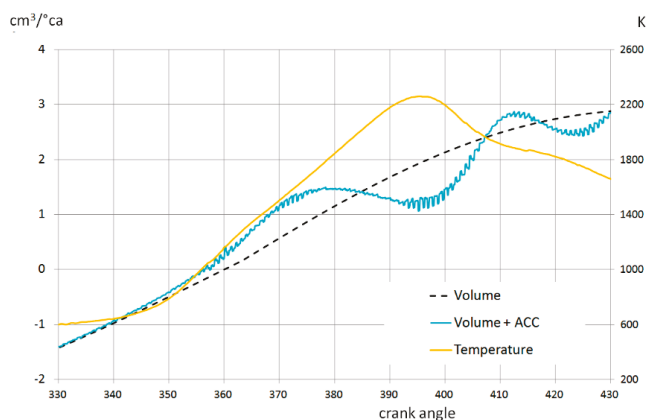


Figure 19. Temperature course and volume changes (1500 rpm, 10 bar).

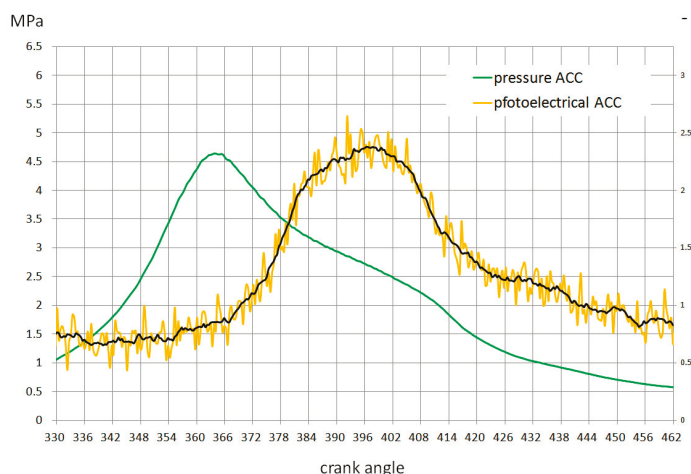


Figure 20. The change in indicated pressure and radiation intensity recorded by the phototransistor.

It should be added that the photoelectric tests conducted were not used to determine a specific value of the maximum temperature, but to show its curve during the combustion process, in the aforementioned verification of the simulation model, because the model showed a clear shift of the temperature maximum in the ACC engine compared to those reported in the literature [7–10]. Comparing the simulation curve in Figure 20 with the curve of the radiation intensity change in Figure 20, it can be concluded that the simulation model reproducing the temperature from real measurements is correct, and most importantly, phase and characterologically compatible with the photoelectric measurement, which confirms the control concept using the ACC system. The use of a compression ratio above 14:1 in the theoretical analysis may also raise objections, due to the possibility of exceeding the so-called GI in the real engine, self-ignition temperature around TDC point, and inducing detonation combustion. However, the actual engine, as shown in the example of the ACC engine, will require significantly higher values of the ignition advance angle, by at least 5 ca to 15 ca compared to the typical values, which in turn, combined with the very fast response of the ACC system, completely prevents this phenomenon from occurring. An additional situation is improved by the very intense flow of the combustible mixture forced by the additional piston from the ACC system, which eliminates the areas favorable for detonation at full load. During the research conducted, it was always possible to “approach the limit of the detonation”, while avoiding it. For this purpose, a “fine-tuning” effect was used, the control of the supply volume and pressure in the ACC system. It should be added that the Vibe functions were selected due to their frequent use in zero-dimensional models, but in engines equipped with mechanical oscillators (ACC motor), these should undergo additional modifications as the intervention of the ACC system increases. In summary, lowering the combustion temperature while maintaining or increasing the overall efficiency of the engine is possible only by, for example, using mechanical oscillators whose task is a specific manipulation of additional volumetric work in the cycle. This method may in the future lead to the return of the concept of quasi-adiabatic engines, and even engines implementing a cascade of several thermodynamic cycles, increasing the scale of Carnotization. It would be interesting to conduct an analysis on an even more basic level, such as in [14–16]. In general, obtaining a certain maximum temperature using the combined kinematics of the crank-piston and the complementary system is theoretically entirely possible, as indicated by the conducted analysis. Unfortunately, the analysis also reveals the following fact, namely that each change in rotational speed requires different kinematics from the complementary system, as shown in Figures 8–10. A similar situation takes place when the load changes, as shown by the empirical results in Figures 17–20, and although this case was not considered in the analysis, assuming the maximum (rated) load in advance, its changes should also be taken into account in a broader context.

The conclusion to use the Fourier series to decompose the kinematics of the complementary system into it has a specific purpose. This goal is to identify technical devices that can replace it and reveal physical limitations, related to the masses and frequencies of these devices, for example, on the basis of which it is possible to predefine the geometric dimensions enabling installation in the engine. For these reasons, among others, the number of five words was used, which was considered adequate for the problem under consideration. Moreover, this series can also be optimized by additionally reducing the number of words, for example to three, maintaining a high correlation coefficient. The main assumption in the theoretical considerations was that the functions describing the combustion processes are continuous: therefore, the Vibe function was selected as very good material with which to modify the theoretical cycle.

Although the considerations completely omit the heat transfer and the influence of the additional system on the Vibe combustion function itself, it does not matter for the analysis itself: when the conditions for the continuity of the function and the assumptions of the analysis are met, the issues mentioned also describe continuous functions. What is worth emphasizing in the empirical results is that continuity was inherently maintained despite the interference of the ACC system, and this can be seen in the graphs from Figures 18–20. The use of mechanical oscillators as equivalents of Fourier series terms made sense, as shown in the graphs in Figures 12–14. On that basis, it could be assumed that a real engine working according to the proposed cycle would always have a lower maximum temperature while maintaining similar or increased efficiency compared to a typical engine with a constant, much lower compression ratio. This is due to the distribution of the supply of the second portion of heat Q_2 in a more even, and thus more advantageous way because it enables the use of a significantly higher compression ratio of the main parameter responsible for efficiency. Additionally, taking into account the fact that some of the energy is lost by the complementary system outside the cylinder, one should always expect a lower temperature in the actual engine. The aforementioned ACC engine, presented schematically in Figure 18, is one of the first examples of this type that to a large extent implements the theoretical concept, which clearly confirms the correctness of the assumptions made. Figures 18–20 show a clear correlation between the theoretical and real charts. On this basis, it can be concluded that the concept of controlling the maximum temperature in the internal combustion engine as proposed, using one or more mechanical oscillators, is technically feasible. Importantly, the use of even one nonlinear oscillator with pneumatic support gives measurable results, provided that its parameters are properly selected [40,41]. It is very important not to forget that the oscillator influence process consists of two main phases, the accumulation phase, and the energy recuperation phase. For example, in an ACC motor, energy is returned to the motor, and this must in turn be efficient. At this point, the authors propose that this criterion should be the value of the efficiency of the theoretical cycle [1], which is implemented by the real engine. This should be understood more generally as follows, that the lowest efficiency of the energy transformation performed by mechanical oscillators, for example. In order to obtain a maximum temperature of a specific value in real engines, first of all, it is necessary to indicate the place of change in the volume of the complementary system and to determine experimentally its minimum value.

Based on the theoretical considerations, it can be done only initially, while on the basis of empirical studies, the required volume changes at full load can be quantified and located in a specific place in the engine cycle. According to preliminary studies for the ACC engine, the maximum volume changes are not that large, as they constitute about 15% of the swept volume and approximately occur in places coinciding with the theoretical analysis in which the Vibe function was used. This can be seen especially in Figure 19, where the highest compliance occurs at the PAE supply pressure of 11 bar. The graphs with lower pressures in Figure 20 are no longer so consistent, which in turn can be directly explained, before the early or too late reaction of the ACC system, especially for the recuperation phase [42,43].

The use of a compression ratio above 14:1 in the theoretical analysis may also raise objections, due to the possibility of exceeding the so-called GI in the real engine self-ignition temperature around the TDC point and inducing detonation combustion. However, the actual engine, as shown in the example of the ACC engine, will require significantly higher values of the ignition advance angle, by at least 5 ca to 15 ca compared to the typical values, which in turn, combined with the very fast response of the ACC system, completely prevents this phenomenon from occurring. An additional situation is improved by the very intense flow of the combustible mixture forced by the additional piston from the ACC system, which eliminates the areas favorable for detonation at full load. During the research conducted, it was always possible to “approach the limit of the detonation”, while avoiding it. For this purpose, the “fine-tuning” effect was used, meaning the control of the supply volume and pressure in the ACC system. It should be added that the Vibe functions were selected due to its frequent use in zero-dimensional models, but in engines equipped with mechanical oscillators (ACC motor) it should undergo additional modifications as the intervention of the ACC system increases.

Summarizing, lowering the combustion temperature while maintaining or increasing the overall efficiency of the engine is possible only by increasing the degree of Carnotization of the engine cycle, by using mechanical oscillators whose task is a specific manipulation of additional volumetric work in the cycle [44–46]. This method may in the future lead to the return of the concept of quasi-adiabatic engines, and even engines implementing a cascade of several thermodynamic cycles, increasing the scale of Carnotization. It would be interesting to conduct an analysis at an even more basic level, such as in [14–16].

5. Conclusions

The general conclusion is that obtaining a certain maximum temperature using the combined kinematics of the crank-piston and the complementary system is theoretically completely possible, as shown by the analysis carried out. Unfortunately, the analysis also reveals the following fact, namely that each change in rotational speed requires different kinematics from the complementary system, as shown in Figures 7–9. Thus, it obviously eliminates constant kinematic systems. The conclusion to use the Fourier series to decompose the kinematics of the complementary system into its terms has a specific purpose. This goal was to identify technical devices that could replace it and to reveal physical limitations, e.g., related to the masses and frequencies of these devices. On this basis, it is possible to predefine, for example, geometric dimensions that enable installation in an internal combustion engine. Among other things, for these reasons, the number of only five terms of the Fourier series was used, which was considered adequate to the problem under consideration. The rest of the series can also be optimized by further reducing the number of words, e.g., to three while maintaining a high correlation coefficient, or in extreme cases to one word, which is the case in the ACC engine. The main assumption in theoretical considerations was that the functions describing the combustion processes are continuous, therefore the Vibe function was selected as very good material to modify the theoretical cycle. Although the considerations completely omit the heat transfer, which is very approximate for the ACC engine at the moment, and the influence of the additional system on the Vibe combustion function, it is of no great importance for the analysis itself—when the conditions of the function continuity are met. What is worth emphasizing in the empirical results, the continuity is inherently preserved despite the interference of the ACC system, and this can be seen in the graphs from Figures 14–16. On this basis, it should be assumed that similar results would be obtained by functions taking into account heat transfer or the mentioned alternative kinematic systems. Therefore, the use of mechanical oscillators as equivalents of the terms of the Fourier series makes sense. Based on Figures 14–16, it could be assumed that the real engine operating according to the proposed cycle will always have a lower maximum temperature while maintaining a similar or increased efficiency compared to a typical engine by constant significantly lower compression ratio. This is due to the distribution of the supply of the second portion of heat Q_2 in a more even

manner, allowing the use of a significantly higher compression ratio of the main parameter responsible for efficiency. Taking into account the fact that some of the energy is lost by the complementary system outside the cylinder, you should expect an even lower temperature in the actual engine. The aforementioned ACC engine, presented schematically in Figure 13, is one of the first examples of this type that to a large extent implements the theoretical concept, which clearly confirms the correctness of the assumptions made.

Moreover, in Figures 18 and 19, resulting from the combination of empirical results and simulations, we observed trends in the course of the maximum temperature. In turn, the graph from Figure 20 can be considered as the final proof that the temperature could be kept at a constant level during the second part of the combustion process. On this basis, it can be concluded that the concept of controlling the maximum temperature in the internal combustion engine as proposed, i.e., using one or more mechanical oscillators, is technically feasible. In the case of oscillators, it is important not to forget that the process of their influence consists of two basic phases, i.e., the accumulation phase and the energy recuperation phase, and this in turn must be done with appropriate efficiency. At this point, the authors propose that this criterion should be the value of the efficiency of the theoretical cycle [27], which is implemented by the real engine. The general conclusion that can be drawn from the analysis regarding the temperature is to obtain the maximum temperature of a specific value in real engines. First of all, the place of the change in the volume of the complementary system should be indicated, as it was done in the analysis. Then, determine the minimum possible value by means of an experiment, e.g., following the NO_x emission because, based on the theoretical considerations, it can only be done initially. According to preliminary studies for the ACC engine with 17.1:1 cr, the maximum volume changes constitute about 15% of the swept volume and approximately occur in places coinciding with the theoretical analysis in which the Vibe function was used. This can be seen especially in Figure 15, where the highest compliance is achieved at the PAE feed pressure of 11 bar and 1500 rpm. The graphs with lower pressures, Figure 15, are no longer so consistent, which in turn can be directly explained before the early or too late reaction of the ACC system, especially for the recuperation phase.

Summarizing, lowering the combustion temperature while maintaining or increasing the overall efficiency of the engine is possible only by increasing the degree of Carnotization of the engine cycle, which may occur when we use mechanical oscillators of the first high compression ratio. Several thermodynamic cycles increasing the scale of Carnotization. It would be interesting to carry out an analysis on an even more basic level, such as in [5–7]. Therefore, it can be concluded that the concept of maximum temperature control in the internal combustion engine in the proposed one, i.e., using one or more mechanical oscillators is technically achievable. What is important—the use of even one nonlinear oscillator with pneumatic support gives measurable results, provided that its parameters are properly selected.

6. Patents

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Glogowski, M. U.S. Patent No. 8,720,397 B2, 2014.

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Practical Aspects of Cylinder Deactivation and Reactivation

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Abstract: Cylinder deactivation is an effective measure to reduce the fuel consumption of internal combustion engines. This paper deals with several practical aspects of switching from conventional operation to operation with deactivated cylinders, i.e., gas spring operation with closed intake and exhaust valves. The focus of this paper lies on one particular quantity-controlled stoichiometrically-operated engine where the load is controlled using the valve timing. Nevertheless, the main results are transferable to other engines and engine types, including quality-controlled engines. The first aspect of this paper is an analysis of the transition from fired to gas spring operation, and vice versa, as well as the gas spring operation itself. This is essential for mode changes, such as cylinder deactivation or skip-firing operation. Simulation results show that optimizing the valve timing in the last cycle before deactivating/first cycle after reactivating a cylinder, respectively, is advantageous. We further show that steady-state gas spring operation is reached after approximately 6 s regardless of the initial conditions and the engine speed. The second aspect of this paper experimentally verifies the advantage of optimized valve timings. Furthermore, we show measurements that demonstrate the occurrence of an unavoidable torque ripple, especially when the transition to and from the deactivated cylinder operation is performed too quickly. We also confirm with our experiments that a more gradual mode transition reduces the torque drop.

Keywords: cylinder deactivation; control strategy; engine control; mode switching; experimental analysis; gas spring operation; skip-firing

1. Introduction

It is an inescapable fact that, for the foreseeable future, road and off-road transport will be characterized by a mix of solutions involving internal combustion engines (ICEs), battery, and hybrid powertrains, as well as conventional vehicles powered by IC engines [1,2].

Following governmental regulations for the protection of the environment, a low fuel consumption is an important goal for all car manufacturers. In Europe, from 2020 onwards, a fleet averaged CO₂ limit of 95 g/km will come into effect for a reference vehicle of 1372 kg. By 2030, a further reduction to 59.4 g/km is required. The fuel consumption of vehicles equipped with a combustion engine must, therefore, be reduced significantly.

In addition, the CO₂ emissions of these vehicles can be further reduced when biofuels and synthetic fuels are used. As renewable energy sources, such as solar- or wind-power, are intermittent, researchers have proposed to use excess electric energy produced during periods of low demand to produce synthetic fuels [3]. In Reference [4], a Well-to-Miles analysis is performed to quantify the demand of renewable energy for vehicle operation with different synthetic fuels. Retrofitting gasoline engines to operate with natural gas can lead to a reduction in the CO₂ emissions by 20% [5] and 23% [6], respectively, for passenger cars and a reduction of 26% [7] for a non-road application.

One method to reduce the fuel consumption of spark ignition engines is cylinder deactivation. By using fewer cylinders to supply the same brake load, parasitic losses

are reduced, which increases efficiency. Numerous publications [8–20] about cylinder deactivation describe these effects. Fuel savings by up to 25% for steady-state engine operation [13,14] and by up to 15% in driving cycles [14] or 20% in urban driving [9] are reported.

Cylinder deactivation can improve the operation of Diesel engines, as well. Here, the goal is not only to increase the efficiency at low loads but also to increase the exhaust gas temperature and, thereby, the temperature of the exhaust gas after treatment system, as well as to reduce engine-out emissions [21–27].

The publications cited describe the benefits of cylinder deactivation in great detail. Regarding the mode changes, hardware-specific requirements like timing issues to avoid damage to the system [18,28,29] typically crop up. Other authors mention that changes of the actuation of the throttle, the ignition, and possibly the cam timing [11,30–32] are necessary, or they state that improved software [18,26,33] is required for mode changes. While the authors of Reference [17] provide detailed information regarding the control strategy, these results are only valid for the particular system they use, and the control strategy is also only provided schematically with no experimental verification.

In Reference [12], a control strategy is presented with an experimental verification; however, the same ignition angle appears to be used for both cylinder groups during the mode transition. The authors do not provide sufficient detail about the two engine revolutions during which the actual mode transition occurs, but this is crucial, as shown in Reference [34].

Other authors [31,35] recognize a torque ripple during the mode changes and suggest opening the torque converter lock-up clutch as a countermeasure. However, they do not address the cause of the torque ripple.

In a previous paper, the present authors introduced a control strategy to influence the torque ripple directly [34]. The properties during the transition to and from cylinder deactivation were analyzed with regard to cycle-averaged torque and fuel consumption. It was shown that the drawback of an increased fuel consumption during the transitions can be neglected when a variable valve-train is used. The analyses also showed that the specific increase/decrease in each cylinder group has a significant impact on the cycle-averaged engine torque. It must be noted that the key point of the control strategy presented in Reference [34] is the selection of beneficial reference values for the load of the individual cylinders. Tracking these reference values will lead to a smooth transition regardless of the particular controllers that are used on the engine control unit (ECU).

This paper can be seen as a follow-up to Reference [34]. We provide insight into the gas spring operation of the unfired cylinders and show that regardless of the initial conditions, steady-state operation is reached after only 10 engine cycles for the considered spark ignition engine. Then, we present different possibilities for the transition from fired operation to gas spring operation and vice versa. The advantages that result from using optimized valve timings made possible by a fully variable valve-train are quantified. In the experimental analysis part of the paper, we provide experimental verification for the control strategy presented in Reference [34]. In that study, it was stated that using a particular control strategy which limits the torque gradients of each cylinder group during deactivation and reactivation, respectively, will reduce a torque ripple during the transition phase. In this study, we show experimentally that whether the mode transition is done quickly or gradually has, in fact, a significant impact on the torque drop during deactivation and reactivation processes, respectively. It must be noted that this study does not assess problems associated to noise, vibration, and harshness (NVH) as this is very specific to drivetrain, application, and vehicle. This study shows how the transition should be done whenever it is allowed.

For the simulation part of the study, we use two types of engine simulation models for the analyses presented here. They employ different levels of detail, and, while one is more detailed and, hence, only used for offline analysis of the fundamental effects, the other is strongly simplified and suitable for implementation on the ECU. For the

experimental part and also to parameterize the simulation models, a passenger car engine equipped with the fully variable FlexWork valve-train [36] and operated with natural gas is used.

This paper is structured as follows: First, we present the experimental setup that was used for the measurements shown in this publication. Next, we present the simulation models. Then, we analyze fundamental properties of the engine operation in the deactivated state, i.e., gas spring operation. Thereafter, we present simulation results of different strategies for the deactivation and reactivation of a cylinder, respectively. Finally, we provide experimental verification.

2. Experimental Setup

Experiments were performed on a Volkswagen engine with four cylinders and a displacement of 1.4 L. The engine operated on port-fuel-injected natural gas and was controlled to stoichiometric conditions. The complete camshaft system, composed of the camshafts, gears, chain, and timing adjustment, was removed and the engine was fitted with the electro-hydraulic FlexWork valve-train [36], which is shown in the photograph in the top of Figure 1. This valve-train has one actuator, a magnetic coil, for each pair of intake and exhaust valves, respectively. Figure 1 also shows a schematic of the engine test bench, and Table 1 gives an overview of the experimental setup.

Table 1. Main characteristics of the experimental setup.

Engine basis	Volkswagen 1.4 L, production code EA111
Bore/stroke/number of cylinders	76.5 mm/75.6 mm/4
Compression Ratio	10
Fuelling	Port fuel injection of natural gas, Bosch NGI2 injectors
FlexWork valve-train	Self-developed, electro-hydraulic valve-train for intake and exhaust
Valve lift measurement	Linear encoders (Elgo Electronic), using custom-developed converters
Valve lift variability intake	All cylinders 1–9 mm
Valve lift variability exhaust	All cylinders 1–9 mm
Valve timing variability intake	Cylinder-individual
Valve timing variability exhaust	Cylinder-individual
Engine control	In-house developed, rapid-prototyping system (dSPACE)
In-cylinder pressure transducers	M10 piezoelectric, water-cooled (Kistler 6061B) in all cylinders
Stoichiometry sensors	Wide band (Bosch LSU 4.9)
Fuel flow measurement	Coriolis sensor (Rheonik RHM015)
Combustion air flow measurement	ABB Sensyflow P
Torque measurement	HBM T40 torque transducer
Flywheel/gearbox	Standard dual mass/no gearbox
Elastic shaft/joints	GKN Löbro type 111/CV15
Coupling	Voith type K

The working principle of the valve-train is basically a hydraulically excited spring-mass-system [37] which is able to recover energy during the valve closing phase. Detailed description of the valve train can be found in Reference [38,39], and the results of the engine operation in Reference [36,38–40]. For the experiments conducted for this research, only three main properties of the valve-train are relevant:

- The valve lift for the intake and exhaust valves, respectively, can be set for all cylinders.
- Both opening and closing timings can be set individually for each cylinder at the intake and exhaust side, respectively.
- The valves are naturally closed, which means that each pair of intake and exhaust valves can be deactivated individually.

These properties allow an investigation of the deactivation and reactivation process of a cylinder without the typical constraints cam-based systems have. These constraints comprise the inability to change the valve timing of two cylinder groups individually and—generally speaking—the lack of variability in the exhaust valve opening duration.

The engine control software allows variation of valve timings, the mass of fuel injected and the ignition timing for each cylinder-individually from one actuation to the next. This makes a fast and cylinder-individual load control possible. Thus, the new values are chosen for each cylinder individually in each cycle according to the load that is required in that instant. The condition for the deactivation of a cylinder simply is if the desired mean effective pressure is smaller than 0.5 bar. In order to realize a deactivation, the corresponding “enable” signals are set to zero right after the last actuation of the fuel injection, intake valve and exhaust valve, respectively, of a particular cylinder.

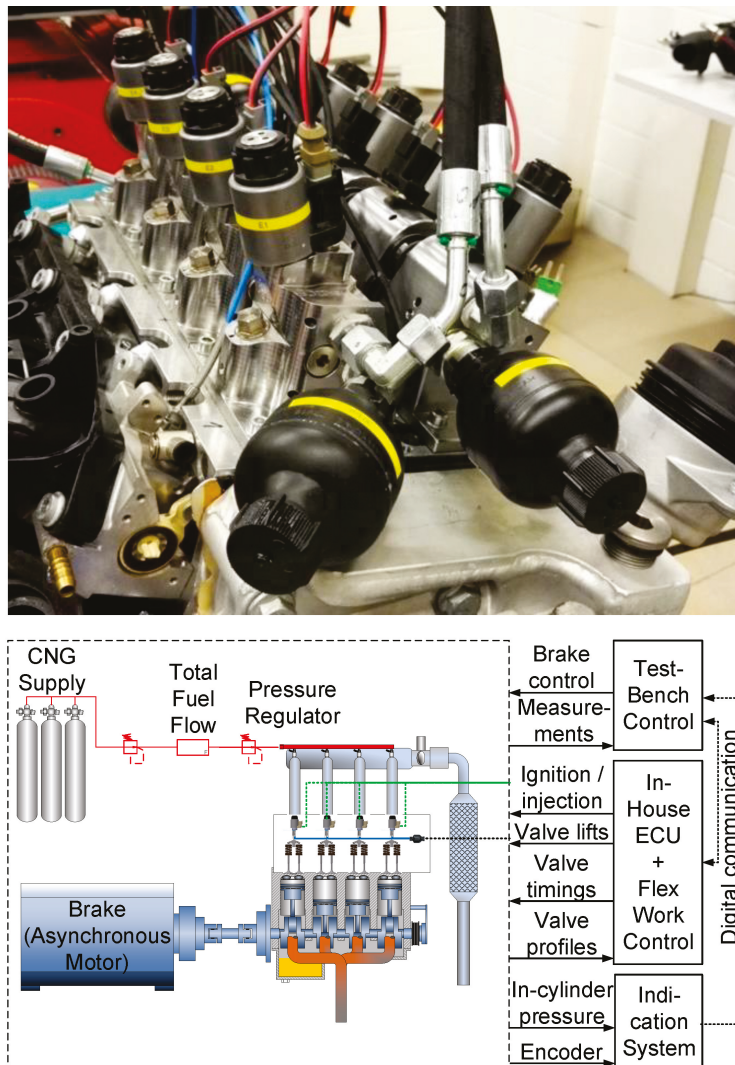


Figure 1. Photograph and schematic of the experimental setup. “Measurements” comprises pressures and temperatures at various locations, as well as volumetric flow rates of the hydraulic valve actuators, and the air and fuel for combustion.

To motivate for the benefits of cylinder deactivation, some information on the brake efficiency is provided in Table 2 for low load operating points at an engine speed of 2000 rpm.

Table 2. Brake efficiency at 2000 rpm in 4-cylinder and 2-cylinder operation.

p_{me} (bar)		1.0	1.5	2.0	2.5	3.0	3.5
brake efficiency (%)	4-cylinder operation	15.00	20.51	24.14	26.19	27.81	28.96
	2-cylinder operation	20.31	24.43	27.18	28.85	30.00	31.05
	relative improvement	35.46	19.12	12.56	10.15	7.87	7.20

No sensors to assess problems associated to NVH are installed at the testbench as this is very specific to drivetrain, application, and vehicle. However, the authors can state that there was no audible noise during the transition, and the sound of the engine was soft in full-cylinder operation, as well as in two-cylinder operation, with an audible difference between them. A visual inspection indicated that there were no fouled spark plugs after the measurements in two-cylinder operation and deactivation/reactivation measurements. However, the engine was only operated with two cylinders for rather short periods of time (several minutes) for the required measurements. Whether extended two-cylinder operation can lead to the occurrence of such problems can only be determined in long-term studies, which is not the scope of our research.

3. Simulation Models

Two different types of engine simulation models are used in this work. They employ different levels of detail, and, while one is more detailed and, hence, only used for offline analysis of the fundamental effects, the other is strongly simplified. This model is invertible and allows the ECU to calculate feed-forward control trajectories.

The process simulation is a 0-dimensional engine simulation tool. The intake manifold pressure, the exhaust manifold pressure and the pressure inside the crank case, respectively, are constant, while the in-cylinder pressure can change. The mass flow over intake valves, exhaust valves and piston rings is modeled as compressible flow. There are three gas species in the model—air, unburned mixture, and burned gas—which have temperature dependent gas properties. The combustion is modeled as a Vibe type heat release rate [41] and wall heat losses are implemented using the Woschni correlation [42]. The parameters for the valve flow coefficients, heat release rate, wall heat losses, and blow-by properties are identified using measurements obtained at the test bench. To identify the blow-by parameters, at least two measurements are necessary: One measurement at conditions where no reverse blow-by occurs, e.g., supercharged operation, and one measurement in the gas spring operation where reverse blow-by has a major influence on the in-cylinder mass and pressure level. The root mean square error of the indicated efficiency at a given indicated mean effective pressure is as low as 0.18%, and the root mean square error of the mean indicated pressure at given valve timings is 0.136 bar in the relevant operating region. This model will be used for the analysis of the gas spring operation and the transition from and to fired operation, respectively.

The simple engine model is a discrete event model based on the Willans approximation presented in Reference [41]. For one engine speed, the mean effective pressure p_{me} is modeled as:

$$p_{me} = e(p_{me}) \cdot p_{m\varphi} - p_{me0,g}(p_{me}) - p_{me0,f}(p_{me}) \quad (1)$$

$$p_{m\varphi} = \frac{H_l \cdot m_\varphi}{V_d}, \quad (2)$$

where e is the Willans efficiency, i.e., the efficiency of the conversion of chemical energy in the fuel to indicated work of the high pressure part of the engine cycle, $p_{me0,g}$ represents the pumping losses, and $p_{me0,f}$ represents the friction losses. The mean effective fuel power $p_{m\varphi}$ is defined by the fuel's lower heating value H_l , the mass of fuel injected per cycle m_φ , and the engine displacement volume V_d .

The in-cylinder pressure measurements and the torque measurement were used to evaluate the summands of Equation (1) as

$$e = \frac{p_{mi,hp}}{p_{m\varphi}} \quad (3)$$

$$p_{me0,g} = -p_{mi,lp} \quad (4)$$

$$p_{me0,f} = p_{mi} - p_{me}, \quad (5)$$

where p_{mi} is the indicated mean effective pressure over the entire engine cycle, $p_{mi,lp}$ is the indicated mean effective pressure of the low-pressure loop, i.e., the gas exchange, and $p_{mi,hp}$ is the indicated mean effective pressure of the high-pressure loop, i.e., the compression and combustion strokes. The engine model shown in Figure 2 is based on steady-state load variations measured at the test bench which were carried out at an engine speed of 2000 rpm. The valve lift is kept constant at 3 mm for both intake and exhaust valves, and the throttle is fully open for all measurements. Using a higher valve lift would result in a slight increase of the achievable load and a slight reduction of throttling losses at the expense of combustion stability at very low loads.

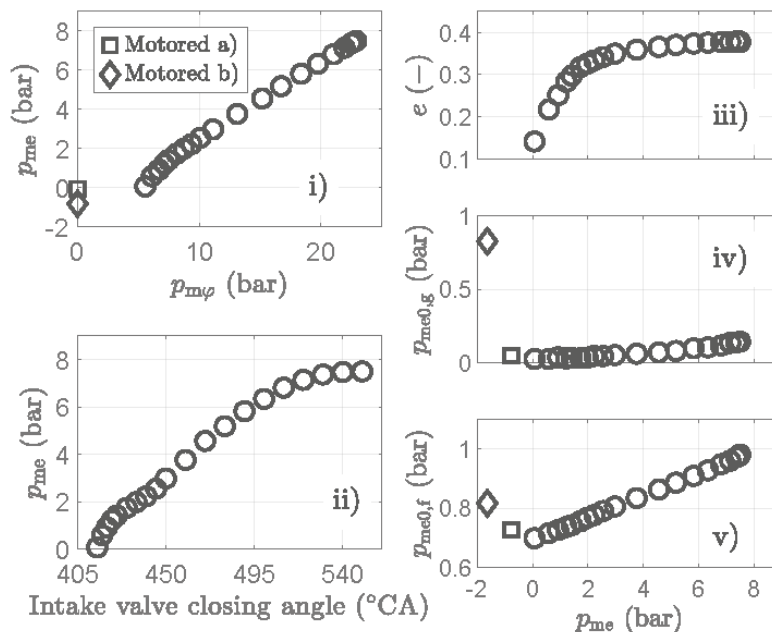


Figure 2. Willans-based Discrete Event Model. The circles depict measurements in fired operation. Motored (a) refers to motored operation with all valves closed (gas spring operation) and motored (b) refers to motored operation with only the intake valves closed. Subfigures (i,ii) show the mean effective pressure over the fuel mean effective pressure and the intake valve closing angle, respectively. Subfigures (iii–v) show the Willans efficiency, the gas exchange losses, and the friction losses, respectively, as a function of the brake mean effective pressure.

The load is controlled by changing the valve timings, mainly the intake valve closing angle. At low loads, the intake valve closes as early as 55 °CA after gas exchange top dead center. When the load reduces from 7 to 2 bar p_{me} , the combustion efficiency e also slightly reduces, which is due to a decrease of turbulence. However, this effect is overcompensated by the benefit of lower pumping losses when compared to throttled operation, as was analyzed in Reference [40] for this exact same engine and valve-train. At loads lower than $p_{me} = 1.8$ bar, the center of combustion had to be delayed to avoid high cycle-to-cycle variations of the mean indicated pressure.

Since the valve timings, the fuel injection and the ignition angle can all be changed from one engine cycle to the next for each cylinder and the intake manifold pressure remains constant at all loads, no further level of detail is required for this model besides the respective delays which are summarized in Table 3. The measured curves are implemented directly in the ECU. By means of linear interpolation, we obtain the respective valve timings, injection duration, and ignition angle for the desired load of each cylinder.

Table 3. Delays of the discrete event model.

Delay	Segments
injection-to-power-stroke	2
ignition-to-power-stroke	1
power-stroke-to-exhaust	1

4. Gas Spring Operation in a Deactivated Cylinder

Before we analyze the transitions to and from the deactivated cylinder operation, we address the properties of a deactivated cylinder itself. Simulations are carried out using the process simulation model. All simulations are over 1000 engine cycles at 2000 rpm and start at top dead center. The cylinder only contains burned gas at a pressure of 1.1 bar, which is chosen to represent the in-cylinder pressure during the exhaust stroke. Both intake and exhaust valves are closed from the beginning, and there is no heat release from combustion. While the given engine has a compression ratio of 10, compression ratios of 13 and 16 were also analyzed for comparison.

In the upper and middle plots of Figure 3, the cycle-averaged in-cylinder mass m_{cyl} is shown. The upper plot shows that, even if the identified blow-by flow cross sections are doubled or halved, the in-cylinder mass converges to the same value. The smaller the flow cross section, the slower the process. However, after a maximum of 250 engine cycles (which corresponds to 15 s), the steady-state is reached. However, the variation of the in-cylinder mass over one engine revolution increased, the larger the flow cross section is. This is shown in the middle right plot.

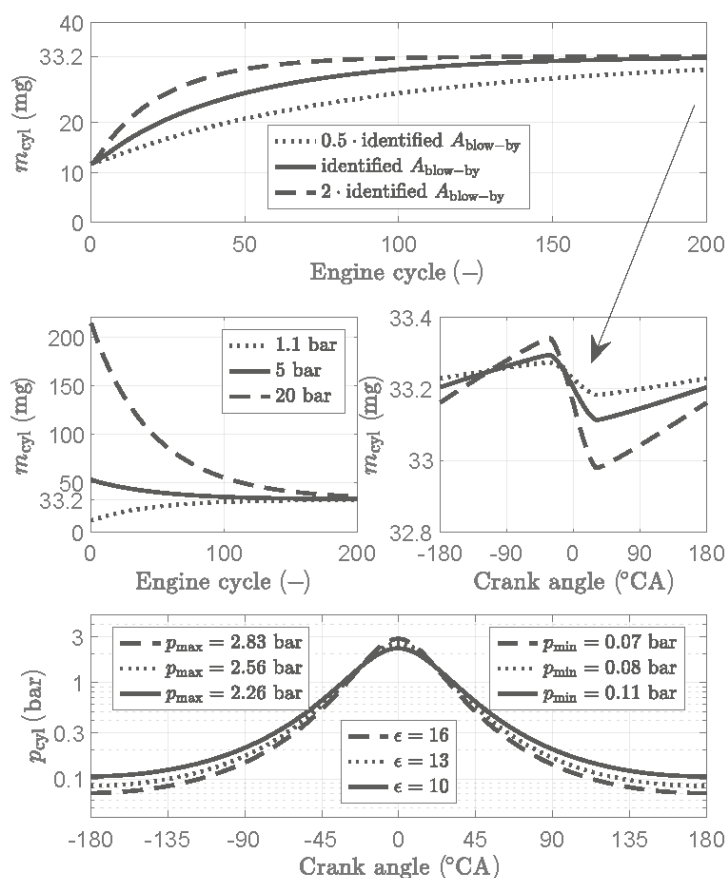


Figure 3. Gas spring operation with different model parameters. Upper plot and middle right plot: different blow by properties (double and half of the identified parameters). Middle left plot: different initial in-cylinder pressures. Bottom plot: different compression ratios.

The middle left plot shows that, for different initial pressures, the in-cylinder mass also converges quickly to the same steady-state value. The p_{mi} of the gas spring operation is -0.042 bar in steady-state. Except for the smallest blow-by flow cross section, the simulations converge after approximately 100 engine cycles, which corresponds to 6 s. The gas spring p_{mi} value, however, stabilizes within 5% of its final value in less than 10 engine cycles.

Finally, the bottom plot of Figure 3 shows the cylinder pressure trace in steady-state conditions for different compression ratios. Increasing the compression ratio increases the pressure amplitude. The peak pressure at top dead center increases, while the minimal pressure at bottom dead center reduces simultaneously. The peak pressure for the given compression ratio of 10 is only approximately 2.3 bar, while the minimum pressure is slightly above 0.1 bar.

An additional set of simulations at 4000 rpm reveals that it takes approximately the same amount of time and, hence, twice as many engine cycles until the steady state is reached. The minimum and maximum pressure levels, as well as the cycle-averaged in-cylinder mass, remain largely unchanged. However, the p_{mi} of the gas spring operation reduces to -0.037 bar in steady-state due to the smaller fluctuation of mass over one engine revolution.

These analyses show that, after only a short period of time—approximately 6 s—the steady-state gas spring operation is reached, regardless of the initial conditions and the engine speed. This corresponds to less than one second of operation in the deactivated state. In addition, these simulations reveal the in-cylinder pressure level which is relevant for the deactivation and reactivation process of a cylinder.

5. Analysis of the Deactivation and Reactivation Process of a Cylinder

Regardless of the engine configuration, the number of cylinders, the type of fuel, the strategy used, or the load of the last combustion cycle: whenever any individual cylinder is deactivated, it must undergo a transition from fired operation to gas spring operation. During the reactivation, the inverse holds true and is again independent from the aforementioned criteria. Several basic options are available to influence the deactivation and reactivation process, respectively. In order to analyze these transitions in a general way, this section analyzes the deactivation and the reactivation using the process simulation. These results will be validated in the following section using experimental data.

5.1. p_{mi} Calculation

Since the deactivation of a cylinder can be seen as a very fast transient process in which significant changes occur from one engine cycle to the next it becomes relevant how the p_{mi} is calculated. On the contrary, in steady-state operation, it is not relevant in which stroke the p_{mi} calculation starts. In that case, the resulting average p_{mi} over several engine cycles is always the same. As in Reference [34], the p_{mi} calculation is started with the intake event (intake-compression-combustion-exhaust). Since, during the deactivating/reactivating engine cycle, some particular deviations from conventional pV-diagrams occur, it is important to keep this in mind.

It must also be mentioned that, regardless of how the p_{mi} is calculated, there is only one pressure in the combustion chamber that acts on the piston; therefore, the way in which the p_{mi} is calculated cannot change the result.

5.2. The Deactivating Cycle

There are two fundamentally different approaches, when it comes to the deactivation of a cylinder: either exhaust gas or air can be trapped inside the cylinder before deactivation.

This choice will lead to different possibilities in the operating strategy upon reactivation. In case air is trapped inside the cylinder, a combustion can occur before the reactivation of the valves. For a direct injected engine, the fuel injection may cause enough turbulence and entrainment for a combustion to take place. If the engine is port injected,

an air-fuel mixture has to be trapped inside the cylinder for this option which would, however, be difficult to ignite due to the resulting lack of charge motion.

Since the engine used for this research is only operated with port injected natural gas, the option to trap air or mixture is not considered. Therefore, simulations are made of three different methods for deactivating a cylinder that all trap exhaust gas inside the cylinder:

- Do not expel the burned gas of the last combustion and, thus, trap exhaust gas at high pressure.
- Conduct the last exhaust stroke and deactivate the exhaust valves thereafter. Thus, trap exhaust gas at low pressure.
- Close the exhaust valve during the last exhaust stroke and hence only partially expel the burned gas. Thus, trap exhaust gas at medium pressure.

Method (c) is only possible with sufficient variability in the valve-train. To compare these transition methods, several engine cycles must be taken into account because, as shown in Section 4, it takes several engine cycles until the steady-state gas-spring operation is reached. The value Σp_{mi} is a sum of mean indicated pressures which is used to quantify each method. It is comprised of the p_{mi} of the last conventional combustion cycle, the cycle in which the cylinder is deactivated, and the following 138 engine cycles.

An optimization was carried out to find the best valve timing for method (c). An exhaust valve closing angle of 25 °CA before top dead center directly initializes the gas spring operation at the desired pressure level for this engine, since, at this crank angle, the in-cylinder pressure is approximately at the value it will have in steady-state gas spring operation. This valve timing depends on the blow-by properties of the specific engine and may vary for different engines. However, this effect is largely independent of the load of the last combustion cycle, as the measure only affects the last exhaust stroke and the gas spring operation.

The upper three plots of Figure 4 show the valve lift curves and the cylinder pressure over several engine cycles after deactivation for each deactivation method. The engine cycle which spans from 0 to 1 in each simulation is referred to as “the deactivating cycle”. The bottom two plots show the p_{mi} trace of the first deactivated cycles and a double logarithmic pV-diagram of the deactivating cycle of each method, respectively.

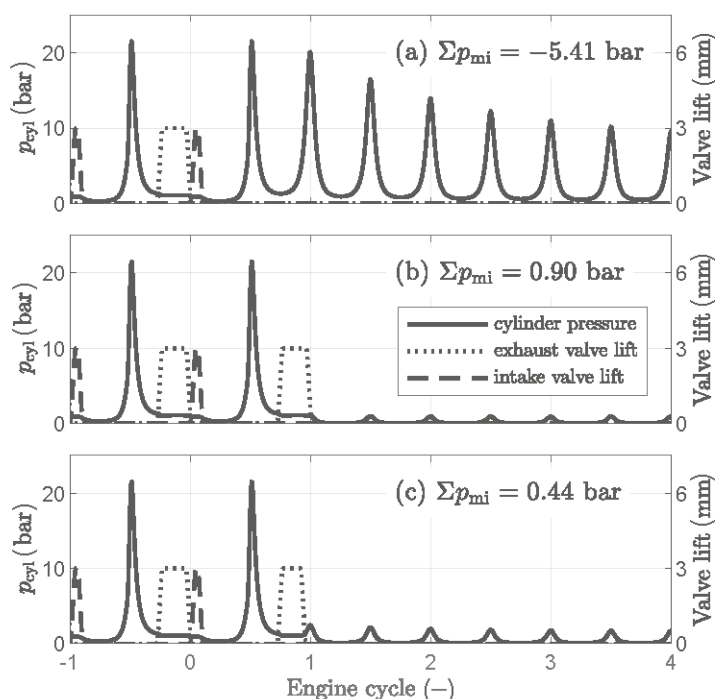


Figure 4. Cont.

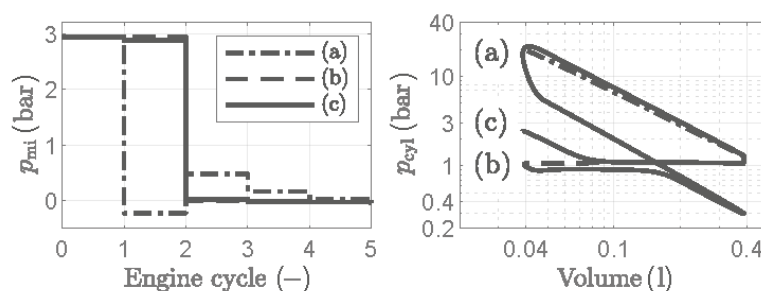


Figure 4. Upper three plots: valve lift curves and cylinder pressure of different cylinder deactivation strategies. Bottom two plots: p_{mi} traces and a double logarithmic pV-diagram of the deactivating cycle.

The main difference between these methods is that they result in a different in-cylinder pressure after the deactivating cycle, which can be seen in the bottom right plot. Comparing the p_{mi} traces of all methods, the bottom left plot shows that, while methods (b) and (c) are rather similar, method (a) differs strongly. The negative p_{mi} in the deactivating cycle for method (a) results from the high pressure during the exhaust stroke with closed exhaust valves. The positive p_{mi} values directly thereafter result from the fact that the p_{mi} calculation starts at top dead center, and the cylinder content loses mass until steady-state is reached. The Σp_{mi} value for this deactivation method is the lowest of the three. Comparing methods (b) and (c), one difference is that a slightly lower p_{mi} value results for method (c) in the deactivating cycle. The reason for this difference is that, late in the expansion stroke, when the exhaust valve is closed, the pressure increases. This can be seen in the pV-diagram and is counted as a negatively enclosed area. The following p_{mi} values in the gas-spring operation until steady-state is reached are very similar but slightly lower for method (c). The Σp_{mi} value is 0.46 bar lower. Therefore, a trade-off exists between increasing the initial losses and avoiding an underpressure in the initial phase of the gas spring operation.

5.3. The Reactivating Cycle

When exhaust gas is trapped inside the deactivated cylinder, there are two fundamentally different approaches, when it comes to reactivating a cylinder: either the intake valves or the exhaust valves can be opened first.

In case the exhaust valves are opened first, the cylinder content is scavenged to the exhaust before opening the intake valves. The other option is to open the intake valve first. In that case, the burned gas fraction is higher for the first combustion as the in-cylinder pressure at the corresponding top dead center is approximately 2 bar. Both intake valve opening and closing angle need to be adjusted for this option.

Simulations are made of three different methods for reactivating a cylinder:

- (a) Reactivate the exhaust valves first using conventional valve timing.
- (b) Reactivate the exhaust valves first using a very early exhaust valve opening angle.
- (c) Reactivate the intake valves first using a modified intake valve opening and closing angle.

Options (b) and (c) are only possible with sufficient variability in the valve-train. An optimization was carried out to find the best timing for the opening of the exhaust valve for method (b). An exhaust valve opening angle of 20 °CA after top dead center maximizes the p_{mi} of the reactivating cycle for this engine, since, at this crank angle, the in-cylinder pressure is approximately equal to the exhaust manifold pressure. This angle depends on the blow-by properties of the specific engine and may vary for different engines. However, this effect is largely independent of the load of the first combustion cycle, as the measure only affects the last expansion and exhaust stroke of the gas spring operation. For method (c), both intake valve timings were optimized. An opening angle of 20 °CA after top dead center minimizes the losses, while an adjusted closing angle leads to the desired p_{mi} . In this case, the last gas spring cycle is not modified, and both valve timings affect the first combustion cycle, as the p_{mi} calculation starts with the intake

stroke. In cases when the load of the first combustion is very low, this method may run into the problem that the required opening duration of the intake valve may become shorter than possible with the hardware. In such cases, a later intake valve closing angle must be accepted and counterbalanced by ignition retardation.

The upper three plots of Figure 5 show the valve lift curves and the cylinder pressure over several cycles after reactivation for the three different reactivation methods. The engine cycle that spans from 0 to 1 for methods (a) and (b) is referred to as “the reactivating cycle”. Method (c) does not have a separate reactivating cycle, as the modification of the first intake valve opening can be seen in the first cycle with combustion. The bottom two plots show the p_{mi} trace of the first reactivated cycles and a double logarithmic pV-diagram of the reactivating cycle of each method, respectively.

Comparing the p_{mi} traces of these methods shows that the only visible difference occurs during the reactivating cycle. Unlike for the deactivating process, the reactivating cycle itself is sufficient to quantify the differences between these methods.

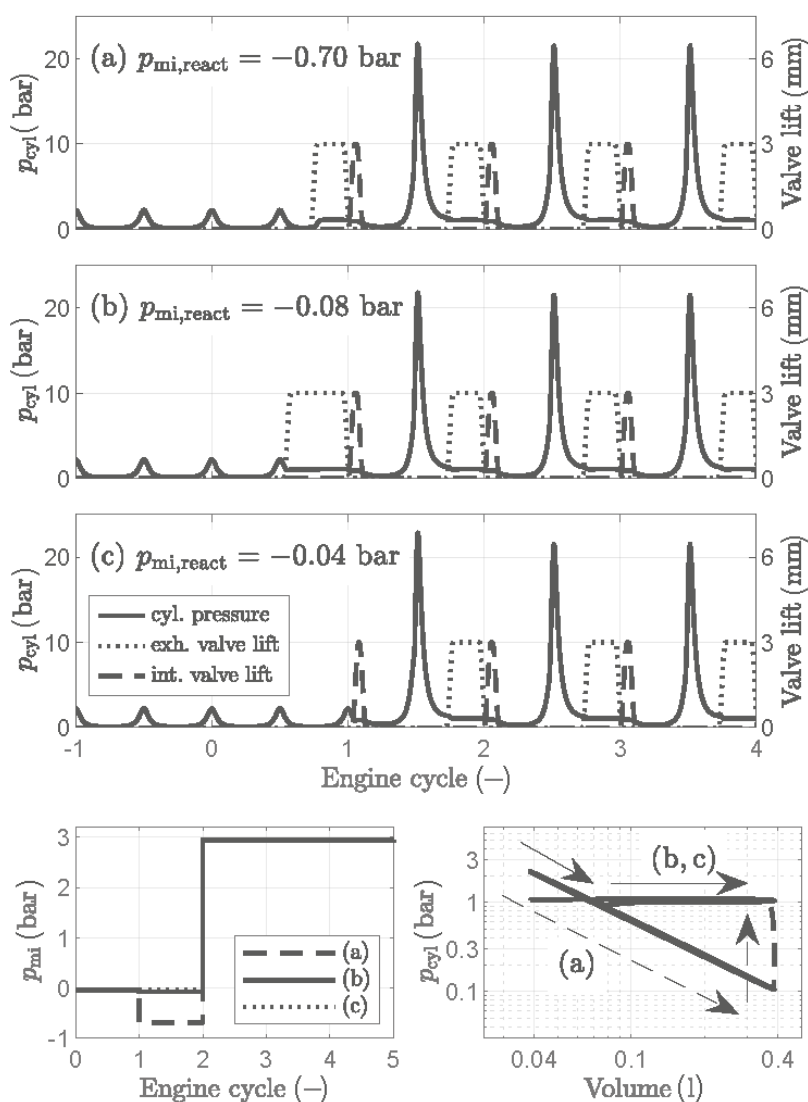


Figure 5. Upper three plots: valve lift curves and cylinder pressure of different cylinder reactivation strategies. Bottom two plots: p_{mi} traces and a double logarithmic pV-diagram of the deactivating cycle.

In method (a), the gas spring operation ends in bottom dead center, and the in-cylinder pressure increases to the level of the exhaust manifold pressure as soon as the exhaust valves open. In contrast, for method (b), the gas spring operation ends early, during the downward stroke; therefore, the negatively encountered area in the

pV-diagram is much smaller. For method (c), the gas spring operation also ends early, during the downward motion of the piston, but one revolution later compared to method (b), i.e., during the intake stroke. This is also the reason that methods (b) and (c) appear similar in the pV-diagram. These results clearly show that methods (b) and (c) are superior to method (a) since the losses are much lower. The difference between methods (b) and (c) results from the fact that, in method (b), there is essentially one entire engine revolution where the exhaust valves are open. Since exhaust gas is being aspirated into the cylinder and then pushed out into the exhaust manifold again, this method shows slightly higher losses.

The process simulation model does not take into account a possible deterioration of the combustion due to increased internal burned gas fraction or lack of turbulence. Therefore, the practicability of the proposed methods will be put to the test during the experimental part of the study in Section 6.

6. Experimental Results

In this section, we experimentally verify the different strategies for the deactivating and reactivating cycle, respectively. We also show that a torque drop occurs, when the transition to and from deactivated cylinder operation is done too quickly and that this torque drop can be reduced when the transition is done more gradually.

6.1. Cylinder Deactivation

Since the fully variable FlexWork valve-train allows for changes in valve timings of both intake and exhaust from one actuation to the next, the two most promising of the previously discussed cylinder deactivation methods are presented. Thereafter, a so-called “instantaneous” cylinder deactivation, which represents the fastest way to change to two-cylinder operation, is analyzed followed by measurements of more gradual transitions for comparison. This was proposed in Reference [34], and, using this strategy, one cylinder group increases the load linearly, and the other cylinder group decreases the load linearly.

For all pV-diagrams shown in this paper (fired operation and gas-spring operation), the offset correction of the in-cylinder pressure is done during the compression stroke when all valves are closed and before ignition by comparing the measured pressure trace to that of a polytropic compression [43]. For the gas spring operation, this is the only option as all valves are closed.

The left plot of Figure 6 shows the steady-state gas spring operation of 20 engine cycles. The reason that the lines diverge near the bottom dead center is the double logarithmic representation of the pV-diagram. The deviations at bottom dead center are in fact not larger from those at top dead center in the same plot. The average p_{mi} of the steady-state gas spring operation of all four cylinders is -0.051 bar.

The middle and right plot of Figure 6 show the deactivating cycle and several gas spring cycles after the deactivation when conventional and optimized valve timing (partial exhaust stroke) is used, respectively. The measurements clearly show that, in the middle plot, the pressure during the gas spring operation increases over time. The first gas spring cycle starts at top dead center with the in-cylinder pressure similar to that of the exhaust manifold after the exhaust stroke, i.e., slightly above 1 bar. Then, the pressure level increases to the steady-state value shown in the left plot due to the increasing in-cylinder mass which results from inverse blow-by. In the right plot, this is not the case, and the steady-state gas spring operation is reached almost instantly. Despite the fact that the difference between the two methods can clearly be seen in the pV-diagram, it was not possible to confirm the difference in the Σp_{mi} value presented in Section 5.2. The reason is that, due to the stochastic nature of the last combustion before deactivation and the measurement uncertainty (which is rather large at this low pressure level), such small differences vanish.

The additional advantage for the optimized valve timing is that the in-cylinder pressure is higher during the transition to steady-state gas spring operation, which is

helpful in order to reduce the aspiration of engine oil into the cylinder which occurs when the pressure below the first piston ring is higher than the in-cylinder pressure [44]. Therefore, the optimized exhaust valve timing for the deactivating cycle is used in the following cylinder deactivation measurements.

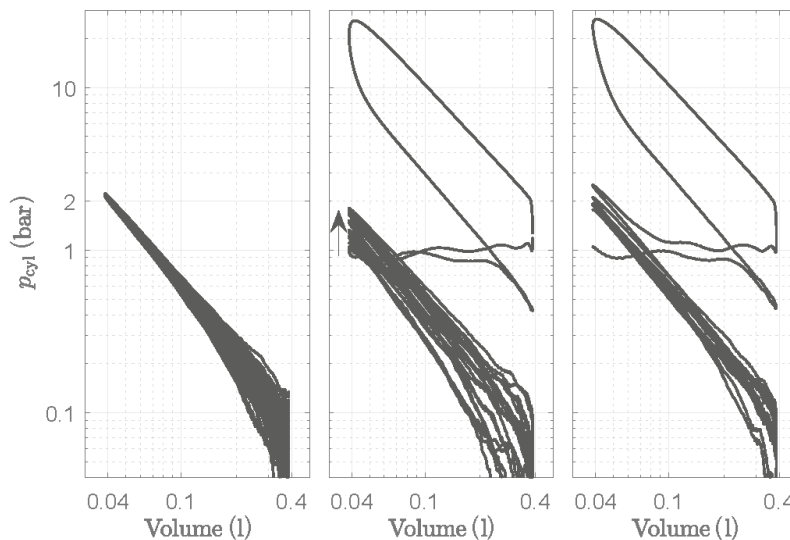


Figure 6. Measurements of the gas spring operation (**left** plot) and 2 methods for cylinder deactivation (**middle** and **right** plot). The middle plot shows the pressure trace of the last cycle with combustion and several engine cycles thereafter when the exhaust valves are deactivated after a full exhaust stroke. The right plot shows the same number of engine cycles when a partial exhaust stroke is performed in the deactivating cycle.

Figure 7 shows the result of an instantaneous cylinder deactivation. The upper plot shows the intake valve closing angle for each cylinder group. This value represents the desired engine load, as shown in Figure 2. The middle plot shows the measured p_{mi} values that resulted from the combustion cycle that used the intake valve closing angle shown in the upper plot in the same segment. For deactivated cylinders, no intake valve closing angle is shown, while the p_{mi} measurement of the gas spring cycle is shown. The bottom plot presents the cycle-averaged p_{mi} , which is the mean value of the current segment p_{mi} and the last three p_{mi} values. On the right y-axes in the bottom plot, the torque measurement is shown. This measurement is done with a frequency of 50 Hz and is, therefore, not segment-synchronous.

Figure 7 clearly shows in the upper and middle plot that the load changes from one cycle to the next for each cylinder group. The p_{mi} of the firing cylinder group directly changes to the targeted value, while the p_{mi} of the deactivated cylinder group directly changes to the gas spring operation. The cycle-averaged p_{mi} features two distinct drops that indicate a lack of torque. The torque measurement confirms that the drops in the p_{mi} are also seen at the gearbox input in the form of a significant torque drop (35% lower torque).

Figure 8 shows the result of a more gradual cylinder deactivation that used a linearly increasing/decreasing torque ramp for the desired load in the two cylinder groups, respectively. This results in two intermediate combustion events for the firing cylinder group and a single intermediate combustion event for the cylinder group that will be deactivated. However, the reduction of the cycle-averaged p_{mi} is clearly visible compared to the instantaneous deactivation and there is almost no drop in the torque measurement (only 8% lower torque). These experimental results clearly show that the amplitude of the deviation of the cycle-averaged p_{mi} and the torque drop can be reduced significantly by the strategy presented in Reference [34]. Despite the fact that no model of the drivetrain

and the dual mass flywheel were used, these results show that, when just three combustion events are placed between the four-cylinder and two-cylinder operation, the fluctuations of the cycle-averaged p_{mi} are small enough so that no torque oscillation of the dual mass flywheel results.

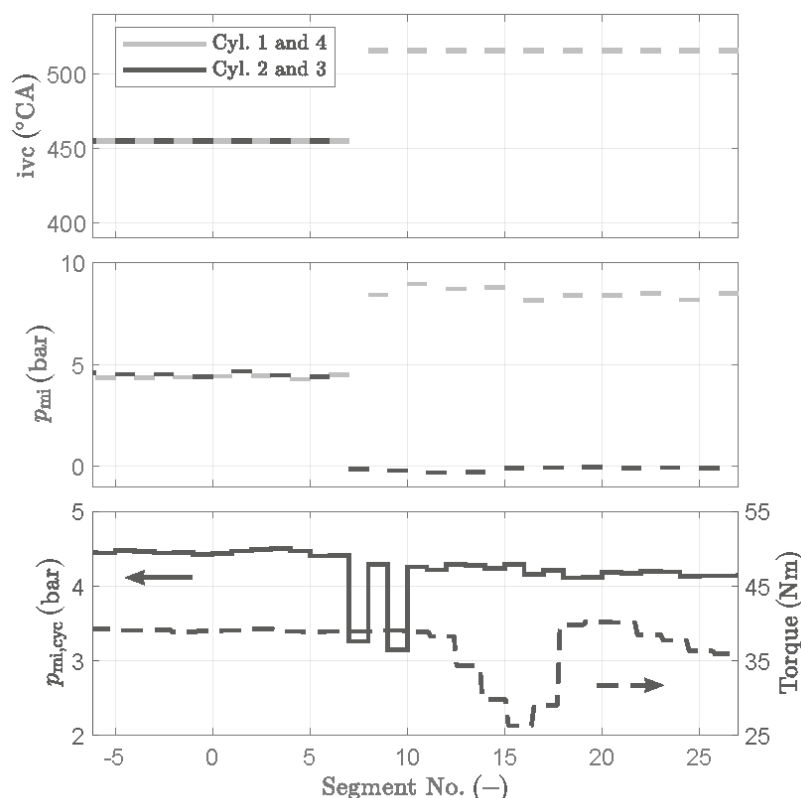


Figure 7. Measurement of an “instantaneous” cylinder deactivation. Top to bottom: intake valve closing angle (which represents the desired load), measured p_{mi} , cycle-average p_{mi} , and measured torque. For deactivated cylinders, no intake valve closing angle is shown, while the p_{mi} measurement of the gas spring cycle is shown. The term “instantaneous” is used here because this represents the fastest way to change into two-cylinder operation.

It must be noted that, due to the stochastic nature of the combustion process and especially in fast transient load changes as required here, it is not possible to place the three combustion events such that exactly the desired p_{mi} results. However, a clear reduction in the cycle-averaged p_{mi} and the torque drop could always be observed.

In order to test whether a smaller gradient of the torque ramps leads to an even smaller torque drop, additional measurements were made. The torque ramps are such that each cylinder increases/decreases the load by only 0.2 bar p_{me} from one actuation to the next. The resulting transition from four-cylinder operation to two-cylinder operation takes 1.1 s, and the result is shown in Figure 9. The results clearly show that, during this slower transition, even smaller deviations in the torque can be reached.

The trade-off between a small torque drop and a short mode transition time that was identified in Reference [34] is confirmed by these results. However, the experimental results also show that even very short mode transitions result in a small torque drop as long as an instantaneous deactivation is avoided.

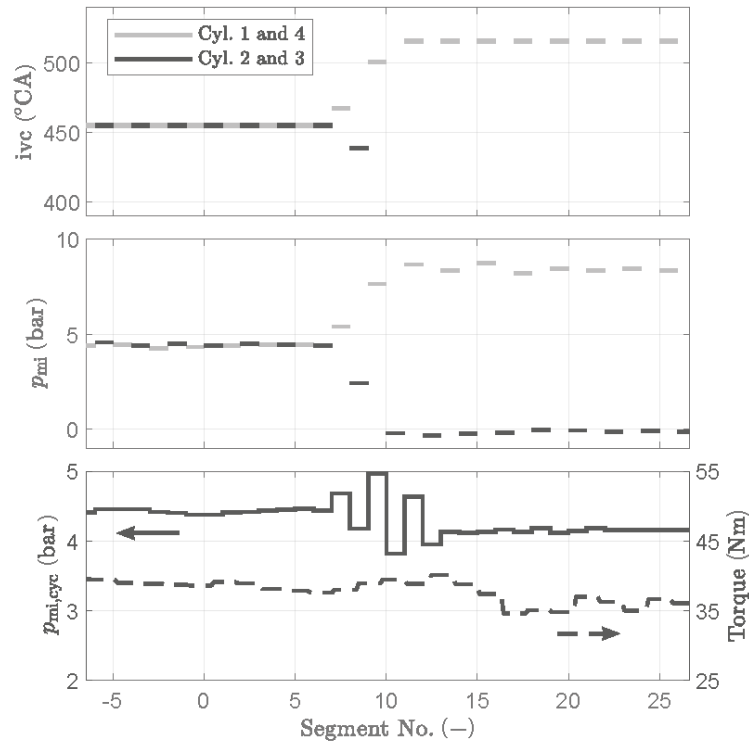


Figure 8. Measurement of a more gradual cylinder deactivation. Top to bottom: intake valve closing angle (which represents the desired load), measured p_{mi} , cycle-average p_{mi} , and measured torque. For deactivated cylinders, no intake valve closing angle is shown, while the p_{mi} measurement of the gas spring cycle is shown.

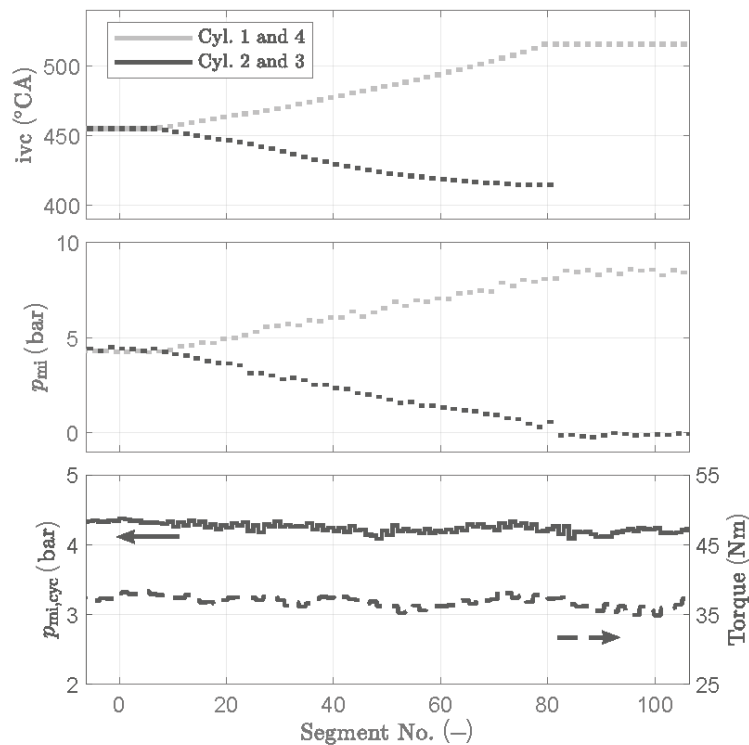


Figure 9. Measurement of a very slow cylinder deactivation. Top to bottom: intake valve closing angle (which represents the desired load), measured p_{mi} , cycle-average p_{mi} , and measured torque. For deactivated cylinders, no intake valve closing angle is shown, while the p_{mi} measurement of the gas spring cycle is shown.

6.2. Cylinder Reactivation

All three previously discussed cylinder reactivation methods are presented here. Then, as for the cylinder deactivation, an “instantaneous” reactivation is presented and discussed, followed by the analysis of a more gradual mode transition.

Figure 10 shows the reactivating cycle and one combustion cycle when the conventional (left) and the optimized (middle) exhaust valve timing is used, respectively. The right plot shows the reactivation when the intake valve is opened first. The measurements confirm that there is a significant difference in p_{mi} of the reactivating cycle. The measured $p_{mi,react}$ value in case of the conventional exhaust valve timing is only off by 0.03 bar. In case of the optimized exhaust valve timing, the difference is 0.08 bar, which may result from an underestimation of the flow coefficients of the exhaust valves for reverse flow direction or from neglecting the flow conditions in the exhaust manifold. For the strategy where the intake valve opens first, it was not possible to get the mixture to ignite. This may be different for direct-injected engines, which is why the general idea and valve strategy is presented here anyway. Both cases, when exhaust and intake valve timing are optimized, respectively, show lower losses than using the conventional exhaust valve timing. For the following cylinder reactivation measurements, the optimized exhaust valve timing is used for the reactivating cycle.

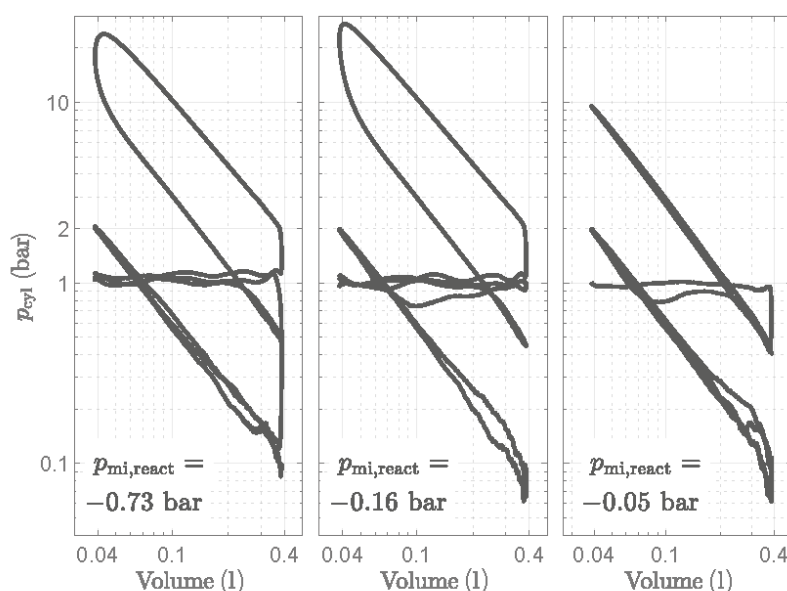


Figure 10. All plots show measurements of pressure traces of the reactivating cycle followed by one combustion cycle. The conventional exhaust valve timing is used on the **left**, while the **middle** plot shows the result when the optimized exhaust valve timing is used in the reactivating cycle. The **right** plot shows the reactivation when the intake valve is opened first.

For the cylinder reactivation strategy, one small change is made compared to the strategy presented in Reference [34]. In that publication, the p_{mi} of the reactivating cycle was assumed to be -0.8 bar, which corresponds to steady-state motored operation with deactivated intake valves and activated exhaust valves. Since with the fully variable valve-train optimized valve timings can be used to reduce these losses, the slight load increase of the firing cylinder group before the cylinder reactivation is not required here.

Figure 11 shows the results of an instantaneous cylinder reactivation. The results are very similar to those of an instantaneous cylinder deactivation.

Figure 12 shows a more gradual cylinder reactivation. The results are again very similar to those of a gradual cylinder deactivation. A soft reactivation reduces fluctuations of the cycle-averaged p_{mi} and the torque drop.

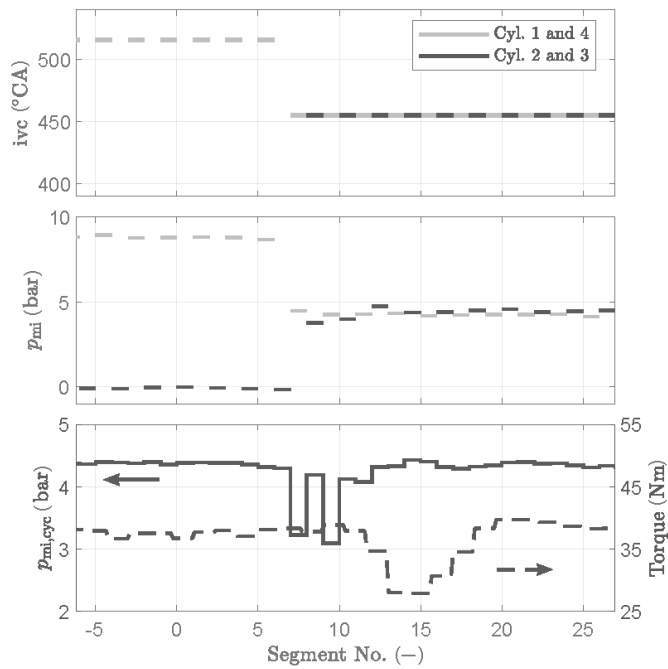


Figure 11. Measurement of an “instantaneous” cylinder reactivation. Top to bottom: intake valve closing angle (which represents the desired load), measured p_{mi} , cycle-average p_{mi} , and measured torque. For deactivated cylinders, no intake valve closing angle is shown, while the p_{mi} measurement of the gas spring cycle is shown. The term “instantaneous” is used here because this represents the fastest way to change back into four-cylinder operation.

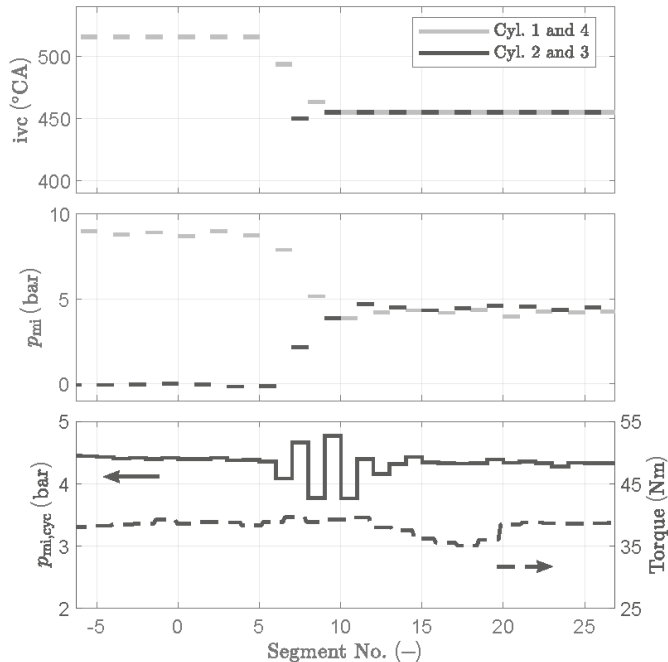


Figure 12. Measurement of a more gradual cylinder reactivation. Top to bottom: intake valve closing angle (which represents the desired load), measured p_{mi} , cycle-average p_{mi} , and measured torque. For deactivated cylinders, no intake valve closing angle is shown, while the p_{mi} measurement of the gas spring cycle is shown.

It must be noted that, due to the stochastic nature of the combustion process, especially for the first combustion after operation in the deactivated state and especially at very low loads, the desired p_{mi} is not always achieved. Using fuel enrichment for the

first combustion helps to avoid misfiring, but the reactivation from the deactivated state remains challenging. However, also for the cylinder reactivation, a clear reduction in the cycle-averaged p_{mi} and the torque drop could always be observed.

7. Conclusions

The transition from fired operation to gas spring operation, and vice versa, as well as the deactivated cylinder state, are essential for mode changes, such as cylinder deactivation or skip-firing operation.

A process simulation model was used to analyze how a cylinder behaves in the deactivated state. The results show that a different initial pressure before deactivating a cylinder or different blow-by properties still lead to the same steady-state in-cylinder mass. The steady-state is reached after approximately 6 s, which corresponds to 100 engine cycles at 2000 rpm and 200 engine cycles at 4000 rpm, respectively. The corresponding steady-state pressure amplitude changes for different compression ratios. The in-cylinder pressure level is important for the deactivating and reactivating cycle, respectively.

The simulation model was also used to analyze the last cycle before deactivating and the first cycle after reactivating a cylinder, respectively. Results show that optimizing the exhaust valve closing timing for the deactivation and the exhaust valve opening timing for the reactivation process are beneficial. While the advantage during the deactivation is in avoiding under-pressure, the advantage during the reactivating cycle over conventional valve timing is approximately 0.6 bar p_{mi} .

Experimental results confirm the advantages of optimized valve timing. For the reactivating cycle, the clear advantage can be measured, and the difference for the deactivating cycle is clearly visible in the pV-diagram.

Further experimental results for different types of cylinder deactivation/reactivation processes show how the specific strategy that is used can help to reduce an unavoidable torque drop. When the deactivation/reactivation process is done as quickly as possible, a large torque drop occurs. When the torque of the deactivating/firing cylinder group is decreased/increased more gradually, this torque drop can be reduced significantly.

This research shows that the transition to and from the gas spring operation must be taken into account and highlights the benefits resulting from an optimization of the valve timing. Beyond cylinder deactivation, this is relevant for skip-firing operation and other non-conventional engine operating modes, such as 8-stroke and 12-stroke operation, which will be part of future research. Furthermore, the experimental results show that maintaining low oscillations in the cycle-averaged p_{mi} during mode transitions avoids a large drop in torque at the output shaft. This insight may be helpful when managing other mode transitions, such as changing between 4-stroke and 8-stroke operation, which will be analyzed in future research, as well.

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Conflicts of Interest: The authors declare no conflict of interest.

Abbreviations

The following abbreviations are used in this manuscript:

NVH	noise, vibration, and harshness
ICE	internal combustion engine
ECU	engine control unit
rpm	revolutions per minute
cyc	cycle
deact	deactivation
react	reactivation
ivc	intake valve closing angle
p_{me}	mean effective pressure
p_{mi}	mean indicated pressure
p_{mq}	fuel mean effective pressure
e	Willans efficiency
$p_{me0,g}$	gas exchange losses
$p_{me0,f}$	mechanical friction losses

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Article

ANN Prediction of Performance and Emissions of CI Engine Using Biogas Flow Variation

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Abstract: Compression ignition (CI) engines are popular in the transport sector because of their high compression ratio. However, in recent years, it has become a major concern from an environmental point of view because of the emission and depleting fossil fuel. The advanced combustion concept has been a popular research topic in the CI engine. Low-temperature combustion with alternate fuel has helped in reducing the oxides of nitrogen (NO_x) and soot emission of the engine. Biogas is a popular substitute of energy especially deduced from biomass because of its clean combustion properties, as well it being a renewable energy source compared to non-renewable diesel resources. In experiments with dual fuel, i.e., conventional diesel and alternate fuel (biogas) were carried out through them. In the present study, an artificial neural network model was used to estimate emissions and check the attributes of performance. Different algorithms and training functions were used to train the models. However, the best training algorithm was Levenberge Marquardt and the training function was Tansig (Hyperbolic tangent sigmoid) and Logsig (logarithmic sigmoid), which showed the best result with regression coefficient ($R > 0.98$) and Mean square error ($MSE < 0.001$). The best model was trained by evaluating MSE and regression coefficient. Experimental results and artificial neural network (ANN) prediction showed that the experimental results were similar to each other and lie at the same intervals. The ANN model helped in predicting experimental data that were earlier difficult to experimentally perform using interpolation and extrapolations. It was observed that there was an increase in Brake Specific Energy Consumption (BSEC) and a decrease in Brake thermal efficiency (BTE) with improved biogas flow rate and reduced NO_x emission in the combustion chamber. Carbon monoxide (CO) and hydrocarbon (HC) emissions increase linearly with the increase in biogas flow rate, whereas smoke opacity decreases. It could be concluded that this study helps in understanding the effect of dual fuel (diesel-biogas) combustion under different load conditions of the engine with the help of ANN, which could be a substitute fuel and help to protect the environment.

Keywords: biogas; dual fuel; emissions; diesel; ANN modeling; biomass

1. Introduction

In recent years, the interest in alternate fuels and renewable energy [1] has increased because of an increase in energy demand [2] and stringent environmental policies [3], which are because of the depleting fossil fuels, and an increase in the price of fossil fuels for the internal combustion engine [4]. A compression ignition engine is being used by the transportation sector and holds a major share [5]. It was also observed that conventional compression ignition (CI) engines had higher NO_x and soot. To control the emissions, exhaust gas recirculation and an after-treatment method (diesel particulate filter, diesel oxidation catalyst, and selective catalytic reduction), or both, were incorporated together. Incorporating these systems in the engine increases the complexity and cost of the engine. Advance combustion concepts and specific fuels were the topics of interest for many

researchers [6]. Energy demand has been increasing [7] and demand for decarbonization is important for the environment [8]. Therefore, an urgent need for alternate fuels is required for the CI engine [9], both liquid as well as gases [10,11]. In the case of IC engines, gaseous alternate fuel can be considered because of their high compression ratio [12] and good mixing characteristics, which in turn would decrease the emission and increase brake thermal efficiency [13].

For the CI engine, advanced combustion strategies such as:

- Reactivity controlled compression ignition (RCCI)
- Homogeneous charge compression ignition (HCCI)
- Partially premixed combustion (PPC) were used

NO_x and soot emissions were reduced by partial premix combustion when applied to the IC engine. PPC helped in reducing heat transfer losses and shortened the combustion duration compared to conventional diesel engines. Other researchers mixed low-temperature combustion with alternate fuels (ethanol, butanol, natural gas, butanol, bio-methane, etc.) [14,15]. Bio-methane has been a popular alternate fuel in Poland and Italy. It could be developed from animal slurry, biodegradable waste, and from maize grown on agricultural land [16,17]. This combination showed improvement in NO_x , soot emissions and efficiency. Generally, in dual-fuel combustion, one of the fuels is of high reactivity and the other is of low reactivity [18]. Biomethanol, as an alternative fuel in the transport and industrial sector, needs to be investigated [19].

A potential renewable energy is biogas, which could be produced from organic material under natural degradation by micro-organisms without the use of oxygen. Organic substances are converted to biogas by anaerobic digestion, which is used as fuel for vehicles and to generate heat and electricity. Biogas mainly constitutes of methane and carbon dioxide [20]. Biogas for industrial purposes is developed at (1) landfills, (2) agricultural organic waste digestion plants, (3) sewage treatment plants, and (4) sites with industrial processing units [21].

Biogas is environment friendly [22] and is available abundantly [23,24]. The use of biogas in the CI engine is difficult, as there is no spark plug for combustion and ignition. Biogas is also low in cetane number and has a high self-ignition temperature [25]. CO_2 composition of biogas helps in combustion at low temperature, which reduces the chance of NO_x emission formation at elevated temperature during combustion in dual fuel mode [26,27]. BTE remains unchanged for intermittent loads, whereas, at reduced loads, it decreases and increases at maximum loads [28,29]. Thus, by using biogas as a dual fuel mode [30], smoke and NO_x emission were reduced and controlled [31,32].

Karthic et al. used an artificial neural network to foreshow the performance and emission of a diesel engine. The ANN model has input parameters such as the load on the engine, the injection pressure of fuel, fuel flow rate and injection timing of fuel; and the output parameters of the model were brake thermal efficiency and emission. It was evident that the experimental results and the ANN prediction were similar for the dual-fuel engine in terms of emission and performance [33]. Gul et al. obtained the optimum combination of engine speed, operating load and fuel nature by Taguchi DOE in a diesel engine that was run by 100% waste cooking oil and B20 (i.e., 20% biodiesel and 80% diesel). Experimental results and ANN simulation computed the best combination by guaranteeing refinement of the output response factors, thus ratifying the Gray–Taguchi method in curtailing emissions and enhancing combustion and performance simultaneously [34]. To conduct the experiment, Kumar used RSM based on box–Behnken experimental design. For the production of jatropha-algae oil, parameters such as the molar ratio, reaction time, catalyst concentration, and reaction temperature were optimized. The predicted results showed a correlation with the RSM outcomes [35]. Samuel et al. modeled the production of coconut oil ethyl ester by RSM and ANN. It was observed that the predicted yield by ANN agreed with the output of the experiment [36]. Calik et al. (2018) used corn, sunflower and canola biodiesel blends in a diesel engine, injected hydrogen through a manifold inlet and predicted the emission, noise and vibration level with the help of

a support vector machine and artificial neural network. It was concluded that ANN predicted better results than SVM [37]. Najafi et al. (2019) experimented on a CI engine, which was simultaneously run by pilot fuel (oxygenated additive) and main fuel (natural gas). Artificial Neural Network and genetic algorithm modeling were used to reduce emission by establishing the ratio of pilot fuel in respect to biodiesel, gaseous fuel, and additive [38]. Javed et al. used hydrogen fuel with ZnO nano additives biodiesel in a diesel engine. An artificial neural network was utilized to forebode noise with different engine criteria. ANN was also used so that extensive experimentation could be avoided.

This paper examined the performance and emission features under the influence of diesel and biogas used together at varying engine loads at different gas flow rates. The prediction of performance and emission was carried out by an artificial neural network [39]. Biogas was introduced into the combustion chamber through an inlet manifold. The comparative analysis of the prediction and the actual data are presented in this paper.

2. Experimental Setup

DAF8 Kirloskar make, single cylinder engine, four stroke-naturally aspirated, with a power of 6 kW was employed in the experiment. The experimental setup is shown in Figure 1. In India, such single-cylinder CI engines are mostly used in agriculture and for commercially generating electricity applications in rural areas. Table 1 depicts the specification of the CI engine employed for the experiment. Properties of different fuels used for the experiment are enlisted in Table 2 as per the American society for testing and materials (ASTM) standards. Table 3 describes biogas and its composition that was produced locally from vegetable and fruit waste along with animal leftovers. After a 45 to 60 day cycle, biogas was formed by anaerobic fermentation of the waste materials. The introduction of the gas mixture at the time of suction stroke takes place through the inlet air manifold. Flowmeter was used to evaluate the discharge of biogas, attached at inlet manifold pipe through a venturi meter. The fuel control mechanism regulated the fuel flow in the engine.

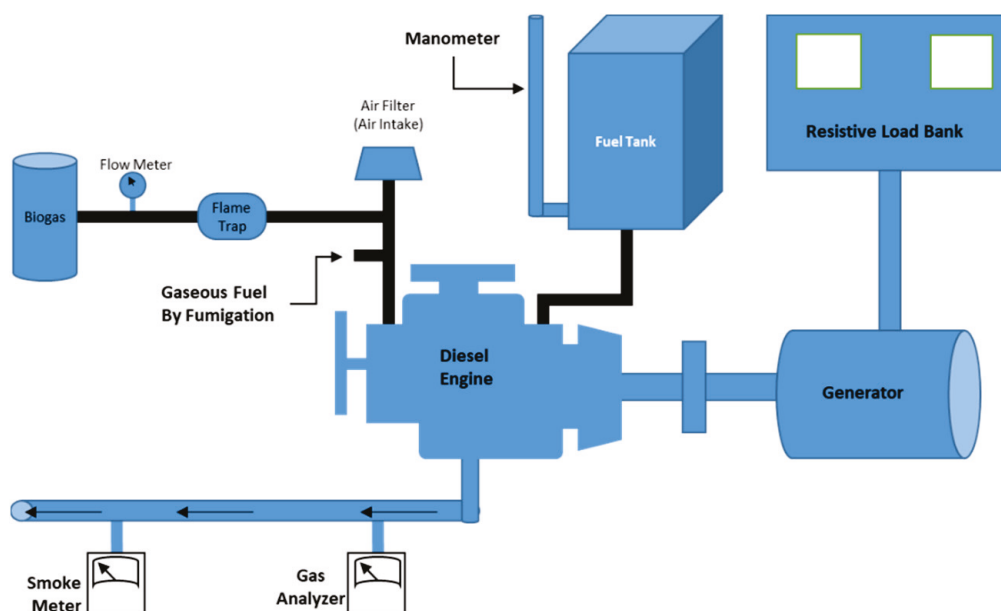


Figure 1. Experimental setup block diagram.

Table 1. Engine specification for experimentation.

Company	Kirloskar Oil India Ltd.
Model	DAF8
Bore × stroke	95 × 110 mm
Power	8 BHP
Speed (RPM)	1500
Inlet valve opening (degree)	4.5°
Inlet valve closed (degree)	35.5°
Exhaust valve opening (degree)	35.5°
Exhaust valve closed (degree)	4.5°
Injection type	Direct Injection
Nozzle opening pressure	200 bar
No. of holes (diesel injector)	4
Cylinder's	1
Compression ratio	17.5:1
Cylinder volume	780 cc
Static injection timing	26° bTDC
Alternator Specifications	
Company	Kirloskar Private Limited
Dynamometer	AC alternator, 50 Hz, single phase
Current rating	21.7 A
Rated output	5 kVA
Voltage rating	230 V
Rated speed	1500 rpm
Boundary Conditions	
Intake temperature	Room temperature
Boost pressure	205–210 bar
Injection quantity	0–10 kg/h
Injection strategy	Single Point

Table 2. Fuel attributes.

Attributes	Biogas	Diesel
Chemical Composition	CH ₄ (60%), CO ₂ (40%) (volume)	C ₁₂ H ₂₆
Cetane Number	-	45–55
Density (kg/m ³)	1.1	840
Auto-ignition temperature (K)	1086	553
Lower Calorific Value (MJ/kg)	20.67	42
Stoichiometric air-fuel ratio	10	14.92

Table 3. Constitution of biogas.

Name	Formula	Amount (%)
Methane	CH ₄	50~70
Hydrogen	H ₂	5~10
Water Vapor	H ₂ O	0.3
Carbon Dioxide	CO ₂	30~40
Nitrogen	N ₂	1~2
Hydrogen Sulfide	H ₂ S	Traces

Injection pressure and timing had been kept constant for engine operation under dual-fuel mode because of recommendation by the manufacturer. The experiment was conducted at a steady-state condition and a determined angular speed of 1500 rpm was attained. Standard data were generated with the help of conventional diesel fuel at various engine loads. Engine load was increased by 20% step by step, up to 100% for both

the fuel operations. The parameters that were recorded at different engine operations were:

- fuel flow;
- air consumption;
- biogas flow rate;
- temperatures;
- power output;
- exhaust tailpipe emissions.

3. Performance Analysis

In the previous literature, performance characteristics were calculated [40,41] with the help Equations (1)–(3), shown below:

$$\text{Brake Power} = \frac{\text{Voltage (V)} \times \text{Current (I)}}{\eta \times 1000} \text{KW}, \quad (1)$$

where η = Efficiency.

$$\text{Brake Thermal Efficiency} = \frac{B.P \times 3600 \times 100}{(n_{bio} \times C_{bio} + n_d + C_d)}, \quad (2)$$

where n_{bio} and n_d are mass of biogas (kg/h) and diesel fuel. C_{bio} and C_d are the lower calorific value of biogas (kJ/kg) and diesel fuel (kJ/kg).

$$\text{Brake Specific Energy Consumption} = \frac{\left\{ \sum (m_{totalfuel} \times CV_{totalfuel}) \right\}}{\text{Brake Power (B.P)}} \text{KW}, \quad (3)$$

where $m_{totalfuel}$ and C_{total} fuels are the mass and lower calorific value of the total amount of fuel.

3.1. Exhaust Gas Emissions Analysis

Data recording of the exhaust gas emission was carried out by AVL Digas 444N, connected at the tailpipe. O_2 , CO_2 and CO were measured in % vol., NO_x and HC were measured in gm/kW hr. Equipment AVL 437C evaluated the smoke opacity. ASTM-D6522 protocol was used for the measurement of gas emission.

3.2. Uncertainty Analysis

Table 4 shows the uncertainty percentage that is associated with the measuring equipment. The uncertainty percentage in Table 4 has been enlisted from the equipment specifications as provided by the equipment make after a quality check. During the experiment, different parameters were measured and the error associated with them was calculated with the help of uncertainty analysis equations. Experimental data recording was repeated thrice for an average value and to maintain high accuracy. Uncertainty was calculated as shown by the below Equations (4) and (5) [2,42]:

$$\text{Overall uncertainty} = \sqrt{\frac{(Fuel\ flow\ rate)^2 + (Flow\ properties)^2 + (CO)^2 + (CO_2)^2 + (NO_x)^2 + (HC)^2 + (Smoke\ opacity)^2 + (Engine\ load)^2 + (Temperature\ indicator)^2}{}} \quad (4)$$

$$= \sqrt{(1.0)^2 + (1.0)^2 + (0.2)^2 + (0.2)^2 + (0.5)^2 + (0.1)^2 + (1.0)^2 + (0.5)^2 + (0.15)^2} = \pm 2. \quad (5)$$

Table 4. Uncertainty analysis.

Equipment Name	Units	Uncertainty Percentage
Fuel flow rate	mL	±1.00
Fuel properties	-	±1.00
CO	vol. %	±0.20
CO ₂	vol. %	±0.20
NO _x	ppm	±0.50
HC	ppm	±0.10
Smoke opacity	vol. %	±1.00
Engine Load	-	±0.50
Temperature Indicator	°C	±0.15

4. Artificial Neural Network

Artificial neural network was extensively employed for predicting the different thermal application in this research. ANN can help in the prediction of different characteristics of the engine.

ANN model has many processing elements known as “neurons”, which are similar to the human brain. They are interlinked to each other and data is fed into the neurons and processing is carried out. Weights are defined to the neurons, based on which learning is carried out by the network (i.e., for learning, testing and validation). Mean square error is the performance function on which the ANN model is evaluated. Refinement of the activation level is carried out by determining the error weights [34]. This method of determining the error weights is called feed forward back propagation network. Hidden layers had to be included in the input and output layer because of the non-linear experimental data, therefore, a multilayer neural network was created [43]. The model was trained using Gradient descent with adaptive learning rate (TRAINDA) [44]. Hyperbolic tangent sigmoid (TANSIG), Logarithmic sigmoid (LOGSIG), and Linear (PURELIN) were the transfer functions for the model’s output [45].

5. Data Normalization

The performance of the ANN model depends on the presentation of the input layer data. Input and output data have to be graded to maximize the performance. The model used in this study is a back propagation model. The performance was tremendously affected by scaling of input and output data. The logistic sigmoid transfer function was used to generate data between 0 and 1 [46].

$$\eta v_i = 2 \times \left(\frac{v_{\min} - v_i}{v_{\min} - v_{\max}} \right) - 1, \quad (6)$$

$$\eta v_i = 0.8 \times \left(\frac{v_{\min} - v_i}{v_{\min} - v_{\max}} \right) - 0.1. \quad (7)$$

Normalization of both the data in the range 0.1–0.9 were carried out using a simple method [47]. Normalization formula used by different literature are shown by Equations (6) and (7), and input and output values were normalized by Equation (7). Before training the model, randomization of data was carried out and 70% of data was selected for model learning. After training the model, authentication was carried by 15% data and the remaining 15% was used for efficacy testing of the model [45,48].

6. Modelling and Simulation

Artificial neural network model was developed by using Matlab R2018. Biogas flow rate and the load were the input for all the experimental trials used in the ANN model. Emission and performance data acquired during the experimental process was used as an output parameter in the model. ANN configuration diagram is shown in Figure 2.

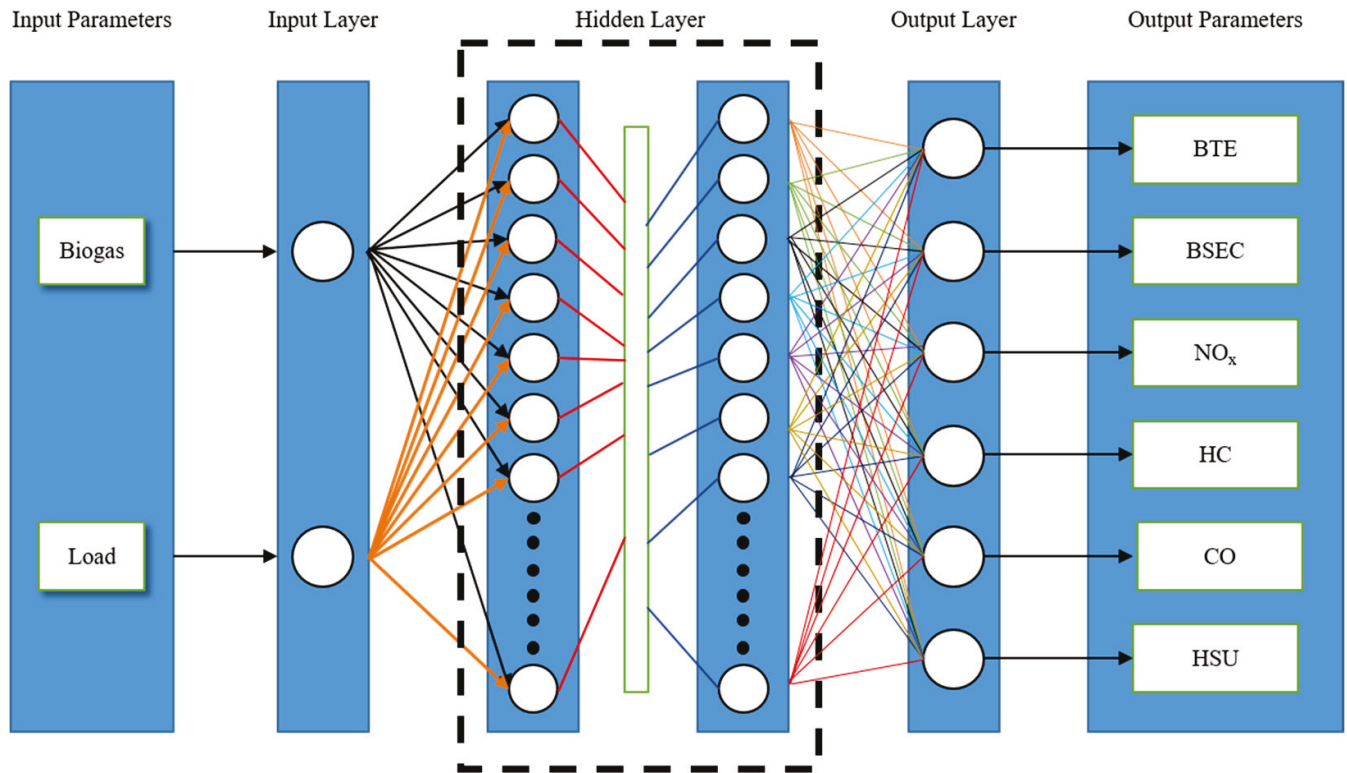


Figure 2. Artificial neural network structure for experiment.

Artificial neural network model was trained in MATLAB R2018 by employing different training algorithms, functions, and changing the neurons in different layers. Tansig (Hyperbolic tangent sigmoid) and Logsig (logarithmic sigmoid) transfer functions were used in the layers of the model. Weights and bias were randomly chosen by MATLAB and initially the network executed 100 iterations. The minimum gradient was 10^{-7} and the stopping criteria for the network were 10,000 epochs [45].

To understand the output of the model, a simulation of all input data matching was carried out. The ANN model was evaluated by regression coefficient Equation (8), Mean square error Equation (9) by employing targets and outputs of the model:

$$\text{Regression Coefficient} = \sqrt{1 - \left\{ \frac{\sum_{i=1}^n (T_i - O_i)^2}{\sum_{i=1}^n O_i^2} \right\}}, \quad (8)$$

$$\text{Mean Squared Error} = \frac{1}{n} \left\{ \sum_{i=1}^n (T_i - O_i)^2 \right\}. \quad (9)$$

The two statistical expressions value ($R > 0.98$ and $\text{MSE} < 0.001$) were used together to evaluate the model. After satisfying the iterations, it would be terminated. Figure 3 represents the developed ANN algorithm in MATLAB and the predicted data from the ANN model is given in Table 5.

Table 5. Predicted data from the ANN model.

S.No	Bio Gas (kg/h)	BSEC (MJ/kWh) Predict				
		20% Engine Load	40% Engine Load	60% Engine Load	80% Engine Load	100% Engine Load
1	1	38.51	27.53	22.28	20.00	18.27
2	2	41.44	29.17	22.77	20.97	18.48
3	3	44.81	30.79	23.06	21.93	18.97
4	4	47.74	32.95	23.56	22.49	19.56
5	5	50.50	35.03	24.73	23.04	20.17
6	6	53.51	37.04	26.10	23.84	20.86
7	7	56.50	38.95	27.60	24.54	21.61
8	8	61.10	40.91	29.02	25.28	22.20
9	9	65.25	43.07	30.19	26.02	22.71
10	10	69.51	45.10	31.08	26.75	23.11
11	11	72.56	46.62	33.55	27.44	23.36
12	12	75.49	48.78	34.05	27.99	23.95
13	13	78.25	50.87	35.22	28.54	24.56
14	14	81.26	52.88	36.59	29.34	25.26
15	15	84.26	54.78	38.09	30.04	26.00

S.No	Bio Gas (kg/h)	Brake Thermal Efficiency (Predicted)				
		20% Engine Load	40% Engine Load	60% Engine Load	80% Engine Load	100% Engine Load
1	1	8.57%	14.92%	18.79%	19.93%	21.99%
2	2	8.02%	14.59%	18.50%	19.95%	21.91%
3	3	7.49%	14.14%	18.11%	19.35%	21.49%
4	4	6.96%	13.68%	17.52%	18.99%	21.32%
5	5	6.45%	13.38%	16.75%	18.37%	20.86%
6	6	5.98%	13.06%	16.05%	17.70%	20.46%
7	7	5.54%	12.52%	15.47%	17.59%	20.12%
8	8	5.16%	11.86%	14.99%	17.25%	19.79%
9	9	4.92%	11.51%	14.43%	16.81%	19.51%
10	10	4.75%	11.29%	13.87%	16.38%	19.18%
11	11	4.46%	11.04%	13.37%	15.85%	18.83%
12	12	4.08%	11.06%	12.89%	15.52%	18.50%
13	13	3.84%	10.55%	12.33%	15.08%	18.22%
14	14	3.66%	9.85%	11.77%	14.65%	17.89%
15	15	3.38%	9.48%	11.27%	14.12%	17.54%

S.No	Bio Gas (kg/h)	NOx (g/kWh) Predict				
		20% Engine Load	40% Engine Load	60% Engine Load	80% Engine Load	100% Engine Load
1	1	42.78	35.26	26.31	20.55	14.40
2	2	41.54	33.59	24.66	19.72	13.69
3	3	39.51	32.11	23.39	18.87	13.01
4	4	37.39	30.20	22.40	17.89	12.26
5	5	35.78	28.35	21.49	17.06	11.41
6	6	34.41	26.56	20.48	16.44	10.52
7	7	32.94	24.82	19.44	15.77	9.64
8	8	31.47	23.29	18.98	14.94	8.44
9	9	29.65	22.15	18.20	14.01	7.09
10	10	26.49	20.70	17.21	13.15	6.02
11	11	25.52	18.45	15.68	12.42	5.56
12	12	24.16	16.67	14.67	11.80	4.68
13	13	22.68	14.93	13.63	11.14	3.79
14	14	21.21	13.40	13.17	10.31	2.60
15	15	19.40	12.26	12.39	9.38	1.23

Table 5. Cont.

S.No	Bio Gas (kg/h)	CO (% Vol) Predicted				
		20% Engine Load	40% Engine Load	60% Engine Load	80% Engine Load	100% Engine Load
1	1	0.312	0.119	0.061	0.090	0.170
2	2	0.320	0.128	0.061	0.093	0.175
3	3	0.320	0.127	0.066	0.094	0.180
4	4	0.324	0.135	0.067	0.096	0.182
5	5	0.330	0.137	0.070	0.098	0.185
6	6	0.331	0.140	0.071	0.100	0.190
7	7	0.337	0.142	0.073	0.101	0.193
8	8	0.340	0.147	0.075	0.103	0.196
9	9	0.345	0.150	0.076	0.106	0.199
10	10	0.346	0.155	0.078	0.107	0.205
11	11	0.350	0.159	0.081	0.110	0.211
12	12	0.353	0.162	0.086	0.112	0.212
13	13	0.357	0.167	0.087	0.113	0.217
14	14	0.362	0.176	0.090	0.115	0.221
15	15	0.367	0.184	0.092	0.117	0.226

S.No	Bio Gas (kg/h)	HC (g/kWh)				
		20% Engine Load	40% Engine Load	60% Engine Load	80% Engine Load	100% Engine Load
1	1	1.18	0.78	0.66	0.47	0.28
2	2	1.32	0.85	0.70	0.54	0.28
3	3	1.42	0.92	0.76	0.58	0.28
4	4	1.56	0.99	0.78	0.60	0.28
5	5	1.65	1.05	0.77	0.67	0.29
6	6	1.77	1.09	0.80	0.73	0.30
7	7	1.87	1.17	0.86	0.77	0.32
8	8	1.95	1.24	0.89	0.78	0.34
9	9	2.00	1.29	0.95	0.79	0.37
10	10	2.09	1.35	0.98	0.83	0.38
11	11	2.20	1.42	1.00	0.87	0.39
12	12	2.29	1.47	1.03	0.93	0.41
13	13	2.41	1.51	1.09	0.98	0.43
14	14	2.51	1.59	1.12	1.02	0.45
15	15	2.59	1.66	1.18	1.08	0.47

S.No	Bio Gas (kg/h)	Smoke Opacity (% HSU)Predict				
		20% Engine Load	40% Engine Load	60% Engine Load	80% Engine Load	100% Engine Load
1	1	15.75%	23.87%	29.89%	36.12%	44.93%
2	2	15.14%	23.41%	28.03%	34.25%	44.21%
3	3	14.54%	22.97%	26.97%	32.01%	41.98%
4	4	13.86%	22.36%	25.78%	29.89%	40.78%
5	5	13.03%	21.38%	24.09%	28.69%	39.10%
6	6	11.98%	20.00%	22.03%	28.00%	37.90%
7	7	10.77%	18.50%	20.46%	27.00%	36.40%
8	8	9.77%	16.84%	19.91%	25.85%	34.86%
9	9	9.30%	15.85%	19.19%	24.90%	33.34%
10	10	9.13%	14.55%	16.90%	23.91%	31.80%
11	11	8.11%	13.99%	15.75%	21.90%	30.10%
12	12	7.55%	13.43%	15.19%	21.10%	28.89%
13	13	6.99%	12.87%	14.47%	20.15%	27.40%
14	14	6.43%	12.31%	13.92%	19.06%	25.85%
15	15	5.78%	11.73%	12.77%	18.11%	24.33%

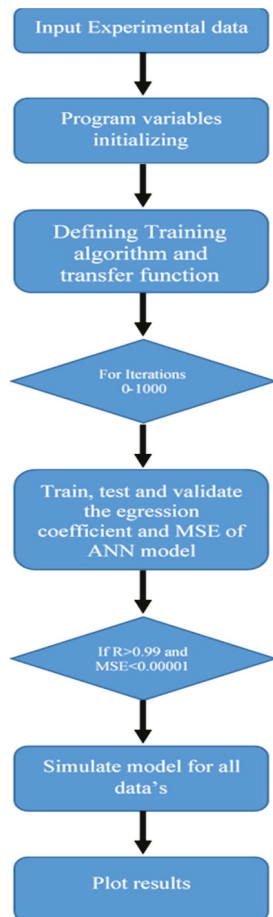


Figure 3. ANN model flowchart.

7. Results and Discussion

The model was trained with different algorithms and training functions, but the best training algorithm was Levenberge Marquardt and the training function was Tansig (Hyperbolic tangent sigmoid). The Logsig (logarithmic sigmoid) showed the best result with ($R > 0.98$) and ($MSE < 0.001$). The best model was trained by evaluating the mean square error and regression coefficient.

ANN predictions were used for the experimental values with regression coefficient and predictions. ANN predictions were matched with the actual data. Regression coefficient for emission for BSEC, BTE, NO_x , CO, HC and smoke opacity were 0.99939, 0.99866, 0.99699, 0.99942, 0.99706, and 0.99865 respectively.

Figure 4 shows the accuracy of the training data, validation of the data, and test data. The closeness of data points with the Fit line shows that the accuracy of the predicted data and the regression coefficient ($R = 0.99939$) will be higher. The variation of BSEC at different engine loads is given in Figure 5. From the study, it is clear that BSEC was the highest and as we increase the flow of biogas from 1 kg/h to 15 kg/h, the BSEC increases linearly. At 100% engine load, BSEC was the lowest. The predicted values of BSEC at 20% load 1 kg/h biogas was 38.51 MJ/kWh, whereas at 15 kg/h BSEC was 84.26 MJ/kWh. As the engine load was increased, the BSEC decreased. BSEC at 100% engine load at 1 kg/h was 18.27 MJ/kWh. By increasing the biogas to 15 kg/h, the BSEC increased to 26.00 MJ/kWh. It can be concluded that the increase of biogas flow rate resulted in an overall lower heating value. From the figure, the maximum value was obtained for 20% engine load at 15 kg/h biogas mass flow rate. Figure 6 shows the regression coefficient ($R = 0.99866$) for BTE predictions for the accuracy of the training data, validation of the data, and test data. A variation of BTE with the variation of the mass flow rate of biogas at different engine loads is given in Figure 7. From the figure, it is clear that BTE is lower at

20% engine load, whereas at higher engine load BTE also increases. At 20% engine load at 1 kg/h biogas mass flow rate, the value of BTE was 8.57% and on increasing the gas flow rate, the BTE reduced to 3.38%. The highest BTE was at 100% engine load at 1 kg/h biogas flow rate with 21.99% BTE. On increasing the flow rate, the BTE was reduced to 17.54% at 15 kg/h. It could be observed that upon increasing the engine load, BTE increased. It was because poor utilization of gaseous fuel mixture resulted in reduced BTE under dual fuel mode. Figure 8 shows the regression coefficient ($R = 0.99966$) for NO_x prediction. This shows how closely the training data and test data match closely with each other. Variation of NO_x with the variation in the mass flow rate of biogas at different engine loads is given in Figure 9. From the figure, it is clear that at a lower engine load, NO_x emission is higher. At 20% engine load and 1 kg/h biogas flow rate, the value of NO_x was 42.78 g/kWh and on increasing the gas flow rate, it reduced to 19.40 g/kWh. At 100% engine load and 1 kg/h biogas flow rate, NO_x was 14.40 g/kWh and on increasing the flow rate further, it reduces to 1.23 g/kWh. Increasing the engine load reduces the NO_x . It further reduces on increasing the biogas flow rate in the engine. It could be justified as the use of biogas diminishes the harmful emissions.

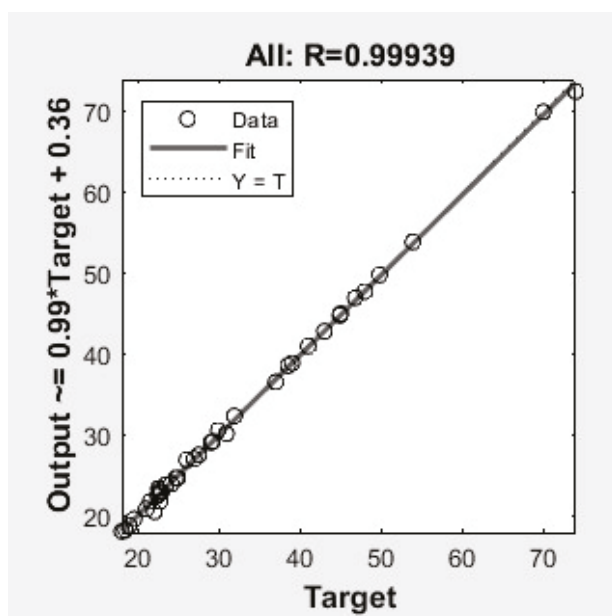


Figure 4. Accuracy of the target and output data of BSEC using a Regression Coefficient metric.

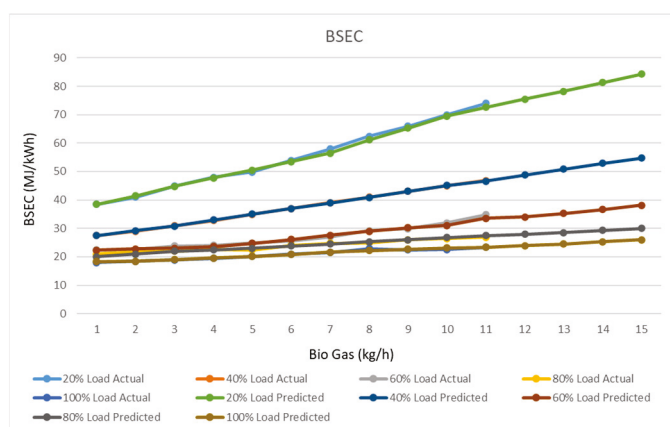


Figure 5. Effect on BSEC by change in engine load.

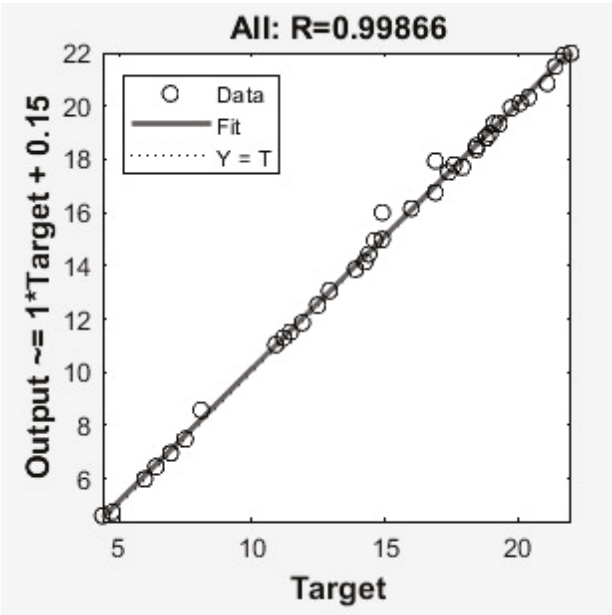


Figure 6. Accuracy of the target and output data of BTE using a Regression Coefficient metric.

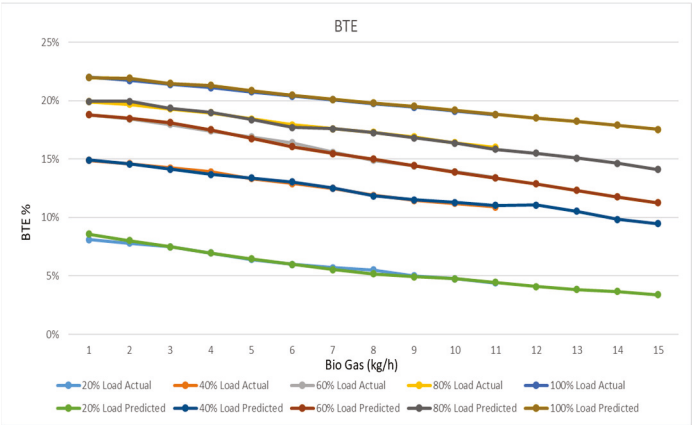


Figure 7. Effect on BTE with change in engine load.

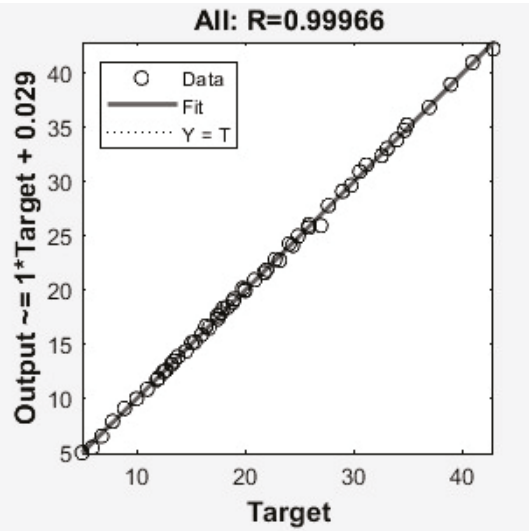


Figure 8. Accuracy of the target and output data of NO_x using a Regression Coefficient metric.

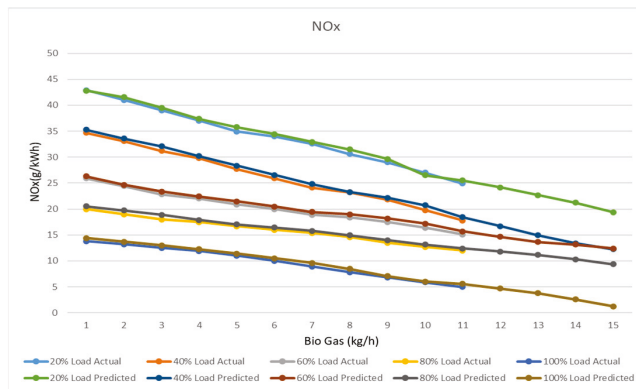


Figure 9. Effect on NO_x with change in engine load.

Figure 10 shows the regression coefficient of CO emission, i.e., $R = 0.99942$. CO emission increases with an increase in engine load and biogas flow rate. CO emission increases with a decrease in load and increase in biogas flow into the engine, as shown in Figure 11. At 20% engine load and 1 kg/h biogas flow rate, CO emission was 0.312% Vol, increasing the biogas flow rate to 15 kg/h increases the CO to 0.367% Vol. On increasing the engine load at 1 kg/h biogas flow rate, CO emission were 0.119% Vol, 0.061% Vol, 0.090% Vol, and 0.170% Vol, respectively for 40%, 60%, 80%, and 100% engine load. Increasing the mass flow rate of biogas in the engine decreases the oxygen supply in the engine, which leads to higher CO emissions.

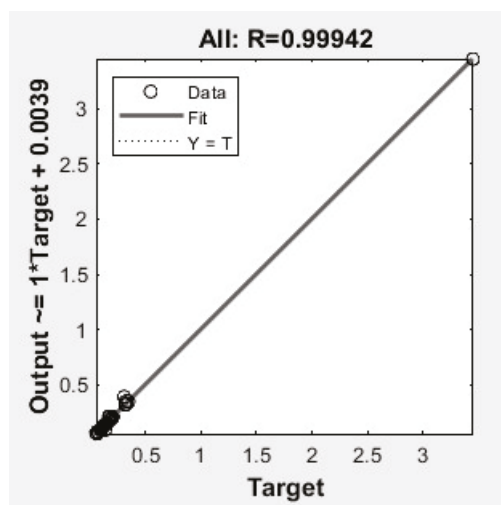


Figure 10. Accuracy of the target and output data of CO using a Regression Coefficient metric.

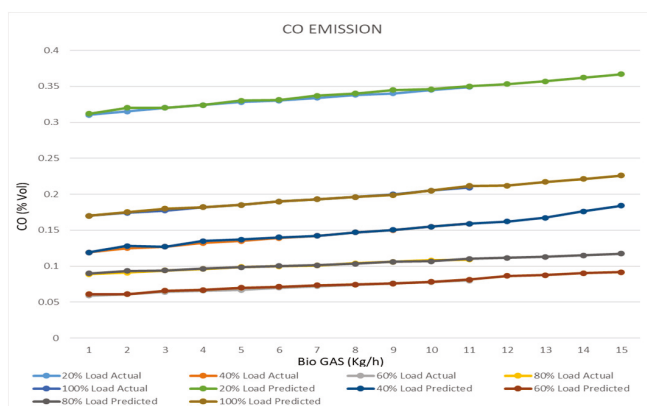


Figure 11. Effect on CO with change in engine load.

Figure 12 shows the regression coefficient ($R = 0.99706$) for HC prediction. Variation of HC with the variation of the mass flow rate of biogas at different engine loads is given in Figure 13. From the figure, it is clear that HC emissions are lower at lower engine loads. At 20% engine load and 1 kg/h biogas flow rate, the value of HC was 1.18 g/kWh and on increasing the gas flow rate it increases to 2.59 g/kWh. At 100% engine load and 1 kg/h biogas flow rate, the HC was 0.28 g/kWh and on increasing the flow rate further, it increases to 0.47 g/kWh. Increasing the engine load reduces the HC. It increases by increasing the biogas flow rate in the engine. At 15 kg/h biogas flow rate, HC was 2.59, 1.66, 1.18, 1.08, and 0.47 g/kWh for 20%, 40%, 60%, 80%, and 100% engine load, respectively. An increase in HC with an increase in biogas flow rate could be justified by the lower flame velocity of biogas. The regression coefficient for smoke opacity was 0.099865, as shown in Figure 14. Variation of smoke opacity with the variation of the mass flow rate of biogas at different engine loads is given in Figure 15. From the figure, it is clear that smoke opacity increases the engine load. At 20% engine load and 1 kg/h biogas flow rate, smoke opacity was 15.75%. Increasing the biogas flow rate to 15 kg/h reduces the smoke opacity to 5.78%. At 100% engine load, 1 kg/h biogas flow rate smoke opacity was 44.93% and increasing the biogas flow reduced the smoke opacity to 24.33%. A decrease in smoke opacity with an increase in biogas is because of the absence of aromatic compounds in the biogas composition.

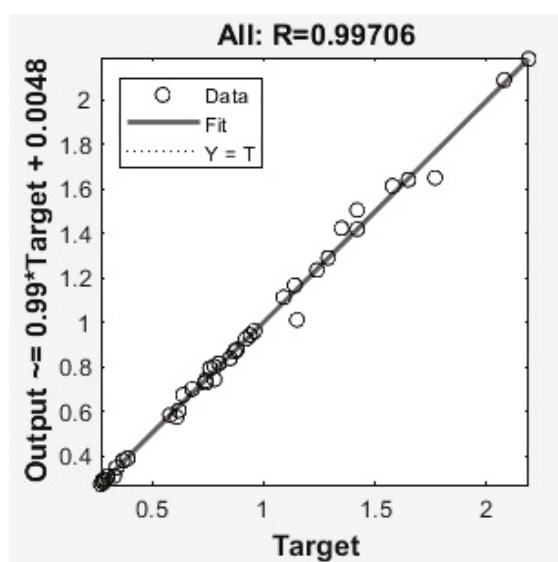


Figure 12. Accuracy of the target and output data of HC using a Regression Coefficient metric.

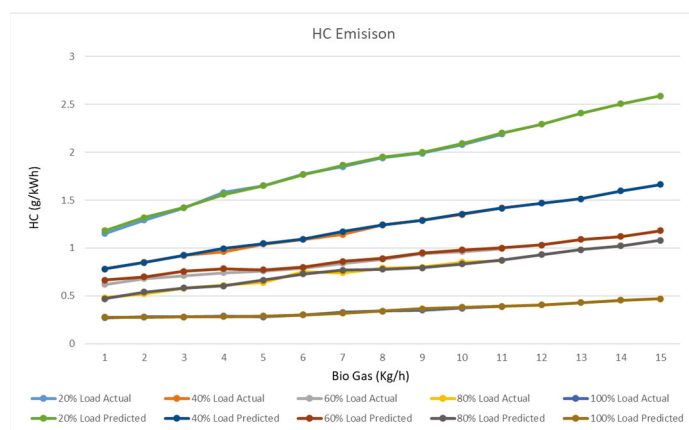


Figure 13. Effect on HC with change in engine load.

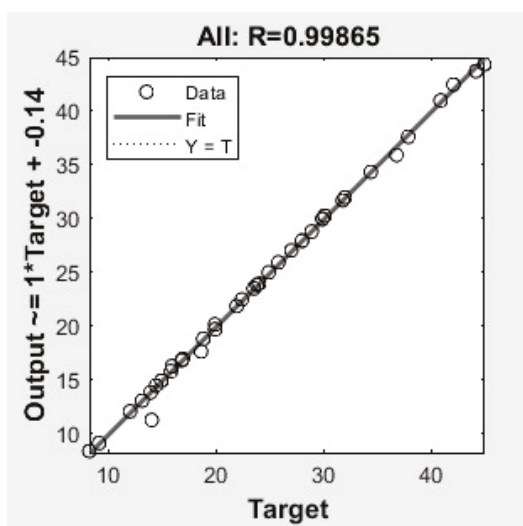


Figure 14. Accuracy of the target and output data of smoke opacity using a Regression Coefficient metric.

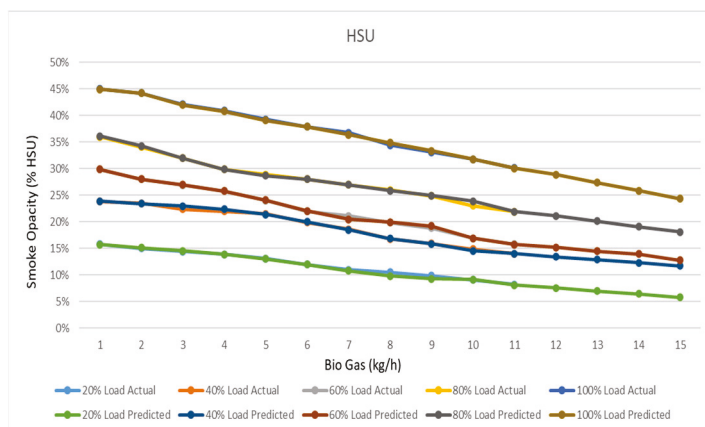


Figure 15. Effect on smoke opacity with change in engine load.

8. Conclusions

Performance and emission characteristics of a dual-fueled compression ignition engine were predicted using an artificial neural network, by varying the mass flow rate of biogas. Biogas is a clean fuel (obtained by anaerobic digestion from agricultural wastes) and a better alternative to conventional fuels, which reduces the NO_x emissions gases without a major change in the existing diesel engine. ANN helped in predicting the performance and emission characteristics of the CI engine at different biogas mass flow rates. The comparative study shows that the experimental results are clearly identical to the ANN results. Based on the study it was noted that:

- ANN model was trained using the Levenberge Marquardt algorithm and training function was Tansig (Hyperbolic tangent sigmoid) and Logsig (logarithmic sigmoid).
- ANN model was evaluated on the basis of the regression coefficient ($R > 0.98$) and Mean squared error ($\text{MSE} < 0.0001$).
- BSEC was highest at 20% engine load and as we increase the flow of biogas from 1 kg/h to 15 kg/h BSEC increases linearly. At 20% engine load and 1 kg/h biogas flowrate, BSEC was 38.51 MJ/kWh, whereas at 100% load it was 18.27 MJ/kWh. Increasing the biogas flow rate increases the BSEC to 84.26 MJ/kWh and 26.00 MJ/kWh for 20% engine load and 100% engine load, respectively. It could be justified as an increase in biogas resulted in a lower heating value.

- BTE at 20% and 100% engine load and 1 kg/h biogas flow rate was 8.57% and 21.99%. On increasing the gas flow rate to 15 kg/h, BTE was reduced to 3.38% and 17.54%. Poor utilization of gaseous fuel may be blamed for lower BTE under dual fuel mode.
- NO_x prediction showed that at 20% engine load and 1 kg/h biogas flow rate, the value of NO_x was 42.78 g/kWh and on increasing the gas flow rate, it reduced to 19.40 g/kWh. At 100% engine load and 1 kg/h biogas flow rate, NO_x was 14.40 g/kWh and on increasing the flow rate further, it reduces to 1.23 g/kWh.
- CO emission increases with a decrease in load and an increase in biogas flow into the engine. Biogas decreases the oxygen supply in the engine, which leads to higher CO emissions.
- Prediction showed that the trend of HC was similar to that of the experimental value. Increasing the engine load reduces the HC and it increases on increasing the biogas flow rate in the engine. The lower flame velocity of biogas was the reason for the increase in HC.
- The absence of aromatic compounds in the biogas decreased the smoke opacity with an increase in biogas mass flow rate.

It could be concluded that this study helps in understanding the effect of dual fuel (diesel-biogas) combustion under different load conditions of the engine with the help of ANN, which could be a substitute fuel and help to protect the environment.

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Abbreviations

Nomenclature

ANN	Artificial Neural Network
ASTM	American Society for Testing and Materials
BSEC	Brake Specific Energy Consumption
BTE	Brake Thermal Efficiency
CI	Compression Ignition
CO	Carbon Monoxide
DOE	Design of Experiments
HC	Hydrocarbon
MSE	Mean Squared Error
NO _x	Nitrogen Oxides
RSM	Response Surface Methodology
SVM	Support Vector Machines

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Article

Experimental Studies of Dust Suction Irregularity from Multi-Cyclone Dust Collector of Two-Stage Air Filter

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Abstract: The necessity of two-stage (multi-cyclone-baffle) inlet air filters used for motor vehicle engines operating under dusty conditions was demonstrated. The advantages of a set of several dozen cyclones (multi-cyclone) used for filtering inlet air in the engines were shown. The problem of dust accumulation separated by the cyclones in the multi-cyclone dust collector and the necessity of its removal on a current basis were analyzed. Methods of removing dust from the dust collector by means of the Q_S suction flow were described. It was demonstrated that the most rational method of forcing the suction flow is the use of the suction flow released from the engine. The study results are presented in this paper and show that the removal (by suction) of the accumulated dust from the dust collector additionally causes a noticeable increase in cyclone filtration efficiency. It was shown that the effective suction flow must not exceed 10–15% of the outlet flow value from the cyclone. It was evidenced that the suction of dust from the dust collector of the multi-cyclone, whose range covers several dozen cyclones, causes a differentiation in the value of the suction flows from the individual cyclones, which decreases the effect of suction and interferes with the proper operation of the cyclone. The use of a proprietary methodology, an experimental study of the irregularity of the suction from the cyclones of two multi-cyclones differing in construction, was performed. Suction flows from the cyclones, which were located at a significant distance from the main suction duct, showed 50% lower values. Possibilities of reducing the irregularity of the suction flows from the several dozen cyclones forming the multi-cyclone are presented.

Keywords: engine; air filter; multicyclone; dust suction; dust suction irregularity; air filtration efficiency and accuracy

1. Introduction

The use of trucks and special vehicles, including tracked military vehicles and working machinery, takes place mostly on unpaved, sandy, and desert terrain, where the concentration of air dust is considerable and often exceeds 1 g/m^3 . The main air pollutant sucked in by the engines of these vehicles is polydisperse mineral dust (often called road dust), which is lifted from the ground by the moving vehicles or by the wind. The primary constituents of the road dust are silica SiO_2 and corundum Al_2O_3 , whose proportion in dust reaches 95%, as well as Fe_2O_3 , MgO , and CaO . In addition, the dust contains oxides: K_2O , Na_2O , and SO_3 [1,2]. The chemical and polydisperse dust composition is influenced by many factors, mainly substrate composition and type, as well as location and height above the ground, climatic factors (winds, rains, snow, frost, droughts, etc.), as well as dust precipitation (industrial, from forest fires and volcanic ash precipitation) [3].

The polydisperse composition of dust determines the relative contribution of the individual size groups to the total dust mass.

The dust grains' residence time in the air depends on their fall velocity, which is determined by their size and density, and is a result of the mutual relationship between the aerodynamic drag force and the gravity force. Falling speed increases significantly with grain diameter. For example, SiO_2 silica grains with sizes of 10, 50, and $100 \text{ }\mu\text{m}$

(density 2650 kg/m³) fall at the following speeds: 0.08, 0.19, and 0.7 m/s, respectively [4]. This means that only dust grains with a 1 µm diameter can remain in still air all the time (practically unlimited), but their share in the total dust amount is usually small.

Due to the low falling speed, dust grains with dimensions $d_p = 2\text{--}10\text{ }\mu\text{m}$ stay in the air for a long time and are, thus, sucked by the engines. Dust grains with $d_p = 10\text{--}50\text{ }\mu\text{m}$ in the air are sucked in by the engines when the vehicle is operated under high-dust-concentration conditions in the air. In agitated dust clouds, dust grains with diameters not exceeding $d_p = 50\text{--}80\text{ }\mu\text{m}$ constitute 80–100%. Dust grains larger than 50 µm are found in the air on training grounds, quarries, construction sites, and mines [5].

The dust concentration in the air around a moving vehicle is variable and depends mainly on ground type and condition, precipitation, wind direction, vehicle driving conditions (velocity, single vehicle, or column), and types of wheels and steering system (wheeled or tracked). Therefore, dust concentrations around a moving vehicle assume variable values (Table 1).

Table 1. Dust concentrations under various air conditions.

Author	Environmental Conditions	Value [g/m ³]
[6,7]	Clean village environments	from 0.00001
	Tracked vehicles' column movement under desert conditions	about 20
[8]	Dusty environments	0.001–10
[9]	A few meters from the sandy road on which all-terrain vehicles were moving	0.05–10
[10]	On highways	0.0004–0.1
[10]	Driving vehicles' columns on sandy terrain	0.03–8
[11]	During helicopter take-off or landing on an accidental landing site at the height of the helicopter's propeller end—0.5 m above the ground	3.33
	At the inlet to the intake system of vehicle internal combustion engine	no more than 2.5
[6,8,9]	Limited visibility	0.6–0.7
[12]	Zero visibility	about 1.5

Considerable amounts of particulate matter come from the emission of dust from opencast mining and pose a serious threat to human health, especially dust with aerodynamic diameters <10 µm (PM10), which are inhaled through the respiratory tract. Particles <2.5 µm (PM 2.5) and fine particles up to 1 µm (PM1) are inhaled deeply into the lungs [13].

Dust settles by gravity on photovoltaic panels, reducing their efficiency [14]. The dust deposition phenomenon on solar mirrors significantly reduces energy production in solar power plants [15].

Mineral dust features strong abrasive properties. When it moves into the engine cylinders together with the air, it is the most frequent cause of accelerated wear in two frictionally mating parts, such as the P-PR-CW (piston—piston rings—cylinder wall) of the engine. On the ten-grade Mohs scale of hardness, where 10 is ranked as diamond hardness, silica has a hardness of 7 and corundum of 9. Furthermore, the dust grains are irregularly shaped lumps with sharp edges. Dust that is sucked in with the air moves above the piston, and this is what results in the most wear: the upper part of the cylinder, piston, and the upper piston rings. Abrasive wear of the engine components is caused mainly by particles of 1–40 µm, with the dust grains of 1–20 µm being the most dangerous [2,8,11,16]. The authors in [16] stated that circa 30% of the pollutants entering the engine may escape from the cylinders to the exhaust system in an unchanged form, together with the exhaust gases, thus increasing the emission of particulate matter (PM) from the engine. Only 10–20% of dust that enters the engine with the air through the intake system settles on the cylinder liner walls. This dust part settles on the cylinder liner, where it forms a kind of abrasive paste together with the oil, which causes abrasive

wear when it comes into contact with the surfaces of engine mating parts, for example, the piston—piston rings—cylinder wall (P-PR-CW) mating parts.

By contrast, the mineral dust grains, with a melting point much lower than the peak temperature in the combustion chamber (2000–2500 °C), melt, and, together with the exhaust gases, then enter the exhaust system and deposit on the walls of the catalytic reactor, forming a glassy surface that impairs its performance [17].

The most dangerous for the two mating parts are dust particles whose diameter d_p is equal to the oil film thickness h_{min} between the two surfaces at any given time. In an internal combustion engine, there are many mating parts lubricated with engine oil, in which the oil film thickness depends on the conditions and parameters of the engine operation and oil properties, and therefore assumes varied values within the range $h_{min} = 0\text{--}10\text{ }\mu\text{m}$ [18].

The excessive wear of the P-PR-CW mating parts is the cause of an increase in blow-offs of fresh charge into the crankcase, which results in a decrease in compression pressure and engine power, an increase in the blow-off of exhaust gases into the crankcase, which accelerates oil ageing, and the increased wear of oil penetrating above the piston, which, in turn, increases the emission of particulate matter [19–21].

For the purpose of ensuring proper air cleanliness at the inlet to truck engines, for special vehicles (tanks, armored personnel carriers, and infantry fighting vehicles) and working machinery operating under the conditions of high dust concentrations in the air, two-stage filters are used [22–25]. The first (initial) stage of filtration is then the multi-cyclone, and the second is a porous baffle usually in the form of a cylindrical insert from pleated filter paper. Figure 1 shows the two-stage (multi-cyclone-porous baffle) air filter from a Leopard 2 tank.

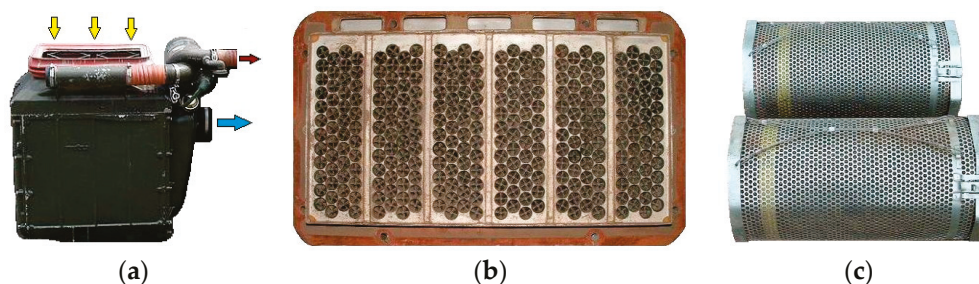


Figure 1. Two-stage (multi-cyclone-porous partition) air filter in Leopard 2 tank: (a) filter view, (b) multi-cyclone—cyclone inlets, (c) filter cartridges.

The multi-cyclone is a set of individual cyclones arranged in parallel (side by side), fixed by their ends in the common bottom and top plates, which guarantee a common air inlet and outlet. Aerosol is fed to the cyclone by a collective duct or can be sucked directly from the environment to the individual cyclones. Dust separated by the cyclones is collected in a dust collector that is common for all cyclones. The multi-cyclones may be constructed from the reverse-flow cyclones with a tangential or axial inlet or from the cyclones (Figure 2). Due to their simple and robust design, a lack of moving parts, high operational safety, stable and low pressure drop, as well as comparatively low investment and operating costs, the cyclones have been a very efficient and popular device for separating solids from the air flow in the industry for 100 years. In addition, cyclones can operate under extreme conditions, including high-gas-flow volumes, as well as high solids concentrations, temperatures, and pressures, which are their advantages over other solid–gas separation technologies.

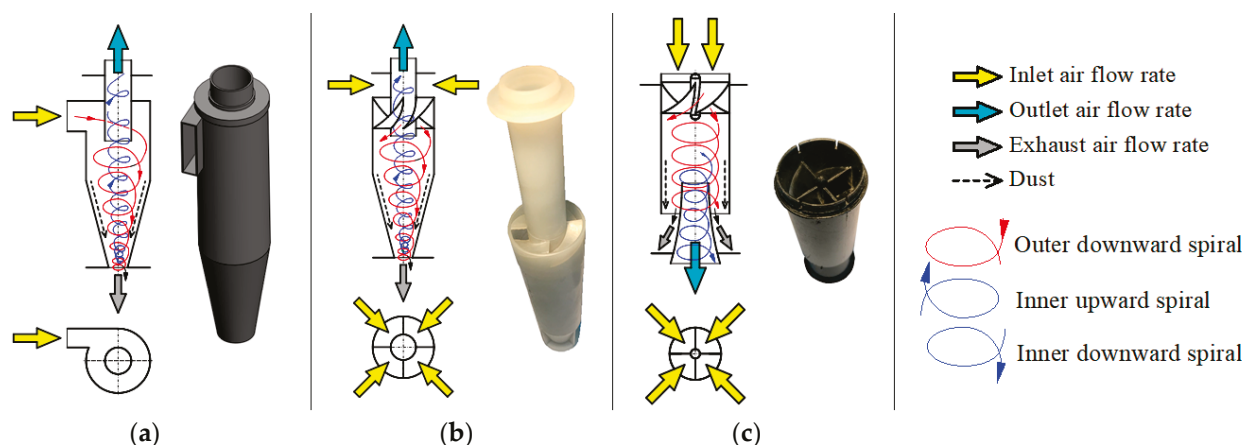


Figure 2. Cyclone types used for filtering the intake air in the motor vehicles: (a) reverse-flow cyclone, (b) reverse-flow with axial inlet, and (c) axial flow cyclone (vortex tube separators).

The tangential inlet return-flow cyclone, due to its significant advantages, is widely used in the aerosol filtration processes of many industries. Large, single cyclones with body diameters of up to several meters are used in sawmills [26], cement plants [27], oil refineries [28], coking plants, metallurgy plants, building materials plants, and cereal processing plants.

As the cyclone cylindrical part diameter D decreases, its separation efficiency increases. As the cyclone size decreases, the minimum particle size that can be separated by the cyclone decreases. Therefore, assemblies (several or a dozen) of smaller-diameter cyclones connected in parallel, instead of one large cyclone, are already widely used in industry.

For inlet air filtration in the internal combustion engines of the vehicles, agricultural and construction machinery use assemblies of several dozen or even several hundred cyclones with internal diameters not exceeding $D = 40$ mm.

Due to the principle of operation, the cyclones can be divided into reverse-flow cyclones, where the aerosol changes the direction of flow by 180° (Figure 2a,b)—clean gas and collected particles leave the device on the opposite sides; and pass-through cyclones, where the aerosol does not change the direction of flow, and then gas and particles pass through the cyclone in one direction only, being discharged from the same end of the device (Figure 2c).

Increasing the quantity of the parallel cyclones and reducing their size at the same time improves the multicyclone efficiency without changing the cross-section of the flow and without having an impact on its pressure drop, provided that the aerosol is distributed evenly to each one. This will avoid a bypass flow through the particulate outlets from one cyclone to another [29].

It is commonly accepted that the incoming stream is evenly distributed over several cyclones, which automatically causes each cyclone to work identically—with an equal gas and solids flow. As the cyclone efficiency increases with the gas volumetric flow rate, the design of the cyclone should be such as to ensure that each cyclone will operate under or near optimal operating conditions. Therefore, an uneven flow stream through adjacent cyclones causes one or more of the cyclones to operate above the optimal state and others below it, thereby reducing the overall cyclone efficiency. As a result, the operating parameters of the multi-cyclone, while maintaining the same flow characteristics on average, tend to be much worse than the operating parameters of individual cyclones, of which it is made. It is written [30] that the dust mass collected by three identical and parallel cyclones was in a ratio of 2:1.5:1, which was caused by uneven aerosol distribution for individual units. The same study reported that the parallel connection of 14 identical cyclones, each showing an efficiency of 96%, reduced the overall efficiency of this set to 92.2%. The likely cause of this efficiency decrease was that, when combined into the

assembly, one or more cyclones operated below the optimal efficiency condition, and the rest operated above. It is stated [31] that if the gas–solid flow in parallel cyclones is uneven, it reduces the overall separation efficiency. The authors in [30,32–37] investigated the unevenness of gas flow and solid particles through a set of several cyclones.

The authors in [30] investigated, using an analytical model, the gas and solid flow distribution in two identical and parallel connected cyclones, each with a diameter of $D = 101.6$ mm. By comparing the experimental data, they confirmed the consistency of the proposed analytical model. They showed that the heterogeneous gas and solids flow occurs mainly with a high load of solids (over 0.01%). Gas contamination can significantly affect the improper distribution of gas and solids through identical cyclones.

The authors in [32] studied the gas and solids flow through six parallel cyclones with a tangential inlet located asymmetrically on the left and right walls of the common tank to which the cyclone inlets were connected. The outlets of three cyclones are also connected with a common collector. The research shows that the gas and solid stream flow distribution was uneven in three cyclones on one side. The middle cyclone on each side showed higher particle velocities. The authors in [33] performed numerical calculations of gas and solid flow hydrodynamics in a multicyclone with six parallel cyclones using the Euler–Lagrangian model and computational dynamics of solid particles (CPFD). The geometry of the six cyclones was either axially symmetric or point symmetric. The cyclones' gas flow took place from a common chamber. It was shown that the solids concentrations in four cyclones located in the corners of the chamber were higher than those in others. It was found that the “axially symmetric” cyclones connection system is from the point of view of even solids distribution, more advantageous than the “symmetric-point” system. The authors in [34] investigated gases and solids hydrodynamics in six parallel-connected cyclones. The inlets of three cyclones are connected with the right, and the remaining ones with the left wall of the common chamber, creating the axial symmetry system and the central symmetry system of the cyclone arrangement. Central cyclones on both sides of the inlet chamber have been shown to have a higher solids velocity and higher efficiency than the corner cyclones. An uneven gas and solid flows distribution mainly occurs in three cyclones on one side. The authors in [35] also investigated the gas and solid flow behavior in six cyclones in parallel in a ring oven. They observed a flow irregularity between the six parallel cyclones in the ring oven.

One study [37] presented numerical tests of exhaust gas filtration, which is a product of fossil fuels combustion in a power plant, by a multicyclone consisting of four separate return cyclones with a tangential inlet with a diameter of $D = 0.889$ m and a height of $H = 2.45$ m. Cyclone inlets are connected circumferentially by a common cylindrical channel. Cyclone outlets are connected by a common cylindrical chamber. The model tests were carried out at a mass flow rate of 1 kg/s, which was approximately 30% of the total exhaust gas flow. The particles were modeled using the law of spherical resistance using the logarithmic function of Rosin Raimler. The particle density was 600 kg/m³ and the achieved separation efficiency was about 94.7%.

The multi-cyclone is characterized by the possibility of separating large masses of dust from the polluted air, without increasing the flow resistance, but with a much lower efficiency (87–95%) and accuracy ($d_p > 15\text{--}35$ μm) than those of a paper filter. The remaining insignificant mass of dust is directed to a filter cartridge most frequently made of pleated paper having a low absorption (within the range of 200–250 g/m²), where the dust grains are retained with high accuracy above $d_p = 2\text{--}5$ μm [8,38–40]. As a result, air of the required purity is supplied to the engine cylinders and the operation time of the filtration system is extended, and, thus, the service interval is extended, which is limited by the achievement of the specified value of the flow resistance—the permitted resistance Δp_{fdop} .

The dust separated by the cyclones is collected in the dust collector that is common to all cyclones, from where it must be systematically removed. The mass of dust that is retained by the cyclones and stored in the dust collector depends on the size of the air

stream, the concentration of dust in the air, and the efficiency of the cyclones. Therefore, depending on the filter type and vehicle operating conditions, the mass of dust separated by the cyclones varies. An engine of a truck tractor (articulated vehicle) with a displacement of $V_{ss} = 15.8 \text{ dm}^3$ used at an average velocity of $v = 60 \text{ km/h}$ on public roads ($s = 1 \text{ mg/m}^3$) sucks, during 50,000 km of driving, approximately 1.5 kg of dust, out of which a multi-cyclone working with efficiency $\varphi = 85\%$ retains more than 1.2 kg of dust. A cartridge with a paper surface of 2 m^2 is sufficient to stop the remaining mass of dust. This is particularly important in the case of trucks and special vehicles that are powered by the high-powered compression ignition engines and have a high (1000–6000 m^3/h) volume demand of air. For example, the T-72M tank engine draws in 3400 m^3/h from the environment, and the Leopard 2 tank engine—6000 m^3/h of air [41]. Therefore, at a dust concentration of $s = 1 \text{ g/m}^3$, the air filter multi-cyclone in the T-72M tracked vehicle engine with a displacement capacity of $V_{ss} = 38.8 \text{ dm}^3$ is able to separate about 170 kg of dust from the air sucked into the engine during 1000 km of run.

The storage of such a large dust mass in the dust collector of the multi-cyclone is not advisable, due to:

- The need to use a large-volume dust collector, which increases the dimensions of the filter and makes it difficult to place in the vehicle;
- Unnecessary loading of the filter structure with additional mass;
- dust re-suction during vehicle shocks and sudden air flow changes due to rapid engine velocity changes;
- entrainment (of already separated dust) and decrease in efficiency of the cyclones of the multicyclone in the case of complete filling of the dust collector.

Therefore, it is necessary to continuously remove dust from the dust collector in the multi-cyclones of the vehicle engine air filters. This operation is performed by the special devices: fans or blowers and ejectors, which are optional equipment of the engine.

Separated and collected dust removal (by suction) from the cyclone settling tank causes a noticeable separation efficiency increase. On the other hand, the analysis of available multi-cyclone design solutions and their dust extraction systems from the settling tank show that dust connection outlet openings of several dozen cyclones with a common sealed space (dust settler), from which the extraction takes place pointwise (with two or one stub pipe), result in a lower separation efficiency for the same cyclones, but working individually. Different suction streams and stream flowing values going through individual cyclones may then occur, which interferes with the multi-cyclone correct operation.

This phenomenon is explained by the air stream uneven distribution into individual cyclones, and even by possible reverse flows in the cyclones located on the multicyclone periphery, which causes different adjacent cyclones characteristics [42].

Even the air stream distribution problem into individual cyclones, especially in vehicle multi-cyclone air filters, where dust extraction is used, is not fully recognized and sufficiently described in the available literature. Although ejection suction is commonly used for the current dust removal from inertial filters, this problem is not accompanied by theory development that would fully explain the phenomena occurring during air flow through the dust collector. Therefore, it is advisable to work on improving the dust extraction system organization from the multi-cyclone settling tank, which will increase the multi-cyclone efficiency and, thus, reduce the flowing dust mass with the air to the second filtration stage. As a result, with a porous partition limited absorbency, the filter's service life will be extended.

This study scientific objective was to experimentally evaluate the suction stream value effect on cyclone filtration characteristics and to experimentally identify changing internal dust collector structure possibilities with a limited height for a cyclone battery, ensuring the minimization of the filtration efficiency and differentiation of the accuracy characteristics in individual cyclones in the entire multicyclone. There are no theoretical analyses and experimental studies explaining this problem in the available literature.

2. The Problem of Dust Removal from a Multi-Cyclone Settler in the Literature

Dust removal from the multi-cyclone dust collector is an operation that, in older air filter solutions, is performed by the driver during each daily maintenance operation (Figure 3a). In currently used air filter solutions, dust is removed from the settling tanks automatically:

- Through unsealing rubber drain cones (after each engine immobilization) (Figure 3b);
- In a continuous manner, thanks to the creation (using the ejection effect) of the air flow—suction flow Q_S (Figure 3c).

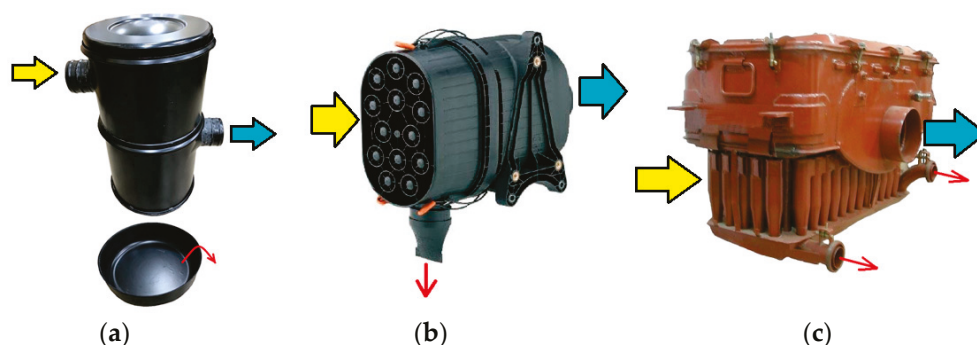


Figure 3. The manners of dust removal from the multi-cyclones: (a) manually (filter of combine harvester engine), (b) with a rubber drain cone (Mann + Hummel filter), (c) ejectively (filter of military tracked vehicle).

The dust suction is carried out by creating an air flow Q_S , which is part of the inlet (polluted) flow Q_0 to the multi-cyclone (Figure 4):

$$Q_0 = Q_G + Q_S, \quad (1)$$

where Q_G is the filter outlet (cleaned) air flow—inlet air flow to the engine.

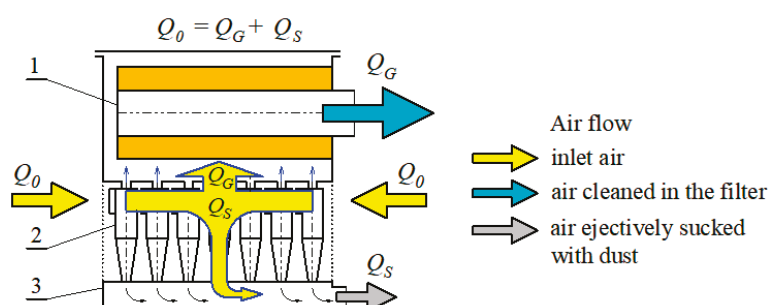


Figure 4. The division of the inlet air flow in the filter with the ejective dust suction from the dust collector: 1—filtration baffle, 2—multicyclone, 3—dust collector.

The special fans or blowers are commonly used to create the suction flow. The disadvantage of this type of equipment is the necessity of being driven, usually by an electric motor or, more rarely, by a mechanical transmission, from the crankshaft of the engine. The mechanical drive makes the characteristics of the fan directly dependent on the engine velocity, which varies over a wide range from the idle to maximum power velocity. The fan should then be in proximity of the engine, where there is usually no space for an air filter. Furthermore, a required continuity of fan operation places high demands on the durability and reliability of the drive. The electric drive of the fan eliminates the disadvantages of the mechanical drive, but is a considerable burden to the alternator that generates the electric energy for the other equipment in the vehicle. The electrically driven fan is used to create an ejector flow for the dust suction from the dust collector of

an air filter of the Leopard 2 tank and to suck dust from the dust collector of the air filter multi-cyclones of helicopters [43] (Figure 5).

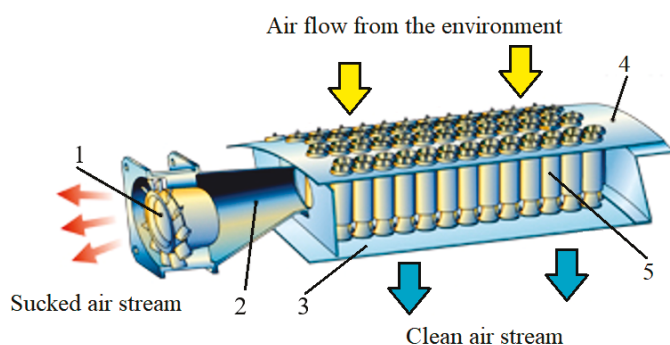


Figure 5. The system for ejective pollution suction from the air filter multi-cyclone in helicopter by means of a fan: 1—fan, 2—ejector, 3—dust collector, 4—multicyclone, 5—multicyclone housing.

For this reason, in the solutions of many air filters, the proper ejectors utilizing the energy of the compressed air flow [44–46] or the energy of exhaust gases flowing out of the engine exhaust system [47–49] are used to create the suction flow as a forcing device.

The suction flow Q_s , flowing through the dust collector, lifts the dust grains penetrating there, and then it is carried away by means of the ducts and directed outside the vehicle.

A characteristic feature of an ejector is the existence of two flows, between which momentum is transferred to each other. The flow called active has a larger value of the momentum vector and its return is strictly determined. The return of the momentum vector of the second (passive) flow is usually the same. Due to different manners of feeding the active and passive flows, there are various configurations of the ejectors (Figure 6). In each ejector, the following components are distinguished: active stream inlet duct, passive flow inlet duct, and mixing chamber. As can be noticed, significant differences occur not only in the mutual position of the flow inlet ducts, but also in their dimensional proportions.

The ejector forces or intensifies the flow of the suction flow between two spaces, which are usually the thermodynamic open systems. It must therefore be connected to these spaces by suitable ducts. The active flow is also fed to the ejector through a suitable duct from a pressurized space, which may be a closed system.

The duct of the active flow can take various forms determined by the design possibilities, but it always ends in a nozzle (convergent or convergent-divergent) or has a suitable narrowing. The shape of the passive flow duct depends mainly on the mutual positioning of the ejector and the device for which it is intended. The aim is to shape both ducts in such a manner as to minimize flow losses. The mixing chamber, or a significant part thereof, is usually in the form of a cylinder. The ejector, together with the ducts that connect it to the relevant spaces, forms the ejector system.

The authors of the paper [47] believe that the most rational method of forcing the suction flow is to eject it into the flow of exhaust gases flowing out of the engine. The ejector configurations shown in Figure 6B,C are then used in the engine exhaust system.

Forcing the suction flow through the flow of exhaust gases discharged from the engine is characterized by considerably easier technology of the ejective suction system and lower mass, which is significant in the case of the usually vibrating exhaust system. The ejection suction method is much more economical, both in terms of energy and manufacturing costs, and is characterized by high reliability.

The ejector system is characterized by a very simple construction, a low quantity of components and a lack of moving parts. It requires an inappreciable operational supervision, coming down to a periodical visual inspection of the ejector technical condition

and tightness of the pipe connecting the ejector with the inertia filter of the dust collector, whereas the use of the ejector requires an increase in the energy devoted to the suction.

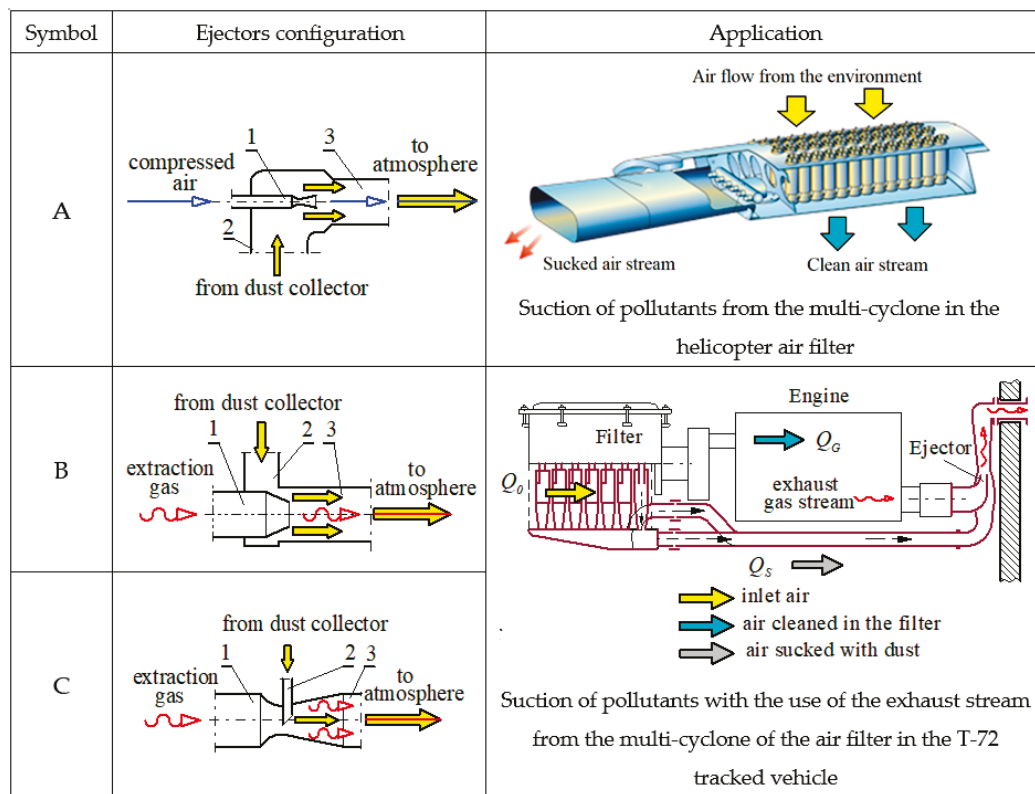


Figure 6. Ejector configurations used in suction systems to force an ejection stream: (A)—compressed air [50], (B,C)—exhaust gases: 1—active stream inlet duct, 2—passive stream inlet duct, 3—mixing chamber.

3. Experimental Studies of Dust Suction from a Settling Tank

3.1. Experimental Studies of the Influence of Dust Extraction from the Settling Tank on the Filtration Efficiency and Flow Resistance of Cyclones

The measure of the intensity of the dust suction from the dust collector of the multi-cyclone (cyclone) is the suction rate m_0 usually defined as the quotient of the flow Q_S (m^3/h) in the suction system to the size of the flow flowing from the multi-cyclone (cyclone), and in the event where the multi-cyclone is the first stage of air cleaning in a filter, to the size of the flow discharged from the air filter Q_G (m^3/h)—entering the engine [47,48,51–53]:

$$m_0 = \frac{Q_S}{Q_G} 100\%. \quad (2)$$

The author of the study [54] defined the suction rate m_0 as the quotient of the size of the suction flow Q_S to the size of the inlet flow Q_0 into the cyclone.

The removal (by means of the suction) of the accumulated dust from the multi-cyclone dust collector causes a noticeable increase in its filtration efficiency, and thus an increase (even up to 160%) in the operating time of the two-stage air filter (multi-cyclone—porous baffle) limited by the determined value of the permitted resistance Δp_{flop} [55].

The multi-cyclone efficiency increase is a smaller dust mass directed to the second filtration stage, which is usually filter paper. This results in a lower intensity of the flow resistance increase, which, with a limited and established paper dust absorption, extends the filtration system operation time.

This is confirmed by the results of research (carried out by the author) on the characteristics of the flow resistances $\Delta p_w = f(m_D)$ and the filtration efficiency $\varphi_w = f(m_D)$ of the

paper filtration baffle operating in the “cyclone—filter cartridge” system as a function of dust mass m_D sucked into the system (Figure 7).

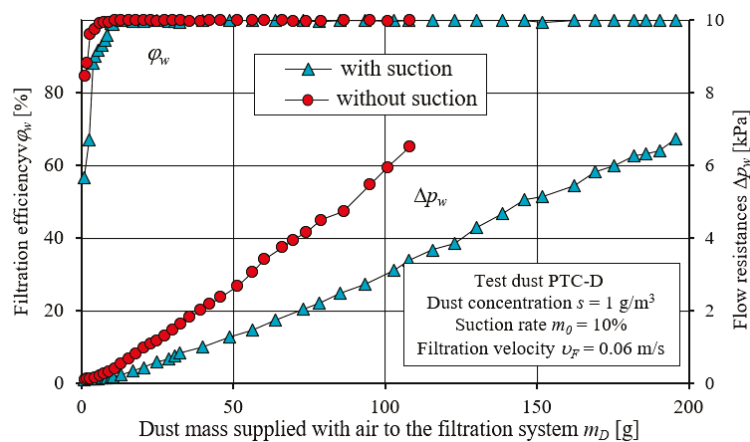


Figure 7. The change in the flow resistances $\Delta p_w = f(m_D)$ and filtration efficiency $\varphi_w = f(m_D)$ of the paper filter cartridge operating while positioned in series behind the pass-through cyclone with the ejective dust removal and without any dust removal.

The first stage of filtration was a through-flow cyclone with an axial inlet with an internal diameter of $D = 36$ mm, and the second was a cylindrical cartridge (Figure 8) positioned in series behind the cyclone, with a separation area of $A_w = 0.1534$ m². The filter material was cellulose paper with parameters given in the Table 2.

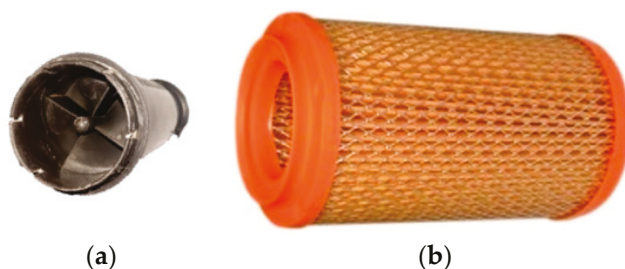


Figure 8. “Filter set” elements: (a) through cyclone, (b) cellulose cartridge.

Table 2. Cellulose filter medium parameters.

Parameters	Value
Permeability q_p (m ³ /m ² /h) at 200 Pa	3015
Basis weight g_m (g/m ²)	121
Thickness g_t (μm)	610
Por average diameter (μm)	76

The filter cartridge operating in the “cyclone—filter cartridge” system with an average efficiency $\varphi_w = 99.95\%$ reached the flow resistance $\Delta p_w = 6.52$ kPa after being fed with the air into the system of $m_D = 108$ g of dust. The same cartridge operating in the “cyclone—filter cartridge” system, but with suction, achieved a similar value of the flow resistance $\Delta p_w = 6.74$ kPa after feeding $m_D = 196$ g of dust into the “system” together with the air, i.e., nearly twice as much. The explanation of the phenomenon of the increase in the cyclone efficiency as a result of the occurrence of the suction is presented in Figure 9.

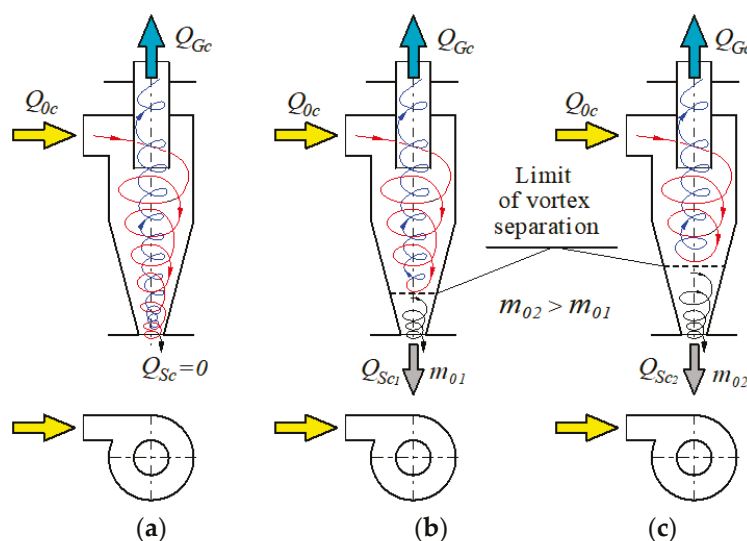


Figure 9. The flow of the air through a reverse-flow cyclone with a tangential inlet: (a) without suction and (b,c) with suction of different intensities.

The creation of the suction flow Q_S in the cyclone causes the inlet air flow Q_0 , which, with a helical motion (external vortex), moves to the bottom of the conical part of the cyclone, fails to turn back fully toward the outlet of the cyclone as a return flow—the internal vortex. In the area of the bottom of the conical part of the cyclone, a radial vortex is created from the wall to the cyclone axis and then toward the inlet opening to the dust collector. As a result, a certain part of the air flow with a Q_S value (Figure 9b) separates and flows out of the area of the conical part of the cyclone through the outlet to the dust collector, entraining the separated dust grains of the larger sizes with it, which increases the efficiency of the cyclone.

An increase in the value of the suction flow Q_S (suction rate m_0) causes the conventional limit separating the two vortices to be located at an increasingly greater distance from the outlet (Figure 9), and the area covered by the suction flow becomes increasingly larger. At the same time, there is an increase in the axial velocity of the flow in the area of the bottom of the conical part of the cyclone, which results in an increase in velocity (toward the dust collector) of the dust grains located in this zone. Thus, the numerical share of the larger dust grains in the outlet flow and their size systematically decreases, and the cyclone efficiency has an increasing value.

Figure 10 shows the stand where the characteristics of a through-flow cyclone with axial inlet were tested. The station was additionally equipped with a particle counter for recording dust grains in the air stream in the range of $d_p = 0.7\text{--}100\ \mu\text{m}$. PTC-D test dust has been used as a substitute for AC fine dust. The dust has been dosed into the dust chamber at such a flow rate as to ensure dust concentrations of $1\ \text{g}/\text{m}^3$ at the cyclone inlet. The characteristics of the cyclone have been performed within the air flow range $Q_{Gc} = 5\text{--}35\ \text{m}^3/\text{h}$ resulting from the air demand of the engine within the rotational velocity range $n = n_{\min}\text{--}n_{\max}$.

For the purpose of determining the filtration efficiency φ_c of the pass-through cyclone, the mass method has been applied by measuring the mass of dust retained by the cyclone m_{Zc} and the mass of dust m_{Dc} delivered to the tested cyclone, as well as the mass of dust retained by the absolute filters m_{A1} and m_{ZA2} within a specified time t_p . The mass of dust delivered to the cyclone and the mass of dust retained by the absolute filters have been determined as a difference between the masses of the dust container and the filters prior to and upon the measurement. The measurements have been performed for a fixed value of the air flow entering the cyclone Q_0 . Seven values of the air flow have been determined within the range $Q_0 = 5\text{--}35\ \text{m}^3/\text{h}$ with the equal intervals. For each determined value of the air flow $Q_{G,j} = 5$, the measurement cycles have been carried out, during which the

dust masses necessary to determine the cyclone filtration efficiency have been determined. For this purpose, an analytical balance with a measuring range of 220 g and an accuracy of 0.1 mg has been used.

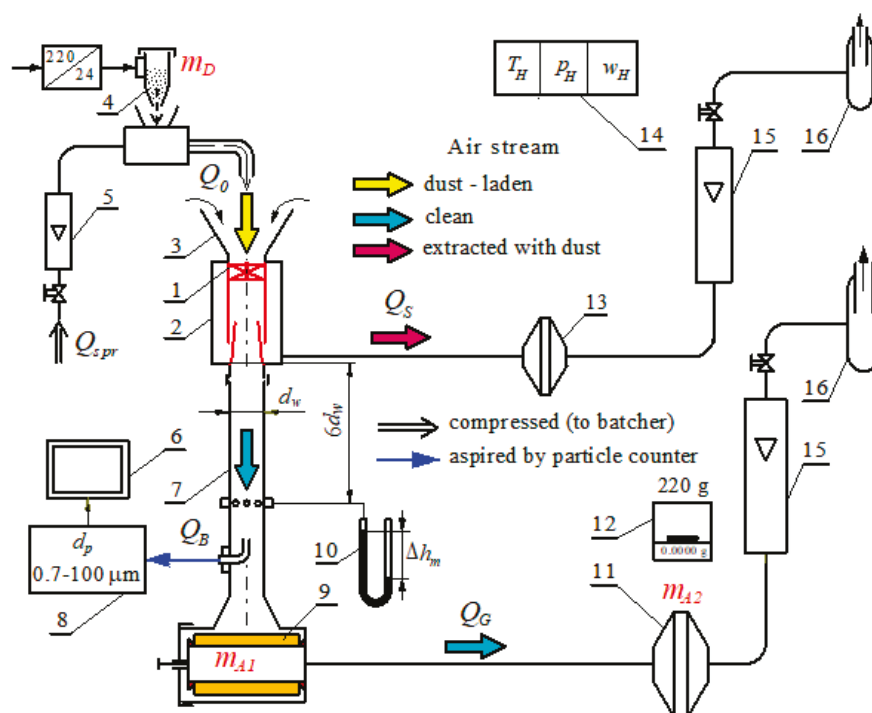


Figure 10. The functional diagram of the test stand for axial-flow cyclones: 1—tested axial flow cyclone, 2—dust collector, 3—dust chamber, 4—dust container, 5—compressed air measuring rotameter, 6—computation unit, 7—outlet pipe from cyclone, 8—particle counter, 9—measuring absolute filter A1 of main stream, 10—U-tube liquid manometer, 11—measuring absolute filter A2 of the main stream, 12—analytical balance, 13—filter securing suction flow rotameter, 14—set for measuring humidity, ambient air temperature, and pressure, 15—rotameters for measurement of main air flow Q_G and suction Q_S , 16—fans forcing air flow.

Upon each measurement cycle j , the filtration efficiency has been determined using the following formula, and then an average of the measurements has been calculated:

$$\varphi_{cj} = \frac{m_{Zcj}}{m_{DCj}} \times 100\%. \quad (3)$$

The mass of dust retained by the cyclone m_{Zc} has been calculated with the use of an indirect method from the following relationship:

$$m_{Zcj} = m_{Dcj} - (m_{A1j} + m_{A2j})[\text{g}] \quad (4)$$

The pressure drop has been determined after each measurement cycle in the tested pass-through cyclone Δp_{cj} on the basis of the measured static pressure drop Δh_{mj} in the outlet pipe behind the cyclone at a distance of $6d_w$ from the end of the cyclone outlet pipe, where d_w is the internal diameter of the cyclone outlet pipe. Δh_{mj} has been read in mm H₂O on a U-tube liquid manometer, and then the pressure drop value in Pa was determined.

On the basis of the measurements of the dust grain sizes in the air behind the cyclone, the filtration accuracy has been determined as the largest dust grain size $d_{pj} = d_{pmax}$. The

percentage share of particular fractions of the dust grains in the air behind the cyclone for a given test cycle has been determined from the formula:

$$U_{zi} = \frac{N_{zi}}{N_z} = \frac{N_{zi}}{\sum_{i=1}^{32} N_{zi}} 100\%, \quad (5)$$

where: N_{zi} —number of the dust grains in the air flow behind the cyclone (passed through the cyclone) within the measurement intervals limited by the diameters (d_{pimin}/d_{pimax}); $N_z = \sum_{i=1}^{32} N_{zi}$ —total number of the dust grains passed through the filter (from all measurement intervals) in the research cycle.

Figure 11 shows the results of the studies related to the following characteristics: efficiency $\varphi_c = f(Q_G)$ and filtration accuracy $d_{pmax} = f(Q_G)$, as well as flow resistance $\Delta p_c = f(Q_G)$ of the pass-through cyclone, which is an element of the air filter multi-cyclone of the truck engine.

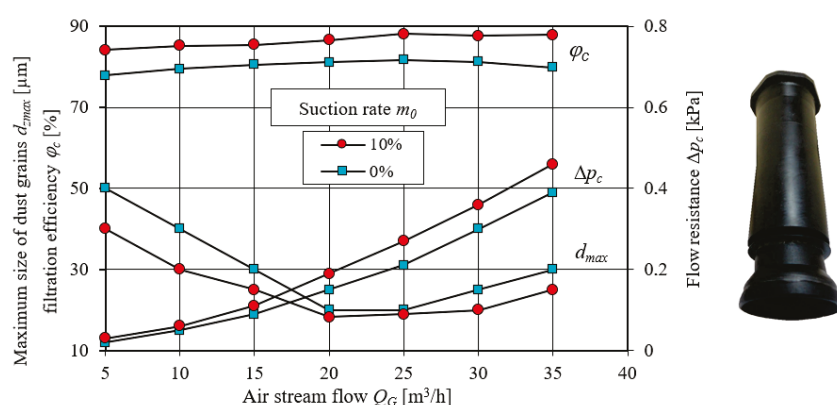


Figure 11. The characteristic of the filtration efficiency $\varphi_c = f(Q_G)$ and the flow resistances $\Delta p_c = f(Q_G)$ of the pass-through cyclone filter of the Volvo truck with dust suction ($m_0 = 10\%$) and without any dust suction.

The characteristics-related tests have been performed with the rate of ejective suction from the dust collector of the cyclone $m_0 = 10\%$ and without any suction.

With the increase in the rate of the air flow Q_G , the filtration efficiency increases, and after reaching the maximum value of $\varphi_{cmax} = 81.8\%$ for $Q_G = 25 \text{ m}^3/\text{h}$, it slightly decreases. Such a course of the cyclone characteristic $\varphi_c = f(Q_G)$ is in conformity with the results of other studies presented in the literature [56–61]. An increase in the air flow Q_G causes, at the same time, a decrease in the maximum size of the dust grains d_{pmax} (increase in the filtration accuracy of the cyclone), and upon determining the minimum value $d_{pmax} = 20 \text{ μm}$ at $Q_G = 20\text{--}25 \text{ m}^3/\text{h}$, an increase in d_{pmax} value takes place. From the course of these two characteristics, it can be noticed that the most favorable cyclone working conditions occur within the range of the air flow $Q_G = 15\text{--}30 \text{ m}^3/\text{h}$.

The ejective suction of dust at the rate of $m_0 = 10\%$ gives a characteristic curve of the filtration efficiency $\varphi_c = f(Q_G)$ of the cyclone to assume higher values in the whole range of the air flow Q_G and to differ slightly in its course. The use of the suction has resulted in an increase in the filtration efficiency of the cyclone to $\varphi_{cmax} = 88.1\%$, i.e., by 11.6%. The dust suction from the dust collector has also increased the filtration accuracy of the cyclone—a decrease in the maximum size of dust grains d_{pmax} , but mainly within the range of lower and higher values of the air flow Q_G . As the air flow Q_G increases, a parabolic increase in the flow resistances Δp_c of the cyclone results from the increasing inlet velocity v_0 .

The inlet velocity v_0 , the average velocity, is defined as the quotient of the rate of the air flow entering the cyclone Q_0 and the cross-sectional area A_0 of the inlet element at its narrowest point.

The increase in the flow resistance of the cyclone with the suction is more intense. For the same rate of the air flow Q_G , an increase in the suction flow rate Q_S increases

the inlet air flow rate Q_0 according to the following formula. The inlet velocity v_0 thus assumes the larger values.

$$Q_0 = Q_G + Q_S. \quad (6)$$

Figure 12 presents changes in the cyclone filtration efficiency and accuracy at a constant air flow Q_G as a function of the suction rate m_0 . An increase in the rate of suction to 12–16% resulted in an intensive increase in efficiency and a decrease in the maximum size of dust grains d_{pmax} . With a further increase in m_0 , the increases in the filtration efficiency and the cyclone accuracy were not as intensive. An increase in the suction rate $m_0 = 18\%$ resulted in a 10% increase in efficiency and a decrease in the value of the maximum size of the dust grains from 25 to $15.8 \mu\text{m}$.

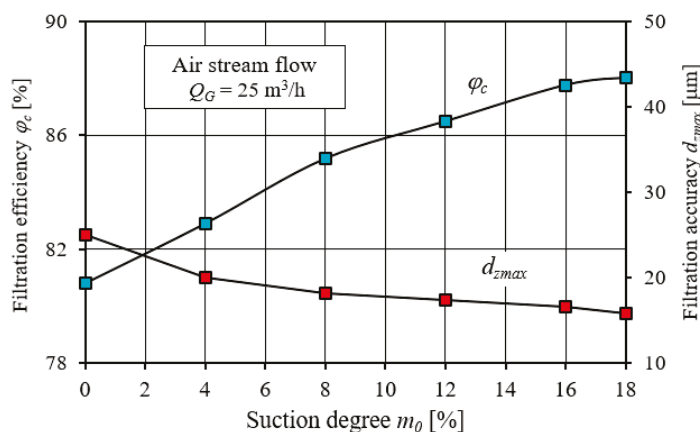


Figure 12. The impact of the suction rate m_0 on the filtration efficiency and accuracy of the pass-through cyclone.

The phenomenon of the dust suction has a significant influence on the fractional composition of dust in the cyclone outlet air. It can be noticed in Figure 13 that at a constant value of the air flow Q_G in the cyclone, the higher the value of the suction flow (suction rate m_0), the greater the share of the dust grains below $3 \mu\text{m}$ in the cleaned air. On the other hand, the proportion of grains above $3 \mu\text{m}$ decreases. This indicates that an increase in the suction rate results in increasingly smaller dust grains being removed from the air.

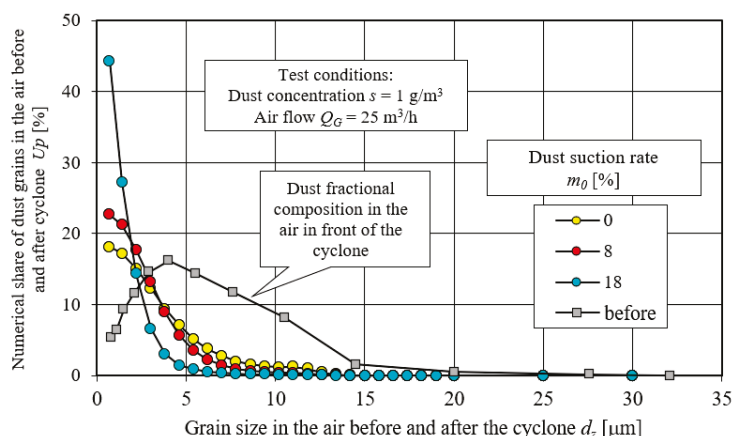


Figure 13. The impact of the suction rate m_0 on the numerical share U_p of the dust grains in the outlet air from the pass-through cyclone before.

The results of similar experimental studies on the tangential inlet reverse-flow cyclone for the various rates of the ejective suction are shown in Figure 14 [62]. The characteristics of the filtration efficiency $\varphi_c = f(Q_G)$ and flow resistance $\Delta p_c = f(Q_G)$ is

shifted almost parallel to higher values with the increase in the suction rate m_0 . A significant (5%) increase in the filtration efficiency and an increase in the flow resistance (11%) of the tangential inlet return-flow cyclone occurs when changing the suction rate m_0 within the range of 0–8% (Figure 14). An increase in the suction rate m_0 by a further 8% results in a minor (1%) increase in the filtration efficiency of the cyclone, but a significant (13%) increase in the flow resistance.

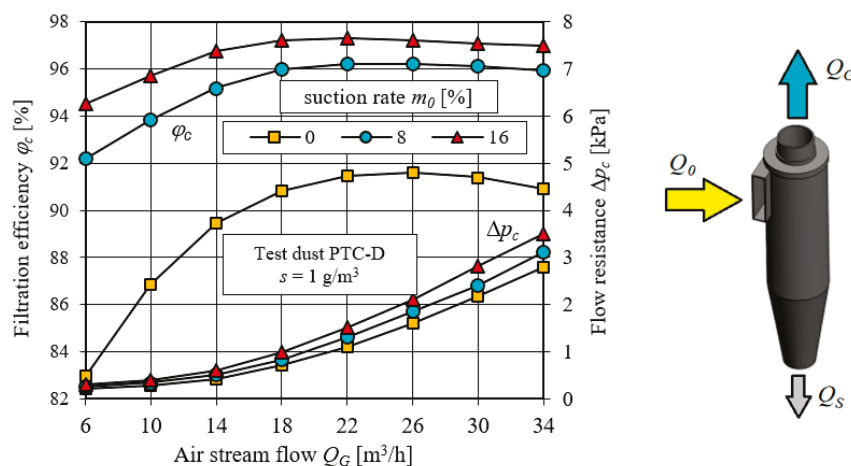


Figure 14. The impact of the suction rate m_0 on the efficiency characteristics $\phi_c = f(Q_{Gc})$ and the flow resistance. $\Delta p_c = f(Q_{Gc})$ of the reverse-flow cyclone with the tangential inlet D-40.

Due to the high filtration efficiency and accuracy, the cyclones should operate within a certain range of air flows and at a rate of the ejective suction m_0 not exceeding an optimum value. This is the value beyond which the increase in the filtration efficiency of the cyclone is no longer intensive, but there is a significant increase in the flow resistance. The optimum value of the suction rate m_{0opt} must be determined experimentally for each type of the cyclone.

During the tests of the return-flow cyclone with a tangential inlet in for a constant outlet flow $Q_G = 80 \text{ dm}^3/\text{h}$ and with the increasing suction rate within the range $m_0 = 4.8\text{--}20\%$, a decrease in the diameter of the limit grain from 1.74 to $1.05 \mu\text{m}$ and an increase in the flow resistance of the cyclone within the range between 522 and 821 Pa have been obtained [54].

3.2. Experimental Tests of Uneven Dust Suction from the Settling Tank

3.2.1. Research Problem

The analysis of available construction solutions related to the multi-cyclones and their dust suction systems from the dust collector indicates that if the dust collector covers several dozen of the cyclones and the dust suction from the dust collector is performed by two or one stub-pipes, then [29,42]:

- A differentiation of the value of the suction flows from the particular cyclones takes place;
- A mutual interaction of the swirled flows flowing from the cyclones and flowing to the common compartment of the dust collector occurs;
- There is a possibility of the return flow in the cyclones located on the periphery of the multi-cyclone.

The aforementioned phenomena may be the cause of the lower filtration efficiency of the multi-cyclone than what would result from the applied suction rate for the individual cyclones. Moreover, with the amount of the dust mass supplied with the air to the multi-cyclone and the occurrence of the said phenomena:

- (1) A permanent deposition of dust on the outlet elements of the cyclone, which blocks the outflow of the sucked dust. As the working time of the multi-cyclone expires,

the mass of dust deposits increases, which results in an obstructed outflow of the suction flow from the cyclone and may cause it to be put out of operation.

- (2) A permanent settling of dust in the corners and around the edges of the dust collector, from where the suction is less effective than from the cyclones in close proximity to the suction duct. The accumulated and deposited dust at the bottom of the dust collector causes difficulties in the outflow of both sucked dust and cleaned air from the cyclone. As a result, the air flowing through these cyclones assumes increasingly lesser values, which results in a decrease in the efficiency. As the working time of the multi-cyclone expires, the mass of the deposited dust on the bottom of the dust collector and on its periphery is increasingly larger, which, in effect, causes blocking of the outlets from these cyclones and their putting out of operation.

Figures 15–17 show the outlet slots of the cyclones before the multi-cyclone has started its operation, after 500 g of dust and 2150 g of dust have been fed to the multi-cyclone. It can be noticed that in spite of the use of the dust suction, dust accumulation occurs and the outlets of the cyclone dust suction slots are blocked. More information on this subject is provided by the author in [63].

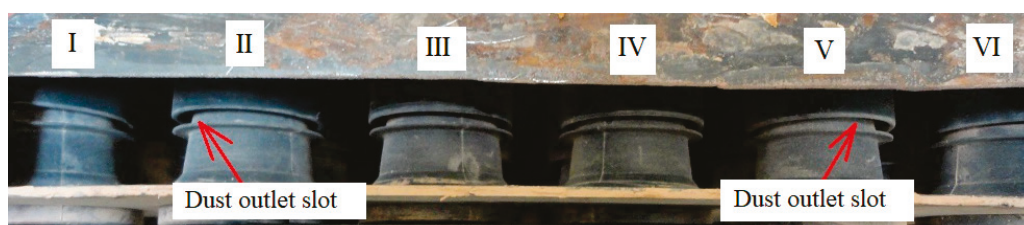


Figure 15. The view of the multi-cyclone from the dust suction collector. Visible outlet slots of the cyclones before starting operation by the multi-cyclone (I, II, III, ... VI—cyclone serial number in the column).

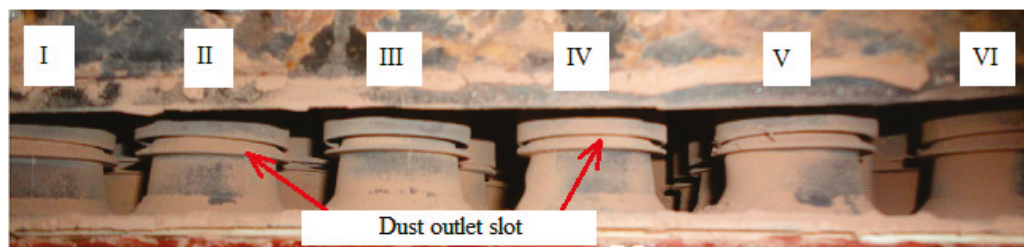


Figure 16. The cyclone outlets of the multi-cyclone after a supply of 500 g of dust into the multicyclone.

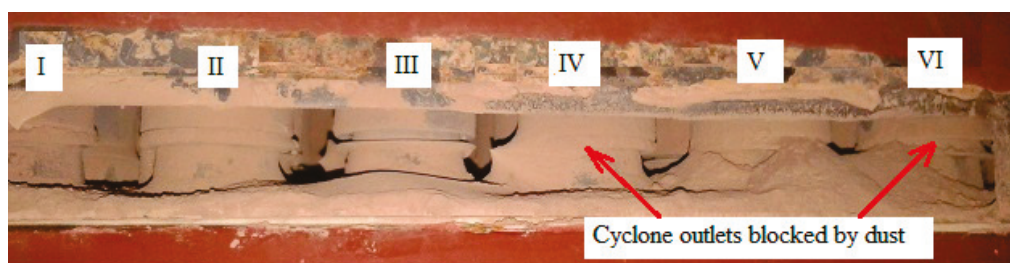


Figure 17. The cyclone outlets of the multi-cyclone after a supply of 2150 g of dust into the multicyclone.

In those cyclones where the suction rate m_0 assumes a low value, the resulting increase in the efficiency is insignificant. On the other hand, the cyclones where the filtration is carried out at a much higher suction rate value m_0 than the optimum value do not achieve proportionally higher efficiency values, and an increase in the flow resistance may occur. The quantity of the cyclones in which the filtration process takes place at the optimum value of the suction rate m_0 is low. This causes the

operating parameters of the multi-cyclone, while maintaining on average the same flow characteristics and the same value of the suction rate, to be significantly worse than the operating parameters of the individual cyclones of which it is composed. The reason for this unfavorable phenomenon is the irregularity of the value of the occurrence of the suction flow from the individual cyclones.

3.2.2. Purpose and Subject of Research

In order to recognize this phenomenon, the experimental study has been conducted. The study aimed to determine the impact of the construction of the multi-cyclone and its dust collector on the values of the sucked flows Q_{Sc} from the individual cyclones comprising the multi-cyclone.

The subject of study has been two multi-cyclones of the air filters of the engines of two special vehicles with the conventional designations T and B. A schematic diagram of the system for supplying air to the engine of the T vehicle and the system for ejective suction of the pollutants from the dust collector of the multi-cyclone, which has been given the working name MT, is shown in Figure 18.

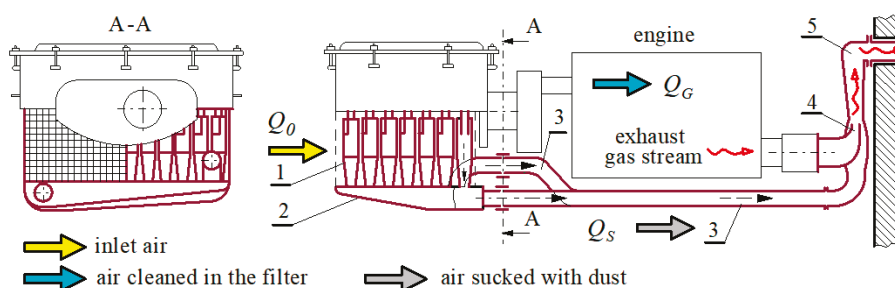


Figure 18. The system of ejective dust suction using the suction flow from the air filter multicyclone of the T-72 tracked vehicle: 1—multicyclone, 2—dust collector, 3—suction ducts, 4—ejectors, 5—exhaust gas pipe.

In this vehicle, air cleanliness is provided by a two-stage filter (Figure 19) operating in the “multi-cyclone—porous baffle” series system. The MT multi-cyclone is constructed of the vertically positioned tangential inlet return-flow cyclones D-40 with an inner diameter of the cylindrical part $D = 40$ mm, the quantity of which is 96 (Figure 20a). The dust suction from the multi-cyclone dust collector is performed by a special system, the basic element of which is an ejector positioned in the exhaust gas flow outlet pipe. The second stage of the filtration is an engine oil-moistened filtration bed in the form of three (bottom, middle, and top) serially arranged cassettes filled with an irregularly pressed steel wire bed of diameter $d_s = 0.25$ mm.

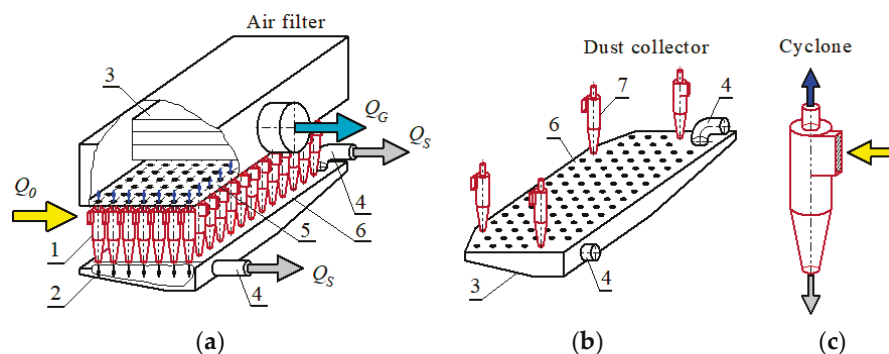


Figure 19. The elements of the air filter of the tracked vehicle T: (a) air filter, (b) dust collector, (c) return-flow cyclone with tangential inlet, 1—multicyclone, 2—dust collector, 3—filter cartridge (cassettes), 4—dust suction ducts, 5—upper fixing plate of cyclones, 6—lower fixing plate (collector upper wall) of collectors, 7—cyclone.

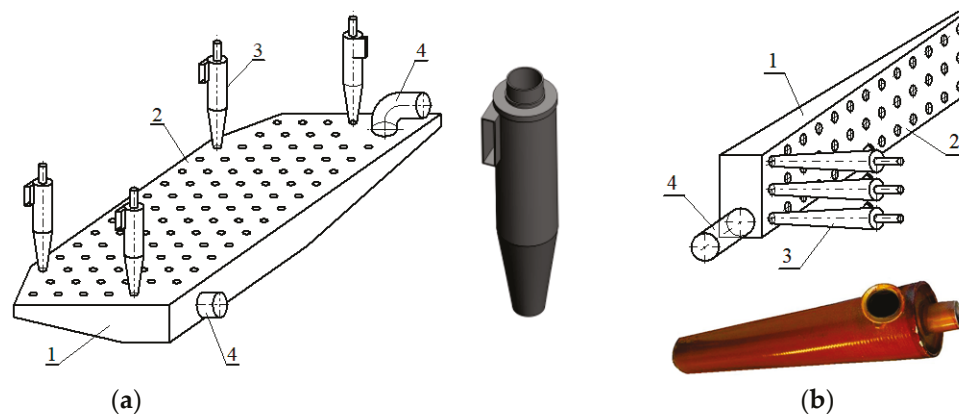


Figure 20. The construction of the dust collectors: (a) M72 multicyclone, (b) MB multicyclone, 1—dust collector chamber, 2—cyclone mounting plate, 3—cyclone, 4—suction stub-pipes.

In tracked vehicle B, air cleanness is ensured only by a single-stage filter. This is a multi-cyclone (working name: MB) built from 39 horizontally aligned return-flow cyclones with a tangential inlet having an internal diameter at its top of $D = 35$ mm.

The general construction of the multi-cyclones selected for testing is similar. On a rectangular plate, which is the upper wall of the dust collector, the cyclones are fixed, arranged in rows and columns (Figure 20).

The size of the mounting plate is determined by the diameter and quantity of the cyclones. The outlet pipes of the cyclones (cleaned air) are installed in a common upper plate, which is the base of the baffle filter cartridge. The multi-cyclones constructed in this manner are the sealed devices.

The dust collectors of the tested multi-cyclones are hollow inside ducts of a shape similar to a flat cuboid. The dust suction from the MT dust collector is realized by two stub-pipes, and from the MB dust collector by a single stub-pipe. The cyclones in relation to the outlet of the suction stub-pipe are placed at the different distances (Figure 20).

The cyclone furthest from the outlet of the suction stub-pipe is located at a distance of 5 to 8 times greater than the cyclone closest to the outlet. This undoubtedly has an impact on the value of the suction flow from the individual cyclones. Such design solutions of the dust collectors and dust discharge systems result from the limited space allocated for the installation of the air filter in the vehicle.

3.2.3. Methodology of Research on Suction Uniformity

The values of the sucked flows Q_{Sc} from the individual cyclones of the air filter multi-cyclone have been determined by an indirect method consisting of a measurement of the value of the inlet flow Q_{0c} to the cyclone, which, for the case where the cyclone outlets flow $Q_{Gc} = 0$, assumes the value $Q_{0c} = Q_{Sc}$. Such an arrangement will occur when the outlet from all cyclones of the multi-cyclone is covered (Figure 21), and a flow Q_{SF} is sucked from the dust collector of the multi-cyclone, which is the sum of the sucked flows Q_{Sc} from the individual cyclones according to the following formula.

$$Q_{SF} = \sum_{j=1}^J \sum_{k=1}^K Q_{Sc_{jk}}, \quad (7)$$

where I, II, III, ..., j , ..., J is a cyclone sequence number in the column—the number of the rows; and 1, 2, 3, ..., k , ..., K is a cyclone sequence number in the row—the number of the columns.

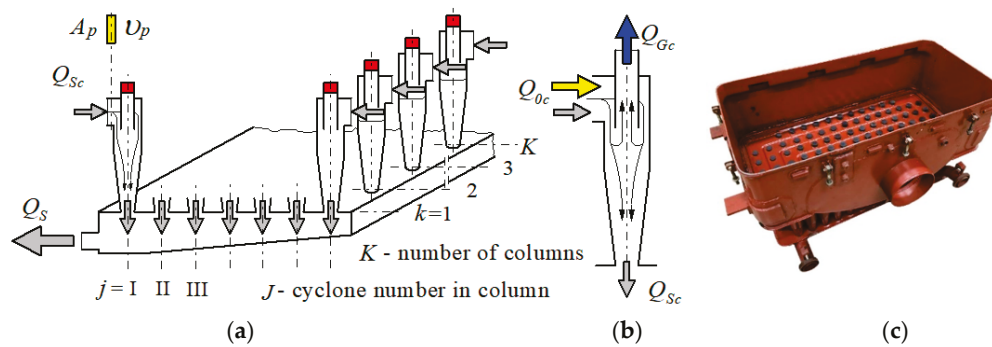


Figure 21. The flow Q_{Sc} in the cyclone: (a) during a measurement (I, II, III, ... j ,... J is a cyclone serial number in the column—the number of the rows; 1, 2, 3, ... k ... K is the cyclone serial number in the row—the number of the columns), (b) during operation, (c) a view of the multicyclone with the blocked cyclone outlets.

The scope of the study has covered the determination of the values of the sucked flows Q_{Sc} only from the individual outermost cyclones of the MT and MB multi-cyclones, as access to the cyclones positioned in the middle is not possible. The multi-cyclones differ in design and size to what results from the quantity of the cyclones used.

The studies of the multi-cyclones have been carried out on a specially built test stand. The values of the flows Q_{Sc} from the individual cyclones have been determined while applying an indirect method consisting of the measurement with a vane probe (generating minor flow resistance) of the air flow velocity v_0 in the inlet stub-pipe of the cyclone subjected to the study, the cross-sectional area of which, at its narrowest, equals A_p . The stand for the measurement of the suction flows from the dust collector of the MT multi-cyclone is presented in Figure 22. The role of the ejector, consisting of generating an appropriate suction flow Q_S , has been fulfilled by the suction fan.

The stand has been equipped with a special measuring system (Figure 23), where the Testo 400 vane probe for the measurements of the air flow velocity has been the basic element. The probe has been placed in cylindrical measuring duct 5 with a diameter d_c , equal to the diameter of the vane of the probe. The inflow of air into the probe is carried out using a lemniscate. In the case of the measurement of the value of the sucked flows Q_{Sc} from the cyclones of the MT multi-cyclone, the outlet stub-pipe of measuring duct 5 has been symmetrically installed in the inlet part of connecting duct 4 with the cross-section of a rectangle A_p —Figure 22. The outlet of the connecting duct has been shaped to match the rectangular cross-section A_0 of an opening of inlet stub-pipe 3 of the cyclone.

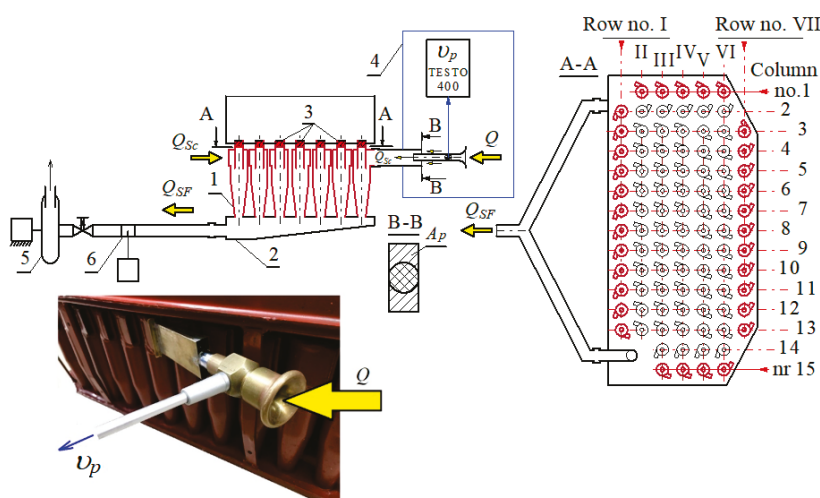


Figure 22. The stand for measurement of the sucked flows from the individual cyclones of the MT multicyclone: 1—multicyclone, 2—dust collector, 3—flow closing plugs, 4—measurement kit, 5—suction fan, 6—sucked flow flowmeter Q_{SF} .

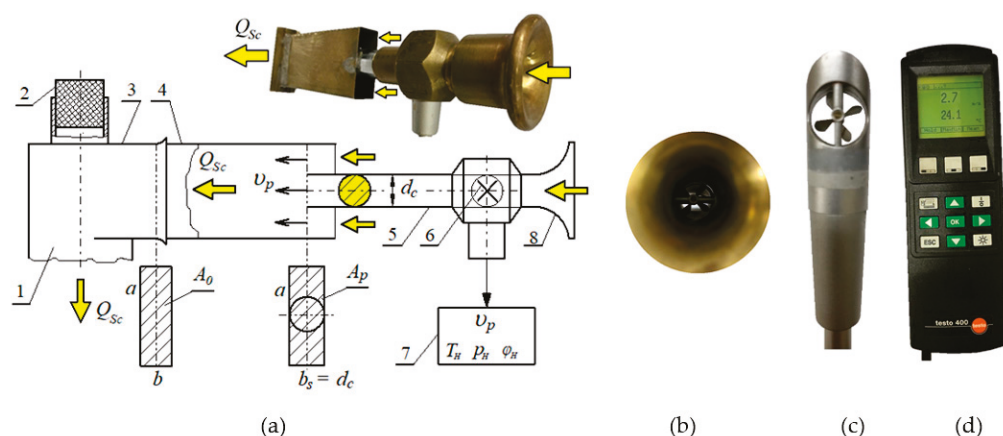


Figure 23. The system for measurement of the sucked flows Q_{SF} from the individual cyclones of the MT multicyclone: (a) functional diagram (1—cylindrical part of cyclone body, 2—plug closing cyclone outlet pipe, 3—cyclone inlet stub-pipe, 4—connecting duct, 5—measuring duct, 6—vane probe, 7—readout panel of velocity value, 8—inlet lemniscate), (b) view of vane probe in measuring duct, (c) vane probe, and (d) reading panel of measured parameters.

The studies have been carried out for the conditions where the main suction flow Q_{SF} flows successively through the duct collector of the multi-cyclone, with the value resulting from the suction rate $m_0 = 4, 8$ and 16% and the air demand by the engine Q_{Silmax} at the rotational velocity of the maximum power.

The values of the extraction flows Q_{SF} from the dust collector of the multi-cyclone, for the suction degrees $m_0 = 4, 8$ and 16% assumed for the study, have been calculated from the formula:

$$Q_{SF} = m_0 Q_{GF}, \quad (8)$$

where: Q_{GF} —the air flow flowing from the multi-cyclone (air filter) equal to the air demand by the engine Q_{Silmax} at the rotational velocity of the maximum power n_N .

The maximum air demand by the engine Q_{Silmax} can be determined experimentally under the laboratory conditions. This is the most reliable method but is also an expensive and time-consuming one. For the comparative analyses conducted as a part of this paper, the maximum air demand by the engine $Q_{Silmax} = Q_{GF}$ has been determined using the theoretical relation [64]:

$$Q_{Silmax} = Q_{GF} = \frac{V_{ss} \cdot n_N \cdot \eta_v \cdot 60}{1000 \cdot \kappa} \left[\text{m}^3/\text{h} \right], \quad (9)$$

where: V_{ss} —engine displacement (dm^3), n_N —rotational velocity of maximum power (rpm), κ —stroke number coefficient (2—for four-stroke engines, 1—for two-stroke engines), η_v —filling factor, the value of which depends on the engine type.

According to the data of the Donaldson Company, Inc., when calculating the air demand of the engine, the filling factor η of the engine cylinders can be assumed as follows [65]:

- 0.85—four-stroke naturally aspirated engines with compression ignition and spark ignition,
- 1.60—four-stroke turbocharged engines with compression ignition,
- 1.85—four-stroke turbocharged engines with compression ignition and intake air cooler.

Calculated according to the aforementioned dependence, the air demand of the engine Q_{Silmax} is equal to the air flow flowing out of the air filter—from the multicyclone. For MT and MB multi-cyclones, the air flow assumes the values: $Q_{GF} = 3400$ and $1250 \text{ m}^3/\text{h}$, respectively. Then, the values of the suction flows Q_{SF} according to formula 4.2 for the ejective suction rates $m_0 = 4, 8$ and 16% assume the values: $Q_{SF} = 135, 270$, and $540 \text{ m}^3/\text{h}$ (for MT multi-cyclone) and $Q_{SF} = 50, 100$, and $200 \text{ m}^3/\text{h}$ (for MB multi-cyclone), respectively.

For the determined values of the main suction flow Q_{SF} , the measuring system has been successively attached to the inlet stub-pipes of the tested cyclones. For each cyclone, at 10 s intervals, seven values of the air velocity v_{pjk} have been recorded, from which an average $\bar{v}_{pjk}$ has been determined. Assuming that the average air velocity in the A_p cross-sectional plane of the connecting duct 4 and at the outlet of the measuring stub-pipe 5 assumes the same value as the measured velocity v_{pjk} in the cylindrical measuring duct, the value of the flow flowing through the stub-pipe has been determined from the formula:

$$Q_{Scjk} = \bar{v}_{pjk} \cdot A_p \quad (10)$$

where: A_p —cross-sectional surface area of the duct connecting the cyclone stub-pipe and measurement duct of the probe.

The developed research methodology enables the measurement of the sucked flows Q_{Sc} from the cyclones of the MT multicyclone only for the cyclones located in the outer rows and columns, and for the MB multicyclone only from the upper column. This is due to the sealed construction of the multi-cyclone and the lack of access to the inlet stub-pipes of the other cyclones.

3.2.4. Analysis of Studies of Irregular Suction from Multi-Cyclone

The flows sucked Q_{Sc} from the individual cyclones of the MT multi-cyclone assume diverse values (Figures 24–26). The highest values of the sucked flows within the range $Q_{Sc} = 2.5\text{--}5.95 \text{ m}^3/\text{h}$ independently of the value of the main sucked flow Q_{SF} are assumed by the cyclones, whose outlets of the sucked flows are located near the stub-pipes from the dust collector of the multi-cyclone (Figure 25). The sucked flows Q_{Sc} from the middle cyclones of the tested row of the multi-cyclone, located at a certain distance from the suction stub-pipes, assume values within the range $Q_{Sc} = 1.05\text{--}2.95 \text{ m}^3/\text{h}$, over 50% lower.

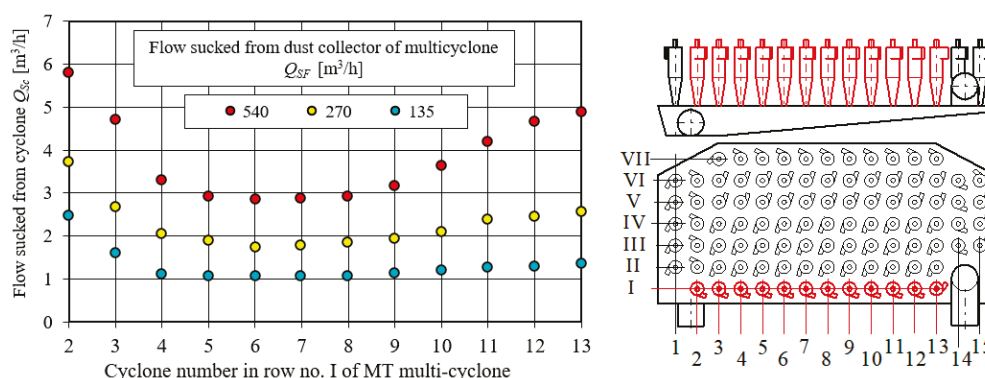


Figure 24. The values of the sucked flows Q_{Sc} from the individual cyclones of the first (I) row of the MT multicyclone.

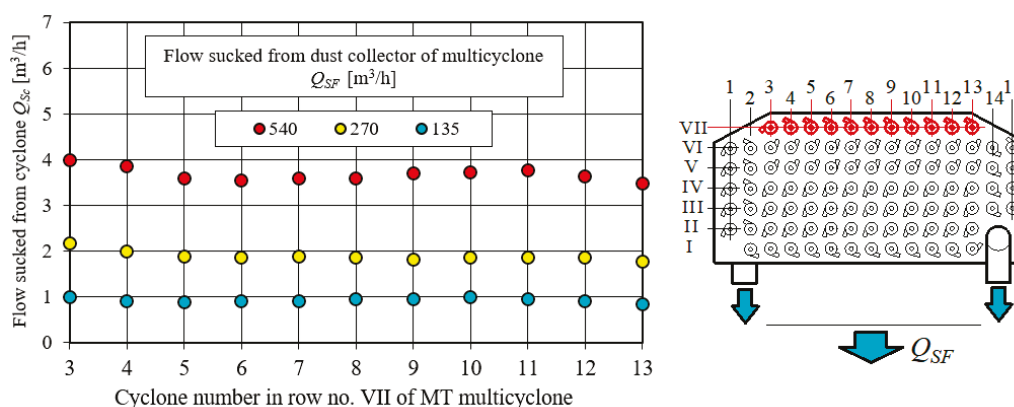


Figure 25. The values of the sucked flows Q_{Sc} from the cyclones of the seventh (VII) row of the MT multicyclone.

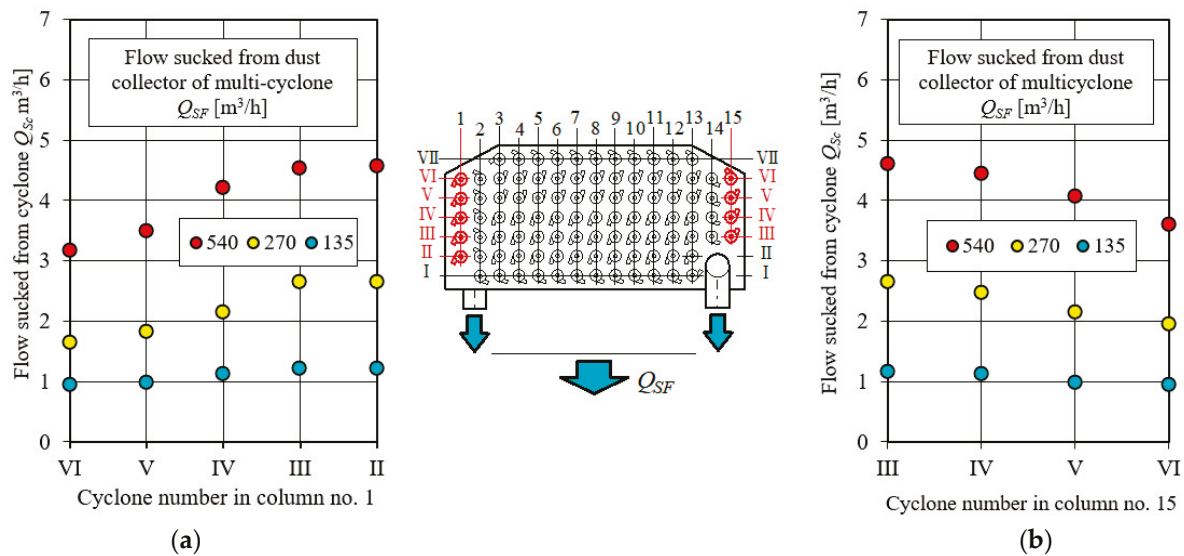


Figure 26. The values of the sucked flows Q_{sc} from the cyclones of the MT multi-cyclone: (a) columns no. 1, (b) columns no. 15.

In the last (VII) row of the cyclones, the sucked flows from the individual cyclones assume values lower than half (Figure 26) those in the first row (I) of the multi-cyclone. The differences between maximum and minimum values of the sucked flows Q_{sc} for the successive main sucked flows $Q_{SF} = 540$, 270, and 135 m³/h assume values of 11%, 14%, and 16%, respectively.

As the cyclones are positioned further from the front part of the dust collector (from the suction stub-pipe) toward the rear wall of the multi-cyclone, there is a systematic decrease in the value of the sucked flows Q_{sc} (Figure 27).

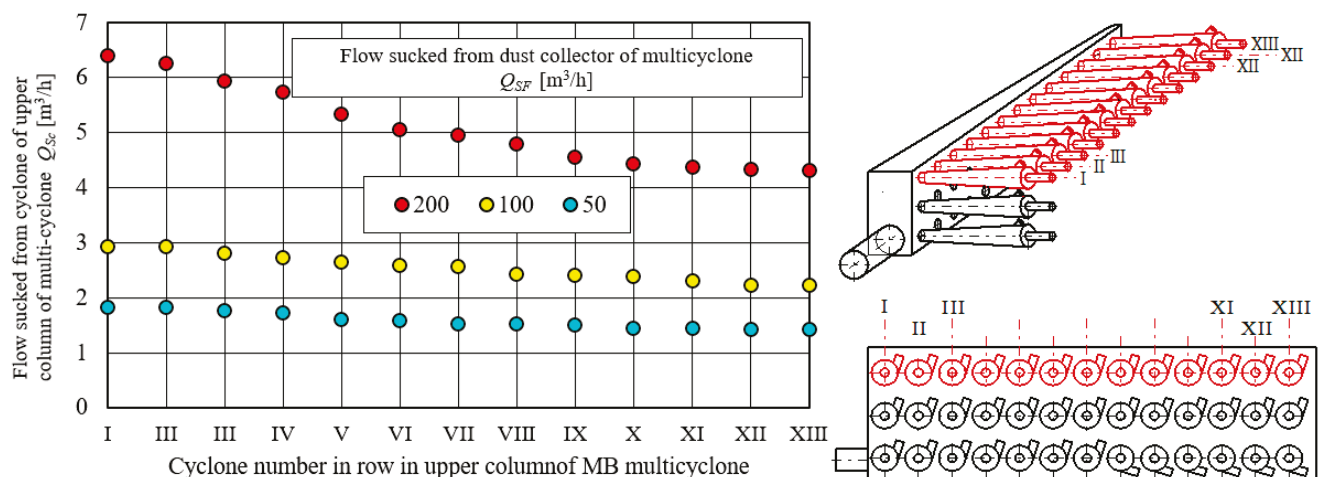


Figure 27. The values of the sucked flows Q_{sc} from the individual cyclones of the upper column of the MT multi-cyclone.

The highest values of the sucked flows Q_{sc} occur for the cyclones located closest to the stub-pipe for the dust suction from the dust collector. As the position of the subsequent cyclones moves farther from the dust suction stub-pipe, the values of the sucked flows Q_{sc} systematically decrease, and for the cyclones positioned the farthest, the sucked flows Q_{sc} assume values 35–50% lower. Therefore, the achieved filtration efficiency of the cyclones may be lower.

In the event of the studies of the value of the studied flows Q_{sc} from the cyclones of the MB multi-cyclone, the measurements have been made only for the cyclones located

in the upper column of the multi-cyclone, as getting access to the inlet sub-pipes of the cyclones of the middle and lower columns of the multi-cyclone is impossible.

The highest values of the sucked flows Q_{Sc} from the cyclones of the upper column of the MB multi-cyclone occur, independently of the value of the main sucked flow Q_{SF} , for the cyclones positioned closest to the outlet of the dust collector outlet of the multi-cyclone, and they are within the range $Q_{Sc} = 1.83\text{--}6.4 \text{ m}^3/\text{h}$ (Figure 27).

Along with the distance of the position of the cyclone from the suction stub-pipe, the values of the sucked flows Q_{Sc} systematically decrease, and for the last cyclone no. XIII, they assume the values within the range $Q_{Sc} = 1.42\text{--}4.32 \text{ m}^3/\text{h}$. This indubitably has an impact on the efficiency of the aerosol filtration in the cyclones.

3.2.5. Possibilities of Organizing Process of Suction from Multi-Cyclones of Air Filters of Motor Vehicles

As past experiences have proven, the main reason for the irregularity of the suction flows from the cyclones may be the variation in the flow resistance in respect of these flows in the dust collector of the multi-cyclone, at the suction cyclone—suction stub-pipe, for the individual cyclones. It results most of all from the nonuniform lengths of the said sections, determined by the location of the cyclone in relation to the suction stub-pipe and the impact of the dust collector walls on the flows sucked from the cyclones positioned in their vicinity, as well as the possible occurrence of the turbulence areas in the total aggregate flow sucked in the dust collector.

There are three basic methods leading to a relative stream uniformity sucked from individual cyclones [44,47,51]:

- Maintaining the symmetry of the position this manner consists of, making the construction of the dust collector so that the symmetry of the position of all cyclones in relation to the suction stub-pipe is maintained;
- Creating an equal flow resistance of the sucked flows by the purposeful, appropriately selected throttling from those cyclones for which the flow resistance is lower than for the remaining ones;
- A combined method—which is an appropriate constructional combination of the two aforementioned procedures.

The only possibility to achieve a symmetry of position is to place all cyclones along the circumference of the wheel. This leaves an unused area in the central part of the plate, with a greater number of cyclones in the multi-cyclone—Figure 28.

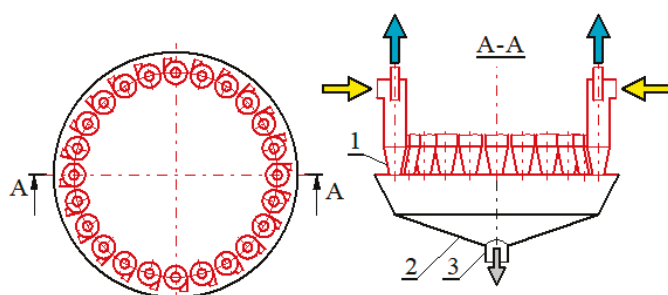


Figure 28. The diagram of dust collector with symmetrical position of all cyclones in relation to the suction stub-pipe: 1—cyclone, 2—dust collector, 3—sub-pipe of suction duct.

A consequence of the symmetrical construction of the dust collector with a large number of the cyclones is the disproportionately large dimensions of the plate of the dust collector on which the cyclones are installed. In the engine chambers of vehicles, there is usually no excess of free space, hence the minor practical usefulness of this type of construction of the multicyclone, except for the units with a low quantity, of the order of a dozen or so cyclones.

The equalization of the flow resistances of the value of the suction flows from the individual cyclones can be obtained by the method consisting of the change in the design of the dust collector of the multicyclone, by dividing the space of the dust collector with several vertical baffles into segments of width a , and the segments into independent, isolated-from-each-other suction ducts, which should be assigned to the specific groups of the individual cyclones—Figure 29. The heights of the ducts h_1 , h_2 , and h_3 , developed by the baffles in the dust collector, should be selected in terms of obtaining the equal flow resistances through the ducts.

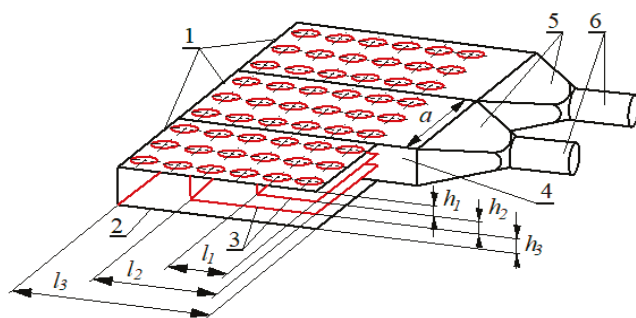


Figure 29. The schematic diagram of a division of the dust collector into the independent segments and ducts: a —width of segment (duct), l_1 , l_2 , and l_3 —length of ducts, 1—segments, 2—bottom of dust collector, 3—horizontal baffles, 4—vertical baffle, 5—through-pass duct of segment, 6—suction stub-pipe of segment.

Each of the individual suction ducts in the duct collector should include a group of the cyclones that is not too numerous. The cyclones may be positioned along the width a of the duct 3...4 and along the length $l_1 \dots 4$, depending on the height of the duct in the location where the suction from the cyclones takes place. If there are more cyclones in the width of the multi-cyclone, the dust collector must be divided (Figure 29) by a tight vertical wall 4 so that the segments developed by the division of the width fit the required quantity of the cyclones. A separate suction stub-pipe should be assigned to each segment. This manner of shaping the dust collector is particularly suitable for modifying the dust collectors in the existing, operated filters, which do not provide the expected dedusting efficiency.

4. Conclusions

The paper presents an experimental evaluation of the influence of suction streams from a multi-cyclone settling tank on the separation efficiency and flow resistance characteristics of a single cyclone, as well as an experimental evaluation of the influence of various dust settling tank designs on the suction streams value. Based on conducted analyses and experimental tests results of a single and multi-cyclone, it was found that:

- (1) Dust suction from the cyclone settling tank causes a noticeable increase in its separation efficiency, but only up to a certain extraction degree value, called optimal, which, depending on cyclone type, is in the range $m_0 = 8\text{--}15\%$. Further increasing the suction rate is uneconomical due to a slight increase in efficiency and an increase in flow resistance caused by an increase in inlet velocity and a decrease in separation accuracy.
- (2) In a multi-cyclone consisting of a large number (several dozen) of cyclones and having a common dust trap, the use of only one or two dust extraction nozzles from the settling tank causes the distances of the outlet openings in individual cyclones sucked off from the extraction system stub to be significantly different, which makes natural differences between flow resistance values and the streams sucked off from individual cyclones. This may have a negative impact on the separation efficiency in individual cyclones, and, therefore, in the multi-cyclone.

- (3) If several dozen of the same cyclones, forming a multicyclone, have a common dust settling device, the multicyclone for the same aerosol flow conditions is less effective than the single cyclones is. The reason for this phenomenon may be the separation process disturbance in the cyclone due to the uneven aerosol distribution to all cyclones and the backflows occurrence between them. Theoretical analyses and experimental studies explaining this problem are not often found in the available literature.
- (4) The highest suction streams values Q_{Sc} from individual cyclones occur, regardless of the main suction stream values from the Q_{SF} settling tank, for the cyclones located closest to the dust suction nozzle from the settling tank. As the subsequent cyclones position moves away from the suction stub pipe, the Q_{Sc} values sucked off streams systematically decrease (Figures 24–27) and, for the furthest cyclones, they have values 35–50% lower.
- (5) The developed suction streams Q_{Sc} measuring methodology of suction streams from individual return cyclones with a tangential inlet of the air filter multi-cyclone, which is the author's original achievement, has been verified only in relation to the rectangular dust settling tank chamber. It can also be used to test uneven suction from circular multi-cyclone settling tanks.
- (6) Uniformed suction streams value from multi-cyclone individual cyclones made of several dozen cyclones can be ensured by making an appropriate design and changing dust settler spatial structure. For this purpose, maintaining symmetry method of all cyclones position in relation to the suction stub pipe (Figure 28) or stream resistances method can be used by dividing the settling tank space into independent segments and channels covering a specific number of cyclones (Figure 29). Specific method application depends on the number and cyclones type that make up multi-cyclone, space limitations in multi-cyclone installation place, and possibility of carrying out appropriate tests range.

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Article

Assessment of the Impact of Applying a Non-Factory Dual-Fuel (Diesel/Natural Gas) Installation on the Traction Properties and Emissions of Selected Exhaust Components of a Road Semi-Trailer Truck Unit

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Abstract: Problems such as global warming and rising oil prices are driving the implementation of ideas to reduce liquid fuel consumption and greenhouse gas emissions. One of them is the use of natural gas as an energy source. It is a hydrocarbon fuel with properties that allow the reduction of CO₂ (Carbon Dioxide) emissions during combustion. Solutions are being implemented that allow for the use of natural gas to means of transport, namely in trucks of various categories and intended use. These installations are used in new vehicles, but also in the form of conversion for used cars, usually several years old. The article presents the results of tests of an engine from a used semi-trailer truck with a mileage of approx. 800 thousand km, with the compressed natural gas supply system installed. This installation (hardware and software), depending on the engine operating conditions, enables the replacement of up to 80% of diesel (base fuel) with natural gas. The impact of changing the fuel supply method on the traction characteristics calculated with the use of external characteristics of both conventional and dual-fuel mode was assessed. The emissions of exhaust gas components were also determined under the conditions of the UNECE Regulation No. 49. The test results confirm that compared to conventional fueling, dual fueling allows for a significant reduction in CO₂ emissions, even in a used vehicle with high mileage. The use of a non-factory installation did not significantly affect the traction properties of the vehicle, and engine wear is of greater importance in this case (comparison with factory data). The work is a valuable supplement to the results of the research in which the impact of the use of a non-factory CNG (Compressed Natural Gas) supply system on the performance of a semi-trailer truck unit equipped with such an installation was assessed compared to a semi-trailer truck unit powered in a classic way with diesel fuel.

Keywords: CNG; dual-fuel supply; engine external characteristics; tractive effort curves

1. Introduction

Along with the development of civilization and the increase in consumerism, the demand for the transport of goods using wheeled vehicles is also increasing [1,2]. This is especially true in cities where people have to deal with the negative health effects of pollution caused by an increasing number of vans and trucks [3,4]. It is estimated that only in Poland, about 50,000 people die each year due to bad air quality, or even twice as much, taking into account the percentage of deaths in Europe caused by exposure to fine particulate matter (PM 2.5) created as a result of burning fossil fuels for a population over 14 years old [5]. Typical airborne dust concentrations can vary from 0.01 mg/m³ in clean rural environments to about 20 g/m³ in desert conditions [6]. The basic components of road dust are silica (SiO₂) and corundum (Al₂O₃), the content of which in the total mass of dust can reach up to 95%. The grains of these minerals are characterized by

high hardness, 7 and 9 degrees on the ten-point Mohs scale, respectively, and irregular shape [7].

Public transport seems to offer a more rational use of the environment [8] by reducing fuel consumption, exhaust emissions [9–11] and noise. The concept of sustainable transport combines high-quality transport with care for the environment [12]. Therefore, these vehicles should have a power source other than diesel or gasoline. Of course, reducing CO₂ emissions from transport can improve quality of life, but other factors such as access, amenities and safety need to be taken into account [13].

Currently, it is particularly important to reduce the emission of harmful exhaust gas components through the development of low-emission urban transport and road transport of goods [14–16]. Importantly, according to the statistics published by the Barcelona Institute for Global Health [17], on the list of the 50 most polluted cities in the EU (European Union), 15 are in Poland.

The data of the European Environment Agency [18] show that transport in the European Union causes a systematic increase in the emission of harmful substances, mainly greenhouse gases. In the years 1990–2015, it reached a level of about 35% [18]. Therefore, the European Commission began to take decisions aimed at reducing the consumption of fossil fuels and their negative impact on the environment. It was assumed that it is necessary to reduce greenhouse gas (GHG) emissions by 2050 by at least 60% compared to 1990 levels. This is particularly important for road transport, as it dominates freight transport in the European Union (about 75% of total transport in 2018) [19].

An example of such activities may be the increasing restrictiveness of requirements for products formed during the combustion of a fuel mixture. Their implementation means that, for example, even relatively new semi-trailer truck units (5 years old) are classified as polluting and their use is subject to additional fees, e.g., increased road fees or extra ecological taxes. This forces vehicle manufacturers to seek technical solutions that reduce harmful emissions. It seems obvious to build trucks with hybrid or electric engines only. However, in this case, range is still a limitation, which for an electric car with medium load capacity is currently around 400 km, while for a semi-trailer truck unit, it is, according to the available data, only 100–300 km [20]. In addition, the price of an electric semi-trailer truck unit in 2021 was five to eight times higher than the price of the same conventional semi-trailer truck (DAF, Volvo and Scania as an example). In addition, the increase in the weight of the semi-trailer truck unit as a result of the use of heavy batteries reduces the weight of the load that can be transported, and the loading process is time-consuming. Both of these elements have a strongly negative impact on the logistic costs of using such a vehicle. Therefore, a good direction of changes is still the introduction of more environmentally friendly solutions in road transport, mainly through the use of ecological or alternative fuels to diesel, including natural gas in various forms such as Compressed or Liquid Natural Gas (CNG—LNG).

The combustion of natural gas, due to its different composition compared to diesel fuel with a lower share of carbon and a greater share of hydrogen, results in lower carbon dioxide emissions and reduces greenhouse gas emissions and the toxicity of the resulting exhaust gases. In the short and medium term, natural gas may become the most important alternative fuel [21,22]. Large natural gas resources increase its importance as an ecological fuel, which can be obtained to a large extent from the countries' own sources. In addition, many European countries have a well-developed infrastructure adapted to distribution of CNG and LNG. Natural gas is also a relatively cheap fuel in relation to diesel. In 2020, the average price of natural gas accounted for approx. 60% of the diesel fuel price. Taking into account its lower energy value compared to petroleum-derived fuels, the average cost of using natural gas to drive heavy transport vehicles is about 40% lower than that of diesel fuel [23].

The properties of natural gas (in particular in the form of compressed CNG) in terms of the use of internal combustion engines in transport/utility vehicles to power reciprocating internal combustion engines, the availability of resources and the growing

sources of supply in individual European countries justify undertaking scientific, research and development works on the use of CNG to power internal combustion engines of heavy transport vehicles. This is in line with the energy policy of the European Union—Directive 2009/33/EC of 23 April 2009—on the promotion of ecologically clean and energy-efficient road transport vehicles. The presented work is an example of such activities. It describes the effects (measured and calculated) of the application of an innovative concept of effective adaptation of a semi-trailer truck engine to CNG in dual-fuel mode. The method of fuel supplying in dual-fuel engines can be implemented in many different ways. For a given application, those that can be used in a given implementation are selected. The basic method of supplying gas fuel is through the intake manifold to the interior of the engine. This solution is the original solution from which it is worth starting the research of a given dual-fuel engine. Further modifications that can be applied to the tested engine include the introduction of direct gas injection into the combustion chamber and multiple fuel injection, allowing the ignition of the mixture in alternative modes of compression ignition, such as Reactivity Controlled Compression Ignition (RCCI) or Direct Dual Fuel Stratification (DDFS) [24]. To achieve mentioned ignition methods, it is required to create a homogeneous mixture of gaseous fuel with air, which requires interference in the structure of the engine intake system and the combustion chamber, in which the level of the mixture swirling should change [25].

The operation of an internal combustion engine in which a homogeneous mixture of fuel and air is self-igniting differs significantly from classic internal combustion engines, primarily in terms of efficiency and emission of harmful exhaust components. Increasing the thermal efficiency of such an engine is related to the reduction of emissions of many harmful exhaust components, the emission of which is difficult to reduce in a long-term manner with the use of an engine's exhaust aftertreatment system. This will have a major impact on human health and environmental pollution in the future.

The article is part of the topicality of the problem of searching for new concepts of propulsion of the means of transport focused on pro-ecological solutions and alternative fuels. The developed methodology of tractive effort curves and ecological properties of CNG-powered trucks will allow for the classification of this fuel for the propulsion of motor vehicles. Currently, most researchers focus on assessing the ecological properties of CNG-powered vehicles, without taking into account driving performance after changing the engine fueling method. In the case of the mass transport of goods with the use of a road semi-trailer truck unit, their mechanics of movement are of great importance, which is discussed in this paper.

2. Purpose, Object and Research Plan

The main goal of the research was to assess the impact of the use of CNG supply on the ecological parameters of vehicles in terms of their use in natural operation in the transport of goods compared to vehicles powered by conventional fuel. Particular emphasis was placed on the comparison of the vehicle's tractive force and their ecological properties in relation to the same vehicle powered by diesel fuel only. During the research, an innovative high-pressure indirect CNG injection system was used to supply the engine. The mentioned CNG injection system was developed in cooperation with the Military University of Technology in Warsaw.

The aim of the research was to experimentally determine how the application of a non-factory diesel/CNG dual-fuel installation to a used tractor unit of a tractor engine with a semi-trailer affects its tractive effort curves and changes the emissions of individual exhaust components: NO (Nitrogen Oxide), NO₂ (Nitrogen Dioxide) NO_x (Nitrogen Oxides), CH₄ (Methane), CO (Carbon Dioxide), CO₂ (Carbon Dioxide), HC (Hydrocarbons) and smoke opacity in the test cycle carried out in accordance with the requirements of the UNECE Regulation No. 49.

The investigation was led on a standard dynamometer stand. The engine was load by the Zöllner PS1-3812/AE water dynamometer with 1250 kW maximum power. Torque

produced by dynamometer was measured with tensometric transducer. Rotational speed was measured with pulse transducer in cooperation with toothed wheel, which was on a dynamometer flange. Measuring of fuel consumption was led with AVL Fuel Balance every 5 s (diesel) and SwirlMaster FSS450 Intelligent Swirl Flowmeter (CNG). Temperature of cooling liquid was controlled on the level $85 \div 90$ °C using an external heat exchanger. Table 1 presents the basic technical parameters of the investigation equipment used.

Table 1. List of investigation equipment used during the investigation.

No.	Name of Device/Measured Quantity	Type	Range	Accuracy
Water dynamometer				
1.	– Torque— M_o , – rotated speed— n	Zöllner PS1-3812/AE	$M_o = (0 \div 700) \text{ N}\cdot\text{m}$, $n = (0 \div 4000) \text{ rpm}$	$\pm 1 \text{ N}\cdot\text{m}$, $\pm 1 \text{ rpm}$,
2.	Fuel weight-meter (diesel)	AVL 733S Fuel Balance	$(0 \div 200) \text{ kg/h}$	$\pm 0.005 \text{ kg/h}$
3.	Fuel weight-meter (CNG)	SwirlMaster FSS450 Intelligent Swirl Flowmeter	$1.2\text{--}104.6 \text{ kg/h}$	$\pm 0.6\%$ measured quantity
Exhaust analyzer—measuring of toxic elements Concentration in exhaust gases				
4.	– hydrocarbons (HC), – nitrogen oxides (NO_x),	AVL CEBII	HC $(1.0 \div 2200) \text{ ppm}$, NO_x $(1.0 \div 6000) \text{ ppm}$	$\pm 0.1\%$ measured quantity
Exhaust analyzer—measuring of toxic elements Concentration in exhaust gases				
5.	– carbon dioxide (CO_2), – carbon monoxide (CO), – nitrogen oxides (NO_x), – nitrogen dioxide (NO_2), – oxygen (O_2).	atmosFIR, emissions monitoring FTIR systems	CO_2 $(0.01 \div 23)\%$ CO $(1.0 \div 11,000) \text{ ppm}$, NO_x $(1.0 \div 6000) \text{ ppm}$ NO_2 $(1.0 \div 6000)$ O_2 $(0.1 \div 21)\%$	$\pm 0.1\%$ measured quantity
6.	Smoke concentration—extinction coefficient of light radiation— k .	AVL Opacimeter 4390	$(0.001 \div 10.0) \text{ m}^{-1}$	$\pm 0.002 \text{ m}^{-1}$
7.	Thermocouple—measuring of exhaust temperature— T	NiCr—NiAl (K)	$(-50 \div 1100) \text{ }^\circ\text{C}$	$\pm 1 \text{ }^\circ\text{C}$
8.	Mass air consumption	SensyMaster FMT430 Thermal Mass Flowmeter	$120 \div 7000 \text{ kg/h}$	$\pm 1.2 \text{ kg/h}$

The tests and calculations were carried out for the D13C460 EURO V EEV (Enhanced Environmentally friendly Vehicle) (338 kW) engine, used in the Volvo FH13 series semi-trailer truck (Figure 1). The mileage of the semi-trailer truck, the unit of which was tested on an engine dynamometer and on which a non-factory diesel/CNG installation was adapted, was 790,500 km. The total operating time of the engine is 11,800 h, the engine has used 229,100 liters of diesel fuel since the beginning of its operation. The tested engine, according to the approval documents, met the requirements of the EURO V standard when it left the factory.

The planned tests were divided into the following steps:

- Determination of the external characteristics of the D13C460 engine when fueled by the basic fuel, which was diesel;
- Determination of the external characteristics of the D13C460 engine with a diesel/CNG supply system installed and optimized on a dynamometric test bench during dual fuel operation;

- An assessment of the impact of the dual-fuel diesel/CNG supply of the D13C460 engine on the tractive effort curves of a semi-trailer truck and semi-trailer combination with a total weight of 40 tons;
- An assessment of the impact of the dual fuel diesel/CNG supply on the emission of the toxic exhaust gas components during the research test in accordance with the requirements of the UNECE Regulation No. 49.



Figure 1. Semi-trailer truck Volvo FH13.

2.1. Determination of the External Characteristics of the Engine

The work began with the determination of the external characteristics of the D13C460 engine, which was dismantled from the semi-trailer truck unit. The test was performed on a dynamometric stand located in the Department of Engines and Exploitation Engineering of the Military University of Technology in Warsaw. Commercial diesel fuel was used to power the engine. Due to the design limitations of the dynamometric brake used, the external characteristics were determined in the speed range from 900 rpm to 2100 rpm (compared to the factory data, the area of low values was omitted—from 600 rpm to 900 rpm).

The engine was equipped with a power supply system that allows two types of fuel (diesel and CNG) to be delivered to the cylinders. In the course of adaptation activities, it was subjected to the optimization process, in which the main criterion was the maximum replacement of diesel oil by CNG, taking into account the influence of the current engine operating conditions (an appropriate example of a gas replacement map is shown later. After this determination, the values of the torque and the effective power of the engine as a function of the rotational speed of the engine crankshaft were again carried out. The test results are summarized in Figure 2 and Table 2 (additionally, for comparative purposes, the manufacturer's catalog characteristics were also taken into account—marked as factory data).

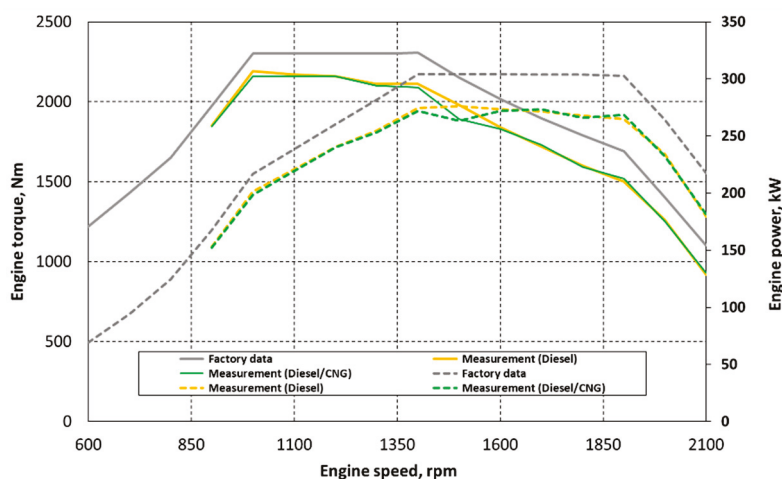


Figure 2. External characteristics of the engine: solid line—torque, dashed line—power.

Table 2. Comparison of the values of the torque and the useful power used in the traction calculations of a set of vehicles.

Engine Speed	Factory Data		Diesel		Diesel/CNG	
RPM	N·m	kW	N·m	kW	N·m	kW
600	1220	76.6	-	-	-	-
700	1430	104.8	-	-	-	-
800	1650	138.2	-	-	-	-
900	1975	186.1	1850	152.9	1845	152.4
1000	2300	240.8	2190	201.5	2160	198.4
1100	2300	264.9	2170	220.8	2160	219.6
1200	2300	289.0	2160	240.6	2160	240.0
1300	2300	313.1	2110	254.7	2100	253.3
1400	2305	337.9	2110	274.5	2090	271.9
1500	2151	337.9	1980	275.9	1890	263.4
1600	2017	337.9	1840	273.5	1830	272.0
1700	1897	337.7	1720	271.7	1730	273.3
1800	1790	337.4	1600	267.6	1590	265.9
1900	1690	336.2	1500	264.8	1520	268.3
2000	1400	293.2	1260	234.1	1250	232.3
2100	1100	241.9	920	179.5	930	181.5

The external characteristics determined in this way were used to assess the impact of the change in the power supply method on the tractive effort curves of a fully loaded (total weight 40 tons) semi-trailer truck and semi-trailer unit.

2.2. Traction Characteristics—Forces and Vehicle Performance

External forces, which are shown in Figure 3, acts on a car moving in a uniform or alternating rectilinear motion on a flat road surface. The most important ones include the following forces: W_V —vehicle weight, F_r —rolling resistance force, F_D —aerodynamic drag force and movement occurring in certain circumstances, namely: F_T —tractive effort, F_g —grading resistance force, F_i —inertia force, F_P —semi-trailer truck pulling force (trailer resistance force) and normal reactions Z_1 and Z_2 under wheels (where 1 and 2 are the first and second axes of the vehicle). Based on the balance of forces, we can write:

$$F_T = F_{r1} + F_{r2} + F_D - F_i + F_P + W_v \sin \alpha \quad (1)$$

$$Z_1 + Z_2 = W_v \cos \alpha + F_P \tan \alpha_H \quad (2)$$

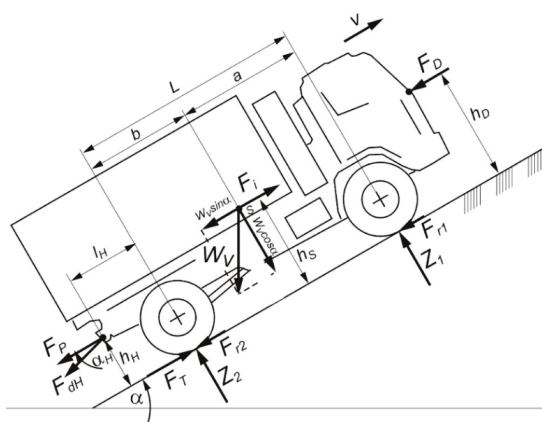


Figure 3. System of forces and reactions acting on the vehicle (description in the text): S—center of mass, h_S —height of the center of mass, h_H —height of the towing hitch, h_D —height of the center of wind pressure, a , b and L —the distances between the center of mass and the axles of one and the vehicle's wheelbase, α —road inclination angle, α_H —tilt angle of drawbar.

The rolling resistance force acts on the road wheels of the vehicle and amounts to:

$$F_r = f_r \cdot (W_v \cos \alpha + F_P \tan \alpha_H) \quad (3)$$

In the estimated force calculations, the rolling resistance coefficient can be determined from the empirical equation [26]:

$$f_v = f_r \cdot [1 + (0.0216v)^2] \text{ for } v, \text{ m/s} \quad (4)$$

The aerodynamic drag force affects the vehicle's running properties at high speeds (for speeds $v < 30$ km/h, the effect of the F_D force is often neglected [26]). It is an aerodynamic force and acts on the body in the opposite direction to the vehicle's direction. The value of the aerodynamic drag force is determined by [27,28]:

$$F_D = C_D \cdot A_x \cdot \rho \cdot \frac{V_r^2}{2} \quad (5)$$

where: A_x —frontal area of the vehicle body; ρ —air density; C_D —quantifies the aerodynamic properties of the car and is usually called the aerodynamic drag coefficient or the aerodynamic coefficient; V_r —relative speed of vehicle and wind velocity component in the vehicle moving direction.

Surface A_x is usually calculated as an approximation [26]:

$$A_x = \gamma_P \cdot H \cdot b_w \quad (6)$$

where: γ_P —cross-sectional filling factor, which for heavy goods vehicles is 0.9–1.1; H —vehicle height; b_w —track width; $A_x = 9.0 \div 12.0 \text{ m}^2$ —for vehicles with a load capacity of more than 15 tons.

The grading resistance is the component of gravity parallel to the road surface:

$$F_g = W_v \sin \alpha \quad (7)$$

When negotiating gradients and descents, the pressure of the vehicle's wheels on the road is reduced. This reduces the rolling resistance value at the expense of the increase in the force F_g . The combined approach to the uphill and rolling resistance forces creates the so-called road resistance:

$$F_\Psi = F_r + F_g = W_v (f_r \cos \alpha + \sin \alpha) = W_v \Psi \text{ gdzie } \Psi = f_r \cos \alpha + \sin \alpha \quad (8)$$

The amount of road resistance makes it easier to jointly capture the impact of the road on the vehicle, especially at the initial stage of vehicle calculations or evaluation of their tractive effort curves.

The force of inertia is opposed to changes in the speed of the vehicle. There are two basic components of the vehicle inertia resistance resulting from translational and rotational motion:

$$F_{i1} = -m \frac{dv}{dt} = -\frac{W_v}{g} \frac{dv}{dt} \text{ and } M_i = -\sum_k^k I_k \frac{d\omega_K}{dt} = F_{i2} r_D \quad (9)$$

where I_k —moment of inertia of the k th rotating element (including moving parts of the engine and drive system), reduced to the driving wheels axis; ω_K —angular velocity of the drive wheel with dynamic rolling radius r_D ; $\omega_K \approx v/r_D$.

Therefore, the force of inertia can be described by an approximate:

$$F_i = F_{i1} + F_{i2} = -\delta \frac{W_v}{g} \frac{dv}{dt} \quad (10)$$

where δ (rotating mass factor) stands for the ratio of the rotating masses of the engine and drivetrain. Its value can be approximated as follows [29]:

$$\delta = 1 + (0.04 \div 0.06) \left(1 + i_{gb}^2\right) \frac{W_{cv}}{W_v} \quad (11)$$

where W_{cv} —current vehicle weight; W_v —nominal vehicle weight; i_{gb} —gear ratio in the gearbox.

Trailer drag force results from the sum of the resistance to motion of the trailer towed by the vehicle. The force acting directly on the trailer towbar is F_{dH} (Figure 3). The trailer's drag force F_P directed parallel to the ground is considered:

$$F_P = F_{dH} \cos \alpha_H \quad (12)$$

The aerodynamic drag force of a trailer with a height not exceeding that of the semi-trailer truck usually reaches up to 35% of the semi-trailer truck aerodynamic drag forces value.

Reactions to the road surface and the limitations resulting from the grip of the tires change constantly while driving. The values of normal reactions that vary while driving are often compared to those calculated in the state of static equilibrium (the vehicle is affected by the force of gravity W_V and the normal and tangential reactions to the road surface). A significant limitation of the available driving force is the adhesion of the wheels to the road surface and the load per drive axle.

This limitation has not been taken into account in the study, because this factor is not related to any changes resulting from the way the engine is powered.

The equation of motion of the car results from the sum of projections (on the direction parallel to the road surface and the longitudinal axis of the vehicle) of the forces acting on the vehicle and the balance of power in the vehicle drive system (power available on the wheels) can be written by the equation:

$$P_T = P_e - P_{loss} = F_T v = N_e \eta_{dl} = P_{RT} \quad (13)$$

where $P_{RT} = \sum F_{RT} v$ is the power of resistance to motion (demanded power). In this equation, P_{loss} expresses the power losses occurring in the drive system, when it is transferred from the engine to the driving wheels. These losses are included in the form of the overall efficiency of the drive system η_{dl} . The individual components of the power of resistances to motion can be expressed as follows from the equation of motion:

$$P_{RT} = \sum F_{RT} v = (F_r + F_D + F_g - F_i) \cdot v = P_r + P_D + P_g - P_i \quad (14)$$

In order for the vehicle to move, there must be a relation $P_e \cdot \eta_{dl} \geq P_{RT}$, where the value of the engine power is read from its external (speed) characteristic. On the other hand, the difference $P_e \cdot \eta_{dl} - P_{RT} = \Delta P$ is the excess power that can be used for acceleration, acceleration of the vehicle or overcoming the resistance to motion in difficult road conditions or the resistance to motion of the trailer.

Tractive effort curves are often used to compare performance of vehicles of different designs. It is presented in the form of a diagram of the relation between the driving force F_T on the wheels and the driving speed v , which is calculated as follows [28,30]:

$$F_T = \frac{T_e \cdot i_{dl} \cdot \eta_{dl}}{r_D} \text{ and } v = \frac{2\pi n_S}{60 i_{dl}} r_K \quad (15)$$

where T_e is the engine torque, i_{dl} and η_{dl} are the gear ratio and efficiency of the driveline, r_D is the dynamic wheel radius, n_S is the rotational speed of the engine crankshaft and r_K is the kinematic radius of the wheel, which is usually assumed to be equal to the dynamic

radius r_D (in calculations later in the work for simplification, rolling of wheels without skidding was assumed).

An example of the tractive effort curves is shown in Figure 4a. The number of visible branches (black lines) corresponds to the number of constant gear ratios that can be obtained in the car's drive system. The figure also shows the course of the drag forces resulting from the interaction of the rolling resistance and the air drag force with a blue line. Its point of intersection with the branches of individual gears shows the value of the maximum speed that the vehicle can reach under given conditions.

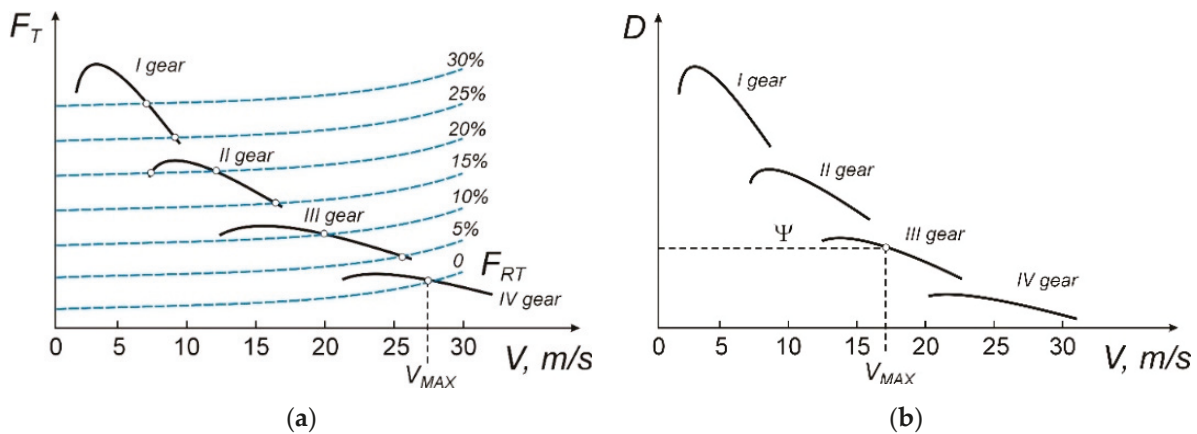


Figure 4. Examples of (a) the tractive effort curves—driving force as a function of speed for a car equipped with a four-speed gearbox, with the marked influence of the angle α of the road inclination on the maximum speed achieved (marked points of intersection) and (b) the dynamic characteristics $D = f(v)$ of a car with the same gearbox.

Figure 4a additionally shows the lines of force of resistance to movement while climbing hills with increasing inclination (inclination expressed as a percentage). Such a combination allows you to show not only what maximum speed can be achieved, but also which gear allows you to overcome a given incline. Starting from the equation of the car motion $F_T = \sum F_{RT} = F_r + F_D + F_g - F_i$, the concept of a performance factor is introduced [31]:

$$D = \frac{F_T - F_D}{W} \quad (16)$$

where W is the weight of the vehicle (semi-trailer truck with semi-trailer). The diagram of the relation $D = f(v)$ shown in Figure 4b is called the dynamic characteristic. It is a convenient indicator for evaluating and comparing vehicle performance. The dynamic characteristics can be used to determine the current road conditions:

$$D_{\Psi} = \frac{F_r + F_g}{W} = f_r \cos \alpha + \sin \alpha \rightarrow \Psi \quad (17)$$

where the lines $D_{\Psi} \rightarrow \Psi$ indicate the value of the achievable speed in road conditions, described in a conventional manner by the value of the road resistance coefficient Ψ .

The location of these lines makes it easier to determine the maximum speed in the considered road conditions and the accelerations that can be achieved. Their designation results from the following list [32]:

$$D - D_{\Psi} = \frac{\delta}{g} \frac{dv}{dt} \rightarrow \frac{dv}{dt} = \frac{g}{\delta} [D(v) - \Psi] \text{ and } \left. \frac{dv}{dt} \right|_{\max} \approx \frac{\delta}{g} (D_{\max} - f_r) \text{ dla } \alpha = 0 \quad (18)$$

Using the dynamic characteristics, it is possible to determine the time and path of acceleration of the car:

$$t_a = \int_{v_{\min}}^v \frac{\delta \cdot dv}{g \cdot (D(v) - \Psi)}, \quad S_a = \int_0^{t_a} v dv \quad (19)$$

2.3. Selection of Data for Calculations

The next step of preparations was the selection of the values of the values characterizing the tested set of vehicles and other design factors influencing its traction parameters. Available data, the details of which are described below, were used to determine the resistance to motion, the value of the torque transmitted to the wheels and the power.

Semi-trailer truck and semi-trailer: One of the most important parameters determining the driving properties of a set of vehicles is its permissible total weight, which according to the regulations in force (see Paragraph 3 in [33]) may not exceed the value of 40 tons in the case of a two-axle semi-trailer truck unit and a three-axle semi-trailer (or a three-axle semi-trailer truck and two-axle semi-trailer).

Drivetrain: The Volvo FH13 semi-trailer truck is equipped with 14-speed gearbox VTO2214B [34]. It is a manual gearbox, consisting of three mechanisms: the Splitter Group transmission (two speeds), the main gearbox (four speeds) and the Range Change Group transmission (two speeds) (Figure 5). The letter “O” used in the marking shows that the Splitter Group transmission was made in a different way than the traditional one, i.e., its gear ratio is 1.0 in the “Split Low” position and 0.8 in the “Split High” position. Finally, the ratios of individual gears were determined, the values of which are summarized in Table 3. The letters C and R denote the ratios of the creeping and reverse gears, and the SL, SH, RL and RH, respectively, show the low and high gear ratios of the Splitter and Range Group gears. The gear change time during acceleration was assumed to be 1 second.

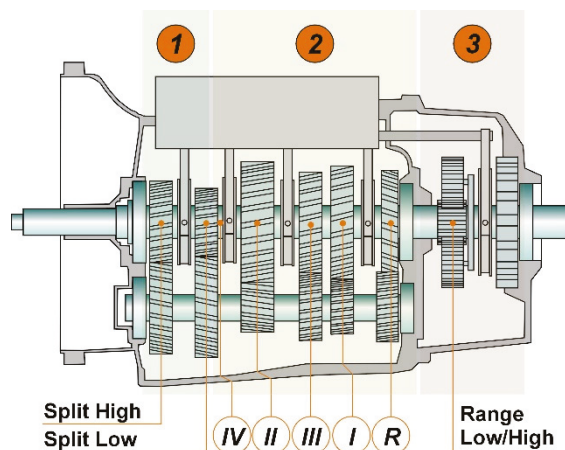


Figure 5. Schematic diagram of the VTO2214B gearbox design system [34]: 1—Splitter Group gearbox, 2—main gearbox, 3—Range Change Group gearbox, Split High—(SH, $i = 0.80$), Split Low—(SL, $i = 1.00$), High Range—(HR, $i = 1.00$), Low Range—(LR, $i = 3.75$).

Table 3. Identification of the gear ratios of the VTO2214B gearbox.

Type of Gearing	Splitter Group			Main Gearbox							Range Change Group			
	SL	SH	I	II	III	IV	W	RL	RH					
Ratio	1.00	0.80	3.605	2.380	1.530	1.000	3.225	3.75	1.00					
Gear Ratio	C1 13.52	C2 10.82	1 8.93	2 7.14	3 5.74	4 4.59	5 3.75	6 3.00	7 2.38	8 1.90	9 1.53	10 1.22	11 1.00	12 0.80
Gear Ratio		R1 12.09			R2 9.68		R3 3.23						R4 2.58	

The torque is transmitted to the wheels of the drive axle via the drive shaft and the main gear located in the drive axle (manufacturer’s code RSS1344C) ratio of 2.85 to 1 [35].

Apart from the transmission ratio, the efficiency of the drive system and the dynamic radius of the wheel also determine the value of the driving force developed on the wheels. The efficiency of the classic system [26,31] is the product of the efficiency of its components,

which include the efficiency of couplings (possible slippage), cylindrical wheels with external and internal gearing, drive shaft with joints, drive axle (bevel gear, differential and hub bearings). The final estimated efficiency of the driveline used for the calculations was $\eta_{dl} = 0.90$.

Semi-trailer truck driving wheels. The dynamic radius of the drive wheel was calculated with the wheel size 315/70R22.5 and the recommendations contained in sources [32,36]. It was also assumed that the wheels would be loaded with the nominal value of the normal load (specified in the tire catalog); hence, its value will be $r_D = 0.49$ m.

Vehicle movement resistance. The rolling resistance coefficient of the size wheels was determined on the basis of the test results presented in [37,38] for semi-trailer truck tires of the size 315/70R22.5 and in [39] for the tires of the semi-trailer size 385/65R22.5. At the same time, the least favorable value was chosen, assuming, in simplified terms, the same for all wheels of the vehicle $f_r = 0.0083$. This procedure was performed due to the verification of only two factors influencing the movement of the vehicle, namely the torque and useful power of the engine powered by various types of fuel. The value of the rolling resistance coefficient was also dependent on the driving speed:

$$f_r(v) = f_r \cdot (1 + 0.000045 \cdot v^2) \quad (20)$$

When calculating the inertia resistance, the dependencies proposed in [26,30] were used, where the rotating mass coefficient δ , related to the gear ratios, was used:

$$\delta = 1 + 0.0175 \cdot i_{gb,k}^2 \quad (21)$$

where $i_{gb,k}$ —ratio on gear k .

By selecting a higher value of the constant factor, Equation (21) takes into account the moments of inertia of the trailer wheels in their rotational motion.

When calculating aerodynamic drag resistance, the dimensional data taken from [35,40] and information on the range of values of the aerodynamic drag coefficient C_D [41–43] were used (the possible range of changes is up to 50%). In addition, apart from the cab structure and the way of setting up the semi-trailer truck and semi-trailer unit (differences in the height of the cab and the body, their distance, etc.), there is also an influence of additional resistance resulting from increased disturbances in the flow of air in a situation when the direction of air movement does not correspond to the direction of move. This significantly increases the value of the C_D coefficient. As shown in [42], for a semi-trailer truck with a semi-trailer, it is an increase of 125% for the same driving speed and a deviation of 10° .

In order to take into account the maximum possible resistance to motion, the value of $C_D = 0.9$ (C_D for $0^\circ = 0.5$ and correction for angular deviation equal to 1.2) was selected for a set of vehicles. The frontal area was also approximated, assuming, based on the data [35,40,44]: width and height of the semi-trailer truck cab—2.49 m; 3.87 m and the height and width of the semi-trailer: 4.00 m, 2.55 m and the cross-section factor of 0.9. This allowed for the calculation of the frontal area, which is $A = 9.16$ m². The air density was also assumed to be $\rho = 1.225$ kg/m³, which corresponds to the standard conditions used in the calculations of aerodynamics.

Another component of the resistance is the longitudinal inclination of the road (resulting from its grade line and marked in Figure 3 by the angle α). Pursuant to paragraph 24 in [45], its value depends on the assumed design speed of the road and should not be greater than that specified in Table 4. A similar range of road inclination to 12% for a set of vehicles and the possibility of moving it up to such a hill five times is specified in the Regulation of the European Commission No 1230/2012 [46]. On this basis, the range of road gradient changes from 0 to 15% was adopted (it was additionally extended to 30% when calculating the maximum speed, taking into account the possible specific operating conditions in mountain terrain).

Table 4. The maximum road gradient depending on the adopted design speed [45].

Design speed of the road (km/h)	120	100	80	70	60	50	40	30
Road grade inclination (%)	4.0	5.0	6.0	7.0	8.0	9.0	10.0	12.0

In the traction calculations performed in the work, the variable factor was the engine with changed fuel supply system. They also included values taken from the catalog data of the vehicle manufacturer [35,47] (labeled “Factory data”), which showed the capabilities of the semi-trailer truck—semi-trailer combination.

2.4. Results of Tractive Effort Calculations

Figure 6 summarizes the tractive effort for the three considered variants of the external characteristics of the engine. In general, the course of the characteristics is very similar to the achieved values of the driving force and speed. Slight differences are related to the stability of the maximum engine torque (in the range of 1000–1400 rpm), which is regulated by the EDC controller. This value remains unchanged on the manufacturer’s (catalog) characteristic, while the results of the measurements performed show a decrease in the torque value by about 3.5%. This tendency is visible regardless of the diesel fuel only or diesel/CNG combination.

Compared to the factory characteristics, the course of the maximum power hyperbola has also changed, which in the F-V coordinate system is calculated as $F_T \cdot v = P_{TMAX} = \text{const}$ (P_{TMAX} is the maximum power supplied to the wheels) and is expressed by the general equation $C \cdot x^{-1}$ [48]. The determined constant values are as follows:

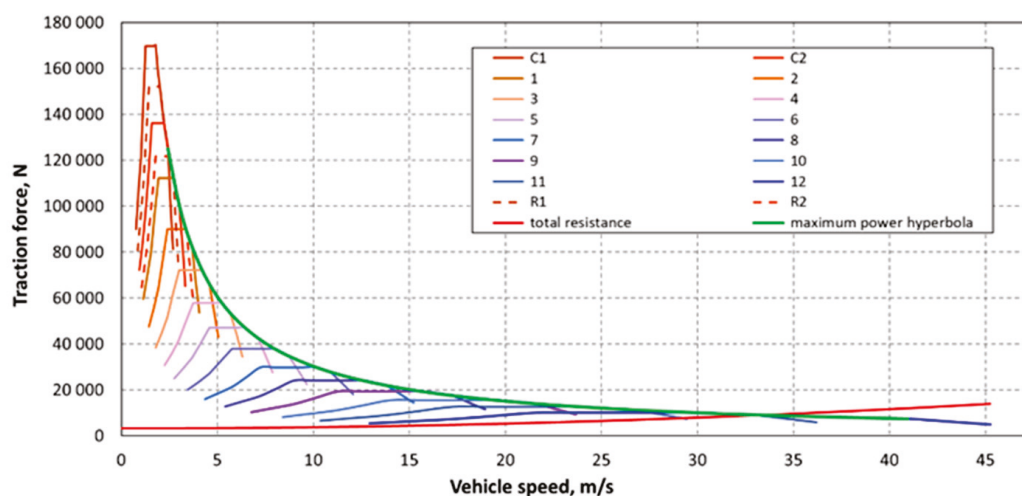
$$C_{\text{Factory data}} = 302,603 \text{ N} \cdot \text{m/s}$$

$$C_{\text{Diesel}} = 275,945 \text{ N} \cdot \text{m/s}$$

$$C_{\text{Diesel/CNG}} = 273,251 \text{ N} \cdot \text{m/s}$$

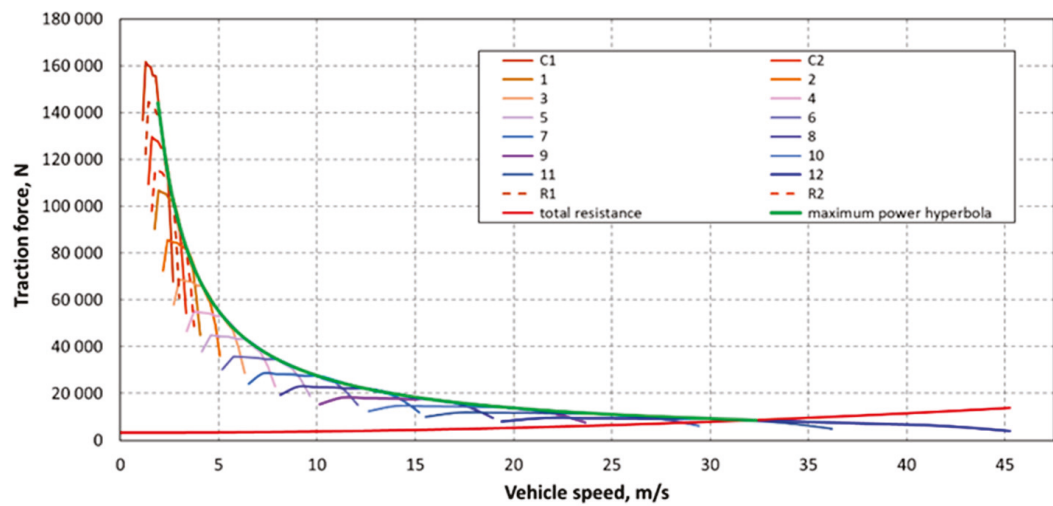
This shows that the achieved values of the driving force or speed will be greater for a vehicle with an engine operating in accordance with the factory data, and the change from diesel fuel to diesel and CNG has a very small (imperceptible) effect on the traction performance of the vehicle.

This conclusion is confirmed by calculations of the predicted maximum speed depending on the road gradient on which the vehicle is used (Figure 7). The use of the measurement results leads to the determination of the speed lower by 4–5% compared to the factory data. The change of power supply does not cause a significant reduction in speed (for a horizontal flat road, it is only 0.15 m/s, and this value decreases with the increase of the road inclination angle).

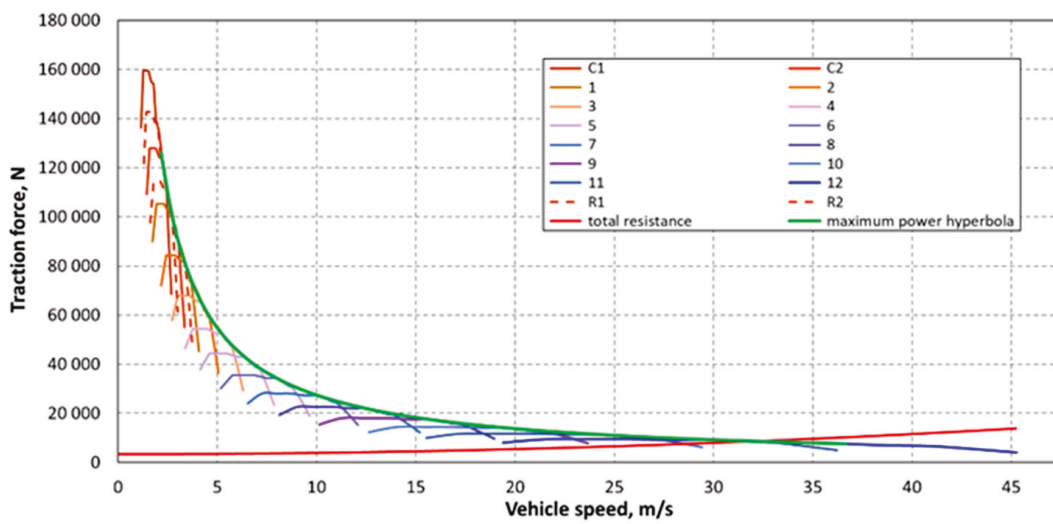


(a)

Figure 6. Cont.



(b)



(c)

Figure 6. Tractive effort curves determined for: (a) factory data, (b) diesel fuel and (c) diesel/CNG.

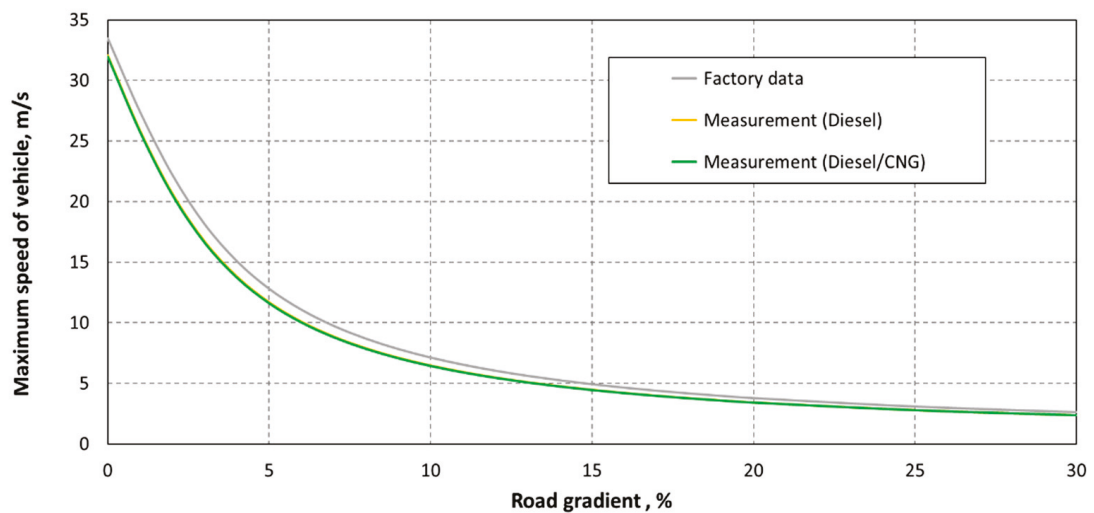


Figure 7. Change of the achieved maximum speed value depending on the road inclination angle.

Figure 8 shows the calculations of the range of gradient changes that can be covered in each of the possible gear ratios. Again, as expected, the highest elevation values are provided by the engine with the factory characteristics (gray bar). The calculations made using the measurements from the dynamometer with diesel fuel (yellow bar) showed a decrease in the percentage of the road gradient, which was about 3.5% for the crawling gear C1, 1.5% for the second gear and similarly for subsequent gears, with gradual reduction differences. The introduction of CNG to power supply caused a further reduction of the value by about 0.5% in gear C1 and about 0.25% in second gear. In the following gears, the differences are even smaller and will not be felt during typical use of the vehicle.

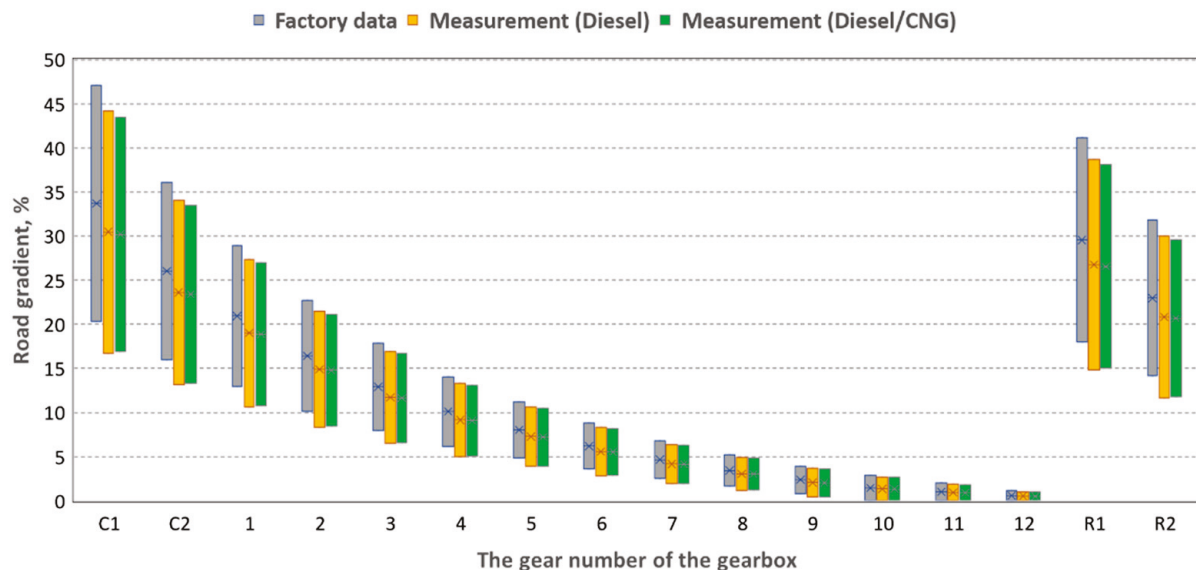
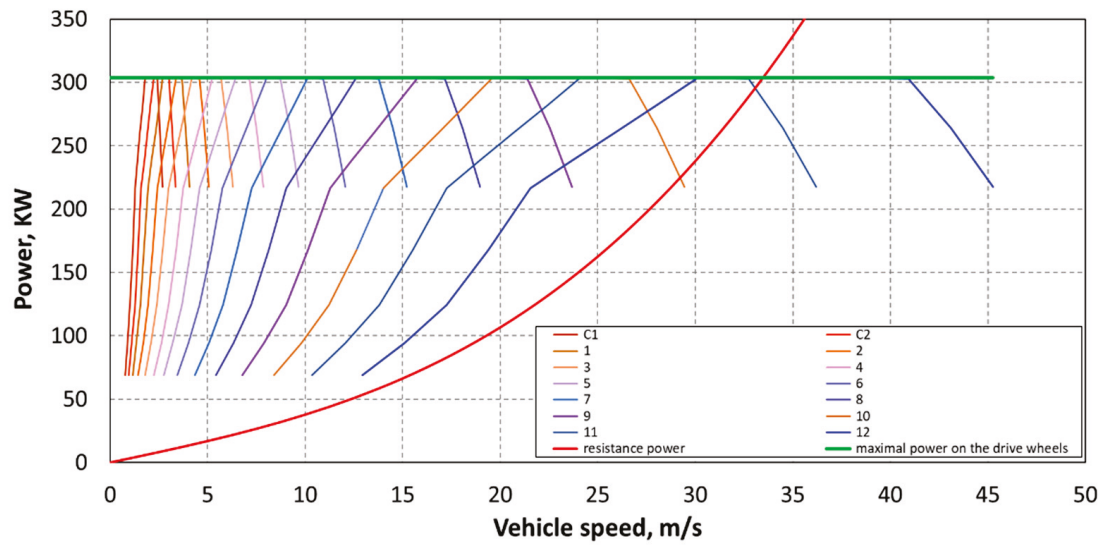


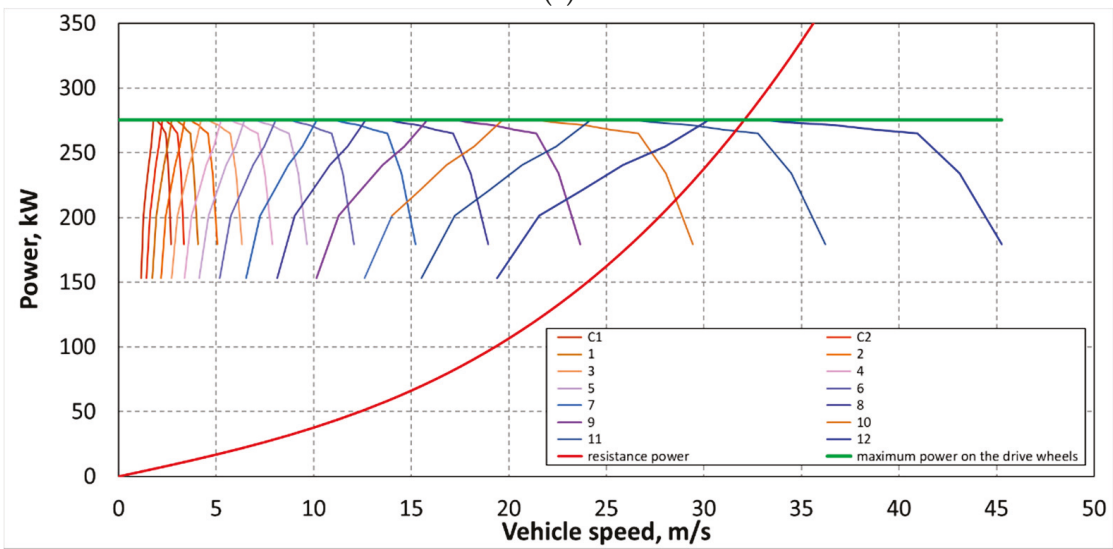
Figure 8. Gradeability in each gear of the gearbox.

Figure 9 shows the power on the drive wheels $P_T = f(v)$. The field of available power is the area under the lines of the $P_T = f(v)$ diagram, while the field of lost power is located between the line of maximum power P_{TMAX} on the wheels and the curves of the characteristics for individual gears $P_T = f(v)$. On their background, a graph of resistance power in steady motion on a horizontal road was drawn (red line). The summary shows that the shaping of the power curve determined in the measurements (its unevenness) results in the formation of the lost power fields (areas shaded in gray in the figure). In the case of diesel fuel supply, this field will appear in the range of the engine speed of 1500–1700 rpm, and for diesel/CNG, it is in a speed range of 1400–1600 rpm. At the same time, it is the largest field, but it is above the economic speed range indicated in by the manufacturer (1000–1500 rpm).

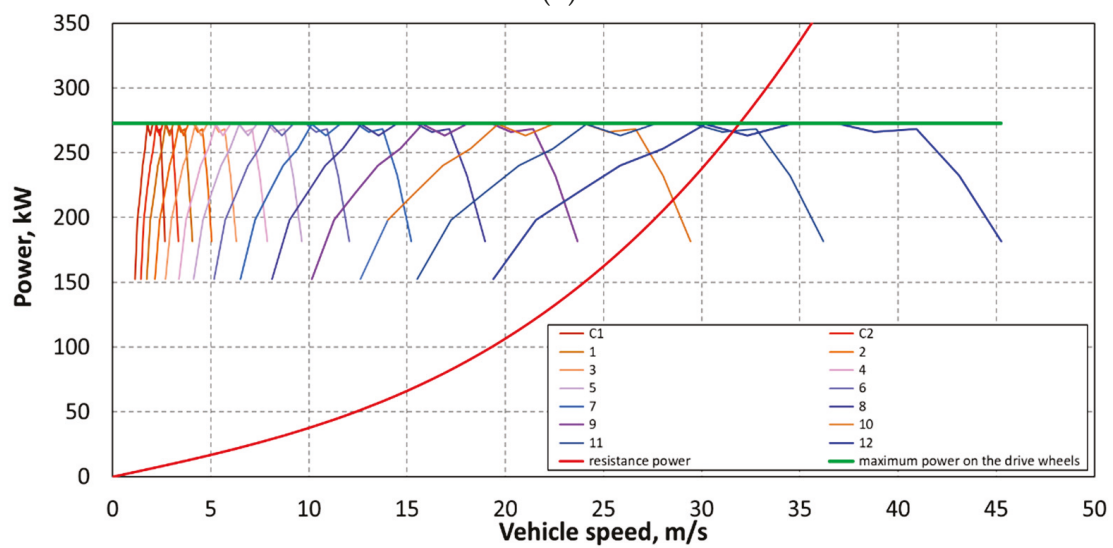
Another important parameter determining the tractive effort of a vehicle set is the performance factor D . The course of changes in its values for individual gear ratios and three considered variants are shown in Figure 10. The characteristic presented in this way allows to indicate with what maximum speed and in which gear the car will be able to overcome the assumed motion resistance. In this approach, the change of fuel supplying the engine does not result in significant differences. The largest discrepancy in values in favor of the supply with diesel only is about 1.5% and it occurs at 1000 rpm and about 5% at 1500 rpm.



(a)



(b)



(c)

Figure 9. Power supply field on drive wheels. (a) Factory data, (b) Diesel and (c) Diesel/CNG.

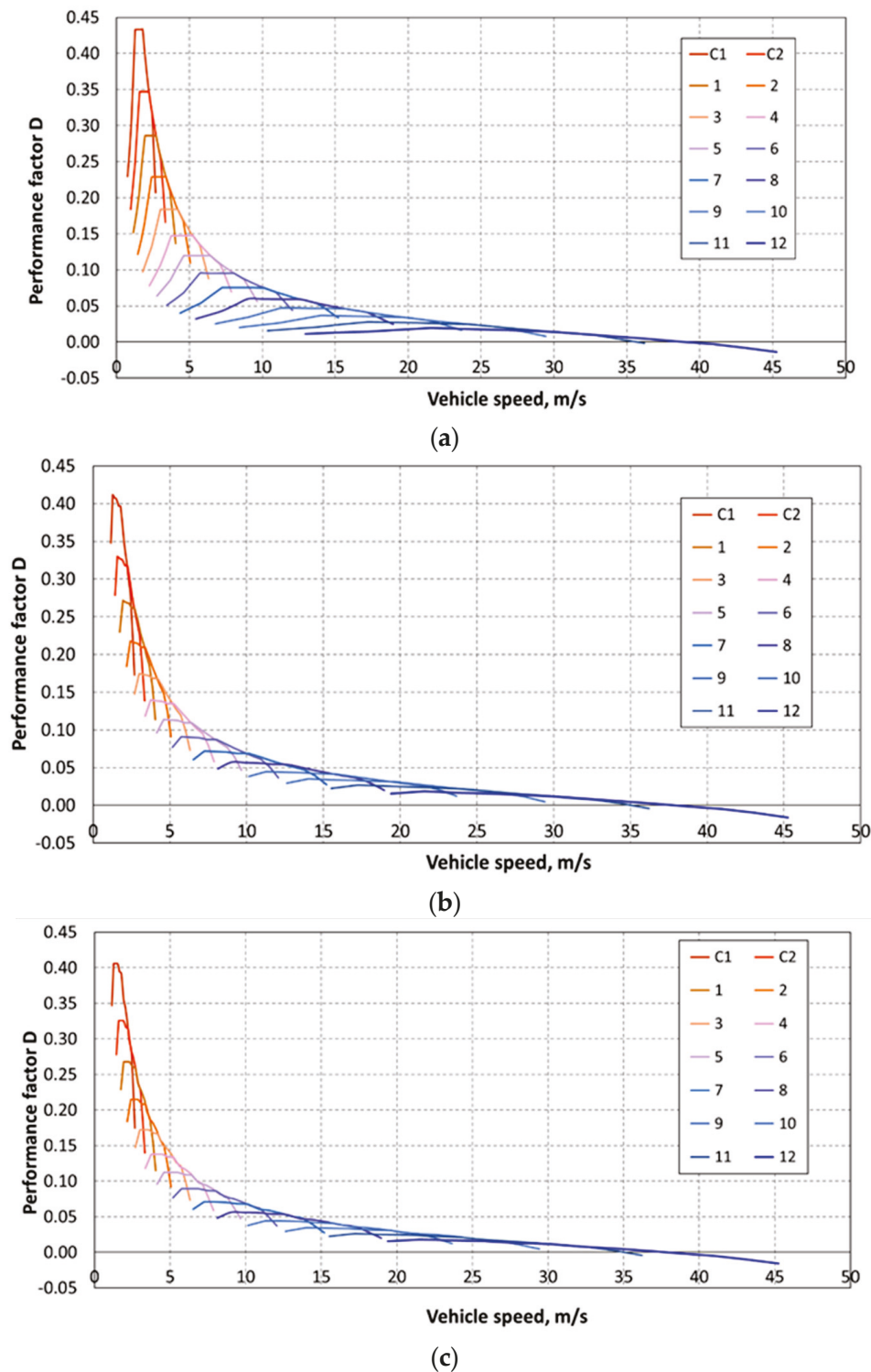


Figure 10. Dynamic characteristics. (a) Factory data, (b) Diesel and (c) Diesel/CNG.

Such a decrease in the value of the dynamic index, resulting from the unevenness of the course of the engine torque, is reflected in the achieved acceleration values. In Figure 11, for one of the gears, a dashed line has been drawn, denoting the engine speed to 1500 rpm. It should be expected that close to this value, a truck powered by diesel/CNG would accelerate 10% slower. This can be important, for example, when entering traffic or when crossing a road intersection (when one needs to start moving from a dead stop).

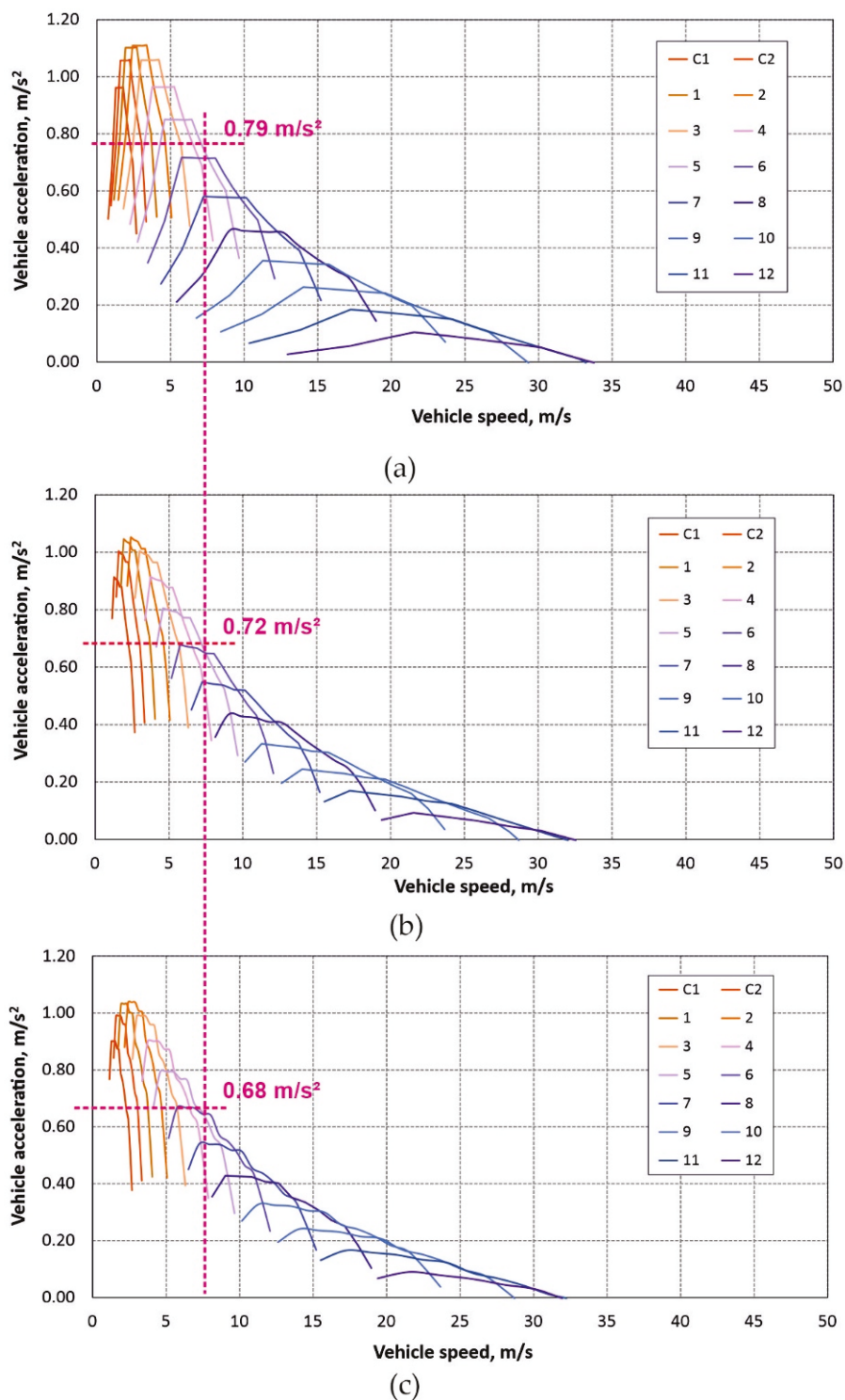
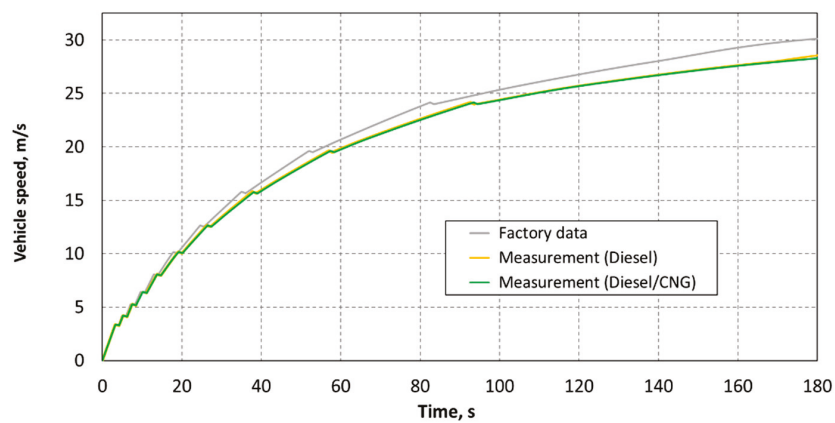


Figure 11. Acceleration achieved in individual gears. (a) Factory data, (b) Diesel and (c) Diesel/CNG.

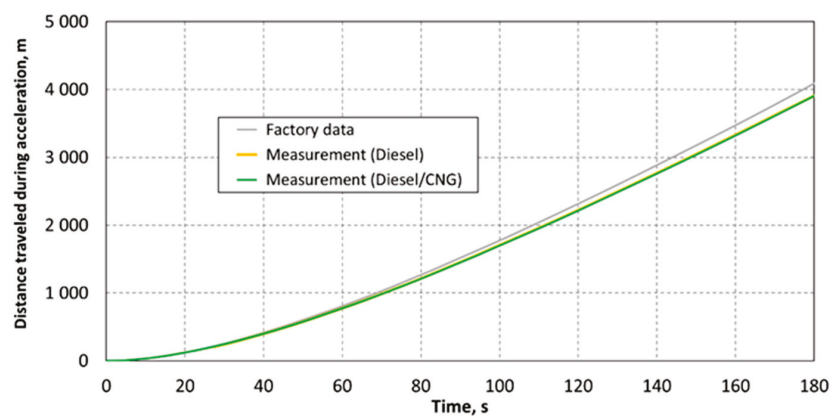
An example of the values from the graphs is presented in Table 5. These values show that the change of fuel supplying the engine does not noticeably affect the traction characteristics of the road semi-trailer truck during acceleration. The results in a graphic form are presented in Figure 12.

Table 5. Parameters of the acceleration process.

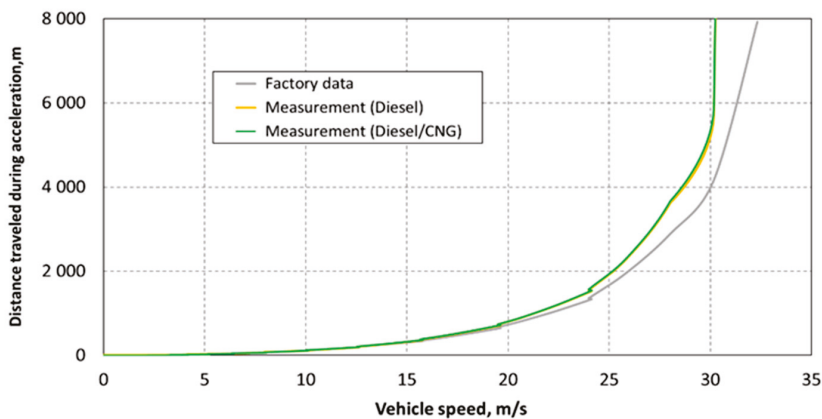
Parameter		Measurment		
		Factory Data	Diesel	Diesel/CNG
Distance to reach speed [m]	50 km/h	248	268	270
	80 km/h	1029	1156	1163
Time to reach speed [s]	50 km/h	29.2	31.5	31.8
	80 km/h	69.2	77.3	77.9
Distance travel time [s]	500 m	44.6	45.8	45.9
	1000 m	68.7	70.6	70.8



(a)



(b)



(c)

Figure 12. Time (a), distance (b) and speed during acceleration of the vehicle set (c).

2.5. Exhaust Emission Tests in Accordance with the Requirements of the UNECE Regulation No. 49

The next step of the research was to determine how the use of a non-factory diesel/CNG dual-fuel installation to supply the engine of a semi-trailer truck unit affects the emission of individual exhaust components, i.e., NO, NO₂, NO_x, CH₄, CO, CO₂, HC and exhaust smoke opacity. To determine these components, the research cycle resulting from the provisions of the UNECE Regulation No. 49 (Table 6), adopted for application in Poland by Resolution No. 97/92 of the Council of Ministers of 31 August 1992, was used.

Table 6. Phases of the test cycle with their importance factors—values required by the UNECE Regulation No. 49 [49].

Phases of the test cycle	1	2	3	4	5	6	7	8	9	10	11	12	13
RPM range	idling		middle			idling			nominal			idling	
RPM during test	600	1300	1300	1300	1300	1300	600	1700	1700	1700	1700	1700	600
Torque (%)	0	10	25	50	75	100	0	100	75	50	25	10	0
Importance factor	1/12	0.08	0.08	0.08	0.08	0.25	1/12	0.1	0.02	0.02	0.02	0.02	1/12

During the tests, commercial diesel fuel from one production batch was used, in accordance with the EN-590 standard (the same that was used to determine the speed characteristics). The normative document (UNECE Regulation No. 49) [49] allows the use of fuel similar to the reference fuel in a situation where the appropriate reference fuel is unreachable. The advantage of choosing such a procedure is the determination of the actual values of the engine operation characteristics with a random fuel composition (from random petrol station). It was also remembered to minimize the impact of fuel properties on the result of the comparative tests, which is why diesel fuel from one delivery was used for all tests.

The measurements were made with the use of the prepared research version of the CNG dosing controller, in which the optimal individual control functions (injection maps, timing, etc.) were determined. The following were adopted as components of the diesel replacement function. The optimal CNG dose: gas injection start, diesel/CNG exchange ratio, correction for pressure in the intake manifold, correction for power developed, correction for temperature and gas pressure. Figure 13 shows the exchange ratio as a function of rotational speed and fuel dose. The visible numerical values on a red background show the percentage of CNG in the total fuel dose (vertical axis) supplied to the engine at various values of the engine RPM and its load represented as the calculated dose of diesel fuel for one work cycle. The diesel/CNG exchange ratio was variable and depended on the operating conditions—it ranged from 20 to 75%.



Figure 13. T Map of Diesel/CNG replacement factor.

The results of the tests are illustrated by comparing them in the following figures of changes in the emission of the components of individual gases: NO and NO₂ (Figure 14), NO_x and CH₄ (Figure 15), CO and CO₂ (Figure 16) as well as HC and exhaust smoke as the coefficient of extinction of absorbed radiation (Figure 17). To facilitate the analysis process and to present the changes in a numerical manner, the results were also presented as relative, comparing the results obtained for diesel/CNG supply with diesel fuel only (on the right side of the figures). The emission of individual gaseous components and the smoke opacity were determined directly at the engine exhaust pipe, just before the exhaust gas treatment system.

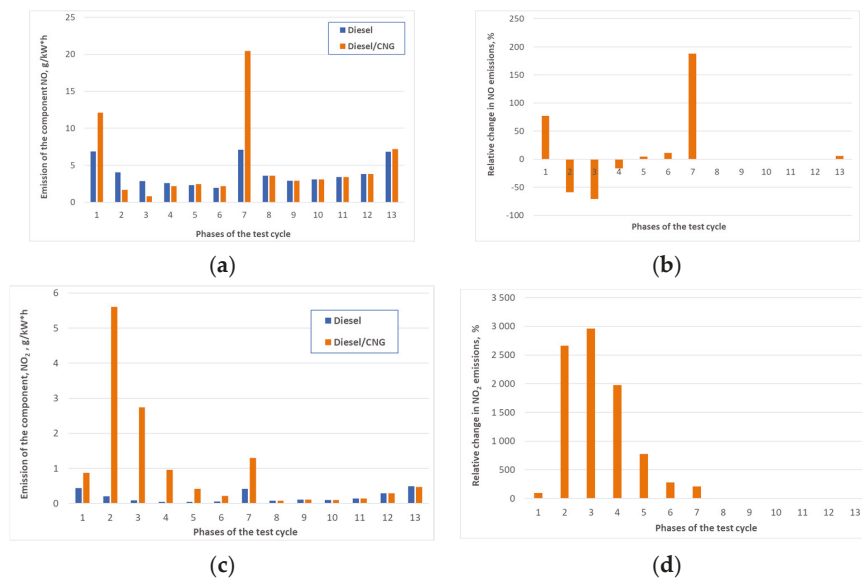


Figure 14. Emission of gaseous exhaust components during test on diesel fuel and diesel/CNG supply for individual phases of the test: (a) emission of NO, (b) relative change in NO emission, (c) emission of NO₂ and (d) relative change in NO₂ emission.

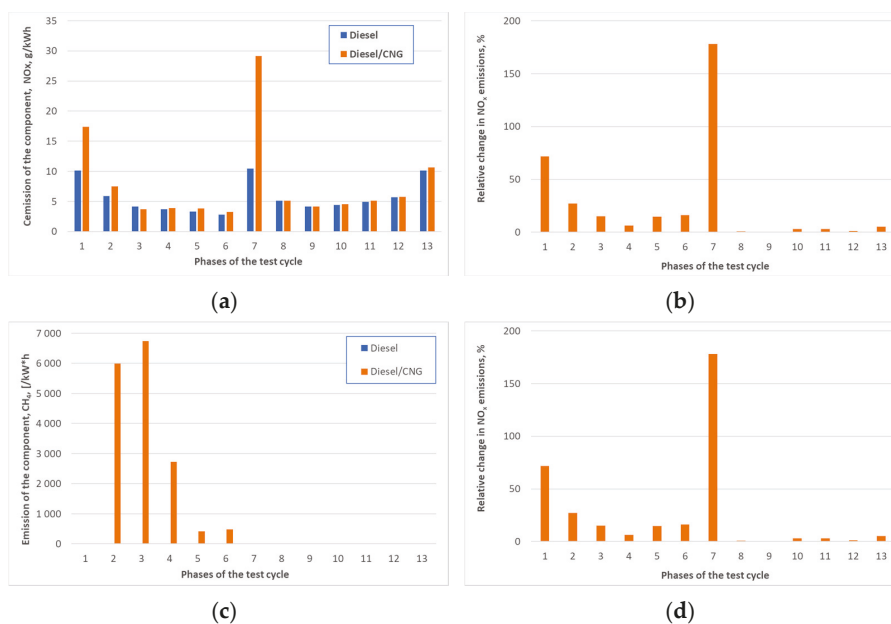


Figure 15. Emission of gaseous exhaust components during the R49 test on diesel fuel and diesel/CNG supply for individual phases of the test: (a) emission of nitrogen oxides (NO_x), (b) relative change in NO_x emission, (c) emission of CH₄ and (d) relative change in CH₄ emission.

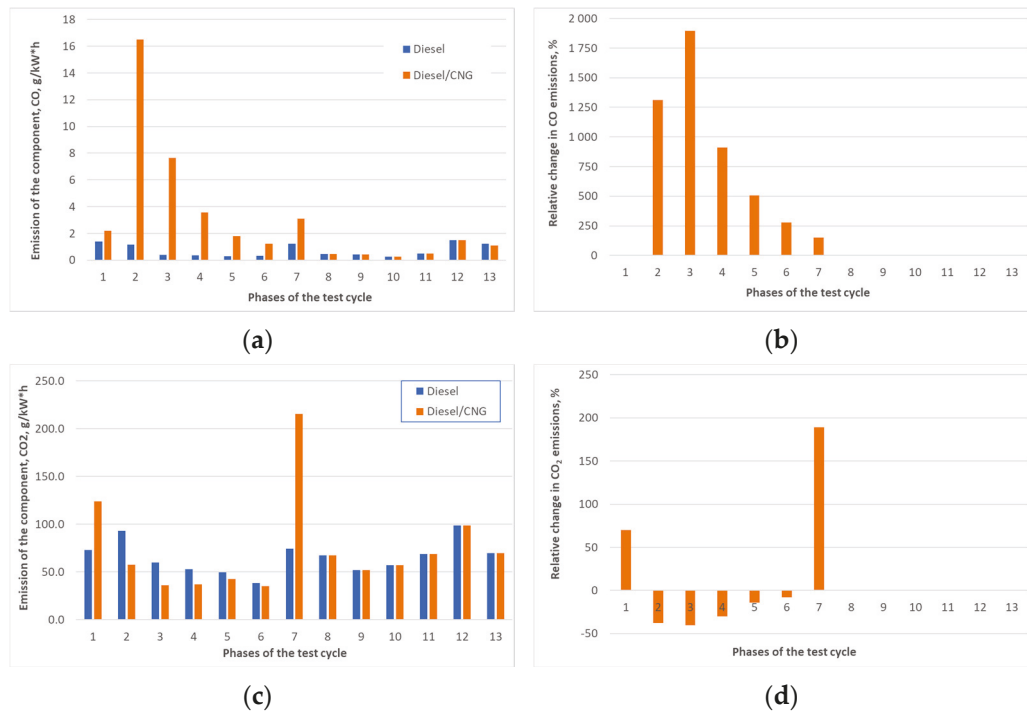


Figure 16. Emission of gaseous exhaust components during the R49 test on diesel fuel and diesel/CNG supply for individual phases of the test: (a) emission of CO, (b) relative change in emission of CO, (c) concentration of CO₂ and (d) relative change in emission of CO₂.

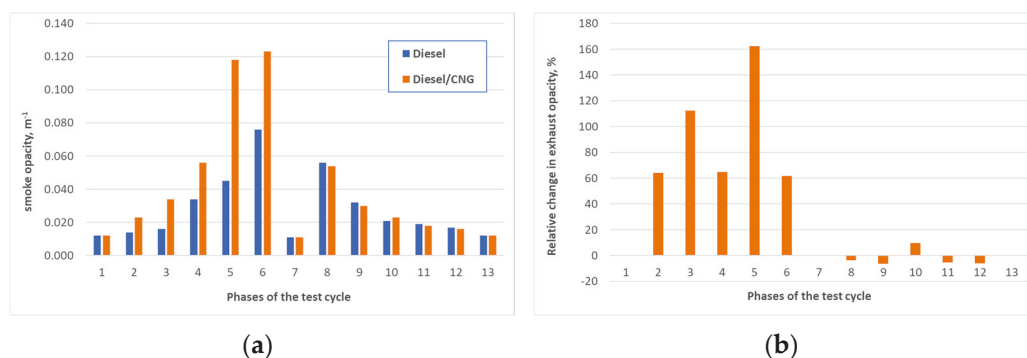


Figure 17. Extinction coefficient of absorbed radiation during the R49 test on diesel fuel and diesel/CNG supply for individual phases of the test: (a) extinction coefficient of absorbed radiation and (b) relative change in the extinction coefficient of absorbed radiation.

Should be noted that the obtained results of emissions of individual exhaust components do not take into account the deterioration factors resulting from the European Union Commission Regulation No. 2018/1832 of 5 November 2018 amending Directive 2007/46/EC of the European Parliament and of the Council, Commission Regulation No. 692/2008 and European Union Commission Regulation No. 2017/1151.

The change in the share of individual nitrogen oxides is illustrated in Figure 14 and also in Figure 15a,b. In the case of NO, at the rotational speed of the intermediate gear and low values of the torque (up to 25%), a significant decrease in the content of this gas in the exhaust gas was observed. After exceeding the engine load of above 50%, this value increases slightly; at a speed of 1700 rpm, no changes in the composition of exhaust gases were recorded. For NO₂, at an intermediate speed, tens of times increase in the content of this gas in the exhaust gas is visible, which decreases with increasing engine torque. This is related to the reduction of CNG in the total fuels dose (sum of diesel and CNG doses). As in the case of NO, at the speed of 1700 rpm, the change in the fueling method did not

significantly affect the exhaust gas composition. The observed changes in NO_x emission at an engine speed of 1300 rpm are in the area of 20–30%.

The change in the share of methane gas in the exhaust is shown in Figure 15c,d. In the case of this gas, while supplying the engine with diesel fuel only, CH₄ emission fluctuated at the level of 0.002–0.004 g/kW·h. The use of diesel/CNG resulted in a significant increase in the concentration of this gas in the exhaust gas up to 7 000 g/kW·h. The reason for this is methane entering the exhaust manifold during the “cylinder air flush” on the intake stroke. In order to minimize or eliminate this disadvantage, it will be necessary to develop further changes including the redesign of valve timing system.

The use of alternative fueling caused an increase in CO content, which decreased with increasing engine load, and no changes were observed at the engine speed of 1700 rpm. For CO₂, CNG supply had a very positive effect on reducing the emission of this greenhouse gas in the area of intermediate rotational speeds and up to 75% of the engine torque value.

The use of dual-fuel supplying resulted in an increase in exhaust gas opacity by 50–150%, depending on the load at the speed of the maximum torque. It is related to the reduction of global AFR and the occurrence of local oxygen deficiencies, which favors the formation of solid particles [50].

Based on the results obtained in individual test phases, the total emission of gaseous components was also determined (the Importance factors in the calculations were shown earlier in Table 6) and compared with the requirements of Euro IV and Euro V (Table 7).

Table 7. The obtained results of emissions of individual exhaust gas components and the applicable Euro IV and Euro V standards limits.

Exhaust Component	e CO	e NO _x	E HC
	g/kW·h	g/kW·h	g/kW·h
Emission on Diesel fuel	0.340	3.221	0.073
Emission on Diesel/CNG	1.66	3.49	0.91
Emission in accordance with Euro IV requirements	1.50	3.50	0.46
Emission in accordance with Euro V requirements	1.50	2.00	0.16

3. Conclusions

The information contained in the previous parts of this work allows for the following conclusions:

1. Factory data (published by the manufacturer) concerning the maximum value of torque and power, as well as the course of their changes in the area of the tested engine RPMs, differ significantly from those determined in the measurements on the dynamometer, both with diesel fuel and Diesel/CNG. This affected the traction calculations, in which the semi-trailer truck (towing semi-trailer) with an engine with factory parameters always had the best results. For this reason, they were treated as achievable values; however, the comparisons themselves were made only for the results of laboratory measurements.
2. Changing the supply fuel (from Diesel fuel to Diesel/CNG) does not significantly reduce the traction characteristics of the vehicle, and the changes are so small that the driver should not notice the differences indicated in the calculations (very similar acceleration values, distance and time traveled, hill-climbing ability, use of the same gear in the same state of movement, etc.).
3. The engine power values determined in the laboratory tests show differences in relation to the factory characteristics, which is the result of the natural processes of engine wear. As a result, irrespective of the fuel used, the power field losses occurred, which are greater when supplied with Diesel/CNG. This can lead to a slight increase in fuel consumption when driving with the use of higher engine speeds—above 1500 rpm for diesel fueling and above 1400 rpm for Diesel/CNG.

4. Similar fluctuations occur with torque. The biggest disadvantages for Diesel/CNG supply, compared to Diesel fuel only, seems the lower torque values, close to 1500 rpm. As calculated, it may result in a temporary reduction of the achieved acceleration value up to 10%.
5. A decrease in the CO₂ content was noticed during with the Diesel/CNG mixture, which is related to the change of the elemental hydrogen and carbon particles ratio in the outgoing exhaust gas. It is a favorable phenomenon in terms of the emission of toxic and greenhouse gases. The changes in the CO₂ content are proportional to the changes in the Diesel/CNG replacement factor.
6. There was a dozen or so percent increase in the content of NO. It involves changing the combustion process to similar to combustion in typical spark ignition engines. The introduction of CNG into the intake manifold reduces the excess air ratio and increases the combustion temperature of the air-fuel mixture. Increasing the combustion temperature results in an increase in NO₂ content, which affects the total NO_x emissions. The vehicle's engine is equipped originally with an SCR system that should reduce NO_x emissions to the required level.
7. A several hundred-fold increase in the methane content in the exhaust gas was noticed. This is related to the contraction of the intake and exhaust systems—air flushing of the combustion chamber. This phenomenon may also cause an increase in CO emissions during homologation tests and, as a result, exceed the limits resulting from the requirements of EURO emissions standards. This conclusion is confirmed by the several dozen-fold increase in the content of CO in the exhaust gas. This results from the changes in the organization of the combustion process (similar situation to the NO_x concentration). By introducing CNG into the intake manifold, the fuel is mixed with air in advance, resulting in an almost homogeneous mixture which is then supplied to the cylinder. After compression, Diesel fuel is injected into this air-CNG mixture in liquid phase. This causes local oxygen deficiencies—which favors incomplete combustion and the formation of carbon monoxide as well as an increase in smoke occurrence during Diesel/CNG supply in the range of 50–120%, depending on the operating conditions. This phenomenon is very unfavorable from the point of view of environmental protection due to the increased emission of solid particles to the atmosphere.
8. There may be several reasons for the increased opacity of exhaust gases:
 - o Reduction of the excess air factor due to the supply of CNG fuel in gaseous form,
 - o Change in the combustion method of the air-fuel mixture, resulting from the shift of the flammability limit and homogenization of the mixture [25,51],
 - o Local oxygen deficiencies resulting from the co-combustion of CNG and Diesel fuel, which leads to a local reduction of the excess air ratio, especially in the area of diesel fuel combustion.

It should be stated that the engine running on a Diesel/CNG supply did not meet the Euro V standard requirements. However, the engine was tested without the exhaust gas treatment system. This may be confirmed by the fact that the Euro V standard limits were not met also when fueling with diesel fuel as the basic fuel. The final verification of the impact of the use of the Diesel/CNG dual fuel supply on the emissions of individual exhaust components should be carried out by certified laboratory with appropriate authorizations. In future research, it is advisable to carry out tests of an engine equipped with a complete exhaust system—exhaust gas catalysts and an SCR-type NO_x reduction system—in order to assess the impact of Diesel/CNG supply on the operation of exhaust gas treatment systems. The conducted tests and analyzes allowed for the assessment of the impact of applying a CNG supply system to a used semi-tractor. The obtained results confirm the purposefulness of using such solutions in order to reduce CO₂ emissions by vehicles used in the transport of goods. The change in the tractive forces is not significant enough to affect the way vehicles of this type are used.

Moreover, in order to optimize the engine in terms of emissions of individual exhaust components, tests of the correct valve timing and optimization of the CNG injection process should be carried out too.

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Abbreviations

Abbreviations

a	longitudinal distance from center of gravity to front axle
a_v	vehicle acceleration
A_x	frontal area of the vehicle body
b	longitudinal distance from center of gravity to rear axle
b_w	front track width
CNG	Compressed Natural Gas
CO ₂	Carbon Dioxide
CO	Carbon Oxide
C_D	aerodynamic drag coefficient
D	performance factor
dv/dt	vehicle acceleration
F_D	aerodynamic drag force
F_{dH}	force acting in drawbar
F_g	grading resistance
F_i	inertia force
F_P	pulling force
F_r	rolling resistance
f_r	rolling resistance coefficient
F_T	traction force
F_{RT}	total resistance force
g	standard gravity
H	vehicle height
h_S	height of the center of mass
h_D	height of the center of wind pressure
h_H	height of towing hitch
i_{gb}	gear box ratio
I_k	moment of inertia of k-th rotating element reduced to the driving wheel axis
i_{dl}	driveline ratio
L	wheelbase
LNG	Liquid Natural Gas
m	vehicle mass
NO _x	Nitrogen Oxides
NO ₂	Nitrogen Dioxide

P_e	engine power
P_t	traction power
n_s	engine speed (rpm)
r_D	dynamic rolling radius
r_K	kinematic rolling radius
S	center of mass
Sa	acceleration distance
SCR	selective catalytic reduction
t_a	acceleration time
T_e	engine torque
T_i	inertia torque
v	vehicle speed in m/s
v_r	relative speed of vehicle and wind velocity component in the vehicle moving direction
W_T	trailer weight
W_V	vehicle weight
Z_1, Z_2	normal force under front and rear wheels, respectively
α	road inclination angle
α_H	tilt angle of drawbar
γ	fill factor of the frontal area of the vehicle body
δ	rotating mass factor
η_{dl}	driveline efficiency
ρ	density of the ambient air
Ψ	road resistance
ω_w	angular velocity of the driving wheel

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Article

Theoretical and Experimental Studies of Uneven Dust Suction from a Multi-Cyclone Settling Tank in a Two-Stage Air Filter

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Abstract: In this paper, the uneven air stream distribution problem of individual cyclones is studied in the multi-cyclones of intake air filters in special vehicles' engines. This problem increases in multi-cyclones, in which several dozen cyclones have a common dust trap from which the collected dust is continuously removed by ejection suction. The aim of this study is the recognition of the theoretical and experimental possibility of reducing the streams' unevenness, which should result in an efficiency increase in multi-cyclone separation. The methods that led to obtaining a relative stream uniformity from the suction of individual cyclones was analyzed. The method for creating equal pressure drops between the suction streams in the channels was used to achieve this goal. For this purpose, the internal structure of the multi-cyclone settler was changed. The multi-cyclone settling tank space was divided by vertical partitions into independent segments. The settling tank segment was then divided with horizontal shelves into suction channels of different heights, which were assigned a specific number of individual cyclones. The suction channels' height was theoretically selected in terms of the equal resistance to air stream flow through the channels. For this purpose, the multi-cyclone dust settler segment model was developed. The theoretically determined suction channel's height was verified by performing experimental flow tests in four (A, B, C, D) dust settler variants. Suction streams of satisfactory uniformity from the cyclones of the variant D settling tank were obtained at a level of 5%. In the next stage, experimental tests of the segment cyclones were carried out with dust before and after the division into suction channels of variant D for the settling tank. A significant increase was achieved from 93.73% to 96.08% in the cyclones' separation efficiency, which were located as far away from the suction stub as possible and led to a reduction in the non-uniformity of cyclone efficiency in the segment. It follows that the multi-cyclone dust settling segment's internal structure change gave the expected results.

Keywords: air filter; multi-cyclone; dust extraction; uneven extraction; air separation efficiency and accuracy; engine

1. Introduction

Military motor vehicles (tanks, armored personnel carriers, infantry fighting vehicles) are mainly operated off-road, in sandy or desert terrain, where mineral dust concentration in the air is particularly high and often exceeds a value of 1 g/m^3 . The authors [1] state that the dust concentration when vehicles are column driving on a sandy area reaches $0.03\text{--}8 \text{ g/m}^3$. During the take-off or landing of a helicopter at an accidental landing site, dust concentration at the height of the tip of the CH-53 helicopter's propeller (0.5 m above the ground) may reach a value of $s = 3.33 \text{ g/m}^3$ [2]. According to the authors [3–6], the maximum dust concentration at the inlet to the inlet system of the vehicle's internal combustion engine usually does not exceed a value of 2.5 g/m^3 . There are two basic dust components: SiO_2 silica and Al_2O_3 alumina, whose composition of the dust reaches 95%, and they are also characterized by having the highest hardness. In the ten-degree Mohs scale, silica has a hardness of seven and corundum is a nine. Thus, their hardness exceeds that of the structural materials used for the components (cylinder

liners, crankshaft journals and camshafts) in internal combustion engines. Moreover, mineral dust grains have very irregular shapes with sharp edges. Most dangerous for two cooperating engine components are dust particles whose diameter d_p is equal to oil film thickness h_{min} between two surfaces at that moment. In an internal combustion engine, there are many elements that cooperate in movement and are lubricated with oil. Oil film thickness depends on the engine operating parameters and oil properties, and therefore they assume various values—most often in the range of $h_{min} = 0\text{--}10\text{ }\mu\text{m}$ [7,8]. Dust sucked in with the air enters the piston, which makes the cylinder and piston top degrade the most, as well as the top piston rings. Abrasive engine component degradation is caused mainly by particles with a size of $1\text{--}40\text{ }\mu\text{m}$, with the most dangerous dust grains in the range of $1\text{--}20\text{ }\mu\text{m}$ [9–11]. Dust grains below $50\text{--}100\text{ }\mu\text{m}$ are simply sucked in with the intake air. It follows that the intake air filters in internal combustion engines should retain all dust grains above $1\text{ }\mu\text{m}$. This is a complicated problem considering that military vehicles are most often equipped with combustion engines with high power and high air demand, with which a significant dust mass enters. For example, the engine of a PT-91 tracked vehicle with a capacity of 700 kW draws in more than $3400\text{ m}^3/\text{h}$ of atmospheric air per hour. If the dust concentration in the air is 2.5 g/m^3 , the motor draws in more than 8 kg of dust per hour along with the air.

To remove such a large dust mass in a short time with the required efficiency of 99.5% and an accuracy above $1\text{ }\mu\text{m}$, two-stage air filters are used, which differ in their principle and operation mode, design, separation barrier type, and operation efficiency. The first stage of air separation is a multi-cyclone, which is a group of individual cyclones arranged in parallel (next to each other), fixed with their ends in common lower and upper plates, which guarantee a common air inlet and outlet—Figure 1.

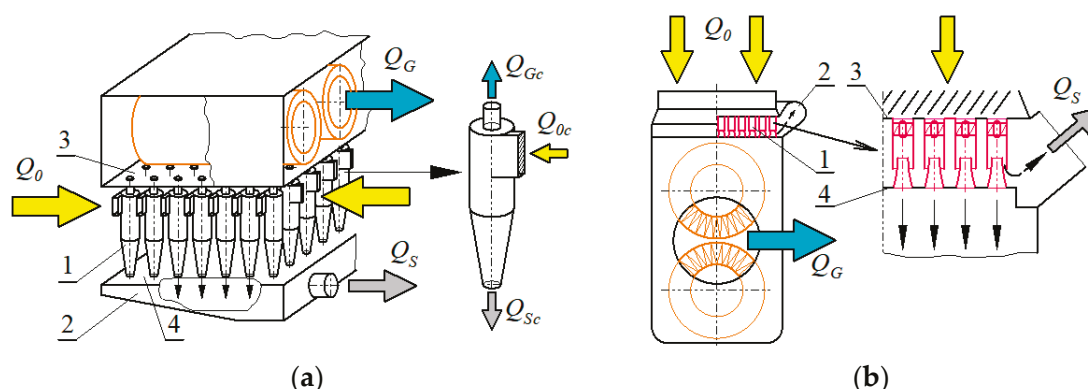


Figure 1. Two-stage air filter with a multi-cyclone: (a) return cyclones with tangential inlet, (b) continuous cyclones, 1—cyclone, 2—dust trap, 3—cyclone mounting top plate, 4—cyclone mounting bottom plate, Q_0 —inlet stream air, Q_G —outlet stream, Q_S —multi-cyclone suction stream.

The second stage of air separation is a partition element (filter cartridge), which, due to space limitations and the desire to increase filter material surface area, is most often made of a pleated filter paper. The operation essence of the “multi-cyclone-porous partition” filter is that the multi-cyclone is characterized by the possibility of separating large dust masses from the polluted air without increasing pressure drop, but with much lower efficiency (87–95%) and accuracy ($d_p > 15\text{--}35\text{ }\mu\text{m}$) than a paper filter. The remaining insignificant dust mass is then directed to the filter element, most often made of pleated paper, having a low and limited absorbency (in the range of $150\text{--}250\text{ g/m}^2$), but characterized by high accuracy (above $d_p = 2\text{--}5\text{ }\mu\text{m}$) and high separation efficiency (above $\varphi = 99.5\%$) [11–16]. As a result, air is supplied to the engine cylinders with the required purity and, at the same time, the operation time of the separation system is extended. Thus, the vehicle’s service interval, which is limited by the achievement of a certain pressure drop value, is the permissible resistance Δp_{fdop} .

Significant dust masses separated by the cyclones are collected in a multi-cyclone dust settling tank common to all cyclones. During eight hours of PT-91 vehicle use, the multi-cyclone filter, operating with 90% efficiency, can separate over 60 kg of dust, the storage of which requires a sedimentation tank with a capacity of over 25 dm³. Storing a large dust mass in a multi-cyclone settler is not recommended due to the unnecessary load on the filter structure and the additional mass in the vehicle. The use of a large-capacity sludge trap increases the filter dimensions, which is especially undesirable for military vehicles. In addition, there are unfavorable phenomena:

- During vehicle shocks and sudden changes in the engine speed: There are rapid changes in the air stream, which causes undesirable suction by the cyclone of dust accumulated in the settling tank,
- As a result of the settling tank completely filling: The dust does not settle in the settling tank, but it is entrained, and thus cyclone separation efficiency decreases.

For this reason, in two-stage multi-cyclone air filters in vehicle engines used in areas of high dust concentration conditions in the air (tanks, infantry fighting vehicles, armored personnel carriers), systematic dust removal from the settling tank in a continuous manner is used. For this purpose, an additional air stream is used—the Q_S suction stream—and the phenomenon of ejection is used to create it. The Q_S flux value is obtained by using special fans or blowers driven by electric motors and powered by an on-board battery, which adversely affects the vehicle's energy balance. For this reason, in many air filter solutions for generating suction stream, appropriate ejectors that use the energy of the compressed air stream are used as a device for forcing the flow [17–20] or exhaust gases out of the engine exhaust system [21–23].

Removal (by suction) of the separated and collected dust from the cyclone settling tank causes a noticeable increase in its separation efficiency, but only to a certain value of the suction degree—which usually does not exceed $m_0 = 10\text{--}15\%$ [24,25]. On the other hand, the available multi-cyclones design solutions analysis and their dust extraction systems from the settling tank, as well as the studies presented in [26], show that the connection of the dust discharge holes of several dozen cyclones into a common and tight space (dust settler) from which extraction takes place pointwise (with two or one stub pipe) and then causes different values (up to 50%) for the suction streams. In a multi-cyclone consisting of a large number (several dozen) of cyclones with a common dust trap, the use of only one or two dust extraction nozzles from the settling tank causes the outlet opening distances of the individual cyclones, which suction away from the extraction system stub, to be unequal. This naturally creates differences between the resistances in the flow and individual stream values that are suctioned out. This may have a negative impact on the proper operation of the multi-cyclone, as it reduces the separation efficiency of individual cyclones and, consequently, of the entire multi-cyclone. According to the author [27,28], this phenomenon results from the uneven distribution of the air stream into individual cyclones, and even the reverse flow occurrence in the cyclones is located on the periphery of the multicyclone, which causes different characteristics in the adjacent cyclones.

The highest suction stream values Q_{Sc} from individual cyclones occur, regardless of the value of the main suction stream from the Q_{SF} settling tank, for cyclones located closest to the dust suction nozzle of the settling tank. As the position of successive cyclones moves away from the suction stub pipe, the suction stream values Q_{Sc} systematically decrease, and for the furthest cyclones they assume values that are lower by 35–50% [26].

The uniform suction stream values from the individual cyclones of a multi-cyclone made of several dozen cyclones can be ensured by using a method that maintains the symmetry of the position of all the cyclones in relation to the suction nozzle, or by a method of creating suction streams that are equal to the resistance by dividing the settling space into independent segments and channels covering a specific number of cyclones. The application of a specific method depends on the number and type of cyclones, space

limitations at the place of multi-cyclone installation, and the possibility of carrying out an appropriate scope of tests.

This work [29] presents a multi-cyclone design intended to study the minimization effect on suction uniformity of individual cyclones by removing the suction stream method from the separation chamber (dust settler). The multi-cyclone consists of 24 through-flow cyclones with an axial inlet, closed in a rectangular cassette, to which a dust trap is tightly attached at the bottom. Dust extraction from the common settling tank takes place through four separate nozzles, shaped in such a way that they form narrowing channels. The same air flow rate was ensured in each stub pipe. Experimental studies of the multi-cyclone for various suction rates, and with a variable air flow velocity through a single cyclone, showed that for a single cyclone, under comparable flow conditions and the same suction degree, the multi-cyclone separation efficiency would be 5–7% lower [29]. This simple dust removal solution from the settling tank significantly reduces the uneven suction from cyclones but does not completely eliminate it.

According to the authors [30], a more effective method of reducing the unevenness of the suction from the cyclones is the multi-cyclone modular structure. The module presented [30] is a cassette with a square cross-section in which there are four symmetrically placed through-flow cyclones with an axial inlet—depicted in Figure 2. A multi-cyclone consists of six such modules, which are arranged in two rows next to each other.

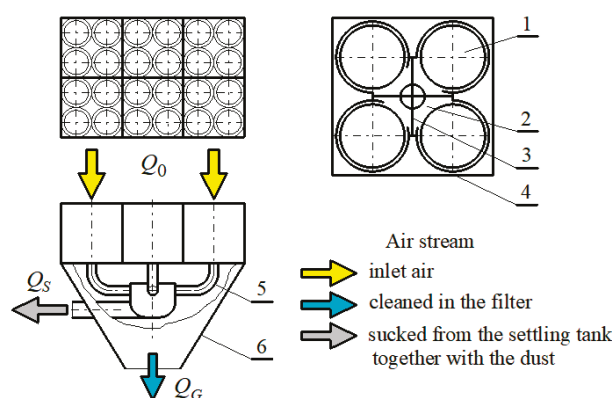


Figure 2. Structural diagram of a modular multi-cyclone: 1—cyclone, 2—dust trap, 3—partition, 4—module cassette, 5—suction pipes, 6—exhaust air collector [30].

The module dust collector has been divided into partitions so that the outlet windows of the individual cyclones are isolated from each other. Each module has a separate settler from which dust is sucked through a centrally located duct. Dust extraction ducts from the six modules are connected into one common collector, from which extraction is performed by one duct. The presented design steps were dictated by efforts to eliminate the interaction of the swirled streams flowing out of the individual cyclones in the module. The performed experimental tests on the module showed that its characteristics did not differ significantly from the characteristics of a single cyclone, both in terms of course and value, which means that the suction from all four cyclones is identical. This means that the pollutants suction method from the dust collector of the module is appropriate. The experimental performance studies of the entire modular multi-cyclone also did not show a decrease in efficiency in relation to a single cyclone. However, due to the use of a rather complex system of suction lines inside the connector, the pressure drop of this multi-cyclone has increased by about 15% compared to a single module.

The problem of organizing the suction process and minimizing uneven suction from the settling tank of the radial filter, which is composed of three identical modules, is presented in this work [31]. The dust separated from the air stream flowing through the filter collects in the separation channels from where it enters the dust trap common for all modules, which is a pipe with a constant cross-section—depicted in Figure 3. The

separation efficiency is determined for the rated conditions of air flow through the filter, which was within the range of 33–40%, and, when compared to the efficiency of a single module (60–61%), is a very small value. The individual performance of each module, as part of the filter assembly, showed that module *a*, which was the closest to the outlet of the sedimentation tank, achieved an efficiency of 64%, and showed that module *b* achieved an efficiency of 43%, while module *c*, the furthest from the outlet of the suction port, only had an efficiency of 20% [31]. Such significant differences in the modules' efficiencies are the result of the divergent suction levels in the modules, and this is due to the unequal distances of their separation channels from the suction stub of the dust settler. The authors [31] solved the problem of equalizing the suction streams in this type of filter by dividing the dust trap into three parts (channels) according to the diagram shown in Figure 3b.

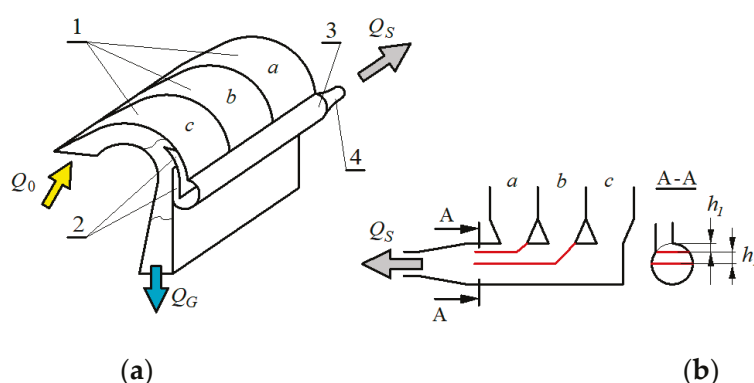


Figure 3. Modular inertial radial filter: (a) functional diagram, (b) dust trap division into suction channels *a*, *b* and *c*, 1—filter modules, 2—separation channels, 3—dust trap, 4—extraction connector [31].

After appropriate selection of the heights h_1 and h_2 , similar values for the suction streams from all modules were obtained. Thus, a significant improvement in the uniformity of operation of all the modules was achieved (maximum efficiency differences did not exceed 7%) and there was a clear improvement (up to 60.6%) in the efficiency of the entire radial filter.

The above-mentioned examples show how important an influence on the operation of the inertial filter is for development of the correct suction system. Regardless of the equalizing suction stream values method, an experimental verification of the system is always required, consisting of measuring the values of the individual suction streams from all cyclones, segments, or multi-cyclone modules.

The problem of dust accumulation in a single cyclone settler and the impact of the settler dimensions on the efficient operation of the cyclones is presented in this work [32–36].

The authors [32], after examining the influence of the dimensions of the cylindrical dust trap (height and diameter) on cyclone operation, showed that the optimal design of the sedimentation tank and the dipleg (the extended part of the bottom part of the cyclone, connecting the cyclone with the dust trap) leads to a better separation efficiency than a conventional cyclone. The authors [33], using the CFD program, investigated the influence that the dipleg shape (cylindrical, inverted cone, conical and diamond) has on the flow characteristics and cyclone operation—depicted in Figure 4.

It was found that the dipleg geometry significantly influences the cyclone pressure drop. The cylindrical dipleg generated the highest pressure drop, while the lowest pressure drop was observed for the cyclone without a dipleg. Among tested geometries, the highest separation efficiency was found for the cyclone with a conical dipleg, and diamond dipleg was the lowest.

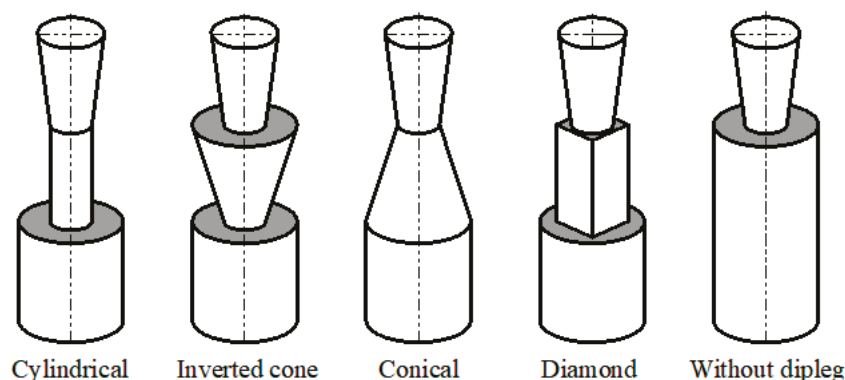


Figure 4. Diplegs and dust settler's geometry. Drawing made on the base [33].

The authors of [34] numerically examined the particle separation process inside two cyclones extended with a dipleg. They found that the dipleg length significantly influences the cyclone's separation efficiency by providing more separation space and minimizing re-lift of the separated particles.

Paper [35] presents research results on three cyclones separation efficiency with different lengths of cylindrical diplegs and with a dust trap of a constant shape. It was shown that tangential velocity, axial velocity and turbulent kinetic energy in the settling tank decreased significantly when the dipleg length was extended. This means that a dust collector with a dipleg can effectively prevent the re-entrainment of already separated dust. In addition, the elongated part of dipleg increases dust separation space, and thus separation efficiency increases, at the expense of a slightly increased pressure drop.

Authors [36] investigated experimentally cyclones with three different geometries of dust outlet from cyclone: a cyclone with a dust trap, with a cone at cyclone outlet, and a cyclone with a cylindrical dipleg. They found that in the case of a cyclone with only a dust trap, the swirling stream passes from the cyclone cone to the settler, which will cause the collected particles to be entrained again. In case of cyclone with a tip cone, efficiency improvement in cyclone has been achieved. They showed that separation efficiency can be significantly improved by changing dust collector geometry.

One of the methods of increasing cyclone separation efficiency is to attach an additional element (cone) at the dust outlet from the cyclone to the settling tank. The task of this element, also known as the apex cone, is to limit the secondary circulation of the separated dust particles. It is very important to precisely select cone diameter, the apex angle and its proper position. Incorrect selection of these parameters may block the bottom outlet of the cyclone and make it difficult for dust to fall into the tank, and thus reduce cyclone separation efficiency. Therefore, [37] influence of fifteen variants of the geometrical configurations of the counter-cone on cyclone efficiency was analyzed. The cone geometry was determined by the following parameters (Figure 5): cone diameter D_s , apex angle α , and distance from the cone base to the cyclone bottom outlet with diameter B_h , for cyclone outlet diameter B .

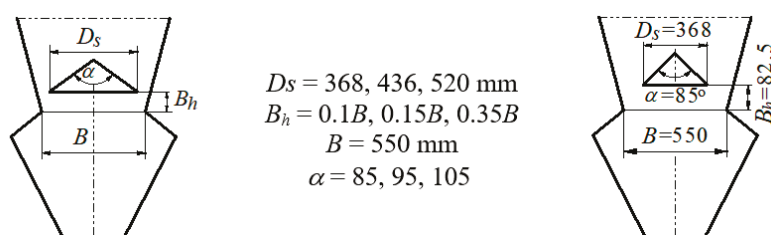


Figure 5. Cone geometry in lower part of the cyclone. Main dimensions: D_s —cone diameter, B_h —distance from the cone base to the cyclone bottom outlet, B —cyclone outlet diameter, α —cone apex angle.

It was found that cone use in each of investigated geometric variants led to cyclone efficiency increase. The results show that the optimal position of the cone (assuming separation efficiency is maximized) should be above the bottom outlet of the cyclone at a distance of $B_h = 0.15B$ (82.5 mm). The apex angle of the cone should be $\alpha = 85^\circ$. Improvement in separation efficiency was observed for particles smaller than 60 μm , and, in particular, for particles up to 15 μm in diameter.

The authors of [38–40] investigated the apex angle of the counter-cone at the dust outlet from the cyclone in the range of $\alpha = 40\text{--}80^\circ$. They found that the optimal value of the angle regarding the maximum separation efficiency is $\alpha = 70^\circ$. Moreover, they observed that increasing the gap between the cone and the wall of the cyclone conical part increased the number of small particles that entered the dust tank.

Two design variants of the cone were analyzed [36]. In the first variant, the cone (apical angle $\alpha = 90^\circ$) was located under the cyclone lower outlet. In the second variant, the cone ($\alpha = 120^\circ$) was located above the cyclone lower outlet. The latter solution led to an increase in the separation efficiency by 2%, and an increase in the pressure drop from about 1200 Pa to 1400 Pa.

The author of [41] compared with CFD the two suggested locations of the cone (below the outlet of the cyclone and in cyclone conical part). He concluded that placing the cone below the cyclone conical portion is more beneficial, as it helps to increase the particle separation efficiency.

The above-mentioned examples show that single cyclone operation is significantly influenced by appropriate dimensions selection of dust collector and attachment of an additional element (cone) at cyclone dust outlet. However, the modifications of cyclones presented in the literature were made in single cyclones and tested mainly towards their use for industrial purposes. Optimizing settling tank dimensions does not take dust extraction into account.

Air stream even distribution problem into individual cyclones in multi-cyclones of vehicle air filters is not fully recognized and sufficiently described in available literature. The problem is increasing the number of multi-cyclones where dust suction is used. Although ejection suction is commonly used for current dust removal methods from inertial filters, this problem was not accompanied by the development of a theory that would fully explain the phenomena occurring during air flow through the dust collector. Therefore, it is advisable to work on improving the organization of the dust suction system from the multi-cyclone settling tank, which will increase multi-cyclone efficiency. As a result, the dust mass flowing with the air to the second separation stage will be reduced, which, with the limited absorbency of the porous partition, will extend the filter's service life.

The aim of this study is the recognition of the need to reduce the theoretical and experimental possibility for the unevenness between the suction streams from the multi-cyclone settling tank by changing its internal structure, which consists of dividing the multi-cyclone settling tank space into independent segments, and dividing each segment into suction channels covering a specific number of cyclones. Dust settling tank segment model development and the application of the method of equal pressure drop of the sucked streams in the channels will enable preliminary determination of their height. Model calculations will be verified during flow experiments. Final verification of the dust collector segment will be carried out during the cyclone effectiveness tests with the test dust.

2. Model and Experimental Research

2.1. Possibilities of Organizing Dust Extraction Process from Multi-Cyclone Filters

As experience so far has shown, the main reason for the suction streams' unevenness from the cyclones may be the differentiation of the pressure drop differentiation of these streams in the filter settling tank and on the cyclone-suction stub section, for individual cyclones. This results mainly from the unequal lengths of the above-mentioned sections,

determined by the cyclone position in relation to the suction nozzle and the impact of the settling tank walls on the suction streams from cyclones located in their vicinity, as well as the possible presence of turbulence areas in the total suction stream from the settling tank.

In the theoretical and experimental studies [18,21,42,43], three basic methods were proposed to obtain a relative uniformity of the suction streams from individual cyclones:

- maintaining the symmetry of all cyclones' positions in relation to the suction nozzle;
- creating equal pressure drops for suction streams through appropriate outflow throttling from those cyclones for which the pressure drop is lower than the remaining ones;
- a combined method consisting of an appropriate constructional combination of both of the above-mentioned procedures.

The only possibility to obtain position symmetry is to place all the cyclones along the circle circumference—depicted in Figure 6.

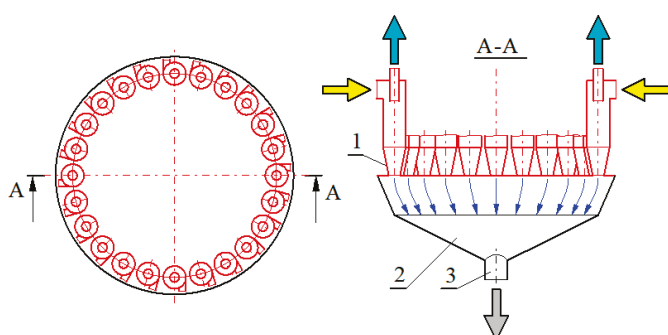


Figure 6. Diagram of the sedimentation basin maintaining position symmetry of all cyclones in relation to the suction stub: 1—cyclone, 2—sedimentation basin, 3—suction pipe stub.

In the central part of circular plate, the more unused area that remains, the greater the number of cyclones that are included in the multi-cyclone. Due to the symmetrical structure of the multi-cyclone with a large number of cyclones, the settling plate on which the cyclones are mounted adopts disproportionately large dimensions. There is usually no excess space in a vehicle's engine compartments, therefore the practicality of this type of multi-cyclone structure is low.

The pressure drop equalization of the cyclone suction streams by throttling and directing selected streams can be performed by the following methods:

- by placing an intermediate inlet to the suction nozzle in the multi-cyclone sedimentation tank;
- by dividing the settling chamber into independent channels;
- by means of throttling elements for the suction stream flow located at the outlets of selected cyclones.

As can be seen from the above, the first two methods require special design form development of settling chamber. In the last one, the shape of the settler does not play a decisive role. In any case, however, it is necessary to carry out the complex and repeated experimental studies.

An intermediate inlet of suction streams from the cyclones in the dust collector can be created by introducing a shaped partition (deflector) into the sedimentation basin, above the suction stub pipe inlet. It should ensure the desired distribution of the current lines for the suction streams flowing out of the individual cyclones. A proposal for such a solution is shown on the example of an axisymmetric settler—depicted in Figure 7.

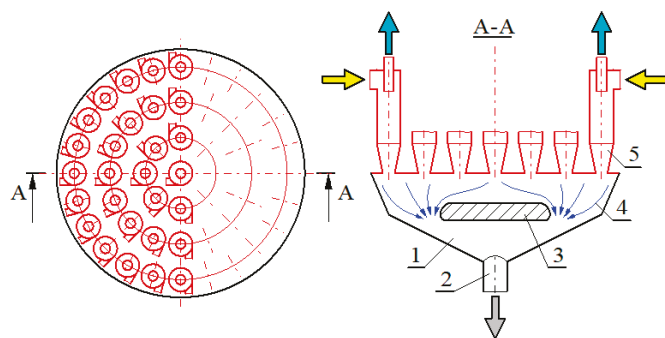


Figure 7. Sedimentation tank diagram with an intermediate inlet to the suction pipe: 1—sedimentation tank, 2—suction hose connector, 3—partition (deflector), 4—stream lines, 5—cyclone.

The deflector dimensions and shape are selected so that the distances of the individual cyclones from the intermediate inlet differ as little as possible. If, after taking the measurements, it turns out that deflector shape selection does not ensure the same degree of suction from all cyclones, which may be particularly difficult to meet in the case of cyclones located centrally in the multi-cyclone, then the course of the streamline can be corrected by making a bore in the center of the deflector, through which an additional air stream will enter, and thereby “enhancing” the suction from the central zone.

It should be emphasized that the described design solution is difficult to develop as it requires the use of a complex computational model, as well as complex and time-consuming experimental research. It can only be recommended for a large production series of filters.

Another method leading to value uniformity of the suction streams from individual cyclones consists of dividing the settling tank with several partitions, and thus creating independent and isolated suction channels inside the settling tank, which are assigned to specific groups of individual cyclones—depicted in Figure 8. The channel heights h_1 , h_2 , h_3 , that are formed by partitions in settling tank should be selected in terms of obtaining the same pressure drop throughout each channel. Achieving full resistance compliance, by selecting a channel’s geometry solely by computation, is not possible due to the rather complex flow. Therefore, the final channel height selection should be made during the experimental tests.

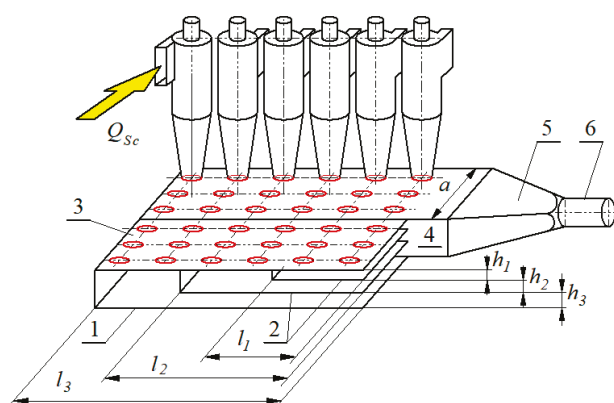


Figure 8. Schematic division diagram of the settling tank into independent channels: 1—cyclone mounting plate, 2—settling tank bottom, 3—partitions, 4—vertical wall, 5—transition channel, 6—suction connector.

Each individual suction channels in the settling tank should include a small group of cyclones. There may be 3–4 cyclones along channel width, and 1–4 along the length, depending on the channel height at the point where the suction from the cyclones takes place. If there are more cyclones along the filter width, then the sedimentation chamber

should be divided (Figure 8) with a tight vertical wall so that the required number of cyclones is contained within the segments resulting from the width division. Each segment should be assigned a separate suction port. This is practically an implementation of a modular filter structure, but in this case, the module may contain many more cyclones than previously described.

This method of shaping the settler, unlike the previous one, is much easier to design. Independent channels creation is particularly suitable for modifying the settling tanks in existing, operated filters, which do not provide the expected separation efficiency.

Yet another method aimed at achieving even suction may be the use of appropriately selected throttling elements (conical adapters) for the suction stream flow, mounted directly on the outlets of the cyclone suction stream—depicted in Figure 9.

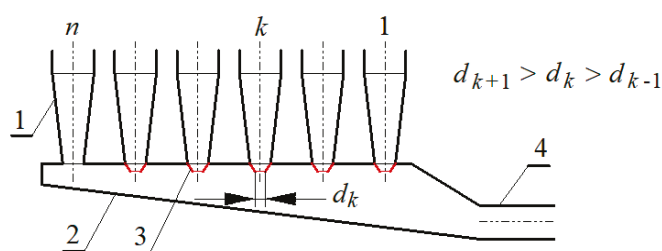


Figure 9. The principle of selecting a choke adapter: 1—cyclone, 2—sedimentation basin, 3—cap, 4—suction port.

By changing aerodynamic resistance with its help, it is possible to regulate the size of the suction stream of the cyclone. The cap diameter must be selected depending on the position of a single cyclone in relation to the suction nozzle so that the same suction jets are obtained in all cyclones.

On the cyclones proceeding from the last to the first, caps of decreasing diameter are placed, and d_k is introducing an increasing level of throttling of the suction stream. It should be noted that the cap's dimension selection can be very laborious, as it requires many consecutive analytical calculations alternating with their experimental verification. As can be seen, this method is suitable for use in filters with a small number of cyclones and settling tanks of any size and shape.

Specific method choice and method used depends not only on their efficiency, but also on the form of the multi-cyclone and the spatial limitations at the place of its installation, as well as on the possibility of carrying out an appropriate scope of tests. It is very likely that the combined use of two or more of these structural measures will be the most effective.

Figure 10 shows an example of a combined method used to obtain a relative uniformity of the suction streams from individual cyclones in the multi-cyclone. The multi-cyclone cyclones are distributed over the circuits of four coaxial circles. The axial-symmetric sedimentation tank has one central suction port. Within a group of cyclones arranged in a given circle, their position is approximately the same in relation to the pipe stub, but different in each group. Between the outer and inner walls of the settling tank there is a partition, due to which two annular channels are formed—the outer and the inner one—which carry away the suction streams from the two pairs of cyclone units. In the partition wall, above the inlet of the suction connector, there is an opening for removing the suction stream from the internal channel.

To equalize streams, throttle caps can be placed on the cyclone outlets from the inner circle. It is a relatively easy procedure as all the caps will be the same due to the symmetry. The suction stream values from the external and internal channels must be proportional to the number of cyclones they “serve”. This requirement can be met by placing an appropriately selected orifice in the opening.

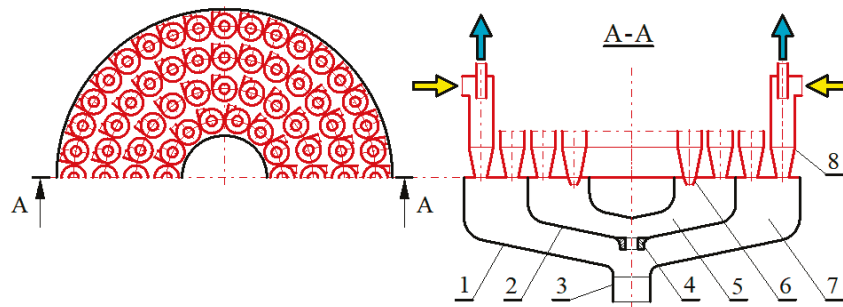


Figure 10. Suction system concept according to the combined method: 1—settling tank bottom, 2—partition, 3—suction port, 4—flange, 5—external channel, 6—internal channel, 7—choke cap, 8—cyclone.

The advantage of the described multi-cyclone concept is the use of the cyclone-mounting plate surface, with their number of the order—one hundred or more—and small settling tank height, i.e., the relatively small dust collector dimensions.

2.2. Dust Extraction System Concept from a Multi-Cyclone

The decision was made to design a dust extraction system from the multi-cyclone dust collector on the basis of a two-stage filter (multi-cyclone-partition) from a T72 vehicle. The choice of this multi-cyclone for testing is supported by the fact that it is used in heavily-used vehicles, and thus it is possible to verify how the design was carried out experimentally. To facilitate the analysis of the results, the tested multi-cyclone was designated MT.

To ensure streams uniform suction from individual MT multi-cyclone cyclones, the most appropriate method seems to be to change the internal structure of the multi-cyclone dust settler by dividing the dust settler space into segments with vertical partitions. The segment space should be divided by horizontal shelves into independent and isolated suction channels, to which specific groups of individual cyclones should be assigned.

Figure 11 shows the idea of dividing the MT multi-cyclone dust settler into segments, and then the segments into suction channels.

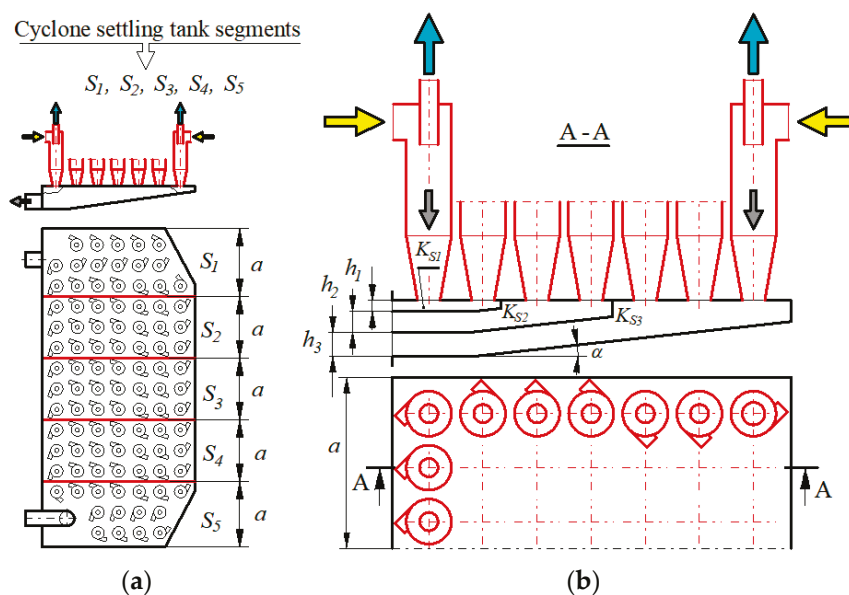


Figure 11. Dust suction concept from MT multi-cyclone: (a) schematic division diagram of multi-cyclone dust settler into segments, (b) chamber dividing principle of segment dust settling chamber into suction channels.

Dust settling chamber of the MT multi-cyclone is divided into widths with vertical internal walls (partitions) in five equal and independent parts (S segments) with a width of a . Each segment (module) includes three columns of cyclones (seven cyclones in a column). The chambers of the segment settlers were divided by horizontal (parallel to each other) partitions into three separate suction channels (K_{S1}, K_{S2}, K_{S3}) with different heights h_1, h_2, h_3 . A segment may include a different number of suction channels, and a different number of cyclones may be assigned to each channel.

The basic problem when dividing the segment settling chamber into suction channels is the setting of horizontal partitions at appropriate heights $h_1, h_2, \dots, h_i$. The channel heights of the dust settler segment should be determined by calculation, and then verified experimentally during the tests of the actual segment in dust settler.

2.3. Dust Settler Model Tests

2.3.1. Model Research Purpose and Subject

The research aim was to determine the height of the suction channels formed by the horizontal partitions in the segment's dust settling chamber. The subject of the research was a model of the segment from MT multi-cyclone dust settler, including 18 cyclones arranged in three identical columns.

2.3.2. Dust Collector Segment Model

The schematic segment diagram in the motor vehicle air filter multi-cyclone settling tank (Figure 12) can be used for the calculation of different settling tanks, but with the same construction principle, i.e., with a division into segments and suction channels.

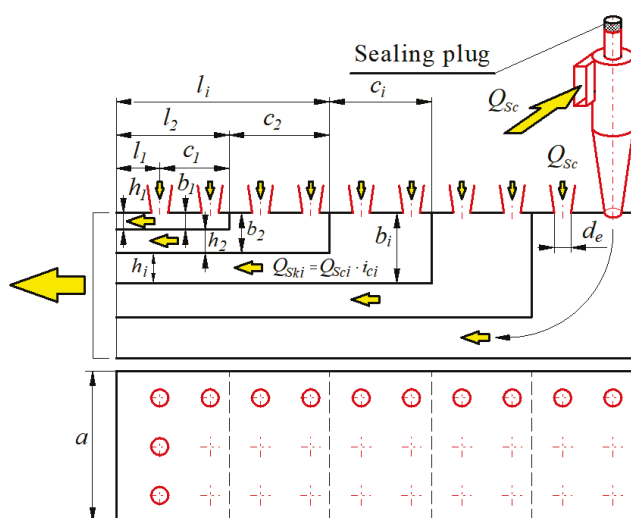


Figure 12. Dust collector segment schematic diagram: a —segment width, b —chamber height, c —chamber length, l —exhaust channel length, d_e —discharge opening diameter, h —settling tank total height, h_i — i -th channel height, i —channel sequence number.

The following assumptions were made during the calculations:

- the suction channels should be numbered in sequence from the cyclone mounting plate;
- the relevant channel walls are parallel to each other;
- channel No.1 sucks air from six cyclones arranged in the first two rows, channel No. 2 sucks air from six cyclones from the next two rows, and channel No. 3 is assigned the last three rows of cyclones—depicted in Figure 12;
- the suction channel is divided into a chamber of length c_i and an exhaust channel of length l_i , where i stands for the sequence number of the channel;

- the boundary of the chamber division and the inlet channel is the plane where the chamber space ends with a cross-sectional step in the flow direction from the chamber height b_i to the height of the channel h_i ;
- in channel one, due to the lack of a step (the height of the channel and the chamber is the same $h_1 = b_1$), the cross-section of the suction channel in axis plane of the first-order cyclones was assumed as a chamber border and outlet channel—depicted in Figure 12;
- all cyclones are the same, and the diameter of the cyclone dust extraction (suction) opening is d_e .

The calculations are based on a simplification in relation to the real object of the dust settler segment—fragments inclination to the lower segment walls has not been taken into account (Figure 11b). Therefore, the dust collector is a low-height, rectangular cassette. As a result, the segment chamber has a different volume than in the real system, and the channel is rectilinear. In the case of inflow of an air stream from the cyclones (diameter d_e) to the chamber, a value change of the cross-sectional area of the flow has a major influence on the pressure loss. The cases cited in the literature consider, first of all, the value change of the flow cross-sectional area, regardless of the chamber shape and volume [44,45]. Thus, replacing the actual form of the chamber with a cuboid does not introduce a significant error. The assumption of a rectilinear channel (omitting the flow direction change in the channel) is justified because this change is very gentle, and, in the outlet channel of the first suction channel, the flow direction does not change at all. The small angle of the lower walls' inclination ($\alpha = 6^\circ$) allows for the same height of the channel along its entire length. With such a slight deviation in the flow direction, pressure losses are negligible. If the chamber height and the duct in the first segment are the same, there is no pressure loss in that segment due to air outflow from the chamber into the duct.

Due to the low flow velocities occurring in the suction channels (below 4 m/s) and very small relative pressure drops in relation to the environment, the air density value in each case can be assumed to be the same—equal to surrounding air density.

The air flows directly into each suction channel chamber from the cyclones with suction openings with a diameter of d_e and an air stream (with separated dust particles) of size:

$$Q_{Ski} = Q_{Sci} \cdot i_{ci}, \quad (1)$$

where: Q_{Sci} —air stream sucked from a single cyclone, i -th suction channel, and i_{ci} —number of cyclones covered by the suction channel.

Air flow Q_{Sc} suctioned from a single cyclone is determined by:

$$Q_{Sc} = \frac{Q_G \cdot m_0}{i_{cm}}, \quad (2)$$

where: Q_G —filter outlet air stream, m_0 —suction level, i_{cm} —cyclone number in the multi-cyclone.

Main issue in settling tank flow calculations is to determine the pressure losses in the suction channels of the settling tank segment. Calculations considered two main causes of pressure losses occurring in the real object—i.e., the suction channel. These are the swirls of the air stream that occur from sudden changes in the flow channel cross-section and the air friction against the channel walls.

During air stream flow through the suction channel from the dust settling tank segment, three types of pressure losses were assumed—depicted in Figure 13:

Δp_e —local pressure loss occurring with Q_{Sc} streams inflow from the cyclones to the segment chamber, resulting from a sudden increase in the channel cross-section. The air from the cyclones enters the duct chamber through circular openings with a diameter d_e in the upper wall of the chamber and fills the chamber's cuboidal space. This results in a local pressure loss due to the reduction of the jet velocity and streams mixing.

Δp_w —local flow Q_{Sk} pressure loss flowing into the channel, wherein the inlet is located in the side chamber wall. The transverse dimensions of the outlet duct (inlet) are much smaller than the side wall dimensions of the chamber. In this case, it can be considered that there is a flow through the conduit with a sudden cross-sectional narrowing.

Δp_k —pressure loss resulting from friction during stream flow through a channel with a constant cross-section.

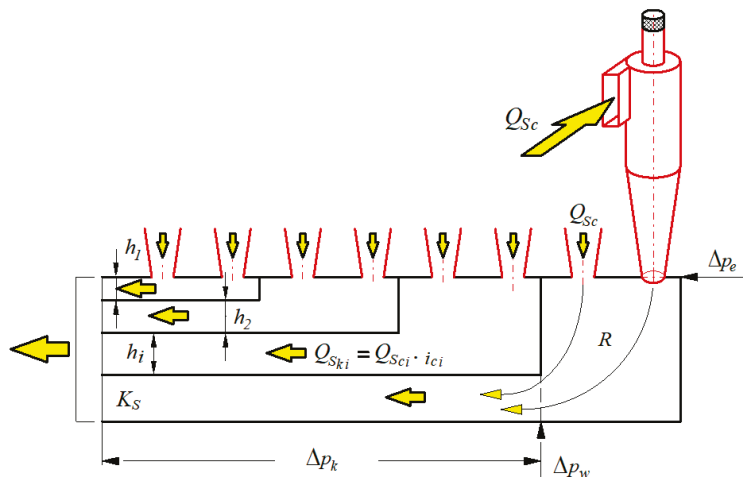


Figure 13. Schematic diagram of the air filter multi-cyclone settling tank segment: K_S —suction channel, R —chamber.

The total pressure loss during the air flow through the i -th suction channel is determined by:

$$\Delta p_{Ksi} = \Delta p_{ei} + \Delta p_{wi} + \Delta p_{ki}, \quad (3)$$

The local pressure loss Δp_{ei} at the air stream Q_{Sci} inflow into the chamber of the i -th channel can be represented by:

$$\Delta p_{ei} = \zeta_e \frac{\rho}{2} u_e^2, \quad (4)$$

where: ρ —air density, ζ_e —pressure drop coefficient, v_e —air flow velocity to the suction channel chamber—air velocity in the outlet opening of the cyclone suction (bleed) stream with a diameter d_e calculated from:

$$u_e = \frac{Q_{Sc}}{A_e}. \quad (5)$$

However, the pressure drop coefficient ζ_e should be determined from:

$$\zeta_{ei} = \left(1 - \frac{A_e}{A_{ci}} \right), \quad (6)$$

where: A_e —the cross-sectional area of suction stream outlet opening (diameter d_e) from the cyclone, A_{ci} —chamber upper wall area (Figure 14), in which the cyclones are attached, calculated from:

$$A_{ci} = a \cdot c_i. \quad (7)$$

The local pressure loss during the suction stream flow Q_{Ski} through the rectangular channel opening with dimensions a and h_i (Figure 11) is represented by:

$$\Delta p_{wi} = \zeta_{wi} \frac{\rho}{2} u_{ki}^2. \quad (8)$$

where: v_{ki} —air velocity in the duct of height h_i and width a , ξ_{wi} —pressure drop coefficient, depending on the ratio of the transverse area of the outlet duct opening A_{hi} to the area of chamber front wall A_{bi} [44]:

$$\xi_{wi} = \frac{1}{2} \left(1 - \frac{A_{hi}}{A_{bi}} \right). \quad (9)$$

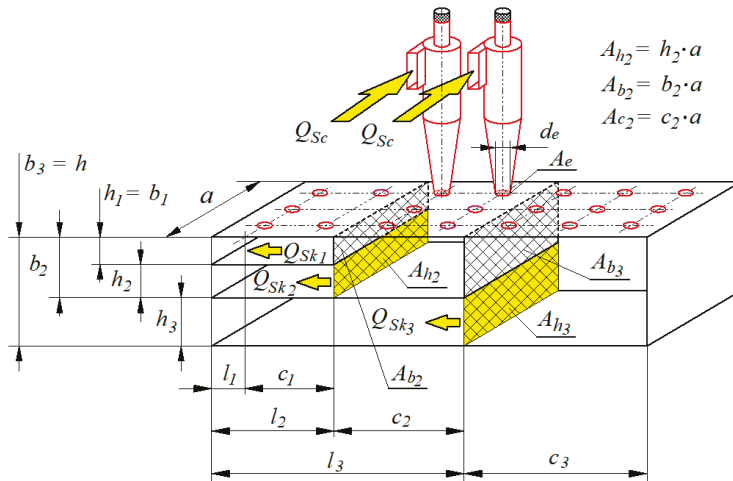


Figure 14. Dust collector segment model with three suction channels.

The air velocity in a duct of height h_i and width a should be calculated from:

$$u_{ki} = \frac{Q_{ski}}{a \cdot h_i}. \quad (10)$$

The pressure loss in the outlet channel is captured by:

$$\Delta p_{ki} = \lambda_{ki} \frac{\rho}{2} u_{ki}^2 \frac{l_i}{d_{ri}}, \quad (11)$$

where: λ_{ki} —duct pressure loss coefficient, l_i —duct length, d_{ri} —equivalent duct diameter.

To calculate the coefficient λ_{ki} the Mises formula was used, which makes its value dependent on the channel transverse dimensions, flow type, and material roughness from which the channel walls are made [45]:

$$\lambda_{ki} = 0.0096 + \sqrt{\frac{2k_R}{d_{ri}}} + 1.2 \sqrt{\frac{2}{Re_{ki}}}, \quad (12)$$

where: k_R —coefficient characterizing channel walls roughness, Re_{ki} —Reynolds number defined by:

$$Re_{ki} = \frac{d_{ri} \cdot u_{ki}}{\nu}, \quad (13)$$

where: ν —air kinematic viscosity.

Due to the exhaust duct cross-section's rectangular shape, its equivalent diameter d_r should be calculated according to:

$$d_{ri} = \frac{2 \cdot a \cdot h_i}{a + h_i}. \quad (14)$$

2.3.3. Model Research Methodology

Flow model tests consist of making total pressure losses calculations in each suction channel separately, assuming the initial assumption of the channel numbers k and their equal heights h_i using:

$$h_1 = h_2 = h_i = \frac{h}{k}. \quad (15)$$

The task is to calculate such heights $h_1, h_2, h_3, \dots, h_i$ and segmented suction channels (with a constant segment width a and a constant height h of the settling chamber), so that the total pressure losses in the suction channels on the section between the outlets from the cyclones and the outlet from the channel suction were equal to:

$$\Delta p_{Ks1} = \Delta p_{Ks2} = \Delta p_{Ks3} \dots = \Delta p_{Ksi}. \quad (16)$$

The calculations were made using the successive approximations method, a calculation program specially designed for this purpose—depicted in Figure 15.

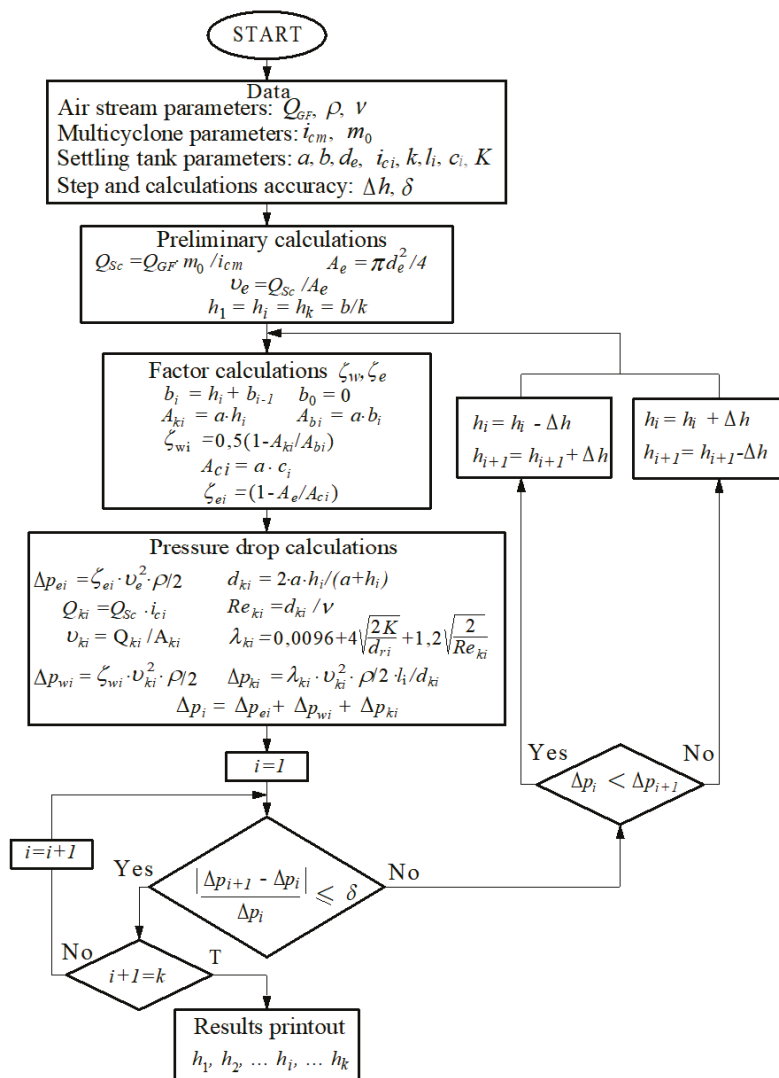


Figure 15. Algorithm for calculating channel heights in the suction chamber of the dust collector segment.

Table 1 presents data for the numerical calculations, including the parameters of the air stream, MT multi-cyclone, and segmented dust trap.

Table 1. Data for numerical calculations for the segment of the MT multi-cyclone dust settler.

Parameter Name		Value	
Air stream	Air flow leaving the multi-cyclone	$Q_{GF} = 3400 \text{ m}^3/\text{h}$,	
	Air density	$\rho = 1.29 \text{ kg/m}^3$	
	Kinematic air viscosity	$\nu = 1.7 \times 10^{-5} \text{ m}^2/\text{s}$	
Multicyclone	Number of cyclones in a multi-cyclone	$i_{cm} = 96$	
	Suction degree	$m_0 = 8\%$	
Segment dust collector	Segment width	$a = 0.095 \text{ m}$,	
	Settling tank height at the outlet	$h = 0.057 \text{ m}$	
	Diameter of cyclone discharge opening	$d_e = 0.019 \text{ m}$	
	Number of cyclones assigned to the channels	No. 1	$i_{c1} = 6$
		No. 2	$i_{c2} = 6$
		No. 3	$i_{c3} = 9$
	Number of established channels		$k = 3$
	Exhaust duct length	No. 1	$l_2 = 0.024 \text{ m}$
		No. 2	$l_2 = 0.096 \text{ m}$
		No. 3	$l_2 = 0.192 \text{ m}$
	Suction channel chamber length	No. 1	$c_2 = 0.072 \text{ m}$
		No. 2	$c_2 = 0.096 \text{ m}$
No. 3		$c_2 = 0.144 \text{ m}$	
Suction channel wall roughness		$K_S = 0.5 \times 10^{-8} \text{ m}$	
Calculation step and accuracy	Calculation step	$h = 1 \text{ mm}$	
	Calculations accuracy	$\delta = 5\%$	

The numerical calculation results, including the channel's height at the segment outlet in the MT multi-cyclone dust settler, are presented in Table 2.

Table 2. Numerical calculations results for dust settling tank segment in multi-cyclone MT.

Channel Number	Air Velocity in Exhaust Duct	Reynolds Number	Local Pressure Loss Δp (Pa)			Total Pressure Loss	Calculations Accuracy	Channel Height
	v_{ki} (m/s)	Re	Δp_{ei}	Δp_{wi}	Δp_{ki}	Δp_{Ksi} (Pa)	δ (%)	h (mm)
1	5.96	6679	3.445	0	1.256	4.701	$\delta_{1-2} = 4.46$	12
2	2.65	6088	3.731	0.697	0.483	4.911	$\delta_{2-3} = 0.386$	19
3	2.27	8158	3.731	0.753	0.408	4.892	$\delta_{1-3} = 4.06$	26

Obtained channel heights h_i at dust multi-cyclone segment outlet MT have the following values: $h_1 = 12 \text{ mm}$, $h_2 = 19 \text{ mm}$, $h_3 = 26 \text{ mm}$.

The dimensions that were determined in this way were used to construct the dust settling tank segment intended for further (experimental) flow tests—depicted in Figure 16.

2.4. Dust Settler Segment Experimental Tests of Air Filter Multicyclone

2.4.1. Purpose and Subject of Flow Tests in Dust Settler

The aim of the experimental flow tests was to evaluate the proposed changes concept in the internal structure of the air filter multi-cyclone MT dust settling tank segment due to the improvement in uniformity of suction from the individual cyclones, which should positively affect the separation efficiency of the entire multi-cyclone.

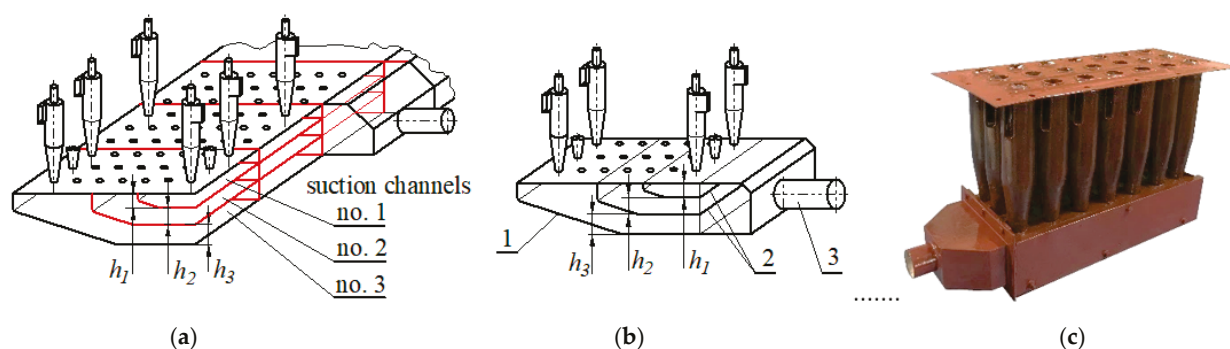


Figure 16. Dust settler schematic diagram: (a) MT multi-cyclone, (b) MT multi-cyclone segment, (c) segment prepared for flow tests, 1—sedimentation tank, 2—horizontal partitions, 3—suction port.

Research subject was dust settler segment of the MT multi-cyclone air filters built on the basis of the model test results of the dust settler. A schematic structure diagram of the MT multi-cyclone composed of the segments is shown in Figure 13.

In the inertial filters, made of several dozen cyclones, it is possible to use various methods for organizing the ejection suction. The choice of a specific method and the method used, aimed at ensuring the suction streams have the same values for all individual cyclones, depends not only on their efficiency, but also on the form of the multi-cyclone, number of cyclones and dust separator configuration, space limitations in the place of its installation, and the possibility of carrying out an appropriate research range.

The uniform value of the suction streams from the individual cyclones can be ensured by using the equal resistances method of suction streams through appropriate structural changes to the multi-cyclone dust settler, which consists of dividing the settling space into independent segments and channels covering a specific number of cyclones.

The calculation algorithm developed by the author enables the initial selection of the height at the outlet $h_1, h_2, \dots, h_i$ and subsequent suction channels for the dust settling tank segment, in terms of equal resistance to the air flow streams through the channels. The final selection of the number and height of $h_1, h_2, \dots, h_i$ and the channels should be made during the experimental tests of the multi-cyclone separate segment.

The algorithm for calculating the segment of multi-cyclone settling tank in a motor vehicle air filter can be applied to the calculation of the different settling tanks, but with the same construction principle—i.e., with a division into the segments and suction channels. A segment may contain a different number of suction channels, and each may be assigned a different number of cyclones.

Multi-cyclone structure made of segments significantly facilitates the design process and allows to give it the most convenient shape. The number of segments in a multi-cyclone can be easily adapted to the needs of the engine properties. Basic research is then confined to a segment, making it much simpler and less costly. The tests of the multi-cyclone, as a whole, usually concern the possible corrections of the arrangement and shape of the suction pipes and the method of their connection with the suction stream collector. The segmented (modular) design of the multi-cyclone gives the possibility of versatility by “assembling” the multi-cyclone from the number of modules resulting from the air consumption of the engine, e.g., one segment for every $300 \text{ m}^3/\text{h}$ or one segment for 50 kW of engine power.

2.4.2. Flow Research Methodology

Tests were carried out in a laboratory, reproducing the work of inertial filters in the real conditions of a vehicle user. Q_{Sc} values of suction streams from the individual cyclones of the dust settling tank segment were measured, in which the internal structure of the settling tank was changed by dividing it into suction channels. On this basis, an

assessment was made for the improvement of suction uniformity, which was caused by the division of the multi-cyclone dust settling chamber into segments and the division of the settling chamber into suction channels. Research course and subsequent design changes of the dust collector are presented, marking them successively A, B, C, and D for the MT multi-cyclone segment, until a satisfactory suction uniformity is obtained.

The flow tests of the MT multi-cyclone dust settler segment were carried out in two stages at the stand, the diagram of which is shown in Figure 17. Tests were carried out according to the methodology used to measure the suction stream values from the single cyclones of a complete multi-cyclone. During the tests of the MT multi-cyclone segment, suction stream values from the left (L) and right (R) cyclones of the segment, as well as from the first (I) and last (VII) cyclone rows, were measured (Figure 17).

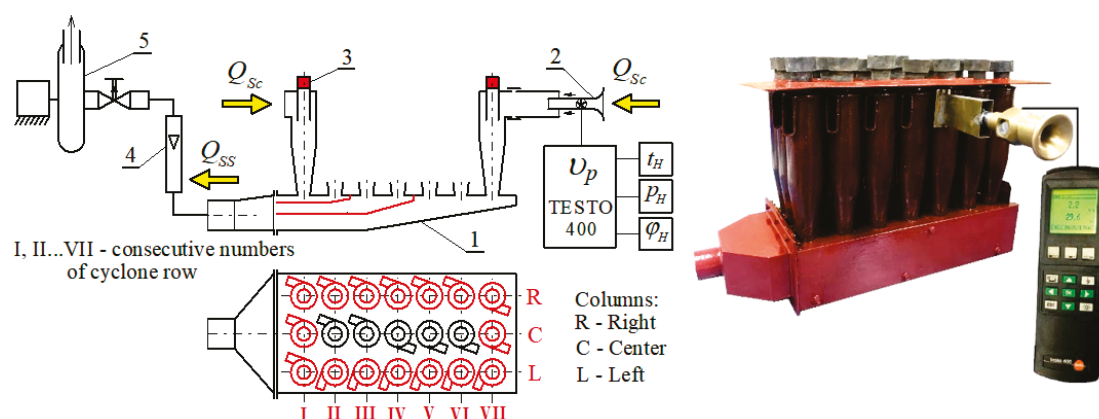


Figure 17. Stand diagram for measuring the Q_{Sc} sucked off streams from single cyclones of the MT multi-cyclone segment: 1—segment of the dust settler, 2—measuring system for the suction stream, 3—closing plug, 4—rotameter, 5—suction fan.

From the segment dust collector, the Q_{Ss} suction stream, being the sum of Q_{Sc} suction streams from the individual cyclones.

$$Q_{Ss} = \sum_{i=1}^J Q_{Sc,i} \quad (17)$$

where: I, II, ..., j ... J —consecutive cyclone number in the column.

The main suction stream Q_{Ss} from the settling tank of the multi-cyclone segment was determined from:

$$Q_{Ss} = \frac{Q_S}{m_s}, \quad (18)$$

where: m_s —number of segments into which the multi-cyclone dust collector has been divided, Q_S —suction stream from the multi-cyclone settler, determined for the rotational speed n_N of the engine and the extraction rate $m_0 = 4, 8, i 16\%$.

In the first stage, the values of the Q_{Sc} suction streams from the individual cyclones of the selected rows and columns of the segment with a dust trap in the original version were determined. In the second stage, the values of the Q_{Sc} suction streams were measured from the same individual cyclones of the segment in which the internal structure of the dust settling chamber was changed. The settling chamber was then divided into suction channels with heights $h_1, h_2, h_3 \dots h_i$ resulting from the model calculations and with the appropriate, and previously established, assignment of the cyclones. At this stage, several variations for dividing the segment's settling chamber into suction channels were investigated.

The initial variant (variant A) of the MT multi-cyclone segment is a dust collector, the chamber of which is divided into three suction channels with heights at the outlet: $h_1 = 12$ mm, $h_2 = 19$ mm, and $h_3 = 26$ mm.

2.4.3. Test Results Analysis of MT Multi-Cyclone Segment

Suction stream values from the left (L) and right (R) cyclones of the MT multi-cyclone segment for the same rows of cyclones differ slightly, and therefore, the measurement results of the suction streams from the cyclones from only one, the left column (L) of the multi-cyclone segment, were used for further analysis (Figure 18). The highest values of the Q_{Sc} suction streams from the individual cyclones of the segment with a dust settling chamber in the original version (before the division of the settling chamber into channels) were recorded, regardless of the value of the total Q_{Ss} suction stream, for the first cyclone from the middle column (S).

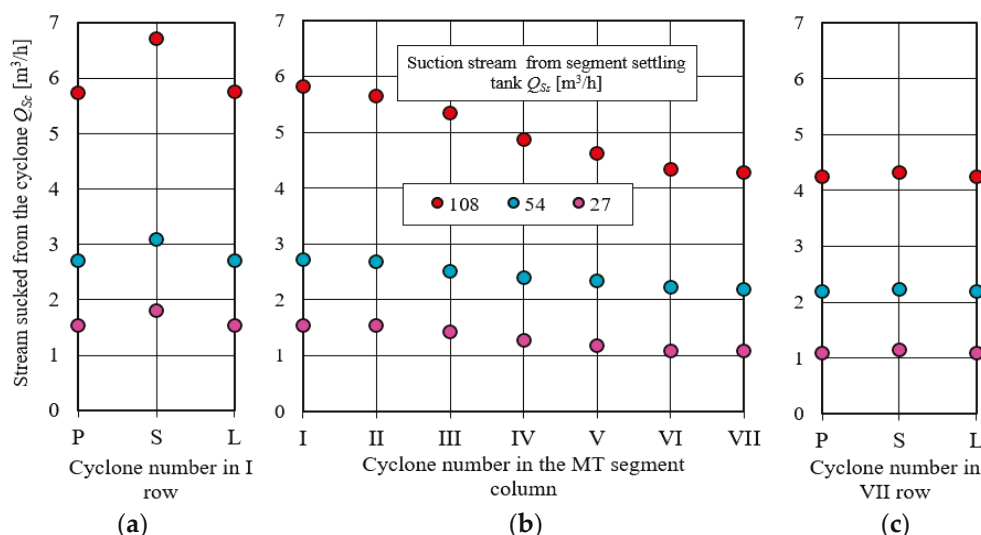


Figure 18. Q_{Sc} values sucked off streams from single cyclones of the original S_{MT} segment: (a) in the row No. I, (b) in the left column (L), (c) in the row No. VII.

Segment dust collector chamber structure change (variant A) with the division into three suction channels (Figure 19) resulted in a significant improvement in the suction uniformity. There was a decrease by about 13% of flow values sucked off Q_{Sc} from the first two cyclones (channel 1) and a significant increase by 16% of the flows in the channel No. 3. In channel No. 1 there were large (about 14%) differences between stream values sucked off from cyclone No. I and No. II. In the remaining channels, these differences do not exceed 4%. The stream sucked off from cyclone No. II with its small value $Q_{Sc} = 4.6 m^3/h$ clearly differs from the suction streams from the remaining cyclones of the segment, the value of which fluctuates within the range of 5.2–5.4 m^3/h .

The reason for such a low value of the Q_{Sc} flux when suctioned from the second cyclone may be the small height (12 mm) of channel No. 1, and the lack of a chamber into which the suction streams from the cyclones flow, as is the case with the other channels, as well as the large distance of the second-order cyclones from the outlet stub, which suppressed the air flow Q_{Sc} . The above analysis shows that channel 1 should not be assigned a larger number of cyclones. In the case of the tested settler, it should only be the first row of cyclones.

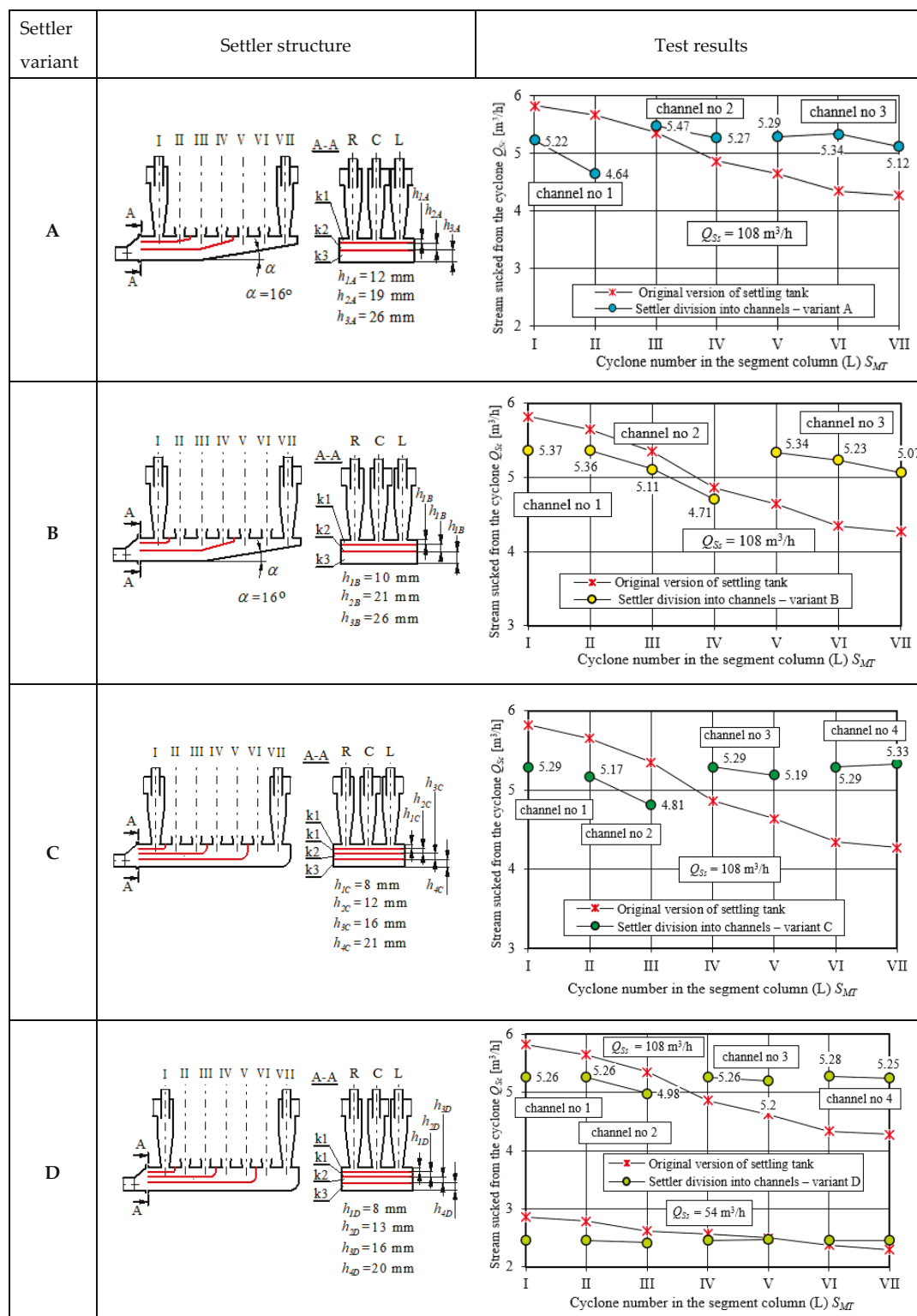


Figure 19. Q_{sc} suction stream values from individual cyclones from left column (L) of the segment before and after the division of the dust settling chamber into suction channels for variants (A–D).

Variant B of the dust settler has a three-channel structure for the dust settling chamber, but with a different assignment of cyclones—depicted in Figure 19. Due to this structure, values of the suction streams Q_{sc} from channel No. 1, and the first cyclones of each of the remaining channels, assume the same—and at the same maximum time values. In channel No. 2 and No. 3, it can be observed that as the position of subsequent cyclones

moves away from the rear wall of channel No. 1, the values of the Q_{Sc} suction streams systematically decrease. The suction stream from the last cyclone in channel 2 is 13% lower, and the suction stream from the last cyclone in channel 3 is 9% lower than the others. The reason for such a low value of the flows Q_{Sc} when suctioned from the last cyclones may be the oblique channel bottom shape, which reduces the chamber volume into which the flows can be suctioned from the cyclone's flow, which suppresses the outflow of the air stream Q_{Sc} . The test results of variant B of the dust settler suggest that assigning more than two cyclone rows to the channel is not recommended. At the same time, assigning a smaller number of cyclones to one channel forces the division of the settling chamber into a larger number of channels, which complicates the design and increases the production cost.

Table 3 presents a summary of the changes in the height of the dust trap channels during the variant tests, and the number of cyclones assigned to the individual channels.

Table 3. Settling tank variants of S_{MT} segment in MT multi-cyclone.

Segment Variant	Channel No. 1		Channel No. 2		Channel No. 3		Channel No. 4	
	Cyclone Number	h_1 (mm)	Cyclone Number	h_2 (mm)	Cyclone Number	h_3 (mm)	Cyclone Number	h_4 (mm)
A	I, II	12	III, IV	19	V, VI, VII	26	-	-
B	I	10	II, III, IV	21	V, VI, VII	26	-	-
C	I	8	II, III	12	IV, V	16	VI, VII	21
D	I	8	II, III	13	IV, V	16	VI, VII	20

This problem was analyzed by examining variant C of the segment dust collector—depicted in Figure 19. The structure of the segmented dust collector chamber (variant C) resulted from the division of this chamber into four suction channels with heights: $h_{1C} = 8$ mm, $h_{2C} = 12$ mm, $h_{3C} = 16$ mm, and $h_{4C} = 21$ mm with appropriate cyclone assignments—depicted in Figure 16. The shape of the settling tank has been changed in relation to the original. The bottom of the settling tank and the subsequent partitions are parallel along their entire length to the upper cyclone mounting plate.

As a result of the introduced changes, a significant reduction in the differences between the values of the suction flows Q_{Sc} from the individual cyclones of the segment was achieved—Figure 19. The suction streams from cyclones No. II and III of channel No. 2 have significantly lower (cyclone No. III by more than 8%) values than the suction streams from cyclones covered by channels No. 3 and No. 4, although the distance of these cyclones from the outlet of the channels (section A-A) is much greater.

The analysis of the structure of the designed settling tank (variant C) shows that both the volume of the chambers of channels 3 and 4 and the height b_3 and b_4 of these chambers (the distance between the outlet openings from the cyclones and the bottom of the chamber) are two and three times larger than the volume of channel chamber No. 2 and its amount b_2 , respectively. Thus, the attenuation of the streams outflow from the cyclones of channels No. 3 and No. 4 is much lower, which can be seen in the form of higher values of the suction streams. The highest values have the suction streams from the cyclones of channel No. 4. The results of the research and the above analysis suggest changes in the structure of the settling chamber within channels No. 2 and No. 4.

Variant D of the chamber of the settling tank, in relation to variant C, has a similar height to channel 2, but higher by 1 mm, and is similar to the height of channel 4, but lower by the same value.

The measurement results of the suction flow values Q_{Sc} from the individual cyclones for the left column (L) of the segment (variant D) of the dust settling chamber, for two values of the suction flow $Q_{Ss} = 108$ m³/h and $Q_{Ss} = 54$ m³/h, show (Figure 19) a clear improvement in the uniformity of the suction between cyclones in the channels, and

within the entire column. The differences between the suction streams from adjacent cyclones in channels 3 and 4 do not exceed 3%. For a lower value of the suction stream ($Q_{Ss} = 54 \text{ m}^3/\text{h}$), these differences are insignificant.

2.5. Separation Efficiency Experimental Studies of Single Cyclones in the Multi-Cyclone Segment

2.5.1. Purpose and Scope of Research

The efficiency research for dust removal from the cyclones in the S_{MT} segment of the MT multi-cyclone was aimed at experimentally verifying the developed concept of the changes that can be made in the design of the internal structure of the dust settling chamber of the segment. An experimental evaluation of the developed concept of the changes that can be made in the internal structure of the dust settling chamber of the S_{MT} segment of the MT multi-cyclone was performed due to the efficiency of dust removal in individual cyclones. The separation efficiency values of single cyclones from the left column (L) of the S_{MT} segment were determined. The research was carried out in two stages:

- stage I—dust removal efficiency measurements from the cyclones of the segment with the original dust trap;
- stage II—dust removal efficiency measurements from the segment cyclones after making changes to the internal structure of the dust collector chamber according to variant D;

2.5.2. Research Methodology and Conditions

The dust removal efficiency tests for single cyclones of the S_{MT} segment were carried out on the stand, the diagram of which is shown in Figure 20. The tested segment, being the fifth part of the MT multi-cyclone, includes 21 D-40 cyclones arranged in seven rows and three columns in the same way as in the original multi-cyclone. In the first stage of research, the segment settling chamber has the shape of a multi-cyclone settling chamber. On the other hand, in stage II of the tests, the sedimentation tank is divided into four suction channels—as is shown in variant D. Directly at the outlet of the segment's sedimentation tank, in the suction line, there is a safety filter that stops the dust separated by the cyclones. The cyclone outlet nozzles of the segment (purified air) are connected by a common chamber, in which there is a measuring filter cartridge designed to retain the m_{ZA} dust mass carried along with the exhaust air from the cyclones.

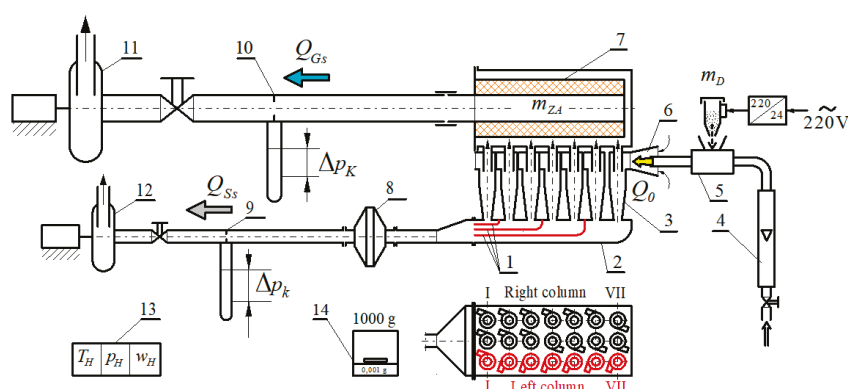


Figure 20. Test stand diagram of the air filter multi-cyclone segment: 1—partitions of dust settling channels, 2—dust settler, 3—cyclone, 4—compressed air stream rotameter, 5—dust dispenser, 6—dust chamber, 7—measuring filter, 8—absolute filter of the suction stream, 9—measuring orifice of the suction stream, 10—measuring orifice of the main stream, 11—suction fans, 12—device for measuring temperature, pressure and humidity of the ambient air, 13—analytical balance.

The test dust was dosed to the inlet stub of the tested cyclone, which ended with a dust chamber where it was mixed with the inlet air stream Q_0 . The separation efficiency

was determined successively for the extreme single cyclones (No. I, II, . . . VII) of the left column (L) of the multi-cyclone segment under the following conditions:

- air outlet stream from a single cyclone $Q_{Gc} = 34 \text{ m}^3/\text{h}$, resulting from the number of cyclones used in the filter multi-cyclone and the air demand at the rotational speed $n_N = 2000 \text{ obr/min}$ in vehicle engine;
- air stream suctioned from a single cyclone $Q_{Sc} = 2.72 \text{ m}^3/\text{h}$ for the degree of the ejection dust extraction from the settling tank $m_0 = 8\%$;
- dust concentration in the air $s = 1 \text{ g/m}^3$;
- measurement time—time of uniform dust dosing to the cyclone $t_{pom} = 3 \text{ min}$;
- PTC-D test dust, being a domestic replacement for AC-fine dust.

For each cyclone from the right column, starting from cyclone No. I up to cyclone No. VII, $j = 5$ for the separation efficiency measurements φ_c , which were performed successively according to the following methodology:

- m'_{Dj} the mass of the dust dispenser container and the m'_{ZAj} were determined, before each measurement;
- suction fans for the main and suction pipes were started, and then, on the basis of the indications from the U-tube water pressure gauges and the gauge diagrams of the measuring orifices, the required value of the air streams Q_{Gs} and Q_{Sc} was determined—depicted in Figure 18;
- dust was evenly dosed to the set cyclone during time t_{pom} ;
- air streams flow Q_{Gs} and Q_{Sc} were closed, and then the mass m'_{Dj} of the dust dispenser container and the mass m'_{ZAj} of the measuring filter cartridge after the measurement were determined;
- dust mass loss from the dispenser was determined, and thus the dust mass introduced with the air stream into the cyclone according to the formula:

$$m_{Dj} = m'_{Dj} - m''_{Dj}. \quad (19)$$

- dust mass retained on the filtering element of the measuring filter was determined according to:

$$m_{ZAj} = m''_{ZAj} - m'_{ZAj}. \quad (20)$$

- dust mass retained by the cyclone was determined from:

$$m_{cj} = m_{Dj} - m_{ZAj}. \quad (21)$$

- The separation efficiency was calculated as the quotient of the dust mass retained by the cyclone and the dust mass introduced into the cyclone on the basis of:

$$j_{cj} = \frac{m_{cj}}{m_{Dj}} \cdot 100\%. \quad (22)$$

After cycle $j = 5$, the efficiency measurements for a given cyclone were performed and the average value of the separation efficiency and dust concentration of inlet air to the cyclone were calculated.

2.5.3. Research Results Analysis

The test results of the separation efficiency values for single cyclones from the left (L) column in the S_{MT} segment of the MT multi-cyclone are shown in Figure 21.

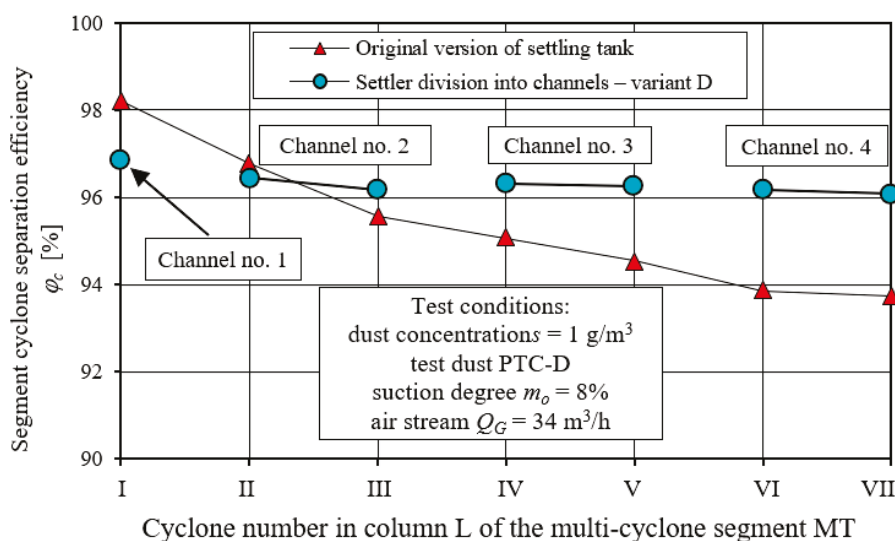


Figure 21. Separation efficiency values for single cyclones of left column (L) in MT multi-cyclone segment for the original version of the settler and after division into suction channels—variant D of settling tank.

As the cyclone's position moves away from the suction nozzle, the cyclone separation efficiency values of the segment, whose settling chamber is a common part for all cyclones, are decreasing. For the first (I) cyclone, a separation efficiency of 98.2% was recorded in the column, and for the last (VII), a separation efficiency of 93.7%. There was a 5% difference in cyclone separation efficiency, which results is mainly caused by the decreasing values of the suction streams (suction rates) from these cyclones—as demonstrated by the flow tests of the multi-cyclone and its dust settling device. A decrease in the cyclone separation efficiency may also be caused by the interaction of the swirled streams flowing out of the cyclones and flowing into the common chamber of the dust settling chamber, and can even lead to the possibility of reverse flows in the cyclones located on the periphery of the multi-cyclone.

The separation efficiency values for the same cyclones in the segment in which the dust settling chamber structure was changed (the chamber was divided into suction channels) show much smaller differences, although a slight decrease in the cyclone efficiency is noticeable with their location being further from the suction connector—depicted in Figure 21. The efficiency separation values from cyclone No. I, from which dust is suctioned through channel No. 1, is clearly higher than the efficiency of the other cyclones in the column and has a value of 96.8%. The cyclone separation efficiency, from which suction takes place through the separate channels 2, 3 and 4, is slightly lower and ranges from 96.1–96.4%. After changing the structure of the dust settling chamber, the separation efficiency of the first two cyclones (I and II) decreased, while the remaining cyclones achieved higher efficiency values, with the highest increase (about 3%) recorded for the last (VII) cyclone in the column.

A slight decrease in the cyclone separation efficiency of the tested dust settling column with a chamber structure that guarantees even suction may be caused by the impact of the main (purified air) Q_G streams flowing from the cyclones in the chamber area, where the measuring filter insert is located. The different distances of the cyclone outlet pipes from the segment of the main stream outlet can cause different pressure drops for these streams (the further the cyclone is located, the greater the pressure drop) in the cyclone in the outlet stub pipe section, and for the individual cyclones. The air stream exiting the cyclone furthest from the main stream outlet port may be less than the value of the air stream exiting the cyclone directly at the outlet port. The lower value of the air stream means a lower speed and, at the same time, a lower inertial force, and thus the cyclone obtains efficient dust removal at a lower level.

3. Summary

The aim of this study was to develop a concept for dust suction organization from the multi-cyclone dust collector that would ensure the minimization of the differences between the suction streams from individual multi-cyclone cyclones. It has been shown that the uniformity values of the suction streams from individual cyclones can be achieved by the appropriate shaping of the internal structure of the multi-cyclone dust collector. The method consists of dividing the dust collector of the multi-cyclone into several vertical partitions, thus creating identical segments of which the internal space is then divided by horizontal, independent, and isolated suction channels. Each channel has been assigned a specific group of single cyclones. The heights of channels h_1 , h_2 , and h_3 formed by partitions in the settling tank were selected in terms to obtain the same values of pressure drop through the channels by using this purpose-built model of the segmented dust settling tank. It was not possible to achieve the equal values of pressure drop by only selecting channels geometrically by calculation, due to the rather complex flow in the sedimentation tank channels. Therefore, the final selection of the channel height was made during the experimental flow tests. Verification of the developed model for the dust settling tank segment was performed during the separation efficiency tests of the individual cyclones of the segment, using the appropriate values for mineral dust concentration in the suctioned air.

Based on analysis of the results, modeling, and experimental studies of a single cyclone in the multi-cyclone segment, it was found that:

1. In the case of a multi-cyclone consisting of a return cyclone with a tangential inlet, to which air flows from an unlimited space, uniform suction stream values from the individual cyclones of the multi-cyclone can be ensured using an equal suction streams resistance method. Such an effect can be achieved by an appropriate multi-cyclone dust settler structure change, consisting of dividing the settling chamber into independent segments and channels covering a specific number of cyclones.
2. A uniform value of the suction streams from the individual multi-cyclone cyclones can be ensured using a method of equal suction streams resistances through an appropriate structure change of the multi-cyclone dust settler, consisting of the division of the settling space into independent segments and channels covering a specific number of cyclones.
3. The initial height selection at the outlet $h_1, h_2, \dots, h_i$, of the subsequent suction channels of the dust settler segment, in terms of equal resistance to the flow of air streams through the channels, can be made using a model of the multi-cyclone segment settler. The final selection of the number and height $h_1, h_2, \dots, h_i$, and the channels, should be made during the flow experiments of a separate multi-cyclone segment.
4. Q_{Sc} values for the suction streams from individual cyclones of the S_{MT} segment before the dust settling chamber structure change have significantly different values in the range of 5.82–4.28 m³/h, which is 36%. The change in the multi-cyclone dust settling chamber structure, by dividing it into segments and dividing the segment settling chamber into independent and isolated suction channels, reduces these differences to 3–5%, which should result in higher cyclone separation efficiency values.
5. The algorithm for calculating the multi-cyclone settling tank segment of a motor vehicle air filter can be applied to the calculation of the different settling tanks, but with the same construction principle, i.e., with a division into segments and suction channels. A segment may contain a different number of suction channels, and each may be assigned a different number of cyclones. It is not advisable to assign more than two cyclone rows to a channel. At the same time, assigning a smaller cyclone number to one channel forces the settling chamber division into a larger number of channels, which complicates the design and increases the production cost.

6. The multi-cyclone structure made of segments significantly facilitates the design process and allows it to have the most convenient shape. The number of segments in a multi-cyclone can be easily adapted to the needs of the engine's characteristics, e.g., one segment for every 300 m³/h or one segment for 50 kW of engine power.
7. The segment cyclones having a common dust collector chamber achieve different values for the separation efficiency (in the range of 98.2–93.7%), wherein the smaller their distance from the suction port the greater the value, which is caused by the unevenness (different values) of the suction streams from the individual cyclones. Other factors, such as the interaction of swirled streams flowing out of the cyclones and flowing into a common chamber of the dust settling chamber, or the occurrence of reverse flows between cyclones, cannot be excluded.
8. The significantly smaller differences in the range of 96.8–96.1% for the achieved separation efficiency values of the single cyclones in the segment with a modernized dust settler confirms the results of the conducted flow tests for the multi-cyclone segments and suggests that the presented method of calculating the dimensions of the dust settling channels is correct.

It follows that the change in the dust settling chamber structure (division of the settling chamber into channels and assigning them a specific number of cyclones) gave the expected results in the form of more uniform dust removal efficiency values in the individual cyclones but did not significantly increase the efficiency of the segment separation. However, even such a slight (0.9%) increase in separation efficiency of the segment (multi-cyclone) gives measurable effects in the form of increased retention of dust mass. For example, a multi-cyclone air filter from a T-72 tracked vehicle operated for 50 h (service interval) at a dust concentration of 1 g/m³ will retain (when changing efficiency from 95.4% to 96.3%) more than 1500 g of dust, which constitutes 20% of the dust in the air flowing out of the cyclones and influencing the second separation stage—i.e., the filter paper. With limited absorbency of the filter paper, which for standard paper has a value of $k_m = 190\text{--}220 \text{ g/m}^2$, these are measurable results in the form of an extended service life of the air filter and limited by the value of the permissible resistance Δp_{fdop} .

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