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CONVERSION OF TURBOCHARGED INTERNAL COMBUSTION ENGINE TO GASEOUS FUEL USED IN AGRICULTURAL MACHINERY

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This study focuses on the conversion of a turbocharged compression-ignition (diesel) engine UR-II 8602.12, originally installed in the Zetor 16145 tractor, into a spark-ignition (gas) unit operating on gaseous fuel. The study presents experimental results of performance, fuel economy, and emission characteristics under full-load speed sweep as well as part-load operation. The converted engine achieved a maximum corrected power output of 106.5 kW at 2200 rpm, representing an 8.5% reduction relative to the original diesel engine (116.5 kW). The corrected peak torque reached 462 N·m, which is 21% lower than the baseline diesel torque of 583 N·m. From an environmental perspective, carbon monoxide emissions remained very low (~0.05 vol. %), while hydrocarbon concentrations decreased with increasing mixture enrichment (from 377 ppm to 245 ppm). However, nitrogen oxides (NO_x) exceeded the threshold of 500 ppm across the entire speed range, peaking at 2282 ppm at the maximum speed. Exhaust gas temperatures behind the turbine reached up to 650 °C, which, after accounting for thermal gradient, implies that the recommended limit of 630 °C at exhaust valves is exceeded at higher loads. The findings demonstrate that natural gas conversion can retain favourable power characteristics while lowering carbon-based emissions, yet further optimisation is required to control thermal stress and mitigate NO_x formation.

Keywords: combustion engine optimisation; natural gas; biogas; agricultural machinery

The development of society observed in past decades is associated with increasing demands on the environment and its resources. This increase is mainly due to the ever-increasing use of fossil fuels and resources, which poses a serious problem in terms of both local and global environmental impacts. In addition, however, the interest of society, including the scientific community and political leaders, in the impact of human activities on climate change and the environment has also increased. These efforts are aimed at eliminating or mitigating long-term negative impacts on the Earth's climate and the environment as a whole. The greatest impact on the environment is the energy dependence of the world's population primarily on non-renewable energy sources. This fact is clearly confirmed by several factors, whether air quality indicators or pollution or damage to various ecosystems due to

the extraction, processing, leakage, and use of fossil energy sources (UCS, 2025; Bertrand, 2021).

From a global perspective, however, the most significant impact of climate change is due to the rising average temperature of our planet's surface (OECD, 2025). This phenomenon is largely driven by the increased global production of the greenhouse gas CO₂, which largely limits the escape of heat from

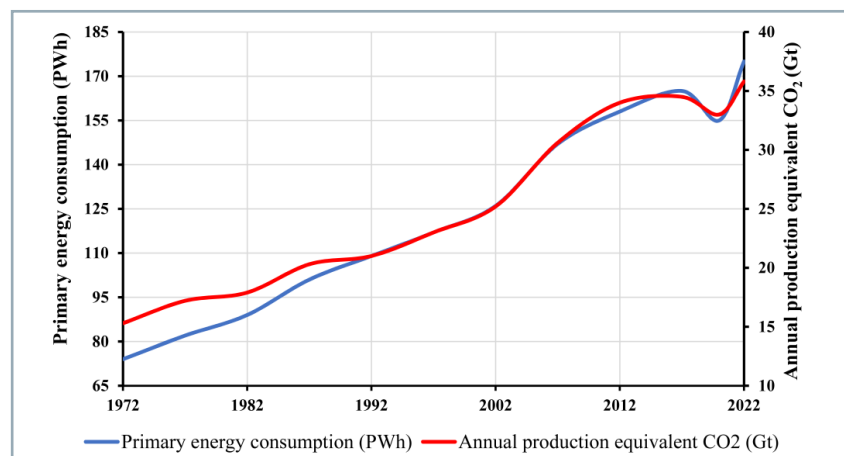


Fig. 1 Comparison of global primary energy consumption and global annual CO₂ equivalent production between 1972 and 2022
Source: Ritchie et al., 2023; IPCC, 1990; NASA, 2025a

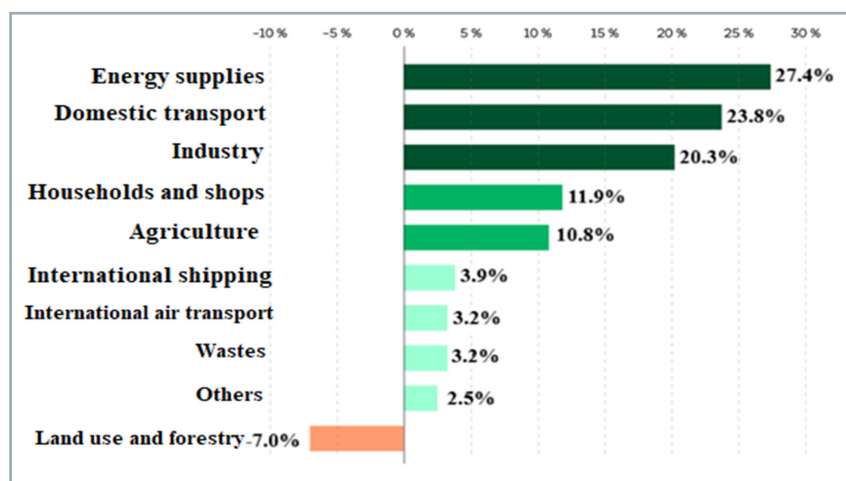


Fig. 2 Amount of greenhouse gases produced in the European Union in 2022 by sectors

Source: OECD, 2025

the surface of our planet (Čupera et al., 2010; NASA, 2025b; NOAA, 2025).

Figure 1 shows how rising energy intensity directly drives global fossil fuel production, energy consumption, and atmospheric carbon dioxide concentrations (Ritchie et al., 2023; OECD, 2025; IEA, 2021). This phenomenon subsequently drives global warming and large-scale climate change (NASA, 2025b; NOAA, 2025).

Figure 2 shows the amount of greenhouse gases produced in the EU-27 countries based on different sectors. The majority of CO₂ production is made up of the energy sector, which includes the production and distribution of energy. A similarly

large part is domestic transport, which has been undergoing considerable turbulence in recent years due to the ever-changing regulations and emission standards of the European Union. In summary, it could be said that all the efforts of the European Commission since 2020 have led to the gradual elimination of internal combustion engines as greenhouse gas producers.

Another major contributor to Europe's total CO₂ production – agriculture – is also closely linked to domestic transport. Since agriculture is one of the very important parts of the national interests of every country, it needs to be dealt with increased priority. Like all sectors,

greening and efforts to eliminate fossil fuels also affect agricultural machinery and mechanisms, which can be achieved in several ways (Karavalakis et al., 2012).

One way to reduce these negative impacts is to increase the share of renewable energy sources, reduce the consumption of fossil fuels and use waste, along with efforts to reduce its generation. These objectives can also be achieved through the use of the energy potential of unseparated waste. The topic of the use of gases generated during waste processing in piston combustion engines has long been the subject of study and research at the Institute of Automotive Engineering and Design of the Faculty of Mechanical Engineering of the Slovak University of Technology.

Biogas is one such alternative; however, the utilization of these fuels presents various challenges, and renewable energy sources – particularly biogas plants – are not always perceived positively by the public (Khan et al., 2015). Despite its multiple advantages and considerable production potential, biogas remains underutilised and relatively unknown. Nonetheless, it is the third fastest-growing renewable energy source globally, surpassed only by photovoltaics and wind energy. European Union data shows that in 2020, 18,855 biogas plants and 726 biomethane plants were installed in Europe, producing 176 TWh of biogas and 26 TWh of biomethane. According to The Regulatory Office for Network Industries (URSO) data, there were 111 biogas plants in operation in Slovakia in 2022 with a total installed electrical capacity of 103.14 MWh. On average, a single biogas plant has an installed electrical capacity of 929 kW. In the production of biogas, the quality of the input raw materials, i.e. waste, is crucial. In terms of sustainability and carbon footprint, feedstocks that are otherwise unusable for human consumption or animal nutrition should be used as a priority for biogas production. However, their availability, quality (e.g., the absence of undesirable pollutants such as plastics), and usability or suitability for the technology remain challenging. Furthermore, supporting domestic agricultural land for food cultivation

Table 1 Approximate biogas yield from different substrates (Čupera et al., 2010)

	Dry matter content (%)	Amount of biogas (Nm ³ ·t ⁻¹)
Oils and fats	99	1200
Glycerine	100	846
Bread and pastries	88	651
Molasses	77	490
Maize silage	33	190
Grass haylage	35	120
Fresh grass	25	100
Sugar cutlets	22	100
Kitchen waste	16	95
Cow manure	20	86
Fruits and vegetables	15	57
Poultry droppings	15	56
Burnouts	8	48
Whey	5	34

presents a challenge compared to international markets, as domestic production cannot compete on price with subsidised foreign food (Sahoo et al., 2009).

Therefore, in practice, raw materials are utilised, such as silage from purposely grown crops (maize, grasses, mixed crops), livestock excrement with or without bedding (cattle manure, poultry droppings, cattle or pig slurry), plant by-products, and industrial residues (distillery dregs, bran, waste bread and pastries, chaff, pomace, sugar beet pulp, molasses, lecithin sludge, fruit and vegetables and their peels, glycerine, etc.). The energy potential of individual inputs is shown in Table 1. Experimental investigations of internal combustion engines focusing on performance, fuel efficiency, and emission characteristics have also been conducted by several authors (Tkáč et al., 2025; Tulík et al., 2024; Phu et al., 2024; Shu et al., 2024).

The presented article deals with the measurement and optimisation of the parameters of the turbocharged tractor engine UR-II 8602.12 NG after its conversion to natural gas or biogas propulsion, intended as the drive unit of the tractor, which will be used as a means of transport in livestock production. The engine design is based on the concept of the DS Martin UR II 8602.12 diesel engine, which has been structurally modified to run on natural gas. The formation of the mixture is provided by a gas mixer.

Natural gas consists primarily of methane (CH_4), typically 92–96 vol. %, with smaller amounts of ethane, propane, and inert gases. Its lower heating value is approximately $48\text{--}50 \text{ MJ}\cdot\text{kg}^{-1}$. Biogas typically contains 50–70 vol. % methane (CH_4) and 30–50 vol. % carbon dioxide (CO_2), along with trace amounts of hydrogen sulphide (H_2S) and nitrogen (N_2). The calorific value of biogas is usually in the range of $18\text{--}25 \text{ MJ}\cdot\text{m}^{-3}$.

The novelty of this manuscript lies in the first experimental conversion and full-load characterisation of the UR-II 8602.12 turbocharged diesel engine – specific to Zetor 16145 tractors widely used in Central European agriculture – into a monofuel spark-ignition natural gas/biogas unit without lambda feedback or catalysis. This work quantifies monofuel performance

trade-offs (8.5% power loss, 21% torque drop) under lean-to-enriched mixtures ($\lambda = 1.64\text{--}1.38$), identifies critical NO_x peaks (up to 2282 ppm) and valve thermal exceedance (post-turbine 650°C), and provides baseline data for cost-effective retrofits in biogas-rich regions like Slovakia (111 plants, 103 MW capacity). This advances tractor-specific gaseous fuel optimisation by highlighting needs for adaptive mixing and thermal management.

Material and Methods

The experimental work was conducted on the UR II 8602.12 NG internal combustion engine, which was developed by converting a Zetor 8602.1 diesel engine. This section presents the measurements of performance, economic, ecological, and other monitored engine parameters under the conditions of the main speed characteristic. The engine was operated on a lean mixture within a speed range of 1000 rpm to 2000 rpm (or 2200 rpm), with a constantly adjusted ignition advance angle of $\varphi_{\text{ign.}} = 18^\circ \text{ CA}$ before TDC across all engine speeds. In addition to torque and exhaust gas analyser accuracy, the measurement uncertainty of temperature sensors was $\pm 1.5^\circ\text{C}$, pressure sensors $\pm 1\%$, and flow measurement $\pm 2\%$. The overall experimental uncertainty was estimated according to standard propagation methods

and is within acceptable limits for engine performance and emission measurements.

Basic characteristics of an internal combustion engine

The conversion of the UR II 8602.12 NG internal combustion engine to the ignition principle of operation and combustion of gaseous fuel (natural gas, methane, or biogas) required a change in the preparation of the mixture by means of an air-gas fuel mixer, which also allowed manual regulation of the mixture composition characterised by the excess air coefficient λ . During the conversion, the original diesel injection system was replaced by an ignition system with optimised spark energy for gaseous fuel, and the throttle was installed on the discharge side of the turbocharger. The engine was equipped with the necessary sensors (position sensor, temperature, pressure, and flow sensors) for the electronic engine control module. A schematic representation of the engine is shown in Fig. 3.

The reference fuel, which was natural gas, can be burned in spark-ignition engines in a wide range of fuel mixture richnesses, to which the design parameters of the engine (compression ratio, resistance of individual parts to thermal loads) must be adapted. To cope with a smooth combustion process, the compression ratio of the UR II 8602.12 NG engine has

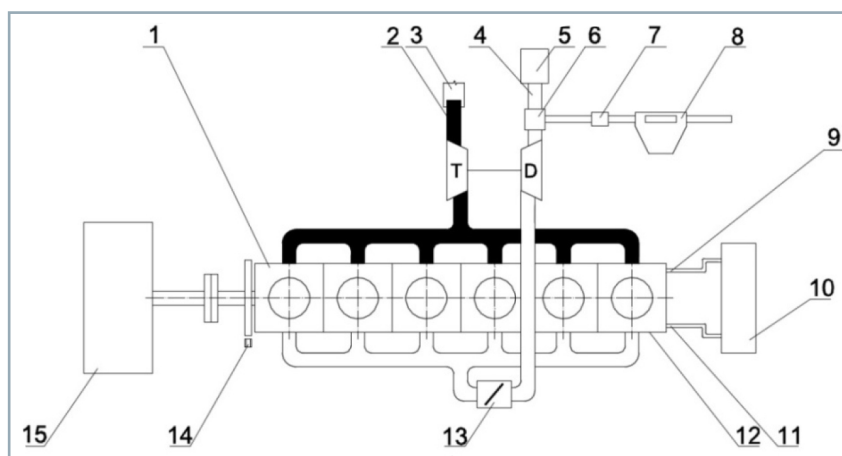


Fig. 3 Diagram of the UR II 8602.12 NG internal combustion engine
1 – UR II 8602.12 NG engine; 2 – exhaust gas temperature sensor; 3 – sampling port for exhaust gas analyser; 4 – intake air temperature sensor; 5 – air filter; 6 – mixer with diffuser; 7 – zero gas regulator; 8 – volumetric flow meter for gases fuels; 9 – engine outlet temperature sensor; 10 – radiator; 11 – engine inlet temperature sensor; 12 – engine oil temperature sensor; 13 – throttle; 14 – crankshaft speed sensor; 15 – Hofmann I4a eddy current dynamometer

Table 2 Characteristic data of the combustion engines

Engine fuel	UR II 8602.12 NG natural gas	DS Martin UR II 8602.12 oil
Position and arrangement of cylinders	standing, row	standing, row
Duty cycle	four-stroke, petrol	four-stroke, diesel
Method of mixture preparation	Mixer Hatrac.com ignition system	direct injection
Number of cylinders	6	6
Filling turbocharger	supercharged K27-98/02	supercharged K27-98/02
Cooling	liquid	liquid
The order of work of the rollers	1-5-3-6-2-4 (first cylinder at the radiator)	1-5-3-6-2-4 (first cylinder at the radiator)
Drilling (mm)	110	110
Stroke (mm)	120	120
Displacement (cm ³)	6842	6842
Compression ratio	12 : 1	17 : 1
Maximum brake power* (kW) at revolutions per minute	106.5 2200	116.5 2200
Maximum brake torque (N·m) at revolutions per minute	462 2200	583 1600

*/maximum ISO brake power (without fan), reduced for the UR II 8602.12 NG engine to a bar pressure of 99 kPa and a temperature of 298 K

been reduced to 12 : 1. The summary characteristics and parameters of the internal combustion engine before and after conversion are shown in Table 2.

Very low amounts of pollutants in the exhaust gases are produced by engines burning stoichiometric

mixtures in conjunction with a catalytic converter, as well as by engines burning lean fuel mixtures, in which the content of pollutants is reduced by the oxidation of CO and CH_x.

The layout of the engine and the range and complexity of its accessories are influenced by

the requirements for the engine' operating parameters, as well as the requirements for harmful emission levels.

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In this case, the UR II 8602.12 NG was equipped with a mixer without reverse regulation of the composition of the fuel mixture using a lambda probe and without a catalytic converter, as it is a more economically advantageous application of the engine for the use of a tractor drive. The engine and dynamometer assembly is shown in Fig. 4.

The performance parameters of the engine were measured with a Hofmann I4a engine brake, which was calibrated according to the procedure given by the brake manufacturer before measurement. The accuracy of the measured torque ranged within $\pm 1\%$. The individual components of exhaust gases as well as the excess

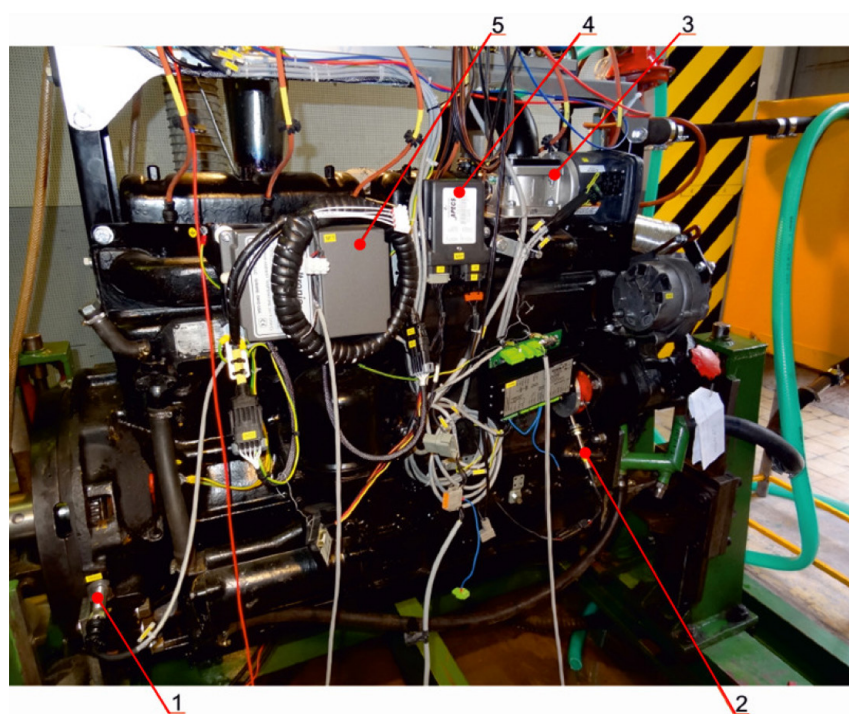


Fig. 4 Installation of the UR II 8602.12 NG engine at the Hofmann I4a brake station
1 – crankshaft speed sensor for the control unit; 2 – speed sensor for the ignition system; 3 – throttle; 4 – internal combustion engine control unit; 5 – ignition system control

air coefficient λ were measured (respectively λ calculated) by the SUN DGA 1000 analyser with an accuracy of $\pm 3\%$ (OIML Class I accuracy).

Results and Discussion

The measured values of the UR II 8602.12 NG internal combustion engine in the main speed characteristic mode are shown in Fig. 5, which presents the measurement results when operating a natural gas engine with a constant pre-ignition angle and with a preset value of mixture richness using a control screw before the fuel supply to the diffuser. The results of the power parameters show that the maximum measured effective power (111.3 kW) is achieved at 2200 rpm. The maximum reduced brake torque (483 N·m) is also achieved at the maximum operating speed of 2200 rpm. After correcting the performance parameters according to Regulation No 85 of the Economic Commission for Europe of the United Nations (2014) (operating temperature 25 °C and operating pressure 99 kPa), the corrected maximum power was adjusted to 106.5 kW and the maximum torque was corrected to 462 N·m. The graph also shows economic parameters related to the operation of the internal combustion engine, such as hourly volumetric or mass fuel consumption (M_p) and brake specific fuel consumption (m_{pered}).

The minimum brake specific fuel consumption in the main speed characteristic is at 1500 rpm and is 216 g·kW⁻¹·h⁻¹. In addition to the above-mentioned parameters, the values of operating parameters such as cooling water temperature at the engine inlet t_{water} , engine oil temperature t_{oil} , air temperature in the intake manifold t_{intake} , and exhaust gas temperature behind the turbine $t_{exhaust}$ during each operating mode were also recorded. During the measurements, the cooling water temperature ranged from 78 °C to 83 °C, engine oil temperature from 92 °C to 98 °C, intake air temperature from 20 °C to 22 °C, and exhaust gas temperature from 485 °C to 650 °C depending on operating conditions. These results confirm the significant thermal loading of the engine, especially at higher speeds and richer mixtures.

During the testing, exhaust emission parameters (CO [% vol.], CO₂ [% vol.], CH_x [ppm], NO_x [ppm], O₂ [% vol.]) were also measured. Based on these values, the flue gas analyser software calculated the air excess coefficient (λ [-]) using the Bretschneider formula for the SUN DGA 1000 analyser.

Figure 6 shows exhaust emission composition as a function of engine speed. As the speed increases, the natural gas-air mixture becomes richer because the fuel system lacked automatic air-fuel ratio regulation. At engine idle speed (800 rpm), the richness of the mixture was set to the value of the excess air coefficient $\lambda = 1.61$ using the fuel supply control screw. The measured

course of emissions from the supply point downstream of the turbine shows that the values of carbon monoxide (CO) production are at a low level (0.05 vol. %). The formation of carbon dioxide (CO₂) increased due to the change in the excess air coefficient closer to the stoichiometric mixture with the change in engine speed. The lowest value of CO₂ (6.35 vol. %) was at 1000 rpm at the value of the excess air coefficient $\lambda = 1.636$. On the contrary, the highest CO₂ value (7.63 vol. %) was at 2200 rpm, when the excess air coefficient was $\lambda = 1.377$. The measured emission profile downstream of the turbine shows that carbon monoxide (CO) production remains low at 0.05 vol. %. Carbon

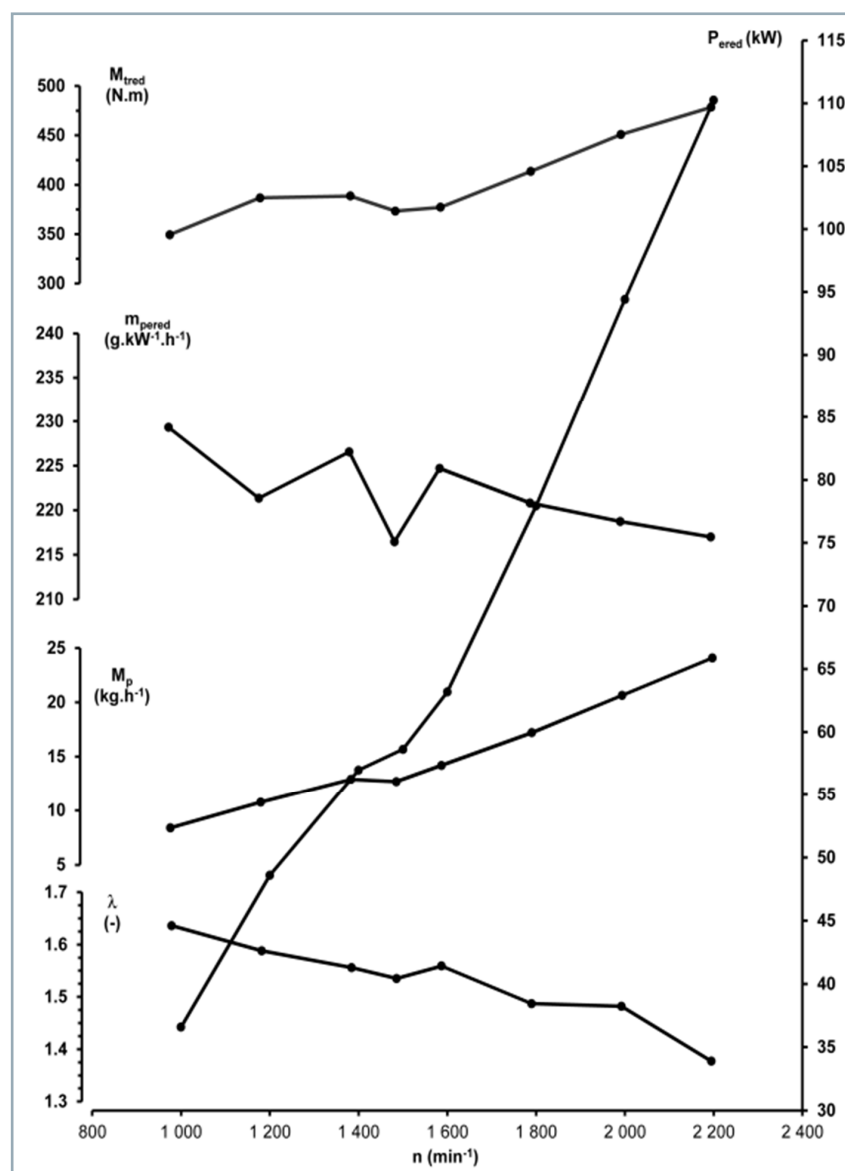


Fig. 5 Main speed characteristics of the UR II 8602.12 NG internal combustion engine

dioxide (CO_2) formation increased as the air excess coefficient shifted closer to a stoichiometric mixture with changing engine speed. The minimum CO_2 value (6.35 vol. %) was recorded at 1000 rpm with an air excess coefficient of $\lambda = 1.636$. Conversely, the maximum CO_2 value (7.63 vol. %) occurred at 2200 rpm, where the air excess coefficient dropped to $\lambda = 1.377$. The increase in the formation of the perfect combustion product is related to the increase in the proportion of fuel in the mixture with air and the perfection of the mixture burnout. Residual oxygen (O_2) in exhaust gases decreased from the original value of 9.6 vol. % for $\lambda = 1.636$ to 7.4 vol. % for $\lambda = 1.377$ by decreasing the excess air coefficient. Unburned hydrocarbon (CH_x) emissions peak at 377 ppm for the lean mixture ($\lambda = 1.636$). Conversely, the minimum CH_x concentration drops to 245 ppm at an air excess coefficient of $\lambda = 1.377$, where the fuel combusts more completely. The last monitored component of exhaust gases was nitrogen oxide production (NO_x). It ranged from 473 ppm for the excess air coefficient $\lambda = 1.636$, when the exhaust gas temperature measured behind the turbine was 485 °C. Increasing the proportion of fuel in the mixture with air ($\lambda = 1.377$) increased the production of nitrogen oxides (2282 ppm), which

is related to temperature increase during combustion.

The temperature increase during combustion is also reflected in the exhaust gas temperature, which reached 650 °C in this case.

The results align with Paul et al. (2017), which indicates that converting diesel engines to CNG generally causes a 5–15% power drop, with torque reductions of up to 25%. These findings corroborate previous literature, where the greatest losses are observed in torque rather than power output. This trend is also confirmed by Banapurmath et al. (2008), who documented a 10–12% decrease in power and an 18–22% reduction in torque when converting an agricultural tractor to CNG/biogas mode.

A comparison with dual-fuel systems is critical. Owczuk et al. (2019) documented that in dual-fuel mode (diesel and biogas) on a tractor engine, biogas containing 50 vol. % CO_2 replaced up to 40% of the diesel fuel while maintaining stable engine operation. The authors noted a performance decrease of only 5–7%, which is less than that observed in the monofuel conversion presented in this work. Similar conclusions were published by Yoon and Lee (2011), who found that the dual-fuel system preserved up to 92–95% of the diesel engine's original power.

Reducing the compression ratio of the UR II 8602.12 NG from the original 17 : 1 to 12 : 1 was necessary for natural gas operation, but it has a significant impact on performance parameters. Wei and Geng (2016) analysed the effect of compression ratio on the performance of CNG engines and found that the optimal compression ratio for converted engines ranges from 11 : 1 to 13 : 1. With a compression ratio of 12 : 1, the engines achieved 8–10% less power compared to the diesel version with a compression ratio of 17 : 1, which exactly corresponds to the results presented in this study. By contrast, specialised CNG engines designed from the ground up for natural gas (such as the Fiat F1C engine used in the Steyr Natural Power tractors) use a compression ratio of 12.5 : 1–13 : 1 and achieve nearly identical performance characteristics to their diesel equivalents. This fact suggests that the conversion of existing diesel engines is always a trade-off between conversion costs and performance parameters.

The most critical finding of this study is the elevated NO_x emissions, which exceed the limit of 500 ppm across the entire measured speed range and peak at 2282 ppm at maximum speed. This finding is consistent with several studies. Cho and He (2007) documented that methane engines naturally have a higher tendency to produce NO_x due to higher combustion temperatures close to the stoichiometric mixture.

Doğan et al. (2019) analysed the effect of the mixture composition on the formation of NO_x in CNG engines. They found that when operating on lean mixtures with $\lambda > 1.5$, NO_x emissions dropped below 500 ppm, albeit at the expense of engine performance, which decreased by 15–20%. For enriched mixtures ($\lambda = 1.3$ – 1.4), similar to the presented study, they documented an increase in NO_x to values of 1800–2500 ppm, which corresponds with the measured data.

Conclusion

Based on the measurements of the main speed characteristics, it can be concluded that the UR II 8602.12 NG engine after conversion to natural gas power achieves a maximum corrected power of 106.5 kW at 2200

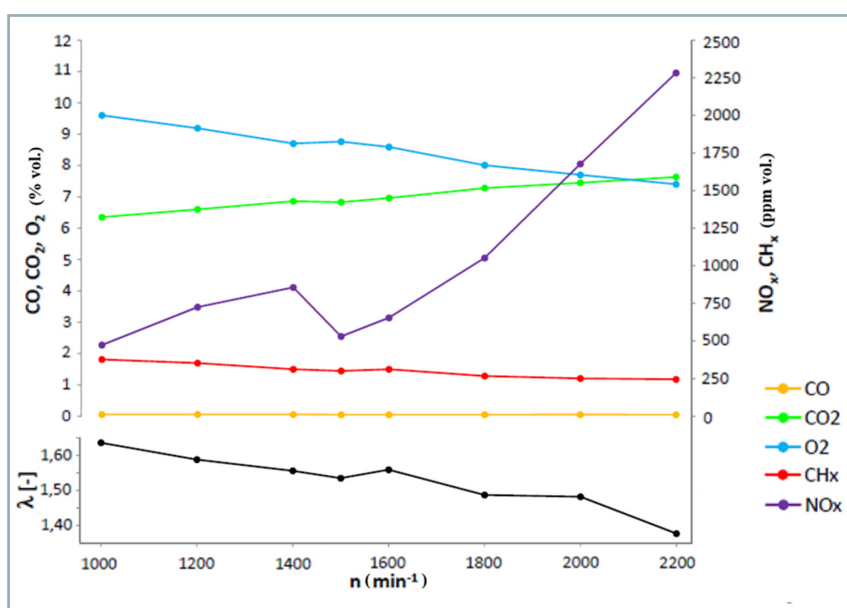


Fig. 6 Composition of exhaust emissions depending on the change in engine speed n (operating conditions: maximum load, ignition advance angle 18° CA before TDC)

rpm, which is a decrease of approximately 8.5% compared to the original diesel engine (116.5 kW). The maximum corrected torque reaches 462 N·m at 2200 rpm, a reduction of around 21% compared to the original diesel version (583 N·m at 1600 rpm). In terms of exhaust gas temperature, values from 485 °C to 650 °C were recorded in the range of the main speed characteristics. After taking into account the thermal gradient at the turbine (approx. +100–130 °C), it is clear that the recommended limit of 630 °C can be locally exceeded in the medium and higher speed range, which poses a potential risk to the service life of exhaust valves and seats. Therefore, it is necessary to ensure operation in the area of moderately lean mixtures, where the temperature of gases is below the limit value.

In terms of emission parameters, the measured values of carbon monoxide (CO) are very low (≈ 0.05 vol. %), while carbon dioxide (CO₂) increases with increasing speed from 6.35 vol. % to 7.63 vol. %, which correlates with gradual enrichment of the mixture and a higher degree of combustion perfection. Concentrations of unburned hydrocarbons (CH_x) decrease from 377 ppm to 245 ppm as the mixture is enriched. From a regulatory standpoint, nitrogen oxides (NO_x) are the most critical component, exceeding the 500 ppm limit across the entire measured speed range and peaking at 2282 ppm at maximum engine speed.

In conclusion, the converted engine retained acceptable performance characteristics, demonstrating a power loss under 9% and a torque decrease under 21% relative to its diesel counterpart. Nonetheless, certain operational risks remain. Specifically, the exhaust gas temperature surpassed the recommended 630 °C threshold under high load conditions, and NO_x formation increased, necessitating subsequent optimisation of mixture composition regulation and thermal management.

The presented conversion of the UR II 8602.12 engine to natural gas achieves comparable results with international studies in the field of conversion of diesel tractor engines and creates a suitable basis for further research in this area.

Future research should focus on the optimisation of combustion control strategies, particularly ignition timing and mixture composition, to reduce NO_x emissions while maintaining engine performance. Furthermore, the implementation of exhaust gas aftertreatment systems, such as three-way catalysts or Exhaust Gas Recirculation (ERG) systems, represents a key challenge for practical deployment. Another important area is the long-term durability of engine components under increased thermal load, especially exhaust valves and turbocharger components. Finally, the integration of biogas with variable composition requires adaptive control strategies to ensure stable and efficient engine operation.

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