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
____
« ___» __________ 2025 p.

QUALIFICATION WORK

MODERNIZATION OF THE FUEL SYSTEM OF A LOW-SPEED MARINE ENGINE BY USING A HYDRAULICALLY DRIVEN SYSTEM

Speciality 142 – Power engineering

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

Head of work Boris TYMOSHEVSKY

Education applicant Vyacheslav BORISENKO

Mykolaiv 2025

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

Vyacheslav Oleksandrovych Borysenko
(surname, first name, patronymic)

1. Work topic *Modernisation of the fuel system of a low-speed marine engine by using a hydraulically driven system*

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 Ne = 21,600 kW, rotational speed n = 104 rpm. The engine is installed in the engine room of a container ship; the fuel system is being modernised, namely, a hydraulically driven fuel system is being used.*

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. Modernisation of the internal combustion engine fuel system with the development of a hydraulically driven system diagram. Calculation of the high-pressure fuel pump and injector elements. 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 view of the vessel and MV. 2. General layout of equipment in the MV of the vessel. 3. Cross-section of the modernised 6DKRN 80/230 engine. 4. Diagram of the control oil system for the hydraulic drive of the high-pressure fuel pump. 5. Hydraulically driven fuel 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.	*Modernisation of the internal combustion engine fuel system with the*		
	development of a hydraulically driven system diagram. Calculation of t		
	he high-pressure fuel pump and injector elements.		
4.	*Economic justification of technical solutions*		
5.	*Analysis and development of labour and environmental protection*		
	measures for engine applications		

Education applicant ______ (signature) *Vyacheslav BORISENKO* (surname and initials)

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

АНОТАЦІЯ

Кваліфікаційна робота на тему: «***Модернізація паливної системи малообертового суднового двигуна шляхом застосування системи з гідравлічним приводом»*** присвячена підвищенню потужності малообертового суднового двигуна 6ДКРН 80/230 виробництва фірми MAN B&W та зниженню питомої ефективної витрати палива за рахунок модернізації його паливної системи

Представлено короткий опис та основні характеристики судна і малообертового двигуна 6ДКРН 80/230, а також виконано розрахунок робочого циклу з модернізованою паливною системою для номінального режиму роботи.

Виконано дослідження можливих шляхів модернізації паливної системи малообертового суднового дизеля MAN B&W 6ДКРН 80/230 з метою забезпечення підвищення потужності та зменшення витрати палива. Виконано розробку схеми паливної системи з гідравлічним приводом паливного насосу високого тиску, а також розраховано її елементи.

Проведено техніко-економічне обґрунтування модернізації паливної системи малообертового суднового дизеля MAN B&W 6ДКРН 80/230.

Розроблено основні вимоги з охорони праці при здійснені ремонту та обслуговування суднового двигуна. Виконано аналіз шкідливої дії шуму та вібрації, а також визначено їх рівень.

Кваліфікаційна робота виконана англійською мовою та містить 56 аркушів розрахунково-пояснювальної записки та п’яти креслень формату А1.

Ключові слова: дизельний двигун; робочий цикл; паливна система; насос високого тиску; ефективні показники; гідравлічний привід.

ABSTRACTS

Qualification work on the topic: ***«Modernisation of the fuel system of a low-speed marine engine through the use of a hydraulically driven system»*** is devoted to increasing the power of the 6DKRN 80/230 low-speed marine engine manufactured by MAN B& W and reducing its specific effective fuel consumption by modernising its fuel system.

A brief description and main characteristics of the vessel and the low-speed engine 6DKRN 80/230 are presented, and the working cycle with the modernised fuel system for nominal operation is calculated.

A study of possible ways to modernise the fuel system of the low-speed MAN B&W 6DKRN 80/230 marine diesel engine was carried out with the aim of increasing power and reducing fuel consumption. A fuel system diagram with a hydraulic drive for a high-pressure fuel pump was developed, and its elements were calculated.

A technical and economic justification for the modernisation of the fuel system of the low-speed MAN B&W 6ДКРН 80/230 marine diesel engine was carried out.

The basic occupational safety requirements for the repair and maintenance of the marine engine were developed. An analysis of the harmful effects of noise and vibration was performed, and their levels were determined.

The thesis is written in English and consists of 56 pages of explanatory notes and five A1 drawings.

Key words: diesel engine; working cycle; fuel system; high pressure pump; efficiency indicators; hydraulic drive.

CONTENT

INTRODUCTION

Improving the operational and environmental performance of diesel engines involves a wide range of tasks, the solution of which is closely related to the operation of the fuel system. Among the main problems are ensuring optimal speed characteristics and fuel injection parameters, stabilizing the cycle fuel supply, and improving the uniformity of its supply in all sections of the high-pressure fuel pump, especially during mode changes. It is also important to eliminate the phenomenon of additional fuel injection and identify the factors that cause it to occur when upgrading existing engines or creating new ones and fuel systems.

Today, considerable attention is paid to researching the operation of diesel fuel systems in unstable modes. The results of the studies show that under such conditions, the unevenness of the cycle fuel supply and the injection advance angle between cylinders and individual cycles increases significantly, which leads to periodic loads on the engine and a reduction in its service life. In addition, there is a noticeable discrepancy between the cyclic fuel supply in unstable and similar stable modes at the same crankshaft speed and fuel rail position, which complicates the process of regulating engine operation.

When forcing diesel engines, the effect of these deviations becomes even more noticeable, so it's super important to study how different types of fuel systems work in unstable modes.

With the increasing use of multi-fuel internal combustion engines, there is a need for similar experimental and theoretical studies of fuel supply processes when operating on different types of fuel.

Developers of modern diesel fuel systems are also actively studying the possibilities of purposeful changes in their characteristics by improving the design of high-pressure fuel pumps and their components (in particular, delivery valves). However, design changes do not always have the same effect on the characteristics of the injection system, and the nature of this effect can vary. Therefore, research in this area remains extremely relevant and in demand.

PART 1
DESCRIPTION OF THE DESIGN OF THE MAN B&W 6DKRN 80/230 LOW-SPEED MARINE ENGINE

1.1. General characteristics of the vessel and power plant based on the 6DKRN 80/230 engine

The low-speed 6DKRN 80/230 engine is installed on a container ship with unlimited navigation area (according to the Shipping Register). The minimum outside air temperature in the vessel's navigation areas is approximately –25 °C. In terms of its architectural and structural design, the vessel is a self-propelled, steel, single-deck vessel with a stern-mounted power plant. The vessel also has living quarters, a control station, double bottom and sides, one longitudinal bulkhead in the diametrical plane of the cargo section and a transom stern with a minimum freeboard. The bow is bulbous with a bulbousness coefficient of 7.

Main characteristics of the container ship:

Maximum length of the ship, m…………………………………….……..….180

Length between perpendiculars, m……………………….………………..…...172

Ship width, m…………………………………………………….……..……….28

Ship side height, m ……………………………..……………………..……..16.85

Calculated draught of the vessel, m……………………..……………..9.20

Maximum draught of the vessel, m ……………………………………..…..11.83

Height of the vessel from its main line, m……………………………......39.70

Minimum theoretical height between deck spaces

in the ship's living and service areas, m…..…………………….……..….2.70

Height of the ship's double bottom at position 17–77 m in DP, m.…......2.59

Value Height of the ship's double bottom at 77–244 m, m…………..….1.98

Deflection of the ship's upper deck beams, m …………………………...0.15

The vessel's speed during testing is not less than 14.2 knots in deep water with a draught of 9.2 m, with a flat keel, sea waves no higher than 2 points, wind force no more than 3 points on the Beaufort scale, propeller rotation at a frequency of 104 min-1 and the corresponding effective shaft power.

In all operational cases of ship loading, stability complies with the basic rules of specifications and construction of seagoing ships of unlimited navigation area. The unsinkability of the ship is ensured under conditions of loading with all types of cargo to a draught of 9.2 m and when any one compartment (except MB) is flooded.

The ship's rolling period for all loading conditions is maintained within 12–14 seconds. The ship's high manoeuvrability (particularly important for container ships) is ensured by the installation of a tunnel-type steering device in its bow, which provides a thrust of 58.5 kN.

Basic requirements for ship power plants:

– the power plant of a technical fleet vessel must provide a speed of at least 7 knots;

– the ship's power plant must be capable of reversing to ensure the necessary manoeuvrability;

– the ship's power plant must provide at least 70 per cent of the calculated forward rotation speed within 30 minutes when reversing;

– the main and auxiliary mechanisms must be located in the engine room so that there is free access to the exit ladders from their control stations and maintenance areas;

– the width of the passages between the equipment in the engine room along its entire length shall be not less than 0.6 m;

– the width of the gangways at the exit and the width of the hatches at the exits shall be not less than 0.6 m;

– the location of various mechanisms, equipment, fittings and pipelines must ensure free access for maintenance and emergency repairs;

– main mechanisms and their transmissions, thrust bearings of shafting must be fully or partially bolted to the ship's foundations;

– the engine room, shaft tunnel and piping tunnels shall have two exits providing access to the deck to lifeboats or rafts;

– in order to ensure safe navigation and control of the ship, main lighting fixtures shall be installed in all rooms;

– all ventilation ducts shall be protected against corrosion;

– all vertical ventilation ducts and shafts shall pass through watertight decks of the ship within a single watertight compartment;

– all ventilation ducts used to remove explosive, flammable vapours and gases shall be gas-tight and shall not be connected to the ducts of other ship compartments.

The Register's main requirements for main ship engine parameters:

– main ship engines must be capable of stable operation with at least 10% overload of the rated power for at least one hour;

– the minimum permissible stable crankshaft speed of the main engine must not exceed 30% of the rated speed;

– the receiver and air pipes of the main engine must be equipped with a fire detection and alarm system;

– the main engine must have a regulator that is adjusted to ensure that the rotation speed does not exceed the rated speed by more than 15%.

1.2. Description of the design of the low-speed marine engine 6DKRN 80/230

The 6DKRN 80/230 low-speed marine engine shown in Fig. 1.1 is a two-stroke, six-cylinder, reversible single-row engine with vertically arranged working cylinders. The piston diameter is 800 mm, and the piston stroke is 2300 mm. The base engine has a rated power of 18,720 kW at 104 rpm. The air pressure after the compressor is 0.38 MPa, and the specific effective fuel consumption is 173 g/(kW·h).

The foundation frame of the 6DKRN 80/230 engine has a simplified box shape and is made of solid welded steel (Fig. 1.2). The foundation frame of the engine is attached to the ship's hull foundation with bolts with steel wedges. The cross supports of the foundation frame are cast from steel and have holes for the anchor connections. The engine frame bearing inserts are thin-walled steel and filled with babbit.

To drain oil from the crankcase into the circulation oil tank located under the engine in the double bottom of the ship's hull, a special hole with a mesh is provided in the stern part of the foundation frame tray. Also, in the stern of the 6DKRN 80/230 engine, there is a drive compartment with a thrust bearing built into it.

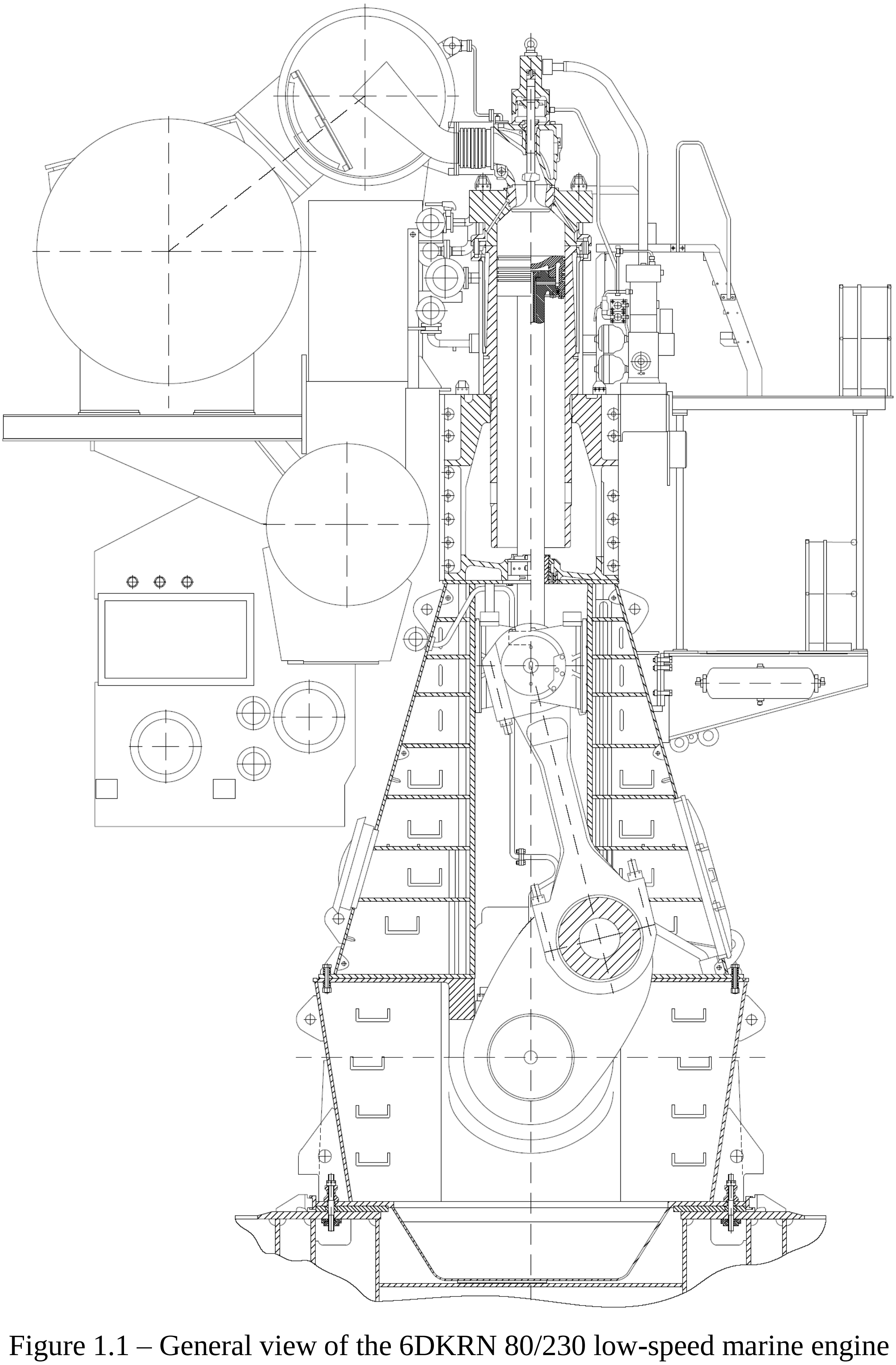

Figure 1.1 – General view of the 6DKRN 80/230 low-speed marine engine

The frame of the 6DKRN 80/230 engine is made of steel and is fully welded, with access hatches to each cylinder compartment and the drive compartment on the control side. The drive compartment also has an exit on the opposite side. Five crankcase relief valves are located in the upper part of the frame on the engine exhaust side and one is located at the front end. Each working cylinder of the engine has four steel crosshead guides welded to the frame structure.

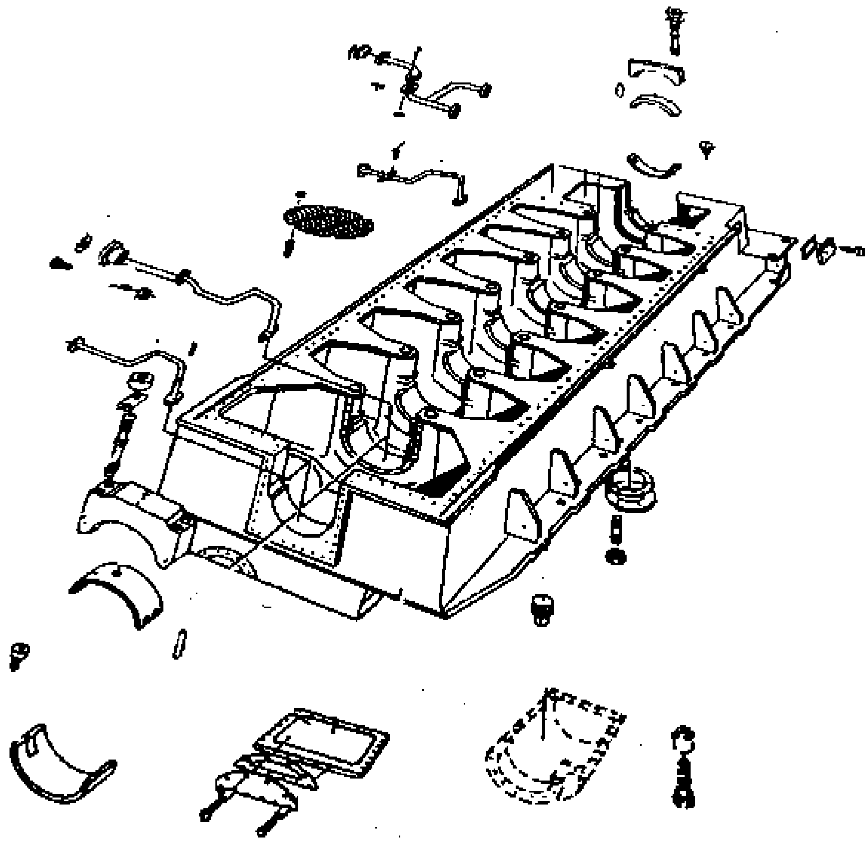

Figure 1.2 – Engine foundation frame 6DKRN 80/230

The engine cylinder block 6DKRN 80/230 is assembled from separate cast iron blocks, which are assembled into a single monoblock using stud bolts. Each engine cylinder block has a mating cylinder liner pressed into it, which consists of two parts with a joint located above the upper level of the block. Both parts of the working cylinder liner are made of modified cast iron. The upper rim of the lower part of the working cylinder liner has holes for eight cylinder lubrication fittings. The upper part of the working cylinder liner is closed from the outside by a hollow cast iron jacket to ensure cooling. In the engine combustion chamber area, the working cylinder sleeve has oblique holes for the passage of coolant. The working cylinder sleeve is sealed by four rubber rings in its lower part and two rubber rings (one ring at the top and bottom of the jacket) in its upper part in the cooling jacket area.

The sealing of the seating between the working cylinder sleeve and the block is ensured by lapping these seats (without using gaskets), and between the working cylinder sleeve and the cover by means of a soft steel sealing ring. Coolant is transferred from the engine block to the cooling jacket via four transfer pipes, and from the jacket to the working cylinder cover via the same transfer pipes.

The engine working cylinder sleeve is solid and made of modified cast iron (Fig. 1.3). The lower part of the working cylinder sleeve has thirty blow-off windows.

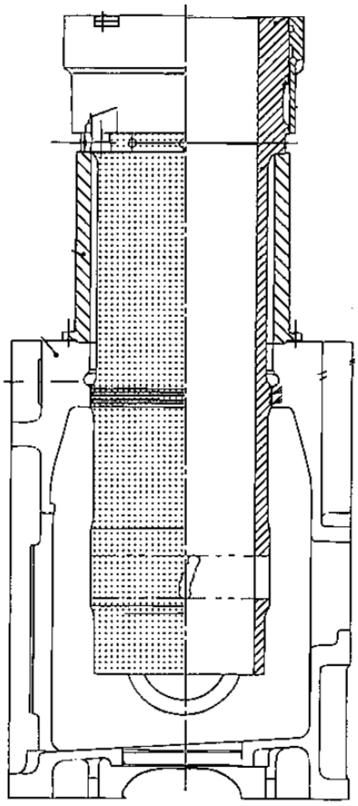

Figure 1.3 – Working cylinder bushing for 6DKRN 80/230 engine

The massive flange of the working cylinder bushing has holes for cooling fluid to flow from the crankcase to the cylinder cover. The lubrication of the engine cylinder sleeve is provided by two rows of holes located in the upper part of the sleeve for the passage of cylinder oil. On the mirror side, each outlet hole has oil distribution grooves.

The engine working cylinder cover is cast from steel, cap-type, with holes for coolant to pass through (Fig. 1.4). The working cylinder cover houses two fuel injectors, an exhaust valve, a safety valve and an indicator tap. The working cylinder cover is attached to the engine cylinder block by means of a hydraulic ring and sixteen pins that pass through the hollow cooling jacket of the upper part of the sleeve.

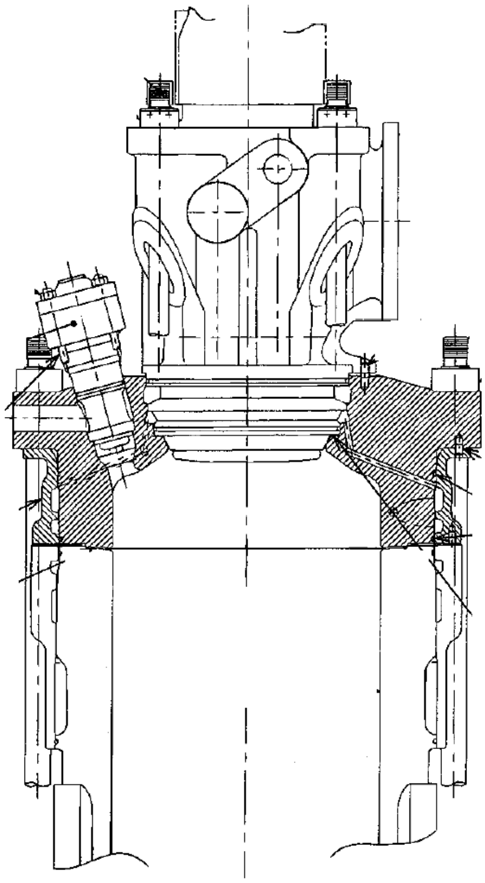

Figure 1.4 – Engine working cylinder cover 6DKRN 80/230

The engine piston 6DKRN 80/230, shown in Fig. 1.5, has a steel head made of chrome-molybdenum steel and a shortened cast iron skirt. The piston has four compression rings and two red copper running-in bands located in the skirt. The piston is cooled by oil, which is supplied and removed through an opening in the crosshead crossbar and a steel tube located inside the rod.

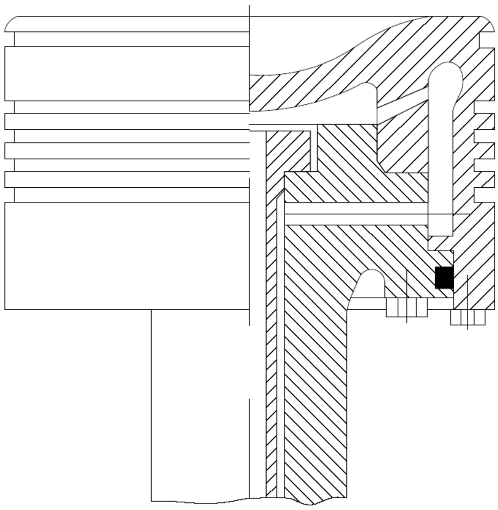

Figure 1.5 – Piston of the 6DKRN 80/230 engine

The crosshead of the 6DKRN 80/230 engine is double-sided and has four babbit-coated sliders (Fig. 1.6). The crosshead crossbar is forged steel and has drilled channels for oil passage. The piston rod bearing, the elbow of the lubrication supply telescope and the engine piston cooling oil drain pipe are attached to the crosshead crossbar with threaded connections.

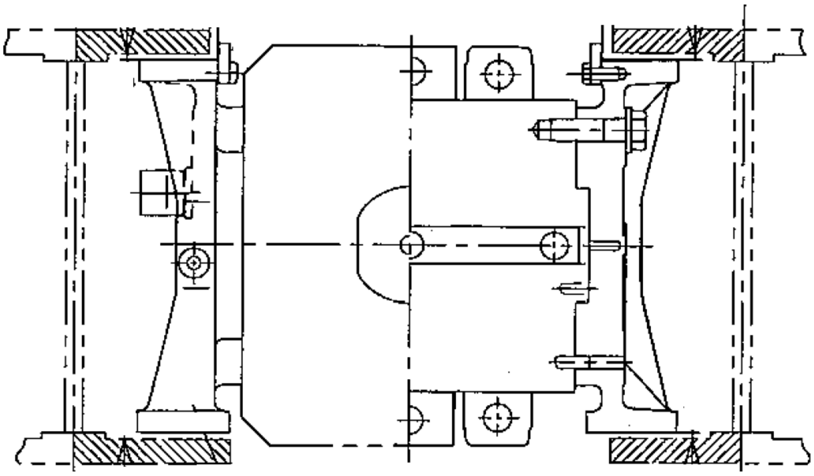

Figure 1.6 – Crankshaft for 6DKRN 80/230 engine

The crankshaft of the 6DKRN 80/230 engine is a steel semi-assembled crankshaft with cast cranks and pressed-in frame journals (Fig. 1.7). The piston of the axial vibration damper is located on the shaft at the front of the engine. A single-row sprocket is also located on this side to drive the auxiliary shafts with balancing balancers. A double-row sprocket for driving the camshaft is attached to the crankshaft at the rear of the engine. A thrust comb with a corresponding thrust bearing is located in the drive compartment.

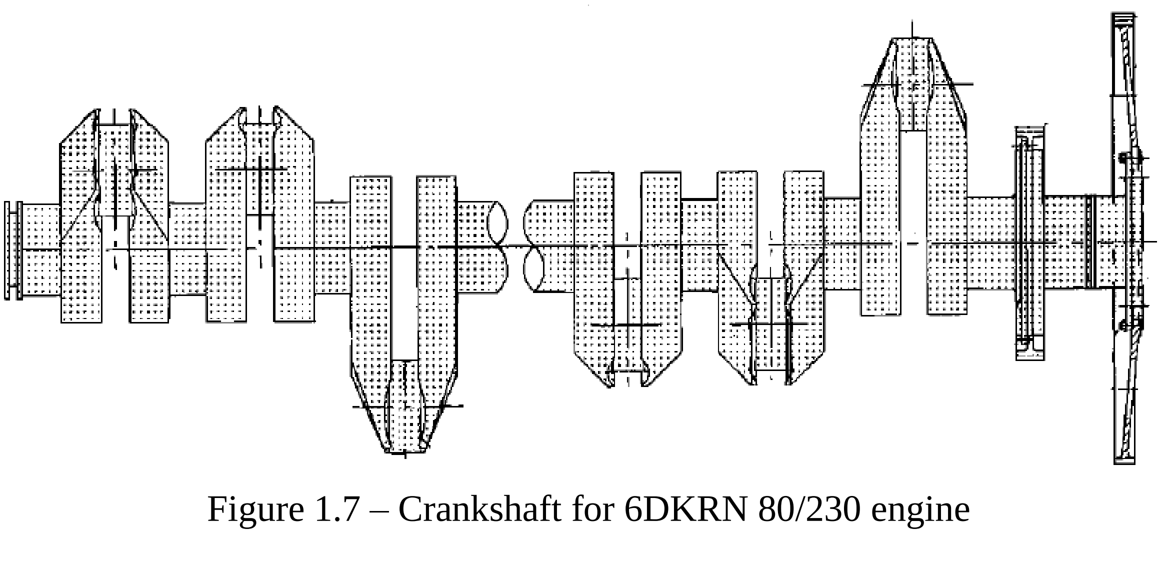

Figure 1.7 – Crankshaft for 6DKRN 80/230 engine

The connecting rod of the 6DKRN 80/230 engine is made of cast steel, followed by forging and machining. The upper head of the connecting rod is of a non-forked type, while the upper and lower heads are integral (Fig. 1.8). The main and connecting rod bearing shells are thin-walled steel with babbit filling. Inside the connecting rod pin there is a bore for oil supply from the main to the connecting rod bearing. The connecting rod pin is relatively short, which reduces the overall height of the engine.

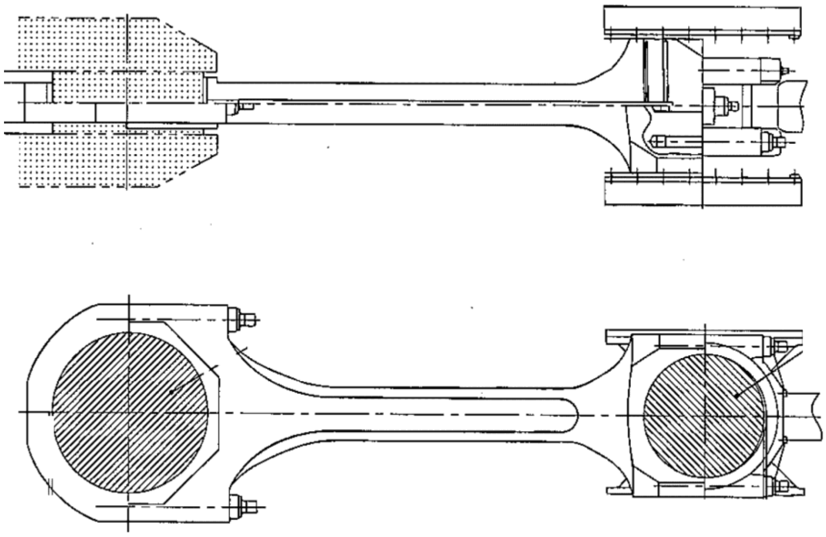

Figure 1.8 – Engine connecting rod 6DKRN 80/230

The engine camshaft is driven by a double-row four-inch chain (Fig. 1.9). Two intermediate sprockets located on the camshaft are used to accommodate balance weights for balancing the moments of second-order inertial forces. The cylinder oil lubricator shaft and the rotation speed regulator are driven by the engine camshaft. The air distributor shaft is attached to the rear end of the engine's camshaft. The fuel and gas distribution cams, as well as the connecting flanges of the camshaft parts, are mounted using hot pressing.

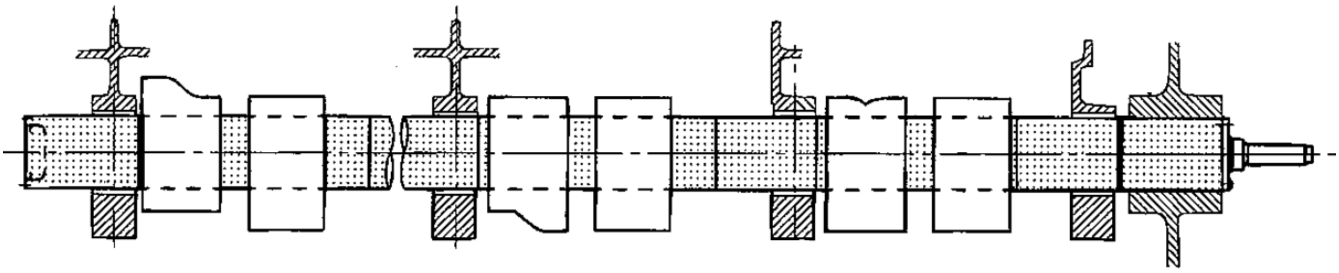

Figure 1.9 – Camshaft of the 6DKRN 80/230 engine

The 6DKRN 80/230 engine has a classic starting system that includes a main starting valve and starting cylinder valves, as well as a slide valve air distributor. The 6DKRN 80/230 engine is reversed by reversing the air distributor and high-pressure fuel pump pushers using actuators located on each pump.

The exhaust valve of the 6DKRN 80/230 engine has a cast iron body, a spindle with an impeller to ensure gas flow rotation, and a cooled seat (Fig. 1.10). The coolant flows through holes in the cylinder cover and through holes in the seat close to the seating ring. The coolant then flows into the cooling cavity of the valve body and exits from the top of the body into the drain pipe. The seating rings of the spindle and valve seat are surfaced with stellite. The valve is opened by a hydraulic piston and closed by a pneumatic piston. The exhaust valve is located in the centre of the engine's working cylinder cover and is attached with four studs and nuts.

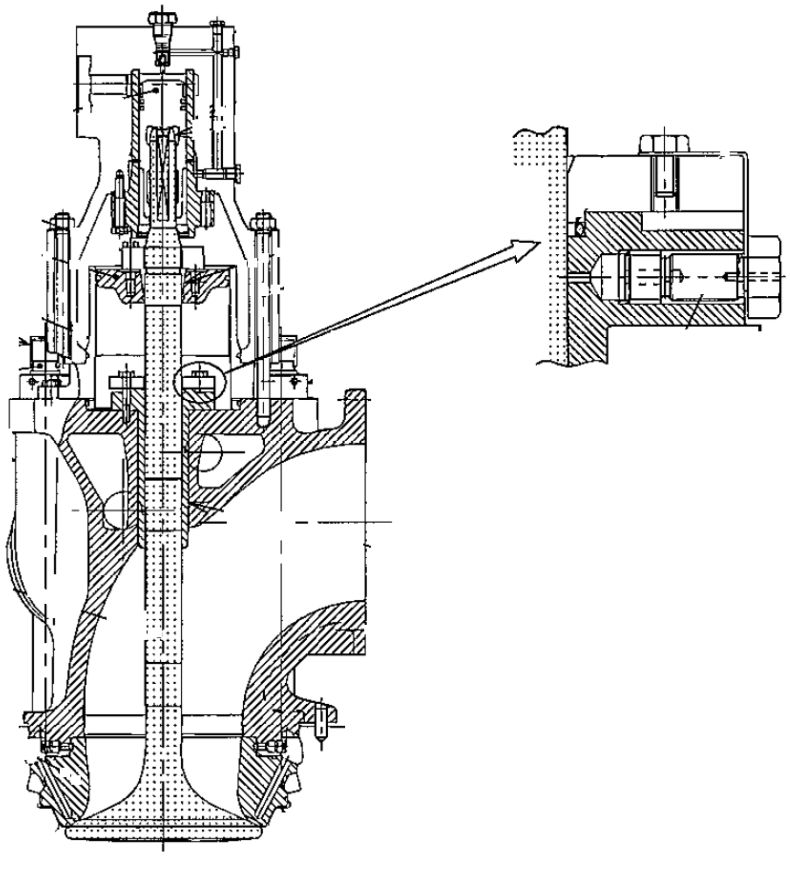

Figure 1.10 – Exhaust valve of the 6DKRN 80/230 engine

The anchor bolts of the 6DKRN 80/230 engine are made of steel and are matched (Fig. 1.11). The bolts consist of two parts and hold the engine block and foundation frame together. The anchor bolt nuts are tightened hydraulically.

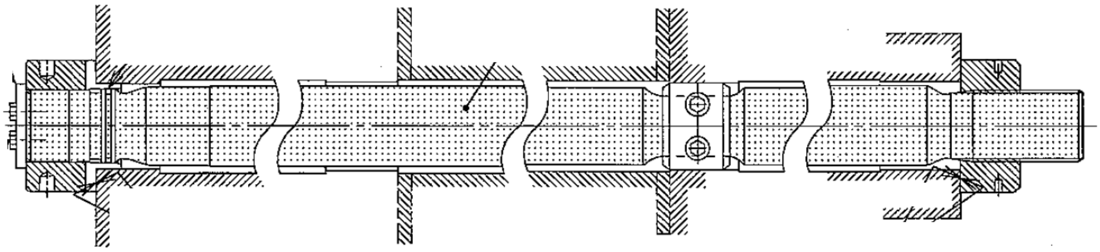

Figure 1.11 – Anchor bolt of the 6DKRN 80/230 engine

1.3. Conclusion to the part

The low-speed two-stroke engine 6DKRN 80/230 manufactured by MAN B&W meets all requirements for operation on a ship as the main engine. The cross-section of the engine and the general view of the engine room with the equipment located therein are presented in the A1 format drawings of the graphic part of the project.

PART 2

CALCULATION OF THE DUTY CYCLE FOR THE NOMINAL OPERATING MODE OF THE MAN B&W 6DKRN 80/230 LOW-SPEED MARINE ENGINE

2.1. Basic requirements for the modernised engine MAN B&W 6DKRN 80/230

In accordance with the task set by the Department of Internal Combustion Engines, Installations and Technical Operation of the Admiral Makarov National University of Shipbuilding, a two-stroke marine engine of the 6DKRN 80/230 type manufactured by MAN B&W is being modernised. The modernisation of the engine should increase its power to 21,600 kW at 104 rpm and reduce specific fuel consumption. The modernised engine with an improved fuel system has different operating parameters from the engine selected as a prototype, which necessitates modelling the operating cycle and determining new performance indicators. To ensure the required power and efficiency, it is necessary to analyse the impact of the main parameters of the operating cycle on these indicators in order to identify the most suitable ones for modification for this type of engine.

The reliability, performance and durability of the engine are determined at the design stage, when the main operating and strength parameters are calculated. At the design or modernisation stage, it is necessary to clearly take into account the specific operating conditions, maintenance and repair requirements. When designing a new engine or modernising an existing one, it is necessary to ensure easy access to the main components and parts, which will simplify repair and maintenance work. Special attention should also be paid to ensuring environmental and sanitary conditions for the personnel who will operate and maintain the engine.

Modernising the engine by changing the fuel system will also reduce the toxicity of exhaust gases, which is undoubtedly a very important factor for modern engines.

2.2. Parameters of main ship engines-analogues

The main low-speed ship engine 6DKRN 80/230 with a rated power of 18,720 kW at a crankshaft speed of 104 rpm was selected as the basis for research and modernisation. Table 2.1 shows the main characteristics of modern engines of similar size and manufacture from different countries around the world. The data presented in the table allows us to assess the degree of effectiveness of the modernisation carried out.

Table 2.1 – Main parameters of analogues of the designed internal combustion engine

Diesel name	Diesel cylinder power, kW	Crankshaft rotation speed, min^{-1}	Average effective pressure, MPa	Specific fuel consumption, g/kW·h
UEC37LA (DKRN 37/88)	515	210	1,554	171
UEC45LA (DKRN 45/135)	882	158	1,559	166
RTA38 (DKRN 38/110)	680	196	1,667	181
RTA48 (DKRN 48/140)	1090	154	1,675	178
S26MCE (DKRN 26/98)	290	250	1.34	171

2.3. Calculation of the operating cycle of a MAN B&W 6DKRN 80/230 low-speed marine engine

When calculating the working cycle of the main two-stroke marine engine 6DKRN 80/230, the initial data were selected based on literature recommendations and the passport data of the prototype engine, taking into account the specified power, efficiency and permissible loads on the parts of the main engine. Table 2.2 below presents the main results of mathematical modelling of the working cycle of the main ship engine 6DKRN 80/230.

Table 2.2 – Calculation results for the main ship engine MAN B&W 6DKRN 80/230

№ s/n	Parameter	Dimension	Value
1.	Crankshaft speed	n, rpm	104.00
2.	Power	N_e, kW	21661.00
3.	Average effective pressure	P_e, MPa	1.8015
4.	Torque	M_e, Nm	1989000.00

5.	Cycle fuel consumption	q_c, g	98.8640
6.	Specific fuel consumption	g_e, kg/(kW)	170.00
7.	Effective efficiency	η_e	0.4900
	Environmental parameters		
8.	Ambient pressure	P_0, bar	1.00
9.	Ambient temperature	T_0, K	295.00
10.	Static pressure behind the turbine	P_{0t}, bar	1.020
11.	Inhibited flow pressure behind the filter	P_{oin}, bar	0.9800
	Supercharging and gas exchange		
12.	Pressure upstream of the exhaust manifold	P_k, bar	3.674
13.	Temperature upstream of the exhaust manifold	T_k, K	311.33
14.	Air flow rate through the ICE cylinders	G_{air}, kg/s	46.00901
15.	Efficiency of the supercharging unit	η_t	0.81701
16.	Average pressure before the turbine	P_t, MPa	0.31701
17.	Average temperature before the turbine	T_t, K	683.0401
18.	Gas flow rate through the ICE cylinders	G_{gas}, kg/s	47.053001
19.	Excess air coefficient	α_{sum}	3.080001
20.	Filling coefficient	η_v	0.8350001
21.	Residual gas coefficient	γ_r	0.03981
22.	Purge coefficient	φ_i	1.28243
	Inlet manifold		
23.	Average pressure in the intake manifold	P_s, MPa	0.3660021
24.	Average temperature in the intake manifold	T_s, K	318.74001
25.	Average wall temperature of the intake manifold	T_{ws}, K	318.74001
26.	Heat transfer coefficient in the intake manifold	α_{ws}, W/(m^2·K)	248.2001
27.	Heat transfer coefficient in the valve channel	α_{wsc}, W/(m^2·K)	80.90101
	Exhaust manifold		
28.	Average static pressure of the gas	P_r, MPa	0.31700121
29.	Average static gas temperature	T_r, K	682.62001
30.	Average gas velocity	W_r, m/s	83.710102
31.	Strouhal number	Sh	10.33300121
32.	Average wall temperature of the exhaust manifold	T_{wr}, K	519.290012
33.	Heat transfer coefficient in the exhaust manifold	α_{wr}, W/(m^2·K)	132.40123

34.	Heat transfer coefficient in the valve channel	α_{wcr}, W/(m^2·K)	168.101237
Combustion			
35.	Excess air coefficient	α	2.7880213
36.	Maximum combustion pressure	P_z, MPa	15.660124
37.	Maximum combustion temperature	T_z, K	1551.8001
38.	Maximum pressure angle	φ_{i_pz}, degrees above UDC	9.000
39.	Maximum temperature angle	φ_{i_tz}, degrees above UDC	22.000
40.	Maximum pressure rise rate	$dp/d\varphi_i$, bar/grad	4.9401210
Injection			
41.	Maximum injection pressure	$P_{inject\max}$, MPa	110.47012
42.	Average droplet diameter	d_{32}, µm	21.957
43.	Injection advance	θ_{ad}, degrees to UDC	6
44.	Duration of fuel injection	φ_{i_in}, degrees	22.000
45.	Ignition delay period	φ_{i_delay}, degrees	0.348
46.	The proportion of fuel that evaporated during the delay period	*Sig*	0.000128010
Environmental indicators			
47.	Hartridge smoke emission	Hartridge	2.052201210
48.	Smoke emission on the Bosch scale	Bosch	0.22401240
49.	Absolute light absorption coefficient of gas according to ECE	K, m^{-1}	0.048860120
50.	Particulate matter emission	PM, g/(kW·h)	0.042001210
51.	Carbon dioxide emission	CO_2, g/(kW·h)	550.6001101
52.	Wet NO_x concentration	NOx, 1/million, (ppm)	742.2301040
53.	NO_x emissions reduced to NO_2	NO_2, g/(kW·h)	2.7601210
54.	Complex of total NO_x and PM emissions	SE	0.536101234
55.	SO_2 emissions	SO_2, g/kW·h	0.0000
Duty cycle parameters at typical points			
56.	Initial compression pressure	P_a, MPa	0.383790010
57.	Compression start temperature	T_a, K	354.6601023
58.	End of compression pressure	P_c, MPa	11.976001
59.	End of compression temperature	T_c, K	911.94001

60.	Pressure at the beginning of release	P_b, MPa	1.375901000
61.	Temperature at the beginning of the release	T_b, K	961.6802490
Parameters that determine the working process			
62.	Compression ratio	ε	14.000
63.	Number of nozzle nozzles	$i_{nozzles}$	3.000
64.	Nozzle nozzle diameter	d_{nozzle}, mm	1.27001
65.	Start of discharge	$\varphi_{s.d}$, deg. to the LDC	40.000
66.	End of exhaust	$\varphi_{e.e}$, deg. to UDC	40.000
67.	Inlet start	$\varphi_{i.s}$, deg. to UDC	87.000
68.	End of the inlet	$\varphi_{i.e}$, deg. behind LDC	57.000
Compressor parameters (high pressure stages)			
69.	Compressor rotor speed	n, min^{-1}	1040.00121
70.	Compressor power	N, kW	7190.6001
71.	Adiabatic compressor efficiency	η_{ad}	0.860100
72.	Air flow through the compressor	G_{khp}, kg/s	46.0100124
73.	Air flow through the KHP	G_{giv}^{khp}, kg/s	796.730874
74.	Air flow through the KHP (corrected)	G_{cor}^{khp}, kg/s	46.1500785
75.	KHP rotor rotation frequency (reduced)	n_{giv}^{khp}, min^{-1}	61.2830102
76.	KHP rotor rotation frequency (corrected)	n_{cor}^{khp}, min^{-1}	1057.921003
77.	Degree of pressure increase in the HP compressor	P_{hp}	3.80000
78.	Total pressure at the KHP inlet	P_o^{khp}, MPa	0.098001
79.	Brake temperature at the KHP inlet	T_o^{khp}, K	295.0000
80.	Total pressure after the HP compressor	P_k, MPa	0.37240001
81.	Brake temperature after the HP compressor	T_k, K	443.51
82.	Thermal efficiency of the HP ONP	η_{Ecool}^{hp}	0.85001
83.	Cooling water temperature in the HP EPC	T_{cool}^{vt}, K	288
84.	Boost pressure at the KHP	P_k^{khp}, MPa	0.3674001
85.	Boost air temperature at the KHP	T_k^{khp}, K	311.33001
Turbine parameters (high pressure stages)			
86.	Rotor rotation frequency	n, min^{-1}	1040.0000
87.	THP power taking into account mechanical efficiency	N_{thp}, kBt	8283.000
88.	Internal efficiency of the HP turbine	η_t	0.91001210
89.	Mechanical efficiency of the HP turbine	η_m	0.97000001

90.	Gas flow through the THP	G_{thp}, kg/s	47.0530012
91.	Gas flow through the THP (reduced)	$G_{giv}{}^{thp}$	387.81110
92.	THP rotor rotation speed (reduced)	$n_{giv}{}^{thp}$, min^{-1}	39.793002
93.	Degree of pressure reduction in the HP turbine	$P_{\ thp}$	3.110001
94.	Relative work of the THP	B_{thp}	27.2940007
95.	Total pressure upstream of the HP turbine	P_t^{thp}, MPa	0.31709
96.	Braking temperature at the inlet to the THP	T_t^{thp}, K	683.0400100
97.	Back pressure behind the HP turbine	$P_{\text{o}}{}^{thp}$, MPa	0.101960001
98.	Gas temperature behind the HP turbine	$T_{\text{o}}{}^{thp}$, K	524.2201010

Based on the calculation of the working cycle of the modernised 6DKRN 80/230 main two-stroke marine engine, a theoretical indicator diagram was constructed, which is shown in Fig. 2.1.

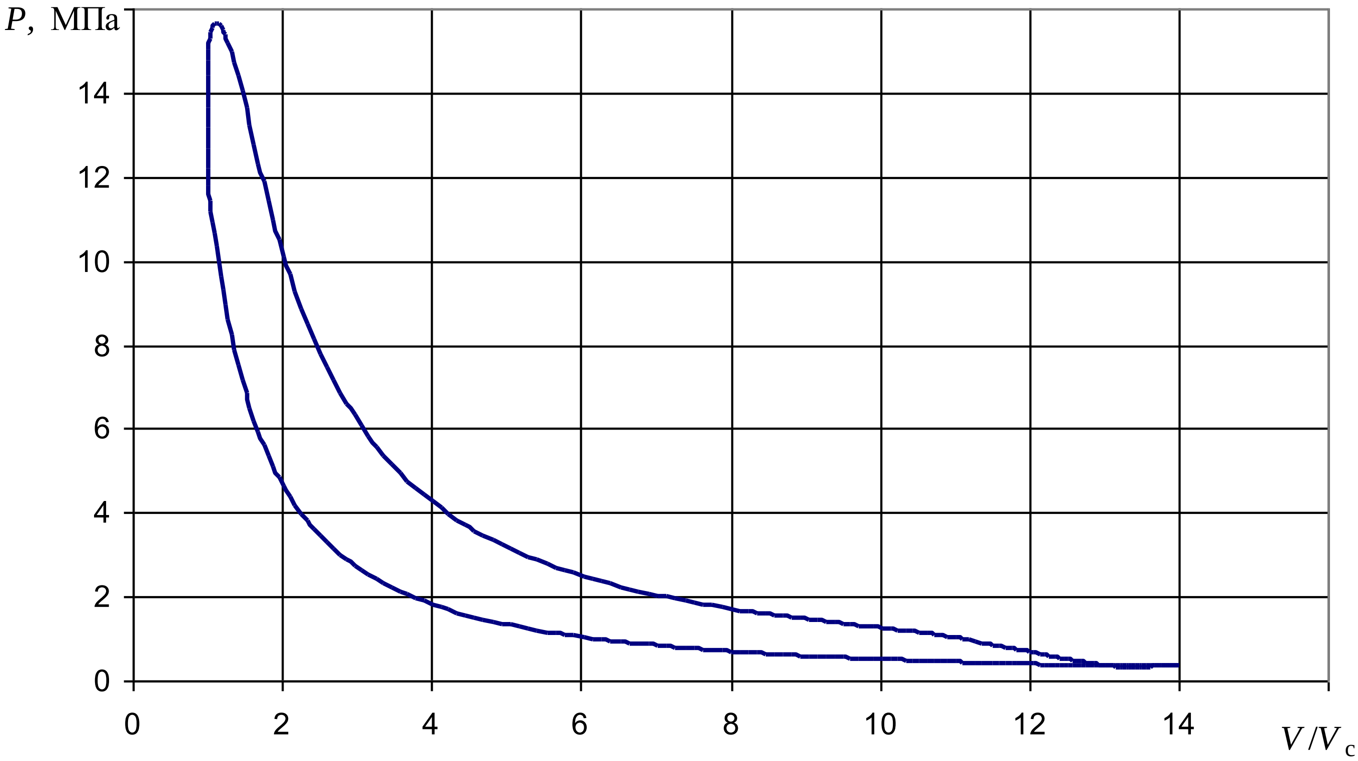

Figure 2.1 – Indicator diagram of the engine MAN B&W 6DKRN 80/230

2.4. Conclusions to the part

As a result of calculations of the working cycle of the main two-stroke marine engine MAN B&W 6DKRN 80/230 manufactured by MAN B& W with a modernised fuel system, all the necessary cycle parameters were obtained, which will be needed for further research and calculation of a new fuel system with a hydraulic drive. The main results of the calculation are summarised and presented in Table 2.3

The simulation results shown in Table 2.3 show that the modernised low-speed engine with a hydraulically driven fuel system outperforms the prototype engine in terms of energy, economic and environmental performance. In particular, the specific effective fuel consumption decreased by 3 g/(kW·h), which is quite significant, and the effective power increased by 2880 kW (15.4%).

Table 2.3 – Parameters of the modernised main ship two-stroke engine MAN B&W 6DKRN 80/230

№ s/n	**Parameter, designation and dimension**	**Value**
1.	Effective engine power N_e, kW	21600
2.	Effective fuel consumption g_e, g/(kW·h)	170
3.	Average effective pressure P_e, MPa	1,8
4.	Effective engine efficiency η_e	0,49
5.	Maximum combustion pressure P_z, MPa	15,66
6.	Maximum combustion temperature T_z, K	1551,8
7.	Supercharging pressure P_k, MPa	3,67
8.	Gas flow rate through the turbine G_t, kg/s	47,053
9.	Air flow rate G, kg/s	46,009
10.	Maximum pressure rise rate, $dp/d\varphi_i$, bar/grad	4.9401210
11.	Hartridge smoke emission, Hartridge	2.052201210
12.	Smoke emission on the Bosch scale, Bosch	0.22401240
13.	Absolute light absorption coefficient of gas according to ECE, K, m^{-1}	0.048860120
14.	Particulate matter emission, PM, g/(kW·h)	0.042001210
15.	Carbon dioxide emission, CO_2, g/(kW·h)	550.6001101
16.	Wet NO_x concentration, NOx, 1/million, (ppm)	742.2301040
17.	NO_x emissions reduced to NO_2, NO_2, g/(kW·h)	2.7601210
18.	Complex of total NO_x and PM emissions, SE	0.536101234
19.	SO_2 emissions, SO_2, g/kW·h	0.0000

PART 3
MODERNISATION OF THE FUEL SYSTEM OF A LOW-SPEED MARINE ENGINE BY USING A HYDRAULICALLY DRIVEN SYSTEM

3.1. General information on the use of fuel pumps with hydraulic plunger drive for low-speed marine engines

When developing new concepts for the working cycle of modern marine engines, a number of manufacturers have opted to replace standard high-pressure fuel pumps with hydraulically driven plunger pumps and electronic fuel supply control. For low-speed engines, such systems were developed by Mitsubishi, while Caterpillar increasingly uses electrically controlled pump nozzles with hydraulic drive for its high-speed engines.

The first low-speed engine with a hydraulic fuel supply system was manufactured by Mitsubishi in 2002 and was designated the UEC Eco-Engine. In such engines, in addition to fuel supply, the electronic system controls the operation of the engine exhaust valves, starting and reversing processes, cylinder lubrication, and crankshaft speed regulation.

Fuel pumps, which are essentially hydraulic pressure multipliers, are used to inject fuel into the working cylinder. Their design is simpler than that of their mechanical counterparts. The use of hydraulics in the engine made it possible to abandon the use of the camshaft and its drive, mechanisms for regulating the cycle fuel supply and controlling the injection phases (VIT). Another advantage is that the PNVT plunger has no regulating edges, which greatly simplifies its manufacturing technology and increases its service life. In addition, the elements used for supplying and delivering fuel to the working cylinder under high pressure do not require any design changes or changes in maintenance. This fuel system uses the same injectors, high-pressure pipes and connecting fittings as traditional injection systems.

The Mitsubishi high-pressure fuel pump consists of a plunger sleeve and a hydraulic cylinder. Inside the hydraulic cylinder there is a piston that is mechanically connected to the plunger of the high-pressure fuel pump (Fig. 3.1). The diameter of the drive hydraulic cylinder is several times larger than that of the plunger itself. This makes it possible to increase the fuel pressure at the pump outlet by the same amount, increasing it from 20...30 to 60...100 MPa. At the same time, unlike mechanically driven high-pressure fuel pumps, the pressure at which fuel is supplied to the injector does not depend on the engine crankshaft speed, which significantly improves the quality of combustion in the working cylinder under partial load conditions.

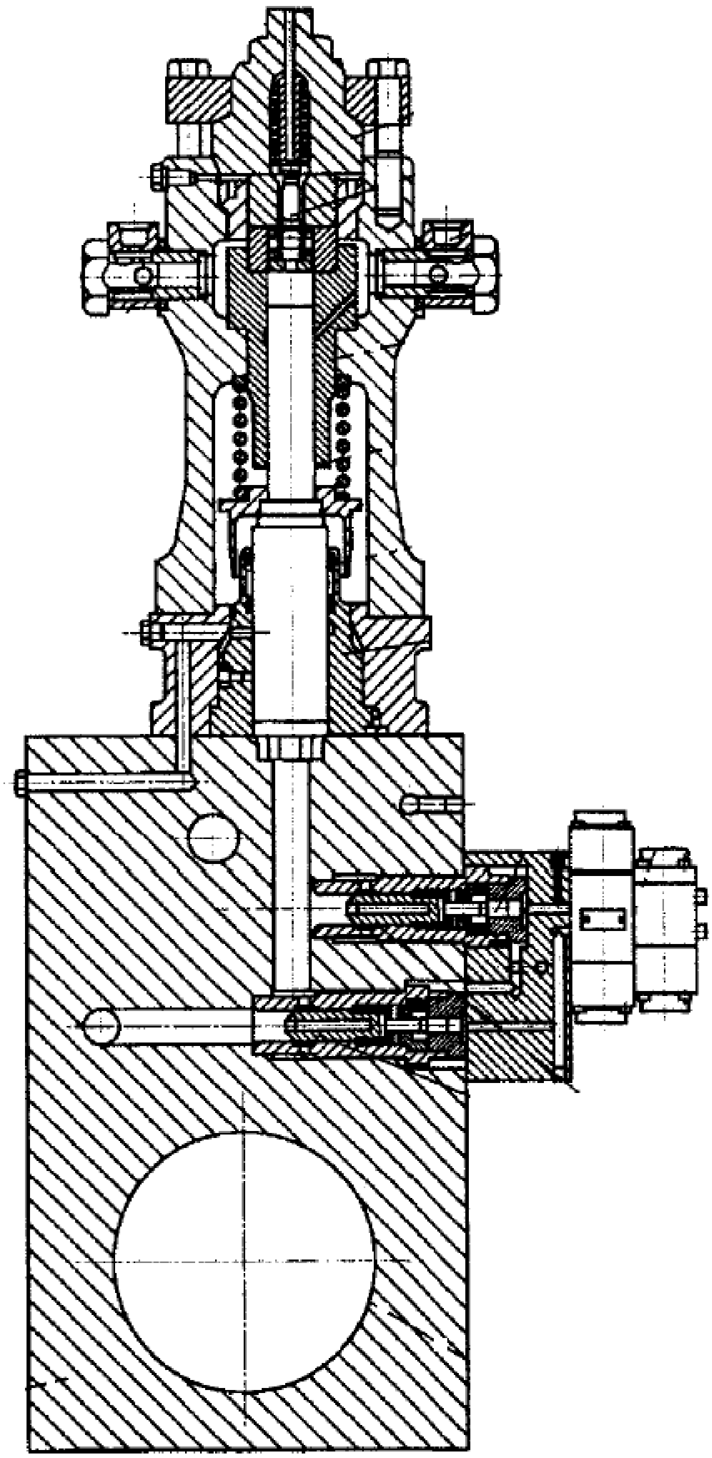

Figure 3.1 – High-pressure fuel pump with a hydraulic drive system manufactured by Mitsubishi, installed on the UEC Eco-Engine

In Mitsubishi's UEC Eco-Engine series engines, the cavity in the rigid pump block housing acts as a pressure accumulator. This solution made it possible to use higher pressure for driving the high-pressure fuel pumps and the exhaust valve and reduced the amount of oil in the system.

Mitsubishi's UEC Eco-Engine series engines use two-stage electrically controlled valve-type distribution devices in their high-pressure fuel pumps. The hydraulic diagram of the high-pressure fuel pump for the UEC Eco-Engine series engines is shown in Fig. 3.2.

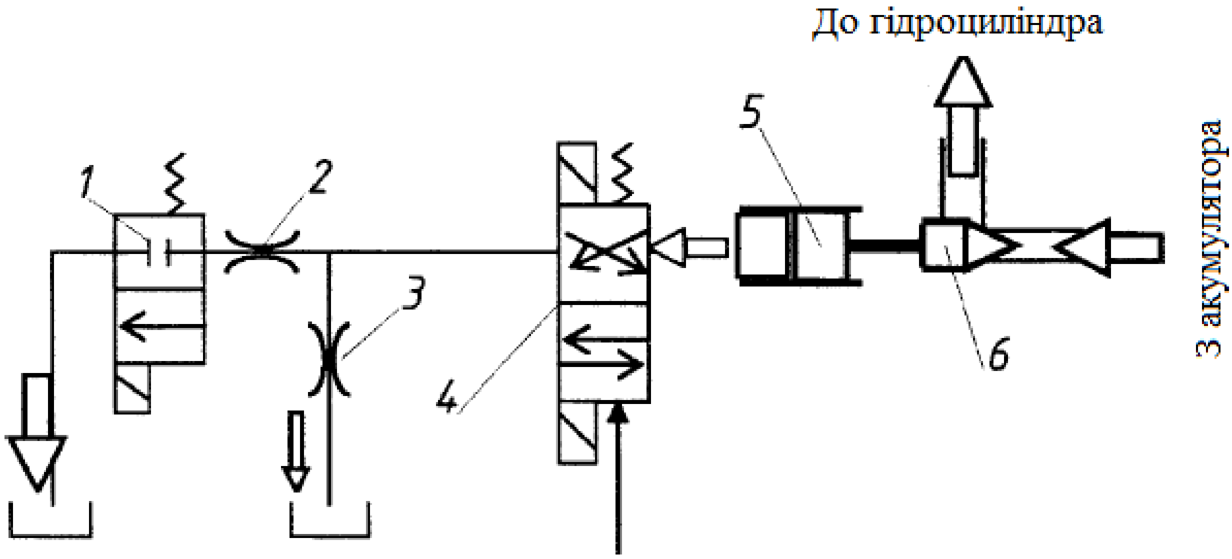

Figure 3.2 – Schematic diagram of the hydraulic drive of the high-pressure fuel pump of the UEC Eco-Engine series manufactured by Mitsubishi:
1 – pilot valve driven by a solenoid; *2* – main throttle; *3* – drain throttle; *4* – main distributor; *5* – main valve drive piston; *6* – main valve

3.2. Modernisation of the fuel system of the MAN B&W 6DKRN 80/230 low-speed marine engine

To minimise changes to the standard fuel equipment of the low-speed marine engine MAN B&W 6DKRN 80/230, a fuel system using high-pressure fuel pumps with hydraulic plunger drive and electronic fuel supply control is proposed. Each high-pressure fuel pump unit is equipped with a gas pressure accumulator with a diaphragm separator, which ensures a stable oil supply and prevents dangerous fluctuations in the control oil system (Fig. 3.3).

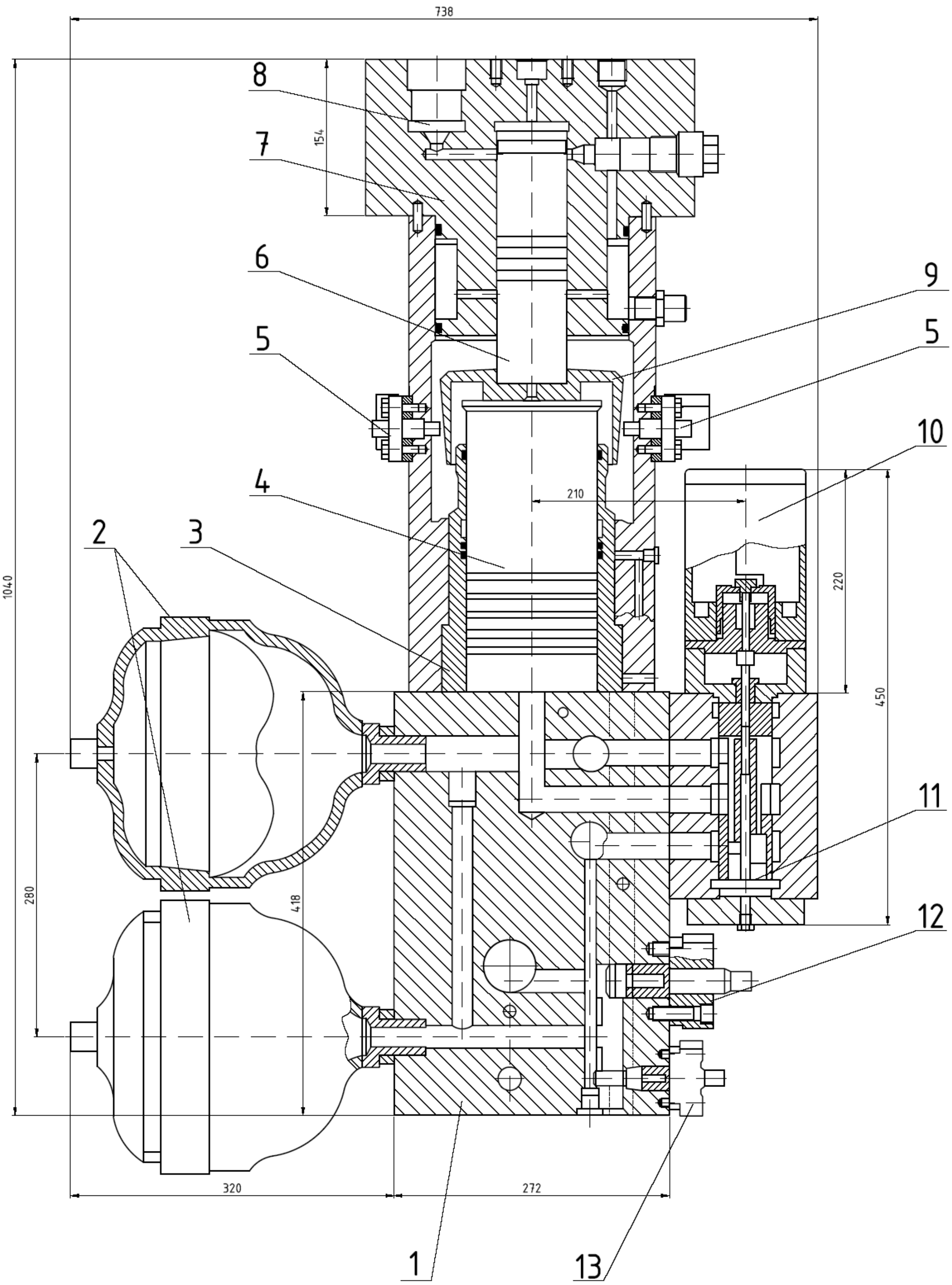

Figure 3.3 – High-pressure fuel pump with hydraulic drive system:
1 – high pressure fuel pump (HPFP) housing; *2* – gas pressure accumulators; *3* – hydraulic piston sleeve; *4* – hydraulic piston of the HPFP plunger drive; *5* – motion sensors; *6* – plunger; *7* – plunger bushing; *8* – high pressure line connection element; *9* – measuring cone; *10* – solenoid; *11* – control spool; *12* – control pressure regulating valve; *13* – control pressure limiting valve

Oil flows to the diaphragm pressure accumulators from the control line and from them to the distribution devices of the high-pressure fuel pump, the exhaust valve actuator, and the lubricator pump.

For the fuel system of the low-speed marine engine MAN B&W 6DKRN 80/230, it is proposed to use a two-stage electrically controlled spool-type distribution mechanism, the diagram of which is shown in Fig. 3.4.

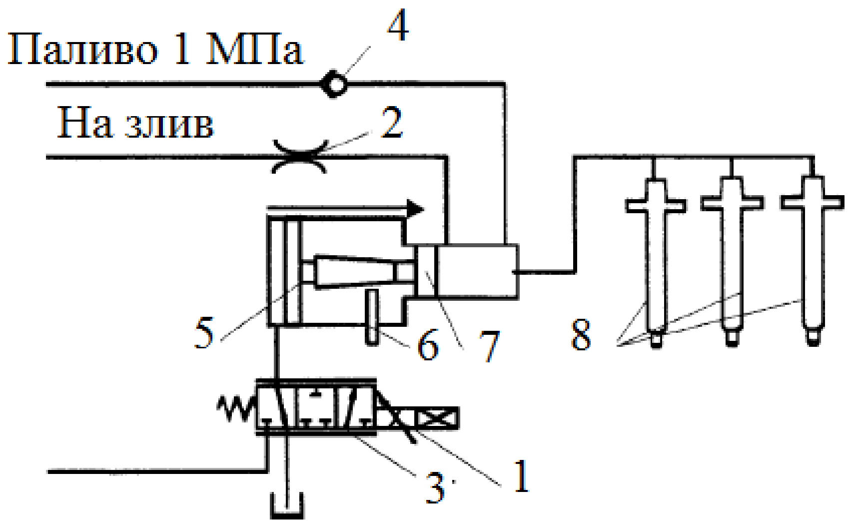

Figure 3.4 – Schematic diagram of the hydraulic drive of the high-pressure fuel pump of the MAN B&W 6DKRN 80/230 marine engine:

1 – pilot valve driven by a solenoid; *2* – drain throttle; *3* – main distributor; *4* – check valve; *5* – hydraulic drive piston; *6* – position sensor; *7* – fuel plunger; *8* – nozzles

The distribution mechanism of high-pressure fuel pumps and exhaust valve actuators has an identical design and, in some cases, is completely unified. The principle of operation and design of the spool distributors of the MAN B&W 6DKRN 80/230 marine engine is shown in Fig. 3.5.

The main spool with distribution grooves cut into it is located in the middle of the distributor housing. The main spool of the distribution device is moved by a protrusion that acts as a hydraulic piston. There are protrusions on both sides of the cavity, and oil flows through a so-called ‘pilot’ spool valve with an electronic drive. If there is no electrical signal, the spool valve assumes its middle position, blocking the connection between the cavity of the drive hydraulic cylinder A and the high-pressure cavity B and the drain cavity B with its central edge. In this case, the pump plunger remains stationary and there is no fuel injection (Fig. 3.5 a).

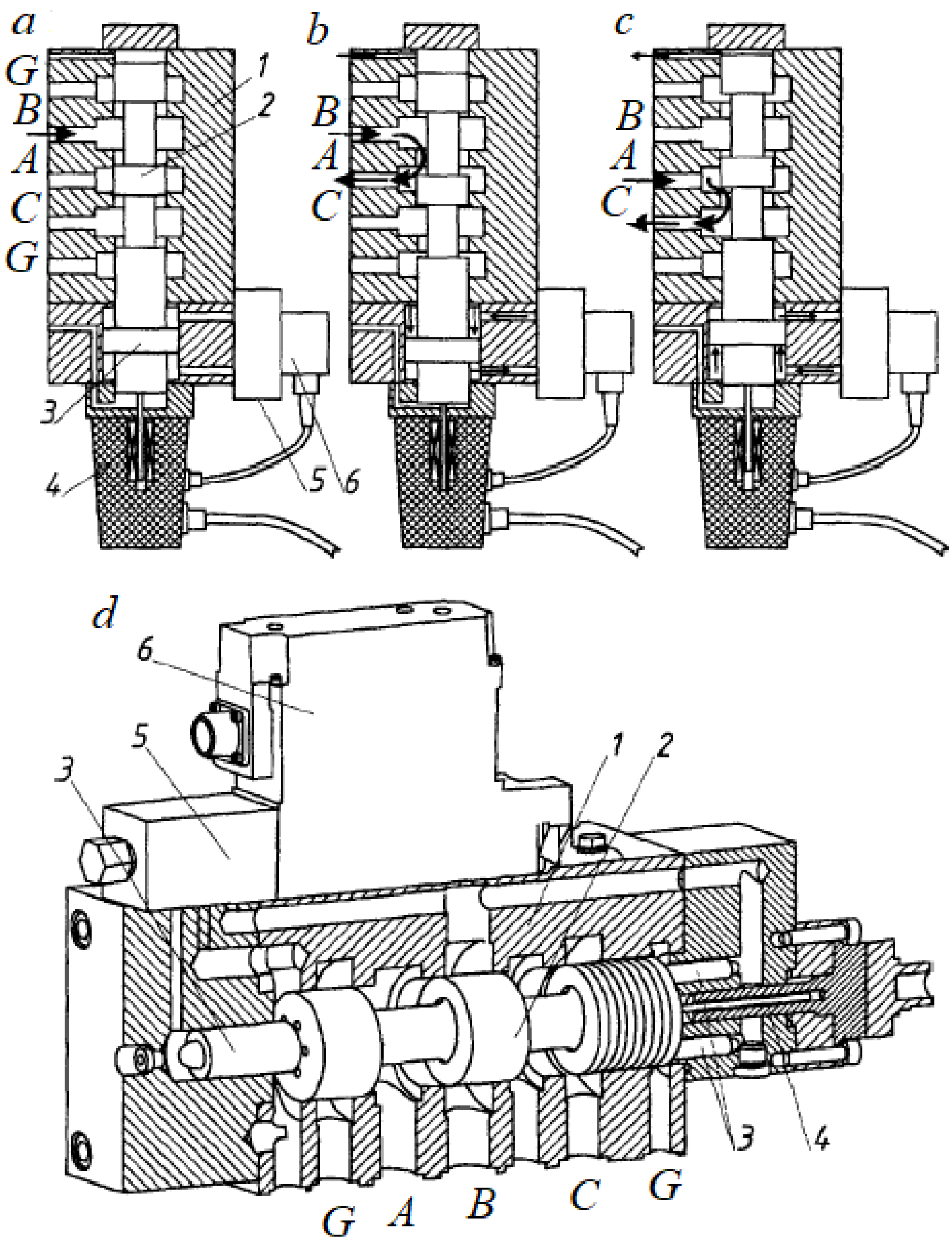

Figure 3.5 – Diagram of the operation of the spool valve of the high-pressure fuel pump of the MAN B&W 6DKRN 80/230 marine engine (*a*, *b*, *c*) and its general view (*d*): *1* – body; 2 – main spool valve; *3* – main spool valve drive piston; *4* – displacement sensor; *5* – pilot valve; *6* – solenoid; *A* – hydraulic cylinder cavity; *B* – high pressure cavity; *C* – drain cavity; *D* – leakage collection cavity

When an electrical signal is received from the control unit, oil enters the cavity above the control protrusion, moving it and the entire spool downwards. As a result, cavity *A* of the hydraulic piston connects to high-pressure cavity *B*, and the piston and plunger begin to move upward (Fig. 3.5 *b*). When the required pressure is reached, fuel injection begins in the working cylinder. The amount of fuel supplied to the working cylinder per cycle directly depends on the stroke of the plunger during the injection stroke, which is determined by the duration of the spool valve opening.

At the end of the injection stroke, the control valve supplies oil to the cavity under the protrusion, while the spool moves upward, connecting cavity *A* with drain cavity *B* (Fig. 3.5 *b*). The oil leaves the cavity of the hydraulic cylinder, fuel injection stops, and the plunger returns to its original position. In the spool pair, the distributor housing has additional drain cavities *G* to collect leaks.

Control over the position of the spool and its movement is provided by a feedback sensor, which generates and transmits a signal to the electronic control unit. Fig. 3.5 *d* shows a general view of a high-pressure fuel pump distributor, in which the pistons for driving the main spool are located at its ends.

The required oil supply torque to the drive piston, as well as the start of its discharge, is determined by the arrival of a control pulse from the electronic unit, which is separately equipped with each cylinder of the MAN B&W 6DKRN 80/230 engine.

The microprocessor-based electronic control system, combined with the hydrovolumetric drive system of the high-pressure fuel pump of the MAN B&W 6DKRN 80/230 marine engine, allows for various methods of fuel delivery to the working cylinder (single injection of the entire cycle dose of fuel, preliminary injection of the so-called pilot portion preceding the main injection, multi-stage injection, etc.). The required fuel supply law can be optimised for a specific operating mode of the marine engine and can be applied when switching to this mode.

The transition from one fuel supply control algorithm to another can be carried out in the interval between two consecutive injections. To ensure normal operation of the hydraulic drive of the high-pressure fuel pump, the exhaust valve and the lubricator pumps, the MAN B&W 6DKRN 80/230 engine is equipped with a control oil system, the schematic diagram of which is shown in Fig. 3.6.

Oil for the hydraulic drive control oil system is taken from the engine lubrication circulation system. Before entering the control line, the oil undergoes additional purification in a self-cleaning filter, which retains mechanical impurities up to 5 μm in size, and then the oil enters the pump unit. Some of the pumps are driven by the engine and supply oil to the system at a pressure of 20 MPa during diesel operation. The other pumps are electrically driven and supply the system with oil at a pressure of 17.5 MPa during engine start-up or when operating in emergency situations. After the pumps, the oil enters the accumulation chamber, which reduces pressure pulsations. The oil then flows through high-pressure pipes to each high-pressure pump unit of the engine.

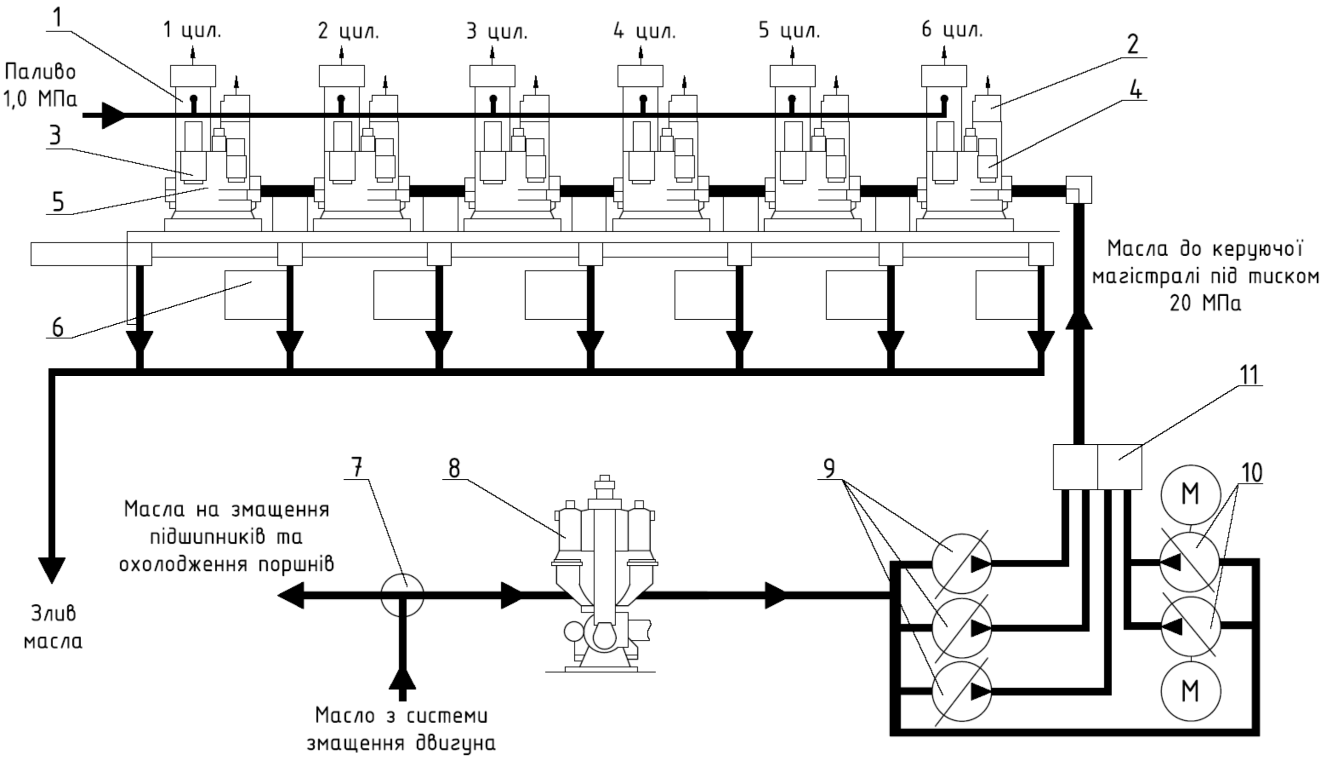

Figure 3.6 – Diagram of the control oil system for the hydraulic drive of the high-pressure fuel pump, exhaust valve actuators and lubricator pumps of the MAN B&W 6DKRN 80/230 engine:

1 – HPFP; *2* – exhaust valve actuator; *3* – HPFP hydraulic drive control valve; *4* – exhaust valve actuator hydraulic drive control valve; *5* – lubricator pump; *6* – electronic control unit; *7* – oil distributor; *8* – automatic fine oil filter; *9* – engine-driven oil pumps; *10* – electric motor-driven oil pumps; *11* – accumulator

To prevent injury to service personnel in the event of a rupture of oil lines, they are manufactured in two layers. Accordingly, the inner cavity of the oil line is used as a working cavity, which is covered with a protective casing on the outside (Fig. 3.7). To ensure oil supply to the control line, axial piston pumps are used on the engine, in which the performance is regulated by changing the angle of the drive disc. The design of an axial piston oil pump is shown in Fig. 3.8. The stroke of the pistons in this type of pump directly depends on the angle of inclination of the drive disc to the axis of the cylinder block.

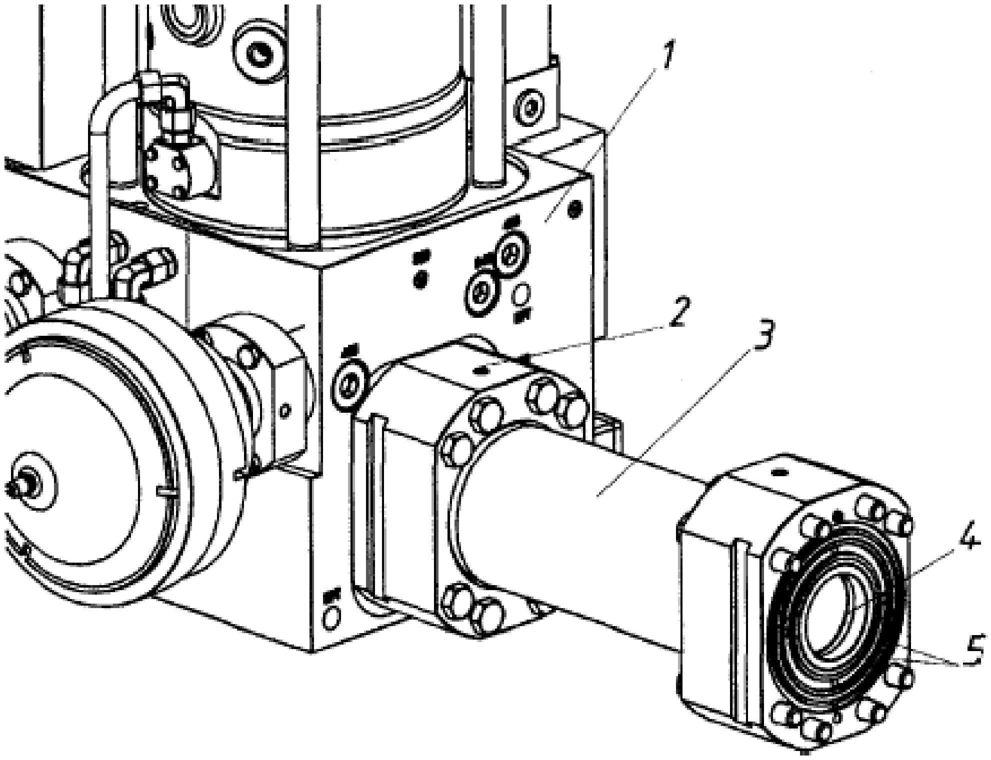

Figure 3.7 – Fragment of the control oil system main line:
1 – high-pressure fuel pump housing; *2* – connecting flange; *3* – protective cover; *4* – working surface; *5* – seal

Changing the angle of the disc, which is achieved by means of a hydraulic mechanism, allows the pump capacity to be adjusted from zero flow (the drive disc is positioned perpendicular to the drive shaft axis) to maximum (the drive disc is deflected to the maximum angle). This makes it possible, regardless of the engine crankshaft speed, to maintain a constant oil flow to the control line, maintaining the required pressure in the system.

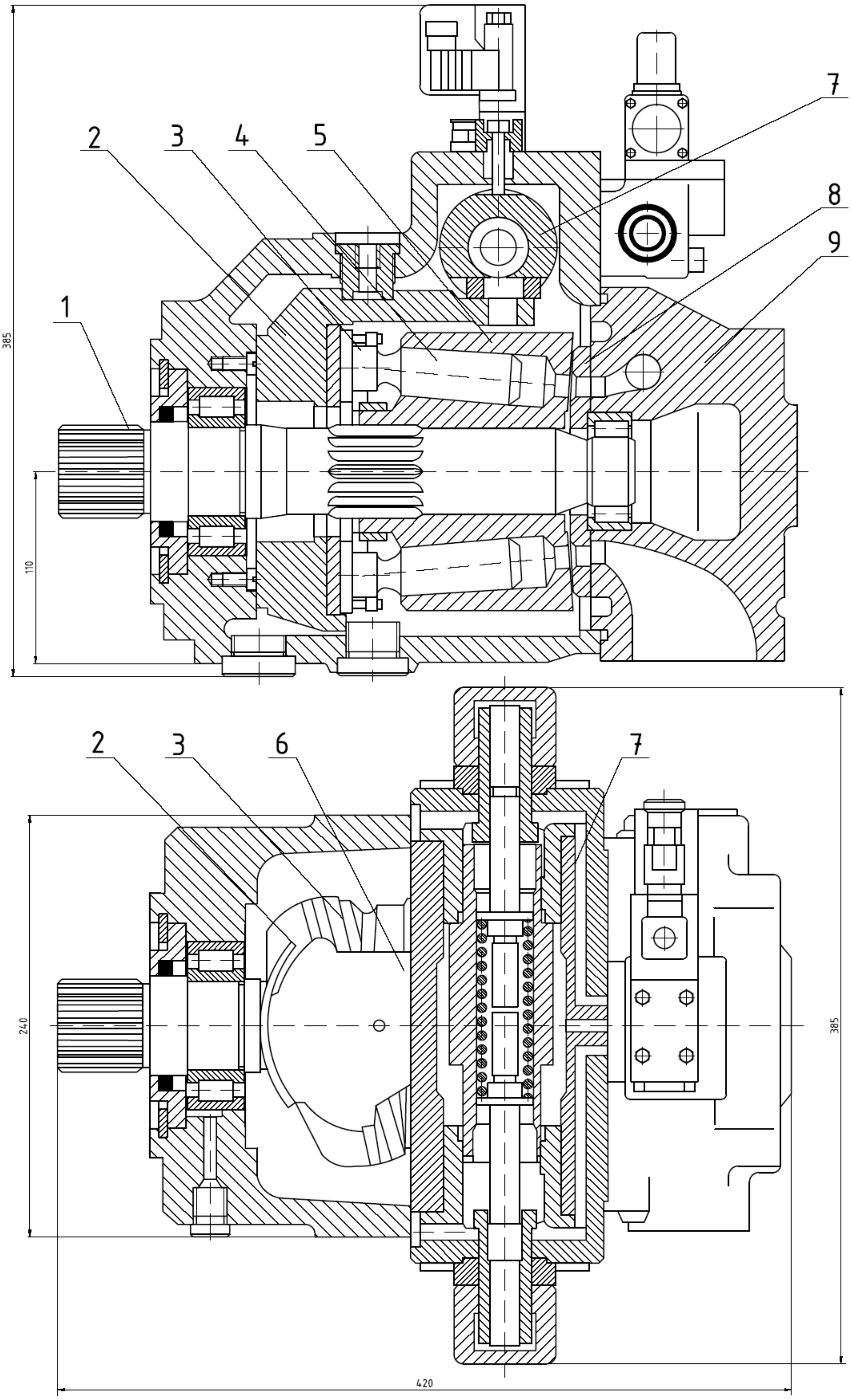

Figure 3.8 – Axial plunger high-pressure oil pump:

1 – pump drive shaft; *2* – inclined drive disc; *3* – ball bearing support; *4* – plunger; *5* – stator; *6* – drive disc inclination change lever; *7* – disc inclination change mechanism; *8* – support and distribution disc; *9* – cover

One of the main advantages of this type of pump is the absence of valve mechanisms. The oil sucked in or displaced by the plungers is distributed through crescent-shaped windows in the support and distribution disc and holes in the cylinder block. At the corresponding dead points of the plungers, the openings of each cylinder are covered by the lower and upper distribution bridges located between the windows. Thanks to this, such pumps have a sufficiently high efficiency with relatively low oil supply unevenness, and when the direction of rotation of the engine crankshaft changes, they maintain the same oil pumping direction.

3.3. Calculation and design of a high-pressure fuel pump for a 6DKRN 80/230 marine engine with hydraulic plunger drive and electronic fuel supply control

The initial data for calculating and designing a high-pressure fuel pump for the main two-stroke marine engine 6DKRN 80/230 manufactured by MAN B&W with hydraulic plunger drive and electronic fuel supply control was obtained in the second section of the project as a result of mathematical modelling of the engine operating cycle.

Table 3.1 – Calculation of high-pressure fuel pump for the main two-stroke marine engine 6DKRN 80/230 manufactured by MAN B&W

№ s/n	Parameter	Dimension	Value
Input data for calculating a high-pressure fuel pump			
1.	Crankshaft speed	n, rpm	104.00
2.	Power of the engine	N_e, kW	21600.00
3.	Average effective pressure	P_e, MPa	1.8015
4.	Number of cylinders	i	6
5.	Cycle fuel consumption of the engine	q_c, g	98.8640
6.	Specific fuel consumption of the engine	g_e, kg/(kW)	170.00
7.	Maximum combustion pressure	P_z, MPa	15.6600
8.	Stroke ratio	z	1.000
9.	Fuel supply duration	$\varphi_{s.d}$, deg. of c. r.	22.000
10.	Cylinder pressure at point c	P_c, MPa	11.9800
Determining the diameter and stroke of a high-pressure fuel pump plunger			
11.	Density of fuel	γ_f, kg/m^3	875.00
12.	Reserve factor	K	1.150

13.	Cyclic fuel supply $Vc = \frac{Neц \cdot g_e \cdot K \cdot V_{ц.осн}}{60 \cdot n \cdot z \cdot \gamma_m}$	V_c, cm^3/cycle	128.901
14.	Coefficient of volume reserve	K_2	4.000
15.	Ratio of plunger stroke to diameter $\frac{S_p}{d_p} = 1...1{,}5$	–	$1{,}5 \cdot d_p$
16.	Diameter of the plunger $d_p = \sqrt[3]{\frac{4 \cdot V_p}{1{,}3 \cdot \pi}} =$	d_p, mm	80.000
17.	Stroke of the plunger $S_p = 1{,}5 \cdot d_p$	S_p, mm	120.000
18.	Volume describing the plunger during the stroke from VDC to LDC $V_p = K_2 \cdot V_c = \frac{\pi \cdot d_p^2}{4} \cdot S_p$	V_p, cm^3	515.6040
19.	Area of the plunger $F_{пл} = \frac{\pi \cdot d^2}{4}$	F_p, mm^2	4989.0690
Calculation of the fuel injection process			
20.	Gas pressure in the diesel 6DKRN 80/230 cylinder at the moment of injection $P_{cyl} = \frac{P_c + P_Z}{2}$	P_{cyl} , MPa	13.8200
21.	Fuel compressibility factor	α_c, MPa^{-1}	8E-04
22.	Flow coefficient of the filling holes of the plunger pair	μ_f	0,8600
23.	Fuel pressure in the low pressure cavity	P_0, MPa	0,4
24.	Diameter of the filling holes	d_f, mm	20.000
25.	Number of filling holes	i_f	4.000
26.	Cross-section of the filling holes $f_f = i_f \cdot \frac{\pi \cdot d_f^2}{4}$	f_f, m^2	0.00126
27.	Spray diameter needle	d_n, mm	30.000
28.	Diameter of the closing cone base of the needle spray	d_c, mm	18.200
29.	Differential plane relative value of the needle spray	δ	0.6400
30.	Needle spray angle of the closing cone	δ_n, deg	60.000
31.	Spray well diameter	d_w, mm	20.000
32.	Spray holes diameter	d_s, mm	0.500

33.	Spray holes number	i_s	3
34.	Needle spray lift	h_n, mm	0.4500
35.	Residual pressure in the system of high-pressure	$P_{r.p}$, MPa	3.000
36.	Start pressure of injection	P_{st}, MPa	40.000
37.	Minimum flow section in the closing cone flow coefficient of the spray nozzle	$\mu_{c.c}$	0.7000
38.	Spray nozzle openings flow coefficient	μ_s	0.6000
39.	Nozzle openings total flow cross-section	f_P, mm^2	0.7850
	First fuel supply period calculation		
40.	Coefficient $K_1^1 = \dfrac{F_П}{\alpha_{СЖ} \cdot 6 \cdot n_K \cdot V_H}$	K_1^1	31.40250
41.	Coefficient $K_3^1 = \dfrac{\mu_H}{\alpha_{СЖ} \cdot 6 \cdot n_K \cdot V_H} \cdot \sqrt{\dfrac{2}{\rho_T}}$	K_3^1	258.7930
42.	Pressure $P_H = \left(\dfrac{K_1^1 \cdot C_П}{K_3^1 \cdot f_H}\right)^2 + P_0$	P_H, МПа	0.41985
43.	Pressure for β $\Delta P_H = \left(K_1^1 \cdot C_П - K_3^1 \cdot f_H \cdot \sqrt{(P_H - P_0) \cdot 10^6}\right) \cdot \Delta\beta$ β: 0.5 1.0 1.5 2.0 2.5	ΔP_H, МПа	0.63338355 1.28966399 1.94594444 2.60222489 3.25850533
44.	First period pressure	P_1, МПа	9.730000
	The second fuel delivery period calculation		
45.	Base cross-section of the closing cone of the needle spray $F_K = \dfrac{\pi \cdot d_K^2}{4}$	F_K, mm^2	52.783001
46.	Spray needle differential area $F_\delta = \dfrac{\pi \cdot \left(d_И^2 - d_K^2\right)}{4} =$	F_δ, mm^2	653.717001
47.	Coefficient $K_1^2 = \dfrac{F_П}{\alpha \cdot 6 \cdot n_K \cdot (V_H + V_Л)} =$	K_1^2	31.4025
48.	Coefficient $K_3^2 = \dfrac{\mu_H}{\alpha_{СЖ} \cdot 6 \cdot n_K \cdot (V_H + V_Л)} \cdot \sqrt{\dfrac{2}{\rho_T}}$	K_3^2	129.396001
49.	Pressure for β	ΔP, МПа	

	$\Delta P=\left(K_1^2\cdot C_П-K_3^2\cdot f_Н\cdot\sqrt{\left(P_Н-P_0\right)\cdot 10^6}\right)\cdot\Delta\beta$ β: 3.0 3.5 4.0		3.03319405 3.68947450 4.34575495
50.	Pressure before closing the filling holes	P, МПа	11.06842
51.	Pressure for β $\Delta P=K_1^2\cdot C_П\cdot\Delta\beta;$ β: 4.5 5.0 5.5 6.5 7.5	ΔP, МПа	23.6260968 24.2823765 24.9386569 25.5949374 26.2512178
52.	Pressure after closing the filling holes	P, МПа	83.9200010
53.	Needle spray lift pressure $P_Н=P_{ВП}-P_С\cdot\frac{F_К}{F_\delta}$	$P_Н$, МПа	39.0300102
54.	Pressure in the high pressure line, $P_l=P_н-P_{зт}$	P_l, МПа	80.920012
Third fuel delivery period calculation			
55.	Coefficient $K_1^3=\frac{F_П}{\alpha_{СЖ}\cdot 6\cdot n_К\cdot\left(V_Н+V_Л+V_Р\right)}$	K_1^3	31.353301
56.	Coefficient $K_2^3=\frac{\mu_Р\cdot f_Р}{\alpha_{СЖ}\cdot 6\cdot n_К\cdot\left(V_Н+V_Л+V_Р\right)}\cdot\sqrt{\frac{2}{\rho_Т}}$	K_2^3	0.0001001
57.	Coefficient $K_С=\frac{\left(\mu_{ЗК}\cdot F_{ЗК}\right)^2}{\left(\mu_{ЗК}\cdot F_{ЗК}\right)^2+\left(\mu_Р\cdot f_Р\right)^2}$	$K_С$	0.99769001
58.	Area $F_{ЗК}=\pi\cdot\left(d_{КО}-h_Н\cdot\sin\frac{\delta_Н}{2}\cdot\cos\frac{\delta_Н}{2}\right)\cdot h_Н\cdot\sin\frac{\delta_Н}{2}$	$F_{ЗК}$, m²	1.4E-05
59.	Pressure for β $\Delta P=\left(K_1^3\cdot C_П-K_2^3\cdot\sqrt{K_С\cdot\left(P_Л-P_Ц\right)\cdot 10^6}\right)\cdot\Delta\beta;$ β:	ΔP, МПа	10,48343742

	8.0 8.5 9.0 9.5 10.5 11.0 11.5 12.0 12.5 13.0 13.5 14.0 14.5 15.0 15.5 16.0 16.5 17.0 17.5 18.0 18.5 19.0 19.5 20.0 20.5 21.0 21.5 22.0		11,13868845 11,79393947 12,44919049 13,10444151 13,75969253 14,41494355 15,07019457 15,72544559 16,38069661 17,03594764 17,69119866 18,34644968 19,0017007 19,65695172 20,31220274 20,96745376 21,62270478 22,2779558 22,93320683 23,58845785 24,24370887 24,89895989 25,55421091 26,20946193 26,86471295 27,51996397 28,17521499 28,83046602
60.	Pressure after closing the filling holes	P, МПа	109.750001
61.	Pressure in the high pressure line $P_{l1+1} = P_{l1} + \Delta P$	P_{l1+1}, MPa	101.880023
62.	Spray holes fuel pressure in front $P_C = \frac{(\mu_{ЗК} \cdot F_{ЗК})^2 \cdot P_Л + (\mu_P \cdot f_P)^2 \cdot P_Ц}{(\mu_{ЗК} \cdot F_{ЗК})^2 + (\mu_P \cdot f_P)^2}$	P_C, МПа	101.680021
63.	The injector needle seating pressure $P_{кв} = P_{нв} \cdot \delta$	$P_{кв}$, МПа	25.600213
Plunger calculation			
64.	Plunger minimum diameter	$d_{p\min}$, mm	76.0000

65.	Compressive stress $\sigma_{СЖ} = \dfrac{P_T}{F_{ПMIN}}$	σ_{cs}, MPa	83.915601
66.	Area of the plunger minimum cross-section $F_{\min} = \dfrac{\pi \cdot d^2_{ПMIN}}{4}$	F_{min}, mm^2	4501.0001
67.	Compression stress in the plunger minimum cross-section ([σ_{com}] = 300 MPa) $\sigma_{СЖ} = \dfrac{P_T}{F_{ПMIN}}$	σ_{cs}, MPa	93.015401
68.	Plunger support end pusher diameter	d_{pop}, mm	60.0000
69.	Plunger support face area $F_{поп} = \dfrac{\pi \cdot d^2_{ПОП}}{4}$	F, mm^2	2826.0001
70.	Plunger and pusher specific load at the flat ends ([K_t] = 2000 MPa) $K_m = \dfrac{P_T}{F_{ПОП}} =$	K_t, MPa	148.147010

3.4. Conclusions to the part

This section develops a schematic diagram of a fuel system with a high-pressure fuel pump with hydraulic plunger drive and electronic fuel supply control for the main low-speed marine engine 6DKRN 80/230 manufactured by MAN B&W, which is installed on a container ship. The results of the engine operating cycle calculation obtained in the second section made it possible to determine the parameters and design the main elements of the high-pressure fuel pump with a hydraulically driven plunger. According to the calculation, all mechanical loads acting on the elements of the high-pressure fuel pump and fuel injector are within acceptable limits.

Drawings of the control oil system for the hydraulic drive, high-pressure fuel pump, axial-plunger high-pressure oil pump and fuel pump spool valve are presented on A1 sheets.

PART 4

ECONOMIC JUSTIFICATION FOR MODERNISING THE FUEL SYSTEM OF A LOW-TURNING SHIP DIESEL ENGINE MAN B&W 6DKRN 80/230

4.1. Technical and economic justification for modernising the fuel system of a low-speed marine engine MAN B&W 6DKRN 80/230 by using a hydraulically driven system

Technical and economic justification for the implementation of the project to modernise the fuel system of the MAN B& diesel engine W 6DKRN 80/230 engine is one of the main stages of the qualification work, providing a detailed justification of the project and an opportunity to comprehensively assess the effectiveness of the technical solutions adopted, the planned measures, and the risks of investing in the engine modernisation project.

The technical and economic justification for the implementation of the project to modernise the fuel system of the 6DKRN 80/230 engine should:

– show that the modernised diesel engine will find its consumer, establish the capacity of the market for modernised engines and the prospects for its development;

– assess the possible risks and costs associated with modernising the diesel engine fuel system and marketing;

– determine the profitability of the modernised marine diesel engine and demonstrate its effectiveness for the shipowner.

When implementing the technical and economic justification for the modernisation of the fuel system of the 6DKRN 80/230 engine, it is essential to provide references to the original sources of the information used. When selecting sources of information, preference should be given to those whose data can be considered reliable and as objective as possible: official statistics, current regulatory and legal documentation, information from recognised well-known analytical agencies, data from specialised industry publications, and Internet resources.

When carrying out a technical and economic justification for the modernisation of the fuel system of the 6DKRN 80/230 engine, it is necessary to carefully consider and take into account existing advanced scientific and technical solutions and achievements, identify and analyse the positive and negative aspects of existing solutions, and develop a set of measures that will give the new marine diesel engine, or in our case the modernised one, significant technical and economic advantages over existing ones.

The use of a hydraulically driven system on the engine reduces specific effective fuel consumption by 3.06 g/(kW·h), which is quite significant, and effective power has increased by 2880 kW. Such a significant reduction in fuel consumption will be reflected in the shipowner's total operating costs when using this diesel engine. Accordingly, when conducting a technical and economic justification for the modernisation of the fuel system of the 6DKRN 80/230 engine, it is necessary to focus on the fuel component of operating costs. When conducting a technical and economic justification for modernisation (see Tables 4.1 and 4.2), we will compare the capital and operating costs of the standard 6DKRN 80/230 marine diesel engine and its modernised version.

Table 4.1 – Technical and economic characteristics of the modernised and basic versions of the 6DKRN 80/230 low-speed marine diesel engine

№ s/n	**Name of indicators**	**Designation**	**Dimensionality**	**Basic 6DKRN 80/230**	**Modernised 6DKRN 80/230**
1	Power	N_e	kW	18720	21600
2	Rotation speed	n	min^{-1}	104	104
3	Diameter of the cylinder	D	m	0,8	0,8
4	Stroke of the piston	S	m	2,3	2,3
5	Specific fuel consumption	g_e	kg/kW·h	0,173	0,170
6	Mechanical efficiency	η_m	–	0,95	0,95
7	Specific oil consumption	g_o	kg/kW·h	0.0006	0.0006
8	Weight	G	kg	705000	705000
9		L	mm	11154	11154
10	Dimensions	B	mm	10374	10374
11		H	mm	13470	13470
12	Price of the engine	P	UAH	850000	950000
13	Price of fuel	P_f	UAH/(kW·h)	1,85	1,85
14	Price of oil	P_o	UAH/kg	9,5	9,5
15	Work hours for the year	t	год	6380	6380
16	Number of control repairs	η	–	1	1
17	Total capital repairs coefficient	ε_{min}	–	1,1	1,1
18	Partial labour cost coefficient	b	–	0,05	0,05
19	Power utilisation coefficient	K_p	–	0,95	0,95
20	Coefficient of part of the cost of capital repairs	$K_{c.r}$	–	0,3	0,3

21	Time of continuous operation	$t_{c.o}$	h	300	300
22	Engine life before the first bulkhead	t_{bul}	h	22000	22000
23	Engine life before overhaul	$t_{o.h}$	h	92000	92000
24	Normative cost-effectiveness comparison coefficient	E	–	0,15	0,15

Table 4.2 – Calculation and results of the technical and economic justification for modernising the fuel system of the 6DKRN 80/230 engine

Name	**Symbols**	**Method of determination**	**Basic 6DKRN 80/230**	**Modernised 6DKRN 80/230**
Annual output	N, kW·h per year	$N = K_П \cdot N_e \cdot t$	113461920	130917600
The service period	T, year	$T = \dfrac{t_{кр}}{t}$	14,42	14,42
Annual current fuel costs	C_f, UAH	$З_п = N \cdot Ц_п \cdot g_e$	36313487,5	41173585,2
Annual operating costs for oil	C_o, UAH	$З_м = g_м \cdot N \cdot Ц_м$	646732,944	746230,32
Annual costs for routine repairs	$C_{r.r}$, UAH	$З_{п.р} = \dfrac{b \cdot t \cdot Ц}{t_{пер}}$	12325	13775
Annual costs of capital repairs	$C_{c.r}$, UAH	$З_{к.р} = \dfrac{K_{к.р} \cdot t \cdot Ц}{t_{к.р} \cdot \varepsilon_{\min}}$	16076,09	17967,39
Annual economic effect for the consumer	U_1, UAH	$U_1 = З_п + З_м + З_{к.р} + З_{п.р}$	36988621,53	41951557,91
	$U_1^{1б}$, UAH	$U_1^{1б} = \dfrac{N_c}{N_б} \cdot U_1^б$	42679178,68	–
Annual economic effect on the consumer from the use of the designed engine	$E_к$, UAH	$E_к = \left[Ц_б \cdot \dfrac{N_c}{N_б} \cdot \left(\dfrac{\dfrac{1}{T_б} + E_б}{\dfrac{1}{T_c} + E_c} \right) + \dfrac{U_1^{1б} - U_1^c}{\dfrac{1}{T_c} + E_c} \right] - Ц_c$		3347969,983
Additional one-off costs for engine upgrades	ΔK, UAH	–	100000	

Annual savings when operating the upgraded engine	ΔS, UAH	$\Delta S = U_1^{1б} - U_1^{c}$	727620,77
Payback period of the upgraded engine	T_{pay}, year/month	$T_{окуп} = \dfrac{\Delta K}{\Delta S}$	0,14/ 1,6

4.2. Conclusions to the section

A theoretical assessment of the economic feasibility of modernising the fuel system of the main low-speed ship engine 6DKRN 80/230 manufactured by MAN B&W, which is installed on a container ship, has been carried out. The calculation results confirmed the economic feasibility of modernising the engine fuel system by using fuel pumps with a hydraulic plunger drive.

The annual economic effect for the consumer from the use of the modernised low-speed engine 6DKRN 80/23 on the ship, compared to the serial one, is 3347969.983 UAH, and the modernisation of the serial engine will pay for itself in just 0.14 years.

PART 5
ANALYSING AND DEVELOPING MEASURES TO ENSURE LABOUR AND ENVIRONMENTAL PROTECTION REQUIREMENTS FOR SHIP ENGINE MAN B&W 6DKRN 80/230 OPERATION

5.1. Occupational safety measures during installation of the 6DKRN 80/230 marine engine

Installation of the main marine engine must be carried out in accordance with the basic occupational safety requirements for shipbuilding (OST 5.0241–78 and OST 5.4110–87).

Work on the installation of the main ship engine is carried out in the equipment-filled engine room, where electric welding and gas cutting of metal can be carried out simultaneously, causing the air space of the compartments to be contaminated with harmful gases and fine dust containing elements that are toxic to humans. The pneumatic tools used in the work generate intense high-frequency noise, which contributes to rapid fatigue and loss of human performance.

When painting ship structures and various elements, toxic vapours of various solvents contained in paints, such as acetone, benzene or xylene, are released into the surrounding air. Therefore, when performing work on a ship, it is necessary to pay close attention to the health of the service personnel, as well as to conscientiously comply with the rules of safety and industrial hygiene.

Only persons who have undergone a special medical examination and have the appropriate doctor's permission, as well as having undergone training in occupational health and safety and fire safety in the workplace, are allowed to work on ships. Service personnel must be provided with protective helmets, special clothing and special footwear.

When installing the main engine, only serviceable tools and equipment specially adapted for ship conditions should be used. When performing work, it is prohibited to carry or lift objects weighing more than 50 kg for men and 20 kg for women. Hand-held power tools must operate at a voltage not exceeding 36 V, and the connection of power tools to the power source must be performed by an electrician. Before starting any work, power tools must be checked for insulation and grounding protection. For local lighting, it is permitted to use portable lamps in explosion-proof design with a supply voltage not exceeding 12 V. When processing parts with hand-held power tools, the parts must be securely fastened. It is prohibited to process parts in a suspended position.

Only workers over the age of 18 who do not have any skin diseases are allowed to work with epoxy resins and plastics based on them. Epoxy resin that has come into contact with the skin must be immediately removed with paper towels or a cloth moistened with dibutyl phthalate, and the skin must be washed with warm water and soap.

Workers who perform cooling and bolting must be provided with heat-resistant gloves.

When testing pipelines, it is necessary to first check that the pneumatic or hydraulic press and fittings are in good working order and that the pressure gauges have not expired. If the pipeline is under excess pressure, it is strictly prohibited to tighten nuts and flanged pipe connections.

To protect workers who are not performing welding work from the effects of the welding arc, their workplaces must be screened off with screens or partitions.

When performing work on the decommissioning of mechanisms on the ship, fire extinguishing equipment must be available in the room. During this work, welding, gas cutting or other work involving open flames is strictly prohibited.

Ventilation of the room during the installation of the main engine must comply with the requirements of OST 5.9971-85, and the level of harmful factors must not exceed the maximum permissible values.

The temperature, relative humidity and air velocity in the working area must comply with the requirements of OST 12.1.005–86.

Repairing marine diesel engines, especially powerful ones such as the 6DKRN 80/230, involves many hazardous factors that require strict adherence to safety measures.

One of the most dangerous factors that mechanics face when repairing marine diesel engines is high pressure and temperature. Significant pressure is generated in the cooling and lubrication systems of the 6DKRN 80/230 engine, which can cause serious injury if the equipment is not handled properly.

The temperature of the coolant and exhaust gases can also be extremely high, which can cause burns and other injuries.

According to the US National Institute for Occupational Safety and Health (NIOSH), in 2020, approximately 10,000 workers suffered injuries related to high pressure and temperature. In the marine industry, these figures may be even higher, as working with marine engines involves additional risks.

To reduce the risks associated with high pressure and temperature, the following safety measures must be observed:

1. Use special tools and equipment designed for working with high pressure and temperature.

2. Perform regular maintenance of the engine cooling and lubrication systems.

3. Monitor the temperature and pressure in the engine systems.

4. Use protective equipment such as gloves, goggles and clothing made of heat-resistant material.

5. Train personnel in the rules for safe working with high pressure and temperature.

Another serious risk associated with the repair of marine diesel engines is toxic gases and vapours. When working with a 6DKRN 80/230 diesel engine, emissions of carbon monoxide, nitrogen oxides and other toxic substances are possible. Inhalation of these substances can lead to poisoning, headaches, nausea and even loss of consciousness.

According to the World Health Organisation (WHO), approximately 4 million people die each year from air pollution, including toxic gases emitted by marine engines.

To reduce the risks associated with toxic gases and vapours, the following safety measures must be observed:

1. Repair the diesel engine in a well-ventilated area.

2. Use respirators with filters designed to protect against toxic gases and vapours.

3. Monitor the level of toxic substances in the air in the work area.

4. Train personnel in the rules for safe handling of toxic substances.

6DKRN 80/230 low-speed marine diesel engines run on fuel, which poses a potential explosion hazard if handled improperly. Fuel tanks and pipelines contain significant amounts of combustible vapours that can ignite in the presence of an ignition source.

According to the International Maritime Organisation (IMO), there were more than 1,000 explosions on ships between 1990 and 2020, resulting in more than 1,000 deaths.

To reduce the risks associated with the potential danger of explosion, the following safety measures must be observed:

1. Repair the diesel engine in a well-ventilated area.

2. Use spark-proof equipment and tools.

3. Prohibit smoking and the use of open flames in the work area.

4. Regularly check the ventilation system of fuel tanks and pipelines.

5. Train personnel in the rules for safe working with combustible materials.

Repairing a 6DKRN 80/230 marine diesel engine is a complex procedure that requires highly skilled and experienced mechanics. If repair work is performed incorrectly, mechanical damage to the engine may occur, which can lead to malfunctions and even accidents.

According to the American Bureau of Shipping (ABS), about 20% of engine-related accidents on ships occur due to improper repair work.

To reduce the risks associated with mechanical damage, the following safety measures must be observed:

1. Use only specialised tools and equipment designed for repairing marine diesel engines.

2. Repair the engine in accordance with the manufacturer's instructions.

3. Use protective equipment such as gloves and goggles.

4. Train personnel in the rules for safe operation of engine mechanisms.

5. Perform regular engine maintenance.

5.2. Calculations of noise and vibration levels of the 6DKRN 80/230 ship engine in the engine room of a container ship

The main engines on a ship are among the noisiest mechanisms in the engine room and are in fact the main source of noise and vibration. Excessive noise and vibration levels have a negative impact on the health of the container ship's engine room personnel, reducing their productivity and dulling their attention.

High noise levels over long periods of time cause fatigue in the human auditory system and can sometimes lead to partial or even complete hearing loss. Therefore, controlling and ensuring sanitary noise levels in the engine room is an extremely important task.

Vibration in the engine room of a container ship is caused by the dynamic imbalance of the masses of the crankshaft mechanism of the main engine. Local vibration causes vascular spasms and impairs blood circulation in the person servicing the main engine. General vibration on a ship with a frequency of 0.7 Hz causes seasickness in humans, and vibration with a frequency of 4...30 Hz can cause damage to the shoulder girdle, as well as most of the internal organs of a person due to the resonance phenomenon.

The noise level in the engine room of a container ship from the main low-speed engine 6DKRN 80/230 is determined by the formula

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

where $n_н$ is the nominal rotational speed of the crankshaft of the low-speed marine engine 6ДКРН 80/230, $n_н$ = 104 rpm; n is the operating rotational speed of the crankshaft of the low-speed marine engine 6DKRN 80/230, n = 104 rpm; N_e is the rated power of the low-speed marine engine 6DKRN 80/230, N_e = 21600 kW.

$$L = \left[54 + 10\lg\left(104 + 21600^{0.55}\right) + 30\lg\left(\frac{104}{104}\right)\right] = 79{,}39 \text{ dB}$$

The noise level in the engine room of the container ship from the main low-speed engine 6DKRN 80/230 is within acceptable limits.

To reduce aerodynamic noise in the engine room, silencers of various designs are used on the engines. The use of a silencer on the main engine reduces the overall noise level in the engine room of a container ship by 10 to 12 dB.

To reduce airborne noise in the engine room from the external surfaces of the main engine, sound-insulating covers are used, which reduce the overall noise level by 10 to 15 dB. To reduce mechanical noise from the main engine, it is necessary to reduce the gaps between its parts and components, improve the technology of manufacturing and processing engine parts, and try to use materials with greater noise absorption.

A sufficiently effective and important way to reduce the harmful effects of noise in the engine room of a container ship on the human body is to use personal protective equipment such as earplugs and headphones, which can reduce the perceived sound by 15-20 dB.

The vibration level in the engine room of a container ship from the main low-speed engine 6DKRN 80/230 is determined by the formula:

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

where m is the mass of the main low-speed engine 6DKRN 80/230, m = 705000 kg

$$L = 44 + 10\lg\left(\frac{104 \cdot 21600^{0.55} \cdot \left(\frac{1+21600}{705000}\right)}{1+\left(\frac{1}{1500}\right)^3 \cdot \frac{705000}{21600}} + 30\lg\left(\frac{104}{104}\right)\right) = 72{,}87\ \text{dB}$$

The vibration level in the engine room of the container ship from the main low-speed engine 6DKRN 80/230 does not exceed the standards.

An effective way to reduce vibration in the engine room of a container ship from the main low-speed engine 6DKRN 80/230 is to install vibration dampers in areas of increased vibration, as well as to strengthen the fastening of doors and hatches in order to reduce their shaking.

5.3. Conclusions to the part

Hazardous and harmful factors affecting humans during the operation, maintenance and repair of the main low-speed engine 6DKRN 80/230 in the engine room of a container ship were analysed. The noise and vibration levels of the main low-speed engine 6ДКРН 80/230 were calculated, according to which the noise and vibration levels are acceptable and amount to 79.39 and 72.87 dB, respectively. An analysis was performed and measures were proposed to reduce the noise and vibration levels in the engine room of the container ship.

CONCLUSION ON THE WORK

Master's thesis on the topic: ***«Modernisation of the fuel system of a low-speed marine engine through the use of a hydraulically driven system»*** has all the necessary sections and drawings.

The first section provides a brief description and main characteristics of the container ship and the main low-speed marine engine 6DKRN 80/230. It has been determined that the main engine 6DKRN 80/230 can provide the necessary operating parameters of the ship and fits well in the engine room of the container ship.

The second section establishes the main parameters of the operating cycle of the modernised low-speed engine 6DKRN 80/230.

The third section explores possible ways to modernise the fuel system of the low-speed 6DKRN 80/230 engine and develops a schematic diagram of the fuel system with a high-pressure fuel pump with hydraulic plunger drive and electronic fuel supply control. The parameters are calculated and the main elements of the PNVT with a hydraulically driven plunger are designed. The modernisation of the 6DKRN 80/230 low-speed engine through the use of a hydraulically driven system has reduced the specific effective fuel consumption by 3.0 g/(kW·h) and increased the effective power by 2880 kW (15.4%).

The fourth section of the technical and economic justification provides a theoretical assessment of the economic feasibility of modernising the fuel system of the 6DKRN 80/230 main ship engine. According to the calculations, the annual economic effect for the consumer from using the modernised low-speed 6DKRN 80/23 engine on the ship, compared to the serial engine, is 3,347,969.983 UAH, and the modernisation of the serial engine will pay for itself in just 0.14 years.

The fifth chapter develops the basic requirements for occupational safety during engine repair and maintenance. An analysis of the harmful effects of vibration and noise on the environment and human health is provided.

REFERENCES

1. Mollenhauer, K. Hanbbook of diesel engines / K. Mollenhauer, H. Tschoeke. – Berlin : Springer-Verlag Berlin Heidelberg, 2010. –634 p.

2. Heywood John B. Internal Combustion Engine Fundamentals / John B. Heywood. - New York.: McGraw Hill New York, 1988. – 930 –. – ISBN 0-07-100499-8

3. Hunkert, J. Liang, M. Willmann // SAE Papers. – 1997. – 972686. – P. 2133–2145.

4. An Innovative 3D-CFD-Approach towards Virtual Development of Internal Combustion Engines. / Chiodi M./ Stuttgart. 2011. 275 pp.

5. Realizing Future Trends in Diesel Engine Development / B. Georgi, S.

6. Марченко А. П., Рязанцев М. К., Шеховцов А. Ф. Двигуни внутрішнього згоряння : серія підручників у 6 т. Т. 1. Розробка конструкції форсованих двигунів наземних транспортних машин / ред. проф. А. П. Марченко та засл. діяча науки України проф. А. Ф. Шеховцова. Харків : Прапор, 2004. –384 с.

7. Суднові двигуни внутрішнього згоряння: Методичні вказівки до виконання графіко-розрахункових робіт / В. С. Наливайко, С. Г. Ткаченко, В.С. Хоменко. – Миколаїв: НУК, 2012. – 72 с

8. Суднові двигуни внутрішнього згоряння: Підручник / В.С. Наливайко, Б.Г. Тимошевський, С. Г. Ткаченко. Миколаїв : видавець Торубара В. В., 2015. – 332 с.

9. Конструктивні вузли та системи суднових двигунів внутрішнього згоряння : навчальний посібник / В.С. Наливайко, Б.Г. Тимошевський. – Миколаїв : Нук, 2013. – 100 с.

10. Митрофанов О.С., Проскурін А.Ю. Основи експлуатації, обслуговування та ремонту двигунів внутрішнього згоряння : навч. посіб. Миколаїв : НУК, 2018. 151 с.

11. Наливайко В. С., Ткаченко С. Г., Хоменко В. С. Конструкція і динаміка двигунів внутрішнього згоряння. Програма, методичні вказівки та контрольні

завдання для студентів заочної форми навчання. – Миколаїв: НУК, 2008. – 64 с.

12. ДСТУ 7239:2011 «Система стандартів безпеки праці. Засоби індивідуального захисту. Загальні вимоги та класифікація».