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 p.

QUALIFICATION WORK

MODERNISATION OF THE FUEL SYSTEM OF A MEDIUM-SPEED ENGINE BY USING ELECTRICALLY CONTROLLED INJECTORS

Speciality 142 – Power engineering

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

Head of work Boris TYMOSHEVSKY

Education applicant Vladimir IOVENKO

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

Vladimir Andreyevich Iovenko
(surname, first name, patronymic)

1. Work topic *Modernisation of the fuel system of a medium-speed engine by using electrically controlled injectors*

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 – 5430 kW; crankshaft speed – 500 min^{-1}. The engine is installed in the engine room of the vessel. The engine power is increased by using electrically controlled injectors.*

5. Contents of the explanatory memorandum (list of questions to be developed)
1. Application of the object of research in the ship power plant. Determination of requirements for the application of internal combustion engines as part of the object of use. 2. Calculation of the operating cycle for the nominal operating mode of the diesel engine. 3. Development and design of a fuel injector with an electromagnetic valve. 4. Economic justification of modernisation. 5. Analysis and development of labour and environmental protection measures for engines.

6. List of graphic material (with an exact indication of the required drawings)
1. General view of the ship. 2. Cross section of the 6HP 46/58 engine. 3. Hydraulic diagram of the accumulator fuel system. 4. Electrically controlled injector 5. High pressure fuel pump. 6. Throttle valve for controlling the fuel supply.

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.	*Development and design of a fuel injector 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 ____________ *Vladimir IOVENKO*
(signature) (surname and initials)

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

АНОТАЦІЯ

Робота присвячена модернізації паливної системи середньообертового двигуна 6ЧН 46/58 шляхом використання електрокерованих форсунок.

Розглянуто можливість застосування дизеля 6ЧН 46/58 на суховантажному судні. Встановлено, що дизель може забезпечити необхідні експлуатаційні параметри та добре розміщується в приміщенні машинного відділення.

Виконано розрахунок робочого циклу для номінального режиму роботи суднового дизельного двигуна 6ЧН 46/58 з штатною паливною системою та з модернізованою (з використанням електрокерованих форсунок та акумуляторів палива). Встановлено, що модернізований двигун за основними ефективними та екологічними показниками значно переважає двигун прототип (підвищення потужності склало 48,5 кВт, а питома ефективна витрата палива знизилася на 4,4 г/кВт·год).

Проаналізовано перспективні напрямки розвитку паливних систем сучасних середньообертових суднових дизелів. Розроблено та розраховано паливну систему та її основні елементи. Отримано основні геометричні та експлуатаційні параметри.

Встановлено, що річний економічний ефект для споживача від використання модернізованого двигуна, порівняно з базовим, становить 330393,07 грн. Модернізація окупиться за 5 місяців.

Проведено детальний аналіз небезпечних і шкідливих факторів для суднових механіків та іншого персоналу в машинному відділенні судна. Проведено узагальнений розрахунок штучного освітлення в машинному відділенні судна, а також проаналізовано вплив шкідливих компонентів у вихлопних газах на здоров'я людини.

Магістерська робота виконана англійською мовою на 74 аркушах розрахунково-пояснювальної записки. Використано 14 джерел. Графічна частина представлена на 6 кресленнях формату А1.

Ключові слова: двигун внутрішнього згоряння, паливна система, електрокерована форсунка, питома ефективна витрата палива, екологічні показники.

ABSTRACTS

The paper is devoted to the modernisation of the fuel system of the medium-speed engine 6CHN 46/58 by using electrically controlled injectors.

The possibility of using the 6CHN 46/58 diesel engine on a dry cargo ship is considered. It is established that the diesel engine can provide the required operational parameters and is well placed in the engine room.

The operating cycle was calculated for the nominal operating mode of the 6CHN 46/58 marine diesel engine with the standard fuel system and with the upgraded one (using electrically controlled injectors and fuel accumulators). It was found that the upgraded engine significantly exceeds the prototype engine in terms of its main efficiency and environmental indicators (power increase was 48.5 kW, and specific effective fuel consumption decreased by 4.4 g/kWh).

The paper analyses the prospective directions of development of fuel systems for modern medium-speed marine diesel engines. The fuel system and its main elements are designed and calculated. The main geometrical and operational parameters are obtained.

It is established that the annual economic effect for the consumer from the use of the modernised engine, compared to the basic one, is UAH 330393.07. The modernisation will pay off in 5 months.

A detailed analysis of hazardous and harmful factors for ship's mechanics and other personnel in the ship's engine room was carried out. A generalised calculation of artificial lighting in the ship's engine room was carried out, and the impact of harmful components in exhaust gases on human health was analysed.

The master's thesis is written in English on 74 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, electrically controlled injector, specific effective fuel consumption, environmental performance.

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INTRODUCTION

The requirements for modern diesel engines in terms of power, fuel efficiency and environmental friendliness are becoming increasingly stringent. To meet these requirements, it is essential to ensure good and high-quality mixture formation. To do this, engines must be equipped with efficient injection systems that not only ensure the finest possible fuel atomisation through high injection pressure, but also precisely control the injection torque and the amount of fuel injected.

The project addresses the issue of modernising the engine's fuel system, namely the use of electrically controlled injectors and fuel accumulators, which will increase the injection pressure.

The design goal is to increase the efficiency of the 6CHN 46/58 engine at a given engine speed for a ship's power plant. The specific effective fuel consumption should be reduced, and the effective efficiency and power should increase compared to the baseline. To achieve this, the engine fuel system is modernised, new fuel equipment is added and the injection pressure is increased. The operating cycle and key performance parameters of the prototype and upgraded engines are calculated. A new fuel system is developed and its elements are calculated.

The engine upgrade should provide better engine performance parameters at rated and partial operating conditions, as well as improve the environmental performance of the internal combustion engine. The use of electrically controlled injectors and fuel accumulators undoubtedly leads to an improvement in the efficiency of internal combustion engines, which is an extremely important task today, as there is a steady shortage of hydrocarbons in the world and they should be saved as much as possible.

PART 1
PECULIARITIES OF APPLICATION OF THE ENGINE 6CHN 46/58 IN THE SHIP'S POWER PLANT

1.1. General description of the vessel

The bow of the ship (fig. 1.1) has ice reinforcements of the hull. The vessel's architectural and structural type is a four-metre twin-screw motor ship with a limited draft, diesel gear power plant, tank, yut, double sides, aft engine room and accommodation superstructure, and a sloping transom stern. The main characteristics of the vessel are presented in table 1.1.

The stability of the vessel in all operational loading cases meets the requirements of the Register Rules for vessels of the I limited navigation area, unsinkability is ensured in case of flooding of any one compartment. A roll-ballast system is provided to assess stability during loading and unloading operations. The rolling period, depending on the load, is 9...11 seconds. To reduce the bumping motion, 300 mm high scull keels and passive dampeners are provided.

The vessel is designed to carry international standard containers, packaged cargo, timber in bulk and in packages, various general, bulk and dry bulk cargoes, including grain, coal and dry lightly flammable cargoes.

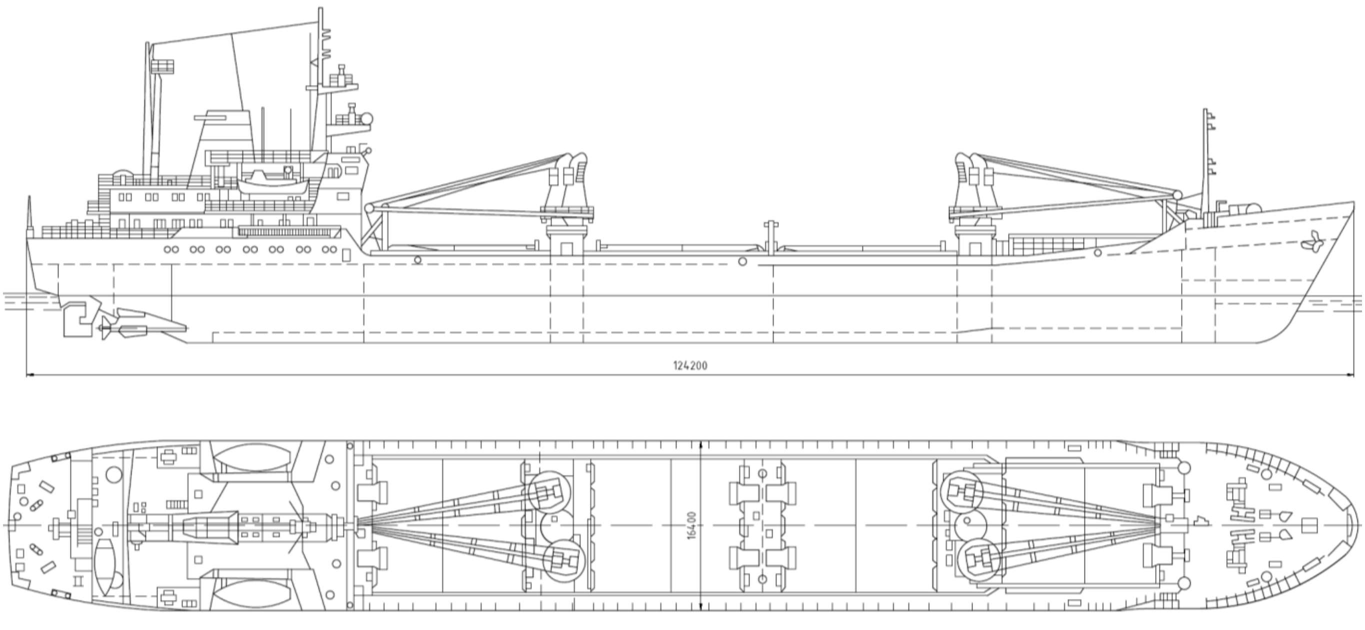


Figure 1.1 – General view of the vessel

Table 1.1 – Main characteristics of the vessel

№ n.o.	Parameter	Value
1	The longest length, m	124,2
2	Length between perpendiculars, m	113,9
3	Width, m	16,4
4	Board height, m	7,5
5	Deposits along the KWL, m	4,5
6	Draft by cargo grade, m	5,5
7	Deadweight at a draft of 4.5 m, tonnes	3370
8	Deadweight at a draft of 5.5 m, tonnes	5030
9	Stroke speed, knots	12,4

With a draft of 4.5 metres and a range of 4,000 miles, the net displacement is 2,940 tonnes, with a range of 6,000 miles, it is 2,860 tonnes. With a draft of 5.5 m and a cruising range of 4000 miles, the net cargo capacity is 4000 tonnes, with a cruising range of 6000 miles – 4520 tonnes. The total deadweight capacity of the holds is 6680 m^3, and the bulk capacity is 6800 m^3. To eliminate transverse ‘pockets’ and speed up loading and unloading operations, the holds have double sides. The vessel is capable of carrying 165 containers (20-foot), including 111 containers weighing 20 tonnes each in the cargo holds and 54 containers weighing 16 tonnes each when the cargo hatches are closed. Timber with a specific cargo volume of 2.32 m^3/t can be transported up to 3,770 tonnes, including 2,875 tonnes in the holds and 895 tonnes on the upper deck. The containers (20- and 40-foot) in the holds and on the hatch covers are mounted on welded fittings. The deck scaffolding is secured by stanchions installed in slots along the bulwarks in the area of holds 2, 3 and 4 and along the longitudinal braces of the cargo hatch of hold 1. The deck scaffolding is secured with lanyards. There are shackles on the upper deck for securing deck cargo. Cargo operations are carried out by four 8/3.2 tonne electro-hydraulic cranes installed in pairs between holds No. 1 and 2, 3 and 4. The boom reach over the nearest side is 13 m and 6 m over the opposite side. The cranes can operate with the rated load at a heel of the vessel up to 5° and a pitch of up to 2°.

The opening and closing of each watertight double-leaf folding manhole cover is carried out by means of a hydraulic drive consisting of four hydraulic cylinders. The covers are fastened in the open (vertical) position by means of locking pins. The strength of the hatch covers is designed for a uniformly distributed load of 1,75 t/m and the installation of 20- and 40-foot containers in one tier on the hatch covers of holds No. 1 and 4 and in two tiers on the hatch covers of holds No. 2 and 3.

In the middle part of the vessel, the bottom, double bottom and upper deck are built in a longitudinal system, while the longitudinal bulkheads, sides, platforms, bow and stern bulkheads are built in a transverse system. The permissible distributed load on the double bottom deck is 7,4 t/m^2 and on the upper deck side passages is 4,0 t/m^2. The main engines of the vessel are two four-stroke, six-cylinder, in-line, gas-turbocharged 6CHN 46/58 engines with a total output of 11,000 kW at a rotation speed of 500 rpm.

The power from each diesel gearbox is transmitted through the intermediate and propeller shafts to a fixed-pitch propeller with a diameter of 2,5 m. The propellers are four-bladed, designed for ice conditions and made of corrosion-resistant steel grade 1X14NDL. The vessel has one spare propeller for each direction of rotation. The propeller and intermediate shafts are connected to each other and to the gearbox shaft of the main diesel gearbox unit by means of flange couplings. The propeller shafts are located aft. The main engines are fuelled by diesel fuel of L-0.2 and L-0.5 grades and motor fuel of DT grade. To meet domestic and technical steam needs, there are 2 auxiliary boilers of the KVA 1.0/5-M brand with a steam capacity of 1 tonne per hour at an operating pressure of 0,5 MPa, fired by diesel fuel, and 2 vertical water-tube recycling boilers of the KUP 80C brand with a steam capacity of 0,75 tonnes per hour, using the heat of the exhaust gases of the main engines.

The ship's electricity needs are met by three diesel generators DGR 200/750 with a capacity of 200 kW and a rotation speed of 12.5 s^{-1}. The emergency power source is a 50 kW DGA 50M-9R diesel generator with a rotation speed of 25 s^{-1}.

The main engines and other engine room equipment are controlled and monitored from the soundproof control centre. The vessel's motion is controlled from the wheelhouse, from where the pre-prepared main engines are started, their operating modes are changed, and the engines are reversed and stopped using the automatic control system. The level of automation and control meets the requirements of the A2 Registry Rules, providing the possibility of unattended maintenance of the engine room.

The vessel's steering is provided by two streamlined semi-balanced rudders of 7 m^2 each, driven by electro-hydraulic machines P16, which enable both rudders to be turned from 35° on one side to 30° on the other in 28 seconds at full forward speed. The wheelhouse is equipped with autosteering equipment. Rudder position indicators are installed in the tiller room, control centre and wheelhouse.

The anchoring and mooring facilities are equipped with two 2,5 tonne Hall anchors with 46 mm calibre anchor chains, 250 m long each; two bow automatic mooring winches manufactured by Poland with anchor tacks, which provide anchor lifting from a depth of 100 m at a speed of 0,17 m/s; one stern automatic mooring winch manufactured by Poland. Each automatic mooring winch provides a pulling force of 78,5 kN on the mooring drums. The speed of mooring rope retrieval at the rated pulling force is 0,31...0,35 m/s.

The rescue equipment includes 2 motor plastic lifeboats with a capacity of 37 persons each, serviced by gravity davits. The vessel is equipped with a working plastic motor lifeboat RSPM 5.5. Loading of provisions, launching and lifting of the workboat is carried out by a 1,6 tonne lifting boom mounted on a vertical column installed in the aft part of the deck of the running bridge. The boom outreach is 2,5 m.

The vessel has 2 block cabins with bathrooms, 2 block cabins with showers, 8 single cabins, 5 double cabins and a pilot's cabin. To ensure normal living and recreational conditions for the crew, the vessel has a crew mess and dining room, crew and crew lounges, a library, a photo laboratory, a sports cabin, a sauna, a sufficient number of showers and a laundry room. The crew's dining room is adapted for film screenings. Cinema equipment is installed in a separate room. The living and public areas are equipped with an air conditioning system with individual room temperature control, served by two automated central air conditioners.

The vessel is supplied with fresh water and fresh water systems. The fresh water supply is stored in two insertion tanks, the inner surface of which is made of stainless steel, and in three tanks made as part of the hull and painted with primer varnish. Fresh water is replenished from the D4U desalination plant with a capacity of 10 tonnes per day. A mineraliser MV-5 is used for mineralisation of desalinated water. The water for domestic needs is taken by a sanitary unit from the Kingston cofferdam.

Bilge oil-containing water is treated by a separation unit USA-1.6 with a purification rate of 15 parts of oil per 1 million parts of mixture. Wastewater and household water are treated and disinfected by the EOS-15 electrochemical treatment unit, after which they are discharged overboard. The sludge is collected in a 5,5 m^3 waste tank, which is designed to drain untreated water in the event of a plant failure. For garbage collection, 3 containers with a total capacity of 1,5 m^3 are installed in the aft part of the yutu deck. The vessel is equipped with modern radio and electrical navigation equipment of domestic production.

1.2. Construction description of the engine type 6CHN 46/58

The engine 6CHN 46/58 is a four-stroke medium-speed engine with a power of 5430 kW (fig. 1.2).

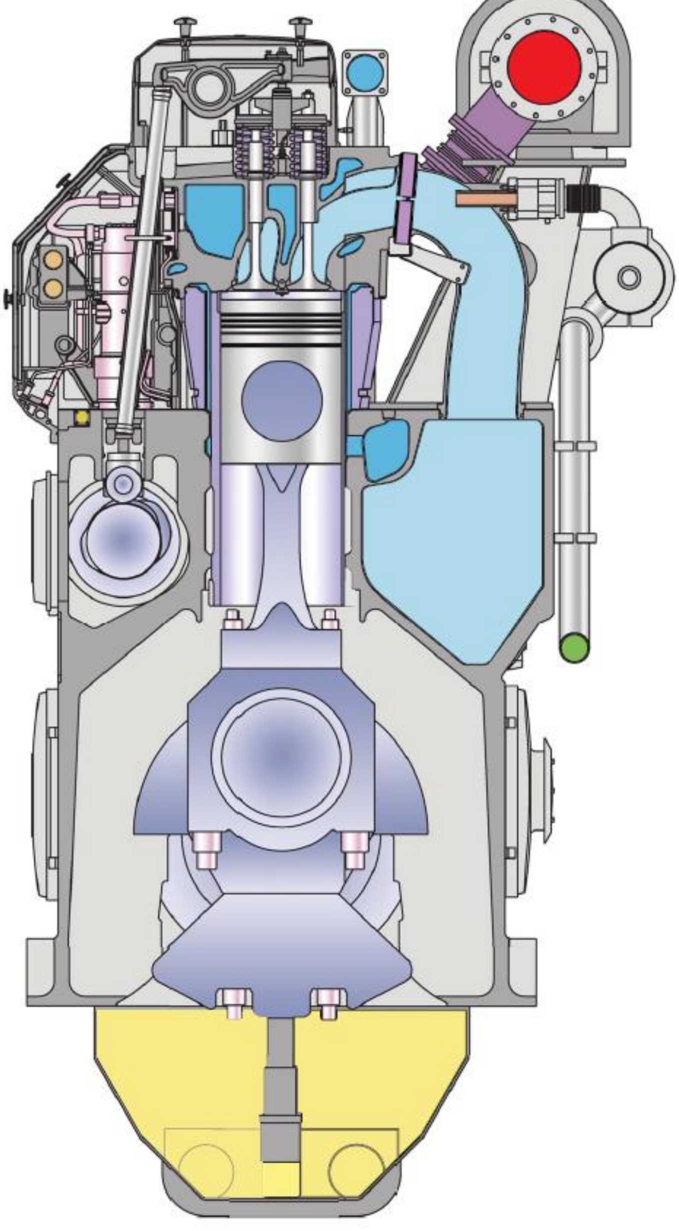


Figure 1.2 – Cross-section of the engine 6HP 46/58

The rotational speed is 500 min^{-1}. The piston speed is 9,7 m/s. The average effective pressure is 2,25 MPa. Engine weight is 95 t. Cylinder diameter – 460 mm, piston stroke – 580 mm. Fuel consumption – 186 g/(kWh).

The piston (fig. 1.3) is designed with a low friction coefficient. The skirt is made of cha-won with spheroidal graphite, the bottom is steel. The piston has a special cooling channel designed to provide efficient cooling and high rigidity for the piston bottom. The design can operate at pressures above 200 bar.

The hardened top ring groove ensures a long service life.

The low coefficient of friction is ensured by the skirt lubrication system:

– a well-distributed, clean oil film that eliminates the risk of abrasion

(scoring) of the piston ring and reduces the wear rate;

– oil rings and grooves are free from corrosive combustion products;

– hydraulically damped inclined movements provide an oil pad between the liner and piston, resulting in reduced noise and wear.

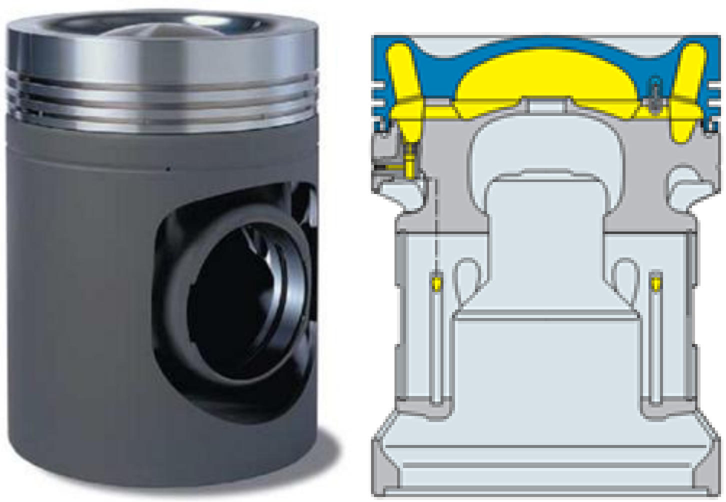


Figure 1.3 – The piston

The design of the crankshaft (fig. 1.4) allows for high combustion pressures and stable bearing loads.

The crankshaft is forged as a single piece and fully machined on a machine tool. The shaft is rigid due to the moderate diameter-to-stroke ratio and the large journal diameters. The crankshaft has built-in counterweights on each crank arm.

Figure 1.4 – Crankshaft

The connecting rod (fig. 1.5) of a marine design consists of three parts, where the combustion forces are distributed over the entire bearing surface and where the relative movement between the mating surfaces is minimal. The three-piece design also reduces the duration of overhauls.

Figure 1.5 – Connecting rod

Overhaul of the piston is possible without dismantling the large (bloodworm) bearing and the bearing can be inspected without dismantling the piston. All nuts are tightened using hydraulic tools.

Due to the special design with a relatively high ring travel, the deformations in the cylinder are very small. To prevent polishing of the liner (fig. 1.6), it is equipped with an oil ring in the upper part. The purpose of this ring is to calibrate the carbon deposit on the top of the piston to a thickness small enough to prevent contact between the inner wall of the liner and the deposit on the top of the piston. Polishing the inner bore can lead to localised wear of the liner and increase oil consumption.

The temperature curve in the cylinder liner is not only important during pressure and deformation, but is also crucial for the wear rate of the cylinder liner. The temperature must remain above the dew point of sulphuric acid to prevent corrosion, but at the same time remain low enough to avoid oil stratification. The composition of the bushing material is based on long experience with special grey cast iron alloys, industrially developed for excellent wear resistance and high strength.

Figure 1.6 – Cylinder bushing

The cylinder cap (fig. 1.7) is designed as a rigid box to reduce the peripheral contact pressure between the cylinder cap and the liner.

Figure 1.7 – Cylinder cover

Four clamping bolts are used to tighten the cover, which simplifies maintenance. Effective water cooling of the exhaust valve seats is also used.

The engine uses piston compression rings with a special anti-corrosion coating (fig. 1.8). The rings are measured and profiled for maximum sealing and pressure balance.

Figure 1.8 – Piston rings

The camshaft is constructed of single-cylinder sections with integrated cams (fig. 1.9). The camshaft sections are connected by means of separate support journals, which makes it possible to remove the shaft sections from the camshaft side. The valve tappets are fitted with rollers. The force from the rods is transmitted to the rocker arms, which in turn exert pressure on the fork with guide pins.

Figure 1.9 – Camshaft and valve mechanism

Both the inlet and outlet valves receive forced rotation from the Rotorcaps during each opening cycle. This uniform rotation ensures even temperature distribution and wear of the valves, and keeps the top seal free of deposits. The result is good thermal conductivity.

The bearings (fig. 1.10) have a thick cage to ensure reliability. Bearing loads have been reduced by increasing the diameters of the crankshaft journals and pins as well as the length.

Figure 1.10 – Bearings

The cylinder block (fig. 1.11) is made of nodular cast iron to provide the rigidity and structural reliability required for elastic mounting. The main bearing is of the suspension type with hydraulic bolt tightening. Side bolts add rigidity to the main bearing housing. The in-line engine is equipped with a common air receiver of increased rigidity.

Figure 1.11 – Cylinder block

The engine has a single exhaust manifold with the SPEX system. This system combines the advantages of a pulsed exhaust system with a constant pressure system. At partial load conditions, the exhaust gas is pushed out, which results in better turbine performance.

The integrated modular exhaust system is robust enough to operate at relatively high pressures and pulsation levels.

The turbocharger (fig. 1.12) has the highest possible efficiency and is equipped with pi-plain bearings without water cooling. The turbocharger is equipped with devices for cleaning the turbine and compressor parts. The exhaust discharge valve and air by-pass are used to obtain the required parameters at the operating range of load characteristics and partial loads.

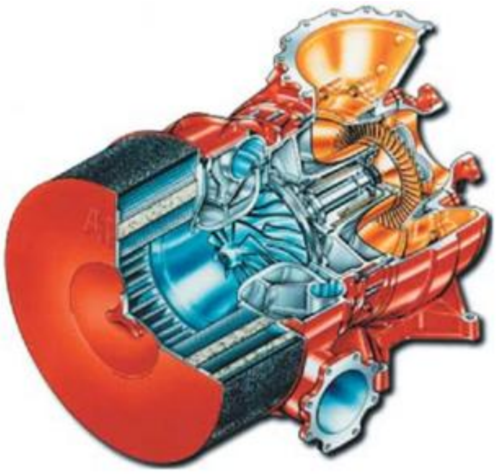


Figure 1.12 – Turbocharger

The entire fuel system is housed in a fully enclosed compartment for maximum safety. All leaks from injectors, pumps and pipes are collected in a closed system.

The high-pressure pump consists of calibrated, replaceable sections (fig. 1.13).

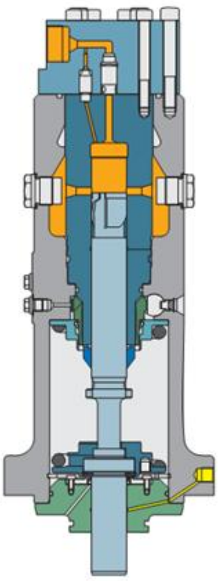


Figure 1.13 – Section of the high-pressure fuel pump

The nozzle is designed with a small heat-absorbing surface on the combustion chamber side. Together with the efficient heat transfer to the cooling water, this elimiates the need for a separate temperature control system for each nozzle.

The cooling system (fig. 1.14) is divided into high and low temperature systems using fresh water. The cooling water of the high temperature system runs continuously to ensure minimal temperature deviations in the cylinder components. To ensure maximum heat recovery, the charge air cooler is divided into high and low temperature sections.

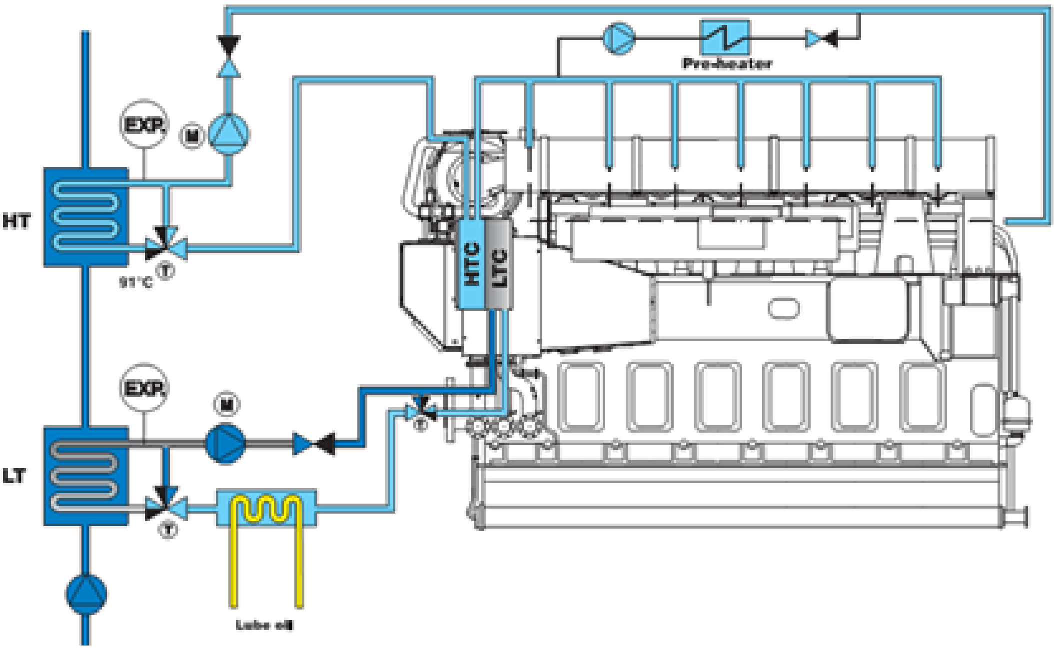


Figure 1.14 – The cooling system

Marine engines have a dry sump (fig. 1.15). The oil is cleaned outside the engine. On its way to the engine, the oil passes through the cooler and oil and cleaning filters.

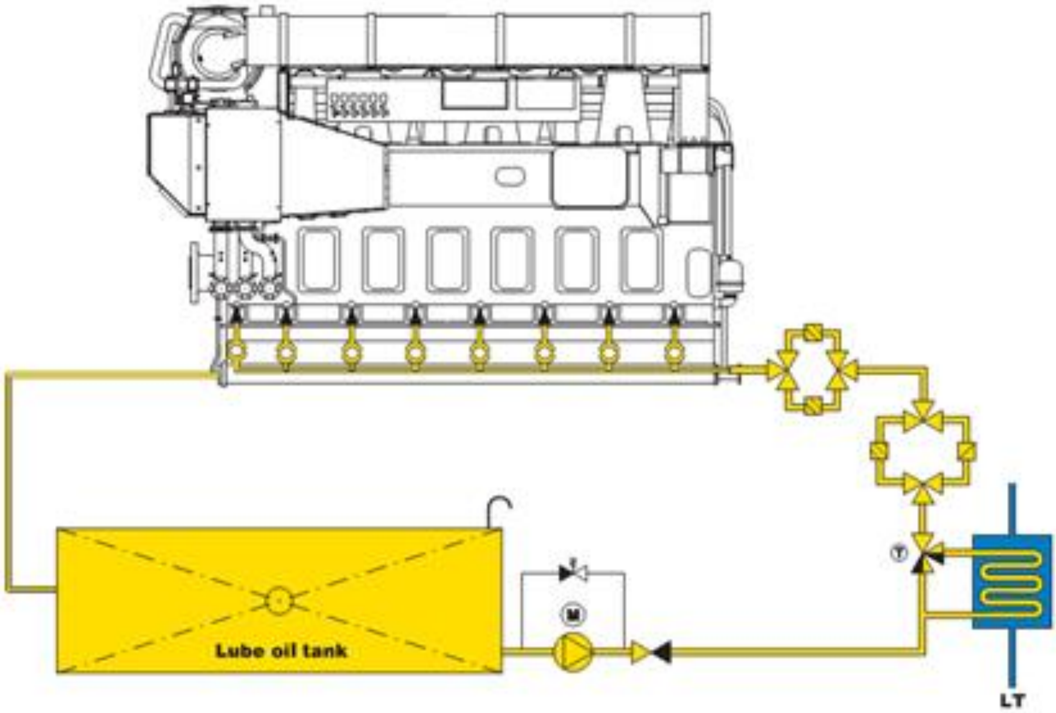


Figure 1.15 – Lubrication system

1.3. Conclusion to the part

The engine meets all the requirements for operation and can be used on the vessel. The cross-section of the engine 6CHN 46/58 and the general view of the vessel are shown in the drawings in A1 format.

To improve the engine's performance, it is necessary to upgrade the engine's fuel system, namely to develop a battery fuel supply system with electrically controlled injectors.

PART 2
CALCULATION OF THE WORKING CYCLE OF THE NOMINAL OPERATING MODE OF THE 6CHN 46/58 ENGINE

2.1. General information

In internal combustion engines, the conversion of chemical energy contained in liquid or gaseous fuels into mechanical work is carried out as a result of a series of periodically repeated processes in the working cylinders. Such a complete set of periodically repeating processes in an engine cylinder is called a working cycle.

The term ‘working fluid’ is used to define a substance or a mixture of substances required for the implementation of the working cycle. It should be noted that the qualitative and quantitative composition of the working fluid in the cylinder during the cycle is not constant. The variability of the working fluid is caused by various circumstances. In particular, quantitative changes in the working fluid are caused primarily by its flow from the intake and exhaust pipelines into the cylinder and vice versa, leakage through the gaps and leaks in the CPG, fuel supply (to the intake pipeline for external mixing or to the cylinders for internal mixing), and the corresponding oxidation reactions of the fuel components.

The combustion of fuel components leads to significant qualitative changes in the composition of the working fluid. While during the intake and compression strokes, the working fluid can be considered as pure air (for diesel engines) or a mixture of fuel vapours and air (for engines with external combustion), starting from the combustion process, it is already a mixture of pure air, vapours and liquid fuel droplets with gradually formed combustion products. The main components of the working fluid are the fuel itself (a carrier of chemical energy) and the oxidiser (air oxygen), which releases this energy.

Engines use as fuel only those combustible substances that easily evaporate and mix with air, and in the process of combustion in the working cylinder do not form harmful solid ash residue. Even a small amount of solid ash residue remaining in the engine cylinder can cause wear on the surfaces of the cylinder mirror, pistons and O-rings. This reduces the engine's service life and can lead to an accident.

Liquid fuels used in internal combustion engines are mostly products of oil refining. They include: diesel fuel (DF), petrol, kerosene, ligroin, diesel oil, etc.

The basic components of any fuel are: carbon C, hydrogen H and oxygen O. Depending on the type of fuel and its production technology, it may contain sulphur S, nitrogen N, lead Pb (as an additive) and some other components. The content of such components in fuel is usually small, so they are neglected when performing thermal calculations of internal combustion engines and taken into account when calculating environmental performance (therefore, the content of foreign components in fuel is limited or even prohibited by standards).

The 6CHN 46/58 engine is a diesel engine (an engine with internal combustion and spontaneous combustion from compression). Diesel engines are characterised by

– high values of the air charge compression ratio (ε=14...18 and above);

– injection of a cyclic dose of fuel directly into the engine's working cylinder at the end of the compression process (when the piston is near the timing);

– spontaneous combustion of the cycle dose of fuel due to the high temperature of the compressed air charge in the cylinder;

– incomplete combustion of the cycle dose of fuel, which can be characterised mainly by the presence of soot and a small amount of carbon monoxide in the exhaust gases.

During engine operation, the pressure and temperature of the working fluid in the cylinder are constantly changing. Changes in the pressure of the working fluid can be experimentally recorded by a special device called an indicator, and the resulting diagram is called an engine indicator diagram. Such an indicator diagram gives a certain picture of the change in the parameters of the state of the working fluid throughout the entire cycle. In this case, the area of the indicator diagram in the “p–V” coordinates corresponds to the cycle operation. Analysing and studying the indicator diagram provides data for analysing the quality of the processes that make up the working cycle, as well as the efficiency of the cycle as a whole.

2.2. Calculation of the nominal working mode of the 6CHN 46/58 engine

The rated mode of a 6CHN 46/58 engine is the mode of its work in which it develops its maximum effective power specified in its manufacturer's passport for a certain period of time at a given crankshaft speed and normal atmospheric conditions (in accordance with ISO 3046-1:2002).

In order to perform adequate duty cycle calculations for the 6CHN 46/58 diesel engine, all the necessary input data for this engine type must be selected correctly.

First of all, the ambient conditions must comply with ISO 3046-1:2002. The ambient air pressure is 100 kPa and the temperature is 25 °C.

The rated engine power, crankshaft speed, charge air pressure, compression ratio, number of cylinders, their diameter, and piston stroke are determined in accordance with the manufacturer's data sheets.

The value of the lower heating value of fuel and its mass composition are selected according to the characteristics of the fuel used in the engine. The mechanical efficiency is chosen from the limits known for 4-stroke ICEs [1–3]. The main parameters of analogue engines are shown in Table 2.1.

Table 2.1 – Main parameters of analogue engines

Diesel name	**Diesel cylinder power, kW**	**Crankshaft rotation speed, min^{-1}**	**Average effective pressure, MPa**	**Specific consumption engine weight, kg/kW**
CHN 40/46	429	520	1,72	204
CHN 36/45	192	500	1,01	214
CHN 31,8/33	736	750	0,75	224
CHN 25/34	55	500	0,79	211
CHN 26/26	110	1000	0,96	204
CHN 21/21	81	1000	1,0	217

The initial data for starting the calculation of the 6CHN 46/58 duty cycle and the calculation results are presented in Table 2.2.

Table 2.2 – Results of engine working cycle calculation with basic fuel equipment

№ n.o.	**Parameter**	**Dimension**	**Value**
1	Crankshaft speed n	rpm	500
2	Power N_e	kW	5494,8

3	Average effective pressure P_e	bar	22,802
4	Torque M_e	Nm	104950
5	Cycle fuel consumption q_c	g	11,445
6	Specific fuel consumption g_e	kg/(kW·h)	0.187
7	Effective efficiency η_e	–	0.451
8	Average indicator pressure P_i	bar	22,356
9	Indicator efficiency η_i	–	0.463
10	Average piston speed S_p	m/s	9.6
11	Ambient pressure P_0	bar	1
12	Ambient temperature T_0	K	288
13	Static pressure behind the turbine P_{0t}	bar	1.010
14	Inhibited flow pressure behind the filter R_{0in}	bar	0.9900
15	Pressure upstream of the exhaust manifold P_k	bar	3.811
16	Temperature upstream of the exhaust manifold T_k	K	326.49
17	Air flow rate through the ICE cylinders G_{air}	kg/s	10.667
18	Efficiency of the supercharging unit η_t	–	0.86
19	Average pressure before the turbine P_t	bar	2.258
20	Average temperature before the turbine T_t	K	767.64
21	Gas flow rate through the ICE cylinders G_{gas}	kg/s	10.825
22	Excess air coefficient α_{sum}	–	2.5724
23	Average pressure of pumping strokes $P_{p.s}$	bar	0.7493
24	Filling coefficient η_v	–	0.977
25	Residual gas coefficient γ_r	–	0.00283
26	Purge coefficient φ_i	–	1.113
27	Average pressure in the intake manifold P_s	bar	3.7992
28	Average temperature in the intake manifold T_s	K	337.37
29	Average wall temperature of the intake manifold T_{ws}	K	377.37
30	Heat transfer coefficient in the intake manifold α_{ws}	W/(m^2·K)	266.97
31	Heat transfer coefficient in the valve channel α_{wsc}	W/(m^2·K)	299.23
32	Average static pressure of the gas P_r	bar	2.1843
33	Average static gas temperature T_r	K	761.06
34	Average gas velocity W_r	m/s	135.56
35	Strouhal number Sh	–	10.842
36	Average wall temperature of the exhaust manifold T_{wr}	K	658.565
37	Heat transfer coefficient in the exhaust manifold α_{wr}	W/(m^2·K)	228.78
38	Heat transfer coefficient in the valve channel α_{wcr}	W/(m^2·K)	979.02
39	Excess air coefficient α	–	2.31
40	Maximum combustion pressure P_z	bar	166.31
41	Maximum combustion temperature T_z	K	1515.2
42	Maximum pressure angle φ_{i_pz}	degrees above UDC	7
43	Maximum temperature angle φ_{i_tz}	degrees above UDC	33
44	Maximum pressure rise rate $dP/d\varphi_i$	bar/grad	3.6151
45	Maximum injection pressure $P_{inject\max}$	bar	826.91
46	Average droplet diameter d_{32} = 19.375	μm	31.185
47	Injection advance θ_{ad} = 14	degrees to UDC	14
48	Duration of fuel injection φ_{i_in}	degrees	36.269
49	Ignition delay period φ_{i_delay}	degrees	1.7359
50	The proportion of fuel that evaporated during the delay period σ	–	0.00018992
51	Duration of combustion φ_{i_comb}	degrees	119.8

52	Hartridge smoke emission Hartridge	–	6.2151
53	Smoke emission on the Bosch scale Bosch	–	0.68166
54	Absolute light absorption coefficient of gas according to ECE, K	1/m	0.15038
55	Particulate matter emission PM	g/(kW·h)	0.15089
56	Carbon dioxide emission CO_2	g/(kW·h)	604.04
57	Wet NO_x concentration NOx,	1/million, (ppm)	795.51
58	NO_x emissions reduced to NO_2, NO_2	g/(kW·h)	5.3184
59	Complex of total NO_x and PM emissions SE	–	1.2627
60	SO_2 emissions SO_2	g/kW·h	0.0000
61	Initial compression pressure P_a	bar	4.1134
62	Compression start temperature T_a	K	368.03
63	End of compression pressure P_c	bar	142.96
64	End of compression temperature T_c	K	962.88
65	Pressure at the beginning of release P_b	bar	14.615
66	Temperature at the beginning of the release T_b	K	1101.3
67	Compression ratio ε	–	14
68	Number of nozzle nozzles $i_{nozzles}$	–	10
69	Nozzle nozzle diameter d_{nozzle}	mm	0.786
70	Start of discharge $\varphi_{s.d}$	deg. to the LDC	59
71	End of exhaust $\varphi_{e.e}$	deg. to UDC	41
72	Inlet start $\varphi_{i.s}$	deg. to UDC	54
73	End of the inlet $\varphi_{i.e}$	deg. behind LDC	31
74	Compressor rotor speed n_{crs}	rpm	12700
75	Compressor power N_{crs}	kW	1650.6
76	Adiabatic efficiency of the compressor η_{ad}	–	0.88
77	Air flow through the compressor G	kg/s	10.667
78	Degree of pressure increase Π_k	–	3.9

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

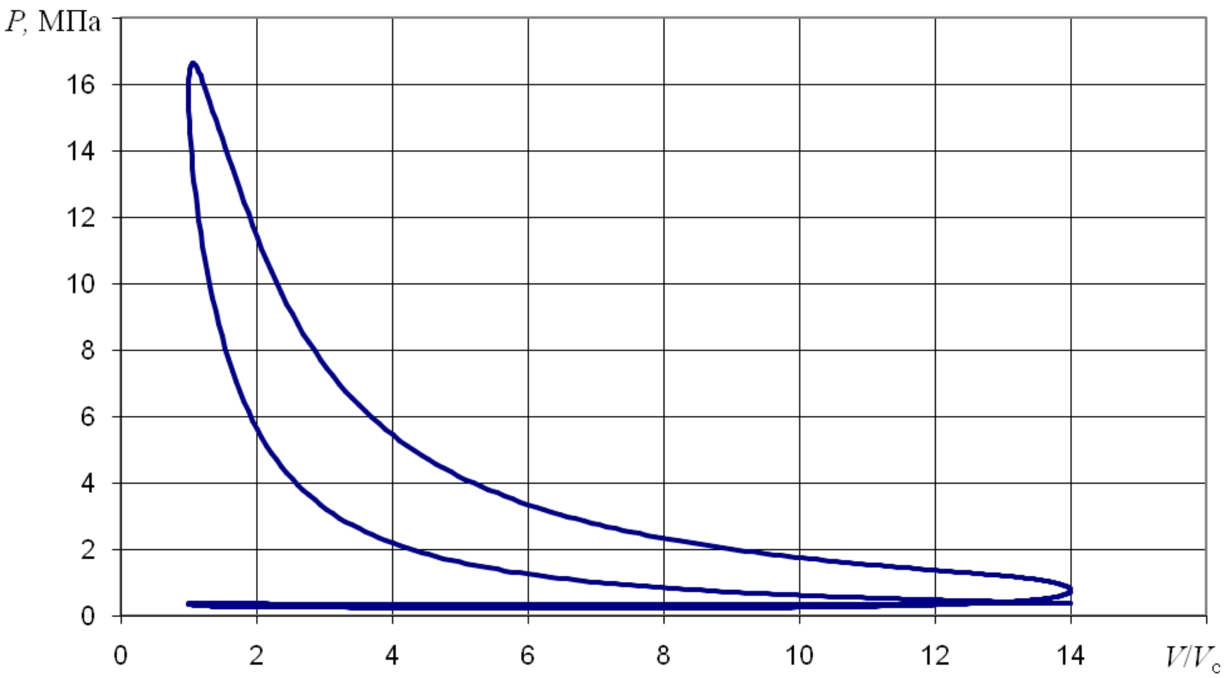


Figure 2.1 – Indicator diagram of the 6HP 46/58 engine with standard fuel equipment

The results of calculating the engine operating cycle using electrically controlled injectors and fuel accumulators are presented in Table 2.3.

Table 2.3 – Results of engine working cycle calculation with basic fuel equipment

№ n.o.	Parameter	Dimension	Value
1	Crankshaft speed n	rpm	500
2	Power N_e	kW	5543,3
3	Average effective pressure P_e	bar	23,004
4	Torque M_e	Nm	105877
5	Cycle fuel consumption q_c	g	11,249
6	Specific fuel consumption g_e	kg/(kW·h)	0.1826
7	Effective efficiency η_e	–	0.453
8	Average indicator pressure P_i	bar	22,58
9	Indicator efficiency η_i	–	0.465
10	Average piston speed S_p	m/s	9.6
11	Ambient pressure P_0	bar	1
12	Ambient temperature T_0	K	288
13	Static pressure behind the turbine P_{0t}	bar	1.010
14	Inhibited flow pressure behind the filter R_{0in}	bar	0.9900
15	Pressure upstream of the exhaust manifold P_k	bar	3.811
16	Temperature upstream of the exhaust manifold T_k	K	326.49
17	Air flow rate through the ICE cylinders G_{air}	kg/s	10.650
18	Efficiency of the supercharging unit η_t	–	0.86
19	Average pressure before the turbine P_t	bar	2.3116
20	Average temperature before the turbine T_t	K	748,56
21	Gas flow rate through the ICE cylinders G_{gas}	kg/s	10.827
22	Excess air coefficient α_{sum}	–	2.612
23	Average pressure of pumping strokes $P_{p.s}$	bar	0.7215
24	Filling coefficient η_v	–	0.977
25	Residual gas coefficient γ_r	–	0.00305
26	Purge coefficient φ_i	–	1.1119
27	Average pressure in the intake manifold P_s	bar	3.799
28	Average temperature in the intake manifold T_s	K	337.21
29	Average wall temperature of the intake manifold T_{ws}	K	377.21
30	Heat transfer coefficient in the intake manifold α_{ws}	$W/(m^2·K)$	266.76
31	Heat transfer coefficient in the valve channel α_{wsc}	$W/(m^2·K)$	298.61
32	Average static pressure of the gas P_r	bar	2.2419
33	Average static gas temperature T_r	K	742,66
34	Average gas velocity W_r	m/s	128,66
35	Strouhal number Sh	–	10.710
36	Average wall temperature of the exhaust manifold T_{wr}	K	646.26
37	Heat transfer coefficient in the exhaust manifold α_{wr}	$W/(m^2·K)$	221.84
38	Heat transfer coefficient in the valve channel α_{wcr}	$W/(m^2·K)$	949.31
39	Excess air coefficient α	–	2.35
40	Maximum combustion pressure P_z	bar	173.04
41	Maximum combustion temperature T_z	K	1546.1
42	Maximum pressure angle φ_{i_pz}	degrees above UDC	8
43	Maximum temperature angle φ_{i_tz}	degrees above	31

		UDC	
44	Maximum pressure rise rate $dP/d\varphi_i$	bar/grad	3.699
45	Maximum injection pressure $P_{inject\mathrm{max}}$	bar	1277.2
46	Average droplet diameter d_{32} = 19.375	μm	27.76
47	Injection advance θ_{ad} = 14	degrees to UDC	13
48	Duration of fuel injection φ_{i_in}	degrees	35.85
49	Ignition delay period φ_{i_delay}	degrees	1.4474
50	The proportion of fuel that evaporated during the delay period σ	–	0.0001499
51	Duration of combustion φ_{i_comb}	degrees	103.8
52	Hartridge smoke emission Hartridge	–	4.9917
53	Smoke emission on the Bosch scale Bosch	–	0.54709
54	Absolute light absorption coefficient of gas according to ECE, K	1/m	0.11979
55	Particulate matter emission PM	g/(kW·h)	0.11377
56	Carbon dioxide emission CO_2	g/(kW·h)	588.5
57	Wet NO_x concentration NOx,	1/million, (ppm)	738.95
58	NO_x emissions reduced to NO_2, NO_2	g/(kW·h)	4.38946
59	Complex of total NO_x and PM emissions SE	–	1.0785
60	SO_2 emissions SO_2	g/kW·h	0.0000
61	Initial compression pressure P_a	bar	4.1138
62	Compression start temperature T_a	K	368.07
63	End of compression pressure P_c	bar	143
64	End of compression temperature T_c	K	963.2
65	Pressure at the beginning of release P_b	bar	14.19
66	Temperature at the beginning of the release T_b	K	1067.3
67	Compression ratio ε	–	14
68	Number of nozzle nozzles $i_{nozzles}$	–	8
69	Nozzle nozzle diameter d_{nozzle}	mm	0.786
70	Start of discharge $\varphi_{s.d}$	deg. to the LDC	59
71	End of exhaust $\varphi_{e.e}$	deg. to UDC	41
72	Inlet start $\varphi_{i.s}$	deg. to UDC	54
73	End of the inlet $\varphi_{i.e}$	deg. behind LDC	31
74	Compressor rotor speed n_{crs}	rpm	12700
75	Compressor power N_{crs}	kW	1647.9
76	Adiabatic efficiency of the compressor η_{ad}	–	0.88
77	Air flow through the compressor G	kg/s	10.65
78	Degree of pressure increase Π_{k}	–	3.9

Based on the cycle calculation, an indicator diagram of the cycle is built (Fig. 2.2).

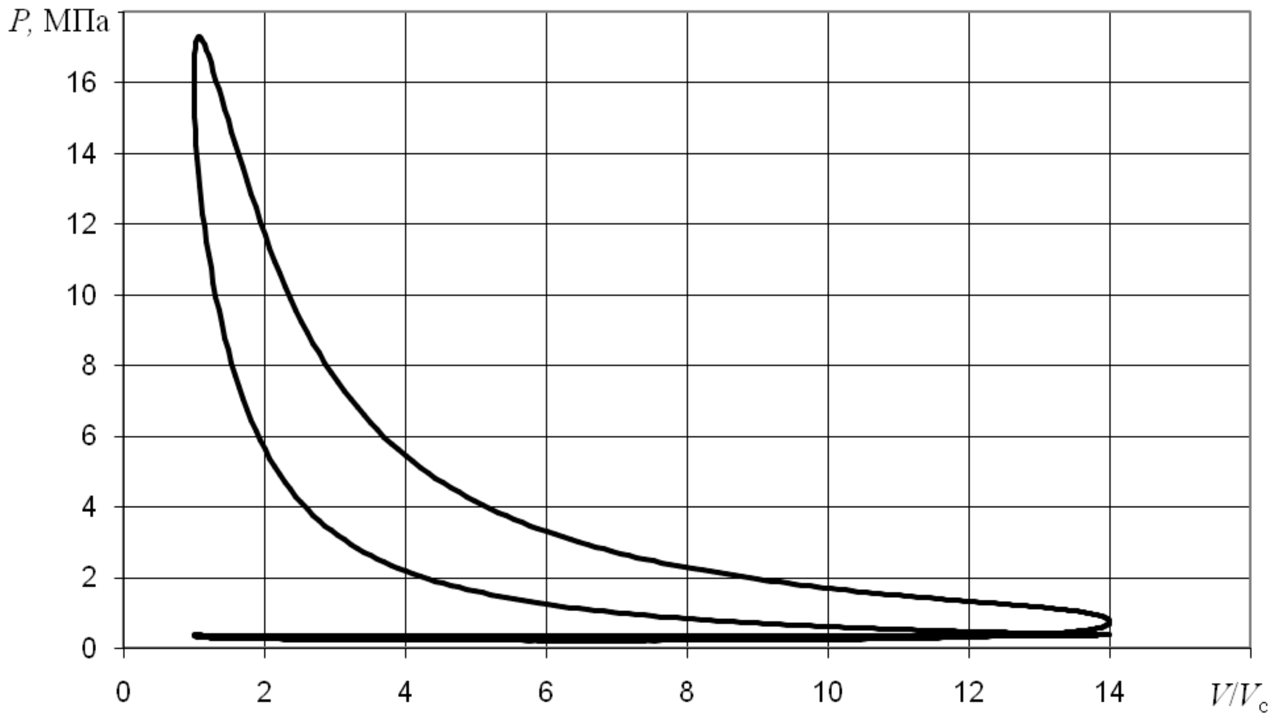


Figure 2.2 – Indicator diagram of the 6ChN 46/58 engine using a fuel accumulator

2.3. Conclusion to the part

As a result of the calculations of the operating cycle of the 6CHN 46/58 engine at different fuel supply pressures using the WC-software, all the main operating cycle parameters that will be needed for further calculations were obtained.

Table 2.4 shows a comparison of the 6CHN 46/58 engine operation parameters when using standard fuel equipment and using fuel accumulators. The table shows that the designed engine with the upgraded fuel system is far superior to the analogue engine in terms of key performance and environmental indicators. Reducing fuel consumption and harmful emissions into the atmosphere is quite relevant nowadays, when the requirements for exhaust gas toxicity and efficiency of internal combustion engines are becoming more and more stringent.

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

Table 2.4 – Comparison of the upgraded engine with the prototype

№ n.o.	Parameter	Standard fuel system	Upgraded fuel system
1	Power N_e, kW	5494,8	5543,3
2	Specific fuel consumption g_e, kg/(kW·h)	0,187	0,1826
3	Average effective pressure P_e, MPa	2,28	2,3
4	Effective efficiency η_e	0,45	0,46
5	Maximum combustion pressure P_z, MPa	16,63	17,3
6	Maximum combustion temperature T_z, K	1515,2	1546,1
7	Pressure upstream of the exhaust manifold P_k, MPa	0,381	0,381
8	Gas flow rate through the ICE cylinders G_{gas}, kg/s	10,82	10,82
9	Air flow rate through the ICE cylinders G_{air}, kg/s	10,66	10,65
10	Maximum pressure rise rate $dP/d\varphi_i$, MPa /grad	0,361	0,369
11	Hartridge smoke emission Hartridge	6,2151	4,9917
12	Smoke emission on the Bosch scale Bosch	0,68166	0,54709
13	Absolute light absorption coefficient of gas according to ECE, K, 1/m	0,15038	0,11979
14	Particulate matter emission PM, g/(kW·h)	0,15089	0,11377
15	Carbon dioxide emission CO_2, g/(kW·h)	604,04	588,5
16	NO_x emissions reduced to NO_2, g/(kW·h)	5,3184	4,8946
17	Complex of total NO_x and PM emissions SE	1,2627	1,0785

PART 3
MODERNISATION OF THE FUEL SYSTEM OF A MEDIUM-SPEED ENGINE 6CHN 46/58 BY USING ELECTRICALLY CONTROLLED INJECTORS

3.1 General information

In recent years, leading manufacturers of medium-speed marine diesel engines have carried out extensive research and development work on battery injection systems for medium-speed engines. As a result of this work, a fundamentally new class of diesel engines has emerged on the market that can meet existing and future standards for the content of harmful substances in exhaust gases. The bulk of the engines were created on the basis of existing models that have proven themselves in operation. At the same time, MAN launched a brand new engine, which was originally designed with a common rail injection system. Subsequently, the injection system developed for this engine was used on other models produced by this company.

When developing common rail systems for medium-speed diesel engines, almost all manufacturers have abandoned the use of a single large-capacity battery, replacing it with several battery modules that provide fuel supply, as a rule, to two or three engine cylinders. This decision is explained by the following considerations:

– medium-speed engines use a wider range of fuels, which require different heating temperatures in the range of 25...150 °C to achieve a given viscosity. This, in turn, causes significant differences in the linear thermal expansion of the battery;

– in a long battery, radial drillings are inevitable for the installation of connecting fittings and pipelines that divert fuel to individual engine cylinders. These drillings are concentrators of significant stresses that weaken the structure. The magnitude of these stresses is proportional to the pressure in the battery and its outer diameter. Reducing the harmful effects of these stresses, for example, by increasing the wall thickness, sometimes leads to an unjustified increase in the size and weight of the fuel supply system as a whole;

– reducing the diameter of the accumulator leads to a reduction in its volume, which, if the cycle feeds are sufficiently large, makes it difficult to achieve the same injection conditions in different cylinders, especially if the fuel is supplied to the accumulator under high pressure from one side. In addition, excessive pressure fluctuations can occur in the accumulator, which disrupts the fuel supply;

– the use of small batteries makes it easier to integrate them into the existing engine design, using standard surfaces for their mounting.

It should be noted that the use of several battery sections does not fully correspond to the name of the common rail system, but traditionally this name is applied to all fuel systems of battery-type diesel engines.

Based on the above, when designing the fuel system of the 6CHN 46/58 engine with electrically controlled injectors, it is necessary to take into account the survey and recommendations of leading manufacturers of medium-speed engines.

3.2. Development of the fuel system of the 6CHN 46/58 engine using electrically controlled injectors

The battery fuel supply system consists of separate modules, which makes it more versatile and allows it to be installed not only on new engines but also used to retrofit existing diesel engines. The latter is achieved by unifying the main elements of the battery system with the elements of the fuel supply systems already used on the engines.

The new proposed system is designed to operate on various fuels, including heavy residual fuel such as HFO. The development of the fuel system for the 6CHN 46/58 engine requires minimal changes to the existing design. In addition, the development of the fuel system must take into account the protection of numerous patents. The general scheme of the accumulator fuel system is shown in Fig. 3.1 General view of the main elements of the system is shown in Fig. 3.2.

Figure 3.2 shows that the arrangement of all system elements is vertical. This makes the system more compact and its components more accessible for maintenance. This arrangement is typical of the standard engine system, so all battery system components have the same connection flanges as the traditional high-pressure pumps they are installed in place of. This reduces the cost of implementing the system and ensures that its cost today is comparable to the cost of the standard system.

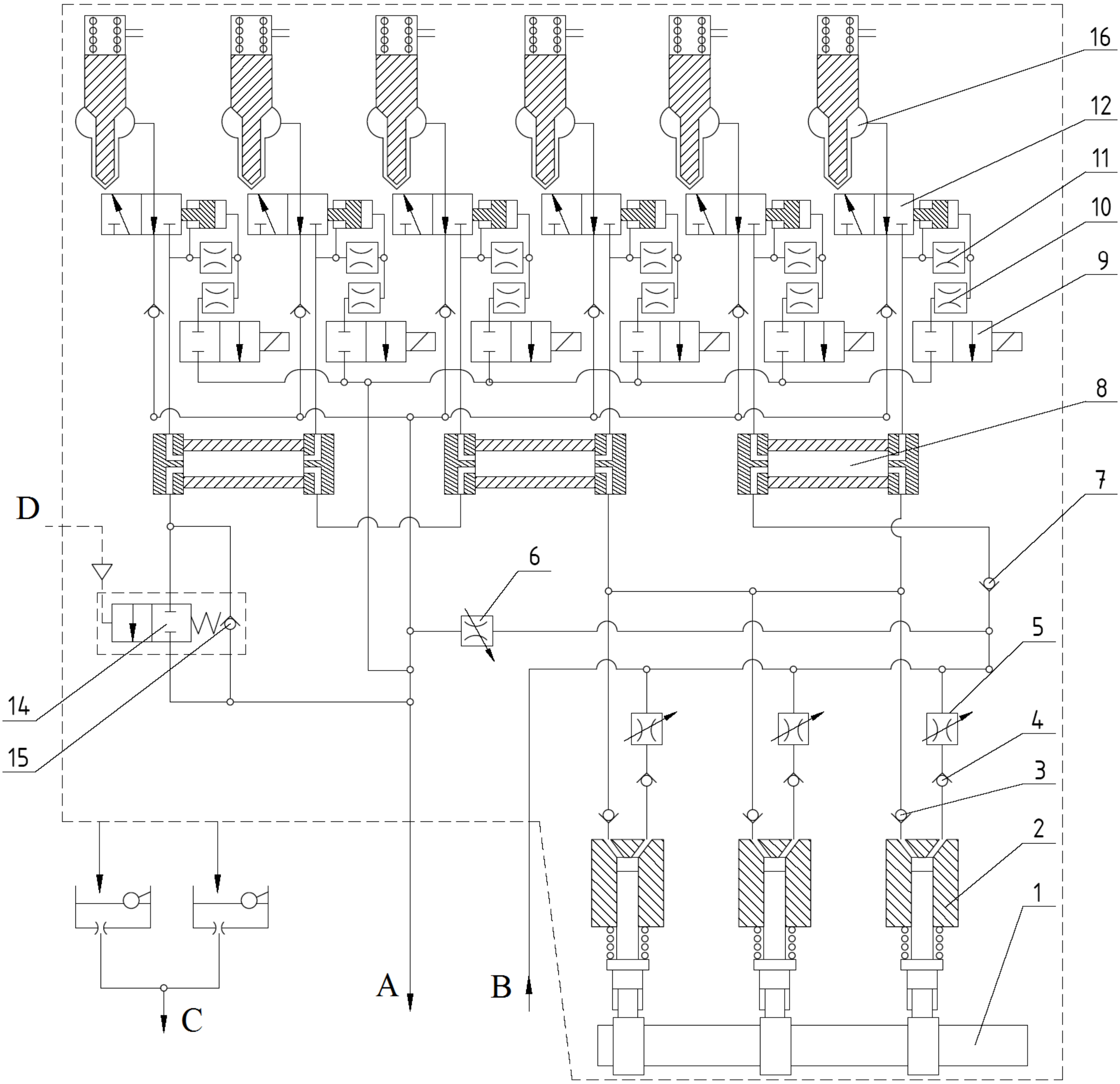


Figure 3.1 – Hydraulic diagram of the engine's storage fuel system:

1 – pump drive shaft; *2* – high-pressure fuel pumps; *3* – discharge valve; *4* – suction valve; *5* – flow control throttle; *6* – pressure drop maintenance throttle; *7* – check valve for increasing flow during system flushing; *8* – accumulator; *9* – starting valve; *10* – starting valve throttle; *11* – throttle of the main flow control valve; *12* – main flow control valve; *13* – check valve of the drain cavity; *14* – system pumping and emergency pressure relief valve; *15* – high pressure limiting valve; *16* – nozzle;

A – fuel supply; *B* – fuel drain; *C* – leakage collection; *D* – control air supply

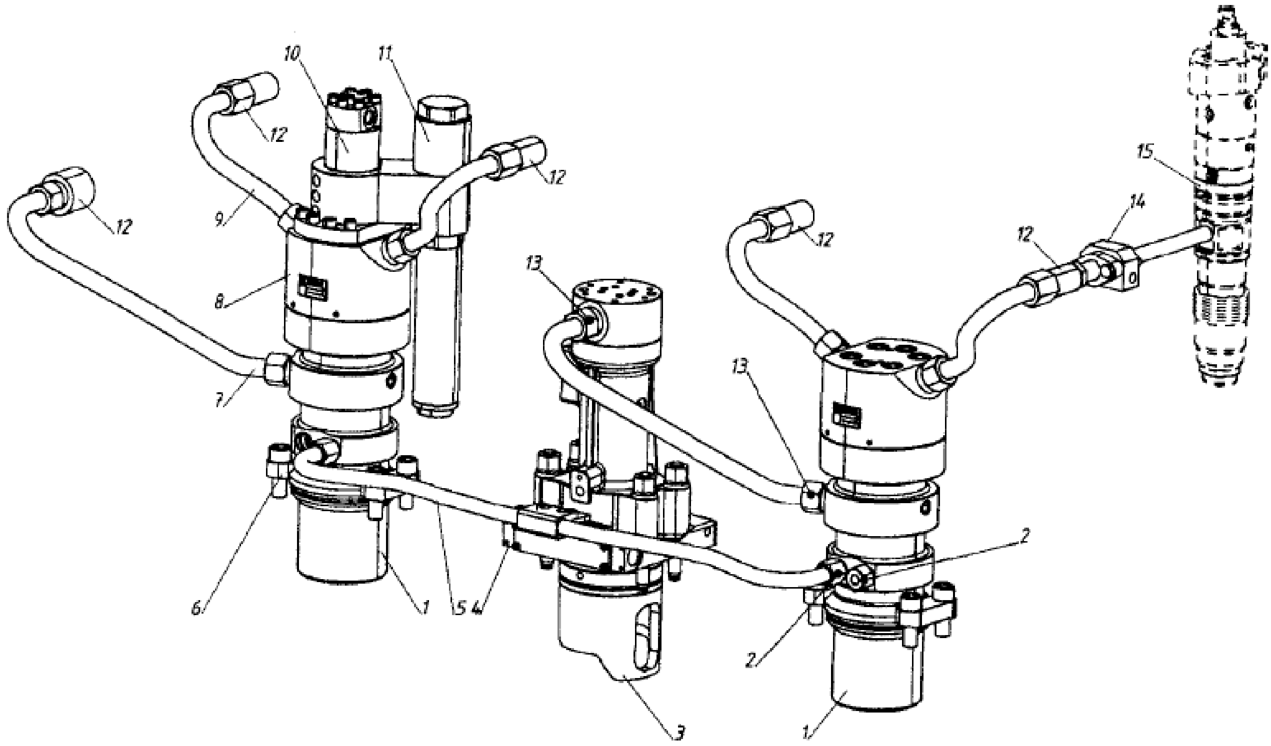


Figure 3.2 – General view of the elements of the accumulator fuel system:
1 – battery installation pad; *2* – connection fittings for high-pressure fuel supply to the battery; *3* – high-pressure fuel pump; *4* – connection flange of the high-pressure pipeline to the fuel pump; *5* – high-pressure pipeline connecting the batteries; *6* – battery connection flange; *7* – high-pressure pipeline from the pump to the battery; *8* – battery housing; *9* – high-pressure fuel supply pipelines to the injectors; *10* – regulator-limiter of the pressure in the battery; *11* – buffer cavity; *12* – injector transition fittings; *13* – high-pressure line connection fittings; *14* – injector fitting; *15* – injector with electromagnetic control of the injection process

The main elements of the fuel system are:

– high-pressure pumps;

– accumulators equipped with a system of valves that control the pressure in the system;

– injectors with electromagnetic flow control.

To drive various valves, a control oil system is provided, which is supplied to the system at a pressure of 22.5 MPa by axial plunger pumps. The oil is pretreated before being supplied.

The presence of a separate control oil system is dictated by the following considerations:

– the reliable operation of the precision pairs of control valves requires a very fine purification of the working fluid, which is easier to achieve with oil than with heavy fuels;

– the use of heavy fuels with a temperature of 150 °C and above as a working fluid can lead to overheating and damage to the solenoid valves located in the injector.

Given that these valves are already located in the cylinder head, where they are exposed to high temperatures, the likelihood of their failure as a result of overheating is quite high. The control oil, having a lower temperature, among other things, cools the valve. The high-pressure fuel pump of the plunger type is designed according to the traditional scheme (Fig. 3.3) and is driven by a camshaft on which a double cam washer is installed. This allowed us to reduce the number of pumps by half, with one pump providing enough fuel to run two engine cylinders. The performance of the pumps is selected in such a way that, if one of them fails, the other pumps will provide a supply sufficient to operate the engine at 80% of the rated power. The fuel pump itself is structurally simpler than its traditional counterpart; the absence of cut-off edges has significantly increased its reliability and compression pressure.

In the proposed fuel system, the pump's operating pressure is in the range of 90...150 MPa, depending on the engine's operating mode. To prevent the plunger from being affected by lateral forces arising from the rolling of the push rod on the cam washer, the pump has a separation between the drive mechanism and the pump section. As a result, only forces directed along the plunger's axis are transmitted to the plunger.

The flow rate of the pumps is controlled by electromagnetically operated butterfly valves (Fig. 3.4). The flow rate at the pump inlet is changed by moving the control cone along the vertical axis. The throttle valve is controlled by an electronic unit, which, depending on the operating mode, changes the flow rate of the pumps and the pressure under which the fuel enters the system. The pressure accumulator is a thick-walled pipe made of high quality steel, closed on both sides with covers that are bolted to the body (Fig. 3.5).

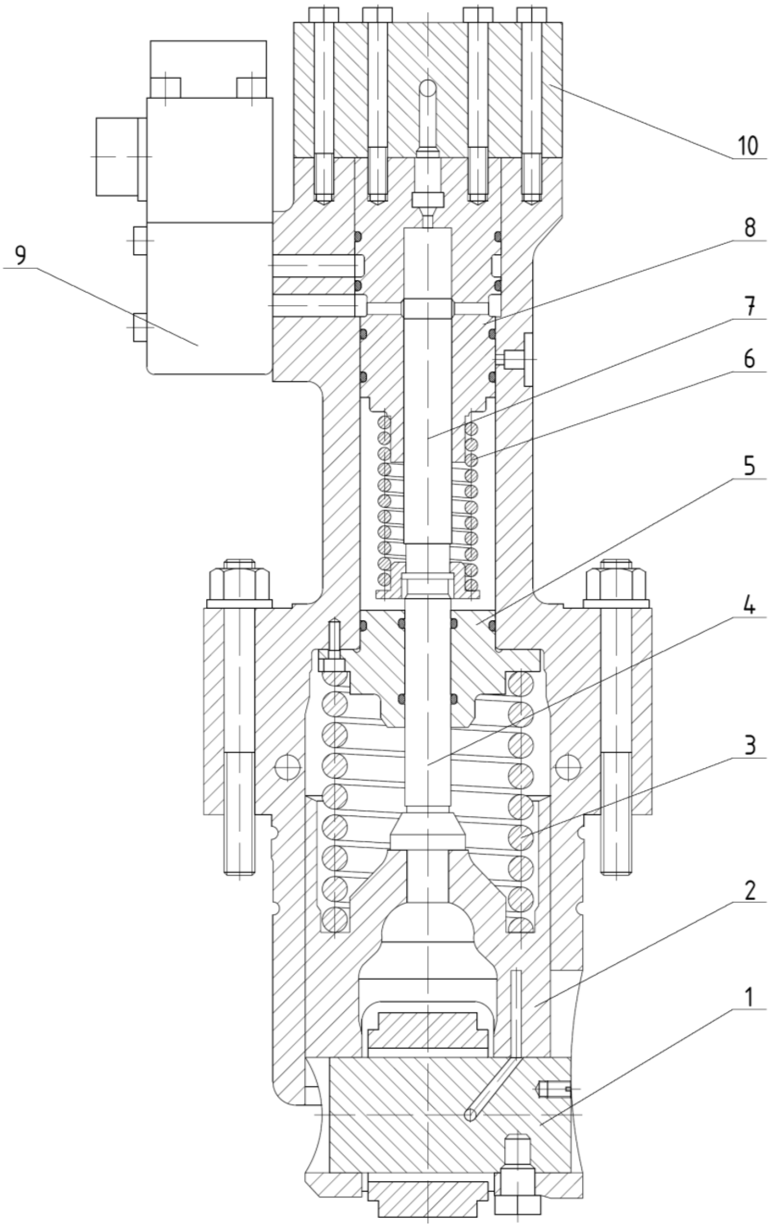


Figure 3.3 – High-pressure fuel pump:

1 – axis of the pusher roller; *2* – pusher; *3* – pusher return spring; *4* – pusher rod; *5* – distributor diaphragm with pusher rod guide; *6* – plunger return spring; *7* – plunger; *8* – plunger bushing; *9* – flow control throttle valve; *10* – pump cover

All connecting fittings and various valves are concentrated on the battery covers. This increases the strength of the housing and ensures its reliable operation at sufficiently high pressures. To maintain the specified operating pressure in the fuel supply system, a pressure regulator is installed on the battery. One regulator is installed on six-cylinder engines, and one regulator for 5…6 cylinders on engines with a large number of cylinders.

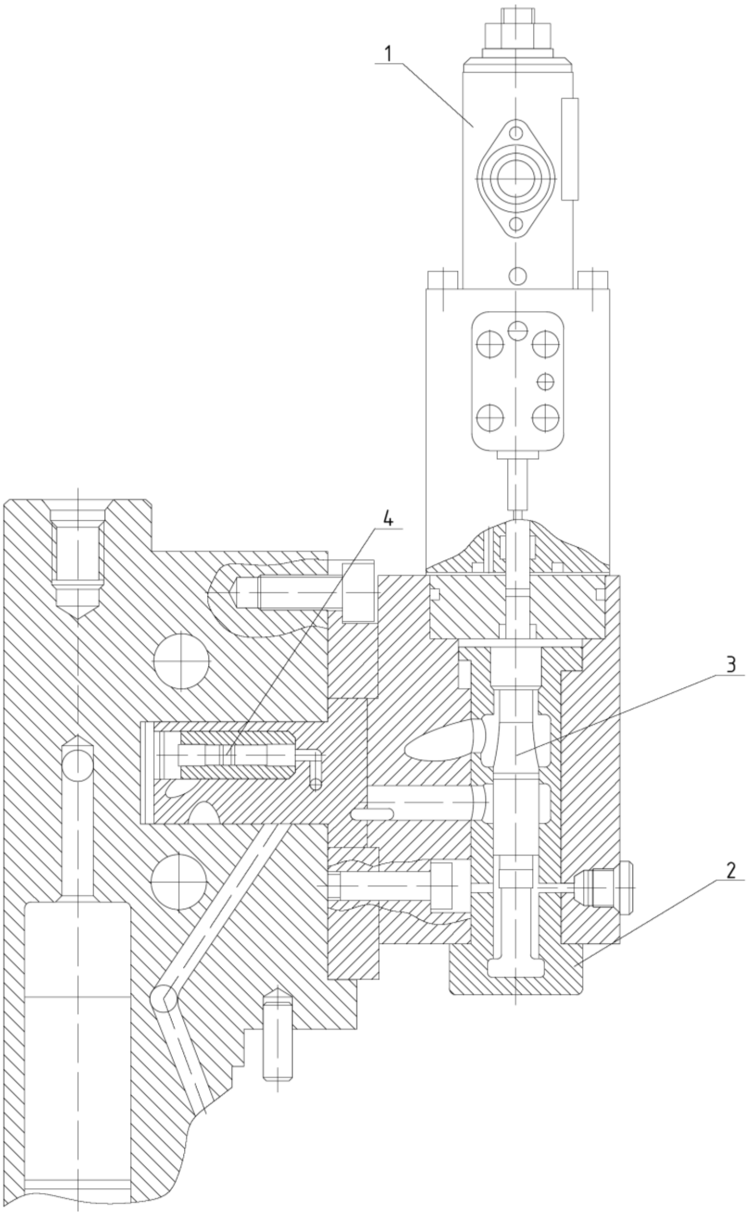


Figure 3.4 – Throttle valve for fuel supply control:

1– throttle valve actuator solenoid; *2* – throttle valve body; *3* – throttling cone of the flow control valve; *4* – safety valve

The pressure limiting regulator is a multifunctional valve system that performs the following functions:

– together with the flow control throttle valve, it maintains the set pressure in the fuel system;

– protects the battery and fuel system from excessive pressure build-up;

– ensures that the system is pumped with hot fuel when the vehicle is parked or before starting the engine;

– prevents pressure waves in the fuel system.

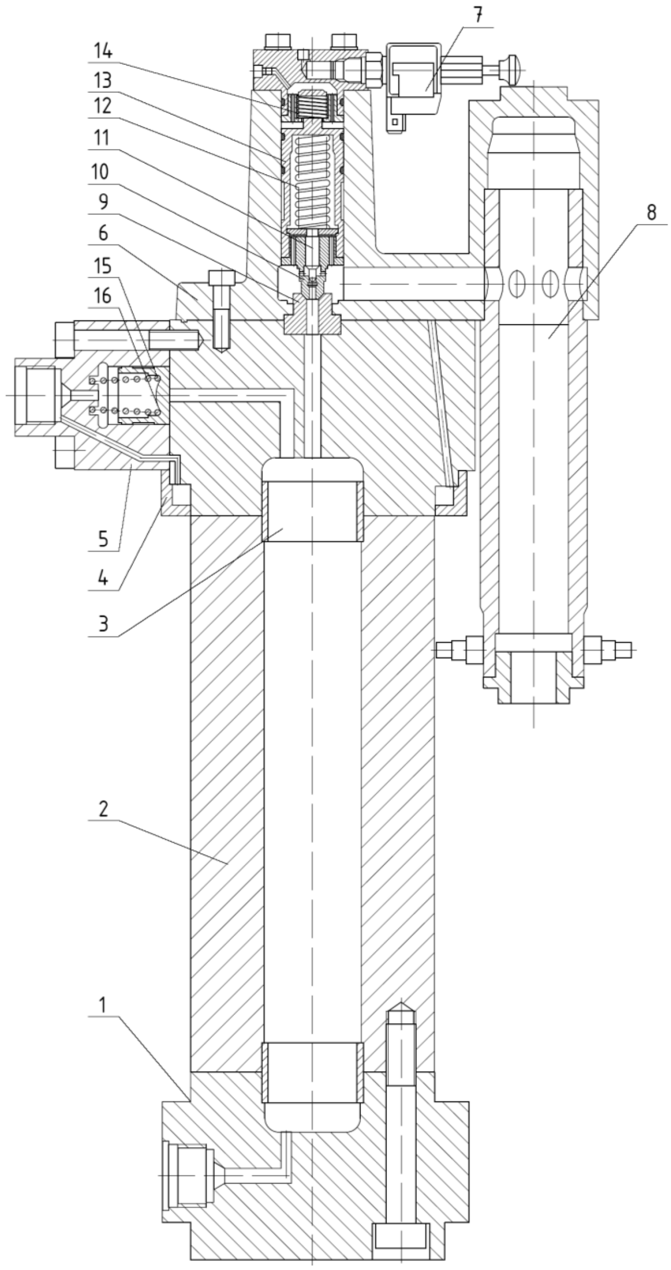


Figure 3.5 – Fuel accumulator:

1 – bottom cover of the accumulator; *2* – accumulator body; *3* – seals; *4* – top cover with a leakage channel; *5* – valve-limiter of the maximum cycle fuel supply; *6* – regulator-limiter of the pressure in the accumulator; *7* – manual regulator of the pressure in the accumulator; *8* – buffer cavity; *9* – pressure regulator valve seat; *10* – pressure regulator valve shut-off cone; *11* – maximum pressure limiting valve shut-off cone; *12* – load spring; *13* – hydraulic piston; *14* – return spring; *15* – return spring; *16* – maximum cycle flow limiting piston

The regulator is controlled by an electronic module through an electromagnetically actuated throttle valve. The working fluid for valve control is oil from the control oil system. The oil is supplied to the limit switch from the control line via a constant cross-section throttle insert and is discharged from the control cavity via an electrically controlled throttle valve. Depending on the control valve's flow path, a certain pressure is set in the cavity, which acts on the hydraulic piston. The latter, via a spring, loads the closing cone of the valve block that separates the accumulator cavity from the low-pressure cavity. The valve block consists of a pressure regulator valve and an emergency pressure relief valve integrated in its housing.

As the movement of the hydraulic piston changes the tension of the load spring, the opening pressure of the control valve changes accordingly. When the pressure exceeds the set pressure by about 15%, the valve opens and allows some fuel to flow into the buffer cavity. Since the valve's locking surfaces are subjected to significant contact loads, the valve seat is made of tungsten carbide. The same material is used to cover the valve's closing cone. The maximum pressure limiting valve is installed inside the control valve body.

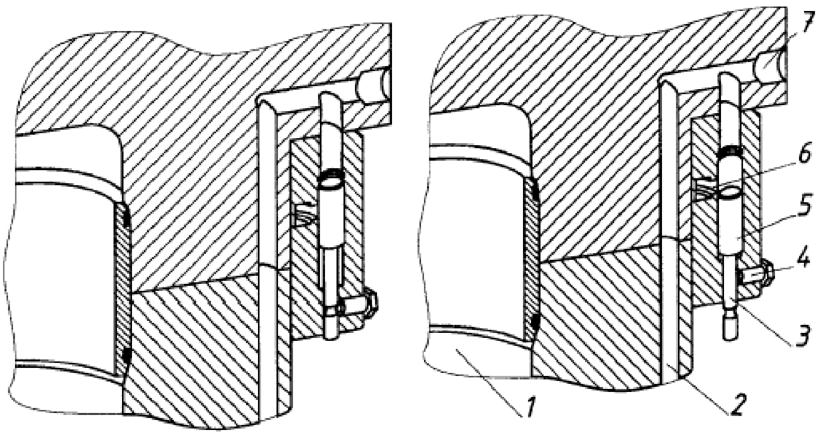
It is triggered in the event of an emergency increase in system pressure. When the system is pumped while the engine is parked or before starting, the pressure in the hydraulic cavity of the restrictor is released, the valve opens under the action of the return spring and the fuel circulates freely through the fuel system. The oil pressure from the control cavity of the restrictor can be released either from the engine control unit or manually. For this purpose, there is a knob on the throttle valve.

When pressure waves exceeding 10 MPa occur in the system, the restrictor valve opens and allows part of the fuel to flow into the buffer volume, where the wave dissipates. This ensures the stability of the fuel supply from cycle to cycle. Operating experience has shown that pressure fluctuations in the fuel system at steady-state conditions do not exceed 4 MPa, which corresponds to 2.7...4.5% of the total system pressure.

The valve-limiter of the maximum cycle flow is made in the form of a cylindrical piston placed in a cylinder. The end surfaces of the piston and cylinder form the valve's shut-off surfaces. The piston is returned to its original position by a conical spring.

There is a thin slot along the piston body that acts as a drainage channel. In the absence of injection, fuel flows through this channel from the accumulator cavity to the valve cavity. The piston moves to its original position under the action of the spring. When injection begins, the drainage channel is unable to provide the required fuel flow through the valve, resulting in a pressure difference between the cavities. This difference causes the piston to move in the cylinder. If the flow does not stop before the piston reaches the end position, for example, due to damage to the high-pressure line, the piston end will rest against the bottom of the valve body and shut off the fuel supply to the damaged line. The leak detection system consists of a network of drainage channels in the top battery cover, each with its own control valve. To improve the safety of the fuel system, all high-pressure connecting and pressure lines are double-walled.

In the event of a seal failure, leaks collect in the space between the walls and are discharged into the drainage system, which has a separate channel for each tube. All channels through the control valve system are connected to the leakage collection line, in which a control sensor is installed. if any of the seals are damaged, the pressure in the drainage channel increases, acting on the control valve piston. The valve, overcoming the force of the ball retainer, moves downwards, connecting the drainage channel with the leakage collection line. A sensor installed in the line signals a malfunction, and the output of the corresponding valve stem indicates the location of the damage (Fig. 3.6).



a *b*

Figure 3.6 – Leak detection system *a* – initial state; *b* – in the presence of leaks:
1 – battery cavity; *2* – drainage channel; *3* – valve stem; *4* – spring retainer; *5* – control piston; *6* – leakage collection channel; *7* – plug

The pumping valve is hydraulically driven by the control oil system (Fig. 3.7), serves to ensure free circulation of fuel through the system during its pumping and to maintain the set pressure in the buffer cavity of the batteries.

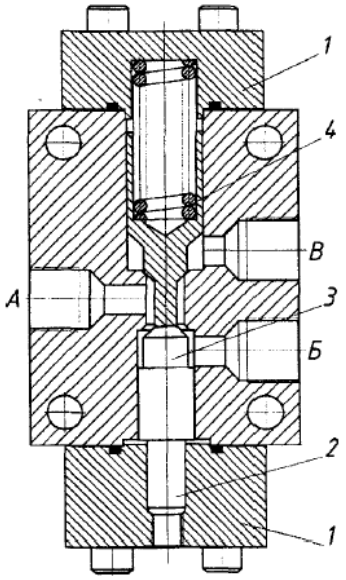


Figure 3.7 – Fuel pumping valve:

1 – cover; *2* – hydraulic drive piston; *3* – valve; *4* – return spring

The fuel injector with electromagnetic flow control is a complex electro-hydraulic device, the hydraulic diagram of which is shown in Fig. 3.8. A longitudinal section of the injector is shown in Fig. 3.9.

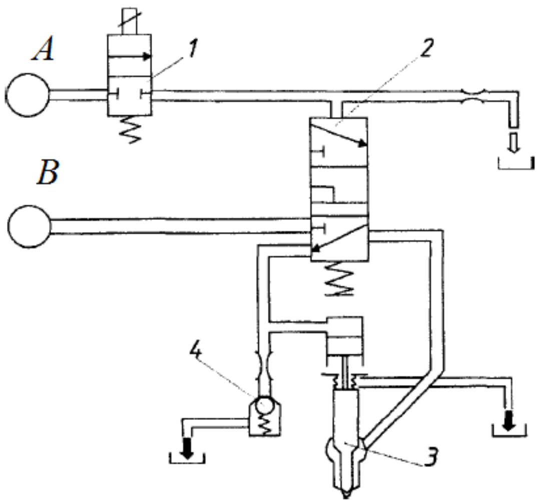


Figure 3.8 – Hydraulic diagram of the nozzle:

1 – starting valve; 2 – main valve: 3 – spray valve; 4 – residual pressure limiting valve; *A – control oil; B – fuel*

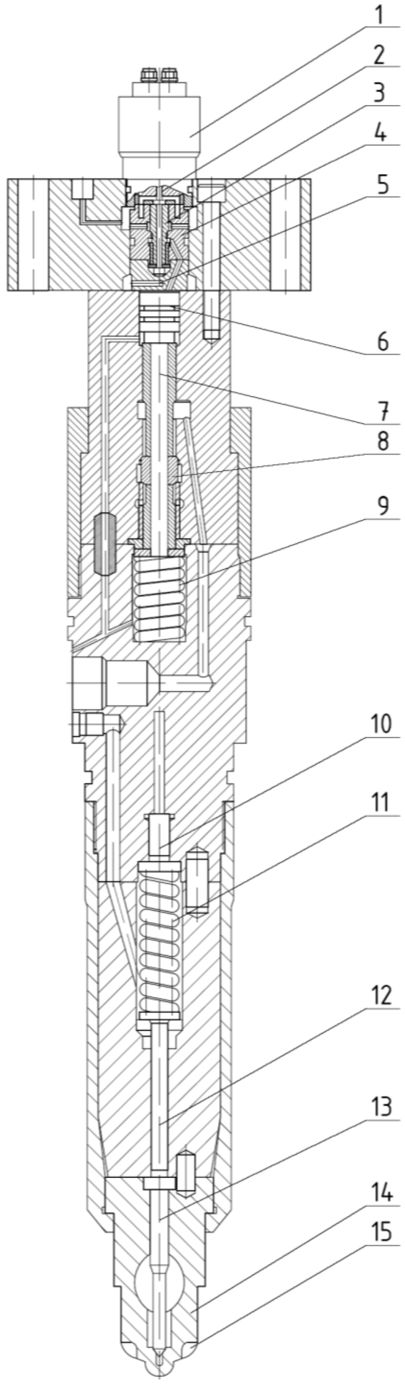


Figure 3.9 – Fuel injector with electromagnetic control of fuel supply:

1 – solenoid; *2* – solenoid armature; *3* – starting valve; *4* – starting valve body; *5* – throttling hole; *6* – main valve actuator piston; *7* – main valve guide; *8* – main valve; *9* – return spring; *10* – spray needle hydraulic locking piston; *11* – spray needle valve load spring; *12* – stem; *13* – spray needle valve; *14* – spray body; *15* – spray cooling cavity

The injector includes three main valves:

– starting valve, with electromagnetic drive;

– main valve, hydraulically driven by the starting valve;

– needle valve of the sprayer.

In addition, the nozzle design includes a valve that limits the residual fuel pressure in the working cavity.

The control system of the battery fuel system is two-level. At the first level is a starting valve controlled by an electronic engine control unit. This valve affects the flow of control oil supplied to it from the control line. On the second level is the main valve, which is hydraulically actuated by the control oil system and controlled by the starting valve.

The design of the main valve allows it to sequentially control the hydraulic closure of the spray needle and the fuel injection process into the combustion chamber. This prevents premature fuel injection until the pressure in the atomiser cavity reaches its maximum value. This solution has significantly improved the quality of fuel spraying, especially in the initial stage of the injection process.

The starting valve of the flow control has an electromagnetic drive from the electronic engine control unit. A cone-type shut-off device is used as the operating body, which is held closed by a load spring.

When the control signal is applied to the solenoid coil, its anchor overcomes the force of the load spring and opens the valve. The cavity of the control oil line is connected to the piston space of the hydraulic cylinder of the main valve actuator. The oil entering the cylinder cavity acts on the piston, moving it along with the main valve guide.

To drain the oil from the hydraulic cylinder cavity, there is a throttle hole in the bottom of the cylinder. When the valve is opened, the oil supply significantly exceeds the oil outflow through the throttle, and when it is closed, the oil gradually flows into the drain line and the piston returns to its original position.

The main valve sequentially controls the hydraulic locking of the spray needle and the fuel injection. A special insert made of tungsten carbide is used as the working element, which is fitted onto a central rod with spacer bushings. Together with the stem, the spacer bushings form the main valve guide.

Tungsten carbide is also used to coat the tapered seats of the main valve, one of which is located in the body and the other in the insert sleeve.

The need to use materials with increased hardness is caused by the fact that, to ensure the required speed, the valve has a very short stroke, and at the moment of its opening and closing, the gap between the seat and the valve is comparable to the size of microparticles that are somehow present in the fuel. At this point, the particles carried away by the fuel flow interact with the valve's working surfaces, causing increased wear.

The operation of the main valve is shown schematically in Fig. 3.10. In the absence of a control signal, there is no pressure in the piston cavity of the main valve actuator and, under the action of the spring, the valve moves to the uppermost position. With its upper working edge, the valve blocks the access of fuel to the spray nozzle and the nozzle needle hydraulic locking piston (Fig. 3.10, *a*). When the starting valve is opened, the control oil enters the cavity of the main valve hydraulic actuator piston. When the valve opens, the fuel flows into the intermediate cavity, from where it enters the spray cavity and the hydraulic cylinder, the piston of which is connected to the intermediate stop of the nozzle needle (Fig. 3.10, *b*).

The piston, overcoming the force of the return spring, descends and blocks the opening of the needle valve via a stopper. This ensures that the injection is delayed as long as necessary to achieve the maximum pressure in the spray chamber.

Further movement of the main valve causes its trailing edge to block the fuel from entering the needle valve's hydraulic locking cylinder. The fuel in this cylinder flows through the throttle opening and the residual pressure limiting valve to the drain, releasing the nozzle needle stop. Under the influence of pressure in the spray cavity, the needle rises, and fuel injection begins (Fig. 3.10, *c*).

After the starting valve closes, the control oil is drained from the hydraulic cylinder of the main valve actuator. This causes the main valve to open and reconnect the intermediate cavity with the hydraulic shut-off cylinder of the needle valve sprayer. Under the influence of fuel pressure, the piston moves and closes the needle valve through the intermediate stop, thus ensuring a sharp cut-off of the flow (Fig. 3.10, *c*). Then the main valve closes, separating the nozzle cavity from the accumulator cavity, and the remaining fuel is drained from the nozzle cavity through the residual pressure relief valve.

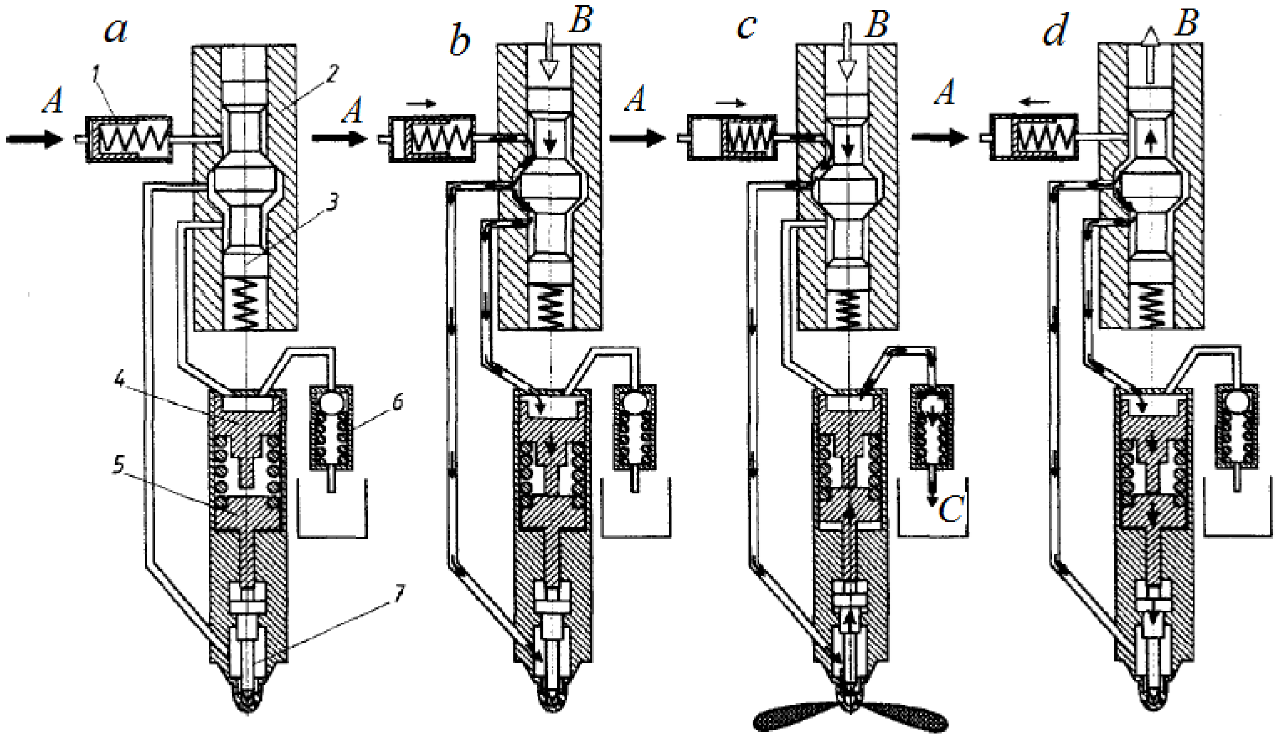


Figure 3.10 – Schematic of the main valve operation:

a – no injection; *b* – preparation for injection; *c* – injection; *d* – flow cut-off; *1* – maximum cycle flow limiter; *2* – main valve body; *3* – main valve; *4* – hydraulic piston with intermediate nozzle needle stop; *5* – hydraulic piston with main nozzle needle stop; *6* – residual pressure limiting valve; *7* – needle spray valve; *A* – fuel supply; *B* – control oil supply; *C* – fuel to drain

The engine electronic control system is a complex that includes several microprocessor modules designed to control the engine and monitor its operating parameters. The system also includes various sensors that collect the information necessary for decision-making and actuators that control the engine.

All system elements have duplication and redundancy, so the failure of one module does not lead to an engine shutdown, and the functions of the damaged module are distributed among the remaining system elements. The same applies to data collection sensors, some of which have multiple redundancies. First of all, this applies to the crankshaft position sensors, without which engine operation is basically impossible. One engine is equipped with two sensors for the current crankshaft position, the main and backup sensors, and two sensors that detect the Crankshaft Operating Point and the NBP. The wiring diagram of the sensors to the engine management and control system is shown in Fig. 3.11.

The first two sensors receive a signal from a special disc mounted on the flywheel, and the second sensors receive a signal from a disc mounted on the end of the camshaft. To generate the signal, 119 indentations are made on the flywheel disc every 3° and one indentation is omitted to form a reference point that corresponds to the CCT. The second disc has a stepped protrusion, the edges of which correspond to the engine shaft positions at the CMT and NMT. Since the camshaft rotates twice as slowly as the crankshaft, the signals from this sensor coincide with the start of each new stroke.

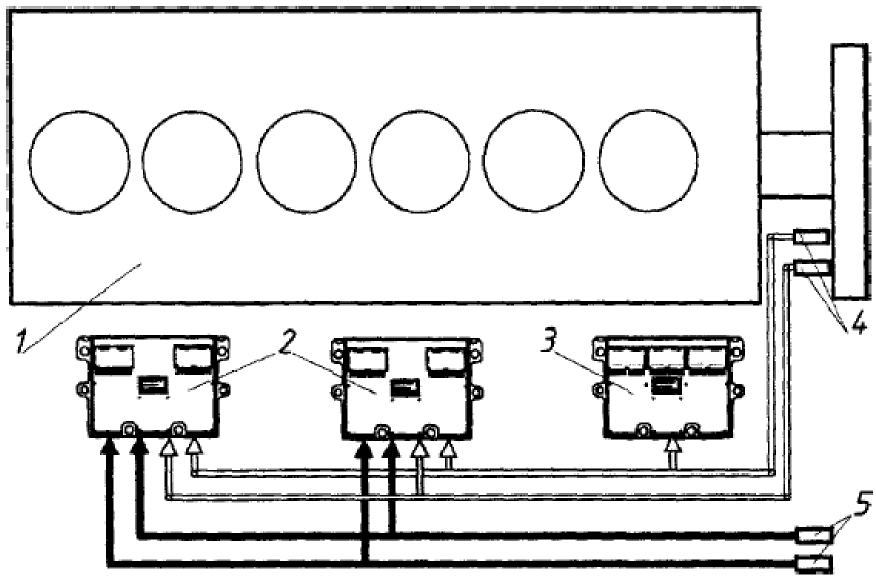


Figure 3.11 – Wiring diagram of sensors to the engine management and control system:

1 – engine; *2* – controllers for controlling the working cylinders; *3* – controller for controlling the fuel system; *4* – crankshaft position sensors; *5* – crankshaft rotation phase sensors

3.3. Calculation of fuel system elements of the 6CHN 46/58 marine engine

According to the second chapter of the master's thesis, the initial data for further calculations of the geometric dimensions and performance parameters of the elements of the new fuel system were obtained from the calculation of the operating cycle of the 6CHN 46/58 marine diesel engine. The results of the calculations are presented in Table 3.1.

Table 3.1 – Calculation of fuel system elements of marine diesel engine 6CHN 46/58

№ n.o.	Parameter	Dimension	Determination method	Value
1	Power of the engine, N_e	kW	–	5543
2	Specific effective fuel consumption of the engine, g_e	g/(kW·h)	–	183
3	Part of the main cycle supply, $V_{c.main}$		–	0.8
4	Rotation frequency, n	rpm	–	500
5	Number of cylinders of the engine i	–	–	6
6	Stroke ratio z	–	–	0,5
7	Maximum combustion pressure, P_z	MPa	–	17.26
8	Fuel supply duration, $\varphi_{s.d}$	deg. of c. r.	–	36
9	Cylinder pressure at point c, P_c	MPa	–	14.3
10	Density of fuel, $\gamma_т$	kg/m^3	–	875
11	Reserve factor, K	–	–	1,05
12	Cyclic fuel supply	cm^3/cycle	$V\text{ц} = \dfrac{Neц \cdot g_e \cdot K \cdot V_{ц.осн}}{60 \cdot n \cdot z \cdot \gamma_m}$	13,496
13	Coefficient of volume reserve, K_2	–	4...8	4
14	Volume describing the plunger of a high-pressure fuel pump during the stroke from VDC to LDC	cm^3	$V_П = K_2 \cdot V_ц = \dfrac{\pi \cdot d_П^2}{4} \cdot S_П$	14.846
15	According to the standards	–	1.0…1.5	1.5
16	Diameter of the plunger of the high-pressure fuel pump	mm	$d_П = \sqrt[3]{\dfrac{4 \cdot V_П}{1,3 \cdot \pi}}$	24
17	Stroke of the plunger	mm	$S_п = 1,5 \cdot d_П$	37
18	Area of the plunger	mm ^{2}	$F_{пл} = \dfrac{\pi \cdot d^2}{4}$	467.8
19	Feed rate of the plunger, η	–	–	0.7
20	Active geometric plunger stroke	mm	$S_{АГ} = \dfrac{Q_T}{\eta \cdot F_П}$	41.2
21	Duration of the active geometric stroke of the plunger	degree	$\beta_{АГ} = (0,6 \div 0,8) \cdot \beta_{ВП}$	13.5

22	Maximum possible plunger speed	m/s	$C_m = 1{,}3\dfrac{S_{AГ}\cdot 6\cdot n_K}{\beta_{AГ}}$	5.95
23	Kinematic coefficient at the starting point of the profile	mm	$X_н = R_0 + c$	415
24	Radius of the pusher roller	s^{-1}	$\omega_K = \dfrac{\pi \cdot n_K}{30}$	26.16
25	Determination of ram acceleration at the first section of the profile	m/s^2	$W_1 = X_H \cdot \omega_K^2 - \dfrac{X_H^2 \cdot \omega_K^2}{R_H + \rho}$	602.855
26	Plunger stroke on the first section of the profile	m	$S_1 = \dfrac{C_{MAX}^2}{2\cdot W_1}$	0.0294
27	Plunger travel on the second section of the profile	m	$S_2 = S_п – S_1$	0.007224
28	Determination of ram acceleration in the second section of the profile	m/s^2	$W_2 = -\dfrac{C_{MAX}^2}{2\cdot S_2}$	-2453.11
29	Kinematic coefficient at the end of the first section of the profile	m	$X_c = R_0 + c + S_1$	0.444
30	Maximum value of the pressure angle	degree	$\delta_{MAX} = arctg\left(\dfrac{C_{MAX}}{X_C \cdot \omega_K}\right)$	27.11
31	Coefficient of excess of the force of the plunger spring	–	$K = \dfrac{(f_0 + S_1)\cdot K_Ж}{m\cdot \lvert W_2\rvert}$	0.041
32	Kinematic coefficient at the end of the profile	m	$X_K = R_0 + c + S_п$	0.4516
33	Radius of curvature at the end point of the profile	m	$R_K = \dfrac{X_K^3 \cdot \omega_K^2}{X_K^2 \cdot \omega_K^2 - W_2 \cdot X_K} - \rho$	-0.099
34	Force from fuel pressure at the plunger position at the VMT	MH	$P_T = P_{ЛО} \cdot F_П$	9.4E-05
35	Spring force at the plunger position at the timing mark	MH	$P_п = (f_0 + S_П)\cdot K_Ж$	0.0029
36	Force transmitted by the roller to the cam	MH	$N = P_T + P_П$	0.002998
37	Maximum permissible radius of curvature at the end point of the profile	m	$R_{KД} = \dfrac{\dfrac{N\cdot E}{b}}{\left(\dfrac{\sigma_Д}{0{,}418}\right)^2 - \dfrac{N\cdot E}{b\cdot \rho}}$	0.00019
38	Angles of the first sections of the forward stroke profile	degree	$\beta_1 = \dfrac{12\cdot n_K \cdot S_1}{C_{MAX}}$	14.81
39	Angles of the second sections of the forward stroke profile	degree	$\beta_2 = \dfrac{12\cdot n_K \cdot S_2}{C_{MAX}}$	3.64
40	Straight stroke profile angle	degree	$\beta_П = \beta_1 + \beta_2$	18.45
41	Cam projection angle	degree	$\beta_B = 2\cdot \beta_П$	36.91

42	Radius of curvature	mm	$R_{2H}=\dfrac{\left(C_{MAX}^2+X_C^2\cdot\omega_K^2\right)^{1.5}}{\left(X_C^2\cdot\omega_K^2+2\cdot C_{MAX}^2-W_2\cdot X_C\right)\cdot\omega_K}-$ $-\rho$	0.0357
43	Coefficient	–	$K_3=\dfrac{W_1\cdot10^3}{72\cdot n_K^2}$	0.0001 3
44	Coefficient	–	$K_4=\dfrac{C_{MAX}}{\beta_1}$	0.4019
45	Gas pressure in the diesel cylinder at the moment of injection	MPa	$P_{Ц}=\dfrac{P_{СЖ}+P_Z}{2}$	15.8
46	Fuel compressibility factor, α_c	MPa^{-1}	–	8E-04
47	Flow coefficient of the filling holes of the plunger pair, μ_f	–	–	0.86
48	Fuel pressure in the low pressure cavity of the fuel pump, P_0	MPa	–	0.4
49	Diameter of the filling holes of the plunger pair, d_f	mm	–	10
50	Cross-section of the filling holes of round shape	m^2	$f_H=i_H\cdot\dfrac{\pi\cdot d_H{}^2}{4}$	0.0003 1
51	Spray needle diameter, d_n	mm	–	9.5
52	Diameter of the base of the closing cone of the spray needle, d_c	mm	–	6.4
53	Relative value of the differential plane of the spray needle, δ	–	–	0.63
54	Angle of the closing cone of the spray needle, δ_n	degree	–	60
55	Diameter of the spray well, d_w	mm	–	12
56	Diameter of spray holes, d_s	mm	–	0.786
57	Number of spray holes, i_s	–	–	8
58	Spray needle lift, h_n	mm		0.275
59	Residual pressure in the high-pressure system, $P_{r.p}$	MPa		5
60	Injection start pressure, P_{st}	MPa		37.5
61	Flow coefficient of the minimum flow section in the closing cone of the spray nozzle, $\mu_{c.c}$	–	–	0.8
62	Flow coefficient of the spray nozzle openings, μ_s	–	–	0.8
63	Total flow cross-section of the nozzle openings, f_P	mm^2	–	3.88

64	Coefficients	–	$K_1^1 = \dfrac{F_П}{\alpha_{СЖ} \cdot 6 \cdot n_К \cdot V_Н}$	11.476
65	Coefficients	–	$K_3^1 = \dfrac{\mu_Н}{\alpha_{СЖ} \cdot 6 \cdot n_К \cdot V_Н} \cdot \sqrt{\dfrac{2}{\rho_Т}}$	938.248
66	First period pressure	MPa	$P_Н = \left(\dfrac{K_1^1 \cdot C_П}{K_3^1 \cdot f_Н}\right)^2 + P_0$	0.4745
67	First fuel supply period	MPa	$\Delta P_1 = \Sigma\left(K_1^1 \cdot C_П - K_3^1 \cdot f_Н \cdot \sqrt{(P_Н - P_0) \cdot 10^6}\right) \cdot \Delta\beta + P_н$	17.09
68	Cross-section of the base of the closing cone of the spray needle	mm^2	$F_К = \dfrac{\pi \cdot d_К^2}{4}$	32.154
69	Differential area of the spray needle	mm^2	$F_\delta = \dfrac{\pi \cdot \left(d_И^2 - d_К^2\right)}{4}$	38.693
70	Coefficients	–	$K_1^2 = \dfrac{F_П}{\alpha \cdot 6 \cdot n_К \cdot (V_Н + V_Л)}$	11.4757
71	Coefficients	–	$K_3^2 = \dfrac{\mu_Н}{\alpha_{СЖ} \cdot 6 \cdot n_К \cdot (V_Н + V_Л)} \cdot \sqrt{\dfrac{2}{\rho_Т}}$	469.12
72	Before closing the filling holes	MPa	$\Delta P = \left(K_1^2 \cdot C_П - K_3^2 \cdot f_Н \cdot \sqrt{(P_Н - P_0) \cdot 10^6}\right) \cdot \Delta\beta$	147.84
73	Spray needle lift pressure	MPa	$P_Н = P_{ВП} - P_С \cdot \dfrac{F_К}{F_\delta}$	25.62
74	Pressure in the high pressure line	MPa	$P_л = P_н - P_{зт}$	142.84
75	Coefficients	–	$K_1^3 = \dfrac{F_П}{\alpha_{СЖ} \cdot 6 \cdot n_К \cdot (V_Н + V_Л + V_Р)}$	11.148
76	Coefficients	–	$K_2^3 = \dfrac{\mu_Р \cdot f_Р}{\alpha_{СЖ} \cdot 6 \cdot n_К \cdot (V_Н + V_Л + V_Р)} \cdot \sqrt{\dfrac{2}{\rho_Т}}$	0.004
77	Coefficients	–	$K_С = \dfrac{(\mu_{ЗК} \cdot F_{ЗК})^2}{(\mu_{ЗК} \cdot F_{ЗК})^2 + (\mu_Р \cdot f_Р)^2} =$	0.6359
78	Cross section	m^2	$F_{ЗК} = \pi \cdot \left(d_{КО} - h_Н \cdot \sin\dfrac{\delta_Н}{2} \cdot \cos\dfrac{\delta_Н}{2}\right) \cdot h_Н \cdot \sin\dfrac{\delta_Н}{2}$	5.13E-06
79	Pressure in the high pressure line	MPa	$P_{л1} + 1 = P_{л1} + \Delta P$	192.1
80	Fuel pressure in front of the spray holes	MPa	$P_С = \dfrac{(\mu_{ЗК} \cdot F_{ЗК})^2 \cdot P_Л + (\mu_Р \cdot f_Р)^2 \cdot P_Ц}{(\mu_{ЗК} \cdot F_{ЗК})^2 + (\mu_Р \cdot f_Р)^2}$	127.91
81	Injector needle seating pressure	MPa	$P_{кв} = P_{нв} \cdot \delta$	24.001

The calculation of parts for a high-pressure fuel pump for a marine diesel engine 6CHN 46/58 is presented in tabular form (Table 3.2).

Table 3.2 – Calculation of parts for a high-pressure fuel pump for a marine diesel engine 6CHN 46/58

№ n.o.	Parameter	Dimension	Determination method	Value
Calculating the plunger spring				
1	Wire diameter, d_w	mm	–	13
2	Diameter of the plunger spring	mm	–	95
3	Pre-tightening of the plunger spring is f_0	mm	–	6
4	Gap between the spring coils in the compressed state, Δn	mm	–	6
5	Permissible torsional stress, τ_0	MPa	–	450
6	Coefficient, K	–	–	1.2
7	Modulus of shear elasticity, G	MPa	–	82500
8	Maximum permissible spring load	N	$P_{\text{пmax}} = \frac{\pi \cdot d_{\text{пр}}^3 \cdot \tau_0 \cdot 10^6}{8 \cdot D_0}$	4084.69
9	Stiffness of one spring coil	N/m	$K_{\text{ж}1} = \frac{G \cdot d_{\text{пр}}^4}{8 \cdot D_0^3}$	343531
10	Full spring deformation	mm	$f_{max} = f_0 + S_{\text{п}}$	43
11	Number of working turns of the spring	–	$i_{\text{рп}} = \frac{f_{max} \cdot K_{\text{ж}1}}{P_{\text{пмax}}}$	7
12	Stiffness of the spring	N/m	$K_{\text{ж}} = \frac{K_{\text{ж}1}}{i_{\text{рп}}}$	42941.4
13	Spring deformation at the moment when acceleration	m	$f_w = f_0 + S_w$	0.0132
14	Spring force	N	$P_{\text{п}w} = f_w \cdot K_{\text{ж}}$	567.839
15	Force of inertia of the plunger actuator parts	N	$P = m \cdot / W_0$	61327.8
16	Coefficient of excess of spring force	–	$K = \frac{P_{\text{п}w}}{Pu}$	0.009
17	Spring length in the free state	mm	$L_{\text{пр}} = (d_{\text{пр}} + \Delta n) \cdot i_{\text{р}} + f_{max} + i \cdot d_{\text{пр}}$	140.717
18	Spring coil pitch	mm	$t = d_{\text{пр}} + \frac{f_{max}}{i_p} + \Delta n$	29.654
19	Spring index	–	$C' = \frac{D_0}{d_{\text{пр}}}$	7.3077
20	Correction coefficient	–	$K_n = \frac{4 \cdot C' - 1}{4 \cdot C' - 4} + \frac{0,615}{C'}$	1.203
21	Maximum spring force	N	$P_{\text{п}\,max} = f_{max} \cdot K_{\text{ж}}$	1830.05
22	Maximum bending and torsional stress in the spring	MPa	$\tau_{max} = \frac{8 \cdot K_n \cdot P_{\text{пmax}} \cdot D_0}{\pi \cdot d_{\text{пр}}^3}$	242.552
23	Minimum spring force	MPa	$P_{\text{по}} = f_0 \cdot K_{\text{ж}}$	257.649

24	Minimum bending and torsional stress in the spring	MPa	$\tau_{\min} = \frac{8 \cdot K_n \cdot P_{no} \cdot D_0}{\pi \cdot d_{np}^3}$	34.1483
25	Average cycle voltage	MPa	$\tau_m = \frac{\tau_{max} + \tau_{\min}}{2}$	138.35
26	Amplitude of the cycle voltage	MPa	$\tau_a = \frac{\tau_{max} - \tau_{\min}}{2}$	104.202
27	Safety margin	–	$n = \frac{1}{\frac{\tau_m}{\tau_s} + \gamma_\tau \cdot \frac{\tau_a}{\tau_{-1}}}$	2.825
28	Check spring for resonance	$\min^{-1}$ /–	$n_c = 2{,}17 \cdot 10^7 \frac{d_{np}}{i_{p.} \cdot D_0^2}$ $\frac{n_c}{n_k}$	11002.6 29.3>10

3.4 Conclusions to the part

This chapter of the master's thesis analyses the promising directions of development of fuel systems for modern medium-speed marine diesel engines. The fuel system and its main elements of a medium-speed marine diesel engine of the 6CHN 46/58 type were developed and calculated. The main geometrical and operational parameters are obtained. The system elements provide all the necessary parameters of fuel injection for the 6CHN 46/58 engine, which are included in the calculation of the operating cycle (**Part 2**).

PART 4
ECONOMIC PART

4.1 General provisions

A correct assessment of the economic efficiency of new or upgraded equipment serves as a basis for making decisions on its manufacture and helps to determine prices for new or upgraded products. In a broader sense, the assessment of economic efficiency makes it possible to draw a conclusion about the technical and economic level of production and to form an objective view of the trends in its technical development. When performing technical and economic calculations, new equipment is understood to mean both completely new (original) models of technical systems or equipment that differ significantly from existing ones and improved ones (i.e. those that were produced earlier but have been modernised). The main criterion for the economic evaluation of new equipment is the annual economic effect. It is also important to distinguish between actual and expected economic effects. The expected economic effect shows the actual economic benefit that will be obtained after the introduction of a new technical system. The total economic effect is attributed to all the organisations involved. Each of the co-implementers is entitled to a part of the total economic effect, or partial effect. If the economic effect of the implementation is calculated taking into account the interests of the country's national economy, it is called the national economic effect, and if it is determined taking into account the work of an individual enterprise, it is called the self-supporting effect. In the process of designing or modernising, several variants of technical solutions for the same technical system are considered. The requirement of rationality and expediency is ensured by a comprehensive comparative analysis of the various options under consideration by groups of certain factors:

– production (logistical and organisational)

– operational

– social

– environmental;

– economic.

The production factor assesses the material, technical and organisational feasibility of a new technical system. Thus, the material and technical feasibility of a new technical system involves comparing certain design properties, the degree of its unification and normalisation, the appropriate structure of the size range of means, the degree of complexity of the kinematic scheme, overall dimensions and weight of the structure, structural material consumption, etc. The organisational feasibility of a new technical system lies in determining the possibility of its manufacture.

Operational feasibility factors are directly related to the purpose of the technical system and are characterised by:

– compliance with the current level of productivity and operating speeds;

– ease of maintenance;

– appearance and ergonomics;

– the ability to adapt to different working conditions;

– the possibility of further modernisation.

Social and environmental factors are reduced to a comparison:

– professional and qualification composition of employees;

– appropriate working conditions.

Thus, organisational, technical, operational and social indicators are the initial information for further economic calculations. For a generalised assessment of the various options that are compared with each other, their advantages and respective disadvantages are reduced to a single conditional indicator – the cost of public labour.

4.2. Determination of the economic effect of upgrading the 6CHN 46/58 engine

The objects of technology to be compared must be identical and comparable to each other. In particular, the requirement of identity implies that the basic and new versions of the technical system perform the same production function. The requirement of comparability between objects is achieved by ensuring that the new and baseline variants are designed for the same amount of work performed. Table 4.1 shows the basic technical and economic indicators of the 6CHN 46/58 serial diesel engine and its upgraded version. The economic effect is determined by comparing the basic 6CHN 46/58 marine engine and the upgraded version (Table 4.2).

Table 4.1 – Technical and economic parameters of the basic and modernised ICE 6CHN 46/58

№ n.o.	Name of indicators	Symbols	Basic ICE	Modernised ICE
1	Power, kW	N_e	5494,8	5543,3
2	Rotation speed, min^{-1}	n	500	500
3	Diameter of the cylinder, m	D	0,46	0,46
4	Stroke of the piston, m	S	0,58	0,58
5	Specific fuel consumption, kg/kW·h	g_e	0,187	0,182
6	Mechanical efficiency	η_m	0,92	0,92
7	Specific oil consumption, kg/kW·h	g_o	0,0006	0,0006
8	Weight, kg	G	95000	95000
9	Dimensions, mm	L	7580	7580
		B	2880	2880
		H	4807	4807
10	Price of the engine, UAH	P	920000	1058000
11	Price of fuel, UAH/(kW·h)	P_f	2,85	2,85
12	Price of oil, UAH/kg	P_o	9,5	9,5
13	Work hours for the year, h	t	6380	6380
14	Number of control repairs	η	1	1
15	Total capital repairs coefficient	$\varepsilon_{\min}$	1,1	1,1
16	Partial labour cost coefficient	b	0,05	0,05
17	Power utilisation coefficient	K_p	0,95	0.95
18	Coefficient of part of the cost of capital repairs	$K_{c.r}$	0,3	0,3
19	Time of continuous operation, h	$t_{c.o}$	300	300
20	Engine life before the first bulkhead, h	t_{bul}	22000	22000
21	Engine life before overhaul, h	$t_{o.h}$	72000	72000
22	Normative cost-effectiveness comparison coefficient	E	0,15	0,15

Table 4.2 – Calculations of the economic feasibility of diesel modernisation

Name	Symbols	Method of determination	Basic ICE	Modernised ICE
Annual output	N, kW·h per year	$N = K_П \cdot N_e \cdot t$	33303982,8	33597941,3
The service period	T, year	$T = \dfrac{t_{кр}}{t}$	3,45	3,45
Annual current fuel costs	C_f, UAH	$З_п = N \cdot Ц_п \cdot g_e$	17749357,63	17427252,15

Annual operating costs for oil	C_o, UAH	$З_м = g_м \cdot N \cdot Ц_м$	189832,702	191508,2654
Annual costs for routine repairs	$C_{r.r}$, UAH	$З_{п.р} = \dfrac{b \cdot t \cdot Ц}{t_{пер}}$	978266,6667	1125006,667
Annual costs of capital repairs	$C_{c.r}$, UAH	$З_{к.р} = \dfrac{K_{к.р} \cdot t \cdot Ц}{t_{к.р} \cdot \varepsilon_{\min}}$	72763,64	83678,18
Annual economic effect for the consumer	U_1, UAH	$U_1 = З_п + З_м + З_{к.р} + З_{п.р}$	18990220,64	18827445,27
	$U_1^{1б}$, UAH	$U_1^{1б} = \dfrac{N_с}{N_б} \cdot U_1^б$	19157838,33	–
Annual economic effect on the consumer from the use of the designed engine	$E_к$, UAH	$E_к = \left[Ц_б \cdot \dfrac{N_с}{N_б} \cdot \left(\dfrac{\frac{1}{T_б} + E_б}{\frac{1}{T_с} + E_с} \right) + \dfrac{U_1^{1б} - U_1^с}{\frac{1}{T_с} + E_с} \right] - Ц_с$		621013,735
Additional one-off costs for engine upgrades	ΔK, UAH	–	138000	
Annual savings when operating the upgraded engine	ΔS, UAH	$\Delta S = U_1^{1б} - U_1^с$	330393,07	
Payback period of the upgraded engine	T_{pay}, year/month	$T_{окуп} = \dfrac{\Delta K}{\Delta S}$	0,42/ 5	

4.3. Conclusions to the part

The above calculation result confirmed the economic feasibility of using the 6CHN 46/58 engine with an upgraded fuel system (use of fuel accumulators and electrically controlled injectors).

The annual economic effect for the consumer from the use of the upgraded engine, compared to the base engine, is UAH 330393.07. Modernisation of the basic 6CHN 46/58 engine will pay off in 5 months.

PART 5
ANALYSING AND DEVELOPING MEASURES TO ENSURE LABOUR AND ENVIRONMENTAL PROTECTION REQUIREMENTS FOR SHIP ENGINE 6CHN 46/58 OPERATION

5.1. Analysis of hazards and harmful factors in the ship's engine room

When operating the ship's power plant (SPP) and various auxiliary equipment and machinery in the ship's engine room (ER), the ship's personnel are exposed to various hazards and harmful factors (factors), which sometimes have a very negative impact on health and well-being, as well as impede safe and productive working conditions for ship's mechanics and other personnel.

Lubricating oil and fuel, and sometimes other gaseous or liquid flammable substances, are always present in significant quantities on any ship. Accordingly, this creates conditions of increased fire and explosion hazard in the ER and on the ship as a whole.

Accordingly, all ships in general and their ERs in particular have a number of relevant mandatory measures aimed at improving safety (preventive measures) and extinguishing fire fires. To ensure effective fire extinguishing of the engine room premises, the vessel is equipped with water and volumetric chemical fire extinguishing systems.

A fire alarm system is provided in the containership's ER to preventively detect possible fires or explosions. In addition, the ER of any vessel must have various primary fire extinguishing elements, such as sand and fire extinguishers, in sufficient quantities.

Rotating or reciprocating elements and parts of the main and auxiliary diesel engines, as well as various equipment and machinery of a container ship, are dangerous and can cause injury to mechanics and other ship personnel. To prevent this, a number of measures are taken on the container ship:

– installing various stops near moving parts;

– equipping mechanisms with fuses that serve to instantly disconnect devices and units in automatic mode;

– installing various blocking structural elements near moving parts to prevent mechanics and other ship personnel from entering the hazardous area;

– installation of warning alarms to inform mechanics and other ship's personnel about the operation of certain equipment, as well as about the occurrence of abnormal and dangerous failures in its operation;

– installation of remote control and automation systems for the container ship's SPP, which enable the ship's personnel to perform control and regulation from locations significantly removed from potentially hazardous areas.

Container ship's ER space, as well as other types of vessels, is characterised by the presence of harmful and hazardous gases: nitrogen oxides, carbon monoxides, water, fuel and oil vapour, etc. To ensure safe maintenance of container ship's electrical equipment in the ER premises, the standards set the concentration limits for the presence of harmful components: carbon monoxide – up to 20 mg/m^3; hydrogen sulphide – up to 10 mg/m^3; nitrogen dioxide – up to 20 mg/m^3; sulphur dioxide – up to 10 mg/m^3; nitrogen oxide – up to 5 mg/m^3; fuel vapours – up to 30 mg/m^3.

Various types of radiation (thermal and electromagnetic) can also be considered harmful factors affecting ship's mechanics and other personnel.

The source of thermal radiation on a container ship is the ship's propulsion system (main and auxiliary engines). The effect of thermal radiation on a container ship can be reduced by insulating heated parts.

The source of electromagnetic radiation on a container ship is radio and radar stations. The effect of electromagnetic radiation on a container ship can be reduced by locating generation sources in places significantly removed from the ship's personnel, as well as by installing special shielding barriers on these sources.

Noise and vibration are also negative factors affecting ship's mechanics and other personnel. The sources of these factors are engines (main and auxiliary), propeller shafting and various ship auxiliary equipment and mechanisms. It is also worth noting that sea waves can also cause some additional vibration on the ship.

The negative impact of vibration and noise on the body of ship's mechanics and other personnel is difficult to overestimate. The level of negative impact directly depends on the duration and frequency spectrum of sound and vibration waves. The presence of these factors in the performance of seafarers' labour activities creates rather negative working conditions.

For example, depending on the intensity and duration of wave oscillations, ship's mechanics and other personnel may experience some changes in the cardiovascular and central nervous systems. The negative impact also extends to the vestibular system.

At the initial stages of a ship's construction, designers set certain noise and vibration limits for the equipment used on board, thereby ensuring that these negative factors are generally reduced.

Noise from engines is mainly caused by the intake of fresh air into the cylinder and the exhaust of exhaust gases – the so-called aerodynamic noise. This noise can be effectively reduced by installing silencers and waste heat recovery boilers in the air supply and exhaust systems. A comprehensive method is to place noise sources in separate sections or compartments. The use of insulating covers and sound-absorbing materials of various compositions is also quite effective.

During the operation of the GD, auxiliary waste heat boilers (WHB), auxiliary diesel generators (DG), there may be leaks of exhaust gases. In addition to safe components of complete combustion, the exhaust gasesof engines includes a number of components of incomplete combustion, as well as a number of other compounds, such as nitrogen oxides, ash particles, benzoids, etc. The concentration of these harmful gases and various compounds in the air of the container ship's interior must be standardised (Table 5.1).

Table 5.1 – Standard concentration limits for harmful components in the ambient air of the engine room

Limits of permissible concentration of harmful components, m^2/m^3 (for an ER ship)									
Soot	**Oxide of carbon**	**Hydrocarbons**		**Aldehydes**		**Oxides of nitrogen**		**Compounds sulphur**	
C	CO	In terms of petrol	C_2OH_1	CH_2O	P_2CO	NO	NO_2	SO_2	H_2SO_4
3,50	20	300	0,0015	0,70	0,05	30	9	0,05	0,10

The microclimate of the ER is characterised by the following indicators: temperature; humidity; content of harmful gaseous components; air flow rate. These indicators are provided by the ship's ventilation system. The ship's ventilation system can be: supply or exhaust; with a heating function or vice versa air conditioning. The norms of microclimate parameters for the ER of a container ship are given in Table 5.2.

Table 5.2 – Normalisation of ship's internal climate parameters

Climatic conditions in the premises of the ER	Air temperature, ^oC	Relative air humidity, %	Air flow rate, m/s
Summer operation period :			
– ER premises with heat generation at workplaces	34 / 25 – 28	70 – 40 / 60 – 40	1.0 – 1.5 / 1.0 – 1.5
– without heat generation at workplaces	20 – 23 / 23 – 25	60 – 40 / 60 – 40	0. – 0.3 / до 0,5
– isolated control stations	20 – 23 / 22 – 28	60 – 40 / 60 – 40	0,2 – 0,5 / 0,5 – 1,0
Winter period of operation:			
– ER working rooms	22	–	1.5
– isolated control stations	25	60 – 40	0.5 – 1.0
*** In the numerator -** for northern and temperate regions, in the denominator - for southern regions			

The most uniform lighting of the container ship's engine room allows for high performance of ship's mechanics and other personnel. Good lighting of the engine room and the workplace reduces eye fatigue, which can occur due to over-adaptation when the vision is transferred from a fairly well-lit area of a surface or object to a poorly lit one.

By its physical nature, light is an electromagnetic wave produced by a source. The power of such a light flux within the visible part of the spectrum is commonly called luminous flux.

Lux (lux) is used as a unit of measurement for assessing the degree of illumination of premises. Fluorescent and incandescent lamps are used as a source of light generation on ships of the sea and river fleet. Table 5.3 shows the sanitary standards for artificial lighting in different areas of a ship of the sea and river fleet.

Table 5.3 - Artificial lighting standards on fleet vessels

ER premises, workplace	Work surface	Illumination standardisation plane	Minimum illumination, lux	
			Fluorescent lamp	Incandescent light bulb
Engine room, premises for auxiliary mechanisms, control stations	instrumentation on the deck	Horizontal	100	50
		Vertical	500	300
	gangway steps	Horizontal	75	30
	instrument scale	Vertical	300	200
	control panel	Vertical	200	150
Boiler room	boiler instrumentation on the deck	Horizontal	100	50
		Vertical	300	200
Gyrocompass, tiller, unit and pump rooms	instrumentation on the deck	Horizontal	75	30
		Vertical	100	50
Battery powered	instrument scale	Vertical	300	200

Achievement of the standards specified in Table 5.3 is the basic task in the implementation of calculations of artificial lighting in the premises of ships. In addition, when providing lighting, based on long-term practical experience, the basic requirements for artificial lighting have been formed:

– ensuring the required level (Table 5.3) and maximum uniformity (homogeneity) of lighting at workplaces;

– ensuring the absence of different shadows or, conversely, glare on work surfaces;

– ensuring the absence of voltage changes in the power supply network and light flux fluctuations.

5.2. Calculation of the required level of artificial lighting in the ship's engine room

The purpose of the calculation is to determine the required number of DRL lamps to ensure lighting standards in the engine room of a container ship. To do this, a well-known method is used, which also takes into account the flux reflected from the ceiling and walls of the container ship's ER room.

The luminous flux that can be obtained from a certain group of lamps Fl, Lm of luminaires with fluorescent lamps installed in them is determined by the following formula:

$$\sum \Phi_{л} = E_{н} \cdot S \cdot z \frac{k}{\eta},$$

where S – is the area of the container ship's engine room that needs to be illuminated, m; E_{n} – is the minimum standardised illumination of the container ship's engine room, Lx; k – is the safety factor; z – is the coefficient that takes into account the minimum illumination of the container ship's engine room, which is actually the ratio E_{medium}/E_{min}; h – is the coefficient that takes into account the use of the available luminous flux from the corresponding lamp.

The value of the area of the engine room of a container ship requiring artificial lighting is

$$S = A \cdot B = 29.5 \cdot 24{,}5 = 722\ м^2$$

The minimum permissible illumination rate for the engine room of a container ship is about 50 lux. The minimum illumination rating factor for DRL lamps is 1.1. The safety factor is – 1.5.

The coefficient that takes into account the use of the available luminous flux from the corresponding lamp depends on: Luminous efficacy; luminous intensity distribution curve; reflection coefficients from the ceiling – $\rho_{rceiling}$; and walls – ρ_{rwall}; luminaire height – H_{p}; and room factor. Indicator of the room:

$$i = \frac{S}{H_p \times (A + B)} = \frac{722}{6{,}5 \times (29{,}5 + 24)} = 2{,}1$$

where H_{p} – is the height of the luminaires above the working surface in the engine room of the container ship (6.5 m); $\rho_{rceiling}$ = 30%; ρ_{rwall}= 10%; η = 0.7.

Then the light flux in the engine room of the container ship, from a group of fluorescent lamps, will be:

$$\sum \Phi_{л} = 50 \times 722{,}75 \times 1{,}2 \frac{1{,}5}{0{,}7} = 92925\ \text{lm.}$$

The light flux from one DRL lamp is F_{l} = 18210 lm. For artificial lighting of the container ship's ER, we choose a DRL-750 lamp. Then we will determine the required number of such lamps in the ER, which will be enough to create the above illumination standards:

$$N = \sum \Phi_л \big/ \Phi_л = 92925 \big/ 18210 = 5 \text{ pieces.}$$

5.3. Ensuring environmental standards with marine internal combustion engines

Running diesel engines are an intensive source of noise, heat and chemical pollution. The problem of reducing pollutant emissions from running diesel engines is one of the most important tasks of the diesel engine industry, which affects human health and the preservation of the gene pool.

According to the Law of Ukraine 'On Protection of Atmospheric Air', our country is obliged to comply with the requirements of the Technical Code for Emissions of Nitrogen Oxides from Marine Engines, which is an integral part of the International Convention for the Prevention of Pollution from Ships (MARPOL 73/78). Therefore, research in the field of comprehensive improvement of the energy and environmental performance of marine diesel engines is relevant.

The most well-known methods of reducing nitrogen oxide emissions in the shipbuilding industry, such as selective catalytic reduction and exhaust gas recirculation, have a number of serious drawbacks, such as significant design complexity and increased fuel consumption. The rise in fuel prices is forcing shipbuilders to look for methods that simultaneously reduce nitrogen oxide emissions without significantly affecting the fuel efficiency of the diesel engine.

Among such methods, those aimed at improving the workflow by influencing the thermodynamics of fuel combustion in the internal combustion engine cylinder deserve special attention.

The most toxic component of diesel exhaust is nitrogen oxides. They are formed in the combustion chamber of a diesel engine by oxidation of air nitrogen and nitrogen from nitrogen-containing fuel molecules. However, the latter makes up no more than

0.2% of the fuel weight, so only the oxidation of atmospheric nitrogen is usually considered.

Chemically inert under normal conditions, nitrogen reacts with air oxygen in the combustion chamber of a diesel engine to form mainly NO oxide at elevated pressures and temperatures (above 2000 K). Nitrogen is oxidised behind the flame front in the area of combustion products formation. This process involves the dissociation of oxygen, nitrogen, hydrogen and water vapour molecules, the products of which are characterised by increased chemical activity.

The conditions for reducing nitrogen oxides NO_x and specific fuel consumption are opposite, since to reduce NO_x, the maximum cycle temperature must be reduced, and to reduce specific fuel consumption, it must be increased. At the same time, the thermal regime largely determines the thermal stress of the cylinder-piston group parts and, consequently, the reliability of engine operation.

Currently, two groups of measures aimed at reducing nitrogen oxide emissions have been identified - primary and secondary. Primary measures are related to improving the combustion process in the engine cylinder. Secondary measures cover three groups of engine system improvements: air charge preparation, fuel preparation and fuel supply, and the use of catalytic converter treatment. Therefore, the technology for creating a low-toxic diesel engine is as follows.

Primary measures:

– workflow organisation;

– control of the fuel injection start angle;

– control of gas distribution phases;

– increasing the compression ratio and optimising the combustion chamber;

– application of Atkinson and Miller cycles;

– catalytic gas cleaning.

Secondary measures:

– fuel preparation and fuel supply;

– multiphase fuel supply;

– injection of water-fuel emulsion;

– controlled combustion of a homogeneous mixture;

– damping of fuel fluctuations in the fuel system.

Air charge preparation:

– reducing the temperature of the air at the cylinder inlet;

– supply of humid air or superheated steam;

– exhaust gas recirculation.

The effectiveness of primary and secondary measures is currently estimated as follows: primary measures can provide up to a 20% reduction in the standardised toxic components in exhaust gases; secondary measures can almost completely eliminate the emission of toxic components. The most effective secondary measures include the injection of a fuel-water emulsion (FEW) into the cylinder, humid air supply to the engine (HAM), and SCR (Selective Catalytic Reduction) catalysts.

5.4. Conclusions to the part

In this chapter, a detailed analysis of hazardous and harmful factors for ship's engineers and other personnel in the engine room of a ship was performed. A generalised calculation of artificial lighting in the ship's engine room was performed, and the impact of harmful components in the exhaust gases on human health was analysed.

CONCLUSION ON THE WORK

The qualification work on the topic: "***Modernisation of the fuel system of a medium-speed engine by using electrically controlled injectors***" contains all the recommended sections and drawings.

In the first part, a brief description and main characteristics of the dry cargo vessel and the 6CHN 46/58 marine diesel engine are given. It is established that the engine can provide the required operating parameters and is well placed in the engine room.

The second part calculates the duty cycle for the nominal operating mode of the 6CHN 46/58 marine diesel engine with the standard fuel system and the upgraded one (using electrically controlled injectors and fuel accumulators). It was found that the upgraded engine with the new fuel system significantly outperforms the prototype engine in terms of its main efficiency and environmental performance. Thus, the power increase is 48.5 kW, and the specific effective fuel consumption is reduced by 4.4 g/(kWh).

The third part analyses promising directions for the development of fuel systems for modern medium-speed marine diesel engines. The fuel system and its main elements of a medium-speed marine diesel engine of the 6CHN 46/58 type were developed and calculated. The main geometrical and operational parameters were obtained. The system elements provide all the necessary fuel injection parameters for the 6CHN 46/58 engine, which were included in the calculation of the operating one.

In the fourth part, it is established that the annual economic effect for the consumer from the use of the modernised engine, compared to the basic one, is 330393.07 UAH. Modernisation of the basic 6CHN 46/58 engine will pay off in 5 months.

Part 5 provides a detailed analysis of hazardous and harmful factors for ship's mechanics and other personnel in the ship's engine room. A generalised calculation of artificial lighting in the ship's engine room was carried out, and the impact of harmful components in exhaust gases on human health was analysed.

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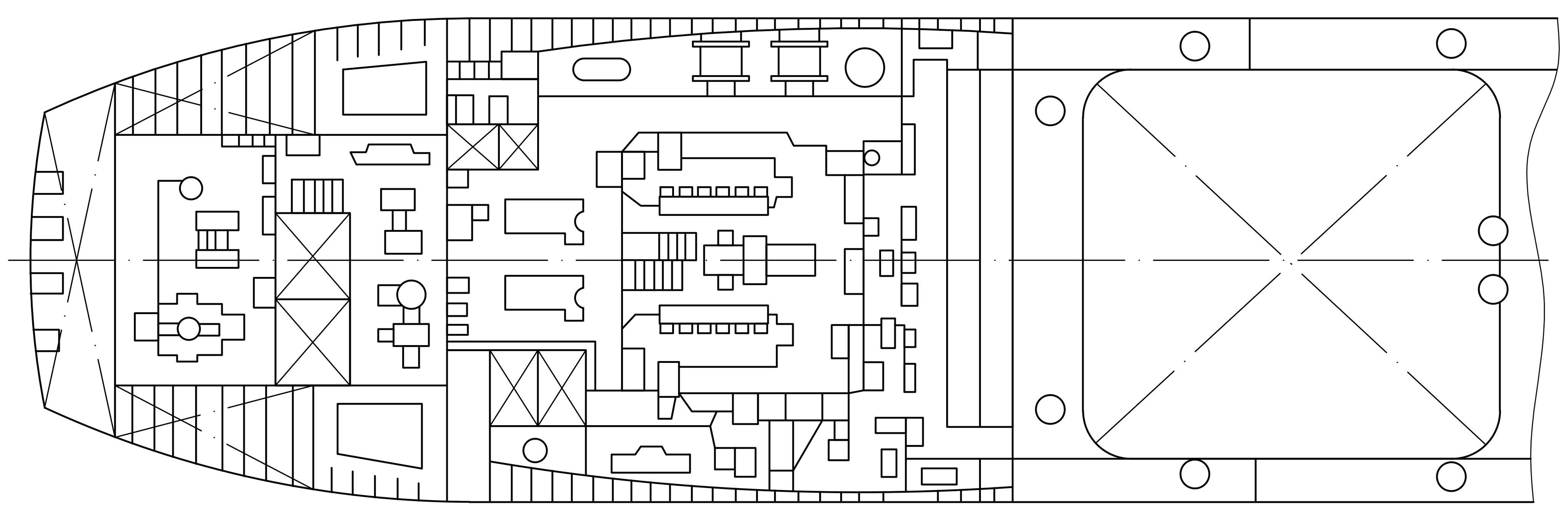

The longest length, m	124.2
Length between perpendiculars, m	113.9
Width, m	16.4
Board height, m	7.5
Draught on the KVL, m	4.5
Draft by cargo mark, m	5.5
Deadweight at draft of 4.5 m, t	3370
Deadweight with draft of 5.5 m, t	5030
Speed, knots.	12,4

Meas.	Letter	№ document	Caption	Date	KWM.142.6221m.07.01
Student		Iovenko V.A.			General view of the ship
Manager		Timoshevskyi B.G.			Let. / Mass / Scale
					Letter / Letters
Head of Depart.		Gogorenko O.A.			National University of Shipbuilding Admiral Makarov

KWM.142.6221m.07.02

Format	Area	Position	Designation	Name	Number	Note
		1		Push rod		
		2		Valve actuator rocker arm		
		3		Cylinder head		
		4		Inlet valve		
		5		Cylinder bushing		
		6		Piston		
		7		Piston pin		
		8		Connecting rod		
		9		Crankshaft		
		10		Lower connecting rod head		
		11		Block crankcase		
		12		Camshaft		

KWM.142.6221m.07.02

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

Cross section of the 6HP 46/58 engine

Let.	Mass	Scale
		1:5

Letter | Letters

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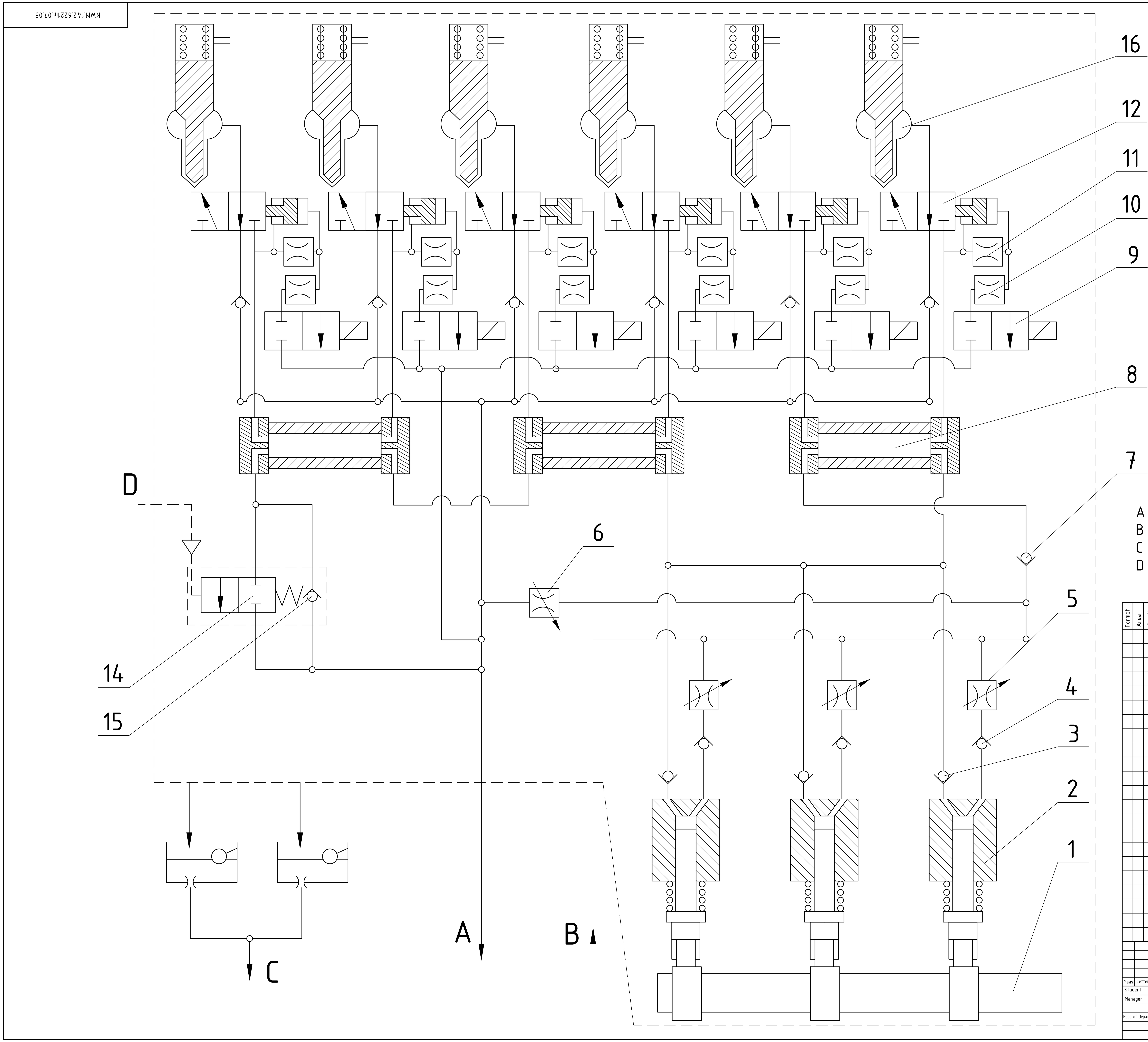

A – fuel drain;
B – fuel supply;
C – collection of leaks;
D – control air supply.

Format	Area	Position	Designation	Name	Number	Note
		1		Pump drive shaft		
		2		High pressure fuel pumps		
		3		Pressure valve		
		4		Suction valve		
		5		Feed control throttle		
		6		Throttle to maintain		
				differential pressure		
		7		Check valve to increase flow		
				rate when flushing the system		
		8		Fuel accumulator		
		9		Starting valve		
		10		Starting valve throttle		
		11		Throttle of the main flow		
				control valve		
		12		Main flow control valve		
		13		Check valve for the		
				drainage cavity		
		14		System pumping and emergency		
				pressure relief valve		
		15		High pressure relief valve		
		16		Injector		

KWM.142.6221m.07.03

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

Hydraulic diagram of the accumulator fuel system

Let. | Mass | Scale

Letter | Letters

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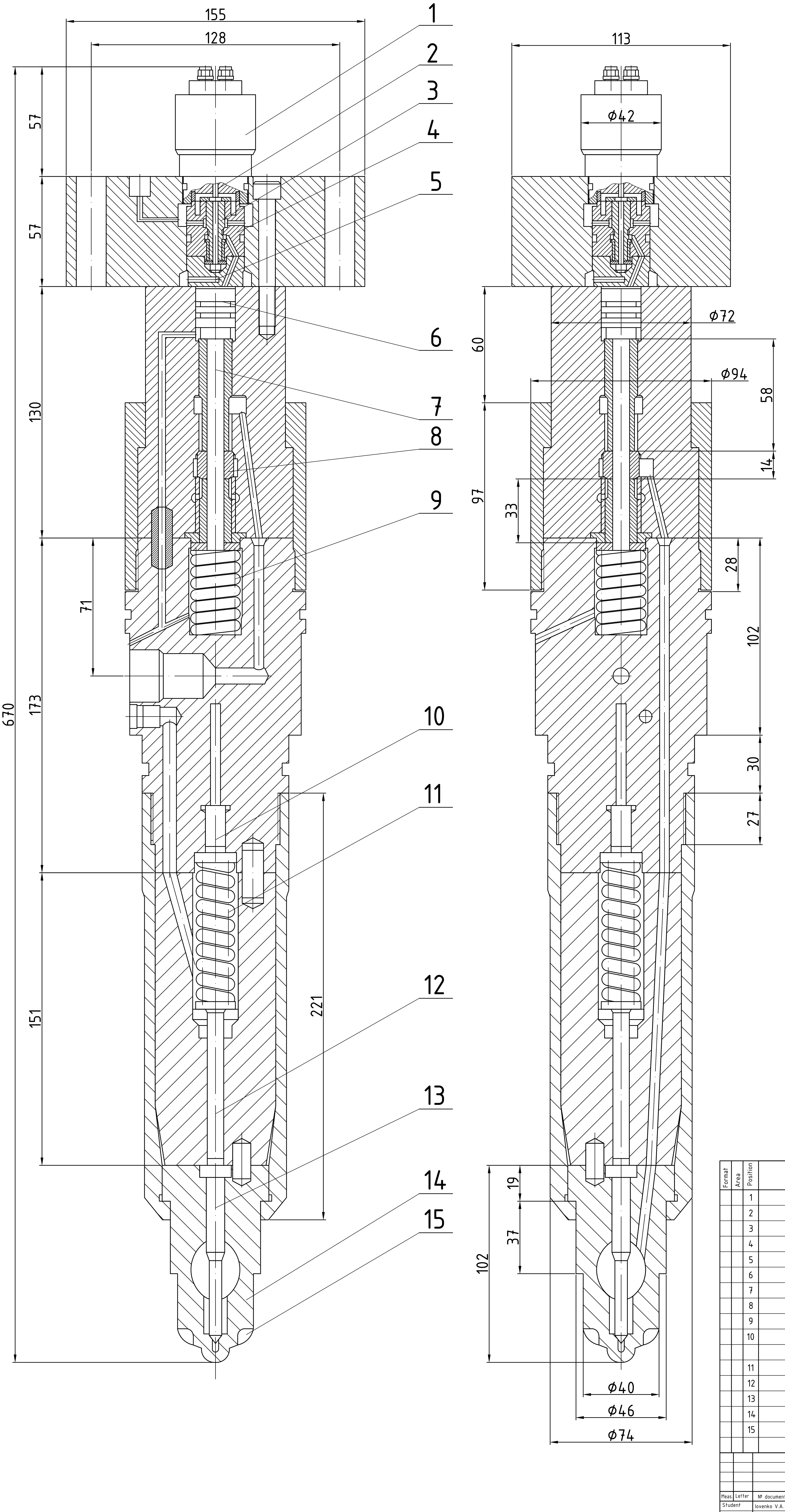
KWM.142.6221m.07.03

KWM.142.6221m.07.04

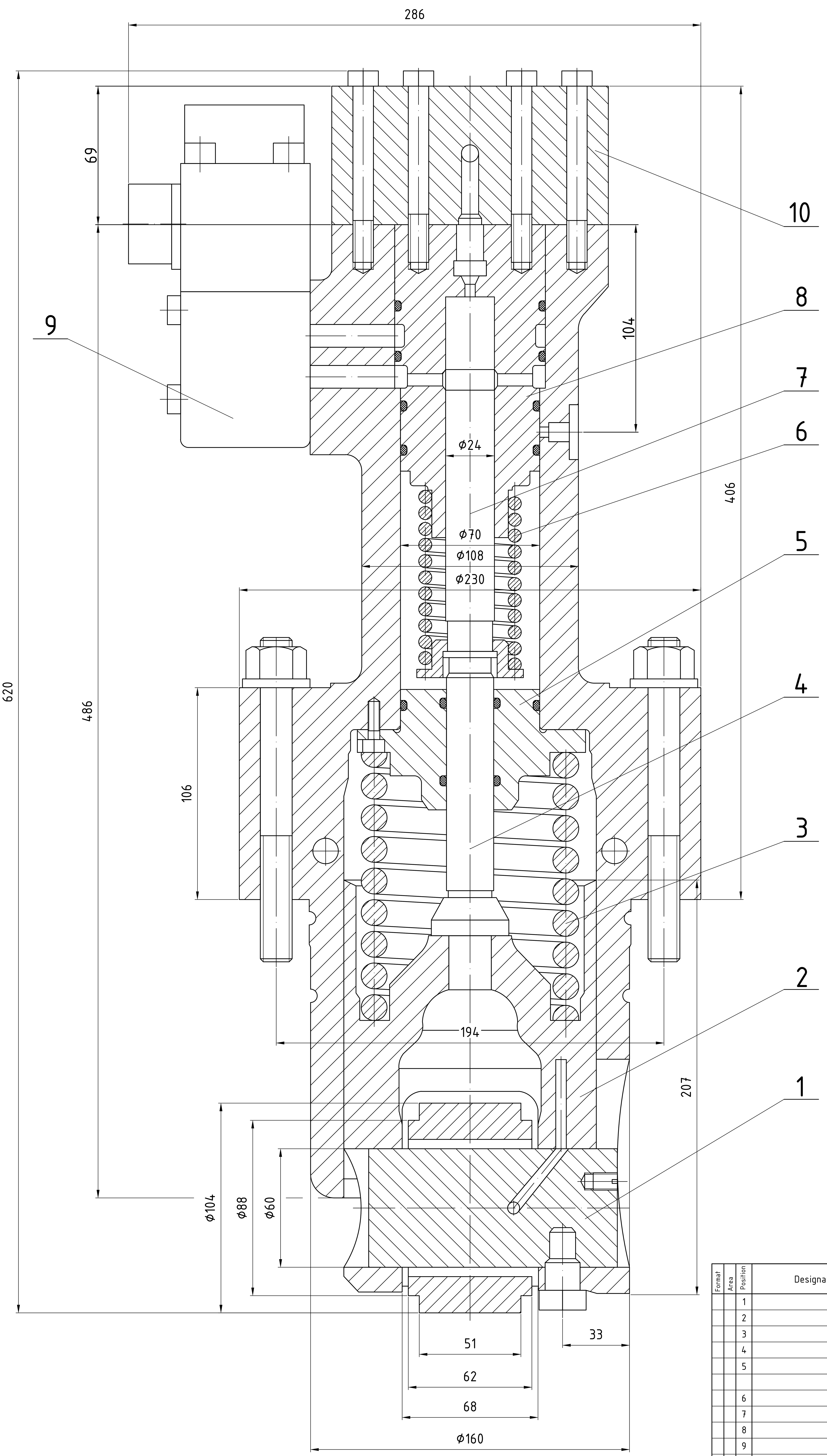


Format	Area	Position	Designation	Name	Number	Note
		1		Solenoid		
		2		Solenoid anchor		
		3		Starting valve		
		4		Starting valve housing		
		5		Valve throttling opening		
		6		Main valve actuator piston		
		7		Main valve guide		
		8		Main valve		
		9		Return spring		
		10		Hydraulic locking piston for the spray needle		
		11		Spray needle valve load spring		
		12		Stem		
		13		Spray needle valve		
		14		Sprayer body		
		15		Spray cooling cavity		

KWM.142.6221m.07.04

Meas	Letter	№ document	Caption	Date		Let.	Mass	Scale
Student		Iovenko V.A.			Electrically controlled injector			1:1
Manager		Timoshevskyi B.G.				Letter	Letters	
Head of Depart.		Gogorenko O.A.				National University of Shipbuilding Admiral Makarov		

KWM.142.6221m.07.05



Format	Area	Position	Designation	Name	Number	Note
		1		Axis of the pusher roller		
		2		Pusher		
		3		Push rod return spring		
		4		Push rod		
		5		Dividing diaphragm with push rod guide		
		6		Plunger return spring		
		7		Plunger		
		8		Plunger bushing		
		9		Flow control throttle valve		
		10		Pump cover		

KWM.142.6221m.07.05				
Meas. Letter	№ document	Caption	Date	
Student	Iovenko V.A.			High pressure fuel pump
Manager	Timoshevskyi B.G.			
Head of Depart.	Gogorenko O.A.			

Let.	Mass	Scale
		1:1
Letter	Letters	

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KWM.142.6221m.07.06

Format	Area	Position	Designation	Name	Number	Note
		1		Throttle valve actuator		
				solenoid		
		2		Throttle valve body		
		3		Throttling cone for the		
				flow control valve		
		4		Safety valve		

Meas.	Letter	№ document	Caption	Date				
					KWM.142.6221m.07.06			
Student		Iovenko V.A.			Throttle valve for controlling the fuel supply	Let.	Mass	Scale
Manager		Timoshevskyi B.G.						4:1
						Letter	Letters	
Head of Depart.		Gogorenko O.A.				National University of Shipbuilding Admiral Makarov		