У результаті проведених теоретичних досліджень виконані розрахунки теоретичної індикаторної питомої витрати палива для різних способів конвертації на газ дизельних двигунів. Розглядались традиційний спосіб дефорсування при конвертації на газ дизельних двигунів за рахунок встановлення додаткових прокладок між головкою і блоком циліндрів, і запропонований спосіб дефорсування за рахунок затримки закриття впускного клапана.

У результаті проведених в лабораторних умовах експериментальних досліджень при різних способах конвертації дизельного двигуна модифікації X17DTL легкового автомобіля Опель встановлено, що при використанні традиційного способу дефорсування двигунів за рахунок встановлення додаткових прокладок між головкою і блоком циліндрів значення індикаторної питомої витрати зросло, в середньому, на 8–9 %. Але при зниженні ступеня стиснення двигуна запропонованим способом за рахунок затримки закриття впускного клапана, експериментальне значення індикаторної питомої витрати не тільки не збільшилось, а й зменшилось, в середньому, на 7–8 %. Зниження ступеня стиснення двигуна за рахунок затримки закриття впускного клапана запропонованим способом здійснювалось шляхом зміни форми кулачків розподільного валу. Для цього кулачки впускних клапанів розподільного валу наплавлялись, а після цього шліфувались до одержання необхідного профілю, при якому відбувалась затримка закриття клапанів у потрібних межах.

Одержані результати дозволяють оптимізувати процеси переведення на газ дизельних двигунів та знизити витрату палива конвертованих двигунів, в середньому, на 15–17 % у порівнянні з газовими двигунами, переобладнання яких здійснено традиційним шляхом.

Ключові слова: альтернативні палива, дизельний двигун, конвертація двигуна на газ, питома витрата палива

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FUEL ECONOMY RAISING OF ALTERNATIVE FUEL CONVERTED DIESEL ENGINES

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1. Introduction

With current production volumes and proved reserves, oil will be enough for mankind for about fifty years. The second energy resource after oil as motor fuel is natural gas. Recently, there has been a tendency of conversion of existing diesel engines to gas and other alternative fuels, but at the moment a number of issues remain unaddressed by producers [1].

In addition, it is indisputable that cars with engines that operate on gas and other alternative fuels have worse fuel consumption compared to similar diesel engines [2]. At present, the issues of economic feasibility in the conversion to alternative fuels in the present conditions are of paramount importance. Therefore, achieving high fuel economy of gas engines is one of the main requirements in the direction of expanding the use of gas fuels.

In general, for the conversion of diesel engines to gas fuel, it is necessary to do the following:

– to install gas cylinder equipment;

– to mount the ignition system;

– to adjust the gas engine control system and optimize ignition timing angles;

– to reduce the compression ratio for the base diesel engine.

The most common solutions to reduce the compression ratio of the base diesel engine in conditions of automobile plants are:

– the use of reduced-compression-height pistons;

– reduction of the piston stroke by replacing the crankshaft.

Easier ways to reduce the compression ratio of the diesel engine in conditions of service stations or autoenterprises are [3]:

– increase in the combustion volume in the head or pistons by milling;

– increase in the thickness or number of gaskets between the engine block and the head.

In this case, the geometric compression ratio is provided in the range of 12–13 units. The specified reduction in the compression ratio leads to a decrease in the indicated efficiency of the gas engine compared to the base diesel engine. In addition, reduction in the indicated efficiency, especially at low loads, causes an increase in throttling losses. This leads to a deterioration of the gas fuel economy, on average, by 10–25 % compared to the base diesel engine. Therefore, a more up-to-date method of reducing the compression ratio of the base diesel engine due to later intake valve closing is proposed. This allows achieving a reduction in the effective compression ratio, making it considerably smaller than geometric. Therefore, work in this direction is undoubtedly relevant. This allows minimizing the problem of detonation combustion of the gas-air mixture and at the same time increasing the indicated engine efficiency and reducing fuel consumption.

2. Literature review and problem statement

Currently, most of the major car manufacturers consider comprehensive solutions to improve the performance of engines: valve timing variation, intake system improvement, engine compression ratio optimization, etc.

Thus, performance improvement of the MTZ-80 wheeled tractor (Belarus) during the conversion of its engine to gas fuel has been studied in [4]. The feature of the technology of converting this tractor engine to gas was the fact that the reduction in the compression ratio was due to the installation of additional gaskets between the cylinder head and block. It should be noted that this method of reducing the engine compression ratio leads to an increased risk of blowing of gaskets between the cylinder head and block.

In [5], the conversion of a diesel engine car to run on natural gas has been studied. In the engine, spark plugs, the electronic ignition system and gas cylinder equipment are installed instead of spray nozzles. The combustion chamber in the piston heads was bored and the compression ratio was reduced from 21 to 13 units. However, the authors have not sufficiently considered the processes of heat exchange in the converted diesel engine.

The study of regulation of the admission period in the gas-converted diesel engines has been made in [6]. It has been found that when the intake valve closing is changed, it is expedient to simultaneously correct the admission period through the valve lift adjustment. But a significant design complication and the associated general significant increase in the cost of the converted engine remained unaddressed.

In [7], the laws of changes in the compression ratio and the intake valve closing and exhaust valve opening angles, depending on the engine load, have been studied. The geometric compression ratio was controlled by a moving-head piston. Comparative calculations of load characteristics of the base and converted engines have been performed. An interesting feature of [7] is the comparison of working processes, taking into account exhaust gas recirculation. The drawback of [7] is the insufficiently fast operation of the piston system with the adjustable engine compression ratio.

The scheme with early intake valve closing has been investigated in [8]. This scheme is based on the intake valve closing at a time when the effective compression ratio reaches a predetermined value. In this case, the limitation of a part of the intake stroke for the mixture admission from the intake manifold is required. Then the valve is closed and the part of the cylinder, which remains at the bottom dead center of the intake stroke, becomes isolated. After the intake valve closing, the charge that appeared in the cylinder first expands to below atmospheric pressure due to the piston movement down during the last part of the intake stroke. Then the charge is compressed again on the compression stroke. The obvious drawback of the proposed scheme is that such early closing of the intake valve reduces admission.

The idea to use the prolonged working fluid expansion cycle instead of the Otto constant-volume cycle, which is the basis for the functioning of the majority of modern spark engines has been implemented in [9]. In this cycle, the expansion ratio was greater than the compression ratio. But such converted engine will have a too complex kinematic diagram of the power mechanism.

The method of implementation of the prolonged expansion cycle due to late intake valve closing to reduce the maximum exhaust gas temperatures has been proposed in [10]. But the converted engine had lower power performance than the base engine.

In [11], the performance of the Opel X17DTL diesel engine converted to run on liquefied propane-butane has been investigated. The reduction of the compression ratio of the converted engine was due to the installation of two additional gaskets between the cylinder head and block. It should be noted that this method of reducing the compression ratio leads to a decrease in the indicated efficiency of the gas engine compared to the base diesel engine.

Thus, the unresolved problem in the conversion of diesel engines to gas is the increased fuel consumption and low indicated efficiency. Therefore, an effective method to ensure a high indicated efficiency of the gas-converted diesel engine by reducing the compression ratio of the base engine due to later intake valve closing has been proposed.

3. The aim and objectives of the study

The aim of the work is to increase the fuel economy of gas-converted diesel engines by reducing the compression ratio of the base diesel engine due to later intake valve closing.

To achieve this aim, the following objectives are set:

– to perform a theoretical research of changes in the indicated specific fuel consumption for gas-converted diesel engines, where the reduction of the compression ratio of the base diesel engine is achieved due to later intake valve closing;

– to carry out an experimental research of changes in the indicated specific fuel consumption for the gas-converted diesel engine, where the reduction of the compression ratio is achieved by the traditional method and the proposed method of later intake valve closing.

4. Calculation of the indicated specific fuel consumption of the converted engine of a new design

The use of prolonged expansion cycles due to late intake valve closing in engines allows:

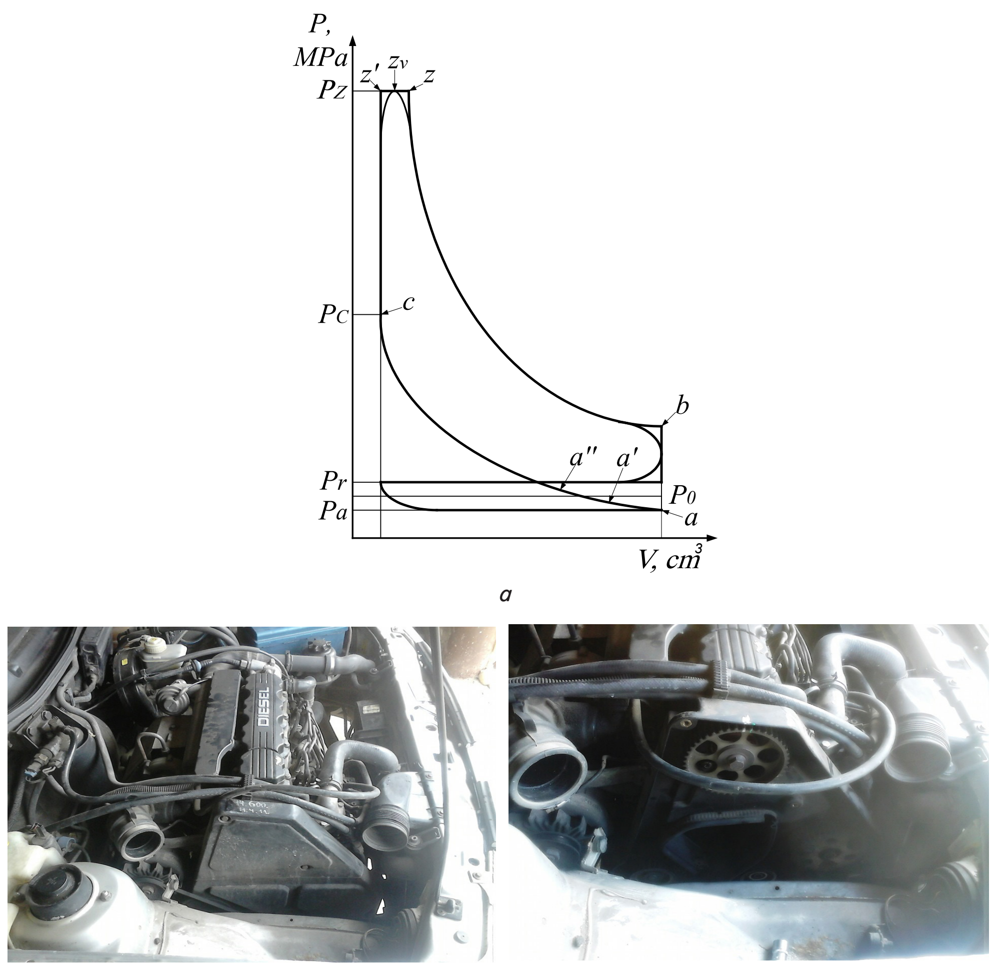
– reducing the requirements to the anti-knock quality of the gas-air mixture when reducing the actual compression ratio in the engine cylinder;

– implementing the concept of prolonged expansion when reducing the actual compression ratio and, accordingly, increasing the cycle efficiency;

– reducing the intake depression due to the reverse release of the fuel mixture, which will reduce pumping losses in all throttling modes during engine operation.

An objective index of the efficiency of any thermal power plant, including diesel engines converted to gas fuels, is the indicated efficiency. High values of the latter directly affect the fuel economy and performance of engines. The method to increase the indicated efficiency by reducing the compression ratio of the base diesel engine due to later intake valve closing is shown in Fig. 1. The scheme in Fig. 1, *a* demonstrates the shift of the intake valve closing point on the indicator diagram. Point *a'* is the moment of closing of the intake valve of the base diesel engine, point *a"* is the moment of closing of the intake valve of the converted engine. To shift the angle of closing of the intake valve of the Opel X17DTL diesel engine (Fig. 1, *b*), cranking of the camshaft drive gear of the valvetrain was carried out (Fig. 1, *c*).

The calculation of the indicated efficiency η_i is based on the comparison of the amount of supplied and removed heat. In the simplified calculations, when determining the indicated efficiency of the cycle, only the heat that is supplied and removed in isochoric and isobaric processes is taken into account, and compression and expansion processes are taken without heat exchange, i. e., as adiabatic. In this calculation, when determining the thermal efficiency of the cycle, heat exchange in the compression and expansion strokes was also taken into account. Similar considerations were also used when determining the average theoretical pressure with different methods of organization of engine cycles.



a

b *c*

Fig. 1. Implementation of the method of reducing the compression ratio of the base diesel engine due to later intake valve closing: *a* – shift of the intake valve closing point *a'* – *a''* on the indicator diagram; *b* – Opel Astra diesel engine converted to the propane-butane mixture; *c* – camshaft drive gear

In the calculations, the authors have made the following assumptions:

– the working fluid is a real gas, the heat capacity of which is constant, i. e. it does not depend on temperature T and pressure P;

– the chemical composition of the working fluid varies, its mass is fixed and depends on the cylinder capacity;

– the duty cycle is closed and reverse, that is, the intake and exhaust periods are not taken into account;

– there is no intake of fresh mixture and release of combustion products;

– there are no energy losses in the cycle, no pumping losses;

– the influence of fuel is taken into account through the lower heating value and the molecular weight of fuel;

– the combustion process is carried out at the end of compression at a constant volume and continues at a constant pressure;

– the cycle is completed at the end of expansion by the process of heat transfer from the working fluid to an external cold source at a constant volume, and then turning the working fluid into the initial state – at a constant pressure;

– the amount of heat transferred is proportional to the mass and combustion heat of a real fuel mixture.

The energy efficiency of engines is estimated by the indicated efficiency η_i, which is defined as the ratio of heat to perform the useful work A_u to the supplied heat Q_s obtained as a result of combustion of the fuel-air mixture

$$\eta_i = \frac{A_u}{Q_s} = \frac{Q_s - Q_r}{Q_s} = 1 - \frac{Q_r}{Q_s}, \tag{1}$$

where Q_r is the amount of removed heat, kJ.

The amounts of heat supplied to engine cylinders during combustion of the fuel-air mixture and removed (through the exhaust system, cooling system, etc.) heat are determined by temperatures in working processes

$$Q_s = m_w C_{v.w.}(T_{z'} - T_c) + m_w C_{p.w.}(T_z - T_{z'}), \tag{2}$$

$$Q_r = m_c C_{v.c.}(T_b - T_a) + m_c C_{p.c.}(T_a - T_{a''}), \tag{3}$$

where $C_{v.w.}$, $C_{v.c.}$ are isochoric heat capacities, respectively, of the working mixture and combustion products; $C_{p.w.}$, $C_{p.c.}$ are isobaric heat capacities, respectively, of the working mixture and combustion products; m_w, m_c are the masses of the working mixture and combustion products; $T_{z'}$, T_c, T_z, T_b, T_a, $T_{a''}$ are the temperatures, respectively, of the working fluid at the beginning of the previous expansion (Fig. 1), at the end of compression, at the end of the previous expansion, at the end of the next expansion, at the beginning of the geometric compression process, at the beginning of the actual compression process.

Then, from (1)–(3), the thermodynamic efficiency coefficient η_i will be

$$\eta_i = 1 - \frac{mC_{v.c.}(T_b - T_a) + mC_{p.c.}(T_a - T_{a''})}{mC_{v.w.}(T_{z'} - T_c) + mC_{p.w.}(T_z - T_{z'})} = 1 - \frac{C_{v.c.}\left((T_b - T_a) + n_2(T_a - T_{a''})\right)}{C_{v.w.}\left((T_{z'} - T_c) + n_1(T_z - T_{z'})\right)}, \tag{4}$$

where n_1 is the average polytropic compression index, n_2 is the average polytropic expansion index.

The polytropic expansion index is assumed to be n_2=1.265, the polytropic compression index – n_1=1.364 [12].

Volumes $V_{a''}$, temperatures $T_{a''}$ and pressures $P_{a''}$ of the working fluid in the relation (4) at the end of late intake valve closing *a"* for the converted engine are calculated as follows:

$$V_{a''} = \frac{V_a - V_c}{2}\left[(1 - \cos\phi) + \frac{R}{4L}(1 - \cos 2\phi)\right],$$

$$T_{a''} = T_a\left(\frac{V_a}{V_{a''}}\right)^{n_1 - 1}, \quad P_{a''} = P_a\left(\frac{V_a}{V_{a''}}\right)^{n_1},$$

where R is the crank radius; L is the crank length; φ is the set angle of late intake valve closing.

Temperatures, pressures and volumes of the working fluid in the relation (4) at the end of the processes of intake a, compression c and expansion b for the converted engine are determined by the formulas:

$$T_a = \frac{P_a T_b}{P_b}, \quad V_a = \frac{V_h \cdot \varepsilon}{\varepsilon - 1} = V_c \cdot \varepsilon,$$

$$T_b = \frac{T_z}{\rho^{n_2 - 1}}, \quad P_b = \frac{P_z}{\varepsilon^{n_2}},$$

$$T_c = T_a \varepsilon^{n_1 - 1}, \quad P_c = P_a \varepsilon^{n_1}, \quad V_c = \frac{V_a}{\varepsilon} = \frac{V_{a''}}{\varepsilon_v},$$

where ε is the geometric compression ratio of the engine, $\varepsilon = V_a / V_c$; V_h is the converted engine capacity; ρ is the ratio of the previous expansion of the working fluid, $\rho = \mu \cdot T_z / (\lambda \cdot T_c)$; δ is the ratio of the next expansion of the working fluid, $\delta = \varepsilon / \rho$; ε_v is the actual compression ratio of the engine, $\varepsilon_v = V_{a''} / V_c$.

Temperatures, pressures and volumes in the relation (4) in the combustion process for the converted engine are calculated as follows:

$$T_z = \frac{T_{z'} V_z}{V_{z'}} = \frac{T_{z'} V_z}{V_c} = T_{z'} \cdot \rho, \quad T_{z'} = \frac{\lambda \rho T_c}{\mu},$$

$$P_z = \frac{P_c \cdot T_z}{T_A}, \quad V_z = V_c\,\rho,$$

where λ is the ratio of pressure increase in the working fluid, μ is the coefficient of molecular change of the working mixture.

By determining the indicated efficiency η_i, the theoretical indicated specific fuel consumption g_i, g/kWh is calculated by the formula:

$$g_i = 3600 / (H_u \cdot \eta_i),$$

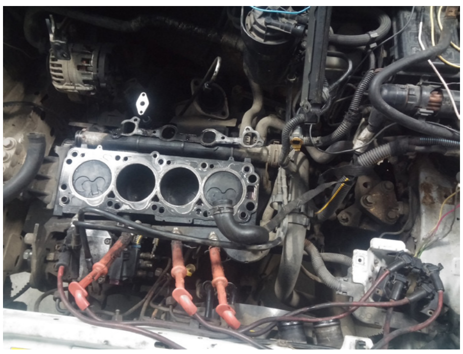
where H_u is the lower heating value, MJ/kg.

5. Methods and materials of the experimental research of fuel-economy characteristics of the diesel and converted engine

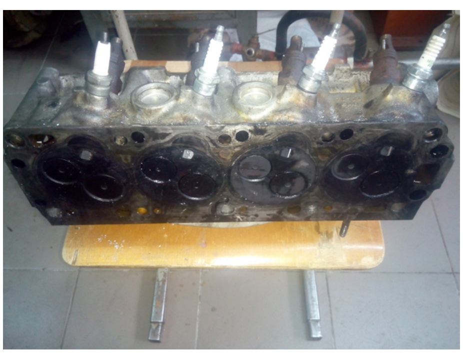
The aim of the experimental research is to determine the indicated specific fuel consumption of the gas-converted diesel engine. To achieve this aim, the Opel X17DTL diesel engine was converted to run on liquefied propane-butane in the Ivano-Frankivsk National Technical University of Oil and Gas (Ukraine).

For the conversion of the Opel diesel engine to gas fuel, gas cylinder equipment of Italian production was installed. In addition, the compression ratio of the converted engine was reduced, the electronic DIS ignition system of own development was installed and the operation of the converted engine control system was optimized (Fig. 2).

a



b



c

Fig. 2. Opel Astra passenger car with the gas-converted diesel engine: *a* – X17DTL engine; *b* – the engine with the dismantled head and installed ignition system; *c* – the head of the converted engine with plugs and diesel nozzles

The first engine conversion and reduction of compression ratio were made by installing additional gaskets under the cylinder head. The experimental research of fuel-economy and environmental characteristics of the converted engine was carried out [11]. The feature of the second conversion is the reduction of the actual compression ratio due to late intake valve closing. Table 1 shows a brief technical description of the Opel Astra X17DTL converted diesel engine.

The indicated specific fuel consumption g, g/kWh was experimentally determined through the engine mechanical efficiency η_{en} and transmission η_{tr}, hourly fuel consumption G and effective wheel power N_e as follows:

$$g = G/\left(N_e \cdot \eta_{en} \cdot \eta_{tr}\right).$$

The effective wheel power N_e, kW was determined by the formula

$$N_e = \pi \cdot \frac{M_e \cdot n_x}{3 \cdot 10^4 \cdot i},$$

where M_e is the effective wheel torque, Nm; n_x is the engine speed, min^{-1}; i is the transmission ratio.

The effective wheel torque was determined using the GOSNITI KI-8964 traction-brake stand [11].

The engine mechanical efficiency η_{en} and transmission η_{tr} were determined through the mechanical loss power of the engine N_{en} and the mechanical loss power of the transmission N_{tr}.

$$\eta_{en} = \frac{N_e}{N_e + N_{en}}, \quad \eta_{tr} = 1 + \frac{N_e + N_{en}}{N_{tr}}.$$

The mechanical loss powers were determined when measuring the power consumption of the electric motor, which was interlocked by the retainer with the drive wheel. The mechanical loss power of the transmission N_{tr} was measured in the neutral position of the gearbox. The mechanical loss power the engine N_{en} and the mechanical loss power of the transmission N_{tr} were measured in the highest gear.

The studied materials and equipment used in the experiments are given in detail in [11].

Table 1

Brief technical description of the Opel X17DTL converted diesel engine

No.	Parameter	Value	
		Reduction of the actual compression ratio by installing additional gaskets under the head [11]	Reduction of the actual compression ratio by late intake valve closing
1	Base engine	Diesel, with Bosch EDC 15M injection system	
2	Converted engine	Gas (propane-butane), with the IFN-TUNG control system	
3	Engine capacity, cm^3	1,669	
4	Maximum power, kW(hp)/rpm, min^{-1}	50(68)/4,400	
5	Maximum torque, Nm/rpm, min^{-1}	130/2,000	
6	Base diesel engine compression ratio	22.0	
7	Base diesel engine intake valve closing angle	30°	
8	Converted engine intake valve closing angle	30°	52.5°
9	Converted diesel engine compression ratio	13.1	13.2
10	Thickness of gaskets under the cylinder head	4.2 mm	1.4 mm (nominal)

6. Results of the research of the indicated specific fuel consumption of diesel engines converted to gas fuel

To obtain the initial data for comparison in order to evaluate the efficiency of the gas-converted diesel engine, the experimental research of three options of the converted engine was carried out. The first option is the gas-converted diesel engine with the base compression ratio of 22.0. The second option is the gas-converted diesel engine, where the reduction of the compression ratio of the converted engine to 13.1 was due to the installation of two additional gaskets between the cylinder head and block. The third option is the gas-converted diesel engine, where the reduction of the compression ratio of the converted engine to 13.2 was due to the 22.5° later intake valve closing.

Fig. 3 shows the dependencies of the indicated specific fuel consumption and torque of the Opel X17DTL gas-converted diesel engine with the base compression ratio of 22.0 on changes in the crankshaft speed.

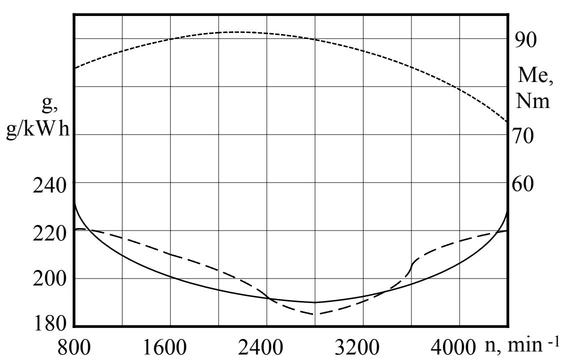


Fig. 3. Dependencies of the indicated specific fuel consumption and torque of the Opel X17DTL gas-converted diesel engine with the base compression ratio of 22.0 on changes in the crankshaft speed: ——— – indicated specific fuel consumption, theoretical dependence; – – – – – indicated specific fuel consumption, experimental dependence; ········ – engine torque, experimental dependence

According to the results of the experiments, the value of the indicated specific fuel consumption at the nominal mode was 220 g/kWh, the minimum value of the indicated specific fuel consumption was 185 g/kWh. Herewith, the value of the theoretical indicated specific fuel consumption at the nominal mode was 229 g/kWh, while the minimum theoretical value of the indicated specific consumption was 191 g/kWh. It should be noted that in order to avoid knocking of the engine with the basic compression ratio of 22.0 when running on gas, the engine torque did not exceed 70 % or 91 Nm. Obviously, achieving the first option of acceptable power characteristics is not possible in principle.

Fig. 4 shows the dependencies of the indicated specific fuel consumption and torque on changes in the speed of the X17DTL gas-converted diesel engine with the compression ratio of 13.1, where the reduction of the compression ratio of the converted engine was due to the installation of additional gaskets between the cylinder head and block.

As a result of the tests, the characteristics of the gas-converted diesel engine to obtain the original reference base were obtained. According to the results of the experiments, the value of the indicated specific fuel consumption at the nominal mode was 238 g/kWh, the minimum value of the indicated specific fuel consumption was 203 g/kWh. Herewith, the theoretical indicated specific fuel consumption at the nominal mode was 247 g/kWh, while the minimum theoretical value of the indicated specific fuel consumption was 211 g/kWh. Thus, with a decrease in the engine compression ratio due to the installation of additional gaskets, the experimental value of the indicated specific fuel consumption, depending on changes in the crankshaft speed, increased in comparison with the first conversion option in the range from 8.1 to 9.7 %. The theoretical value of the indicated specific fuel consumption increased from 3.9 to 7.9 %.

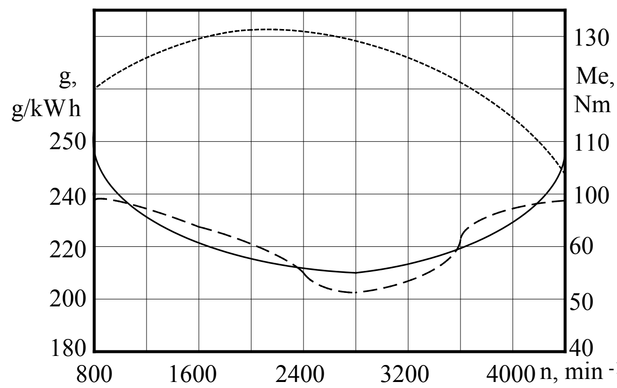


Fig. 4. Dependencies of the indicated specific fuel consumption and torque of the Opel X17DTL gas-converted diesel engine with the compression ratio of 13.1 due to the installation of additional gaskets on changes in the crankshaft speed: ——— – indicated specific fuel consumption, theoretical dependence; – – – – – indicated specific fuel consumption, experimental dependence; ········ – engine torque, experimental dependence

Fig. 5 shows the dependencies of the indicated specific fuel consumption and torque on changes in the speed of the Opel X17DTL gas-converted diesel engine with the compression ratio of 13.2, where the reduction of the compression ratio of the converted engine was due to late intake valve closing.

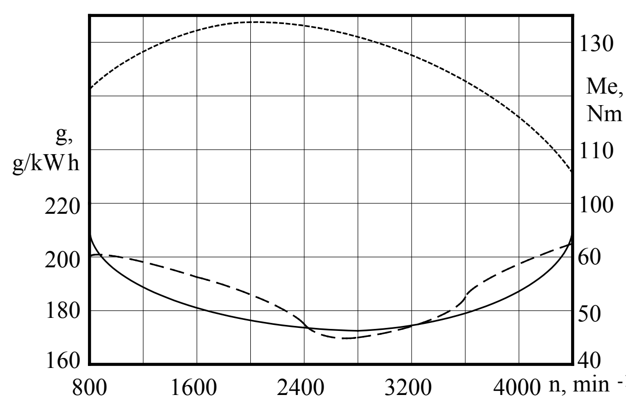


Fig. 5. Dependencies of the indicated specific consumption and torque of the Opel X17DTL gas-converted diesel engine with the compression ratio of 13.2 due to late intake valve closing on changes in the crankshaft speed: ——— – indicated specific consumption, theoretical dependence; – – – – – indicated specific consumption, experimental dependence; ········ – engine torque, experimental dependence

According to the results of the experiments, the value of the indicated specific fuel consumption at the nominal mode was 204 g/kWh, the minimum value of the indicated specific consumption was 160 g/kWh. Herewith, the theoretical indicated specific consumption at the nominal mode was 206 g/kWh, while the minimum theoretical value of the indicated specific consumption was 174 g/kWh. Therefore, with a decrease in the engine compression ratio due to late intake valve closing, the experimental value of the indicated specific consumption decreased in comparison with the traditional conversion option in the range of 7.3–8.1 %. The theoretical value of the indicated specific fuel consumption decreased from 8.8 to 10.4 %.

7. Discussion of the results of the research of the indicated specific fuel consumption of diesel engines converted to gas fuel

The research is a continuation of the study of the methods of conversion of diesel engines to gas fuels [11].

The theoretical calculations and experimental research are useful as they demonstrate the effectiveness of the method of optimizing the engine working process due to later intake valve closing. The theoretical and experimental research shows the possibility of a significant reduction, on average, by 11–14 %, of the indicated specific fuel consumption depending on the engine torque. This, in turn, allows expecting a proportional reduction in the operating fuel consumption for gas-converted diesel engines by the proposed method of reducing the compression ratio by later intake valve closing. It should be noted separately that the specified figures of reduction of fuel consumption of engines with the mechanism of later intake valve closing, as the experiments have shown, will be obtained when providing power at the level of the base engine.

As a significant advantage of the research, it can be noted that the theoretical calculations performed are well correlated with the experimental results. Therefore, the research can be recommended and used by specialists of automobile plants and motor-construction enterprises in the design of new and conversion of existing engines to alternative gas fuels. The results obtained allow optimizing the designs of power systems and valvetrains of gas-converted diesel engines. At the same time, restrictions on the use of the proposed method in terms of the results of fuel economy are obvious. Excessively late intake valve closing may result in reverse fuel emissions into the intake manifold and reduction of the effective compression ratio of the engine. This, in turn, will lead to a deterioration of the combustion process and an increase in fuel consumption.

Therefore, further research into the methods of conversion of diesel engines to gas fuel should be focused on the optimization of valve timing, which is associated with significant experimental difficulties.

8. Conclusions

1. As a result of the theoretical research, the calculations of the theoretical indicated specific fuel consumption for different methods of conversion of diesel engines into gas engines were made. In this case, the calculations showed that for the proposed method of conversion of diesel engines to gas with the 22.5° later intake valve closing, the indicated specific fuel consumption is reduced in the range of 8–10 %.

2. As a result of in vitro experimental research with different methods of conversion of the Opel X17DTL diesel engine, the dependencies of the indicated specific fuel consumption on the crankshaft speed were determined. It was found that using the traditional method of engine derating due to the installation of additional gaskets between the cylinder head and block, the value of the indicated specific fuel consumption increased, on average, by 8–9 %. But with a decrease in the compression ratio of the engine by the proposed method of late intake valve closing, the experimental value of the indicated specific consumption not only did not increase, but decreased, on average, by 7–8 %. The main reason for the reduction of the indicated fuel consumption is that in the engine converted by this method, the expansion ratio will be greater than the compression ratio. And this, accordingly, will increase the indicated efficiency of the engine. In addition, with a decrease in the compression ratio of the engine by this method, the intake depression is decreased, which leads to a reduction in pumping losses in all modes. And this, accordingly, also reduces fuel consumption of the converted engine.

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Розроблено інтегровану систему підтримки функціонування теплонасосного енергопостачання на основі прогнозування зміни температури місцевої води. Зміна витрати пари холодагента, числа обертів електродвигуна компресора відбувається при вимірюванні температури холодагента на виході із конденсатора, тиску випаровування, тиску конденсації та частоти напруги.

Виконано комплексне математичне моделювання теплонасосної системи, що базується на інтегрованій системі підтримки розряду грунту на рівні 10–8 °С. Визначено витрату холодагента, потужність електродвигуна компресора, напругу, частоту напруги, число обертів електродвигуна компресора, коефіцієнт продуктивності теплонасосної системи для встановлених рівнів функціонування. Встановлено параметри конвективного теплообміну в конденсаторі, постійні часу та коефіцієнти математичних моделей динаміки зміни температури місцевої води, витрати пари холодагента, числа обертів електродвигуна компресора.

Здобуто функціональну оцінку зміни температури місцевої води в діапазоні 35–55 °С впродовж опалювального сезону, витрати пари холодагента, числа обертів електродвигуна компресора. Визначення підсумкової функціональної інформації надає можливість приймати наступні випереджуючі рішення: на підтримку зміни тиску випаровування щодо зміни витрати пари холодагента для цифрового управління; на підтримку зміни тиску випаровування щодо зміни витрати пари холодагента та на зміну частоти напруги щодо зміни числа обертів електродвигуна компресора для частотного управління. Тому, запропоновано прогнозування зміни температури місцевої води на основі вимірювання температури холодагента на виході із конденсатора. Саме ця оцінка у співвідношенні з вимірюваним тиском випаровування, входить до складу аналітичних визначень витрати холодагента та числа обертів електродвигуна компресора. Здобуття такої оцінки та вимірювання частоти напруги надає можливість упереджено впливати на узгодження функціонування зовнішнього та внутрішнього контурів теплонасосної системи як при цифровому, так і частотному управлінні

Ключові слова: теплонасосна система, частотне управління, цифрове управління, тиск випаровування, тиск конденсації

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DEVELOPMENT OF ENERGY-SAVING TECHNOLOGY FOR MAINTAINING THE FUNCTIONING OF HEAT PUMP POWER SUPPLY

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1. Introduction

Under conditions for saving natural fuel and decreasing harmful emissions into the atmosphere, heat pump power supply with the use of renewable power sources is gaining further development [1–3]. Thus, for example, the heat pump, using fermented must as a low-potential energy source, is recommended in order to maintain the operation of the biogas plant as a part of a cogeneration power system. Due to additional biogas production, the proposed technology provides an opportunity to increase marketability of a biogas plant and decrease the cost value of production of electricity and heat in the range of 20–30 % [3].

The outside air as a low-potential power source is available in heat pump power supply, but the change of air temperature in a wide range makes it difficult to maintain functioning of heat pump systems [1].

A heating system of the soil–water type requires the construction of special soil heat exchangers for heating the brine – 30 % ethylene glycol solution that is fed to the evaporator of a heat pump [2]. A vertical heat exchanger occupies a smaller area than a horizontal one but requires additional capital investments for drilling wells. It is possible to ensure heat extraction within 30–100 W per one meter of the length of a vertical heat exchanger depending on a soil type at the depth of 40–150 m, where the soil temperature