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MATHEMATICAL MODEL OF THE VAPOR-DIESEL MIXTURE COMBUSTION PROCESS IN MOBILE VAPOR GENERATOR BOILERS UNITS

МАТЕМАТИЧНА МОДЕЛЬ ПРОЦЕСУ ЗГОРАННЯ ПАРО-ДИЗЕЛЬНОЇ СУМІШІ В КОТЛАХ ПЕРЕСУВНИХ ПАРОГЕНЕРАТОРНИХ УСТАНОВОК

The article is aimed at solving the problem of creating a mathematical model of combustion in boilers of mobile vapor generator units of the oil and gas industry of a vapor-diesel mixture, which is a cheaper and more environmentally friendly alternative to diesel fuel for mobile vapor generator units. The calculations showed that the thermal effect of burning a vapor-diesel mixture exceeds the effect of burning the same amount of diesel fuel. The conducted studies showed that the transfer of mobile steam generators to work using products of a vapor-diesel mixture was accompanied by a decrease in fuel consumption during the operation of boilers, and therefore is technically justified.
Keywords: *mobile vapor generator unit, vapor-diesel mixture, fuel combustion, amount of heat.*

Стаття спрямована на вирішення проблеми створення математичної моделі згорання в котлах пересувних парогенераторних установок нафтогазової галузі паро-дизельної суміші, які є більш дешевою та екологічною альтернативою дизельного палива для пересувних парогенераторних установок. Проведені теоретичні дослідження фізико-хімічних процесів при згоранні паро-дизельної суміші в котлах пересувних парогенераторних установок. Сформульовані припущення математичної моделі згорання в котлах пересувних парогенераторних установок паро-дизельної суміші. Створена математична модель згорання паро-дизельної суміші в котлах пересувних парогенераторних установок. Проведені розрахунки показали, що тепловий ефект від спалювання паро-дизельної суміші перевищує ефект від спалювання тієї ж кількості *дизельного палива. Використання паро-дизельних сумішей дозволяє знизити забруднення поверхонь нагріву в котлах сажею. Проведені розрахунки витрат дизельного палива парогенераторною установкою ППУА-1600/100 на різних режимах. Швидкість згоряння паро-дизельної суміші є більшою, ніж товарного дизельного палива, а індикаторний тиск при згорянні паро-дизельної суміші зростає. За умови обмеження максимального тиску згоряння, це призводить до зниження питомої витрати палива. Також треба відмітити, що збільшення швидкості згоряння паро-дизельної суміші призводить до зменшення тривалості горіння палива. Використання паро-дизельних сумішей дозволило підвищити приблизно на 10—15 % ККД котлів пересувних парогенераторних установок та зменшити шкідливі викиди продуктів згоряння. Виконані дослідження показали, що переведення котлів пересувних парогенераторних установок на роботу з використанням продуктів паро-дизельної суміші, супроводжувалося зниженням витрати палива при роботі котлів, а отже є технічно обґрунтованим.*

Ключові слова: *пересувна парогенераторна установка, пародизельна суміш, згоряння палива, кількість теплоти.*

Problem's Formulation

A significant particle of oil and gas technological transport uses technological installations with diesel power systems for attached equipment. This requires significant consumption of expensive diesel fuel during the operation of technological installations, therefore it is expedient to switch to cheaper types of alternative fuels for technological transport. The almost two-time increase in the cost of diesel fuel for 2022 again caused increased interest in the problem of switching existing diesel technological plants to alternative fuel mixtures, primarily those that are characterized by high fuel consumption during technological work.

One of the most energy-consuming process transport units in the oil and gas industry is mobile vapor generator units, which have powerful traction diesel engines to drive the chassis of process units and diesel vapor generator boilers to produce large volumes of vapor. Fuel consumption by vapor generator boilers of the most common installations is more than 100 kg of diesel fuel per hour

One of the common and cheap alternative fuel mixtures is vapor-fuel emulsions. The use of water- and vapor-fuel emulsions has been known for almost a hundred years and allows for an increase in the thermal efficiency of internal combustion engines and reduces harmful emissions of combustion products.

One of the significant problems of using vapor-fuel emulsions in internal combustion engines is the requirement to put a vapor generator on the chassis of the automobile which consumes additional energy and has significant mass-dimensional characteristics

The presence of standard vapor-generating boilers in the mobile technological installations of PPUA will allow, without placing the steam generator on the car chassis, a small part of the produced vapor to be sent to the power supply systems of the traction diesel engine and the vapor -generating boiler, thereby improving their economic and environmental characteristics. One of the steps in solving this problem is the creation of a mathematical model of the combustion of the vapor-diesel mixture in vapor-generator boilers of mobile technological units of PPUA.

Analysis of recent research and publications

Choosing a type of alternative fuel, it is necessary to take into account the conversion of the chemical energy of the fuel into work. The conversion of chemical energy of any type of fuel into work in internal combustion engines and boiler plants is carried out in two stages: in the first stage, it is transformed into heat, and in the second stage the heat is converted into work. During these transformations, the main losses of fuel energy occur, which can significantly increase the efficiency of the internal combustion engine [1].

In particular, applying vapor-fuel mixtures in internal combustion engines makes it possible to increase the efficiency of the use of chemical fuel energy. Work on the use of vapor-fuel mixtures began in the 20th of the last century and continues to this day in many countries of the world [2]. Huge experience has been accumulated in the use of steam-fuel mixtures in internal combustion engines for various purposes and boiler plants.

Studying the experience of foreign and domestic developments shows that diesel engines converted into vapor-diesel engines have high traction-dynamic and economic characteristics, and in terms of environmental safety, they are significantly superior to basic diesel engines [3].

The use of the vapor-diesel mixtures made it possible to increase the efficiency of internal combustion engines by approximately 5—10 % and reduce harmful emissions of combustion products, especially nitrogen oxides [4].

Operational tests of the vapor-diesel mixtures in diesel engines were carried out in the work [5]. The conducted tests showed that the use of vapor-diesel mixtures ensured fuel savings of about 4—5 % (depending on the engine operating mode) with a significant improvement in the environmental characteristics of combustion products and a reduction in soot formation. The wear and reliability of the main systems and parts of the diesel engine were at the same level as when it was running without water.

The water quantity in vapor-diesel mixtures usually does not exceed 20 %, which corresponds to the maximum value of fuel economy. Engine power begins to decrease at 30 % or more of vapor in the vapor-diesel mixtures [6].

Numerous studies have established that the optimal size of water particles in vapor-diesel mixtures is from 5 to 10 microns [7]. When the size increases, the stability of vapor-diesel mixtures decreases with a simultaneous decrease in efficiency. The mechanism of water's effect on fuel combustion has been studied in great detail. Optimally sized water droplets begin to boil before the fuel, causing "micro-explosions" and improving fuel atomization.

Overconsumption of fuel due to contamination of heating surfaces in boilers with soot and coke particles generally reaches 30—35 % [8]. The use of vapor-diesel mixtures allows for to reduction of the contamination of heating surfaces in boilers with soot. Another important factor characterizing the efficiency of using vapor-diesel mixtures is increasing the durability of combustion equipment [9]. In addition, water is a combustion catalyst, especially for carbon (soot), which also increases the completeness of fuel combustion. But starting and stopping diesel engines when using vapor-diesel mixtures should be carried out on fuel without water [10].

It was established [11] using vapor-diesel mixtures allows to increase the fuel combustion coefficient, saves fuel, and reduces harmful emissions of soot, NO, and CO into the atmosphere. The mechanism of this effect is explained as follows. Fuel, entering the burner, is sprayed by a nozzle. If in such a drop of diesel fuel, there are inclusions of smaller water droplets (with a dispersion of about 1 μm), then when heated, such droplets boil with the formation of water vapor.

As a result [12]: the contact surface of fuel with oxygen increases; in the high-temperature zone of the boiler, a water drop explodes and secondary fuel dispersion occurs; centers of turbulent pulsations appear; the number of small fuel droplets increases, which leads to the equalization of the temperature field of the furnace with a decrease in local maximum temperatures and an increase in the average temperature in the furnace; underburning of fuel is significantly reduced with a small coefficient of excess air, which allows reducing heat losses with exhaust gases.

Formulation of the study purpose

The purpose of the work is to create a mathematical model of the combustion processes in the boilers of mobile vapor-generating units of the oil and gas industry of a vapor-diesel mixture for calculating and improving their fuel-economic indicators.

To solve this goal, the following tasks are presented:

– to theoretically investigate the physical and chemical processes during combustion in boilers of mobile vapor generator installations of a vapor-diesel mixture;

– to formulate the assumption of a mathematical model of combustion in boilers of mobile vapor generator installations of a vapor-diesel mixture;

– to create a mathematical model of combustion in boilers of mobile vapor-generating units of a vapor-diesel mixture.

Presenting main material

Purpose and characteristics of industrial steam mobile units PPUA-1600/100M

The PPUA 1600/100M unit is designed for the deparaffinization of oil and gas wells, and underground, and above-ground equipment with saturated steam of high pressure up to 10 MPa. The unit is an autonomous boiler room for generating steam in field conditions. The appearance of the PPUA 1600/100M industrial vapor mobile plant and its steam generator boiler is shown in Fig. 1. Brief technical characteristics of the PPUA-1600/100M industrial vapor mobile plant are given in Tabl. 1 [13].

a)

b)

Fig. 1. The appearance of the industrial vapor mobile plant PPUA 1600/100M (a) and its vapor generator boiler (b)

Table 1. Brief technical characteristics of the industrial vapor mobile plant PPUA-1600/100M

Characteristic	**Indexes**	
	regime I	**regime II**
Heating environment	water	water
Vapor productivity, kg/h.	1600±10%	1600±10%
Heating environment	water	water
Vapor pressure, MPa	9,8	0,78
Vapor temperature, ^{0}C	310	175
Maximum hardness of feed water, μg-eq/kg	10	10
Fuel consumption for the boiler, kg/h.	110	35
Fuel used for operation of the installation	Diesel	Diesel
Fuel pressure, MPa	1,47	0,59
The time required to obtain steam from the moment the installation is started, min., no more	20	20
Unit management	from the car cabin	from the car cabin
The maximum speed of movement of the installation with full weight, km/h.	50	50

Physical and chemical processes during combustion of vapor-diesel mixture in boilers of mobile vapor-generating units

Issues related to the improvement of fuel economy and environmental indicators due to the use of vapor-diesel mixtures (VDS) for diesel engines of automobile transport have not been sufficiently investigated. In the reviewed studies, there are no general recommendations, inherent to all types of diesel engines, from the study of PDS with the aim of improving the economy of engines. Issues related to the mathematical modeling of combustion processes when using vapor-diesel mixtures for boilers of mobile steam generators have not been investigated yet.

In general, the reasons that explain the role of water (vapor) in the process of mixture formation and fuel combustion can be divided into two groups:

– chemical influence (acceleration of the kinetics of chemical reactions, gasification of soot residues);
– physical impact.

The special role of water (vapor) in the processes of mixture formation and combustion is explained as follows. The vapor-diesel mixture is a system consisting of two liquids with different boiling points. The boiling point of water at normal pressure is 100 ^oC. The boiling point of summer diesel fuel is 360 ^oC, winter diesel fuel is 320 ^oC, and arctic diesel fuel is 280 ^oC (not used in Ukraine). Emulsion drops of the "fuel with steam" type are a complex system consisting of diesel fuel, in which water vapor is uniformly distributed in the form of small particles. The difference between the temperature of the surface of the fuel particles and the temperature of the vapor is very significant and reaches 100...200 ^oC. Thanks to this, microparticles of water vapor are additionally heated, increasing the pressure and forming vapor bubbles. At the moment when the pressure, which tends to expand the water vapor droplet from the middle, will exceed the surface tension of the film, which has already weakened as a result of vapor heating, the destruction of the surface of the water vapor droplet will occur, that is, a "micro-explosion". As a result, there is intensive spraying of fuel droplets, their good mixing with air in the boiler or engine cylinder, and rapid evaporation.

Experimental studies by the authors [14] confirmed the existence of the "micro-explosion" phenomenon. Ignition of diesel fuel vapors is preceded by micro-explosions of water vapor particles. The combustion process of the vapor-diesel mixture proceeds violently and takes less time than the combustion of anhydrous diesel fuel.

More complete combustion of VDS is ensured due to the gasification of soot in fuel residues, which usually does not have time to burn when using ordinary diesel fuel. But in the presence of water vapor, carbon black interacts with the latter quite well according to the equation:

$$C + 2H_2O = CO_2 + 2H_2. \quad (1)$$

Free hydrogen reacts with oxygen much faster and more actively than carbon, thereby reducing the effective specific heat of combustion of the fuel. Since the soot burns out intensively, it is not deposited on the outer surfaces of the coil tubes. Clean outer tube surfaces improve heat transfer from combustion products to the coolant in boilers, as soot on the outer surfaces of coil tubes reduces the heat transfer coefficient. The existence of such reactions is clearly confirmed by the fact that when water vapor is injected into the intake manifold of internal combustion engines, soot and soot on the bottoms of the pistons, the cylinder head, and the exhaust manifold are usually not detected.

In general, the physical model of the mechanism of influence of PDS on combustion will have the following form. The presence of vapor in the fuel causes an increase in the volume of the cyclic supply of PDS to the boiler of the vapor generator compared to the supply of pure fuel. Vapor, as an inert body, causes an increase in the surface of the fuel charge, and the greater the vapor content in the mixture. The growth of the surface of the fuel charge causes an increase in the amount of evaporated fuel per fuel injection cycle. The b PDS's combustion speed is greater than commercial diesel fuel's droplets. Therefore, the combustion speed of PDS is greater than commercial diesel fuel, and the indicator pressure during PDS combustion increases, accordingly. If the maximum combustion pressure is limited, this leads to a decrease in specific fuel consumption. Therefore, the combustion of PDS in boilers of vapor generators improves efficiency. It should also be noted that an increase in the combustion rate of PDS leads to a decrease in the duration of fuel combustion.

At the same time, increasing the duration of supply due to diluting diesel fuel with vapor will cause a decrease in specific fuel consumption only until the time of reaching the duration of supply, which is equal to the optimum. Further dilution of diesel fuel with vapor will lead to an increase in specific fuel consumption.

Assumption of the mathematic model aims boilers of mobile vapor generator units of vapor-diesel mixture

The modeling aims to create a mathematical model of the combustion processes in the boilers of the mobile vapor generator units of the vapor-diesel mixture. To solve this problem, the following assumptions are made in the mathematical model:

– heat losses from the outer surface of the boiler of the steam generator are represented by convective heat exchange;

– air humidity is constant and does not affect heat loss from the surface of the boiler of the vapor generator to the environment;

– the change in the physical indicators of the steam-diesel mixture combustion process in the steam generator boiler is a consequence of the physical processes of atomization and evaporation of diesel fuel mixed with an inert body with the thermophysical characteristics of water vapor;

– the chemical participation of the water phase in the process of fuel combustion is the same as that of water vapor entering the steam generator boiler with atmospheric air and formed during fuel combustion;

– combustion of fuel vapors formed during the ignition delay period occurs at a constant volume V = const;

– heat from fuel combustion goes exclusively to increase the temperature of combustion products in the boiler.

Description of the mathematical model of combustion in boilers of mobile vapor generator installations of a vapor-diesel mixture

In the area of ignition, fuel burnout can be ignored with a sufficient degree of accuracy, and the rate of heat release (due to chemical reactions) can be considered close. In the area of rapid combustion up to the moment of P_{Zmax}, the rate of heat release can be calculated from the rate of fuel evaporation, depending on the amount of fuel evaporated during the ignition delay period, the corresponding change in the composition and properties of the working fluid.

The first law of thermodynamics in the area after the start of fuel injection based on 1 kg of fuel can be written as follows:

$$\sum dQ = MdU + udM + pdV, \tag{2}$$

where $\sum dQ$ — the sum of heat sources, which is defined as

$$\sum dQ = dQ_{зг} - dQ_{тв} - dQ_{вип}, \tag{3}$$

where $dQ_{зг}$ — heat release due to fuel combustion; $dQ_{вип}$ — heat loss due to fuel heating and evaporation; $dQ_{тв}$ – heat loss due to heat transfer to the walls of the boiler; M — the mass of the working body, which is equal to:

$$M = M_{пов} + M_{зг} + M_{\nu}, \tag{4}$$

where $M_{пов}$ — current number of moles of air:

$$M_{пов} = \propto L_0 + \gamma_{зг} L_0, \tag{5}$$

where $\gamma_{зг}$ — the particle of fuel burned, which is defined as

$$\gamma_{зг} = {}^{m_{зг}}/_{q_{ц}}, \tag{6}$$

where $m_{зг}$ — a mass of burned fuel; $q_{ц}$ — cyclic fuel supply; M_{ν} — current amount (number of moles) of fuel vapor;

$$M_{\nu} = \left({}^{1}/_{\mu_{\tau}}\right) \cdot (\chi_{\nu} - \chi_{зг}), \tag{7}$$

where $\chi_{\nu} = {}^{m_{\nu}}/_{\chi_{зг}}$ — a particle of fuel that evaporated; μ_{τ} — fuel molecular weight; $M_{зг}$ — current number of moles of combustion products;

$$M_{зг} = \alpha\gamma_{зал} L_0 + \chi_{зг} M_{\alpha}, \tag{8}$$

where $\gamma_{зал}$ — coefficient of residual gases; M_{α}— the number of moles of burned fuel at $\alpha = 1$.

Substituting equations (2.47)-(2.50) into (2.46), we get:

$$M = (\propto L_0 + \gamma_{зг} L_0) + (\alpha\gamma_{зал} L_0 + \chi_{зг} M_{\alpha}) + \left({}^{1}/_{\mu_{\tau}}\right) \cdot (\chi_{\nu} - \chi_{зг}), \tag{9}$$

The mass of the working body at the moment of the start of injection, given that at the moment of the start of injection $\chi_{\nu} = 0$, $\chi_{зг} = 0$, $\tau = 0$ will be equal to:

$$M = (\propto L_0) \cdot (1 + \gamma_{зал}). \tag{10}$$

Considering that at the moment of ignition of the injection mixture $\chi_{\nu} = 1$, $\chi_{зг} = 1$, $\tau = \tau_{зг}$ we will get:

$$M = (\propto +1) L_0 + (\alpha\gamma_{зал} L_0 + M_{\alpha}). \tag{11}$$

Let's transform the equations of the first law of thermodynamics. Because $PV = 8314MT$, therefore

$$pdV + Vdp = 8314(MdT + TdM). \tag{12}$$

Where do we get

$$\sum dQ = pdV + M\mu C_v dT + \mu C_v T dM. \quad (13)$$

Substituting in (2.56) $T = {PV}/{8313M}$ and $dT = \left(\frac{PdV+VdP}{8314} - TdM\right)/M$ we get:

$$\sum dQ = pdV + \mu C_v\left(\frac{PdV+VdP}{8314} - \frac{PVdM}{8314M}\right) + \mu C_v \frac{PV}{8314M} dM. \quad (14)$$

After the transformations, we get:

$$\frac{k-1}{dV}\sum dQ - \frac{dP}{P} - \frac{kdV}{V} + (1-\mu C_v)\frac{dM}{M} = 0. \quad (15)$$

We determine from (2.57) the dependences for the pressures and temperatures of the combustion products in the boiler of the gas generator:

$$\frac{dP}{d\varphi} + k\frac{PdV}{Vd\varphi} = \frac{k-1}{V}\sum\frac{dQ}{d\varphi}, \quad (16)$$

$$\frac{dT}{d\varphi} + (k-1)\frac{TdV}{Vd\varphi} = \frac{1}{\mu C_v M}\sum\frac{dQ}{d\varphi}. \quad (17)$$

The amount of heat needed to heat the coolant to the boiling point at atmospheric pressure:

$$Q_{пдг} = M \cdot C_v(t^{в}_{вх} - t^{в}_{вих}), \quad (18)$$

where $t^{в}_{вх}$ — water temperature at the inlet of the generator boiler coil, ^{0}C; $t^{в}_{вих}$ — water temperature at the outlet of the generator boiler coil, ^{0}C.

Coefficient of heat transfer to water:

$$\alpha_{вд} = 0{,}023 \cdot \frac{\lambda_{вд}}{d_{вн}} Re^{0,8} Pr^{0,4}, \quad (19)$$

where $\lambda_{вд}$ — coefficient of thermal conductivity of water, W/(m·K); $d_{вн}$ — inner diameter of the steam generator boiler coil tube, m; Re, Pr — Reynolds and Prandtl numbers.

Temperature pressure between the steam and the outer wall of the boiler:

$$\Delta t = t_{пар} - 0{,}5\,(t_{пар} - t_{зов}), \quad (20)$$

where $t_{пар}$ — outlet temperature of water vapor at the given mode of the steam generator unit, ^{0}C; $t_{зов}$ — the temperature of the outer wall of the boiler of the vapor generator installation, ^{0}C.

The total thermal resistance of tube walls, sediment layers, and water:

$$R = \frac{d_{зов}}{2 \cdot \lambda_{ст}} \cdot ln\frac{d_{зов}}{d_{вн}} + \frac{1}{\alpha_{вд}} \cdot \frac{d_{зов}}{d_{вн}}, \quad (21)$$

where $d_{зов}$ — outer diameter of the coil tube of the vapor generator boiler, m; where $\lambda_{ст}$ — coefficient of thermal conductivity of the tube wall, W/(m·K).

Average temperature pressure:

$$\Delta t_{сер} = \frac{t^{в}_{вих} - t^{в}_{вх}}{ln\frac{t_{пар} - t^{в}_{вх}}{t_{пар} - t^{в}_{вих}}}. \quad (22)$$

To obtain dry saturated vapor from water at a temperature $t_B < t_S$ under the condition of constant absolute pressure (dry saturated steam has a saturation temperature t_S) the specific amount of heat will be

$$q_{снп} = c_p \cdot (t_S - t_B) + r = c_p \cdot (t_S - t_B) + (i'' - i') = i'' - c_p \cdot t_B, \text{ kJ/(kg K)}, \quad (23)$$

where c_p — specific mass isobaric heat capacity of water, kJ/kg K); for further calculations, we accept $c_p = 4{,}19$ kJ/(kg K); t_B — the temperature of the feed water, which in practical conditions fluctuates within certain limits, ^{o}C; t_S — dry saturated vapor temperature, ^{o}C; r — specific heat of vaporization ($r = i'' - i'$), kJ/kg; i'' — enthalpy of dry saturated steam, kJ/kg; $i' = c_p \cdot t_S$ — enthalpy of boiling water, kJ/kg; $i_B = c_p \cdot t_B$ — enthalpy of feed water, kJ/kg.

To obtain 1 kg of wet saturated vapor with a degree of dryness x from the water having a temperature $t_B < t_S$, the consumption of specific heat will be

$$q_x = c_p \cdot (t_S - t_B) + x \cdot r = i' \cdot (1 - x) + x \cdot i'' - i_B, \text{ kJ/(kg K)}, \quad (24)$$

where x –a measure of the dryness of wet saturated steam (x varies from 0 to 1).

Substituting in formula (24) instead of value $c_p \cdot t_B$ we will get the amount of added specific heat to water to obtain wet saturated steam

$$q_x = i''x + i' \cdot (1 - x) - c_p \cdot t_B \text{, kJ/kg.} \qquad (25)$$

We determine the gross efficiency ratio of the steam generator installation knowing the fuel consumption to obtain dry saturated steam as needed *P*, *t* and *x*, as well as the recommended fuel consumption according to the technical characteristics of the units:

$$\eta_{бр} = \frac{B}{B_{mx}}, \qquad (26)$$

where B_{mx} — fuel consumption for the specified *P*, *t*, and *x*.

The specific consumption (part) of diesel fuel for obtaining 1 kg of steam at different values of the degree of dryness will be

$$q_{nn} = \frac{B_x}{D}, \frac{kg \ \ fuel}{kg \ \ vapor}, \qquad (27)$$

where B_x— actual consumption of diesel fuel to obtain steam at a certain degree of dryness *x*, kg; D — the amount of vapor obtained from spent fuel B_x, kg.

The analysis of the calculated data shows that when the installation is operating at the same mode of P and t, but with different values of the degree of dryness x, the fuel consumption for obtaining wet saturated vapor differs by an amount

$$\Delta B_x = B_{x=0,8} - B_{(x=0-0,6)} \text{, kg/h.,} \qquad (28)$$

where $B_{x=0,8}$ — consumption of diesel fuel to obtain vapor at a dry rate (*x*=0,8), kg/h.; $B_{(x=0-0,6)}$ — consumption of diesel fuel to obtain vapor for the values of the measure of dryness (*x=0; 0,2; 0,4; 0,6*), kg/h.

Calculation results

Let's make some practical calculations.

The amount of heat needed to heat the coolant to the boiling point at atmospheric pressure

We assume that the temperature of the water at the inlet of the generator boiler coil is equal to 30 ^{0}C, the water temperature at the outlet of the generator boiler coil is 100 ^{0}C, the mass of the coolant is 1600 kg. Then the heat load will be:

$$Q_{пдг} = 1600 \cdot 4190(100 - 30) = 469{,}28 \ MJ.$$

Coefficient of heat transfer to water

The inner diameter of the coil tube of the boiler of the generator set ППУА-1600/100 is equal to 3,5 mm. The coefficient of thermal conductivity of water at 0 ^{0}C is 0,551 Вт/(м·К), at 50 ^{0}C is 0,648 W/(m·K), at 100 ^{0}C is 0,683 Вт/(м·К). We will calculate the coefficient of thermal conductivity of water at 100 ^{0}C:

$$\alpha_{вд} = 0{,}023 \cdot \frac{\lambda_{вд}}{d_{тр}} Re^{0,8} Pr^{0,4},$$

The Reynolds number for water:

$$Re = \frac{\omega_{вд} d_{вн}}{\nu_{вд}} = \frac{2 \cdot 0{,}0035}{0{,}419 \cdot 10^{-6}} = 16707.$$

The Prandtl number for water:

$$Pr = \frac{\nu_{вд} C_V \rho_{вд}}{\lambda_{вд}} = \frac{0{,}419 \cdot 4191 \cdot 0{,}419 \cdot 10^{-6}}{0{,}663} = 2{,}5.$$

Then

$$\alpha_{вд} = 0{,}023 \cdot \frac{0{,}663}{0{,}0035} 16707^{0,8} 2{,}5^{0,4} = 15020{,}4 \ \frac{W}{m^2 \cdot K},$$

Temperature pressure between the steam and the outer wall of the boiler:

We take the output temperature of water vapor as the maximum for the PPUA-1600/100 M steam generator unit — 310 ^{0}C. The temperature of the outer wall of the boiler of the steam generator unit is 30 ^{0}C. Then $\Delta t = t_{пар} - 0{,}5 \left(t_{пар} - t_{зов}\right) = 310 - 0{,}5 \ (310 - 30) = 170$ ^{0}C.

The total thermal resistance of tube walls, sediment layers, and water:

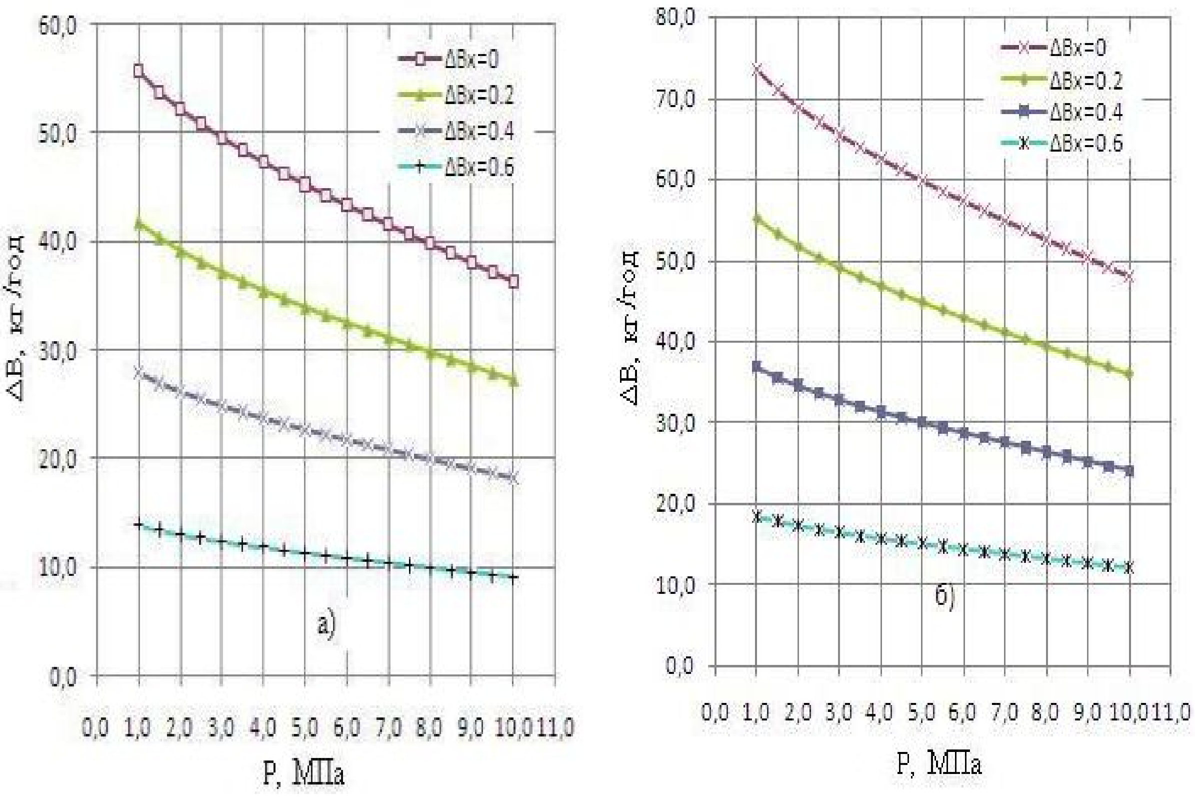
The outer diameter of the PPUA-1600/100 generating set boiler coil tube is 2.8 cm. The coefficient of thermal conductivity of alloy steel 1X2MB of the PPUA-1600/100 generating set boiler coil tube is 45.4 W/(m·K). The coefficient of thermal conductivity of alloy steel 1X2MB of the PPUA-1600/100 generating set boiler coil tube with a scale layer of 0.5 mm is 16.3 W/(m·K). Then

$$R = \frac{d_{зов}}{2 \cdot \lambda_{ст}} \cdot ln\frac{d_{зов}}{d_{вн}} + \frac{1}{\alpha_{вд}} \cdot \frac{d_{зов}}{d_{вн}} = \frac{0{,}028}{2 \cdot 16{,}3} \cdot ln\frac{0{,}028}{0{,}0035} + \frac{1}{15020{,}4} \cdot \frac{0{,}028}{0{,}0035} =$$
$$= 2{,}449 \cdot 10^{-4}\, м^2 \cdot {}^{K}/_{W}\,.$$

Average temperature pressure:

$$\Delta t_{сер} = \frac{100-30}{ln\frac{310-30}{310-100}} = 241{,}4\ {}^{0}C.$$

According to the technical characteristics of PPUA-1600/100 units, their fuel consumption is 110 kg/h. Accordingly, the gross efficiency ratio for installations is equal to $\eta''_{бр} = 0{,}824$. Using the above, we will perform calculations of fuel consumption by steam generator units to measure the dryness of the steam at *x = 0; x = 0,2; x = 0,4; x = 0,6 i x = 0,8*. Fuel consumption $\Delta B_x = f(P, x)$ with different measures of the degree of dryness are shown in fig. 2. Calculation data are summarized in the tabl. 2.



a) b)

Fig. 2. Graphs of PPUA fuel consumption at constant P, t and varying degree of dryness x (a — PPUA-1600/100 installation, regime I; b — PPUA-1600/100, regime II)

Table 2. Regimes, output data and calculations of diesel fuel consumption by the PPUA-1600/100 vapor generator

R, MPa	t, °C	i'', kJ/kg	i', kJ/kg	$B_{x=0}$	$B_{x=0,2}$	$B_{x=0,4}$	$B_{x=0,6}$	$B_{x=0,8}$
1,0	179,88	2777,0	762,6	32,496	50,875	69,254	87,633	106,012
1,5	198,28	2790,4	844,7	36,241	53,993	71,746	89,498	107,250
2,0	212,37	2797,4	908,6	39,156	56,389	73,623	90,856	108,089
2,5	223,94	2800,8	962,0	41,592	58,369	75,146	91,923	108,700
3,0	233,84	2801,9	1008,4	43,709	60,073	76,436	92,800	109,164
3,5	242,54	2801,3	1049,8	45,598	61,578	77,559	93,539	109,520
4,0	250,33	2799,4	1087,5	47,317	62,937	78,556	94,175	109,794
4,5	257,41	2796,5	1122,2	48,900	64,177	79,453	94,729	110,005
5,0	263,92	2792,8	1154,6	50,379	65,325	80,272	95,219	110,166
5,5	269,94	2788,4	1185,1	51,770	66,398	81,027	95,655	110,283
6,0	275,56	2783,3	1213,9	53,084	67,403	81,722	96,041	110,360
6,5	280,83	2777,6	1241,4	54,338	68,354	82,371	96,387	110,403
7,0	285,80	2771,4	1267,7	55,538	69,258	82,977	96,697	110,416
7,5	290,51	2764,7	1293,0	56,692	70,120	83,548	96,975	110,403
8,0	294,98	2757,5	1317,5	57,810	70,948	84,087	97,225	110,364
8,5	299,24	2749,9	1341,2	58,891	71,744	84,597	97,450	110,302
9,0	303,31	2741,8	1364,2	59,940	72,509	85,079	97,648	110,217
9,5	307,22	2733,4	1386,7	60,967	73,254	85,541	97,828	110,115
10,0	310,96	2724,4	1408,6	61,966	73,971	85,976	97,982	109,987

Calculations show that, for example, at measuring the dryness of water vapor x = 0.8; P=10 MPa; t=310 °C, the PPUA-1600/100 installation consumes 109.987 kg/h. of diesel fuel, and at dry vapor x = 0.2 (20 % dry vapor) with the same parameters, the consumption of diesel fuel is 73.971 kg/h.

Conclusions

Physical and chemical processes during the combustion of a vapor-diesel mixture in boilers of mobile vapor generators were theoretically investigated. The proposed mathematical model of combustion processes of vapor-diesel mixture in boilers of mobile vapor generator installations of the oil and gas industry.

In mathematical modeling of the working process of the combustion of a vapor-diesel mixture in boilers of mobile vapor generators, it is advisable to use a thermodynamic calculation based on the use of the first law of thermodynamics and theoretical dependencies of the heat balance.

Setting the combustion law as a function of the fuel supply characteristics and using a series of successive approximations for the heat input laws (heat input with the supply rate, heat input with the evaporation rate, heat input with the rate of formation of a stoichiometric vapor-diesel mixture, taking into account the limitation of the combustion process in the combustion zone) it is possible to determine the characteristic parameters of the process. This approach ensures the adequacy of calculations of indicator indicators and heat release.

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