E3S Web of Conferences **417**, 03018 (2023) https://doi.org/10.1051/e3sconf/202341703018
GEOTECH-2023

Environmental indicators of dual-fuel hydrogen engine

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Abstract. Hydrogen is considered as a promising gas engine fuel for diesel engines. The problems that occur when converting engines to work on hydrogen are presented. Reliable ignition of hydrogen in the engine is achieved by implementing a two-fuel cycle. In this case, hydrogen ignites from the diesel fuel combustion. Calculations of diesel fuel and hydrogen supply effect on the workflow of a dual-fuel engine of the D-245 type were realized. The main indicators of the engine are calculated when the hydrogen supply changes from 0 to 80%. A criterion characterizing the total toxicity of engine exhaust gases is proposed. The optimal supply of hydrogen was 40%. With such a supply of hydrogen, there was a decrease in the smokiness of exhaust gases by 53%, carbon dioxide emissions by 44%, but the emission of nitrogen oxides increased by 27%. With an increase in the supply of hydrogen from 0 to 40%, the maximum calculated effective performance of engine increased by 7.1%.

1 Introduction

The current stage of development of piston engine building is characterized by active research into the possibilities of replacing traditional motor fuels with alternative raw materials fuels [1, 2]. Hydrogen is one of the most attractive alternative motor fuels [3, 4]. Close attention to the use of hydrogen as an energy carrier for various stationary and mobile power plants is explained by the widespread desire to decarbonize the economy [5]. In addition, such qualities of hydrogen as high energy intensity per unit weight, approximately three times higher than the mass energy intensity of petroleum motor fuels, and the absence of incomplete combustion products of hydrocarbons in the exhaust gases (EG), make it possible to use hydrogen in various sectors of the economy without significant harm to the environment [6, 7, 8]. In this regard, global consumption of pure hydrogen is expected to grow from 75 million tons in 2021 to 100 million tons by 2030, or from 119 ml. tn. (the current volume of consumption of this product) to 156 ml. tn. The maximum increase in hydrogen consumption is expected in the transport industry – 12 million tons.

Hydrogen combustion process and the phenomenon of detonation in the engine differ significantly from the hydrocarbon fuels combustion. A very short period of ignition delay of hydrogen-air mixture (stoichiometric and close to it) and a high combustion speed create

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problems of detonation, "rough" combustion and reverse flares in the engine intake tract [9, 10, 11]. Significant differences in the properties of hydrogen and petroleum fuels determine the peculiarities of using hydrogen as fuel for internal combustion engines. Thus, the self-ignition temperature of diesel fuel (DF) and hydrogen in the combustion chamber (CC) of a diesel engine are respectively 250-300 and 510-590 °C, and their cetane numbers are about 45 and 1, the flame expansion rate in mixtures with air at different excess air ratio coefficients are 0.35–0.40 and 1.6-2.6 m/s. Combustion rate of hydrogen-air mixtures is significantly reduced when switching to poor air - hydrogen mixtures [12, 13]. Among the steps to reduce the rigidity of the combustion of hydrogen-air mixtures, we will highlight the EGR organization, increasing the humidity of these mixtures (water supply at the CC), injection advance angle of fuel adjustment (late ignition).

Hydrogen is an environmentally friendly fuel. Since hydrogen fuel does not contain carbon atoms, theoretically there should be no soot (carbon C), unburned hydrocarbons, carbon oxides CO and CO_2 in the composition of its combustion products. The toxic components emissions source is only burnt engine oil. However, the magnitude of such emissions is very insignificant. Hydrogen H_2 is a unique fuel (along with ammonia NH_3), the use of which can almost completely eliminate the emission of carbon dioxide entering the atmosphere in large quantities. And the burning of carbon-based fuels leads to annual emissions of carbon dioxide into the atmosphere, which exceed 32 Gt. The product of combustion (oxidation) of hydrogen is water H_2O – complete hydrogen oxide. But the high combustion temperature of the working mixture and the intense oxidation of nitrogen contained in the air leads to a significant emission of nitrogen oxides NOx during the combustion of hydrogen in the internal combustion engine.

Injecting the ignition dose of diesel fuel into the cylinders is operative method of hydrogen ignition in the combustion chamber [6, 10]. Operation of the engine in non-forced modes is characterized by a reduced temperature of the working mixture. Therefore, the ignition of hydrogen by the heat of combustion of DT provides the required ignition energy Such an organization of the working process makes it possible to solve the problem of hydrogen ignition and its combustion with acceptable indicators of combustion dynamics and with improved indicators of exhaust gas toxicity. The turbocharged D-245 diesel engine (4 H 11 / 12.5) (without charge air cooling) has N_e= 84 kW initial power at n= 2400 min^{-1}. It was selected as the object of research. This engine has a compression ratio of ε= 15, a cast-iron cylinder head and an aluminum piston with a deep combustion chamber of the CNID type (developed by the Central Diesel Research Institute).

2 Computational study of a dual-fuel hydrogen engine

The calculation study was carried out by using the DIESEL-RK software package developed by Professor A.S. Kuleshov at BMSTU. It is intended for thermodynamic calculation and optimization of internal combustion engine work processes [14]. In this study, it is assumed that diesel fuel was injected into the combustion chamber (CC) by a regular diesel nozzle. Hydrogen gas was supplied to the engine intake system near the intake valve of each cylinder. The working process of this engine is investigated at its nominal operating mode (maximum power output). When transferring this engine to work using diesel fuel and hydrogen while maintaining the total calorific value of fuels, a significant increase in maximum combustion pressures and temperatures was noted. In this regard, in order to obtain acceptable pressures and cycle temperatures of this engine, its initial compression ratio ε = 15 was reduced to ε = 8. The fuel injection advance angleθ has been reduced from 14 to 5 degrees of rotation of the crankshaft to the upper top dead centre. In addition, the engine was deforced to the crankshaft speed n = 2200 rpm and, accordingly, to the power N_e = 77 kW.

The calculations investigated the main indicators of the D-245 engine in a purely diesel cycle and in a gas-diesel cycle with diesel fuel supply for ignition and admission of hydrogen equal to 5, 10, 20, 40, 60, 80% (with taking into account the difference in the calorific value of DF and hydrogen H_2). The cyclic supplies of DF m_{DF} and hydrogen m_{H2} for these cases are presented in Table 1.

Table 1. The main indicators of the D-245 diesel engine in the implementation of a gas-diesel cycle with an ignition DF and hydrogen supplies equal to 5, 10, 20, 40, 60 and 80%.

Indicators	The method of organizing the workflow (№ of the energy share of hydrogen in the total fuel supply)						
	№1	№2	№3	№4	№5	№6	№7
The energy share of DF in the total fuel supply E_{DF}, %	100	95	90	80	60	40	20
The energy share of hydrogen in the total fuel supplyE_{H2}, %	0	5	10	20	40	60	80
Cyclic supply of DF, m_{DF}, g	0.0661	0.0628	0.0595	0.0529	0.0397	0.0264	0.0132
Cyclic supply ofH_2, m_{H2}, g	0	0.00117	0.00234	0.00468	0.00936	0.01405	0.01873

When modelling the working process of a dual-fuel hydrogen engine, a three-zone combustion model was used: the jet zone before the end of injection (the volume occupied by the increasing burning jets of the ignition diesel engine); the jet zone during its combustion or the main zone (the rest of the cylinder volume except for the jet zone), in which the homogeneous mixtures of air, residual combustion products of DF and H2; the zone of activation of gas combustion, in which the working fluid is heated to a state, as a result of which the steady combustion of the main zone begins.

The moment of hydrogen ignition in the main zone can be estimated by reaching a set temperature in the activation zone Tactiv= 1100 K, or by accumulating the Livengood-Wu integral (LW) according to the current activation zone parameters [14]:

$$LW = \int_0^{\tau_i} \frac{d\tau}{\tau_{iT}} \tag{1}$$

where, dτ is the time step for calculation [s]; τi is the self-ignition delay period (SIDP) [s]; τiT is the theoretical SIDP depending on the instantaneous values of the activation zone gas parameters [s]. It is assumed that ignition occurs when Livengood-Wu integral becomes equal to 1.

The theoretical SIDP τiT was determined at each computational step by calculating the detailed chemical kinetics of the pre-flame reactions, using the CHEMKIN software package. Due to the high detonation activity of a mixture of hydrogen and air, the assessment of the readiness of this mixture for detonation only by the magnitude of the integral LW is inconvenient, since depending on the conditions in the combustion process, it can vary widely. Therefore, the work uses the LWknock charge knock readiness indicator, which is determined as follows. If the gas-air mixture is poor, the compression ratio is small, then the integral LW in the main zone increases slowly and by the end of the burnout of the fresh charge integral does not reach 1. In this case, it is assumed that there are no conditions for detonation, combustion is detonation-free and LWknock= LWz. As the hydrogen-air mixture is enriched, the combustion rate and the tendency to detonation increase, while the value of the integral LW can reach 1 until the end of the hydrogen burnout. In the scale of the relative duration of hydrogen burnout in the main zone, the moment of reaching LW=1 is designated as φLWz. For the condition φLWz< 1, the indicator of the readiness of the working mixture for detonation is calculated as: LWknock= 2 –φLWz(where φLWzis the relative duration of hydrogen burnout). Thus, the indicator of the readiness of the charge for detonation describes the quantitative and qualitative propensity of the mixture in the cylinder to detonation: at LWknock< 1, detonation does not occur, at 1 <LWknock< 2, detonation occurs during the combustion of hydrogen; at 2 <LWknock, detonation occurs even before the start of its acceptable combustion. The detonation model is identified by clarifying the kknock coefficient in the formula below, which is used to calculate the LW integral when modeling this process:

$$LW = \int_0^{\tau_i} \frac{d\tau}{k_{knock}\, \tau_{iT}} \qquad (2)$$

After analyzing the experimental data [18], $k_{knock}= 1,7$ was assumed.

According to the calculated data obtained for the D-245 engine, an analysis of its main indicators was carried out when implementing a gas-diesel cycle with different energy fractions of H_2 in the fuel supply E_{H2}. At the same time, it was necessary to analyze in more detail the exhaust gas toxicity indicators of this hydrogen engine and determine the factors influencing these environmental indicators. In addition, it is advisable to introduce a generalized criterion for the optimality of the supply of DF and H_2, characterizing the total toxicity of the exhaust gas of the engine, to determine the optimal supply of H_2.

3 Analysis of the main factors affecting the environmental performance of a dual-fuel hydrogen engine

Table 2 presents the parameters of the working process of a dual-fuel hydrogen engine D-245, operating at maximum power at a compression ratio of ε = 8 with an effective power of Ne = 63.8 kW at n = 2200 rpm, obtained in computational studies. The characteristics of its main environmental indicators are shown in Figure 1. These data are supplemented by a number of additional indicators that affect the environmental performance of the engine (see Table 2).

The most significant gaseous exhaust gases toxic component of diesel engines, regardless of their type, class, dimension and design, are nitrogen oxides NO_x. Their share in the total toxic emissions of gaseous harmful substances is 30–80% by weight and 60–95% by equivalent toxicity. Analysis of the Table 2, as well as the data of the works [6, 12, 19, 20, 21], shows that the temperatures in the combustion chamber makes a major impact of nitrogen oxides NO_x in diesel exhaust, in particular the maximum temperature T_{max} of the cycle. Figure 2 shows the dependence of the content of nitrogen oxides C_{NOx} on the maximum combustion temperature T_{max} for the investigated dual-fuel hydrogen engine according to seven points of Table 2. The relationship $C_{NOx} = f\,(T_{max})$ is well approximated by a linear dependence:

$$C_{NOx} = 0.958 \times T_{\max} - 617.2 \qquad (3)$$

The correlation coefficient between the content of C_{NOx} and the maximum cycle temperature T_{max} for the studied dual-fuel hydrogen engine was equal to $R= 0.9994$, which confirms the existence of a close correlation between the parameters C_{NOx} and T_{max}.

Table 2. The main indicators of the D-245 engine in the implementation of a gas-diesel cycle with an ignition DF and different H_2 supplies.

Parameters	№ of the energy share of hydrogen in the total fuel supply						
	№1	№2	№3	№4	№5	№6	№7
The energy share of DF in the total fuel supply E_{DF}, %	100	95	90	80	60	40	20
The energy share of H_2 in the total fuel supply E_{H2}, %	0	5	10	20	40	60	80
Lowest heat of combustion of DF, $H_{U\,DF}$, MJ/kg	42.5	42.5	42.5	42.5	42.5	42.5	42.5
Lowest heat of combustion of H_2, H_{UH2}, MJ/kg	120	120	120	120	120	120	120
Cyclic DF supply, m_{DF}, g	0.066	0.0628	0.0595	0.0529	0.0397	0.0264	0.0132
Cyclic H_2supply, m_{H2}, g	0	0.00117	0.00234	0.00468	0.0093	0.01405	0.0187
Total cyclic supply, m_Σ, g	0.066	0.0640	0.0618	0.0576	0.0491	0.0405	0.0319
Total cyclic fuel energy, E_{cycle}, kJ	2.809	2.809	2.809	2.809	2.809	2.809	2.809
The mass content of carbon in the DF cyclic supply, $m_c=0.87•m_{DF}$, g	0.057	0.0546	0.0518	0.0460	0.0345	0.0230	0.0115
Relative mass content of carbon atoms m_{Crel} in DF and H_2, $m_{Crel} = m_C/m_\Sigma$, fractions	0.870	0.853	0.838	0.799	0.703	0.568	0.361
Relative mass content of carbon atoms m_{Crel} in DF and H_2, $m_{Crel}=(mS/m\Sigma)•100$, %	8.70	85.3	83.8	79.9	70.3	56.8	36.1
Effective power N_e, kW	63.8	64.2	65.1	66.4	68.0	66.6	64.0
Total specific effective fuel consumption g_e, g/(kWh)	275.0	270.7	266.8	261.9	255.7	261.1	271.5
Effective performance factor	0.308	0.311	0.316	0.322	0.330	0.323	0.311

Excess air coefficient	2.020	2.027	2.024	2.017	2.010	2.009	2.012
Excess air coefficient for gas α_{H2} (excluding DF)	-	48.8	24.2	11.9	5.72	3.69	2.68
Maximum combustion pressure p_z, bar	40.3	41.0	42.0	44.8	51.7	56.6	61.1
Maximum combustion temperature T_{max}, K	1518	1535	1570	1618	1753	1868	1997
The average Sauter diameter of DF droplets d_{32}, microns	35.0	35.3	35.7	36.2	37.6	41.2	48.1
Charge detonation readiness indicator LW_{knock}	0.075	0.137	0.218	0.398	0.702	1.070	1.307
Specific mass emission of carbon dioxide e_{CO2}, g/(kWh)	888	832	777	678	496	338	176
Concentration of NOx in the exhaust gas C_{NOx}, ppm	832	860	891	932	1060	1170	1300
Relative concentration of NOx in the exhaust gas $\overline{C}_{\mathrm{NOx}}$	1.000	1.034	1.071	1.120	1.274	1.406	1.563
Smoke in the exhaust gas, % (by Hartridge)	15.0	13.5	12.5	10.5	7.0	4.5	3.0
Relative smokiness of exhaust gas $\overline{K}_{\mathrm{X}}$	1.000	0.900	0.833	0.700	0.467	0.300	0.200
Generalized criteria $\overline{K}_{\mathrm{X}} \cdot \overline{C}_{\mathrm{NOx}}$	1.000	0.931	0.892	0.784	0.595	0.422	0.313

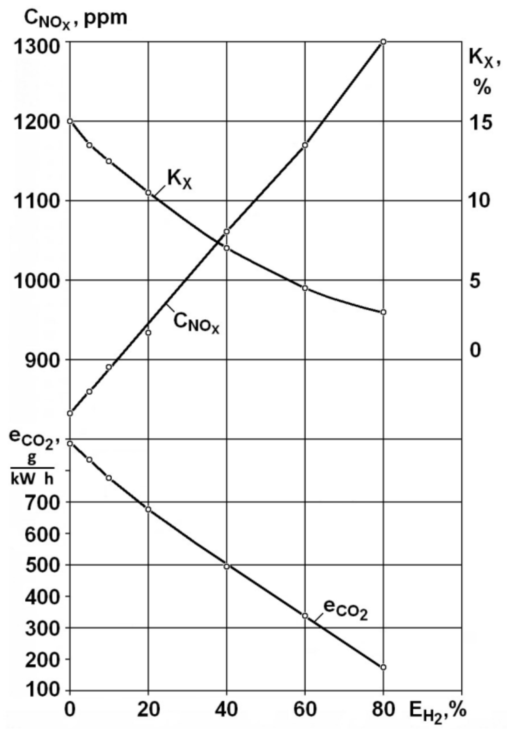

Fig. 1. The dependence of the content of nitrogen oxides C_{NOx}, the smoke content K_X of the exhaust, emissions of e_{co2} carbon dioxide from the exhaust gas on the energy share of H_2 in the fuel supply E_{H2}.

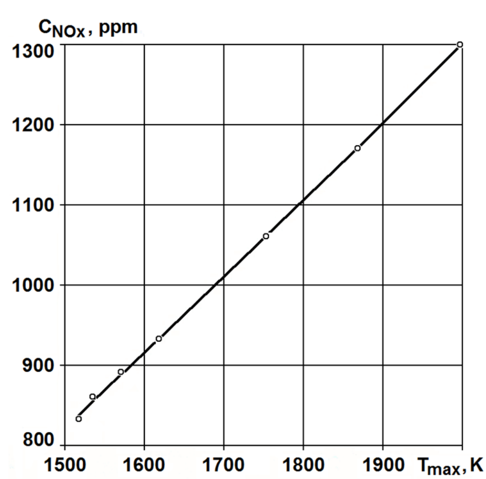

Fig. 2. Dependence of the content of nitrogen oxides CNOx in the exhaust at the maximum combustion temperature Tmax.

Another significant toxic component of diesel exhaust is solid particles. Their emission is closely related to the smoke content and the emission of soot. The main toxic properties of soot are caused not by carbon, but by carcinogenic polycyclic aromatic hydrocarbons settled on soot particles. The most toxic is benz(α)pyrene $C_{20}H_{12}$. In the domestic engine industry, the indicator "smokiness of exhaust gases" is more often used. Data analysis of Table 2, as well as other works [6, 12, 19, 20, 21], shows that the smokiness of the exhaust gas is largely determined by the relative mass content of carbon atoms in fuel molecules (DF and H_2) $m_{C\ rel}$. This parameter was defined as follows. The content of carbon atoms in the molecules of diesel fuel m_C is defined as: $m_C = 0.87 \bullet m_{DF}$, where the factor *0.87* characterizes the mass fraction of carbon atoms C in the molecules of diesel fuel. Further, this indicator m_C is divided by the total mass supply of DF and H_2 fuels m_Σ and expressed as a percentage: $m_{C\ rel} = (m_C / m_\Sigma) \bullet 100\%$.

Figure 3 shows the dependence of the exhaust gas smoke on the relative mass content of carbon atoms in fuel molecules (DF and H_2) m_{Crel} for the studied dual-fuel hydrogen engine by seven points from Table 2. The relationship of these parameters is well approximated by a third-order polynomial dependence:

$$K_X = -14.67 + 1.045 \times m_{C\,rel} - 0.02056 \times m_{C\,rel}^2 + 0.000143 \times m_{C\,rel}^3 \qquad (4)$$

The correlation coefficient between the smoke content of the exhaust gas K_X and the relative mass content of carbon atoms in the molecules of the fuels used (DF and H_2) m_{Crel} for the studied dual-fuel hydrogen engine was equal to $R = 0.9992$, which indicates the presence of a close correlation between the parameters of K_X and $m_{C\ rel}$.

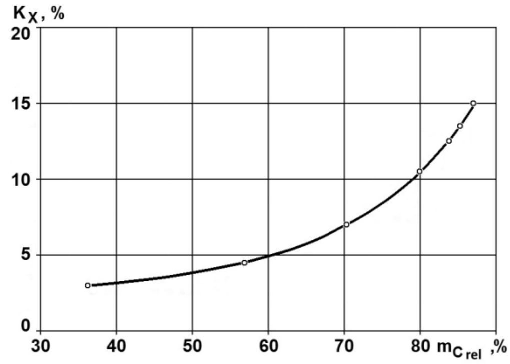

Fig. 3. Dependence of the smoke content of the exhaust gas on the relative mass content of carbon atoms m_{Crel} in diesel fuel and H_2.

4 Methodology of the exhaust gases total toxicity evaluation and optimal hydrogen supply determination

In modern regulatory documents limiting emissions of harmful substances of diesel exhaust, the normalized indicators of exhaust toxicity are unburned hydrocarbons CH_x, nitrogen oxides NO_x, solid particles, carbon monoxide CO [6, 7]. At the same time, the most significant among them are nitrogen oxides NO_x and solid particles. Two more normalized toxic components – CO and CH_x have significantly less toxicological impact. It should be noted that determining the concentration of solid particles in diesel exhaust gases is a complex and difficult technical task. It is much easier and more accessible to determine the smokiness of exhaust gas using inexpensive and common smoke meters. When a diesel engine operating with low excess air coefficients, soot makes up a large part of the total mass of solid particles (up to 95...98%). These two characteristics of exhaust toxicity – the emission of solid particles and the smoke content of exhaust gas are very closely related (with a high correlation coefficient). In this regard, it is proposed to characterize the total toxicity of the exhaust gas of the investigated dual-fuel engine by a generalized criterion in the form of the multiplication of the smoke content

K_X and the nitrogen oxides content C_{NOx}. Since these criteria have different dimensions, the specified product is proposed to be used in a relative form:

$$\overline{K}_X \cdot \overline{C}_{NOx} = (K_X \cdot C_{NOx})_i / (K_X \cdot C_{NOx})_{DF} \tag{5}$$

where $(K_X \cdot C_{NOx})_i$ is the generalized dimensional criterion for the i–th supply of hydrogen; $(K_X \cdot C_{NOx})_{DF}$ is the same criterion when the engine is running only on diesel fuel. The values of this generalized criteria characterizing the total toxicity of engine exhaust gases are presented in Table 2 and in Figure 4.

According to Figure 4, it should be noted that with an increase in the energy share of H_2 in the fuel supply of E_{H2}, the smoke content of the exhaust gas decreases, and the content of nitrogen oxides in the exhaust gas increases. At the same time, the accepted generalized criteria characterizing the total toxicity of exhaust gas monotonically decreases from *1* (when working on a purely diesel cycle) to *0.313* with the energy fraction of H_2 E_{H2} = 80%. In this range of changes in the E_{H2} index, the specific mass emission of the main greenhouse gas, carbon dioxide e_{CO2}, also monotonically decreases in the range from *888* to *176* g/(kWh). But when optimizing the supply of DF and H_2, it is necessary to remember the restrictions on the rigidity of fuel combustion, which increases with the increase in the proportion of H_2.

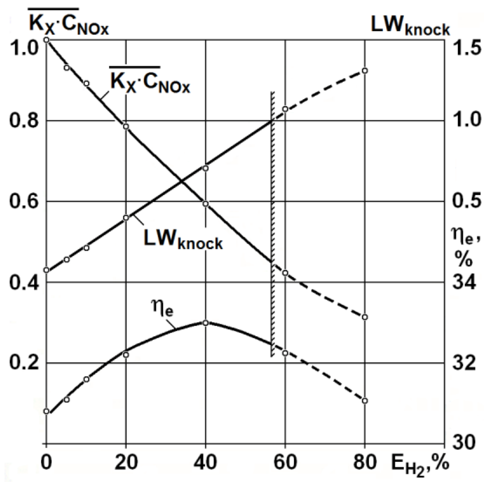

Fig. 4. The dependence of the generalized criteria characterizing the total toxicity of the exhaust gas, the indicator of the readiness of the charge for detonation LWknock and the effective performance of the engine ηe on the energy share of H2 in the fuel supply EH2 (the vertical shaded line shows the permissible maximum supply of hydrogen (from the point of view of the combustion process dynamics), dotted lines – the characteristics of the parameters $\overline{K}_X \cdot \overline{C}_{NOx}$, LWknock and ηe in the area of unacceptably large hydrogen supply).

As noted in this study, the indicator of the readiness of the charge for detonation LW_{knock} is used as an indicator characterizing the rigidity of fuel combustion. It describes the tendency of the working mixture in the cylinder to detonation. When the value of this indicator is $LW_{knock} < 1$, detonation does not occur (the combustion rigidity is acceptable). As follows from the data in Figure 4, such values of the LW_{knock} indicator are provided in the range of E_{H2} variation from *0* to *56%*. For such working mixtures, the optimum for the effective performance of the engine is achieved at $E_{H2} = 40\%$. This ratio of the supply of petroleum DF and H_2 is considered optimal. With such a supply of hydrogen, the effective performance η_e of the engine increased by *7.1%* (to $\eta_e = 33.0$), compared with operation in a purely diesel cycle ($\eta_e = 30.8\%$).

With the selected optimal energy fraction of E_{H2} equal to *40%* (from the point of view of fuel efficiency and indicators of the dynamics of the combustion process), the effective performance of the engine has a maximum value ($\eta_e = 33.0\%$), and the parameter characterizing the rigidity of combustion is equal to $LW_{knock} = 0.702$. The diesel engine transfer from purely diesel cycle to operation with an additive H_2 in the amount of *40%* is conduced by a decrease in the specific carbon dioxide emission e_{CO2} from *888* to *496* g / (kWh), i.e. by *44%*. The smoke content decreased from *15.0* to *7.0*% by the Hartridge scale, i.e. by *53%*. However, in this case, the content of C_{NOx} increases from *832* to *1060* pm, i.e. by *27%*.

5 Conclusion

The object of the study was a dual-fuel H_2 engine of the D - 245 type, in which H_2 was inflamed from the dose of DF. At the same time, H_2 was supplied to the engine intake system, and DF was injected into the engine cylinders by a conventional fuel system. To reduce the rigidity of the mixture combustion, the compression ratio of the engine was reduced from *15* to *8* units, the diesel fuel injection advance angle was reduced from *14*° to *5*° before TDC, the engine was deforced: the crankshaft speed reduced from 2400 to 2200 min^{-1}.

The working process computational study of the engine was carried out by calculation using the DIESEL-RK software package developed by Professor A.S. Kuleshov at the Bauman Moscow State Technical University for thermodynamic calculation and optimization of the internal combustion engines working processes. As a parameter characterizing the rigidity of the working mixture combustion, the readiness of the working mixture for detonation indicator LW_{knock} is selected, defined as $LW_{knock}= 2 - \varphi_{LWz}$, where φ_{LWz} is the relative duration of hydrogen burnout, *LW* is the *Livengood–Wu* integral. When *LW* equal to 1 or more, detonation combustion of the working mixture is noted.

The factors that have the greatest impact on the main factors of exhaust toxicity are determined. They are used for a detailed analysis of the environmental indicators of the investigated dual-fuel hydrogen engine. It is shown that the formation of nitrogen oxides NO_x in the engine is closely related to the combustion temperatures, in particular, with the maximum temperature of the cycle T_{max}. This relationship is well approximated by a linear dependency. The correlation coefficient between the parameters C_{NOx} and T_{max} for the investigated dual-fuel hydrogen engine is equal to $R = 0.9994$, which confirms the existence of a close correlation between these parameters.

The exhaust gases smokiness K_X depends on the relative mass content of carbon atoms in the fuel molecules (DF and H_2) m_{Crel}. This relationship is well approximated by a third-order polynomial dependence. The correlation coefficient between the parameters K_X and m_{Crel} for the dual-fuel hydrogen engine is equal to $R = 0.9992$, which indicates the presence of a close correlation between these parameters.

To determine the optimal combination of diesel fuel and hydrogen, a generalized criteria has been introduced that characterizes the total toxicity of engine exhaust gases. It is defined as the multiplication of the smoke content of exhaust gases K_X and the content of nitrogen oxides C_{NOx}. Since these criteria have different dimensions, the specified product was determined in a relative (dimensionless) form.

With the energy share of H_2 E_{H2}increase in the fuel supply, this generalized criteria monotonically decreases from *1* (when working on a purely diesel cycle – $E_{H2} = 0\%$) to *0.313* at $E_{H2} = 80\%$. In this range of E_{H2}index, the nitrogen oxides concentration C_{NOx} increased from *832* to *1300* ppm, the smoke content of the exhaust gas decreased from *15* to *3%*(Hartridge scale), the specific mass emission of carbon dioxide e_{CO2} decreased from *888* to *176* g/(kWh). But when optimizing the supply of DF and H_2, it is necessary to take into consideration the restrictions of the fuel combustion rigidity, which rises with the increase in the proportion of H_2.

Acceptable combustion rigidity of the mixture in the test is provided at $LW_{knock}<1$, which corresponds to a change in the energy fraction of hydrogen in the total fuel supply E_{H2} from *0* (purely diesel cycle) to *56%.* In this specified range of hydrogen supply, the optimal effective engine performance η_e is achieved at $E_{H2}= 40\%$. Such a supply of hydrogen is considered as optimal.

With such an energy fraction of hydrogen, the effective engine performance$\eta_e$is maximum and equal to *0.330*, which is *7.1%* more compared to diesel cycle. At the same time, the transfer of the engine from diesel fuel to work with an addition of *40%* hydrogen was followed by carbon dioxide e_{CO2} emission decrease from *888* to *496* g / (kWh) (*-44%*). At the same time, the exhaust smoke content K_X decreased from *15.0* to *7.0%* (Hartridge scale) (*-53%*), but the content of nitrogen oxides C_{NOx} raised from *832* to *1060* ppm, (+ *27%*).

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