Ministry of Education and Science of Ukraine
National University of Shipbuilding
Admiral Makarov
Mechanical Engineering Education and Research Institute

Department of Internal Combustion Engines, Installations and Technical Operation

"Admitted to the defence"
Head of the department
Oleksiy GOGORENKO
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« ___ » __________ 2024 р.

QUALIFICATION WORK

IMPROVING THE DESIGN OF A HIGH-PRESSURE FUEL PUMP FOR A MAIN MARINE ENGINE

Speciality 142 – Power engineering

To obtain the second (master's) level of higher education

Head of work Boris TYMOSHEVSKY

Education applicant Oleksandr KOVALCHUK

Mykolaiv 2024

National University of Shipbuilding
Admiral Makarov

Institute, faculty *Mechanical engineering educational and scientific*
Department *Internal combustion engines, installations and technical operation*
Degree *Master's degree*
Speciality *142 Power engineering*
(code and name)
Educational programme *Internal combustion engines*

APPROVED
Head of the Department *ICE, I and TO*
Oleksiy GOGORENKO
«____» ______________20___ year

TASKS
FOR A QUALIFYING JOB FOR AN EDUCATION APPLICANT

Alexander Vladimirovich Kovalchuk
(surname, first name, patronymic)

1. Work topic *Improving the design of a high-pressure fuel pump for a main marine engine*

2. Head of work *D., prof. B.G. Timoshevsky*,
(surname, name, patronymic, academic degree, academic title)

approved by the order of the higher education institution from "___"_____20___ year №___

3. Deadline for submitting a job application ______________________________

4. Initial data for the work *Engine power – 1230 kW; crankshaft speed – 750 min^{-1}. The engine is installed in the engine room of the tug The engine power is increased by upgrading the high-pressure fuel pump, which allows for higher injection pressure.*

5. Contents of the explanatory memorandum (list of questions to be developed)
1. Application of the object of study in a towed power plant. Determination of requirements for the use of internal combustion engines as part of the object of use. 2. Duty cycle calculation for rated diesel operation. 3. Design and engineering of a high-pressure fuel pump with a solenoid valve. 4. Economic justification for modernisation. 5. Analysis and development of labour and environmental protection measures for engine applications.

6. List of graphic material (with an exact indication of the required drawings)
1. General drawing of the tugboat. 2. Engine room. 3. Cross section of the 6CHN31,8/33 engine. 4. Longitudinal section of the 6CHN 31,8/33 engine. 5. The high-pressure pump. 6. The section of the high-pressure pump.

7. Work section consultants

Part	Name, initials and position of the of the consultant	Signature, date	
		The task was issued by	Task accepted
Economic part	*B.G. Timoshevsky*		
Labour's burial ground part	*B.G. Timoshevsky*		

8. Date of issue of the task ____

CALENDAR PLAN

№ s/n	Name of the work stages	Timeframe for completion of work stages	Note
1.	*Application of the object of study in a power plant. Determination of*		
	requirements for the use of an internal combustion engine as part of the		
	object of use		
2.	*Determination of the parameters of the internal combustion engine*		
	operating cycle.		
3.	*Design and development of a high-pressure fuel pump with an*		
	electromagnetic valve		
4.	*Economic justification of technical solutions*		
5.	*Analysis and development of labour and environmental protection*		
	measures for engine applications		

Education applicant ______________ *Oleksandr KOVALCHUK*
(signature) (surname and initials)

Head of work ______________ *Boris TYMOSHEVSKY*
(signature) (surname and initials)

АНОТАЦІЯ

Робота присвячена модернізації паливної системи головного суднового двигуна 6ЧН 31,8/33, а саме вдосконалення конструкції паливного насоса високого тиску. Розглянуто можливість застосування двигуна 6ЧН 31,8/33 на буксирі. Встановлено, що двигун може забезпечити необхідні експлуатаційні параметри судна, добре розміщується в машинному відділенні. Виконано розрахунок робочого циклу для номінального режиму роботи суднового двигуна 6ЧН 31,8/33. Проаналізовано можливих шляхів модернізації паливної системи головного суднового двигуна. Проведено конструювання паливного насосу високого тиску з електромагнітним клапаном. Виконано розрахунок основних геометричних та експлуатаційних параметрів насосу високого тиску, а також розрахунок на міцність його елементів. Встановлено, що застосування паливного насосу високого тиску з електромагнітним клапаном запропонованої конструкції забезпечило зниження питомої ефективної витрату палива на 6 г/(кВт·год), підвищення потужності дизеля на 352 кВт та покращення екологічних показників двигуна 6ЧН 31,5/33. Визначено, що річний економічний ефект від вдосконалення конструкції паливного насоса високого тиску для судновласника складає 30773,83 грн. Модернізація базового двигуна 6ЧН 31,8/33 окупиться за 0,65 року. Проведено аналіз небезпечних і шкідливих виробничих факторів, що впливають на персонал, якій експлуатує та обслуговує суднові ДВЗ. Розраховано рівень шуму й вібрації від двигуна 6ЧН 31,8/33. Проаналізовані та запропановані заходи щодо зниження рівня шуму й вібрації у машинному відділенні судна. Розглянуто характер впливу компонентів, які знаходяться у відпрацьованих газах двигунів на організм людини. Проведено аналіз можливих шляхів зниження токсичності відпрацьованих газів.

Магістерська робота виконана англійською мовою на 77 аркушах розрахунково-пояснювальної записки. Використано 14 джерел. Графічна частина представлена на 6 кресленнях формату А1.

Ключові слова: двигун внутрішнього згоряння, паливна система, паливний насос високого тиску, електромагнітний клапан, питома ефективна витрата палива.

ABSTRACTS

The paper is devoted to the modernisation of the fuel system of the main ship's engine 6CHN 31.8/33, namely, the improvement of the design of the high-pressure fuel pump. The possibility of using the 6CHN 31.8/33 engine on a tugboat is considered. It is established that the engine can provide the required operational parameters of the vessel and is well placed in the engine room. The duty cycle for the nominal operating mode of the 6CHN 31.8/33 marine engine was calculated. Possible ways to modernise the fuel system of the main ship's engine are analysed. The design of a high-pressure fuel pump with an electromagnetic valve was carried out. The main geometric and operational parameters of the high-pressure pump were calculated, as well as the strength of its elements. It was established that the use of a high-pressure fuel pump with an electromagnetic valve of the proposed design provided a decrease in specific effective fuel consumption by 6 g/(kWh), an increase in diesel power by 352 kW, and an improvement in the environmental performance of the 6CHN 31.5/33 engine.It is determined that the annual economic effect of improving the design of the high-pressure fuel pump for the shipowner is UAH 30773.83. Modernisation of the basic 6CHN 31.8/33 engine will pay off in 0.65 years. An analysis of hazardous and harmful production factors affecting personnel operating and maintaining shipboard internal combustion engines was carried out. The noise and vibration levels from the 6CHN 31.8/33 engine were calculated. Measures to reduce noise and vibration levels in the engine room of a ship are analysed and proposed. The nature of the impact of components in engine exhaust gases on the human body is considered. Possible ways to reduce the toxicity of exhaust gases are analysed.

The master's thesis is written in English on 77 pages of calculation and explanatory notes. 14 sources were used. The graphic part is presented in 6 drawings of A1 format.

Key words: internal combustion engine, fuel system, high-pressure fuel pump, solenoid valve, specific effective fuel consumption.

CONTENT

INTRODUCTION

Over the last ten years of the last century, diesel engines have made a leap forward in their development. The sometimes underpowered, noisy engine now fully competes with petrol engines, outperforming them in terms of efficiency. The use of newer open combustion chambers, variable valve timing and intake lengths are now considered promising. Intercooled supercharging is mandatory, and it is regulated not by simply bypassing gases, but by turning the compressor or turbine guide vanes, turning the turbocharger with an electric motor, etc. A special feature of modern diesel engines is the new generation of fuel systems.

The qualitative leap in the diesel industry is also driven by environmental standards. For example, in Europe, emissions of nitrogen oxides and particulate matter from diesel engines have decreased tenfold in 10 years.

In the fight for cleaner exhaust, designers have faced a big problem: most changes to the diesel engine's operating process reduce emissions from only one of the two components. For example, increasing injection advance reduces particulate emissions but increases NO_x emissions. Higher injection pressure and electronic control have solved this problem. Increased pressure improves fuel atomisation, leading to faster and more complete combustion. This explains why for almost 60 years (1927...1985) the maximum injection pressure was 20...50 MPa, and in the last 10 years it has increased to 200 MPa.

One way to increase the injection pressure is to use a high-pressure fuel pump with a solenoid valve. Unlike fuel injection pumps with fuel throttling in the discharge line, there is no fuel loss during injection, injection starts vigorously, there is a clear cut-off, and the control method does not reduce the injection pressure and allows for individual injection depending on the thermal and possibly technical condition of the cylinder.

Fuel systems with this type of fuel injection pump provide fast, precise control over a wide range of parameter changes, including injection characteristics.

PART 1
PECULIARITIES OF APPLICATION OF THE ENGINE 6CHN 31,8/33 IN THE SHIP'S POWER PLANT

1.1. General information. Purpose of the vessel

The multipurpose tugboat fig. 1.1 are designed to perform escort functions at speeds up to 10 knots, participate in rescue operations and firefighting, tow non-self-propelled objects, guide vessels in port waters and berth them. The main characteristics of the tug are shown in table 1.1.

Table 1.1 – Main characteristics of the tug

№ n.o.	Parameter	Values
1	The longest length, m	30,87
2	Waterline length, m	29,25
3	Width, m	11,6
4	Board height, m	5,45
5	Waterline draft, m	3,7
6	Overall draft, m	4,3
7	Displacement, t	680
8	Stroke speed, knots	14,1

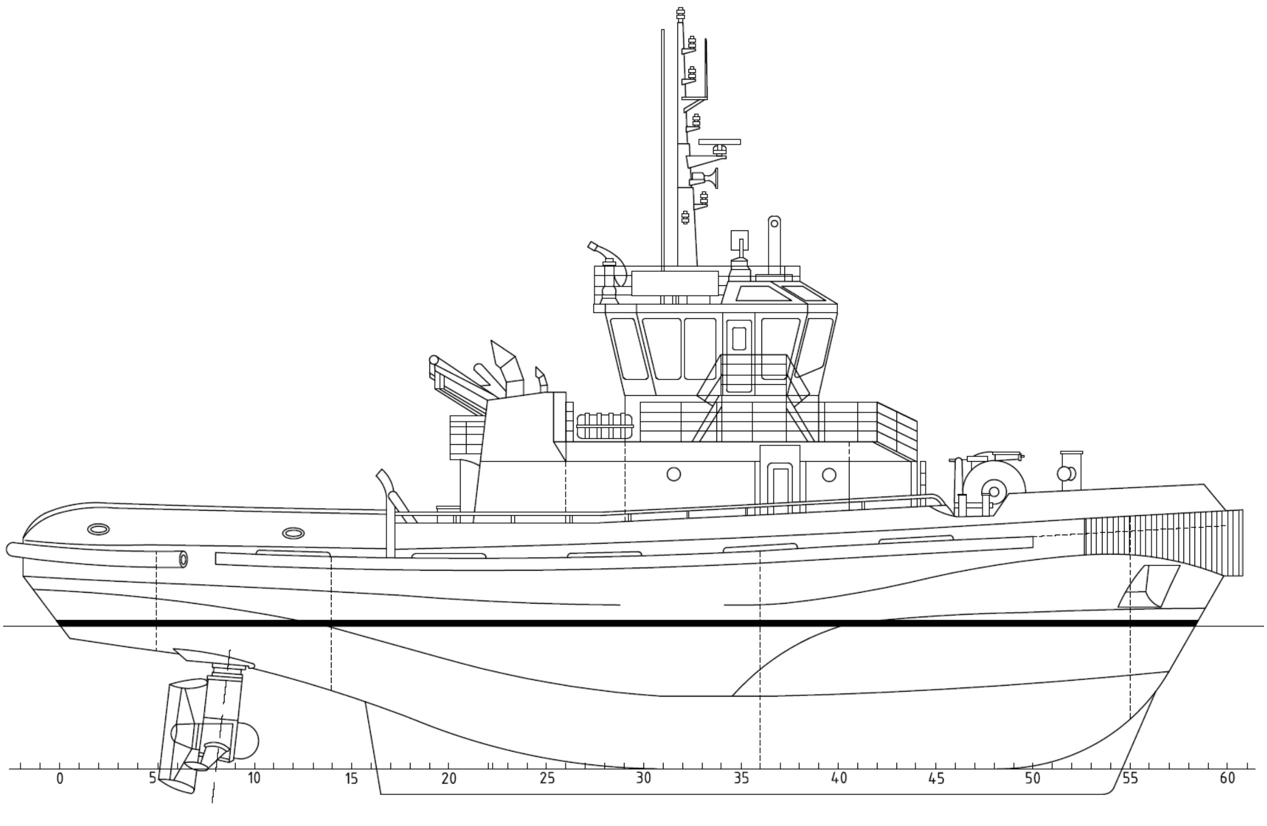

Figure 1.1 – General view of the tug

The vessel can carry 15 tonnes of DF cargo in the stern and up to 10 tonnes of cargo on the half-tank deck and maintain stability with up to 50 tonnes of ice on the decks, superstructures and rigging. Unsinkability is ensured by simultaneous flooding of any two adjacent compartments for all cases of the vessel's operational load.

The circulation diameter at full speed is about five times the length of the vessel's hull. The tug's good handling is ensured by a twin-shaft propulsion system and adjustable pitch propellers.

The vessel has two holds for storing portable equipment and materials required for ship assistance: a bow hold for 50 tonnes of cargo and a stern hold for 120 tonnes of cargo. The fuel tanks can carry 350 tonnes of main diesel fuel and another 80 tonnes by reducing the cargo in the holds to 90 tonnes. The fresh water reserve is 90 tonnes. The vessel has a desalination plant with a capacity of 4.5 tonnes of water per day. The tug has an elongated tank and an advanced superstructure that does not reach the sides.

The bow ends in a bulwark at the top. The vessel's stem is inclined with an icebreaker-type undercut in the lower part, the stern is cruising. The stem is forged in the lower part, above the platform it is made of sheet steel, the aftershield is cast.

The vessel is equipped with two Hall anchors of 1600 kg each and one segmental stern anchor of 1500 kg. The chain length of each anchor is 220 m. There is a special device for washing the anchor chains and sampling them in the keyhole tubes.

The tug is fitted with a streamlined semi-balance rudder with a steel forged ballast. The electro-hydraulic steering machine is controlled from the wheelhouse. It takes about 30 seconds to shift the steering wheel from side to side at full speed.

The vessel has a special steam pipeline system from the heating boiler to supply steam to four ice machines simultaneously. The steam pressure in the system is from 3.5 to 6.0 kg/cm^2 each. An area of up to 30 m^2 can be cleared in an hour with an ice thickness of up to 30 mm.

The vessel's power plant (fig. 1.2) consists of two 6-cylinder diesel engines 6CHN 31.8/33 connected by shaft lines with adjustable pitch propellers. The three-blade screws with a diameter of 2.5 m have removable blades and are made of stainless steel.

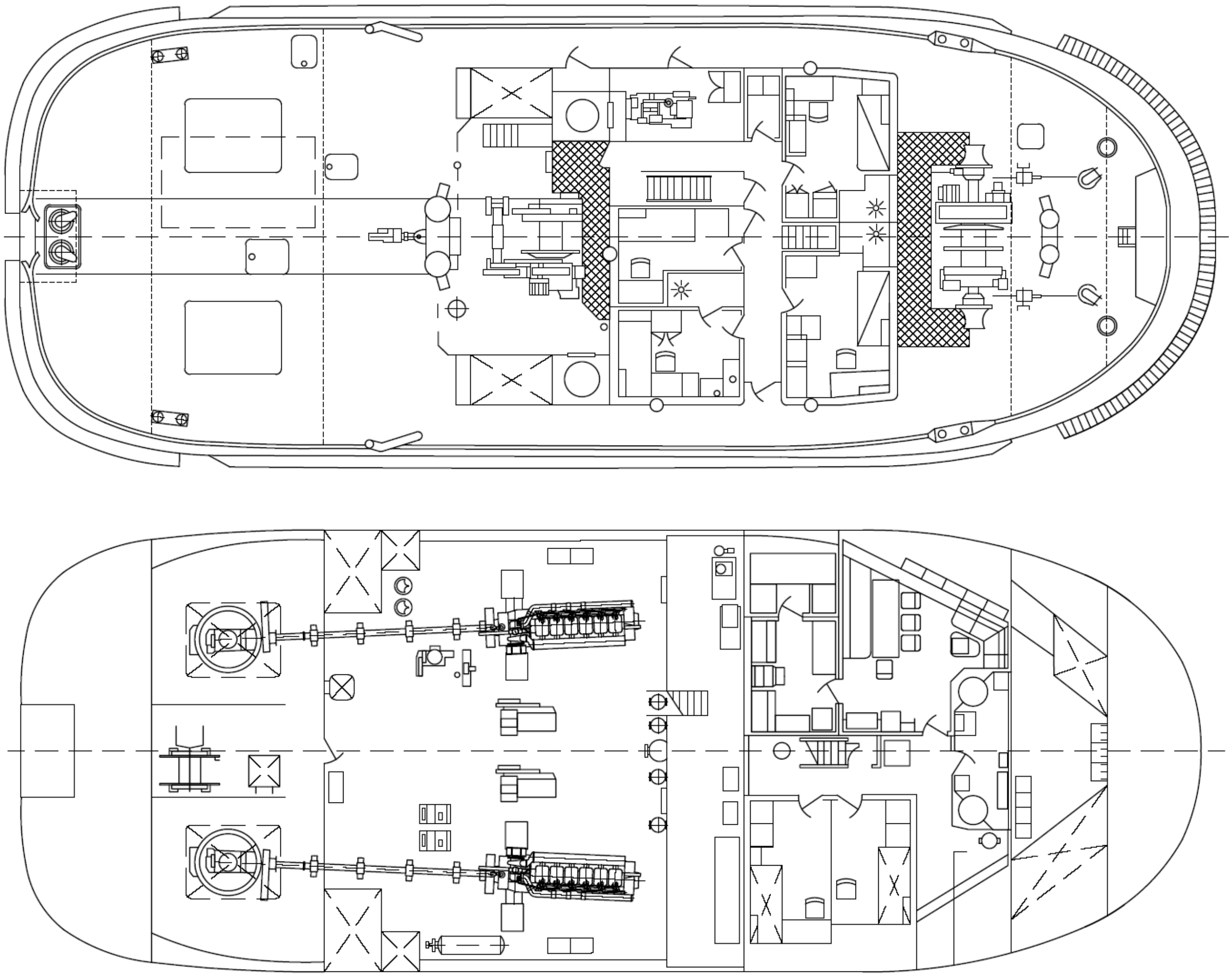

Figure 1.2 – Location of equipment in the engine room towing

Remote control of the power plant and the main control room is carried out from the main control room located in the wheelhouse. A second control panel located on the upper bridge is mechanically connected to the main control panel. The third remote control panel for the railway locomotive, installed in the main engines room, is operated only from the main control panel.

The main engines are started by air with a starting air pressure of 30 kg/cm^2.

The power sources on the vessel are 2 diesel alternating current generators with a voltage of 380 V. All generators are self-excited. The ship's 220 V voltage is supplied to the ship's consumers using three transformers. For heating, the vessel is equipped with an automatically controlled boiler with a capacity of 700 kg of steam per hour at a pressure of 6 atm.

The vessel is equipped with an automatic towing winch with an electro-hydraulic drive with a capacity of up to 25 tonnes. The automatic device ensures that the towing rope is stripped and taken up on the drum at a speed of up to 78.5 m/min. When the load exceeds 25 tonnes, the automatic device switches off. The stern of the vessel is equipped with a special bi-teng and a reinforced clew for jerking ships out of the shallows with a steel or nylon rope. The bi-teng and clew are designed for a working load of 200 tonnes. For towing with several tugs, there are two pairs of reinforced bollards with a pipe diameter of 500 mm in the area of the 103-106 frames on the half-tank.

The vessel has one light alloy lifeboat with a capacity of 35 people each. The boat is equipped with an 8 h.p. diesel engine and can reach a speed of up to 6 knots in calm water.

1.2. Compliance with the requirements of the Register. General requirements for power plants

The power of the main mechanisms of the technical fleet vessels ensures a vessel speed of at least 7 knots.

The vessel's mechanical installation ensures that it is possible to reverse for the required manoeuvrability under all normal operating conditions.

The vessel's mechanical installation ensures that at a constant free reverse speed the vessel can operate at least 70 per cent of the design speed of the forward gear within 30 minutes.

The main and auxiliary mechanisms are located in the machine rooms so that there are free passages to the exit ladders from the control stations and service areas. The width of the aisles along the entire length is at least 600 mm. The width of the gangways at the exit and the width of the hatches in the exits is at least 600 mm.

The arrangement of machinery, ducts, equipment, pipelines and valves ensures easy access to them for maintenance and emergency repairs. The main mechanisms, their gears, and thrust bearings of the shaft lines are fully or partially attached to the ship's foundations with tight-fitting bolts.

The engine room, shaft and pipeline tunnels have two exits each, which provide access to the decks to lifeboats and rafts.

Lighting and ventilation. In all rooms, places and spaces, lighting, which is important for ensuring safety of navigation, control of mechanisms and devices, evacuation of passengers and crew, the main lighting fixtures are installed.

Ventilation ducts are protected against corrosion. Shafts and vertical ventilation ducts pass through watertight decks within a single watertight compartment Ventilation ducts designed to remove explosive and flammable vapours and gases are gas-tight and do not connect to ducts in other rooms.

1.3. Design description of the engine 6CHN 31.8/33

The technical characteristics of the engine 6CHN 31.8/33, laid down by the manufacturer (fig. 1.3), are presented in table 1.2.

The frame of the summer diesel engine is cast iron. The frame connector plane is well above the crankshaft axis. In the transverse baffles, there are main bearing beds with bronze bushings filled with bronze bushings. The outermost main bearing on the flange side is designed as a thrust bearing. The stator of the main generator is attached to the end of the rear end of the frame. The frame sump forms an oil collector.

The cylinder block is cast iron and is attached to the frame with anchor and tie rods. The cylinder bushings are pressed into the block and washed with water. On the right side of the block there is a compartment in which the camshaft bearings are located in the transverse partitions. On the left side of the block are the fuel and water pumps and the fuel filter.

The cylinder cover is cast iron, individual for each cylinder. The cover contains a nozzle, two inlet and outlet valves, an indicator valve with a safety valve.

The cylinder sleeve is cast iron. The upper and lower parts of the sleeve have cylindrical belts that fix it in the block. The water seal of the liner is achieved by lapping its shoulder in the upper part and by three rubber rings in the lower part. At the upper end, the liner has an annular groove that fits the shoulder of the cylinder cover.

The piston is cast from aluminium alloy and is uncooled. The piston is fitted with seven piston rings, including four compression rings and three oil seals. The two upper compression rings are trapezoidal and chrome-plated. The piston pin is steel, cemented, hollow, floating type.

Table 1.2 – Main technical characteristics of the engine 6CHN 31.8/33

Parameters	**Values**
Diesel type	4-stroke direct-injection liquid-cooled engine with gas turbine supercharging and intercooled charge air
Number of cylinders	6
Cylinder arrangement	Ordinary
Cylinder diameter, mm	318
Piston stroke, mm	330
Cylinder operation	1-5-3-6-2-4
Pressure degree	14
Full power output, kW	882
Crankshaft speed corresponding to full power, min^{-1}	750
Specific effective fuel consumption at full power, g/kW·h	214
Basic fuel	DS, DW, DA SS 305-82
Lubrication	M-14 B_2, or M-14 $Г_2$ SS 12337-84
Specific oil consumption at full power, g/ kW·h a) per chad b) total (including draining when changing the oil)	 ≤1,30 ≤1,38
Coolant	Fresh water with additive OP-7 or VNII NP-117/D
Diesel lubrication system: Oil injection pump Oil filter	 gear full-flow fine cleaning
Diesel cooling system Water pump	 centrifugal

The connecting rod is stamped from alloy steel, the connecting rod rod is of I-section. A bronze bushing is pressed into the upper connecting rod head. The lower head has bronze bushings filled with bronze bellows. The lower head cover is attached to the connecting rod with four bolts.

The crankshaft is forged from carbon steel with hollow connecting rods. There is a flange for connection to the generator rotor, and a split gear is mounted on the same side of the shaft to drive the camshaft, fuel pump and water pump.

The diesel camshaft is made of steel and consists of three parts. Each part of the camshaft has support journals. For single-cylinder valves, two cams are placed between the support journals, which are made as one piece with the shaft. The camshaft support bearing liners are filled with bubbling. The camshaft is driven by a gear transmission from the crankshaft.

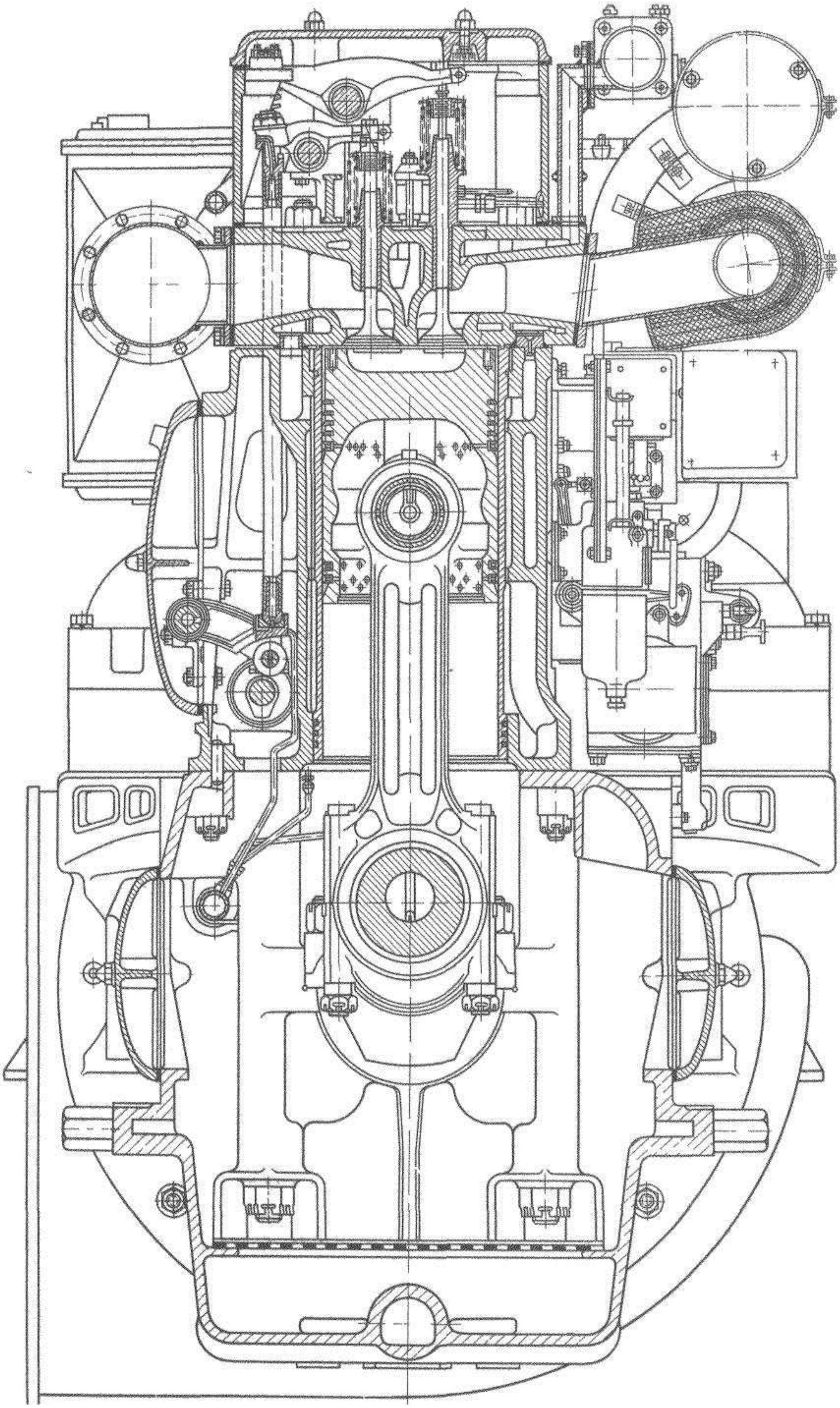

Figure 1.3 – Cross-section of the engine 6CHN 31.8/33

Indirect-acting regulator with hydraulic amplifier, centrifugal speed meter, isodromic feedback, autonomous, with its own oil circulation system, non-reversible, with manual and remote pulse control of the speed setting of the electric type and a stop device by impulse from the protection system. It is driven from the fuel pump shaft via a bevel gear.

In addition to the speed regulator, the diesel engine is equipped with a speed limit regulator that stops the diesel engine when the speed increases to 840...870 min^{-1}.

The diesel fuel system consists of a booster pump, coarse and fine filters, a high-pressure fuel pump, injectors and pipework. The coarse filter is a two-section filter with two sections each and is mounted outside the diesel engine before the fuel pump.

The fine filter is also a two-section filter with filter elements consisting of a set of plates. It is installed between the fuel injection pump and the high-pressure fuel pump. The high-pressure fuel pump consists of six separate sections mounted on a common cast-iron crankcase. The injectors are of the closed type with a hydraulically controlled needle.

The diesel lubrication system is circulating under pressure with a 'wet' crankcase, it includes: a gear oil pump driven by the crankshaft, an oil cooler installed outside the diesel generator; a control valve; an unloading check valve; two coarse filters; two fine filters connected to the system in parallel; a centrifuge installed on the diesel engine; an oil pressure switch.

The cooling system consists of two independent, closed circulating cooling circuits: the diesel cooling circuit and the charge air cooling circuit. The diesel cooling circuit consists of a water-water cooler, a centrifugal water pump, a water manifold, an expansion tank, pipework, valves and valves for filling the system. The charge air cooling circuit consists of a water-water cooler, a separately mounted centrifugal water pump with electric drive, an expansion tank, pipework, valves and taps.

The diesel engine is supercharged by a TK-30B turbocharger. It is cooled and lubricated as part of the diesel cooling and lubrication systems.

1.4. Conclusion to the part

The designed engine can be installed as the main engine on a multi-purpose tugboat. In total, the vessel is equipped with two 6CHN 31.8/33 engines with a required power of 1240 kW at a crankshaft speed of 750 rpm. The engines meet all the operational requirements and can be operated on the vessel. The general view of the vessel and its engine room is as shown in the A1 drawings.

PART 2
CALCULATION OF THE DUTY CYCLE FOR THE NOMINAL OPERATING MODE OF THE ENGINE 6CHN 31.8/33

2.1. Requirements for the designed engine

In accordance with the assignment of the department, a engine 6CHN 31.8/33 with a power of 1230 kW at 750 rpm is designed. The designed engine has parameters that differ from the prototype engine, so the parameters of its operating cycle are calculated and a high-pressure fuel pump is developed to provide new cycle parameters.

To ensure efficiency, attention must be paid to all parameters that affect fuel and oil consumption.

The reliability of the motor is ensured by a rational design and the absence of errors in the calculations, especially for strength. The engine operating conditions that may arise during operation must be taken into account. The permissible noise and vibration levels are ensured by the installation of noise mufflers and structural measures to reduce noise in engine components. The toxicity of engine exhaust gases is reduced by the correct choice of fuel combustion parameters, fuel advance angle, and air excess ratio.

The design should provide easy access to the main components of the engine, ensure the interchangeability of parts as much as possible, and ensure that the piston and connecting rod can be freely dismantled through the cylinder.

The engine should not be cluttered with pipework, and hinged parts should be avoided where possible, where they can be placed separately. Thus, with a combination of design solutions and careful refinement of the engine during testing and operation, it is possible to create a diesel engine that meets the highest modern requirements.

2.2. Main parameters of analogue engines

The engine is designed on the basis of the diesel engine 6CHN 31.8/33; the main parameters of this size and its closest analogues are shown in table 2.1.

Table 2.1 – Main parameters of analogues of the designed engine

Diesel name according to SS	Diesel cylinder power, kW	Crankshaft rotation speed, min^{-1}	Average effective pressure, MPa	Degree of pressure increase in the compressor	Specific fuel consumption, g/kW·h	Specific consumption engine weight, kg/kW
CHN18/22	47…93	750…1000	0,1…1,5	0,5…2,0	207…212	5,7…10,7
CHSN18/20	40…90	1000…1600	0,7…1,0	1,7…2,5	211…230	1,5…3,6
CHN 21/21	81…182	1000…1500	1,0…1,5	1,5…3,5	207…217	4,2…9,2
CHN16/17	58…88	1800…2000	0,85…1,15	1,5…2,7	211…225	1,5…2,5

2.3. Calculation of the engine duty cycle

The cycle calculation was performed using the Calculation-WC software. The Calculation-WC programme is designed to calculate and optimise two-stroke and four-stroke internal combustion engines.

The programme allows for thermal calculation, analysis and research of the following types of internal combustion engines:

– diesel;

– petrol spark engines: carburettor, petrol injection;

– gas spark ignition engines: conventional, fork-chamber.

Calculation-WC belongs to the class of thermodynamic programmes, which means that the engine cylinders are treated as open thermodynamic systems.

As a good thermodynamic programme should, Calculation-WC allows you to study engines with different supercharging systems, match supercharging units to the piston section, study gas exchange processes, including optimisation of the gas distribution phases, and predict various engine characteristics. The programme is constantly being improved, adding new features to solve current problems.

The program implements the LCD model: a model of mixture formation and combustion in diesel engines that allows calculating the heat release rate taking into account

– the shape of the combustion chamber;

– vortex intensity;

– the number, diameter and orientation of the nozzle holes;

– injection characteristics, including multiphase (multiple) injection and PCCI;

– interaction of jets with walls and with each other;

– biofuels and blends of biofuels with diesel fuel in various proportions;

– exhaust gas recirculation systems.

The LCD model allows you to optimise the shape of the combustion chamber and the design of the fuel equipment.

The built-in Fuel Spray Visualisation software allows you to visualise the dynamic interaction of fuel jets with the combustion chamber walls, the air vortex and each other. The software helps to design the shape of the combustion chamber and to select the correct diameter, number and direction of the spray nozzles for a specific fuel supply and air vortex intensity.

The Calculation-WC software interface allows you to explore any combustion chamber shape and any configuration of atomiser nozzles.

For solving optimisation problems, Calculation-WC has a built-in multi-parameter optimisation procedure that includes 14 optimal search methods, as well as 1D and 2D scanning procedures.

The optimisation tools can radically improve the efficiency of computational studies and identify ways to improve engines.

The diesel combustion model allows you to study a multi-fuel engine running on both diesel and biofuels, including their mixtures with diesel in various proportions.

The physical and chemical properties of the fuels can be specified by the user and stored in the software database. A different fuel can be specified for each engine mode.

Each operating mode can be defined and optimised with its own valve stroke diagram, including holding the valve at a given height for a given time.

The system of settings - special data file creation tools - allows you to easily and quickly create a data file for the calculation of any engine, which is especially useful for rapid analysis of internal combustion engines, as well as for the initial setting of empirical coefficients of mathematical models. The setup wizard will set the values of the empirical coefficients based on the extensive 25-year experience of using the programme for engines of various classes and applications.

The programme implements a modern method for calculating nitrogen oxide emissions based on the Zeldovich scheme for conventional engines, as well as on a detailed kinetic mechanism for advanced engines with a high degree of exhaust gas recirculation and multiple injection. The programme includes the calculation of several schemes of the exhaust gas recirculation system.

When calculating the cycle, the initial data were selected based on recommendations, taking into account the need to ensure the specified power at high efficiency and permissible loads on the mechanism.

Below is a copy of the results of calculating the operating cycle of the 6CHN 31.8/33 engine under different fuel supply characteristics using the Calculation-WC programme.

Performance and indicator values

Crankshaft speed n = 750 rpm

Power N_e = 1234.4 kW

Average effective pressure P_e = 12.66 bar

Torque M_e = 15845 Nm

Cycle fuel consumption q_c = 1,921 g

Specific fuel consumption g_e = 0.208 kg/(kW)

Effective efficiency η_e = 0.4

Average indicator pressure P_i = 13.78 bar

Indicator efficiency η_i = 0.44

Environmental parameters

Ambient pressure P_0 = 1 bar

Ambient temperature T_0 = 295 K

Static pressure behind the turbine P_{0t} = 1.0040 bar

Inhibited flow pressure behind the filter R_{0in} = 0.9700 bar

Supercharging and gas exchange

Pressure upstream of the exhaust manifold P_k = 2,084 bar

Temperature upstream of the exhaust manifold T_k = 309.4 K

Air flow rate through the ICE cylinders G_{air} = 2.45 kg/s

Efficiency of the supercharging unit η_t = 0.8

Average pressure before the turbine P_t = 1.61 bar

Average temperature before the turbine T_t = 755.8 K

Gas flow rate through the ICE cylinders G_{gas} = 2.5 kg/s

Excess air coefficient α_{sum} = 2.35

Average pressure of pumping strokes $P_{p.s}$ = 0.21 bar

Filling coefficient η_v = 0.97

Residual gas coefficient γ_r = 0.005

Purge coefficient F_i = 1.09

Inlet manifold

Average pressure in the intake manifold P_s = 2,077 bar

Average temperature in the intake manifold T_s = 323.83 K

Average wall temperature of the intake manifold T_{ws} = 363.83 K

Heat transfer coefficient in the intake manifold α_{ws} = 182.67 W/(m^2·K)

Heat transfer coefficient in the valve channel α_{wsc} = 287.28 W/(m^2·K)

Exhaust manifold

Average static pressure of the gas P_r = 1.59 bar

Average static gas temperature T_r = 752.97 K

Average gas velocity W_r = 89.292 m/s

Strouhal number Sh = 11.3

Average wall temperature of the exhaust manifold T_{wr} = 652.1 K

Heat transfer coefficient in the exhaust manifold α_{wr} = 195.36 W/(m^2·K)

Heat transfer coefficient in the valve channel α_{wcr} = 831.18 W/(m^2·K)

Combustion

Excess air coefficient α = 2.15

Maximum combustion pressure P_z = 95.52 bar

Maximum combustion temperature T_z = 1589.1 K

Maximum pressure angle F_{i_pz} = 8 degrees above UDC

Maximum temperature angle F_{i_tz} = 31 degrees above UDC

Maximum pressure rise rate dP/dF_i = 3.99 bar/grad

Injection

Maximum injection pressure $P_{inject\max}$ = 1425.9 bar

Average droplet diameter d_{32} = 19.375 μm

Injection advance θ_{ad} = 14 degrees to UDC

Duration of fuel injection F_{i_in} = 37.7 degrees

Ignition delay period F_{i_delay} = 3.89 degrees

The proportion of fuel that evaporated during the delay period *Sig* = 0.017

Duration of combustion F_{i_comb} = 103 degrees

Environmental indicators

Hartridge smoke emission Hartridge = 10.229

Smoke emission on the Bosch scale Bosch = 1.11

Absolute light absorption coefficient of gas according to ECE, K, m^{-1} = 0.25 1/m

Particulate matter emission PM = 0.286 g/(kW·h)

Carbon dioxide emission CO_2 = 671.87 g/(kW·h)

Wet NO_x concentration NOx, ppm = 946.55 1/million, (ppm)

NO_x emissions reduced to NO_2, NO_2 = 6.56 g/(kW·h)

Complex of total NO_x and PM emissions SE = 1.89

SO_2 emissions SO_2 = 0.0000 g/kW·h

Duty cycle parameters at typical points

Initial compression pressure P_a = 2.26 bar

Compression start temperature T_a = 354.44 K

End of compression pressure P_c = 66.07 bar

End of compression temperature T_c = 893.24 K

Pressure at the beginning of release P_b = 8.37 bar

Temperature at the beginning of the release T_b = 1097.3 K

Parameters that determine the working process

Compression ratio ε = 12.6

Number of nozzle nozzles $i_{nozzles}$ = 8

Nozzle nozzle diameter d_{nozzle} = 0.38 mm

Start of discharge $\varphi_{s.d}$ = 59 deg. to the LDC

End of exhaust $\varphi_{e.e}$ = 41 deg. to UDC

Inlet start $\varphi_{i.s}$ = 54 deg. to UDC

End of the inlet $\varphi_{i.e}$ = 31 deg. behind LDC

Based on the cycle calculation, an indicator diagram of the cycle is built (fig. 2.1).

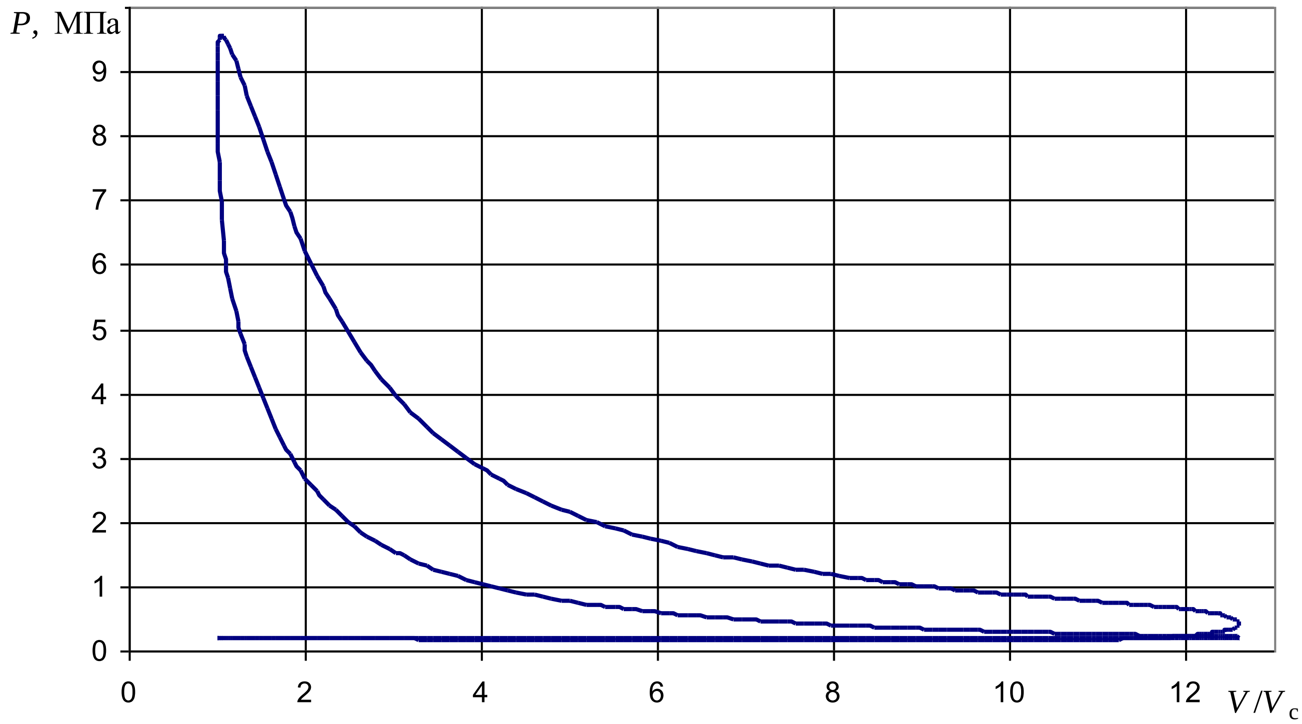

Figure 2.1 – Indicator diagram of the engine 6CHN 31.8/33

Conclusions from the cycle calculations

The designed engine is compared with the prototype engines and the closest analogues by the following parameters: N_{ez}, R_e, P_k, P_z, g_e (table 2.2).

Table 2.2 – Comparison of the designed engine with analogues

The name of the parameter	Units of measurement	Value	
		Projected	Analogues
Cylinder power N_e	kW	205,73	40…182
Crankshaft rotation frequency n	rpm	750	750…2000
Average effective pressure R_e	MPa	1,26	0,7…1,5
Degree of pressure increase in the compressor	–	2,2	1,5…3,5
Maximum combustion temperature T_z	K	1589	–
Maximum pressure in the cycle P_z	MPa	9,55	–
Specific effective fuel consumption g_e	g/(kW·h)	208	207…230

The table shows that the designed engine is higher than the average value for analogues in terms of cylinder power, at the minimum limit in terms of revolutions, in terms of average effective pressure and boost pressure it is within the average limits, and in terms of specific fuel consumption it is almost at the lower limit. The maximum cycle temperature of the engine does not exceed the recommended environmental tolerance limit, and the maximum combustion pressure is within the limits for engines with similar boost pressure.

Thus, the designed engine has a progressive set of key parameters that characterise its operating cycle and consumer qualities.

2.4. Conclusions to the part

As a result of the calculations, we obtained all the main parameters of the operating cycle that will be needed for further calculations, as well as calculated the engine dynamics, and determined the main loads in its mechanism. The data are summarised in table 2.3.

Table 2.3 – Main parameters of the designed engine

№ n.o.	Parameter	Value
1	Effective engine power N_e, kW	1234
2	Effective fuel consumption g_e, g/(kW·h)	0,208
3	Average effective pressure P_e, MPa	1,266
4	Effective engine efficiency η_e	0,4
5	Maximum combustion pressure P_z, MPa	9,55
6	Maximum combustion temperature T_z, K	1589
7	Supercharging pressure P_k, MPa	0,22
8	Air temperature behind the compressor T_c, K	309,4
9	Gas flow rate through the turbine G_t, kg/s	2,5
10	Air flow rate G, kg/s	2,45

PART 3

MODERNIZATION OF THE DESIGN OF THE HIGH-PRESSURE FUEL PUMP OF THE MARINE ENGINE 6CHN 31.8/33

3.1. General information

The criteria for the perfection of fuel supply are indicators of efficiency, power and noise, vehicle dynamics, start-up reliability, emissions of harmful substances from the gas; adaptability factor; compliance with restrictions on cylinder pressure, combustion rigidity, thermal loads, and gas temperature before the turbine.

It is necessary to design fuel equipment that allows:

– flexibly regulate the cycle flow in accordance with the engine speed; provide the required external engine speed characteristics (not necessarily rigidly specified);

– ensure minimal unevenness of the flow rate across cylinders or, conversely, optimal unevenness of the flow rate for each cylinder in accordance with its design, manufacture and current technical condition;

– optimal adjustment of the injection advance angle in accordance with the operating mode;

– automation of start-up, necessary enrichment at start-up, shutdown of fuel supply at forced idling, regulation in transient modes;

– shutdown of cylinders and cycles in partial modes.

Modern emission standards cannot be met only by adjusting or changing the design or operating parameters of the diesel engine, so compliance with them is a controversial task. Fuel injection pumps with increased injection pressure help to meet the emission standards. However, the injection duration should not be reduced as much as possible, since the minimum NO_x emissions with minimum fuel consumption are achieved at the optimal injection duration (fig. 3.1).

Another requirement for fuel injection is the use of flexible and, therefore, electronic fuel injection control. Additional requirements for the fuel supply system include the ability to manufacture the developed fuel pump using the existing equipment of the manufacturer and maximum ease of manufacture.

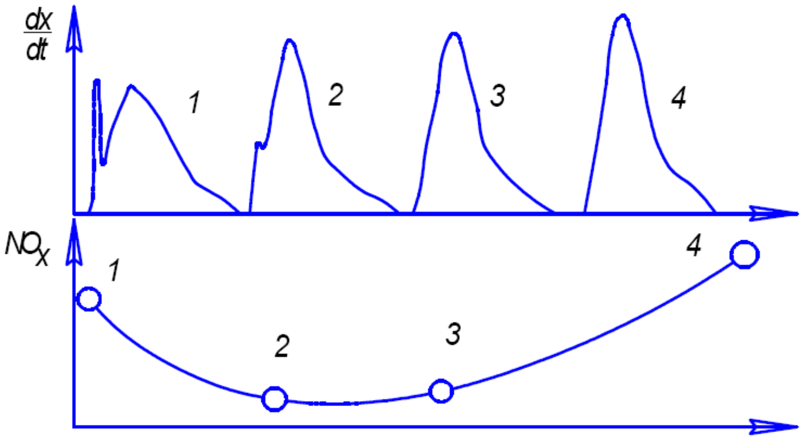

Figure 3.1 – Heat release law and nitrogen oxide emissions as a function of volumetric feed rate in a high-speed diesel engine

3.2. Possible variants of fuel systems

This device combines the fuel pump and injector systems into a single unit. Advantages of pump injectors:

– high injection pressure due to minimisation of the volume of compressed fuel;

– no injection;

– Reduction in the range of parts;

– sharp cut-off of the supply;

– less coking and longer service life of the nozzle;

– lower power consumption;

– reduction of the injection lag relative to the plunger discharge, which reduces the spread of injection advance angles by frequency and reduces the required adjustment range. The pump injectors provide a flatter front end of the injection characteristic and a sharper back end (fig. 3.2).

This helps to reduce combustion harshness, noise, NO_x emissions, large droplets at the end of the injection, and soot formation.

However, injector pumps have a number of disadvantages, the main ones being

– sharp complication of the head layout;

– increased diameter of the injector part;

– large reduction in injection pressure at partial operating modes;

– more complicated and less accurate conditions for regulating the uniformity of the flow over the cylinders.

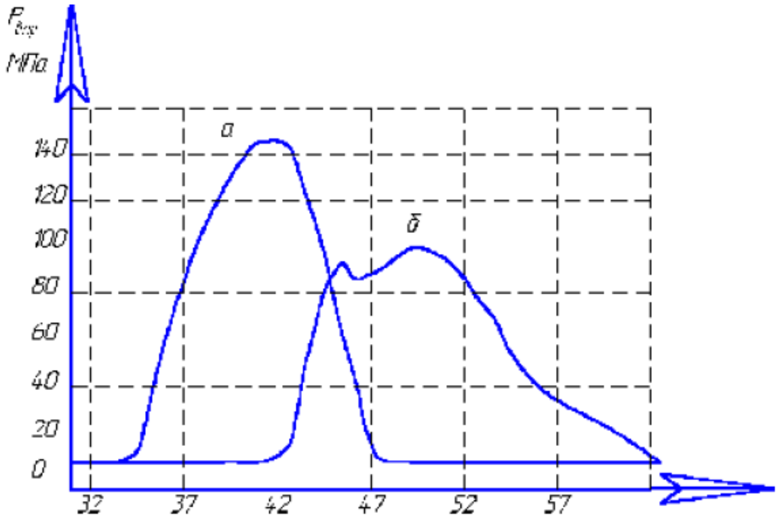

Figure 3.2 – Injection pressure for different fuel supply systems:

a – injector pump;

b – high pressure fuel pump with 580 mm pipeline and injector channel

Distributor pumps (fig. 3.3) became the most widely used in fuel systems of passenger diesel engines at the end of the last century. They are manufactured by BOSCH and are also produced under licence by DIESEL KIKI and IPPON DENSO. This type of fuel injection pump combines a low-pressure slide valve booster pump and a single high-pressure discharge element for all cylinders.

Figure 3.3 – VE type HP HPD by R. Bosch

The booster pump delivers fuel from the tank to the plunger area of the discharge pump. The plunger, performing reciprocating movements, creates a high fuel pressure, and returning, distributes it to the cylinders. the plunger is driven by its contact with the cam washer, which has protrusions according to the number of engine cylinders. the injection duration is changed by a special dosing ring located on the plunger and connected to the accelerator pedal through a system of levers.

In current versions of such pumps, the multi-lever control system for the metering ring has been replaced by an electronic fuel metering system.

The VR series pumps (fig. 3.4) are devices that fulfil all the requirements for a modern diesel fuel system. They create a significantly higher fuel injection pressure than the VE type pumps and ensure compliance with current emission standards.

Figure 3.4 – VR type high-pressure pump

Such fuel injection pumps are installed on modern high-speed engines. As in the previous case, the fuel is sucked from the tank by a slide gate pump. Its rotor is mounted on the common drive shaft of the high-pressure pump. The fuel then flows through channels in the pump body to one of the most important components of the entire device - the internal pressure control valve. Once it enters the high-pressure circuit of the pump, the fuel is compressed to an injection pressure of up to 1000 bar by means of radially arranged plungers.

Fuel dosing in this type of fuel pump is carried out by means of a solenoid valve, which opens at the signal of the control unit.

Fuel systems with battery injection typically include a high-pressure fuel pump (HPFP) that pumps fuel into the battery, a special distributor and an injector. the fundamental difference between these systems and direct injection is that the fuel enters the combustion chamber from the battery rather than directly from the HPFP.

High-pressure fuel pumps are designed to pump fuel into cylinders. The number of fuel pump sections does not depend on the number of engine cylinders and can vary. As a rule, the number of pump sections is 2...3 and rarely 4. The pump operation is usually not synchronised with the engine operation, since neither its speed nor the pump output nor the phases of its operation are determined by the engine operation. The push rods of these pumps are driven by eccentrics or shaft cams that are kinematically connected to the crankshaft. The pumps can be adjustable or non-adjustable. In the latter case, pressure reducing valves are installed on the accumulator to act as pressure regulators.

Cylinders are used to accumulate the fuel pumped by the system pumps. The pressure in the cylinders must be sufficient to ensure high-quality atomisation of the fuel supplied by the injectors to the combustion chamber. Cylinders are thick-walled containers equipped with the necessary fittings. They not only collect fuel, but also dampen fuel fluctuations that occur during system operation and smooth out the pulsations of fuel pumps.

Accumulator fuel systems have a number of advantages over conventional fuel injection systems, including the following:

1. The injection performance of battery fuel systems is highly dependent on the engine speed, which ensures high-quality cutting and stable engine operation at low speeds. This is their main advantage over other systems.

2. They provide high injection pressures and make it easier to deal with unwanted re-injection.

3. The presence of significant fuel pressures in the accumulator makes it easier to start the engine.

4. High-pressure fuel pumps are simpler than direct injection systems. When using high-capacity batteries, the number of revolutions of the pump shaft and the number of plunger elements can be greater than the number of revolutions of the crankshaft and the number of engine cylinders, which allows for large cycle flow rates with relatively small plunger pairs, and, therefore, without the use of bulky drives.

5. The number of working plungers of these pumps is not related to the number of cylinders and can be reduced to even one. The fuel supply by the plungers is not phase-locked to the engine's operating process, so it is not regulated.

These fuel injection pumps do not have a conventional rail or plunger rotation mechanism. The plunger does not have a spool part. In such a fuel supply system, a conventional injector is used, and maintenance and adjustment methods are retained.

In contrast to fuel injection pumps with fuel throttling in the discharge line, there is no fuel loss during injection, injection starts vigorously, there is a clear cut-off, and the adjustment method does not reduce the injection pressure and allows for individual delivery depending on the thermal and possibly the technical condition of the cylinder.

Fuel systems with this type of fuel injection pump provide fast, precise control over a wide range of parameter changes, including injection characteristics.

3.3. Selection of the schematic diagram of the fuel equipment

Since one of the most important requirements for the developed fuel equipment is to ensure the modernisation of the standard high-pressure fuel pump and the entire diesel fuel system, as well as the possibility of manufacturing the developed pump on the existing equipment of the manufacturer's plant and the maximum production rate. We take as a basis type of fuel system – fuel injection pump with a drain control valve. The use of this injection moulding machine is possible without replacing the existing injector and high-pressure line pipework, and, therefore, without changing the design of the usual cylinder head and layout solutions. It is also possible to build an electrically controlled drain valve on the basis of an existing in-line fuel injection valve of the engine 6CHN 31.8/33.

3.4. Calculation and design of the fuel pump

In accordance with PART 2 of the thesis project, the initial data for calculating the geometric dimensions and operating parameters of the fuel pump were obtained from the calculation of the operating cycle of the engine 6CHN 31.8/33 using the Calculation-WC.

Power of the engine 6CHN 31.8/33 N_e = 1234 kW

Specific effective fuel consumption

of the engine 6CHN 31.8/33 g_e = 190 g/(kWh)

Part of the main cycle supply $V_{c.main}$ = 0.8

Rotation frequency n = 750 rpm

Number of cylinders of the engine 6CHN 31.8/33 i = 6

Stroke ratio z = 0,5

Maximum combustion pressure P_z = 95,53 MPa

Fuel supply duration $\varphi_{s.d}$ = 38 deg. of c. r.

Cylinder pressure at point *c* P_c = 68,07 MPa

Cyclic fuel supply

$$V\text{ц} = \frac{Ne\text{ц} \cdot g_e \cdot K \cdot V_{\text{ц.осн}}}{60 \cdot n \cdot z \cdot \gamma_m} = 2{,}184 \text{ cm}^3\text{/cycle},$$

where $\gamma_т$ = 875 kg/m^3 – is the density of fuel; K = 1,11 – is the reserve factor.

Volume describing the plunger of a high-pressure fuel pump during the stroke from VDC to LDC

$$V_П = K_2 \cdot V_ц = \frac{\pi \cdot d_П^2}{4} \cdot S_П = 8{,}816 \text{ cm}^3,$$

where K_2 = 4...8 – is the coefficient of volume reserve, which describes the plunger of the high-pressure fuel pump of the engine 6CHN 31.8/33, over the maximum cycle flow rate by the sum of the volumes: compression in the supraplunger space and in the high-pressure line, deformation of the fuel line, and the volume describing the plunger of the high-pressure fuel pump of the engine 6CHN 31.8/33 when the filling holes are blocked, when the plunger speed is accelerated and decelerated.

We accept the value of the coefficient K_2 = 4.

According to the standards $\frac{S_П}{d_П} = 1 \ldots 1{,}5$.

Assume the value of $S_п = 1{,}5 \cdot d_П$.

The diameter of the plunger of the high-pressure fuel pump of the engine 6CHN 31.8/33 is

$$d_П = \sqrt[3]{\frac{4 \cdot V_П}{1{,}3 \cdot \pi}} = 21 \text{ mm.}$$

The stroke of the plunger will then be

$$S_п = 31 \text{ mm.}$$

The area of the plunger of the engine 6CHN 31.8/33 is

$$F_{пл} = \frac{\pi \cdot d^2}{4} = 330{,}387 \text{ mm}^2.$$

Profiling the straight stroke profile

Straight stroke profiling is performed in two stages:

Stage 1 - determine the maximum possible plunger speed per forward stroke C_{max}, the value of which determines the speed of the plunger during injection, and therefore the injection intensity; 2nd stage - determine the current value of the stroke S, speed C, acceleration W of the plunger and the radius of curvature of the profile R.

Stage – 1 determination of C_{max}

$$S_{АГ} = \frac{Q_T}{\eta \cdot F_П} = 9{,}529 \text{ mm,}$$

where η = 0.7 – feed rate of the plunger

Duration of the active geometric stroke of the plunger

$$\beta_{АГ} = (0{,}6 \div 0{,}8) \cdot \beta_{ВП} = 14{,}3 \text{ degree.}$$

Average speed of the plunger

$$C_m = \frac{S_{АГ} \cdot 6 \cdot n_K}{\beta_{АГ}} = 1{,}505 \text{ m/s,}$$

$$C_{max} = 1{,}3 \cdot C_m = 1.956 \text{ m/s.}$$

Kinematic coefficient at the starting point of the profile

$$X_н = R_0 + c = 85 \text{ mm.}$$

Radius of the pusher roller

$$\omega_K = \frac{\pi \cdot n_K}{30} = 39{,}25 \text{ s}^{-1}.$$

Determination of ram acceleration at the first section of the profile

$$W_1 = X_H \cdot \omega_K^2 - \frac{X_H^2 \cdot \omega_K^2}{R_H + \rho} = 161{,}03 \text{ m/s}^2,$$

Plunger stroke on the first section of the profile

$$S_1 = \frac{C_{MAX}^2}{2 \cdot W_1} = 0{,}0119 \text{ m}.$$

Plunger travel on the second section of the profile

$$S_2 = S_п – S_1 = 0.018893 \text{ m}.$$

Determination of ram acceleration in the second section of the profile

$$W_2 = -\frac{C_{MAX}^2}{2 \cdot S_2} = -101{,}26 \text{ m/s}^2.$$

Kinematic coefficient at the end of the first section of the profile

$$X_с = R_0 + с + S_1 = 0.0969 \text{ m}.$$

Maximum value of the pressure angle

$$\delta_{MAX} = arctg\left(\frac{C_{MAX}}{X_C \cdot \omega_K}\right) = 27{,}222 \text{ degree}.$$

Determination of the coefficient of excess of the force of the plunger spring over the force of inertia of the reciprocating parts of the plunger drive

$$K = \frac{(f_0 + S_1) \cdot K_Ж}{m \cdot |W_2|} = 6{,}0389$$

Kinematic coefficient at the end of the profile

$$X_К = R_0 + с + S_п = 0.1158 \text{m}.$$

Radius of curvature at the end point of the profile

$$R_K = \frac{X_K^3 \cdot \omega_K^2}{X_K^2 \cdot \omega_K^2 - W_2 \cdot X_K} - \rho = 0{,}039 \text{ m}.$$

Force from fuel pressure at the plunger position at the VMT

$$P_Т = P_{ЛО} \cdot F_П = 7E\text{-}05 \text{ MH}.$$

Spring force at the plunger position at the timing mark

$$Р_п = (f_0 + S_П) \cdot K_Ж = 0{,}0019 \text{ MH}.$$

Force transmitted by the roller to the cam

$$N = P_Т + P_П = 0.001953 \text{ MH}.$$

Determination of the maximum permissible radius of curvature at the end point of the profile by the Hertz formula

$$R_{КД} = \frac{\frac{N \cdot E}{b}}{\left(\frac{\sigma_Д}{0,418}\right)^2 - \frac{N \cdot E}{b \cdot \rho}} = 0,0005 \text{ m.}$$

We calculate the maximum permissible fuel pressure in the superplunger volume at the beginning of the second section, while neglecting the spring force and the force of inertia, which are directed towards each other and are close in magnitude.

The angles of the first and second sections of the forward stroke profile

$$\beta_1 = \frac{12 \cdot n_K \cdot S_1}{C_{MAX}} = 27,33 \text{ degree,}$$

$$\beta_2 = \frac{12 \cdot n_K \cdot S_2}{C_{MAX}} = 43,46 \text{ degree.}$$

Straight stroke profile angle

$$\beta_П = \beta_1 + \beta_2 = 70,79 \text{ degree.}$$

Cam projection angle

$$\beta_В = 2 \cdot \beta_П = 141,59 \text{ degree.}$$

Radius of curvature

$$R_{2Н} = \frac{\left(C_{MAX}^2 + X_C^2 \cdot \omega_K^2\right)^{1.5}}{\left(X_C^2 \cdot \omega_K^2 + 2 \cdot C_{MAX}^2 - W_2 \cdot X_C\right) \cdot \omega_K} - \rho = 0,03241 \text{ mm,}$$

$$P_{ТН} = \frac{\left(\frac{\sigma_Д}{0,418}\right)^2 \cdot b \cdot \cos \delta_{MAX}}{\left(\frac{1}{\rho} + \frac{1}{R_{2Н}}\right) \cdot E \cdot F_П} = 152,72 \text{ MPa}$$

Stage 2 – determining the current values of S, C, R, β, P_T

Profiling of the first section of the straight stroke profile:

Coefficient

$$K_3 = \frac{W_1 \cdot 10^3}{72 \cdot n_K^2} = 2E - 05$$

Current value of the plunger stroke

$$S = K_3 \cdot \beta_2$$

Coefficient

$$K_4 = \frac{C_{MAX}}{\beta_1} = 0{,}0716$$

Current value of the plunger speed

$$C = K_4 \cdot \beta$$

Kinematic coefficient

$$X = R_0 + c + S$$

Current value of the radius of curvature at any point in the profile

$$R = \frac{\left(C^2 + X^2 \cdot \omega_K^2\right)^{1.5}}{\left(X^2 \cdot \omega_K^2 + 2 \cdot C^2 - W \cdot X\right) \cdot \omega_K} - \rho$$

Table 3.1 – Current values

β	S	C	X	R	P_T
0,5	3,97606E-06	0,0357845	0,085	0,01793894	123,75
1	1,59042E-05	0,0715691	0,08502	0,01795475	123,8
1,5	3,57845E-05	0,1073536	0,08504	0,0179811	123,89
2	6,3617E-05	0,1431381	0,08506	0,01801797	124,02
2,5	9,94015E-05	0,1789227	0,0851	0,01806536	124,18
3	0,000143138	0,2147072	0,08514	0,01812322	124,38
3,5	0,000194827	0,2504918	0,08519	0,01819155	124,61
4	0,000254468	0,2862763	0,08525	0,01827032	124,87
4,5	0,000322061	0,3220608	0,08532	0,01835949	125,17
5	0,000397606	0,3578454	0,0854	0,01845902	125,5
5,5	0,000481103	0,3936299	0,08548	0,01856887	125,86
6	0,000572553	0,4294144	0,08557	0,018689	126,25
6,5	0,000671954	0,465199	0,08567	0,01881936	126,67
7	0,000779308	0,5009835	0,08578	0,0189599	127,12
7,5	0,000894613	0,5367681	0,08589	0,01911056	127,59
8	0,001017871	0,5725526	0,08602	0,01927129	128,09
8,5	0,001149081	0,6083371	0,08615	0,01944201	128,62
9	0,001288243	0,6441217	0,08629	0,01962267	129,17
9,5	0,001435358	0,6799062	0,08644	0,0198132	129,74
10	0,001590424	0,7156907	0,08659	0,02001353	130,33
10,5	0,001753442	0,7514753	0,08675	0,02022359	130,94
11	0,001924413	0,7872598	0,08692	0,02044331	131,57
11,5	0,002103336	0,8230444	0,0871	0,02067259	132,21
12	0,00229021	0,8588289	0,08729	0,02091138	132,87
12,5	0,002485037	0,8946134	0,08749	0,02115959	133,54
13	0,002687816	0,930398	0,08769	0,02141714	134,22
13,5	0,002898548	0,9661825	0,0879	0,02168394	134,92
14	0,003117231	1,001967	0,08812	0,02195992	135,62
14,5	0,003343866	1,0377516	0,08834	0,02224499	136,33

15	0,003578454	1,0735361	0,08858	0,02253907	137,04
15,5	0,003820993	1,1093207	0,08882	0,02284206	137,76
16	0,004071485	1,1451052	0,08907	0,02315389	138,48
16,5	0,004329929	1,1808897	0,08933	0,02347447	139,2
17	0,004596325	1,2166743	0,0896	0,02380372	139,92
17,5	0,004870673	1,2524588	0,08987	0,02414155	140,64
18	0,005152973	1,2882433	0,09015	0,02448787	141,36
18,5	0,005443226	1,3240279	0,09044	0,0248426	142,07
19	0,00574143	1,3598124	0,09074	0,02520565	142,77
19,5	0,006047587	1,395597	0,09105	0,02557695	143,47
20	0,006361696	1,4313815	0,09136	0,02595641	144,16
20,5	0,006683756	1,467166	0,09168	0,02634395	144,84
21	0,007013769	1,5029506	0,09201	0,02673948	145,51
21,5	0,007351734	1,5387351	0,09235	0,02714293	146,16
22	0,007697652	1,5745196	0,0927	0,02755422	146,8
22,5	0,008051521	1,6103042	0,09305	0,02797327	147,43
23	0,008413342	1,6460887	0,09341	0,0284	148,05
23,5	0,008783116	1,6818733	0,09378	0,02883434	148,64
24	0,009160842	1,7176578	0,09416	0,02927622	149,22
24,5	0,009546519	1,7534423	0,09455	0,02972557	149,78
25	0,009940149	1,7892269	0,09494	0,0301823	150,32
25,5	0,010341731	1,8250114	0,09534	0,03064637	150,84
26	0,010751265	1,8607959	0,09575	0,03111768	151,35
26,5	0,011168752	1,8965805	0,09617	0,03159619	151,82
27	0,01159419	1,932365	0,09659	0,03208183	152,28

Calculation of the fuel injection process

Gas pressure in the diesel cylinder at the moment of injection

$$P_{Ц} = \frac{P_{СЖ} + P_{Z}}{2} = 8{,}18 \text{ MPa.}$$

Fuel compressibility factor α_c = 8E-04 MPa^{-1}

Flow coefficient of the filling holes of the plunger pair μ_f = 0,86

Fuel pressure in the low pressure cavity of the fuel pump P_0 = 0,2 MPa

Diameter of the filling holes of the plunger pair d_f = 5 mm

Number of filling holes of the plunger pair i_f = 2

Cross-section of the filling holes of round shape

$$f_H = i_H \cdot \frac{\pi \cdot d_H{}^2}{4} = 3{,}9\text{E-05 m}^2.$$

Spray needle diameter d_n = 8 mm

Diameter of the base of the closing cone of the spray needle d_c = 4,2 mm

Relative value of the differential plane of the spray needle $\delta = 0{,}64$

Angle of the closing cone of the spray needle $\delta_n = 60$ deg

Diameter of the spray well $d_w = 3$ mm

Diameter of spray holes $d_s = 0{,}36$ mm

Number of spray holes $i_s = 8$

Spray needle lift $h_n = 0{,}45$ mm

Residual pressure in the high-pressure system $P_{r.p} = 3$ MPa

Injection start pressure $P_{st} = 36$ MPa

Flow coefficient of the minimum flow section in the closing cone of the spray nozzle $\mu_{c.c} = 0{,}7$

Flow coefficient of the spray nozzle openings $\mu_s = 0{,}6$

Total flow cross-section of the nozzle openings $f_P = 0{,}814$ mm^2

Calculation of the first fuel supply period

Coefficients

$$K_1^1 = \frac{F_П}{\alpha_{СЖ} \cdot 6 \cdot n_К \cdot V_Н} = 14{,}3304; \quad K_3^1 = \frac{\mu_Н}{\alpha_{СЖ} \cdot 6 \cdot n_К \cdot V_Н} \cdot \sqrt{\frac{2}{\rho_Т}} = 1783{,}98.$$

First period pressure

$$P_Н = \left(\frac{K_1^1 \cdot C_П}{K_3^1 \cdot f_Н}\right)^2 + P_0 = 0{,}26529 \text{ МПа}; \quad \Delta P_Н = \left(K_1^1 \cdot C_П - K_3^1 \cdot f_Н \cdot \sqrt{(P_Н - P_0) \cdot 10^6}\right) \cdot \Delta\beta .$$

β	ΔP
0,5	0,247424254
1	0,503792927
1,5	0,7601616
2	1,016530272
2,5	1,272898945

Then $P_1 = 3{,}80$ MPa.

Calculation of the second fuel delivery period

The second fuel delivery period lasts from the moment the fuel accumulator valve is opened until the spray needle lifts. Cross-section of the base of the closing cone of the spray needle

$$F_К = \frac{\pi \cdot d_К^2}{4} = 13{,}847 \text{ mm}^2.$$

Differential area of the spray needle

$$F_{\delta}=\frac{\pi\cdot\left(d_{И}^{2}-d_{К}^{2}\right)}{4}=36{,}393\ \text{mm}^{2}.$$

Coefficients

$$K_{1}^{2}=\frac{F_{П}}{\alpha\cdot 6\cdot n_{К}\cdot\left(V_{Н}+V_{Л}\right)}=14{,}3304;$$

$$K_{3}^{2}=\frac{\mu_{Н}}{\alpha_{СЖ}\cdot 6\cdot n_{К}\cdot\left(V_{Н}+V_{Л}\right)}\cdot\sqrt{\frac{2}{\rho_{T}}}=891{,}99;$$

$$\Delta P=K_{1}^{2}\cdot C_{П}\cdot\Delta\beta;$$

Then P_2 = 32,97 MPa.

Spray needle lift pressure

$$P_{Н}=P_{ВП}-P_{C}\cdot\frac{F_{К}}{F_{\delta}}=33{,}41\ \text{MPa}.$$

Pressure in the high pressure line

$$P_{л}=P_{н}-P_{зт}=29{,}98\ \text{MPa}.$$

Calculation of the third fuel delivery period

The third period lasts from the moment the injector needle lifts, when fuel is injected into the diesel cylinder through the spray holes, until the cut-off begins.

Coefficients

$$K_{1}^{3}=\frac{F_{П}}{\alpha_{СЖ}\cdot 6\cdot n_{К}\cdot\left(V_{Н}+V_{Л}+V_{P}\right)}=9{,}2651;$$

$$K_{2}^{3}=\frac{\mu_{P}\cdot f_{P}}{\alpha_{СЖ}\cdot 6\cdot n_{К}\cdot\left(V_{Н}+V_{Л}+V_{P}\right)}\cdot\sqrt{\frac{2}{\rho_{T}}}=7\text{E-}04;$$

$$K_{C}=\frac{\left(\mu_{ЗК}\cdot F_{ЗК}\right)^{2}}{\left(\mu_{ЗК}\cdot F_{ЗК}\right)^{2}+\left(\mu_{P}\cdot f_{P}\right)^{2}}=0{,}88967;$$

$$F_{ЗК}=\pi\cdot\left(d_{КО}-h_{Н}\cdot\sin\frac{\delta_{Н}}{2}\cdot\cos\frac{\delta_{Н}}{2}\right)\cdot h_{Н}\cdot\sin\frac{\delta_{Н}}{2}=1{,}98\text{E-}06\ \text{м}^{2}$$

$$\Delta P=\left(K_{1}^{3}\cdot C_{П}-K_{2}^{3}\cdot\sqrt{K_{C}\cdot\left(P_{Л}-P_{Ц}\right)\cdot 10^{6}}\right)\cdot\Delta\beta;$$

β	ΔP	P
19	6,298171992	36,27
19,5	6,463951293	42,73
20	6,629730595	49,36
20,5	6,795509896	56,16

21	6,961289197	63,12
21,5	7,127068499	70,25
22	7,2928478	77,54
22,5	7,458627101	85,00
23	7,624406403	92,62
23,5	7,790185704	100,41
24	7,955965005	108,37
24,5	8,121744307	116,49
25	8,287523608	124,78
25,5	8,453302909	133,23
26	8,619082211	141,85
26,5	8,784861512	150,64
27	8,950640813	159,59

Pressure in the high pressure line

$$P_{л1} + 1 = P_{л1} + \Delta P = 159{,}59 \text{ MPa.}$$

Fuel pressure in front of the spray holes

$$P_C = \frac{(\mu_{ЗК} \cdot F_{ЗК})^2 \cdot P_Л + (\mu_P \cdot f_P)^2 \cdot P_Ц}{(\mu_{ЗК} \cdot F_{ЗК})^2 + (\mu_P \cdot f_P)^2} = 142{,}88 \text{ MPa.}$$

The fourth fuel delivery period starts when the fuel injection valve of the fuel accumulator is cut off. The fuel cut-off is set by the electronic fuel injection control unit depending on the diesel engine load.

The fifth fuel delivery period lasts from the moment the injection valve begins to close until the injector needle is seated on the closing cone of the injector nozzle body. The injector needle seating pressure is

$$P_{кв} = P_{нв} \cdot \delta = 23{,}04 \text{ MPa.}$$

3.5. Calculation of fuel pump parts for the engine 6CHN 31.8/33

Calculation of the plunger spring

The main dimensions of the spring are selected in such a way that the coefficient (K) exceeds the force of inertia of the plunger drive parts moving in a reciprocating motion at the moment when the plunger acceleration becomes negative, and the bending and torsional stress in it does not exceed the permissible value. The diameter of the wire from which the plunger spring is made and its diameter are d_w = 7,5 mm; D_0 = 43 mm. Pre-tightening of the plunger spring is f_0 = 6 mm. The gap between the spring coils in the compressed state is Δn = 1 mm. Permissible torsional stress τ_0 = 450 MPa. Coefficient K = 1.2. Modulus of shear elasticity G = 82500 MPa.

Maximum permissible spring load

$$P_{\text{пmax}} = \frac{\pi \cdot d_{пр}^3 \cdot \tau_0 \cdot 10^6}{8 \cdot D_0} = 1732{,}88 \text{ N.}$$

Stiffness of one spring coil

$$K_{ж1} = \frac{G \cdot d_{пр}^4}{8 \cdot D_0^3} = 410397 \text{ N/m.}$$

Full spring deformation

$$f_{\max} = f_0 + S_{\text{п}} = 37 \text{ mm.}$$

Number of working turns of the spring

$$i_{\text{рп}} = \frac{f_{\max} \cdot K_{ж1}}{P_{пмах}} = 8{,}70768, \text{ accept } 9.$$

Stiffness of the spring

$$K_{ж} = \frac{K_{ж1}}{i_{рп}} = 51299{,}6 \text{ N/m.}$$

Spring deformation at the moment when acceleration

$$f_w = f_0 + S_w = 0{,}0249 \text{ m.}$$

The spring force when the plunger acceleration becomes

$$P_{\text{п}w} = f_w \cdot K_{\text{ж}} = 1277{,}51 \text{ N.}$$

The force of inertia of the plunger actuator parts that move backwards when the acceleration becomes negative

$$P = m \cdot / W_0 = 151{,}612 \text{ N.}$$

Coefficient of excess of spring force, inertial force of reciprocating parts

$$K = \frac{P_{пw}}{Pu} = 8{,}426 \text{ much higher than 1.2.}$$

Spring length in the free state

$$L_{\text{пр}} = (d_{\text{пр}} + \Delta n) \cdot i_{\text{р}} + f_{\max} + i \cdot d_{\text{пр}} = 126{,}0177 \text{ mm.}$$

Spring coil pitch

$$t = d_{пр} + \frac{f_{\max}}{i_p} + \Delta n = 12{,}585 \text{ mm.}$$

Determining the safety margin of a spring:

Spring index $C' = \frac{D_0}{d_{пр}} =$ 5,7333.

Correction coefficient for torsion distribution around the circumference of the coil cross-section and shear stress

$$K_n = \frac{4 \cdot C' - 1}{4 \cdot C' - 4} + \frac{0{,}615}{C'} = 1{,}26572.$$

Maximum spring force

$$P_{п\ max} = f_{max} \cdot K_ж = 1886{,}17 \text{ N}.$$

Maximum bending and torsional stress in the spring

$$\tau_{max} = \frac{8 \cdot K_n \cdot P_{пмax} \cdot D_0}{\pi \cdot d_{пр}^3} = 619{,}958 \text{ MPa}.$$

Minimum spring force

$$P_{по} = f_0 \cdot К_ж = 307{,}797 \text{ MPa}.$$

Minimum bending and torsional stress in the spring

$$\tau_{min} = \frac{8 \cdot K_n \cdot P_{по} \cdot D_0}{\pi \cdot d_{пр}^3} = 101{,}169 \text{ MPa}.$$

Average cycle voltage

$$\tau_m = \frac{\tau_{мax} + \tau_{min}}{2} = 360{,}563 \text{ MPa}.$$

Amplitude of the cycle voltage

$$\tau_a = \frac{\tau_{мax} - \tau_{min}}{2} = 259{,}394 \text{ MPa}.$$

For 50 HFA steel yield strength τ_s = 950 MPa, endurance strength τ_1 = 500 MPa, γ_τ = 1.

Safety margin

$$n = \frac{1}{\frac{\tau_m}{\tau_s} + \gamma_\tau \cdot \frac{\tau_a}{\tau_{-1}}} = 1{,}1132.$$

Check spring for resonance

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Зм.	Лист	№ докум.	Підпис	Дата		

$$n_c = 2{,}17 \cdot 10^7 \frac{d_{np}}{i_{p\cdot} \cdot D_0^2} = 11002{,}6 \text{ min}^{-1} \qquad \frac{n_c}{n_k} = 29{,}3 > 10,$$

therefore spring vibrations are not dangerous.

Calculation of the plunger actuator cam

Determination of contact stresses on the surface of the fuel-injected plunger drive cam at the moment when the fuel pressure in the superplunger volume reaches the maximum value of $P_{f\max}$.

Force acting along the axis of the plunger

$$P = P_p + P_s + P_i = 0{,}00222 \text{ MN}.$$

Force from fuel pressure in the supercharger volume

$$P_p = p_{\text{нmax}} \cdot F_{\text{п}} = 8{,}8\text{E-05 MN}.$$

Spring strength

$$P_s = (f_0 + \text{S}) \cdot \text{К}_{\text{ж}} \cdot 10^{-6} = 0{,}00189 \text{ MN}.$$

The inertial force of the reciprocating plunger drive parts

$$P_i = m \cdot W \cdot 10^{-6} = 0{,}00024 \text{ MN}.$$

Angle δ at the moment when $P_{t\max}$ is reached:

$$\varpi_{\kappa\cdot} = \frac{\pi \cdot n_k}{30} = 39{,}25 \text{ s}^{-1},$$

$$X = R_0 + \rho + S = 0{,}09186 \text{ m},$$

$$\delta = \arctan\left(\frac{C}{X \cdot \varpi_k}\right) = 0{,}49679 \text{ rad (28,48 degrees)}.$$

The force transmitted by the roller to the cam

$$N = \frac{P}{\cos\delta} = 0{,}0025 \text{ MN}.$$

Contact voltage

$$\sigma = 0{,}418\sqrt{\frac{N \cdot E}{b} \cdot \left(\frac{1}{\rho} + \frac{1}{R}\right)} = 384{,}192 \text{ MPa} < \sigma_{\text{д}} = [1200\ldots2000] \text{ MPa}$$

the contact voltages do not exceed the permissible values.

Calculating the cam shaft

The camshaft is calculated for bending and torsion, and its deflection and twist angle are determined.

Bending resistance moment

$$W_{зг} = \frac{\pi \cdot \left(d_B^4 - d_0^4\right)}{32 \cdot d_B} = 6{,}9\text{E-05 m}^3.$$

Bending moment

$$M_{зг} = \frac{N_{MAX} \cdot a \cdot b_B}{l} = 0{,}00021 \text{ MN·m}.$$

Bending stress

$$\sigma_{зг} = \frac{M_{зг}}{W_{зг}} = 3{,}0591175 \text{ MPa}.$$

Maximum torque on the fuel pump cam

$$M_{К\,max} = N_{max} \cdot X \cdot \sin\delta = 0{,}0001 \text{ MN·m}.$$

Torsional resistance torque

$$W_k = 2 \cdot W_i = 0{,}0001375 \text{ м}^3.$$

Torsional stress

$$\tau = \frac{M_{KMAX}}{W_K} = 0{,}8025032 \text{ MPa}.$$

The total applied stress arising in the cam shaft from the combined action of bending and torsional moments, which is determined according to the third theory of strength

$$\sigma_П = \sqrt{\left(\sigma_И + 4 \cdot \tau^2\right)} = 2{,}37385 \text{ MPa}.$$

The total stress value does not exceed the permissible value of 150 MPa.

Moment of inertia of the cam shaft cross section

$$I = \frac{\pi \cdot \left(d_B^4 - d_0^4\right)}{64} = 3{,}1\text{E-06 m}^4.$$

Deflection of the camshaft span

$$Y = \frac{N_{MAX} \cdot l^3}{48 \cdot E \cdot I} = 2{,}8\text{E-06 mm}.$$

Deflection of the cam shaft span does not exceed 1% of the plunger stroke.

Polar moment of inertia of the cam shaft cross-section

$$I_п = 2 \cdot I = 6{,}187\text{E-}06 \text{ m}^4.$$

Shear modulus of elasticity for steel G = 81500 MPa.

Torsion angle in degrees of camshaft rotation

$$\varphi_з = \frac{M_{KMAX} \cdot L}{G \cdot I_П} \cdot \frac{180}{\pi} =$$ 0,03763 degrees of camshaft rotation.

The camshaft twist angle does not exceed the permissible angle [φ_3] = 1 degrees of camshaft rotation.

Calculation of the push rod

Calculation of the roller axis, roller sleeve, and pusher guide surface.

The pusher axis is calculated for bending, shearing and specific pressure on the supports.

Geometrical dimensions of the push rod elements

$l_п$ =	0,068	m;	$d_{то}$ =	0,07	m;
$l_в$ =	0,034	m;	$H_т$ =	0,19	m;
b =	0,03	m;	$d_{пн}$ =	0,038	m;
α =	0,237		$l_о$ =	0,015	М.

Bending stress of the pusher axis

$$\sigma_И = N_{MAX} \cdot \frac{l_П + 2 \cdot l_В - 1.5 \cdot b}{1.2 \cdot (1 - \alpha^4) \cdot d_{ПН}^3} =$$ 3,4936 MPa.

Tangential shear stresses in the sections of the pusher axis

$$\tau_K = \frac{0{,}85 \cdot N_{MAX} \cdot (1 + \alpha + \alpha^2)}{(1 - \alpha^4) \cdot d_{ПН}^2} =$$ 1,9243 MPa

the tangential shear stresses in the sections of the pusher axis do not exceed the permissible values [σ] = 65 MPa.

Specific load in supports

$$K_o = \frac{N_{MAX}}{2 \cdot d_{ПН} \cdot l_0} =$$ 2,21048 MPa

the specific load in the supports does not exceed the permissible value [K_0] = 60 MPa.

Specific load on the inner surfaces of the bushing

$$K_{вв} = \frac{N_{MAX}}{2 \cdot d_{ПН} \cdot b} = 1{,}10524 \text{ MPa}$$

the specific load on the inner surface of the sleeve does not exceed the permissible value [$K_{i.s}$] = 70 MPa.

Specific load on the outer surface of the plunger

$$K_{вн} = \frac{N_{MAX}}{d_{BT} \cdot b} = 40{,}7998 \text{ MPa}$$

the specific load on the outer surface of the bushing does not exceed the permissible value [$K_{o.s}$] = 50 MPa.

Maximum load at the lower edge of the guide surface of the pusher body

$$K_{m} = \frac{2 \cdot N_{MAX} \cdot \sin \delta}{d_{TO} \cdot H_{T}} = 0{,}17559 \text{ MPa}$$

The maximum load at the lower edge of the guide surface of the pusher body does not exceed the permissible value [K_t] = 18,5 MPa.

Calculation of the plunger

Calculation of the plunger for compression in the minimum cross-section and specific load of the support face. Compressive stress in the minimum section of the plunger if the minimum and maximum diameters of the plunger are the same

$$\sigma_{СЖ} = \frac{P_T}{F_{ПMIN}} = 0{,}26529 \text{ MPa}$$

if the minimum diameter of the plunger is $d_{p\min}$ = 17 mm, then

is the area of the minimum cross-section of the plunger

$$F_{\min} = \frac{\pi \cdot d_{ПMIN}^2}{4} = 214{,}022 \text{ mm}^2;$$

$$\sigma_{СЖ} = \frac{P_T}{F_{ПMIN}} = 0{,}40939 \text{ MPa};$$

the compression stress in the minimum cross-section of the plunger does not exceed the permissible [σ_{com}] = 300 MPa.

Specific load at flat ends of the plunger and pusher diameter of the plunger support end d_{pop} = 10 mm

– area of the plunger support face

$$F_{пoп} = \frac{\pi \cdot d_{ПОП}^2}{4} = 78{,}5 \text{ mm}^2;$$

$$K_m = \frac{P_T}{F_{ПОП}} = 1{,}11615 \text{ MPa};$$

the specific load at the flat ends of the plunger and pusher does not exceed the permissible $[K_t]$ = 2000 MPa.

3.6. Construction of the high-pressure fuel pump

The standard high-pressure fuel pump (fig. 3.5) is a classic high-pressure fuel pump with a fuel rail. It has many of the disadvantages of these systems. The pressure and thrust characteristics change undesirably when changing the speed and load conditions, poor spraying at idle and start-up conditions, and difficulty in ensuring the identity of the flow characteristics across sections.

The standard HPFP is equipped with a mechanical regulator and a mechanical system for disconnecting pump sections.

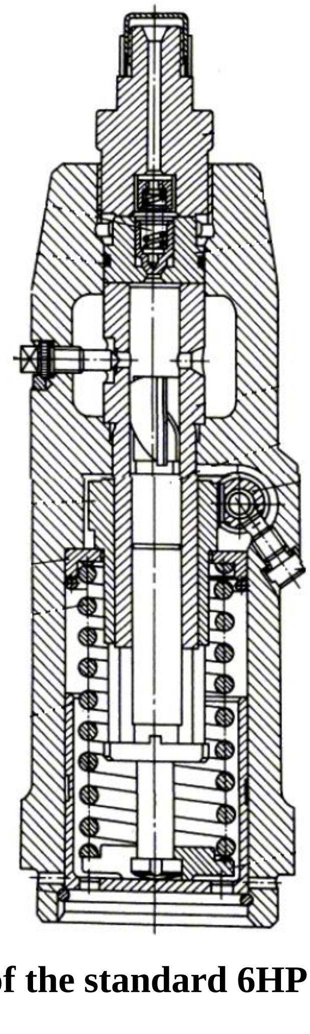

Figure 3.5 – Design of the standard 6HP 31.8/33 engine HPV

The principle of operation of a standard high-pressure pump is as follows. The reciprocating motion of the plunger, which is mediated by the roller pusher, cam shaft, and spring, is alternately used to fill the over-plunger volume with fuel and then to deliver fuel through the valve and discharge pipe to the injector.

Fuel is injected when the plunger moves upwards after the end edge of the plunger head overlaps the inlet channels of the plunger sleeve (liner); the overplunger volume is filled with fuel when the plunger returns to its original position under the action of a spring.

The fuel injection stops when the cut-off edge of the plunger head opens the corresponding window in the sleeve, and then the fuel flows from the superplunger space into the low-pressure pipeline.

The fuel pump of the diesel engine 6CHN 31,8/33 consists of:

– rotary gear sleeve;
– fixing screw;
– discharge valve body;
– sealing ring ;
– valve;
– high pressure connection;
– discharge valve spring;
– plunger sleeve ;
– plunger;
– adjusting rack;
– screw for fixing the rail;
– guide cup;
– pump casing;
– bottom plate of the plunger spring;
– retaining ring;
– plunger spring;
– upper plunger spring plate.

The section of the designed HPV differs from the section of the standard HPV by the presence of an electrically controlled valve *2* (fig. 3.6). The valve is normally open. The feeding process is as follows. Plunger *1* rises from its lower position and pumps fuel into the high-pressure cavity *A* (red). However, at the beginning of the supply, valve *2* is open, and all excess fuel goes to the drain (blue). At a certain point in time, an operating voltage is applied to the solenoid, and the valve retracts to separate cavity *A* from cavity *B*. The pressure in cavity *A* rises and injection occurs. After a while, the solenoid switches off and the valve opens under the action of the spring. The supply ends.

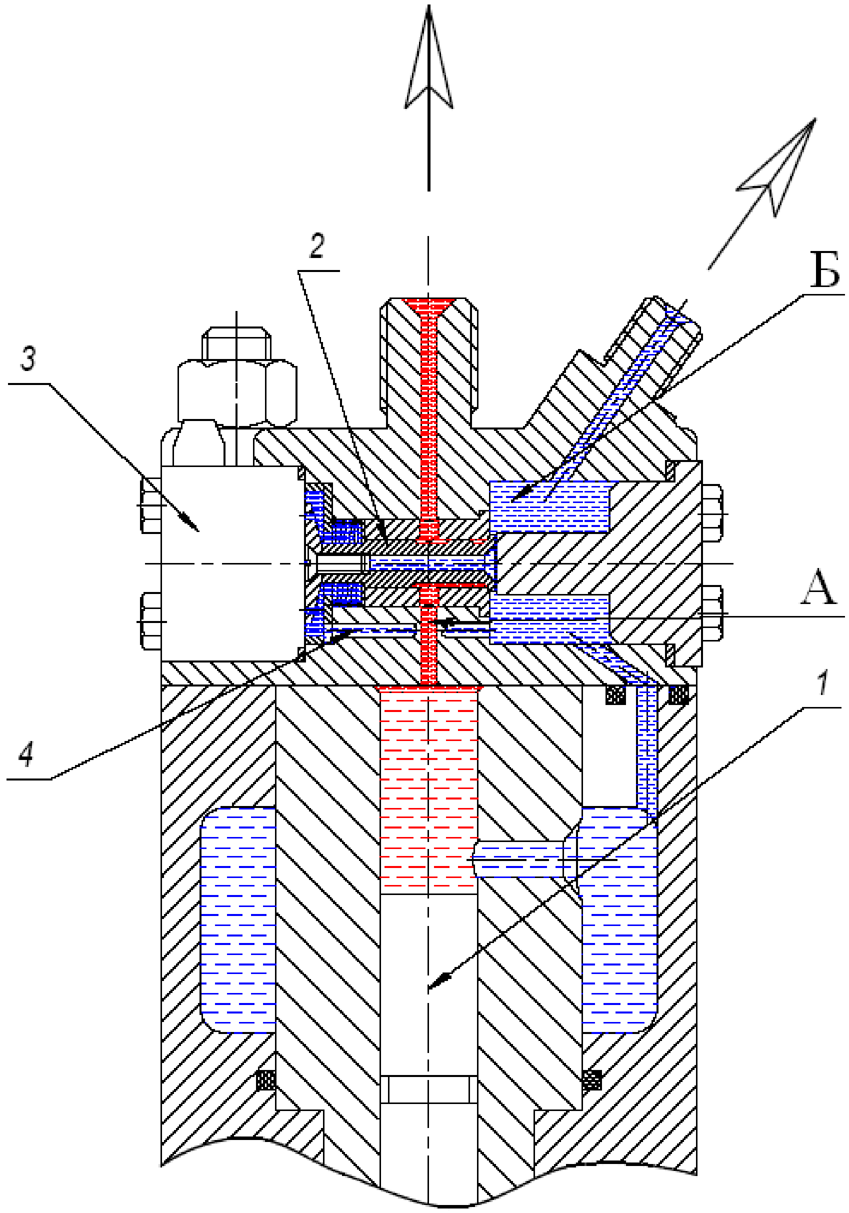

Figure 3.6 – HPV with solenoid valve:

1 – plunger; *2* – electrically controlled valve; *3* – solenoid; *4* – balancing channel; *A* – high-pressure cavity; *B* – booster pressure cavity

The valve is hydraulically relieved both at high and low pressure. To relieve the valve at low pressure, a balancing channel *4* is provided to equalise the pressure on both sides of the valve.

This system allows you to adjust both the fuel injection advance angle (the moment the solenoid is switched on) and the cycle flow (the time during which the valve is closed). The flow parameters are not fixed, i.e. they can vary depending on the diesel engine's operating mode and the current thermal or even technical condition of the cylinder.

The speed of the valve is subject to increased demands; it must operate in both directions within a time of no more than 0,15 ms. To ensure this performance, hydraulic unloading of the valve is necessary, as well as the selection of parameters of the elements that control the valve movement.

The parameters of the electromagnetic valve are as follows:

– valve spring stiffness 6,3 N/mm

– pre-tightening 70 N

– solenoid force 250 N at a stroke of 0,3 mm

– valve weight 31 g

– valve travel 0,3 mm

– minimum stroke 0,1 mm

– gap between solenoid and armature 0,1 mm

3.7. Conclusions to the part

A high-pressure fuel pump with a solenoid valve for a 6ChN 31.8/33 engine was designed and calculated. The main geometric and operational parameters of the high-pressure pump and the parameters of the solenoid valve were obtained (table 3.2).

The fuel pump provides all the necessary parameters of fuel injection for the 6CHN 31.8/33 engine, which are included in the calculation of the operating cycle (Part 2 of the diploma project).

The main elements are designed for strength. According to the calculation, the operating mechanical loads are within the permissible limits.

Table 3.2 – Main parameters of the designed HPV with a solenoid valve

№ n.o.	Parameter	Dimension	Value
1	Diameter of the plunger	mm	21
2	Stroke of the plunger	mm	31
3	Maximum speed of the plunger	m/s	1,95
4	Injection start pressure	MPa	36
5	Residual pressure in the high-pressure system	MPa	3
6	Pressure in the high pressure line	MPa	159,6
7	Fuel pressure in front of the spray nozzles	MPa	142,9
8	Nozzle needle seating pressure	MPa	23
9	Fuel supply duration	deg. of c. r.	38
10	Stiffness of the solenoid valve spring	N/mm	6,3
11	Pre-tightening the solenoid valve spring	N	70
12	Electromagnet force at 0.3 mm stroke	N	250
13	Weight of the solenoid valve	g	31
14	Solenoid valve stroke	mm	0,3
15	Minimum stroke of the solenoid valve	mm	0,1
16	Gap between electromagnet and armature	mm	0,1

PART 4
ECONOMIC PART

4.1. Economic justification for the use of an upgraded tug engine

Objective: analytical justification of the economic efficiency of introducing a modernised engine into operation compared to the existing prototype.

In determining the economic effect of creating a new engine, the parameters of the base model and the upgraded model are compared. Table 4.1 shows the technical and economic indicators of the basic 6CHN31.8/33 engine installed on a tow and the upgraded one.

Table 4.1 – Technical and economic indicators of the basic and upgraded engine

Name of indicators	Symbols	Basic engine	Engineered engine
Power, kW	N_e	882	1234
Rotation speed, min^{-1}	n	750	750
Diameter of the cylinder, m	D	0,318	0,318
Stroke of the piston, m	S	0,33	0,33
Specific fuel consumption, kg/kW·h	g_e	0,214	0,208
Mechanical efficiency	η_m	0,88	0,88
Specific oil consumption, kg/kW·h	g_o	0,0018	0,0018
Weight, kg	G	14500	14500
Dimensions, mm	L	4287	4287
	B	1555	1555
	H	2343	2343
Price of the engine, UAH	P	380000	400000
Price of fuel, UAH/(kW·h)	P_f	0,85	0,85
Price of oil, UAH/kg	P_o	9,5	9,5
Work hours for the year, h	t	4380	4380
Number of control repairs	η	1	1
Total capital repairs coefficient	$\varepsilon_{\min}$	1,1	1,1
Partial labour cost coefficient	b	0,05	0,05
Power utilisation coefficient	K_p	0,95	0.95
Coefficient of part of the cost of capital repairs	$K_{c.r}$	0,3	0,3
Time of continuous operation, h	$t_{c.o}$	300	300
Engine life before the first bulkhead, h	t_{bul}	12000	12000
Engine life before overhaul, h	$t_{o.h}$	72000	72000
Normative cost-effectiveness comparison coefficient	E	0,15	0,15

The economic efficiency of the designed engine is determined by the methodological instruction ’’Methodology for the use of new equipment, inventions and rationalisation proposals in the national economy’’.

Purpose of the method

1. Feasibility study of the best options for the manufacture and introduction of new equipment.
2. Reflection of indicators of efficiency and effectiveness in standards and norms.
3. Calculations of the actual efficiency of the introduction of new equipment.

Calculations of economic efficiency are given in table 4.2.

Table 4.2 – Calculations of economic efficiency

Name	Symbols	Method of determination	Basic engine	Modernised engine
Annual output	N, kW·h per year	$N = К_П \cdot N_e \cdot t$	3670002	5134674
The service period	T, year	$T = \dfrac{t_{кр}}{t}$	16,44	16,44
Annual current fuel costs	C_f, UAH	$\text{З}_п = N \cdot Ц_п \cdot g_e$	667573,3638	907810,3632
Annual operating costs for oil	C_o, UAH	$\text{З}_м = g_м \cdot N \cdot Ц_м$	62757,0342	87802,9254
Annual costs for routine repairs	$C_{r.r}$, UAH	$\text{З}_{п.р} = \dfrac{b \cdot t \cdot Ц}{t_{пер}}$	6935	7300
Annual costs of capital repairs	$C_{c.r}$, UAH	$\text{З}_{к.р} = \dfrac{К_{к.р} \cdot t \cdot Ц}{t_{к.р} \cdot \varepsilon_{\min}}$	6304,55	6636,36
Annual economic effect for the consumer	U_1, UAH	$U_1 = \text{З}_п + \text{З}_м + \text{З}_{к.р} + \text{З}_{п.р}$	743569,94	1009549,65
	$U_1^{1б}$, UAH	$U_1^{1б} = \dfrac{N_c}{N_б} \cdot U_1^б$	1040323,481	–
Annual economic effect on the consumer from the use of the designed engine	$E_к$, UAH	$E_к = \left[Ц_б \cdot \dfrac{N_c}{N_б} \cdot \left(\dfrac{\frac{1}{T_б} + E_б}{\frac{1}{T_c} + E_c} \right) + \dfrac{U_1^{1б} - U_1^c}{\frac{1}{T_c} + E_c} \right] - Ц_c$		277618,15

Additional one-off costs for engine upgrades	ΔK, UAH	–	20000
Annual savings when operating the upgraded engine	ΔS, UAH	$\Delta S = U_1^{1б} - U_1^{c}$	30773,83
Payback period of the upgraded engine	T_{pay}, year/month	$T_{окуп} = \frac{\Delta K}{\Delta S}$	0,65/ 7,8

4.2 Conclusions to the section

The above calculation results confirmed the economic feasibility of using the upgraded 6CHN 31.8/33 engine. The annual economic effect for the consumer from the use of the designed engine, compared to the base engine, is 30773,83 UAH. Modernisation of the basic 6CHN 31.8/33 engine will pay off in 0,65 years.

PART 5
ANALYSING AND DEVELOPING MEASURES TO MEET OCCUPATIONAL HEALTH AND SAFETY AND ENVIRONMENTAL PROTECTION REQUIREMENTS

5.1. Analysis of hazardous and harmful occupational factors affecting diesel maintenance personnel

Occupational health and safety is a system of laws and regulations aimed at ensuring workplace safety and the corresponding socio-economic, organisational, technical and sanitary and hygienic measures.

The objective of occupational health and safety is to minimise possible injuries and illnesses to employees, while ensuring comfort and maximum productivity. Real production conditions are usually characterised by the presence of certain hazardous and harmful production factors.

A hazardous occupational factor is a factor whose impact on an employee under certain conditions will lead to illness or reduced performance. It is not always possible to draw a clear line between hazardous and harmful factors. The same factor can lead to an accident or a decrease in productivity. The operation of an internal combustion engine, as well as various systems and mechanisms that serve the diesel engine, creates a number of occupational hazards that are dangerous to human life and health. These factors are regulated by GOST 12.0.003–83.

Fuel and oil vapours

Fuel and oil vapours penetrate the human body, are irritating and can lead to chronic diseases of the lungs and respiratory tract. According to DSTU 12.1. 005–88, the permissible limits for diesel vapour concentration are 100 mg/m^3 in the air of industrial premises. Engine operation also releases a large amount of harmful substances as a result of fuel and oil combustion. According to SNiP 245–71, the permissible limit of carbon monoxide concentration should not exceed 20 mg/m^3.

The high temperature of exposed engine parts (exhaust manifold, gas turbine, muffler) can cause harm (burns) to people. To prevent this, all hot parts of the engine should be covered with heat-insulating materials.

Fire safety

Fires in the engine compartment are caused by faulty electrical appliances, spontaneous combustion of oily fabrics, malfunctioning shut-off valves, wear and corrosion of fuel equipment elements, the use of open flames, and non-compliance with fire safety standards when working with flammable substances.

Electrical safety

Passing through the body, the current affects:

1. Thermally. It is expressed in burns, heating of blood vessels, nerves and other tissues, and hyperspeeding of muscle tissue.

2. Electrical. It is expressed in a change in the physicochemical composition of blood and other fluids.

3. Biological. It is expressed in the irritation and destruction of body tissues, as well as in the disruption of internal processes.

Lighting

The lighting system plays an important role in the production process. The system is designed to provide lighting in the engine room.

The lighting electrical equipment of the engine room consists of the following elements:

– a lighting panel with circuit breakers. It consumes a voltage of 380 V with a frequency of 50 Hz;

– luminaires with 100 W incandescent lamps;

– luminaires with two 40 W fluorescent lamps;

– emergency lighting fixtures;

– lighting plugs;

– switches.

Ventilation

Toxic compounds (CO, SO_2, CH) can enter the engine room as a result of leaks in the connections between engine parts. To prevent the concentration of harmful substances from rising, a ventilation system is required.

The ventilation system is designed to create normal meteorological conditions of the air environment in the engine room. The ventilation system is designed in accordance with the requirements and norms of SNiP.

Ventilation systems used in the engine room are artificial and natural. Exhaust ventilation from the engine room is carried out naturally through gas-tight partitions and louvres installed on the chimney covers. Supply ventilation of the engine room is provided by a fan.

For effective operation of ventilation systems, it is important that the following technical and sanitary requirements are met:

1. The amount of supply air should correspond to the amount of exhaust air, the differences between them should be minimal.

2. The ventilation system should not create noise in the workplace that exceeds the permissible limits.

3. The ventilation system must be electrically safe, fire and explosion safe, easy to use, reliable in operation and efficient.

Noise

Noise significantly impairs productivity. It affects humans, with an intense noise level, hearing aid fatigue is observed for a long time, which can lead to partial or even complete hearing loss. Sanitary noise standards are given in table 5.1.

Table 5.1 – Sanitary noise standards

b, Hz	63	125	250	500	1000	2000	4000	8000
$\alpha_{про}$, dB	103	96	91	88	85	83	81	80

Vibration

Vibration is caused by the dynamic imbalance of the masses in the crank mechanism of an internal combustion engine. Local vibration causes vascular spasms and impairs blood circulation. General vibration with a frequency of 0,7 Hz causes seasickness, with a frequency of 4...30 Hz it can cause damage to the shoulder girdle and most internal organs due to resonance phenomena. Standard vibration norms are given in table 5.2.

Table 5.2 – Sanitary vibration standards

Direction of vibration standardisation	RMS vibration velocity (numerator, m/s· 10^{-2}), logarithmic vibration level (denominator, dB)						
	1	2	4	8	15	31,5	63
Vertical	20/132	7,1/123	2,5/114	1,3/108	1,1/107	1,1/107	1,1/107
Horizontal	–	3,5/117	1,3/108	0,63/102	0,56/101	0,56/101	0,56/101

Calculations of noise and vibration levels in the engine room

The noise level from the 6CHN 31.8/33 engine is determined by the formula

$$L = \left[54 + 10 \cdot \lg\left(n_{н} + Ne^{0.55}\right) + 30\lg\left(\frac{n}{n_{н}}\right)\right],\ Дб,$$

where n_n = 750 rpm – nominal speed; n = 750 rpm – operating speed; N_e = 1234 kW – rated motor power:

$$L = \left[54 + 10\lg\left(750 + 1234^{0.55}\right) + 30\lg\left(\frac{750}{750}\right)\right] = 83{,}03\ Дб\ .$$

The noise level is within acceptable limits.

To reduce the aerodynamic noise generated by the engine, mufflers of various designs are used. The use of mufflers can reduce the overall noise level by 10...12 dB.

To reduce the airborne noise emitted by the external surfaces of the engine, soundproofing or enclosure boxes are used, which reduce the overall noise level by 10...15 dB, and additionally by 20 dB and above.

To reduce mechanical noise, it is necessary to reduce the gaps between parts and assemblies, and to manufacture structures from materials with greater internal friction and noise-absorbing coatings.

The vibration level for the 6CHN 31.8/33 motor is determined by the formula:

$$L = 44 + 10\lg\left(\frac{n_{н} \cdot Ne^{0.55} \cdot \left(\frac{1+Ne}{m}\right)}{1 + \left(\frac{1}{1500}\right)^{3} \cdot \frac{m}{Ne}} + 30\lg\left(\frac{n}{n_{н}}\right)\right),\ Дб$$

where m = 14500 kg – is the mass of the engine;

$$L = 44 + 10\lg\left(\frac{750 \cdot 1234^{0.55} \cdot \left(\frac{1+1234}{14500}\right)}{1+\left(\frac{1}{1500}\right)^{3} \cdot \frac{14500}{1234}} + 30\lg\left(\frac{750}{750}\right)\right) = 79{,}06\ Дб$$

The vibration level does not exceed the standards.

Measures to reduce noise and vibration

Noise and vibration affect the health of those working in the engine room, reduce productivity and impair attention. Engines are among the loudest mechanisms, and in most cases are the main source of noise and vibration. The most effective and, at the same time, the most complex method of dealing with engine noise is to use the engine itself. This method is based on a special workplace organisation and design of the engine and its components, as well as improved manufacturing technology and processing of engine parts (increased accuracy of gear teeth cutting, general finishing and lapping of parts).

One of the most common methods is sound and vibration isolation. Airborne noise is isolated by means of soundproofing enclosures and partitions, as well as by soundproofing the machine room.

An effective way to reduce noise and vibration is to install vibration dampers in areas of increased vibration, as well as to reinforce doors and hatches to reduce shaking.

A very effective and important way to reduce the harmful effects of noise on the human body is to use personal noise protection equipment: earplugs and headphones, helmets with earplugs, soundproof cabs for engine control. Depending on their design and noise frequency, these types of personal protective equipment can reduce the sound perceived by humans by 15...20 dB.

Development of measures to reduce the impact of hazardous and harmful production factors affecting personnel

Only people who have undergone special technical training and hold a safety certificate authorising them to operate an internal combustion engine are allowed to work on diesel engines.

The following rules must be observed before starting the diesel engine:

– turn the crankshaft of the diesel engine with the gas valves open for at least two revolutions;

– warn those present about the start of the diesel engine;

– to determine the gas content of the engine room using instruments;

– when working at night, ensure normal lighting of the engine room;

– when the diesel engine is running, it is forbidden to open indicator valves without measuring devices attached to them.

5.2. Environmental protection

Environmental protection is a system of legislative acts that ensure the normal functioning of the biosphere when it is affected by natural or anthropogenic factors.

One of the prerequisites for healthy and highly productive work is ensuring clean air and normal meteorological conditions in the working area of the premises.

Preventing the impact of such harmful production factors as gas, steam, dust, excessive temperature and humidity and creating a healthy airspace is an important economic task that should be carried out in a comprehensive manner, simultaneously with addressing the main production issues.

Harmful substances enter the human body primarily through the respiratory tract, but also through the skin and in food. Most of these substances are classified as hazardous and harmful occupational factors because of their impact on the human body.

Environmental pollution arising from the operation of diesel engines

During operation, a diesel engine interacts with the environment: it uses air and water to operate and emits exhaust gases into the atmosphere.

Let's consider the nature of the impact of the components in the engine's internal combustion system on the human body. Hygienists divide them into 6 groups based on this feature.

The first group includes substances that do not have a direct harmful effect on humans and the biosphere as a whole: nitrogen N_2, oxygen O_2, hydrogen H_2, water vapour H_2O, and carbon dioxide CO_2.

The second group includes CO – carbon monoxide or carbon monoxide – a colourless, odourless and tasteless gas, slightly lighter than air (density is 0,97 relative to air). It is practically insoluble in water. Flammable (forms explosive mixtures with air). Getting into the human lungs, and from there into the blood, it displaces oxygen from the blood, as it has 200 times greater solubility than oxygen (a decrease in the oxygen content in the blood leads to suffocation). The harmful effect of carbon monoxide on the human body is also explained by the fact that red blood cells, erythrocytes, absorb CO and lose their ability to participate in the body's gas exchange. As a result, a person suffers from oxygen starvation, associated disruption of the central nervous system, and in severe cases, death. At low concentrations in the air, it causes dizziness and nausea. Since carbon monoxide has almost the same density as air (28 versus 28.7), it is not easily removed from the premises.

The effect of carbon monoxide depends on its concentration in the atmosphere. Up to 0,0016 % by volume, the concentration is harmless, while 0,01 % causes chronic poisoning during prolonged exposure, 0,05 % causes mild poisoning after 1 hour, and 1,0 % causes loss of consciousness after several breaths and death if the person is not removed from the air volume with such a concentration of carbon monoxide and provided with the necessary assistance.

The third group includes nitrogen oxides, mainly nitric oxide (oxide) NO and nitrogen dioxide (dioxide) NO_2. Nitrogen oxide NO is a colourless gas, poorly soluble in water, and oxidises to NO_2 in air. Nitrogen dioxide NO_2 is a red-brown gas with a characteristic pungent odour, heavier than air. In equal amounts, nitrogen oxides are more toxic than CO. When it enters the human body and reacts with water, NOx forms nitric and nitrous acids and their compounds in the respiratory tract, which destroy lung tissue, causing chronic diseases and irreversible changes in the cardiovascular system. NO_x also causes diseases of the mucous membranes of the upper respiratory tract, chronic bronchitis, and nervous disorders. In combination with hydrocarbons, it forms toxic nitroolefins. The consequences of poisoning have a latent period when the poisoned person feels well, but then becomes seriously ill.

Depending on the volume concentration in the air (in %), the effect of NO_2 on the human body is as follows: 0,00001 is the absolute threshold of exposure; up to 0,0003 is the threshold of perception of the smell; 0,0013 is the threshold of mucous membrane irritation; up to 0,002 is the formation of methemoglobin; 0,004...0,008 is pulmonary edema. In addition, nitrogen oxides are involved in photochemical reactions of smog formation.

The fourth group, the most numerous, includes various hydrocarbons (compounds of the SnHm type), which are representatives of all homologous series: alkenes, alkadienes, cyclones, and aromatic compounds, including carcinogens. They have an unpleasant odour, cause many chronic diseases, and have a general toxic and irritating effect.

The fifth group includes aldehydes. They have a pungent odour (especially formaldehyde). At certain doses, they cause irritation of the respiratory tract and mucous membranes of the nose and eyes. Their effect on the human body is characterised by irritating and general toxic effects on the central nervous system and internal organs. VG contains mainly formaldehyde and acrolein.

The sixth group includes soot. Soot consists of small carbon particles ranging from fractions of a micron to tens of microns; the smallest (no more than 10 microns) can remain in the air for several days. Like any aerosol, soot pollutes the air, impairs visibility and can irritate the respiratory tract. Carbon as a chemical substance does not pose a direct threat to the human body. The main danger of soot is that it is a carrier of carcinogenic substances, namely benzopyrene, which is adsorbed by the soot surface and has a stronger effect on living cells than in its pure form. Benzpyrene is found not only in diesel soot, but also in soot from petrol engines, which contains 200 times more of this substance. However, there is much more soot in diesel exhaust gas than in petrol engines.

When using sulphurous fuels, diesel engines also contain inorganic gases, which are sulphur compounds. They have a pungent odour, cause irritation of the upper respiratory tract, and disturb protein metabolism in the body. They are mainly sulphur dioxide SO_2 and hydrogen sulphide H_2S. Sulphur dioxide is a colourless gas with a pungent odour, heavier than air (specific gravity by air 2,264), well soluble in water, and forms sulphuric acid. The effect of SO_2 on the human body, depending on its volume content in the air, %, causes the following effects: 0,0007...0,001 – throat irritation; 0,0017 – eye irritation, cough; 0,04 – poisoning in 3 minutes; 0,01 – poisoning in 1 minute.

Engine exhaust gases consist of 99,0...99,9 % of complete combustion products (carbon and water vapour), residual oxygen and nitrogen in the air. However, it is the remaining part of the fuel gas (no more than 1 % of the total fuel gas consumption, which is almost equal to the air consumption) that determines the environmental level of engines, i.e. the degree of harmful impact on the environment: flora and fauna, humans, and architectural buildings.

The main harmful components of diesel engine exhaust gases are nitrogen oxides, soot, and sulphur compounds.

For petrol engines, carbon monoxide, hydrocarbons, nitrogen oxides and lead compounds (for leaded petrol) are toxic. Hydrocarbons emitted by petrol engines are released into the air both with exhaust gases and crankcase gases, as well as in the form of vapours from the carburettor and petrol tank. For diesel engines, hydrocarbon emissions are usually attributed to exhaust gases only. In petrol engines, the total amount of toxins in the exhaust gas is on average higher than in diesel engines. At the same time, there is a different proportion between the main toxins in the exhaust gas of petrol engines and diesel engines.

In addition to harmful substances, the exhaust gas contains nitrogen as ballast and carbon dioxide CO_2. Regarding nitrogen, it can be noted that while it is harmless to humans, it is a source of harmful oxides in the engine cylinder.

Carbon dioxide does not cause any significant harm to humans in open, ventilated rooms. Until recently, it was not considered to be a harmful component of HG engines. At the same time, the formation of carbon dioxide from fossil carbon fuels leads to an increase in the amount of this gas in the atmosphere. Carbon dioxide is one of the so-called greenhouse gases in the atmosphere. Greenhouse gases are able to trap long-wave heat radiation from the Earth that is not retained by other components of the atmosphere. Greenhouse gases include water vapour, methane, freons and some other compounds. Greenhouse gas molecules perceive energy quanta of long-wave radiation, become excited, and emit radiation themselves. This radiation is directed in all directions, resulting in half of the energy being sent into space and half returning to the Earth.

Greenhouse gases help maintain certain equilibrium temperatures in the Earth's atmosphere and on its surface (for the atmosphere, the average annual surface temperature is +14 °C), which is the result of the heat balance in the Sun–Earth–Space system. Increasing the amount of carbon dioxide leads to an increase in the so-called greenhouse effect, which can affect the planet's climate. At the current rate of growth of the amount of this gas in the atmosphere (and its mass fraction increased from 0,00012 in 1912 to 0,00038 by the beginning of the new century), the average annual temperature of the surface layers of the atmosphere may increase by 2...4 ^{o}C in the next 100...200 years. This is fraught with intensive melting of the planet's glaciers, a rise in sea levels by 6...16 m and related climatic disasters.

One of the negative environmental impacts of transport is smog. This term is widely used to refer to visible air pollution of any kind (not only from transport).

There are three main types of smog:

– crystalline smog (Alaskan type) – a combination of gaseous pollutants, dust particles and ice crystals that occur when fog droplets and steam from heating systems freeze;

– simple smog (London type) – a combination of gaseous pollutants (mainly sulphur dioxide), dust particles and fog droplets;

– photochemical smog (Los Angeles-type, dry) – secondary air pollution resulting from the decomposition of pollutants under the influence of sunlight, especially ultraviolet rays.

The main poisonous component of smog is ozone (O_3). Additional components of smog include carbon monoxide (carbon oxide), nitrogen oxide, acetyl nitrate peroxide, nitric acid and some others. The main source of this type of smog is transport.

All of these substances (except soot) can be fatal if they reach a certain concentration in the air. However, the degree of harmfulness of these substances is different and, accordingly, their permissible concentration in the air is different. Continuous exposure to humans, animals and plants can lead to mutations at the genetic level and hereditary changes in organisms, changing their morphological (external and internal structure) and/or physiological and behavioural characteristics.

The impact of gases and IEDs on plants is caused by the ingress of GHGs both on the surface of plants and into cells (with groundwater). Plants are particularly sensitive to sulphur oxides, nitrogen oxides, and compounds of nitrogen oxides with hydrocarbons.

The impact of exhaust gases from engines on buildings, monuments and other architectural structures is also determined by the deposition of nitrogen and sulphur oxides with water vapour on the surfaces of building elements, as well as the deposition of oily soot particles.

Improvement of work processes and mixing

Increase the intensity of mixture turbulence in the combustion chamber, increase the duration and power of the electric discharge, especially when operating on lean mixtures. This reduces CO emissions on rich mixtures, but can increase NOx emissions when operating at rated loads and on lean mixtures.

Improves fuel atomisation in all operating conditions, including idle, forced idle, partial and transient conditions.

Improves the uniformity of fuel flow through the engine cylinders. This is achieved by using multi-chamber carburettors, fuel injection systems into the air receiver, intake pipes and directly into the engine cylinders.

Reducing the ignition timing reduces NOx emissions, but it also reduces engine efficiency and increases hydrocarbon and CO emissions.

Recirculation of the fuel oil

A small amount of gas (10...12 %) is fed into the intake pipe after the carburettor at the nominal operating mode (according to other sources, up to 20 %). This reduces NO_x emissions by about half by reducing the maximum combustion temperature (according to other sources, by 40...50 %). The reduction is due to a decrease in the speed of the combustion reaction due to the dilution of the combustible mixture with neutral gases and a decrease in the mass of the fresh charge, which, accordingly, reduces the fuel's calorific value. At the same time, the power output is slightly reduced and the engine's efficiency is reduced. CO and CH emissions may increase slightly due to the deterioration of the combustion process. It is believed that recirculation is more profitable at off-nominal modes, although NO_x emissions are already low. Recirculation of HC is more effective for engines with a chamber in the piston than for engines with split chambers.

Fuel and additives

Increasing the cetane number of fuel improves the combustion process, reduces the ignition delay period, reduces the dynamism of the cycle, and harshness of the engine. NO_x emissions are reduced, but smoke and hydrocarbon emissions may increase. The increase in light fractions in the fuel increases its volatility, which means it improves the uniformity of fuel distribution across the combustion chamber. This reduces all types of emissions. The addition of organic-metal additives based on barium, manganese and tetraethyl lead to diesel fuel allows for a several-fold reduction in exhaust smoke and the content of aldehydes and benzo-perenes at higher loads. The smoke emissions of a diesel engine with an undivided combustion chamber are reduced by 40...90 %. The additives do not affect the emission of nitrogen oxides. Most of the barium combines with sulphur contained in the fuel and enters the atmosphere as relatively harmless barium sulphide. Two hypotheses for the additives' effect have been put forward: according to the first one, the formation of soot particles in the combustion process is inhibited; according to the second, the reduction of smoke is achieved by the catalytic effect of additives in the oxidation of soot particles in the process of diffusion combustion, expansion and release.

The use of petrol with a low content of tetraethyl lead can influence the emission of lead compounds and increase the service life of catalytic converters several times.

Converting an engine to gaseous fuel can reduce NO_x emissions by about half and slightly reduce CO emissions. This is due to the fact that gas-fuelled engines can effectively use leaner mixtures that burn at lower temperatures, and the unevenness of the mixture composition across the engine cylinders is reduced.

Neutralization of exhaust gases

Reducing the level of emission of toxic substances in exhaust gases can be achieved due to the neutralization of outgoing gases in special devices – neutralizers. Neutralizers are installed at the gas outlet from engine cylinders and are designed to convert toxic substances into non-toxic ones or to delay (absorb) toxic substances. Accordingly, there are thermal, catalytic and mechanical neutralizers. In thermal and catalytic neutralizers, chemical reactions occur, as a result of which the concentration of gas components of toxic substances decreases.

5.3. Conclusions to the part

An analysis of dangerous and harmful production factors affecting the personnel operating and servicing marine internal combustion engines was carried out. The noise and vibration level from the 6ChN 31.8/33 engine were calculated. Yes, the level of noise and vibration has acceptable values and is 83.03 dB and 79.06 dB, respectively. Analyzed and proposed measures to reduce the level of noise and vibration in the ship's engine room. The nature of the impact of the components found in engine exhaust gases on the human body is considered. An analysis of possible ways of reducing the toxicity of exhaust gases was carried out.

CONCLUSION ON THE WORK

The qualification work on the topic: "***Improving the design of a high-pressure fuel pump for a main marine engine***" contains all the recommended sections and drawings.

The first part considers the possibility of using the 6CHN 31.8/33 engine on a tugboat. It is established that the engine can provide the required operational parameters of the vessel and is well placed in the engine room.

The second part calculates the duty cycle for the nominal operating mode of the 6CHN 31.8/33 marine engine.

The third part analyses possible ways to modernise the diesel fuel system. The design of a high-pressure fuel pump with an electromagnetic valve was carried out. The main geometric and operational parameters of the high-pressure pump were calculated. The strength of the pump elements was also calculated. It was found that the use of a high-pressure fuel pump with an electromagnetic valve of the proposed design provided a decrease in specific effective fuel consumption by 6 g/(kWh), an increase in diesel power by 352 kW, and an improvement in the environmental performance of the 6CHN 31.5/33 engine.

The fourth part determines the economic effect of improving the design of the high-pressure fuel pump of the 6ChN 31.8/33 main ship engine. The annual economic effect for the shipowner from the use of the upgraded 6CHN 31.8/33 engine, compared to the base diesel engine, is UAH 30773.83. The modernisation of the basic 6CHN 31.8/33 engine will pay off in 0.65 years.

Part 5 analyses the hazardous and harmful production factors affecting the personnel operating and maintaining shipboard ICEs. The level of noise and vibration from the 6CHN 31.8/33 engine was calculated. The noise and vibration levels were found to be within permissible limits and amounted to 83.03 dB and 79.06 dB, respectively. The paper analyses and proposes measures to reduce noise and vibration levels in the ship's engine room. The nature of the impact of components in engine exhaust gases on the human body is considered. Possible ways to reduce the toxicity of exhaust gases are analysed.

Summarising the results of the work, a positive conclusion can be made about the relevance and feasibility of the modernisation of the fuel system of the 6CHN 31.8/33 marine engine.

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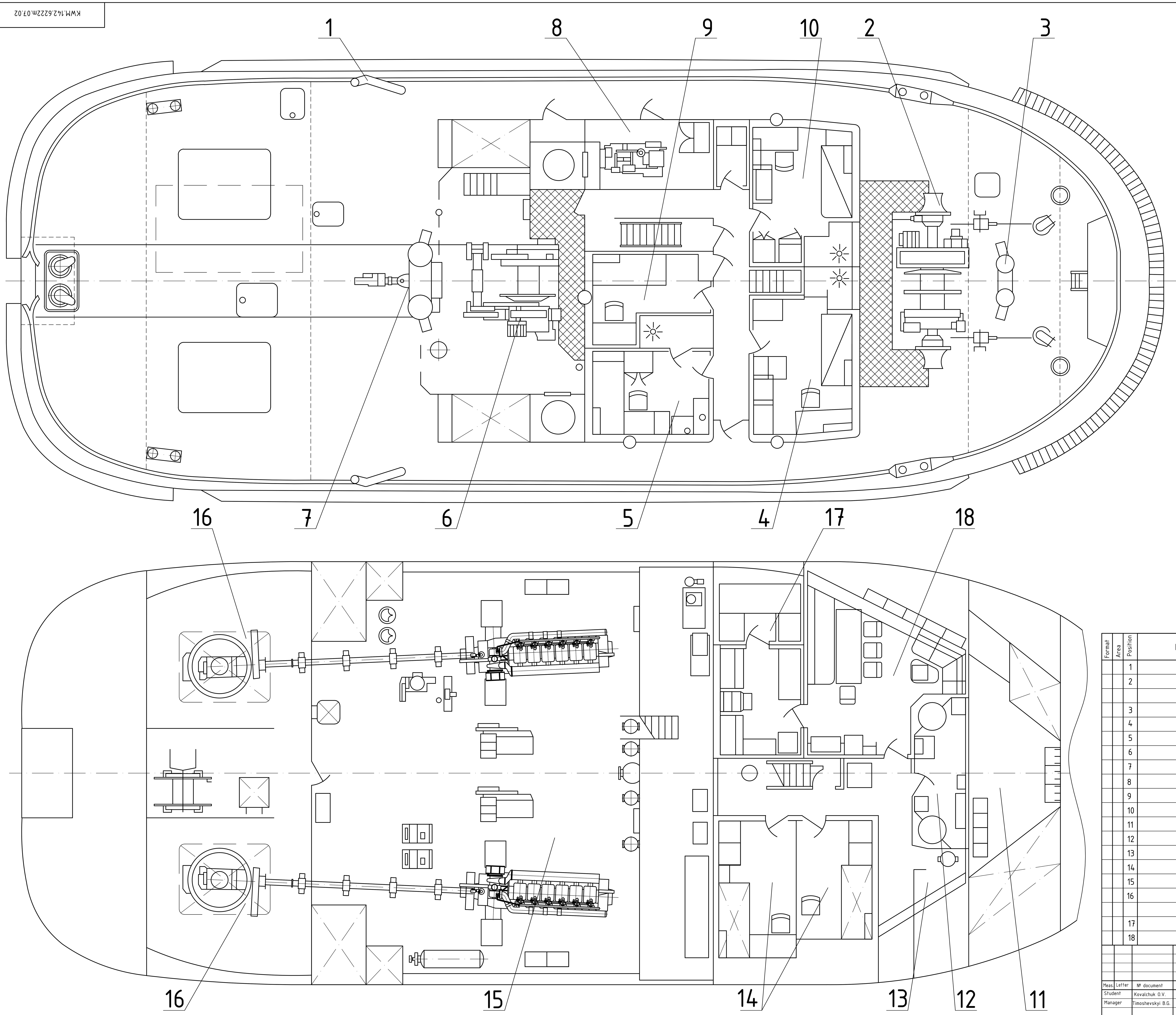
KWM.142.6222m.07.01

Maximum length, m	30.87
Length along the KVL, m	29.25
Width, m	11.6
Board height, m	5.45
Draft along the beam, m	3.7
Overall draft, m	4.3
Displacement, t	680
Speed, kn.	14,1

Format	Area	Position	Designation	Name	Number	Note
		1		Fire monitor		
		2		Hydraulic crane		
		3		Another boat		
		4		Gasket table		
		5		Control panel		
		6		Onboard biting		
		7		Bow anchor and towing		
				escort winch		
		8		Nose biting		

Meas	Letter	№ document	Caption	Date	KWM.142.6222m.07.01			
Student		Kovalchuk O.V.			General drawing of the tugboat	Let.	Mass	Scale
Manager		Timoshevskyi B.G.				Letter	Letters	
Head of Depart.		Gogorenko O.A.				National University of Shipbuilding Admiral Makarov		



Format	Area	Position	Designation	Name	Number	Note
		1		Onboard Biteng		
		2		Bow anchor and towing		
				escort wincha		
		3		Nose Biteng		
		4		Captain's cabin		
		5		XO's cabin		
		6		Stern towing winch		
		7		Towing hook and stern hook		
		8		ADG premises		
		9		Cabin of the 2nd mechanic		
		10		Senior mechanic's cabin		
		11		Storage room		
		12		Laundry		
		13		Bathroom		
		14		2-berth crew cabins		
		15		Engine room		
		16		The premises of the		
				Supreme Council		
		17		Galley		
		18		Salon-dining room		

KWM.142.6222m.07.02

Meas.	Letter	№ document	Caption	Date
Student		Kovalchuk O.V.		
Manager		Timoshevskyi B.G.		
Head of Depart.		Gogorenko O.A.		

Engine room

Let.	Mass	Scale

Letter | Letters

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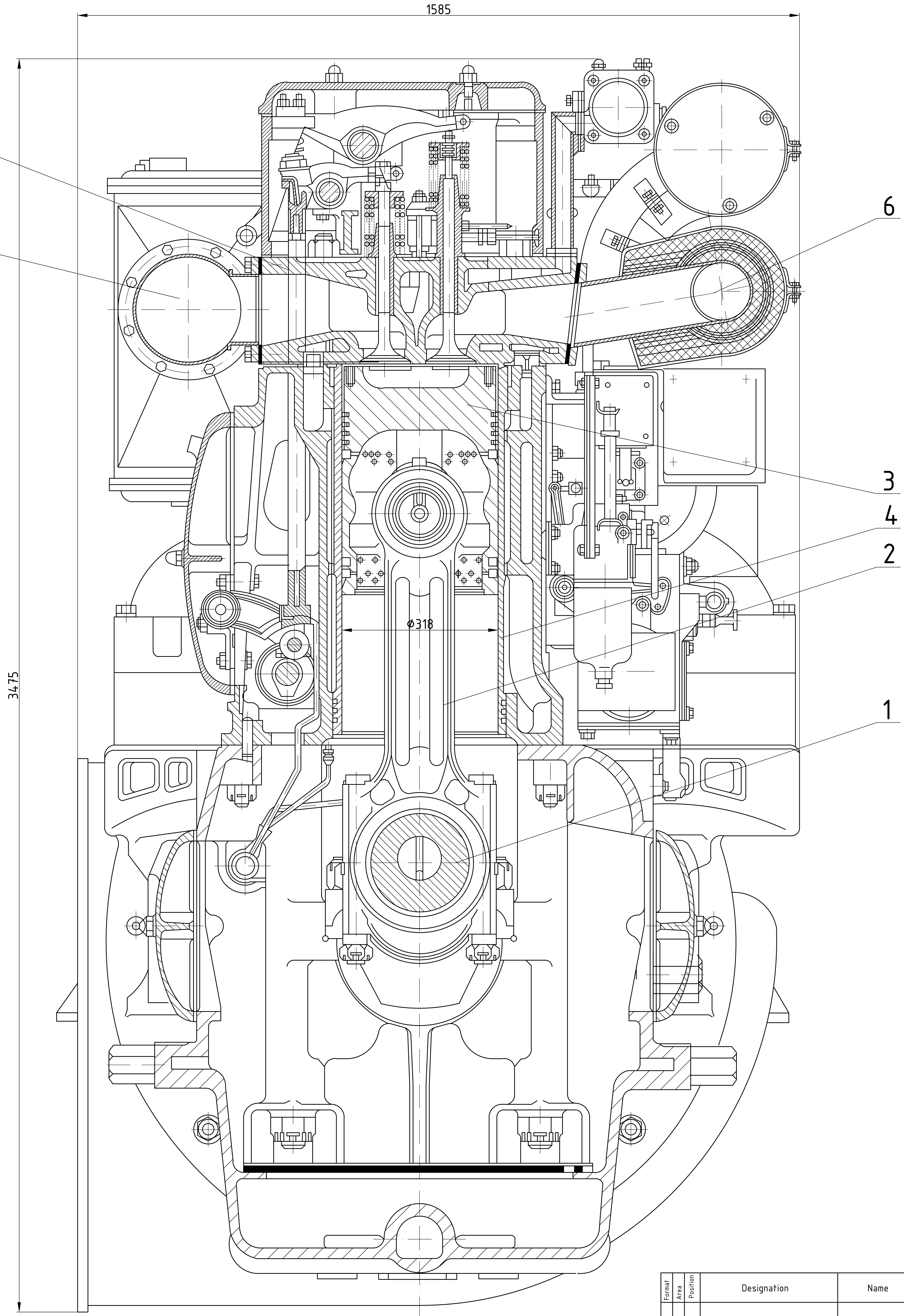
KWM.142.6222m.07.03



Format	Area	Position	Designation	Name	Number	Note
		1		Crankshaft		
		2		Connecting rod		
		3		Piston		
		4		Slave cylinder bushing		
		5		Cylinder cover		
		6		Exhaust manifold		
		7		Inlet manifold		

					KWM.142.6222m.07.03			
Meas.	Letter	№ document	Caption	Date	Cross section of the 6CHN31,8/33 engine	Let.	Mass	Scale
Student		Kovalchuk O.V.						1:4
Manager		Timoshevskyi B.G.				Letter	Letters	
Head of Depart.		Gogorenko O.A.				National University of Shipbuilding Admiral Makarov		

KWM.142.6222m.07.04

Meas.	Letter	№ document	Caption	Date	
Student		Kovalchuk O.V.			
Manager		Timoshevskyi B.G.			
Head of Depart.		Gogorenko O.A.			

KWM.142.6222m.07.04

Longitudinal section of the 6CHN 31,8/33 engine

Let.	Mass	Scale
		1:5
Letter	Letters	

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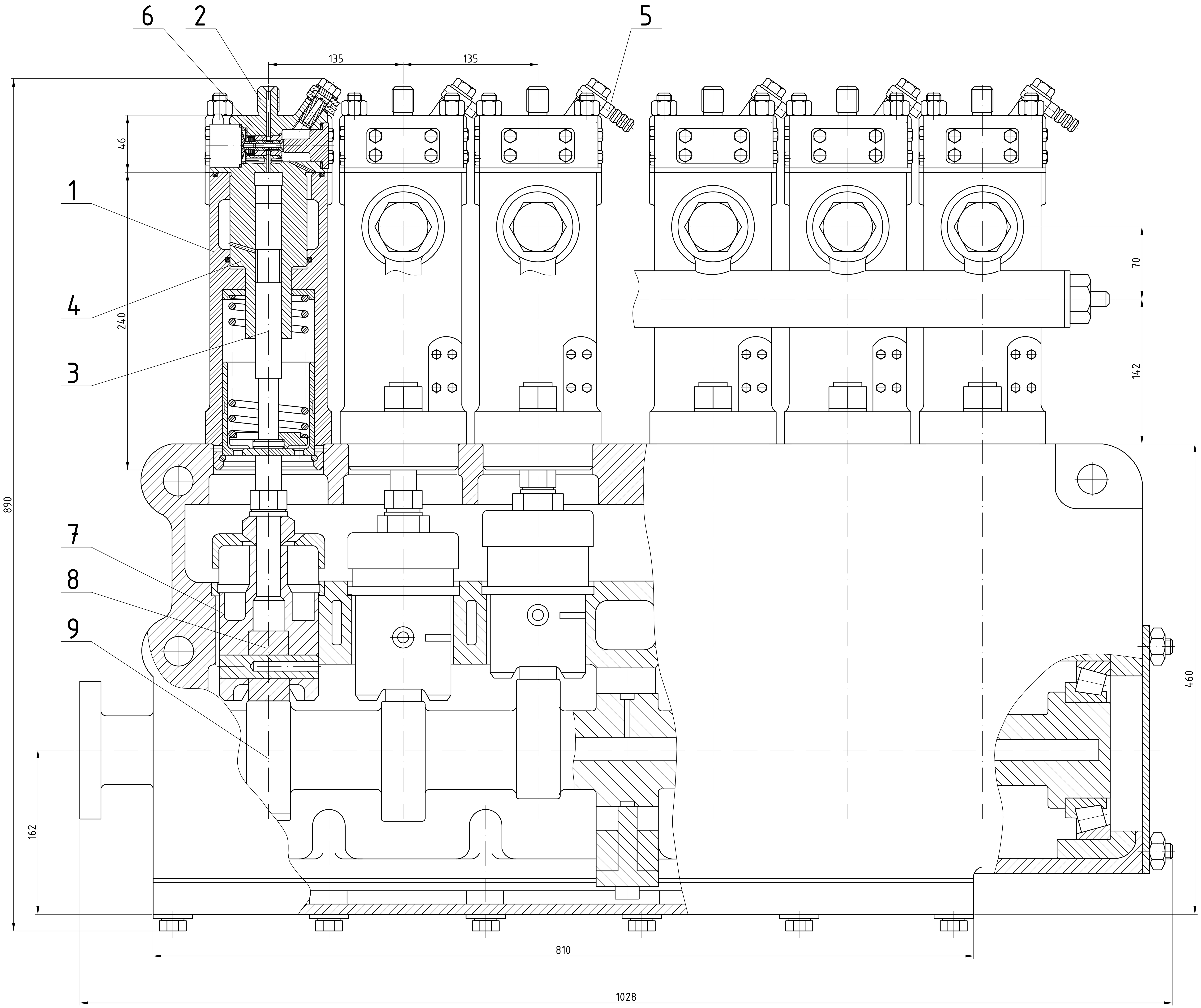
KWM.142.6222m.07.05



Format	Area	Position	Designation	Name	Number	Note
		1		Pump section casing		
		2		High pressure connection		
		3		Plunger		
		4		Plunger bushing		
		5		Fitting		
		6		Electrically controlled valve		
		7		Push rod body		
		8		Roller		
		9		Cam shaft		

KWM.142.6222m.07.05

Meas.	Letter	№ document	Caption	Date
Student		Kovalchuk O.V.		
Manager		Timoshevskyi B.G.		
Head of Depart.		Gogorenko O.A.		

The high-pressure pump

Let.	Mass	Scale
		1:2

Letter | Letters

National University of Shipbuilding
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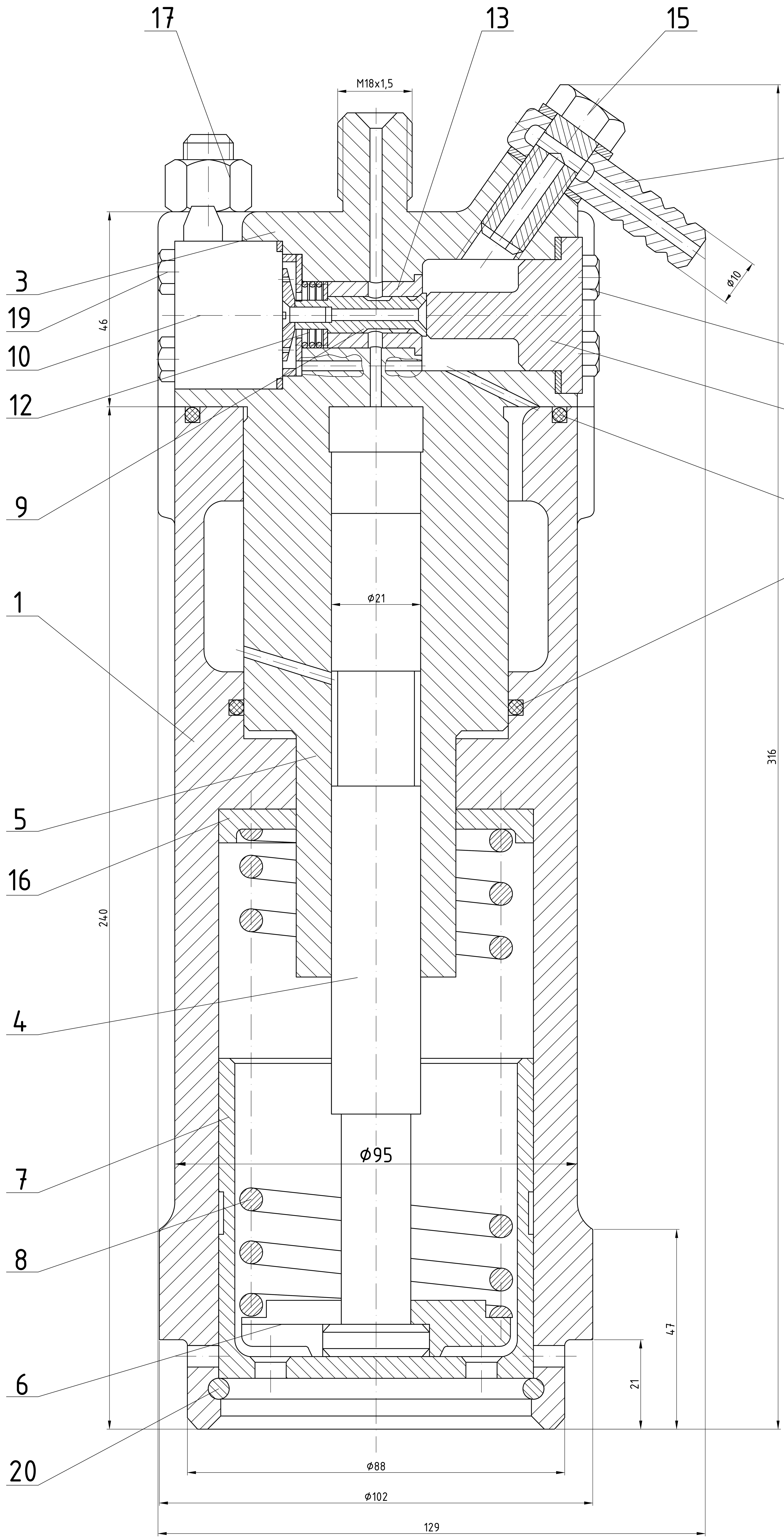
KWM.142.6222m.07.06



1. From the beginning of the pump assembly, the pump elements pos. 4, 5, drain valve, valve bushing pos. 13, cover item no. 11, section cover pos. 3, spring 12 and washer should be thoroughly washed in diesel fuel; other parts in kerosene and cleaning solution.
2. Tighten the nuts item 17 with a torque of 30±3 kgcm.
3. Tighten the bolts pos. 19 shall be tightened with a torque of 15±3 kgf.
4. Tighten the bolts pos. 18 should be tightened with a torque of 20±3 kgf.
5. After tightening the nuts pos. 17, the plunger pos. 4 should move freely in any direction.
6. Pressurise the suction channel with diesel fuel at a pressure of 6...8 kgf/cm^2 for 2 minutes. Fuel leakage in the connections is unacceptable.
7. During assembly, to prevent contamination of the internal channels, put protective parts on the fittings.
8. When assembling, set the valve stroke pos. 9 equal to 0.3 mm using a gasket.
9. After assembly, check the functioning of the drain mechanism by applying 24 V operating voltage to the solenoid pos. 10 with an operating voltage of 24 V. A click should be heard from the valve shim.
11. The colour of the paint is cream, corresponding to the class IY-III group of OMZ operation requirements.
12. Marking is made on the plate attached to the pump casing.
13. Preservation should ensure the safety of the product for 5 years.

Format	Area	Position	Designation	Name	Number	Note
		1		Pump casing		
		2		High pressure connection		
		3		Section cover		
		4		Plunger		
		5		Plunger bushing		
		6		Lower spring plate plunger		
		7		Guide cup		
		8		Plunger spring		
		9		Electrically controlled valve		
		10		Electromagnet		
		11		Cover		
		12		Valve spring		
		13		Bushing		
		14		Fitting		
		15		Bolt of the fitting		
		16		Top plate		
		17		Nut		
		18		Bolt		
		19		Bolt		
		20		Locking ring		
		21		Sealing ring		
		22		Sealing ring		

KWM.142.6222m.07.06

Meas.	Letter	№ document	Caption	Date
Student		Kovalchuk O.V.		
Manager		Timoshevskyi B.G.		
Head of Depart.		Gogorenko O.A.		

The section of the high-pressure pump

Let.	Mass	Scale
		2:1
Letter	Letters	

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