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

Department of Internal Combustion Engines, Installations and Technical Operation

"Admitted to defense" Acting Head of the Department
Oleksii HOHORENKO
____
« ___» __________ 2024 y.

QUALIFICATION WORK

IMPROVING THE EFFICIENCY OF THE SHIP'S
OF A 16 V 48/60 MEDIUM-SPEED MOTOR AT RATED OPERATION

Specialty 142 - Power engineering

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

Head of the work Alexander MITROFANOV

Education applicant Serhii ZABRODIN

Mykolaiv 2024

National University of Shipbuilding
Admiral Makarov National University of Shipbuilding

Institute, faculty *Mechanical engineering educational and scientific*
Department *Internal combustion engines, installations and technical operation*
Degree *Master*
Specialty *142 Power engineering*
(code and name)
Educational program *Internal combustion engines*

APPROVE
Head of the Department *ICE, I and TO*
___________ Oleksii GOGORENKO
«____» ______________20___

T A S K
FOR A QUALIFYING JOB FOR AN EDUCATION APPLICANT

Zabrodin Sergey Georgievich
(last name, first name, patronymic)

1. Theme of the work *Improving the efficiency of the ship's of a 16 V 48/60 medium-speed motor at rated operation*

2. Head of work *D., Associate Professor Mitrofanov O. S.*,
(last name, first name, patronymic, academic degree, academic title)
approved by the order of the higher education institution from "___"__________20___ №________

3.Deadline for submission of the job by the applicant ____________________

4. Initial data for the work *Diesel engine power - 19,200 kW; crankshaft speed – 500 min^{-1}.*

5. Contents of the explanatory memorandum (list of questions to be developed) *1. Peculiarities of using the 16V 48/60 engine as part of a ship's power plant. 2. Calculation of the duty cycle for the rated mode of operation of the diesel generator. 3. Increasing the efficiency of the engine at the rated mode of operation. 4. Economic justification of the modernization of the diesel generator. 5. Analysis and development of labor and environmental protection measures*

6. List of graphic material (with an exact indication of mandatory drawings)
1. General view of the vessel. 2. Overall dimensions of the M.A.N. 16V 48/60 CR engine. 3 Engine type M.A.N. 16V 48/60 CR. 4. Combined graphs of regression functions N_e and P_Z, in the form of equation lines. 5. Turbocharger TK92. 6. Scheme of lubrication of the turbocharger TK92

7. Consultants of work sections

Section	Name, initials and position of the of the consultant	Signature, date	
		task issued	task accepted
Economic section	*Mitrofanov O. S.*		
Labor burial ground section	*Mitrofanov O. S.*		

8. Date of issue of the task ______________________________

CALENDAR PLAN

№	Name of the work stages	Timeframe for completion of work stages	Note
1.	*Features of the 16 V 48/60 engine application*		
	as part of the ship's power plant.		
2.	*Calculation of the duty cycle for the nominal operating mode*		
	of a diesel generator.		
3.	*Increase engine efficiency at rated operating conditions*		
	operating mode.		
4.	*Economic justification for the modernization of the*		
	of the diesel generator.		
5.	*Analysis and development of occupational health and safety measures*		
	and environmental protection measures.		

Education applicant ______________ *Serhiy ZABRODIN*
(signature) (last name and initials)

Head of work ______________ *Alexander MITROFANOV*
(signature) (last name and initials)

ANNOTATION

The thesis is devoted to the study of the design, operation and improvement of the 16V 48/60 marine diesel engine as part of the power plant of a bulk carrier.

The paper analyzes the architectural and technical features of the vessel, considers modern fuel systems of the Common Rail type, which ensure an increase in engine efficiency and compliance with environmental standards. The engine duty cycle was calculated for the nominal operating mode using the Grinevetsky-Masing methodology, which allowed optimizing its performance characteristics.

The engine efficiency was improved, including the use of supercharging systems and gas turbochargers. Based on the data obtained, technical improvements were proposed to increase engine performance and reduce fuel consumption.

The economic justification of the modernization showed the feasibility of implementing changes that can achieve significant savings in operating costs with a short payback period. The study also focuses on labor and environmental protection, including compliance with international safety and environmental standards.

The research results can be used to optimize the operation of ship power plants, ensure their efficiency and environmental safety.

Key words: internal combustion engine, Turbocharging, effective indicators, specific effective fuel consumption, power plant.

TABLE OF CONTENTS

INTRODUCTION

In accordance with the task of the Department of ICE, I and TO: “Improving the efficiency of the ship’s medium-speed motor at rated operation”, the design features of the 16V 48/60 marine engine used as part of the power plant of a bulk carrier were analyzed.

The fuel system, engine operating cycle and the influence of fuel equipment parameters on its efficiency were studied. Particular attention is paid to the improvement of the design taking into account modern requirements for energy efficiency and environmental safety.

The paper also considers the economic feasibility of engine modernization, which allows to increase its performance and reduce operating costs. The paper also examines occupational safety and health measures on board the vessel that ensure the crew's safe work and environmental protection measures aimed at minimizing the environmental impact of power plants.

The results of the research can be used to improve the performance of ship power plants, reduce their environmental impact and increase the efficiency of the fleet.

SECTION 1

PECULIARITIES OF APPLICATION OF THE 16V 48/60 ENGINE AS PART OF A SHIP'S POWER PLANT

1.1. General characteristics of a bulk carrier

The bulk carrier (Figure 1.1) is a double-hulled vessel with a carrying capacity of 20,000 tons. The vessel has four equally sized (28.20 m x 21.00 m) cargo holds separated by double skin transverse bulkheads built on stools, with spaces in between that are used as water ballast tanks and also for storing the hold access barrels. This arrangement allows the reinforcements to be placed inside these tanks and in the double-hull space so that the holds are clear for easy cleaning and maintenance. The design of the tank complies with the Heavy Cargoes notation with a maximum load of 18 tons/m2. Two tiers of steel coils can be loaded and can be serviced by a grab unloading. All holds are mechanically ventilated and can carry dangerous goods. All hatch covers open to the full size of the hold and are fully closed, folding in pairs, end type. Cargo handling is carried out by three NMF 24 t SWL x 28 m electric deck cranes located between the holds.

The upper deck has been kept free of obstructions with all pipes, cables and valves located in passageways above the side ballast tanks or in the double-bottom tunnel. The maneuverability of the vessel is enhanced by a bow thruster with a CP propeller developing 600 kW at 437 rpm.

Electrical requirements are met by three alternators, each with a capacity of 575 kW.

The bunker tanks of the engine room are protected from the sea, and the large capacity of the ballast tanks eliminates the requirement for ballast transportation in the holds and facilitates ballast replacement.

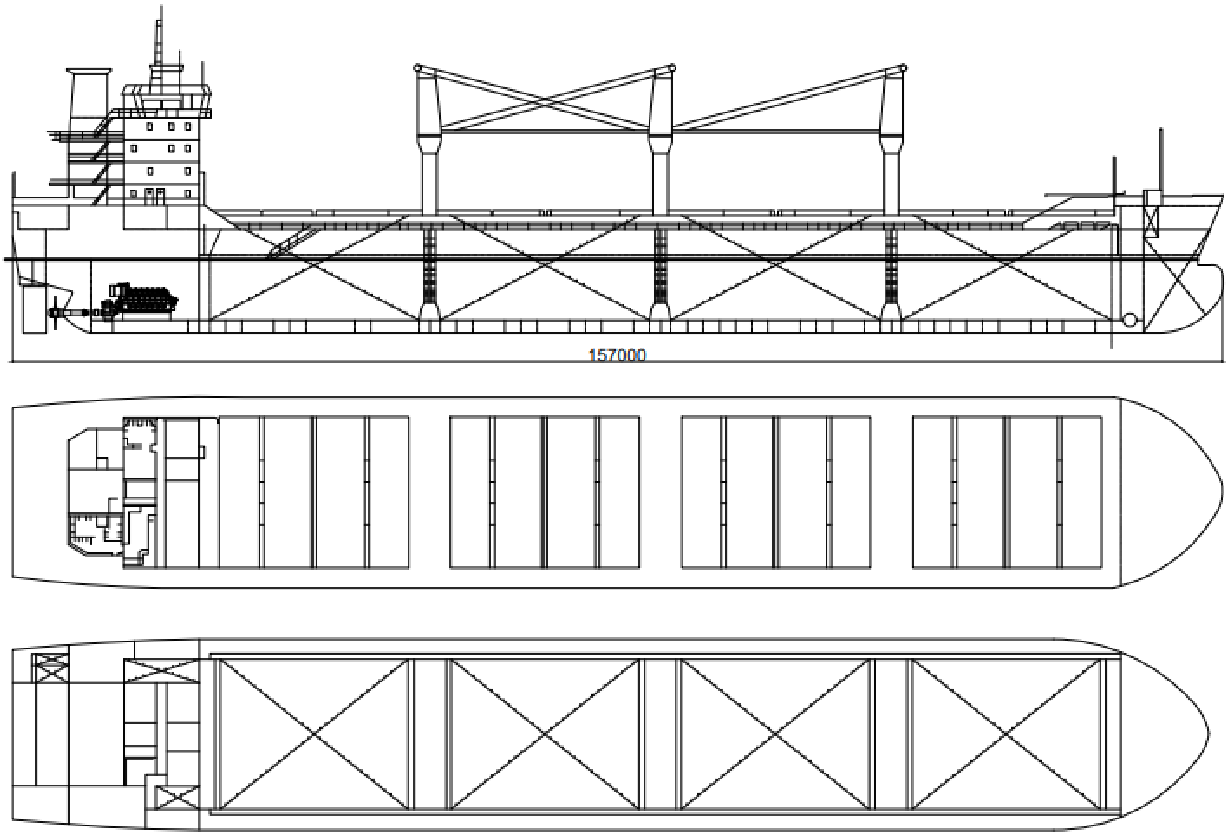


Fig. 1.1 - General view of a bulk carrier

Table 1.1 – Main characteristics of a bulk carrier

№ з.п.	Parameter	Dimensionality	Meaning
1	Own weight of the vessel	т	9650
2	Cargo capacity	т	14000
3	Length.	m	157.901
4	Width	m	23,20
5	Vessel category	–	Bulker
6	Vessel type	–	Ship
7	Maximum speed	nodes	14,0

1.2. Description of the design and characteristics of the ship's power plant

The M.A.N. 16V 48/60CR engine was initially introduced in 1989 with a rating of 885 kW/cylinder, which was gradually increased to 1050 kW/cylinder. The further development of the 48/60 engine design was stimulated by the high tractive power requirements of cruise ships, ferries and large multipurpose cargo ships, as well as strict environmental regulations. As a result, the CR version was launched in 2002 with a 14% increase in specific output of 1200 kW/cylinder at 500 and 514 rpm from a lighter and more compact design. The 48/60 CR program covers a power range from 7200 kW to 21,600 kW with 6, 7, 8 and 9 cylinder inline and 12, 14, 16 and 18 cylinder V-twin models. The 16V 48/60 CR engine has only one common rail exhaust manifold located in the cylinder swing and only one turbocharger. Two charge air coolers are installed. The turbocharger and the two EGRs are combined into a single module.

Figure 1 shows a cross-section of the engine. The engine has a Common Rail fuel system. One of the main features of this system is the independence of the injection processes from the crankshaft angle and engine operation mode, which makes it possible to achieve high injection pressure at partial modes, which is necessary to meet current and future environmental requirements. In a common rail system, a high-pressure pump injects diesel fuel under high pressure (up to 300 MPa, depending on the engine mode) into a common fuel line of a significant volume (accumulator).

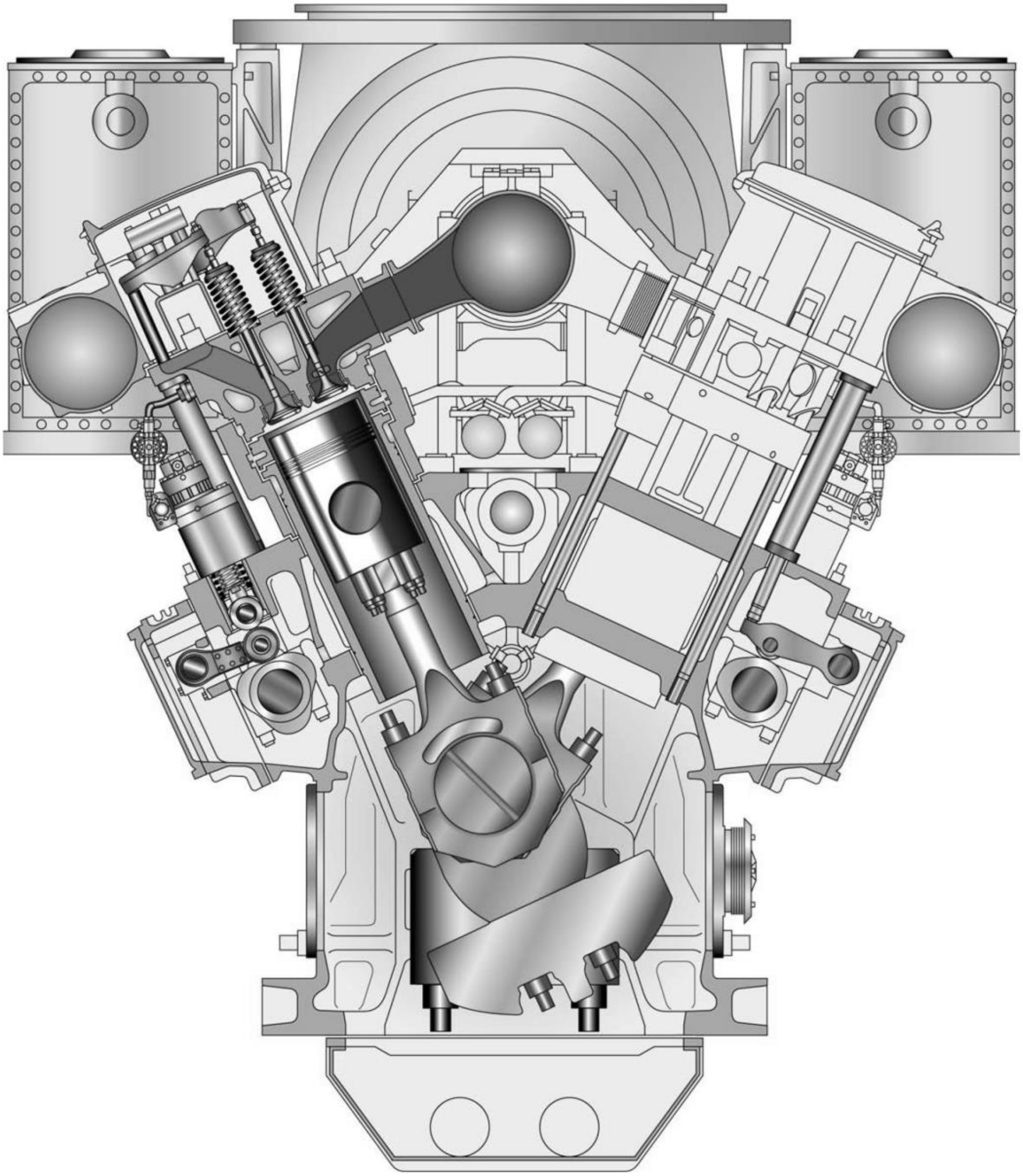


Fig. 1.2 Cross section of the M.A.N. 16V 48/60 CR engine

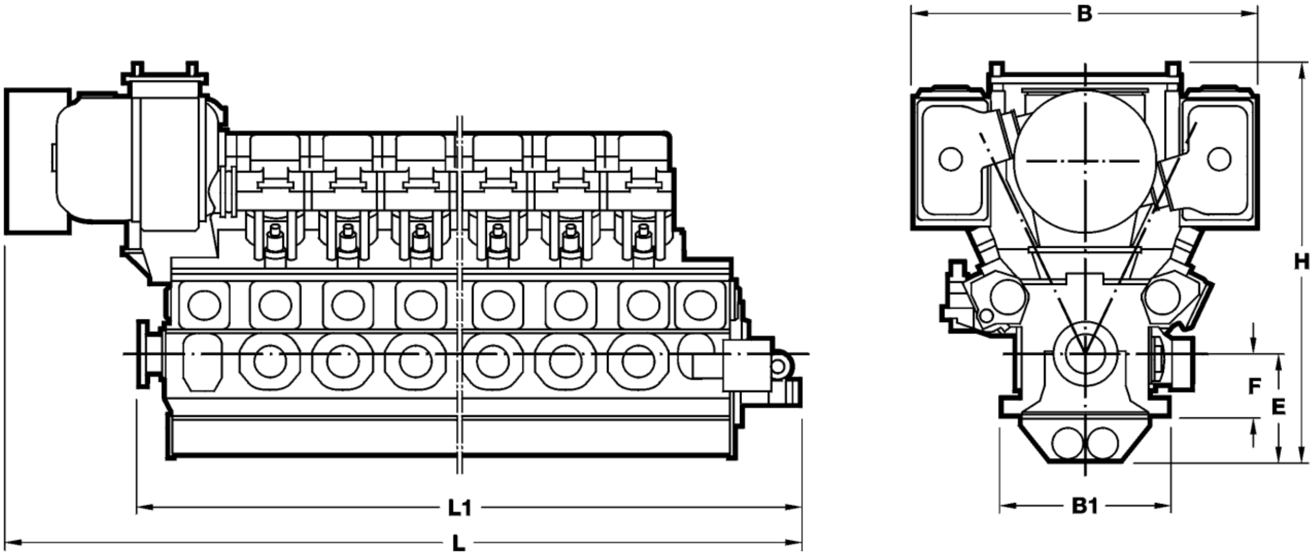


Fig. 1.3 Overall dimensions of the M.A.N. 16V 48/60 CR engine:
L - Motor length without module; L1 - Motor length with module; B - Motor width with module; B1 - Motor base width; E - Height to center of crankshaft rotation axis with motor pallet; F - Height to center of crankshaft rotation axis without motor pallet; H - Total motor height (with module). Values of overall dimensions:

- L = 13,1 m;
- L1 = 10,915 m;
- B = 4,7 m;
- B1 = 2,28 m;
- E = 1,41 m;
- F = 0,83 m;
- H = 5,355 m-code.

Mass values:

- weight of the engine without flywheel: mdw = 236 tons;
- Weight of the engine flywheel: mm = 4.3 tons.

Description of the main engine systems

- Fuel system

The engine is equipped with a common rail fuel system. This is a storage fuel system (common rail), a fuel supply system used in diesel engines. In a common rail system, a high-pressure pump pumps diesel fuel under high pressure (up to 300 MPa, depending on the engine's operating mode) into a common fuel line of a significant volume (accumulator).

- Fuel system injectors

The injectors of the Common Rail system are electronically controlled. Electro- hydraulic injectors with electromagnetic control valves inject diesel fuel under high pressure into the cylinders. A feature of the system is the independence of the injection process from the crankshaft angle and engine operation, which makes it possible to achieve high injection pressure at partial speeds.

- The principle of operation of the Common rail system

The fuel is taken from the fuel tank by the fuel booster pump (low pressure) and through the fuel filter enters the high pressure fuel pump (HPFP). The HPFP delivers fuel to the pressure line, which acts as a pressure accumulator. The control unit regulates the performance of the HPP to maintain the required pressure in the line as fuel is consumed.

The fuel line is connected to the injectors by fuel lines. An electromagnetic valve is built into each injector. When commanded by the control unit, the valve opens, injecting the required portion of fuel into the cylinder.

A general view of the Common rail fuel system is shown in Fig. 1.4

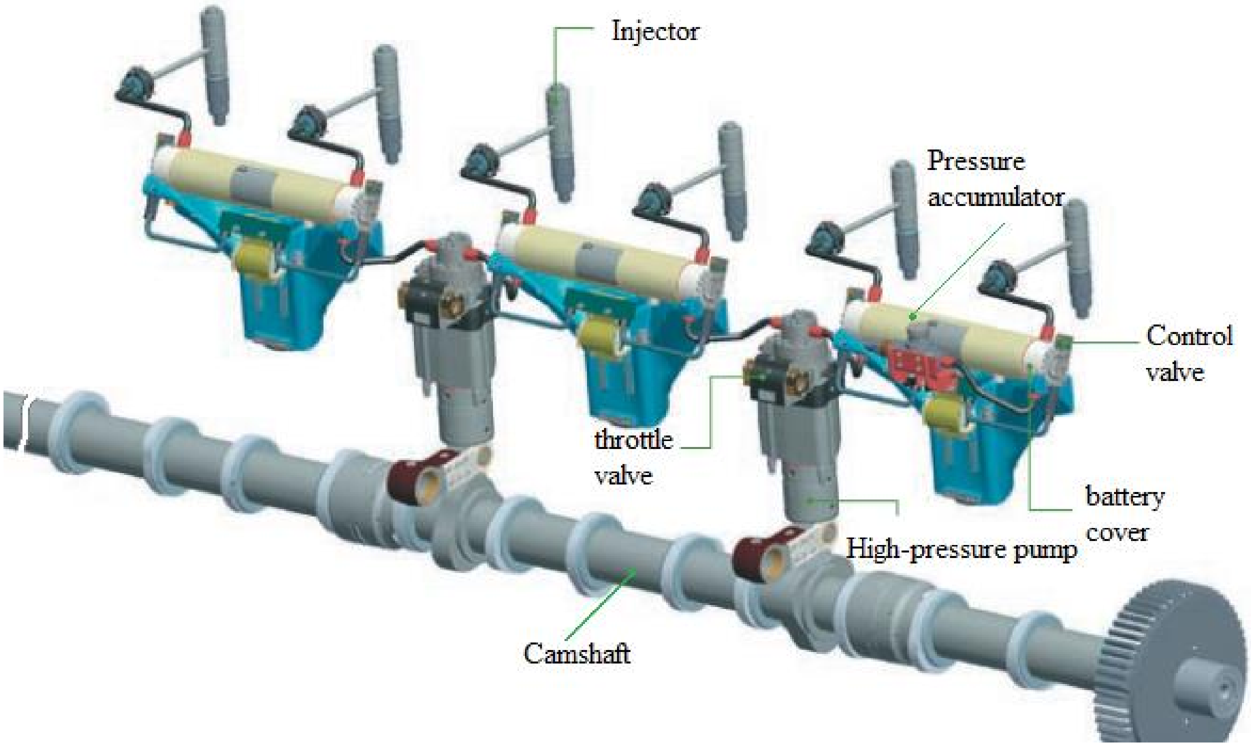


Fig. 1.4 Fragment of the common rail fuel system of the M.A.N. 16V 48/60 CR diesel engine

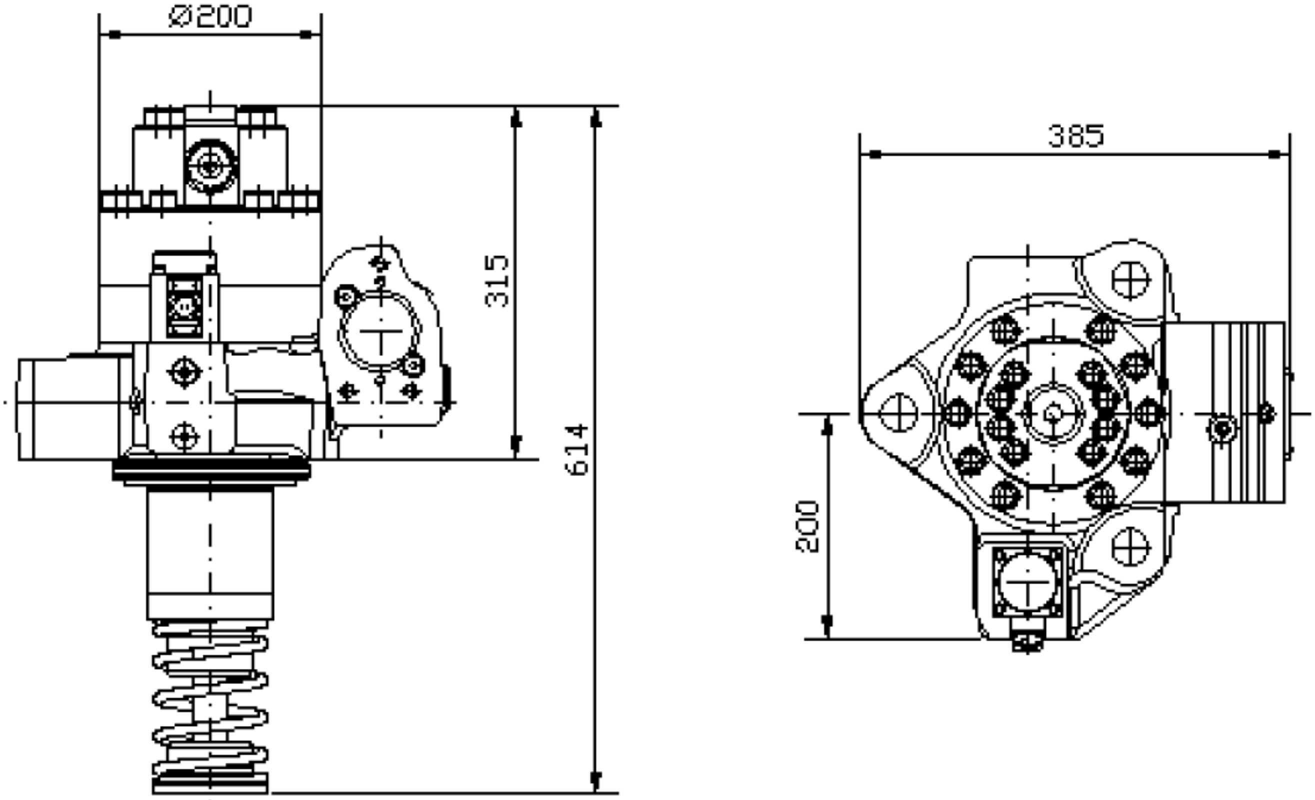


Fig. 1.5 High-pressure fuel pump for the 16V 48/60 CR engine

- Air supply and gas exhaust system (supercharging system)

The air supply system is designed to supply the air required for fuel combustion and cylinder purge in a certain amount and with specified parameters.

The M.A.N. 16V 48/60 CR engine is equipped with a constant pressure supercharging system (isobaric system). It has one large exhaust manifold in the cylinder camber, which is connected to each cylinder by a compensator. The engine has two intake manifolds and two charge air coolers (CAC). It has one centrifugal turbocharger of the TSA-77 type. The turbocharger and two aftercoolers are combined into a single module. Engine boost pressure pk = 420 kPa, i.e., the degree of pressure increase in the compressor Pk = 4.2.

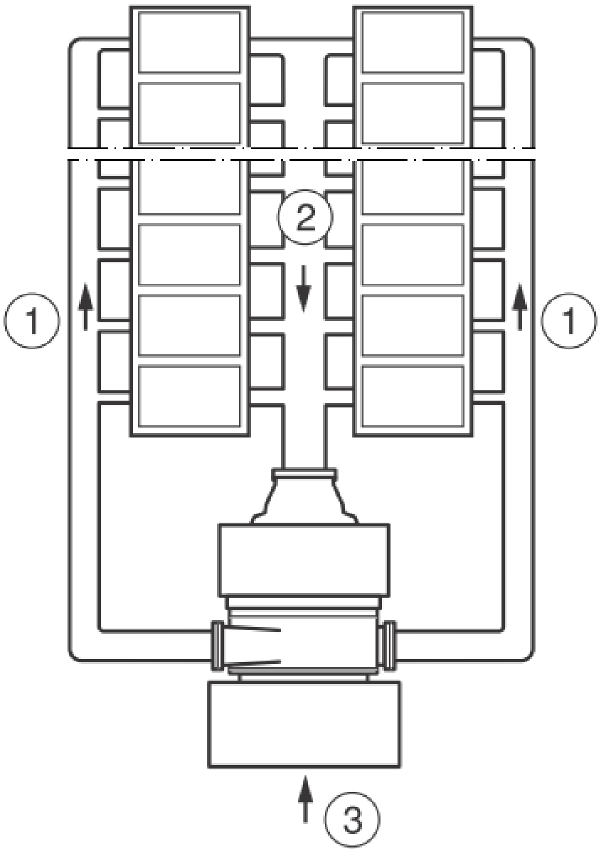


Fig. 1.6 Schematic diagram of the supercharging of the 16V 48/60 CR engine: 1 - charge air; 2 - exhaust gases; 3 - fresh air

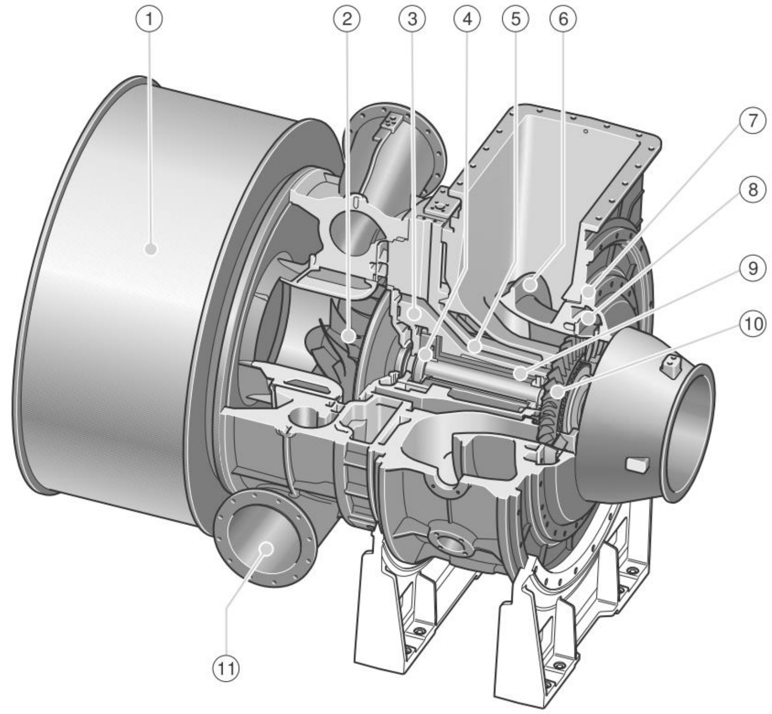


Fig. 1.7 Turbocharger of the TK-92 type.

1 - muffler; 2 - compressor wheel; 3 - receiving housing; 4 - thrust bearing; 5 - built-in air seal systems;

6 - exhaust diffuser; 7 - heat-resistant housing; 8 – nozzle ring;

9 - sliding bearing; 10 - turbine wheel blades; 11 - compressor housing.

Design features of the TK series turbochargers:

- low noise level of the compressor wheel;

- Simple service;

- Uncooled bearing housing;

- Easy access to the stubborn bearing;

- High sealing of air, oil pipes and ventilation system;

- Highly efficient patented exhaust diffuser;

- Optimized compressor casing cross-section;

- Long service life of the nozzle ring.

Turbocharger lubrication system

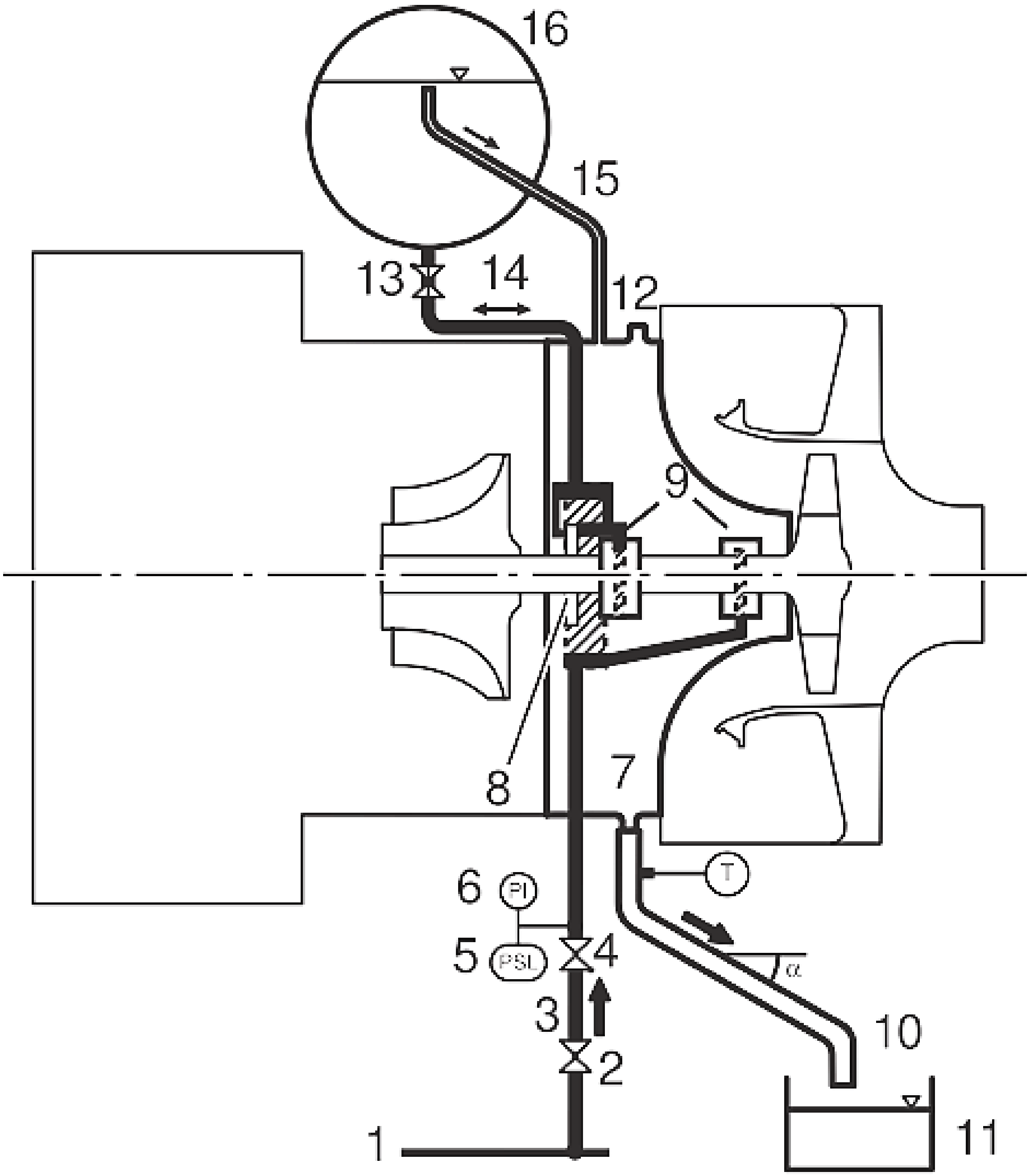


Fig. 1.8 Scheme of lubrication of the turbocharger TK-92

1 - supply pipe; 2 - pressure reducing valve; 3 - pipe leading to the turbocharger; 4 - check valve; 5 - pressure regulator; 6 - pressure gauge;

7 - receiving housing; 8 - thrust bearing with a disk;

9 - bearing bushing; 10 - drainage pipe; 11 - drainage crankcase or tank; 12 - ventilation; 13 - check valve; 14 - drainage pipe; 15 - overflow pipe; 16 - gravity tank.

The gas exhaust system ensures the most rational removal of exhaust gases from the cylinder. Rational exhaust means such an organization of gas exhaust that contributes to the maximum use of the working body energy both in the engine cylinder and outside it, as well as to high-quality cleaning and filling of the cylinders. In addition, it is necessary to ensure that the environmental impact of exhaust gases is as minimal as possible.

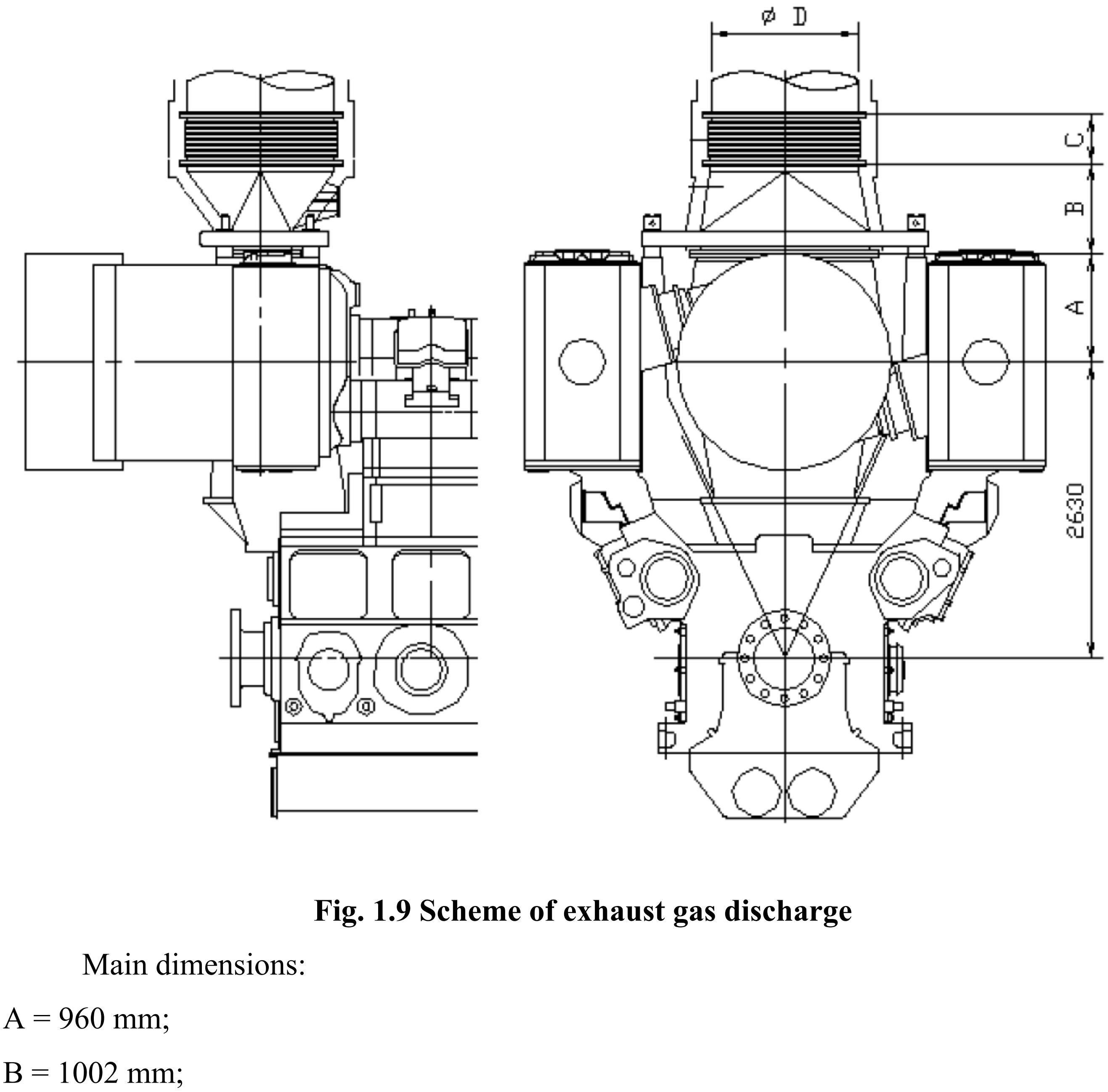


Fig. 1.9 Scheme of exhaust gas discharge

Main dimensions:

A = 960 mm;

B = 1002 mm;

C = 432 mm;

D=1520 mm.

- Engine lubrication and oil drainage system

The 16V 48/60 CR engine has a circulating lubrication system with a dry crankcase. This system is used in all low- and medium-speed engines used as main engines on ships. Two oil drain pipes are located at both ends of the engine oil pan and connected to it. There is an oil drainage tank under the engine. The drainage pipes must be short. The sloping ends of the pipes must be immersed in oil in such a way as to create a fluid seal between the crankcase and the tank.

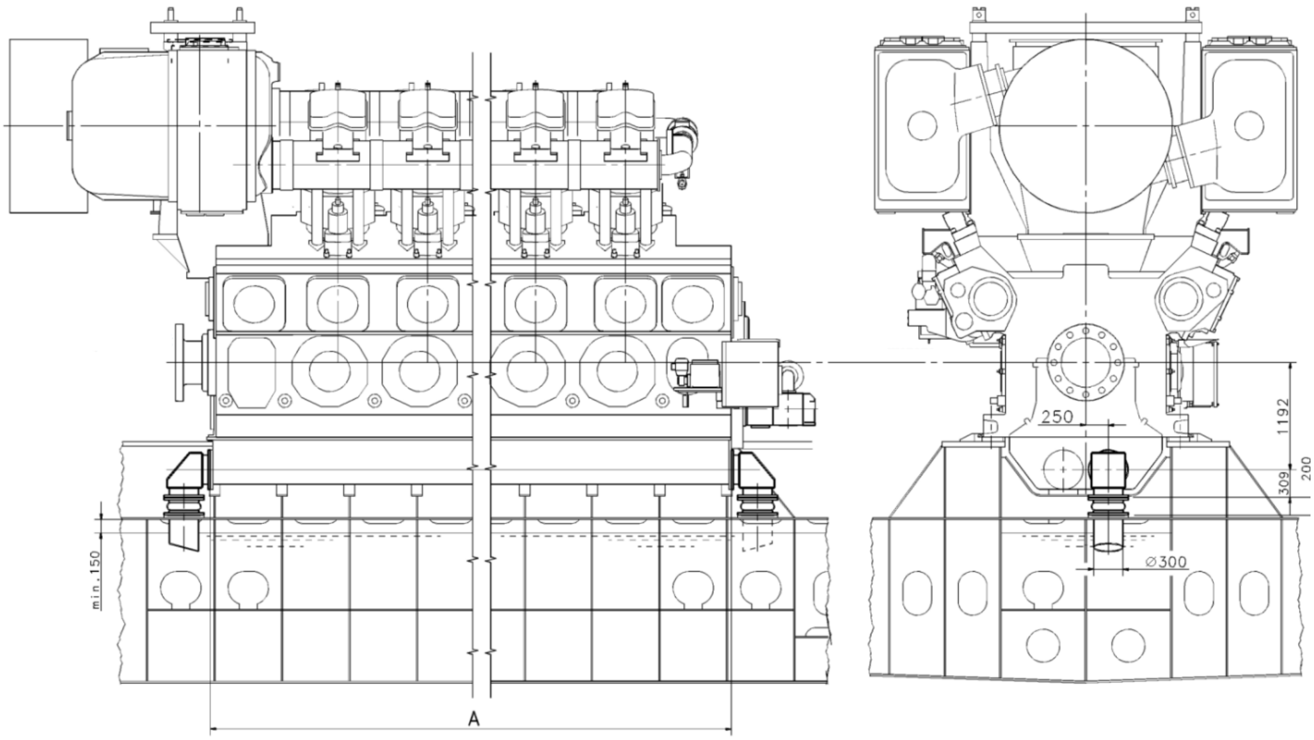


Fig. 1.10 Scheme of lubricant drainage where: A = 9593 mm.

1.3. Peculiarities of the engine operation on the energy consumer

The M.A.N. 16V 48/60 CR engine serves as the main engine on the vessel and is driven by the propeller, so it operates according to the screw characteristic of the adjustable pitch. The main mode of operation of the motor is nominal.

The propeller characteristic is the dependence of power and other engine performance parameters on the shaft speed when it operates with a propeller.

When the engine operates according to the helical characteristic, its power must be completely absorbed by the propeller, for which the power dependence on the rotation frequency is determined by the relationship

$$N = c2 \cdot nm,$$

where c2 – is the proportionality coefficient; m is the degree indicator (for water- isolating vessels m = 3).

A distinctive feature of propeller performance is the sharp decrease in engine power as the speed decreases. In order to provide increased torque (or power) at low speeds, a variable pitch propeller (VP) is used on the ship. Its use allows to operate according to the external characteristic in the entire range of shaft speed change and thus to make the best use of the engine's capabilities.

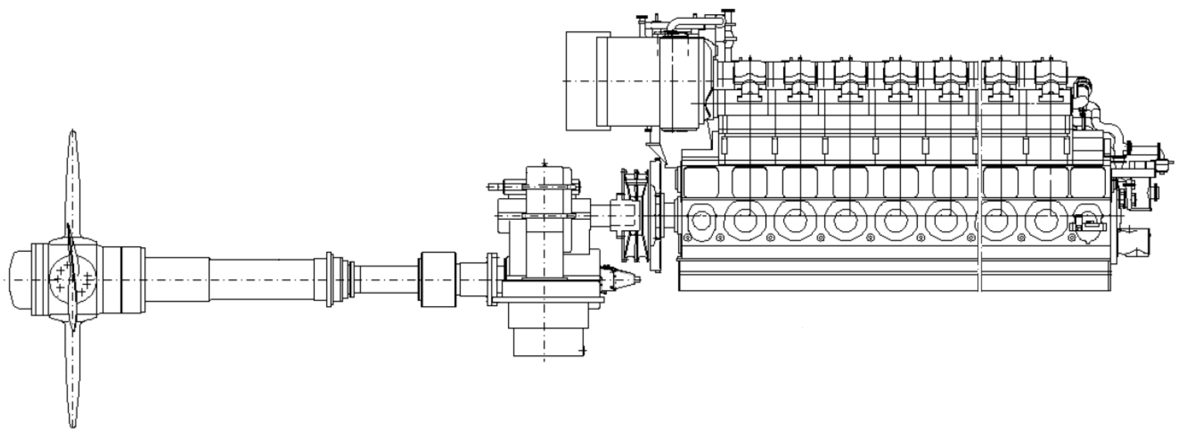


Fig. 1.11 Propulsion system of the vessel with a 16V 48/60 engine CR

The main advantages of an adjustable pitch propeller are its capabilities:

- Changing and adjusting the screw pitch when the shaft is rotating;
- work not only in terms of screw, but also in terms of external characteristics;
- achieving the lowest vessel speed at any operating speed by changing the propeller pitch accordingly;

Changing the direction of the vessel's movement with the help of a propeller, which simplifies the design of the diesel engine;

- Increasing efficiency and range voyage of ships at the optimal combination of diesel and propeller operating modes;

- starting the main diesel engine in the absence of propeller load;

- Ensuring full diesel power in forward and reverse;

- use of automatic control to maintain a constant n or to achieve optimal diesel operation.

starting the main diesel engine in the absence of propeller load;

- Ensuring full diesel power in forward and reverse;

- use of automatic control to maintain a constant n or to achieve optimal diesel operation.

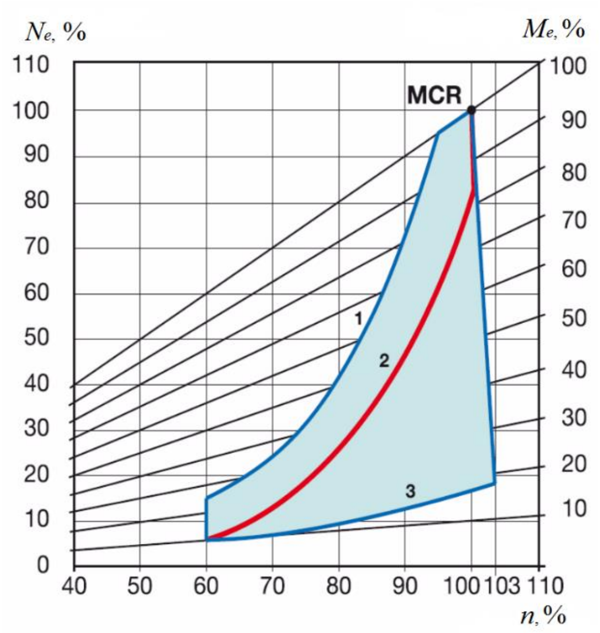


Fig. 1.12 Helical characteristic of the adjustable pitch of the M.A.N. 16V 48/60 CR engine 1 - ultimate load curve; 2 - curve of the recommended combined load; 3 – zero load curve traction;

MCR - maximum load limit; shaded area - operating range for the adjustable pitchscrew.

The combined load curve must be at a sufficient distance from curve 1 to protect against overloading. Load monitoring must be provided. Transmission losses (e.g. gearbox and shaft power) and additional power requirements must be taken into account.

1.4 Conclusion to the chapter

This section analyzes the design features of the bulk carrier, its power plant and technical equipment. The **Common Rail** fuel system is described, which ensures independence of fuel injection from the engine operating mode, which allows for increased efficiency and compliance with modern environmental requirements. The design, functional features of **the M.A.N. 16V 48/60 CR** engine and its systems: supercharging, lubrication and gas exhaust are considered in detail. This ensures optimum performance and durability of the unit.

SECTION 2
CALCULATION OF THE DUTY CYCLE FOR THE RATED OPERATING MODE

2.1 General characteristics of the mathematical model

A mathematical model of an internal combustion engine (ICE) is a simplified representation of the physical and chemical processes occurring in an engine in the form of mathematical equations. It is used to analyze, optimize, and predict the operation of an internal combustion engine and solve technical problems, such as increasing efficiency, reducing emissions, or improving dynamic performance.
The problem of mathematical modeling of the working processes of thermal engines arose in 1824, when the French scientist S. Carnot theoretically founded the working cycle of an ideal thermal machine.

The need to search for ways to improve the organization of the workflow of internal combustion engines leads to the need to analyze the indicators of the work cycle. The aim of the study is to create a generalized methodology for calculating the operating cycle indicators of high-speed diesel engines with different ways of organizing the workflow based on well-studied and reliable methods based on the use of dependencies containing general engine performance indicators. With the development of technology, in particular engine building, and the evolution of engineering thought, as a result of combining and supplementing the methodology for calculating the internal combustion engine workflow, the Grinevetsky-Mazing methodology was obtained as a single general system that is quite reliable in engineering calculations. In his thermal calculation, Professor V. I. Grinevetsky proposed to pre-set the temperature of the final gas T g and the cylinder filling factor ηn and proposed formulas for increasing the temperature of the beginning of combustion Ta and the final gas factor γg. E. K. Mazing, a follower of V. I. Grinevetsky, proposed to accept the estimated temperature of the final gases T g and the final gas coefficient γg, and to take the temperature of the beginning of combustion

Change	Sheet	No. doc..	Signature	Date	

Ta and the coefficient the filling ηn can be calculated by introducing the temperature T′ 0 (air heated by the cylinder walls) into the Grinevitch equation. Currently, there are a large number of different software and analytical complexes that differ in capacity (the number of functions they include), purpose (analysis, diagnostics, verification calculation, design, etc.), and the basic calculation methodology that is the basis of the entire logical complex.

The Hrynevetsky-Masing method is based on considering the dependence of the gas pressure *P* in the cylinder on the volume *V* during the engine's operating cycle. This dependence is analyzed in the form of **an indicator process diagram** (the so-called indicator diagram). Based on this diagram, it is possible to calculate the indicator power, efficiency, energy losses, and evaluate combustion and heat transfer processes.

2.2 Rationale for the choice of input data for the calculation of the duty cycle for the nominal operating mode

Environmental conditions in accordance with ISO 3046-1: 2002

- Ambient air pressure: 100 kPa;
- Ambient air temperature: 25° C. Select the main parameters for calculating the duty cycle:
- Rated power, crankshaft speed, boost pressure, specific effective engine fuel consumption according to the task;
- The compression ratio is given from the engine data sheet; [5, table 2-2].
- the number of cylinders, cylinder diameter and piston stroke are known;
- the values of heat utilization coefficients at points z and b are selected from the specified limits for SOD [2, p. 154];
- the value of the residual gas coefficient is selected from [2, Table 5.1, p. 139];
- the value of air heating from the cylinder walls is selected from [2, page 141];
- select the value of the residual gas temperature from [2, page 142];

- the value of the purge coefficient is selected from [2, p. 162]; - the lower calorific value of the fuel and its elemental composition are selected according to the characteristics of the fuel used;

- The rounding factor of the indicator diagram is chosen from the known limits of 0.92. 0.98 for 4-stroke engines;

- the value of mechanical efficiency is selected from the limits for SOD, and then varied to obtain the required calculation accuracy;

- Other parameters, such as compressor adiabatic efficiency, air cooler efficiency, pressure losses in the air cooler, mechanical efficiency, and air excess coefficient, are selected from the previously performed calculation for the discipline:

"Gas dynamics and supercharging units"

- Calculation of a centrifugal supercharger stage.

2.3 Calculation of the rated operating mode

Rated mode is the mode in which the engine develops the highest effective power specified by the manufacturer for a long time at a given speed and normal atmospheric conditions.

To determine all the necessary engine parameters and build all the necessary diagrams, we calculate the engine duty cycle for the rated mode (Grinevetsky-Masing method)

1). Initial data

1.1. Effective power, kW:	N'e = 19 200;
1.2. Rotational speed, min-1:	n = 500;
1.3. Ambient pressure, MPa:	P0 = 0,1;
1.4. Ambient temperature, K:	T0 = 298;
1.5. Air pressure behind the compressor, MPa:	Pk = 0,42;
1.6. Purging coefficient:	ϕa = 1,05;
1.7. Residual gas coefficient:	γr = 0,025;
1.8. The heat utilization factor at the z:	Z = 0,94;

1.9. The heat utilization factor at the b:	b = 0,97;
1.10. Compression ratio:	ε = 15,3;
1.11. Degree of pressure increase during combustion:	λ = 1,22;
1.12. Charge heating from the cylinder walls, K:	Ta = 20;
1.13. Fraction of the piston stroke lost to gas exchange:	n = 0;
1.14. Rounding factor of the indicator chart:	ζ = 0,98;
1.15. Mechanical efficiency of the engine:	ηm = 0,9284;
1.16. Adiabatic efficiency of the turbocharger compressor:	ηk.ad=0,8;
1.17. Pressure loss in the air cooler, MPa:	Pox = 0,004;
1.18. Air cooling system efficiency:	ηo = 0,8;
1.19. Residual gas temperature, K:	Tr = 800;
1.20. Fuel mass composition (kg/kg):	C = 0,87;
	H = 0,126;
	S = 0;
	O = 0,004;
1.21. Lowest calorific value of fuel, kJ/kg:	QH = 42700;
1.23. Number of cylinders:	i = 16;
1.24. Cylinder diameter, m:	DC = 0,48;
1.25. Piston stroke, m:	Sc = 0,6;
1.26. The tact coefficient:	z = 05;
1.27. Combustion air excess coefficient:	α = 2,09;
1.28. Air adiabatic index:	kв = 1,4;
1.29. Mechanical efficiency of the turbocharger:	ηt.m = 0,97;
1.30. Internal efficiency of the turbocharger turbine:	ηt.ad=0,82;
1.31. Inlet pressure loss coefficient:	a = 0,97;
1.32 Upstream pressure to downstream pressure ratio for the compressor:	t = 0,7619;

2). Calculation of the filling process

2.1. Air temperature behind the compressor, K:

$$T_k := T_0 \cdot \left[1 + \frac{\left[\left(\frac{P_k}{P_0}\right)^{0.286} - 1\right]}{\eta_{k.ad}}\right]$$ $T_k = 487{,}033;$

2.2. Air temperature in the receiver, K:

$$T_s := T_k - \eta_o \cdot (T_k - T_0)$$ $T_s = 335{,}807;$

2.3. Air temperature at the end of filling, K:

$$T_a := \frac{T_s + \Delta T_a + \gamma_r \cdot T_r}{1 + \gamma_r}$$ $T_a = 366{,}641;$

2.4. Air pressure in the engine receiver, МПа:

$$P_s := P_k - \Delta P_{ox.}$$ $P_s = 0{,}416;$

2.5. Pressure at the end of filling, МПа:

$$P_a := \xi_a \cdot P_s$$ $P_a = 0{,}404;$

2.6. Filling rate:

$$\eta_i := \frac{\varepsilon}{\varepsilon - 1} \cdot \frac{P_a}{P_s} \cdot \frac{T_s}{T_a} \cdot \frac{1}{1 + \gamma_r} \cdot (1 - \phi_n)$$ $\eta_н = 0{,}927;$

3). Calculation of the compression process

3.1. Regression equation for calculating the average isochoric molar specific heat of air, kJ/(kmol-K):

$$c'_\upsilon \leftarrow 19.26 + 0.0025\,T$$

3.2. Regression equation for calculating the average isobaric molar specific heat of the exhaust gases, kJ/(kmol-K):

$$c''_\upsilon \leftarrow 20.47 + 0.0036\,T$$

3.3. Determination of the coefficients of the regression equation for the heat capacity of a mixture of gases in a cylinder at the end of the compression process:

$$c''_{\upsilon c} \leftarrow \frac{\gamma_r \cdot c''_\upsilon + \left[\alpha \cdot (1 + \gamma_r) - \gamma_r\right] \cdot c'_\upsilon}{\alpha \cdot (1 + \gamma_r)} \leftarrow a_{\upsilon c} - b_c \cdot T$$ $b_c = 0{,}002513;$

$a_{\upsilon c} = 19{,}274;$

3.4. Compression polytropic index:

$$n_1 - 1 \leftarrow \frac{8.13}{a_{\upsilon c} + b_c \cdot T_a \cdot \left(1 + \varepsilon^{n_1 - 1}\right)}$$ $n_1 = 1{,}359;$

3.5. Pressure at the end of compression, MPa:

$$P_c := P_a \cdot \varepsilon^{n_1}$$ $P_c = 16{,}437;$

3.6. Temperature at the end of compression, K:

$$T_c := T_a \cdot \varepsilon^{n_1 - 1}$$ $T_c = 976{,}128;$

4). Calculation of the combustion process

4.1. Amount of air entering the cylinder, kmol/kg:

$$L := \frac{\alpha}{0.21} \cdot \left(\frac{C}{12} + \frac{H}{4} + \frac{S}{32} - \frac{O}{32} \right)$$ $L = 1{,}034;$

4.2. The theoretical coefficient of molecular change:

$$\beta_0 := 1 + \frac{8 \cdot H + O}{32 \cdot L}$$ $\beta_0 = 1{,}0306;$

4.3. The true coefficient of molecular change:

$$\beta := \frac{\beta_0 + \gamma_r}{1 + \gamma_r}$$ $\beta = 1{,}0298;$

4.4. The proportion of fuel that has burned up to point *z*:

$$x_z := \frac{\xi_z}{\xi_b}$$ $x_z = 0{,}969;$

4.5. Molecular change coefficient at point *z*:

$$\beta_z := 1 + \left(\frac{\beta_0 - 1}{1 + \gamma_r} \right) \cdot x_z$$ $\beta_z = 1{,}029;$

4.6. Specific isochoric average heat capacity at point *z*, kJ/(kmol-K):

$$c''_{\upsilon z} \leftarrow \frac{(m \cdot x_z + \gamma_r) \cdot c''_\upsilon + \left[\alpha \cdot (1 + \gamma_r) - (x_z + \gamma_r) \right] \cdot c'_\upsilon}{\alpha \cdot (1 + \gamma_r) + (m - 1) \cdot x_z} \leftarrow a_{\upsilon z} + b_z \cdot T$$

$$\Delta M := \left(\frac{H}{4} + \frac{O}{32} \right)$$ $\Delta M = 0{,}0316;$

$$L_0 := 0.495 \ \frac{кмоль}{кг}$$

$$m := 1 + \frac{\Delta M}{L_0}$$ $m = 1{,}064;$

$a_{\upsilon z} = 19{,}83969;$

Change	Sheet	No. doc..	Signature	Date	QWM.142.6221м.06.02.EN	Sheet

$b_z = 0{,}00303;$

4.7. Determination of the coefficients of the regression equation for the heat capacity of the working fluid at point b, kJ/(kmol-K):

$$c_{\upsilon b} \leftarrow \frac{(m+\gamma_r)\cdot c''_{\upsilon} + (\alpha-1)\cdot(1+\gamma_r)\cdot c'_{\upsilon}}{\alpha\cdot(1+\gamma_r)+(m-1)} \leftarrow a_{\upsilon b} + b_b\cdot T$$

$a_{\upsilon b} = 19{,}85722;$

$b_b = 0{,}003043;$

4.8. Determination of the temperature at point z, K:

$$\frac{\xi_z\cdot Q_i}{\alpha\cdot L_0} + \left[c'_{\upsilon} + 8.314\lambda + \gamma_r\cdot\left(c''_{\upsilon} + 8.314\lambda\right)\right]\cdot T_c \leftarrow \beta_z\cdot(1+\gamma_r)\cdot c''_{pz}\cdot T_z$$

$T_z = 1966;$

4.9. Maximum pressure in the cylinder, MPa:

$$P_z := \lambda\cdot P_c$$

$P_z = 20{,}053;$

5). Calculation of the expansion process

5.1. The degree of preliminary expansion:

$$\rho := \frac{\beta_z}{\lambda}\cdot\frac{T_z}{T_c}$$

$\rho = 1{,}699;$

5.2. The degree of further expansion:

$$\delta := \frac{\varepsilon}{\rho}$$

$\delta = 9{,}007;$

5.3. Determination of the expansion polytropy and temperature of the working fluid at point b, K:

$$n_2 \leftarrow \left[\frac{8.314\left(\frac{\beta_z}{\beta}\cdot T_z - T_b\right)}{\frac{Q_H\cdot(\xi_b-\xi_z)}{L\cdot(1+\gamma_r)\cdot\beta} + \frac{\beta_z}{\beta}\cdot(a_{\upsilon z}+b_z\cdot T_z)\cdot T_z - (a_{\upsilon b}+b_b\cdot T_b)\cdot T_b}\right] + 1$$

$$T_b \leftarrow T_z\cdot\frac{1}{\delta^{n_2-1}}$$

$n_2 = 1{,}274;$

$T_b = 1076;$

5.4. Pressure at the end of the expansion process, MPa:

$$P_b := \frac{P_z}{\delta^{n_2}}$$

$P_b = 1{,}219;$

6). Determination of cycle indicators

6.1. Theoretical average indicator pressure, MPa:

$$P'_i := \frac{P_c}{\varepsilon-1}\cdot\left[\lambda\cdot(\rho-1) + \frac{\lambda\cdot\rho}{n_2-1}\cdot\left(1-\frac{1}{\delta^{n_2-1}}\right) - \frac{1}{n_1-1}\cdot\left(1-\frac{1}{\varepsilon^{n_1-1}}\right)\right]$$

$P'_i = 2{,}913;$

6.2. True average indicator pressure, MPa:

$$P_i := P'_i \cdot \zeta \cdot (1 - \phi_n)$$ $P_i = 2{,}855;$

6.3. Specific indicator fuel consumption, kg/(kWh):

$$g_i := 433 \cdot \frac{P_s \cdot \eta_i}{\alpha \cdot L_0 \cdot T_s \cdot P_i}$$ $g_i = 0{,}168;$

6.4. Indicator efficiency:

$$\eta_i := \frac{3600}{g_i \cdot Q_H}$$ $\eta_i = 0{,}501;$

7). Determination of effective indicators

7.1. Average effective pressure, MPa:

$$P_e := P_i \cdot \eta_m$$ $P_e = 2{,}651;$

7.2. Effective efficiency:

$$\eta_e := \eta_i \cdot \eta_m$$ $\eta_e = 0{,}465;$

7.3. Specific effective fuel consumption, kg/(kWh):

$$g_e := \frac{g_i}{\eta_m}$$ $g_e = 0{,}1814;$

7.4. Effective motor power, kW:

$$N_e := 13.1 \cdot D_c^2 \cdot S_c \cdot z \cdot P_e \cdot n \cdot i$$ $N_e = 1{,}92 \cdot 10^4;$

7.5. Effective power determination error, %:

$$\Delta N := \frac{N_e - N'_e}{N_e}$$ $\Delta N = 5{,}004 \cdot 10^{-4};$

8). Calculating the power balance of the turbine and compressor

8.1. Determination of air flow rate by the engine, kg/s:

R = 287 J/(kg-K) – is the gas constant of air;

$$G := \left[\frac{\left(\pi \cdot D_c^2\right)}{4}\right] \cdot S_c \cdot \left(\frac{P_s \cdot 10^6}{R \cdot T_s}\right) \cdot \eta_i \cdot i \cdot z \cdot \left(\frac{n}{60}\right) \cdot \phi_a$$ $G = 30{,}423;$

8.2. Gas flow rate through the turbine, kg/s;

$$G_t := G + g_e \cdot \frac{N_e}{3600}$$ $G_t = 31{,}39;$

8.3. Gas pressure in front of the turbine, MPa:

$$P_r := \psi_t \cdot P_k$$ $P_r = 0{,}32;$

8.4. Temperature of gases before the turbine, К:

$$T_t := T_b \cdot \left(\frac{P_r}{P_b}\right)^{\frac{n_2 - 1}{n_2}}$$ $T_t = 807{,}264;$

8.5. Amount of working substance, kmol/kg:

$$M_s := \left(\frac{C}{12}\right) + \left(\frac{H}{2}\right) + \left[0.21 \cdot (\alpha - 1) \cdot L_0\right] + \left(0.79 \cdot \alpha \cdot L_0\right)$$ $M_s = 1{,}066;$

8.6. Gas constant for exhaust gases, J/(kg-K):

$$R_t := \frac{1}{\frac{C}{12 \cdot M_s \cdot 189} + \frac{H}{2 \cdot M_s \cdot 461.6} + \frac{0.21 \cdot (\alpha - 1) \cdot L_0}{M_s \cdot 259.8} + \frac{0.79 \cdot \alpha \cdot L_0}{M_s \cdot 296.8}}$$ $R_t = 287{,}367;$

8.7. Adiabatic index for exhaust gases:

$$k_t := \frac{\left(a_{\upsilon b} + b_b \cdot T_t\right) + 8.314}{\left(a_{\upsilon b} + b_b \cdot T_t\right)}$$ $k_t = 1{,}373;$

8.9. The degree of increase in air pressure in the compressor:

$$\Pi к := \left[1 + \frac{\left(\frac{k_t}{k_t - 1}\right) \cdot R_t \cdot T_t \cdot \left[1 - \frac{1}{\left(\frac{P_r}{P_0}\right)^{\frac{k_t - 1}{k_t}}}\right] \cdot \eta_{t.m} \cdot \eta_{t.ad} \cdot G_t}{G \cdot R \cdot T_0 \cdot \left(\frac{k_в}{k_в - 1}\right) \cdot \frac{1}{\eta_{k.ad}}}\right]^{\frac{1}{0.286}}$$

$\Pi_к = 4{,}2;$

8.10. Design boost pressure, MPa:

$$P_{kd} := \Pi к \cdot P_0$$ $P_{kd} = 0{,}42;$

8.11. Error in determining boost pressure, %:

$$\Delta P_k := \frac{P_{kd} - P_k}{P_{kd}}$$ $\Delta P_k = 5{,}421 \cdot 10^{-3}.$

2.4 Building an indicator chart

We build a theoretical indicator diagram based on the data from the work cycle calculation. We build the diagram using the analytical method, since graphical methods of construction produce large errors.

Indicator diagram in the coordinate system $p - V/V_c$

The ordinates of the points of the compression and expansion polytropes are calculated using the following formulas:

– for the compression process:

$$p = \frac{p_c}{\left(\frac{V}{V_c}\right)^{n_1}}$$

– for the compression process:

$$p = \frac{p_z \cdot \rho^{n_2}}{\left(\frac{V}{V_c}\right)^{n_2}}$$

where $V/V_c = \varepsilon_x$ is the ratio of volumes, which is the current value of the compression ratio; p_c is the pressure at the end of compression; p_z is the maximum pressure in the cylinder; n_1 is the compression polytropy index; n_2 is the expansion polytropy index; ρ is the degree of preliminary expansion.

Results of work in Microsoft Excel

Table 2.1 - Input data

P_r, MPa	P_a, MPa	P_s, MPa	P_k, MPa	P_b, MPa	P_c, MPa	P_z, MPa	n_1	n_2	ρ
0,32	0,404	0,416	0,42	1,219	16,437	20,053	1,359	1,274	1,699

Table 2.2 - Calculation results

V/V_c	$P_{сж}$, MPa	$P_{расш}$, MPa
1	16,437	20,053
1,699	7,99815384	20,053
2	6,40800425	16,2904659
3	3,69330156	9,71836576
4	2,49817597	6,73629797
5	1,84468544	5,06941602
6	1,43984256	4,01865779
7	1,16770817	3,30210452
8	0,97391995	2,78553792
9	0,82986411	2,39740144
10	0,71915508	2,09626276
11	0,63178585	1,85657047
12	0,56132611	1,66176195
13	0,50346993	1,50065855
14	0,45523386	1,3654588
15	0,41449046	1,25056265
15,3	0,40348454	1,21940743

After calculating of the required values we build theoretical indicator diagram of the work cycle and present it in the coordinate $p - V/V_c$ dimensionless in the direction of the volume axis. The theoretical indicator diagram in the coordinate system p - V/V_c is shown in (Fig. 12).

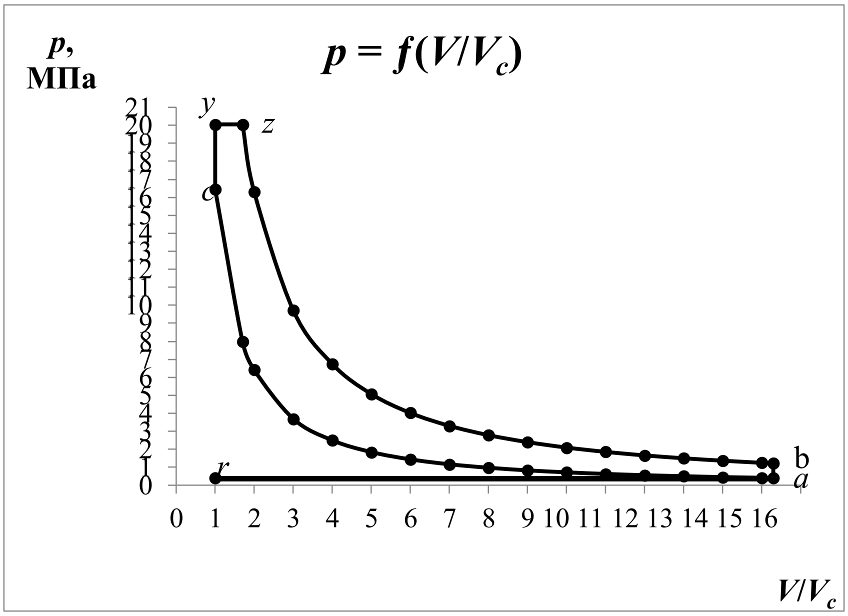


Fig. 2.1 Theoretical indicator diagram of the duty cycle in the coordinate system p – V/Vc

Indicator chart in the coordinate system $T - V/V_c$

The ordinates of the points of the compression temperature and expansion temperature polytropes are calculated using the following formulas:

– for the compression process:

$$T = \frac{T_c}{\left(\frac{V}{V_c}\right)^{n_1 - 1}}$$

– for the expansion process:

$$T = \frac{T_z}{\delta^{n_2 - 1}}$$

where V/Vc = εx is the ratio of volumes, which is the current value of the compression ratio;

T_c is the temperature at the end of compression; T_z is the maximum temperature in the cylinder;

n_1 - compression polytropic index; n_2 - expansion polytropic index;

δ is the degree of further expansion, which is determined from the following expression:

$$\delta = \frac{V}{V_c} \cdot \frac{1}{\rho}$$

where ρ is the degree of preliminary expansion.

The temperature at point y is determined by the formula:

$$T_y = T_c \cdot \lambda$$

where λ is the degree of pressure increase during combustion.

Results of work in the Microsoft Excel environment

Table 2.3 - Input data

T_r, K	T_a, K	T_s, K	T_k, K	T_b, K	T_c, K	T_z, K	λ
807,264	366,641	335,807	487,033	1076	976,128	1966	1,22

Table 2.4 - Calculation results

V/V_c	$T_{сж}$, K	$T_{расш}$, K
1	976,128	1190,876
1,699	806,9885	1966
2	761,0917	1880,071
3	657,9914	1682,385
4	593,4269	1554,863
5	547,7426	1462,644
6	513,0391	1391,371
7	485,4188	1333,827
8	462,6978	1285,908
9	443,5409	1245,071
10	427,0776	1209,641
11	412,7117	1178,46
12	400,0191	1150,697
13	388,688	1125,735
14	378,4834	1103,106
15	369,2241	1082,449
15,3	366,6085	1076,592

After calculating the required values, we build a theoretical indicator diagram of the working cycle and represent it in the *T* - *V/V* coordinate systemc dimensionless in the direction of the volume axis. The theoretical indicator diagram of *the T* - *V/V* coordinate systemc is shown in (Fig. 13).

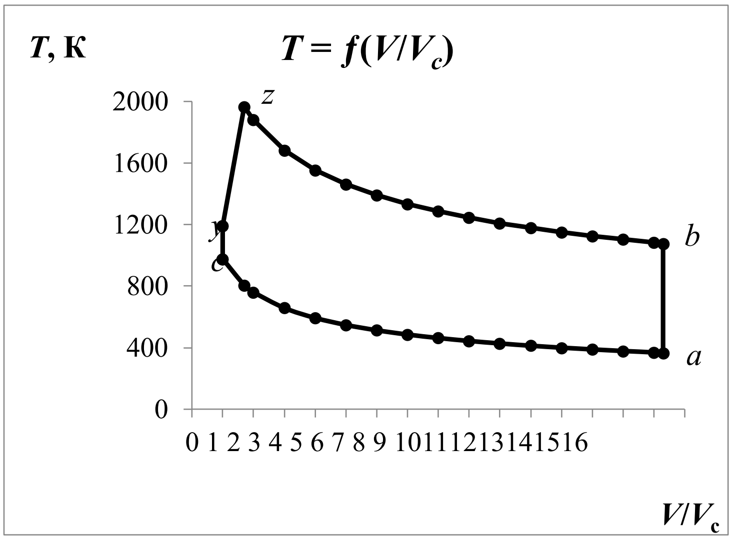


Fig. 2.2 Theoretical indicator diagram of the duty cycle in the coordinate system $T – V/V_c$

Indicator chart in the coordinate system T - S

To determine the entropy, we use the formula:

$$S = \left[c_p \cdot \ln\left(\frac{T}{T_0} \right) - R \cdot \ln\left(\frac{p}{p_0} \right) \right]$$

where c_p is the isobaric heat capacity of the gas mixture; R is the gas constant; T_0 is the ambient temperature; P_0 is the ambient pressure;

$$c_p = f(T)$$

$$c_p = c_v + R$$

where c_v is the isochoric heat capacity of the gas mixture; R = 8.314 kJ/(kg-K).

The current isochoric heat capacity is determined by the formula:

$$c_{vi} = a_{vi} + b_i \cdot T_i$$

where a_{vi}, b_i are the coefficients of the regression equation for determining the heat capacity of the gas mixture; T_i is the current temperature.

– for the compression process:

$$S = \left[c_{p_i} \cdot \ln\left(\frac{T_{сж}}{T_0} \right) - R \cdot \ln\left(\frac{p_{сж}}{p_0} \right) \right]$$

– for the expansion process:

$$S = \left[c_{p_i} \cdot \ln\left(\frac{T_{расш}}{T_0} \right) - R \cdot \ln\left(\frac{p_{расш}}{p_0} \right) \right]$$

Results of work in Microsoft Excel

Table 2.5 - Input data

a_{vb}, kJ (kg-K)	b_b, kJ (kg-K)	a_{vc}, kJ (kg-K)	b_c, kJ (kg-K)	a_{vz}, kJ (kg-K)	b_z, kJ (kg-K)	a_{vy}, kJ (kg-K)	b_y, kJ (kg-K)	R, kJ (kg-K)
19,85722	0,003043	19,274	0,002513	19,83969	0,00303	19,36655	0,0026	8,3144

Table 2.6 - Calculation results

V/V_c	$P_{сж}$, MPa	$P_{расш}$, MPa	$T_{сж}$, K	$T_{расш}$, K	$S_{сж}$, kJ/(kg-K)	$S_{расш}$, kJ/(kg-K)
1	16,437	20,053	976,128	1190,876	-6,2686	-0,91629
1,699	7,998154	20,053	806,9885	1966	-6,42658	20,85894
2	6,408004	16,290466	761,0917	1880,071	-6,42788	20,5782
3	3,693302	9,7183658	657,9914	1682,385	-6,35026	20,06547
4	2,498176	6,736298	593,4269	1554,863	-6,23422	19,84682
5	1,844685	5,069416	547,7426	1462,644	-6,11421	19,75179
6	1,439843	4,0186578	513,0391	1391,371	-5,99882	19,71874
7	1,167708	3,3021045	485,4188	1333,827	-5,89016	19,7201
8	0,97392	2,7855379	462,6978	1285,908	-5,78844	19,74183
9	0,829864	2,3974014	443,5409	1245,071	-5,69326	19,77608
10	0,719155	2,0962628	427,0776	1209,641	-5,60403	19,81822
11	0,631786	1,8565705	412,7117	1178,46	-5,52016	19,86534
12	0,561326	1,661762	400,0191	1150,697	-5,4411	19,91558
13	0,50347	1,5006586	388,688	1125,735	-5,36637	19,9677
14	0,455234	1,3654588	378,4834	1103,106	-5,29554	20,02085
15	0,41449	1,2505627	369,2241	1082,449	-5,22823	20,07445
15,3	0,403485	1,2194074	366,6085	1076,592	-5,20868	20,09056

After calculating the required values, we build a theoretical indicator diagram of the work cycle and represent it in the T - S coordinate system. The theoretical indicator diagram in the T - S coordinate system is shown in (Fig. 14).

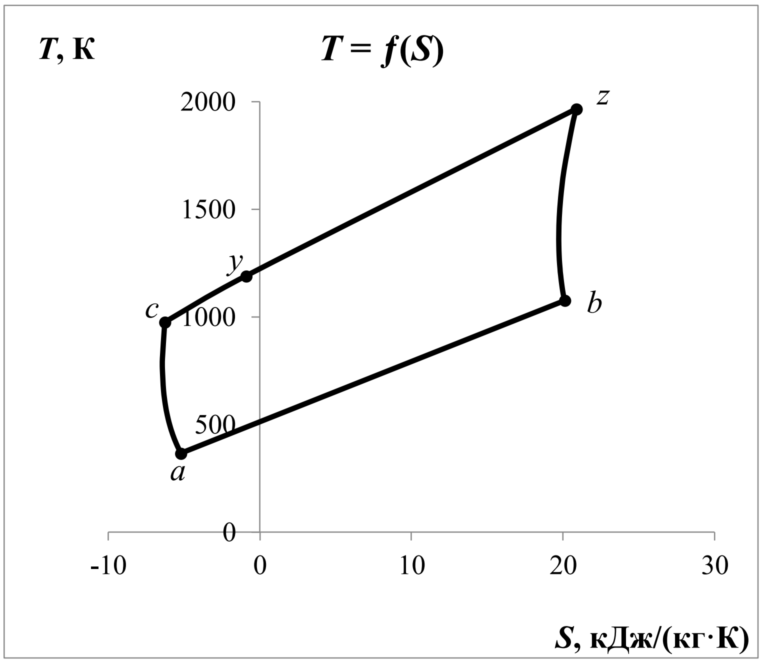


Fig. 2.3 Theoretical indicator diagram of the working cycle in the *T* - *S* coordinate system

2.5 Conclusion to the chapter

In this section, the mathematical analysis and modeling of the engine's operating cycle was carried out using the Grinevetsky-Masing methodology. Key parameters such as average indicator pressure, effective efficiency, and an indicator diagram for the nominal operating mode were determined. This allows optimizing the engine operation, ensuring its efficiency and cost-effectiveness in accordance with the specified conditions.

SECTION 3
INCREASE ENGINE EFFICIENCY AT RATED OPERATION

3.1 Analysis of the use of supercharging in marine diesel engines

Supercharging is a method of increasing engine power by supplying air with a

$$\lambda_н = P_{ен} / P_e,$$

P_e и $P_{ен}$ pressure higher than atmospheric pressure to the working cylinder by means of a cyclic oil supply (cyclic oil supply is the supply of oil in one cycle). The additional fuel supply is a source of additional heat to the working mechanism inside the cylinder, which increases the specific effective work of the cycle. A supercharged gas turbine engine is a combined device consisting of an engine, a gas turbine and a supercharger. Operating in a variable mode, each unit has its own characteristics, which are the links between different indicators of the work process. The forced supercharging of a diesel engine is evaluated by the "supercharging degree" λn:

- Average effective engine pressure without and with supercharging.

In principle, the amount of air in the cylinder can be increased not only by pre-compressing it, but also by lowering the temperature (the specific gravity of air is proportional to Pk and inversely proportional to Tk: γk /γo = Pk To/Po Tk), as well as by increasing the cylinder filling factor etan through better cylinder cleaning. These factors are used to seal the complex. The pre-compressed air is cooled to 30 - 45°C and then fed into the cylinder.

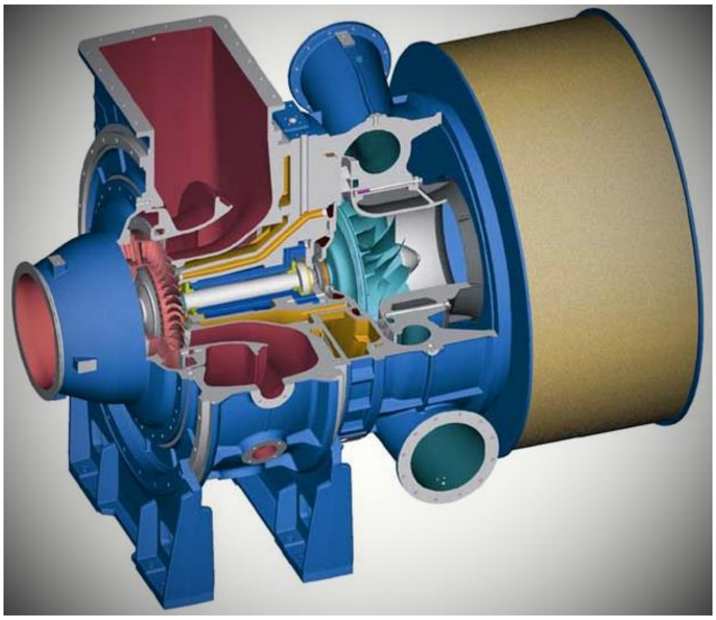


Fig. 3.1 Gas turbocharger of a marine diesel engine

Compared to engines without supercharging, the use of supercharging increases the working volume of diesel engines by 4-5 times, but requires solving many serious technical problems associated with increased mechanical and thermal loads and deterioration of lubrication. Under these conditions, the wear of the cylinder-piston group increases and the characteristics of the supercharger and diesel engine are not harmonized, etc. These problems are constantly faced by diesel engine builders when further boosting engines. Methods of supercharging are distinguished by as follows as follows:

-Inertia;

-mechanical;

-Gas turbines and their combinations.

Attempts to use inertial supercharging took place in the early days of the four-stroke internal combustion engine. In this case, each cylinder is equipped with a specially selected long intake pipe and a corresponding resonant mechanism in it, which increases the air pressure in the cylinder at the final stage of intake. Inertial boosting increases power output by 15-25%. In case of mechanical supercharging air compressor is driven by the engine crankshaft. The supercharger uses a reciprocating, rotary or centrifugal compressor driven directly by the crankshaft or through a transmission (gear, chain, electric).

Gas turbine and combined supercharging methods are the most common in internal combustion engines. With gas turbine supercharging, the energy of the exhaust gases is used to drive the supercharger. The gas turbine and the centrifugal compressor sitting on the same shaft with it are a single unit - a turbocharger for supercharging (TC). The gases from the working cylinders, giving part of their energy to the gas turbine, go on to the utilization boiler and the atmosphere. The air sucked in from the atmosphere is compressed in the compressor to a pressure of Pk, fed to the air cooler and then to the purge receiver and the working cylinders. At low compression in the compressor, when the temperature does not rise above 45-50°C, the refrigerator may not be needed. Combined supercharging refers to a system that uses both gas turbine and mechanical supercharging. It is used in cases where the power of gas turbines is insufficient to drive the supercharger. A special case of a mechanical supercharger is the use of the piston cavities of crosshead engines in conjunction with a gas turbine supercharger.

An assessment of the degree of perfection of a particular supercharging system can be made on the basis of a qualitative mechanical analysis of the engine's efficiency $\eta_{мех}$. For a naturally aspirated engine, we can write:

$$\eta_{мех} = 1 - N_{м}/N_{i}.$$

With inertial supercharging, all other things being equal, the engine mechanical loss power Nm will not change, and the indicator power Ni will increase without any additional energy costs for the air supercharger. Consequently, the mechanical efficiency of the engine will increase. Nevertheless, inertial supercharging has not found application in marine diesel engines due to the bulky intake system and relatively low level of boost.

In a mechanically supercharged engine, the power of mechanical losses increases by the amount N of the cost of driving the air supercharger; the mechanical efficiency is equal:

$$\eta^{мн}{}_{мех} = 1 - \left(\left(N_{м} + N_{в}\right)/\left(N_{i} + \Delta N_{i}\right)\right),$$

Where:

- $N_i + \Delta N_i = N_{iн}$ is the indicator power of the supercharged engine.

Obviously, any increase in diesel power requires an increase in boost pressure Pk. At the same time, the power N to drive the air supercharger also increases. If the indicator power increases more intensively than the mechanical loss power, the mechanical efficiency increases. In this case, as Pk increases, the average effective pressure $p_{ен}$ also increases:

$$p_{ен} = p_{iн}\, \eta^{мн}{}_{мех}.$$

When a certain level of boost is reached, the cost of driving the mechanical air supercharger begins to increase more intensively than the increase in indicator power; the mechanical efficiency of the diesel engine decreases. Despite the increase in Pk, the average effective pressure may even decrease (if the degree of decrease in ηm exceeds the degree of increase in pi). In the extreme case of mechanical supercharging, it is possible to create an engine in which all indicator work is absorbed by the compressor, with mechanical and effective efficiency equal to zero.

According to experimental data, the limit of a reasonable increase in λn under purely mechanical supercharging is within the limits:

$$\lambda_н = 1,2 \div 1,3.$$

At the same time:

$$P_к = 1,3 \div 1,5\, ama,\ \eta^{мн}{}_м = 0,70 \div 0,85.$$

When engines are further boosted, too much power is required to drive the supercharger, which reduces ηm and ηe. Therefore, purely mechanical supercharging is not used in modern engines. It can be found in engines of the old design (ZD-100, 37D, DR 30/50, DR 43/61, etc.).

If we imagine a case where the power to drive the mechanical supercharger $N_в$ is partially covered by a gas turbine (in engines with a combined supercharging system), the mechanical efficiency will be expressed by the following relationship:

$$\eta_{м}^{мн+гт} = 1 - \left(N_{м} + \left(N_{в} - N_{гт}\right)\right)/N_{iн},$$

Where:

$-N_{гт}$ – gas turbine power.

At the same level of engine boost (the same indicator power), mechanical losses with the combined supercharging system will be less than with pure mechanical supercharging by the amount $N_{гт}$. Obviously, the combined system will have a higher ηm. Let's imagine an early case - a turbocharger has completely replaced the mechanical blower (i.e., pure gas turbine supercharging is provided). The mechanical efficiency will increase even more (since $N_{в} - N_{гт} = 0$) and will be determined by the relationship:

$$\eta_{мех}^{гт} = 1 - N_{мех}/N_{iн}.$$

Thus, at approximately the same boost levels, you can record:

$$\eta_{м}^{гт} > \eta_{м}^{мн+гт} > \eta_{м}^{мн}.$$

Such efficiency ratios are found in actual designs of both 4-stroke and 2-stroke diesel engines. The maximum values of mechanical efficiency ηm with gas turbine supercharging reach the values $\eta_{м}^{гт} \approx 0,94–0,96$ (compared to 0.70-0.85 for engines without supercharging). It can be said that the use of gas energy in a gas turbine is the utilization of the heat of the exhaust gases in the diesel cylinder. In order to utilize this energy directly in the cylinder, the piston stroke must be significantly increased. The increase in indicator work would be б reduced to nothing by additional mechanical losses. This is why heat utilization to generate additional work was delayed almost until a gas turbine was developed that could use a working fluid with a large specific volume at low pressure transfers with sufficient efficiency.

Utilization of of heat of exhaust gases y gas turbine blowers was was once possible only in 4-stroke internal combustion engines, where the gas temperature reached 400-450 °C (compared to 260-290 °C in 2-stroke internal combustion engines). At the same time, without any reconstruction of the gas exchange system at the initial stage of using supercharging, it was possible to obtain a boost level of $\lambda_{н} = 1.2\text{-}1.3$ at $P_{к} = 1.25\text{-}1.30$ bar. Modern engines have achieved $\lambda_{н}$ = 2.5-4.0 (2-stroke diesel engines) and $\lambda_н$ = 3.0-5.0 (4-stroke diesel engines). Even higher values have been achieved in prototype designs. Engine boosting is no longer achieved by utilizing the heat of exhaust gases, but by redistributing energy between the cylinder and the gas turbine by opening the exhaust bodies earlier. This way leads to a decrease in the efficiency of the indicator process of the diesel engine itself, which reduces indicator work due to earlier opening of the exhaust bodies. The decrease in thermal and indicator efficiency of the cycle is taken into account by a corresponding decrease in the degree of further expansion of gases in the cylinder = Vb/Vz when the point b on the indicator diagram is shifted to the left. At the same time, further boosting of diesel engines with supercharging leads to an increase in the cycle fuel supply and the removal of combustion to the expansion line, which also reduces the degree of pre-expansion by increasing the degree of pre-expansion; this factor also acts to reduce the indicator efficiency.

The value of the maximum cylinder pressure Pz at ε = const increases less intensively than the charge air pressure Pk and the average indicator pressure pi. Nevertheless, in modern internal combustion engines, the maximum cylinder pressure has reached a value of Pz = 120-160 bar. The desire to reduce or keep Pz at the same level and increase the reliability of the crankshaft bearing requires a reduction in the compression ratio ε. This solution is widely used in the practice of diesel engine construction. However, it leads to a further decrease in thermal and indicator efficiency.

Despite the factors that reduce the indicative efficiency of the diesel engine itself, the effective efficiency and specific effective fuel consumption of high-pressure gas turbine engines are changing for the better:

- Increase in mechanical efficiency;
- Reducing the proportion of heat transferred to cooling water;
- more rational use of gas heat in the turbocharger (increased turbine and compressor efficiency), which reduces the proportion of lost piston stroke to ensure the balance of energy of the gas turbine and compressor for boosting.

3.2 Physical and numerical experiments as a means of studying processes in internal combustion engines

The technique and technology of experimental research of internal combustion engines has been continuously improved over the course of more than 150 years of their development.

A physical experiment can be characterized by the high reliability of the experimental data obtained, since all the real gas-dynamic, hydraulic, and tribological features of the processes under study are taken into account. However, conducting a physical experiment is greatly complicated by significant difficulties in obtaining experimental data, which requires the use of complex and expensive measuring equipment. Thus, despite the significant progress of experimental measurement tools, some parameters of individual physical processes, such as the spatial distribution of temperatures and concentrations of working fluid components during the engine operating cycle, cannot be obtained by direct measurements. Significant disadvantages of a physical experiment are its high cost and significant time spent on the experiment, since it requires the manufacture of the object of experimental research.

With the development of electronic computers since the 60s of the XX century, numerical experimentation has been more widely used in the development and validation of new engine designs. In a numerical experiment, a real engine or physical model is replaced by a mathematical model that describes the processes under study, usually using a system of nonlinear or linear differential equations. For example, numerical experiments are intensively used in the design of individual engine components, taking into account their thermal and mechanical loads; in the study of working processes in an engine cylinder using thermodynamic and gas-dynamic mathematical models; in the study of unsteady wave motion of gas in the gas-air path of an engine; in the study of hydraulic processes in fuel equipment; in the study of heat exchange processes, etc.

The advantages of the numerical experiment include the following:

1) **Full information about the system**

The FFE takes into account all possible combinations of factors and their levels, which makes it possible:

- Investigate the main effects of each factor - to assess how much a change in a particular factor affects the result.
- Identify interactions between factors - some factors may interact, meaning that their combined impact may be significantly different from the sum of their individual impacts.
- Get a complete picture of the process - allows you to deeply understand the behavior of the system under different conditions.

2) **High accuracy and reliability**

The FFE provides sufficient data for statistically sound analysis.

- It helps minimize the likelihood of errors associated with the omission of important factors or their combinations.
- Provides accuracy assessment effects even in the the presence of noise or variability in the experimental data.

3) **Application for a wide range of tasks**

FFE can be used in any area where it is important to investigate the influence of several factors, for example:

- Industrial production (optimization of technological processes);
- Agriculture economy (determination of optimalconditions for growing crops);
- Ecology (assessment of the impact of pollution on various ecosystems);
- Marketing (studying the impact ofpricing policy,advertising, and product design).

3.3 Features of the application of a full factorial experiment for analyzing the parameters of fuel equipment on engine performance

Before conducting research on an internal combustion engine (ICE), regardless of whether physical or numerical methods are used, a planning stage is always carried out. This is an important process that determines the number of experiments and the conditions under which they will be performed to achieve accurate results.

The main goals of planning the experiment:

- Reducing the number of experiments required.
- Parallel variation of all key parameters according to special algorithms.
- Use of mathematical methods to formalize the actions of the researcher.
- Determine a strategy that allows you to make informed decisions after each stage of experimentation.

Experiments can solve two types of problems:

1) **Optimization tasks** – searching for the best conditions for the facility's operation.
2) **Interpolation problems -** analysis influence of individual parameters on the characteristics of the object.

Experiments aimed at optimization are also called extreme experiments because, in mathematical terms, they involve determining the maximum or minimum value of a certain function. In real-world settings, research usually combines both types of tasks.

For the convenience of planning experiments, the object of study is often modeled as a "black box" (see Figure 3.6).

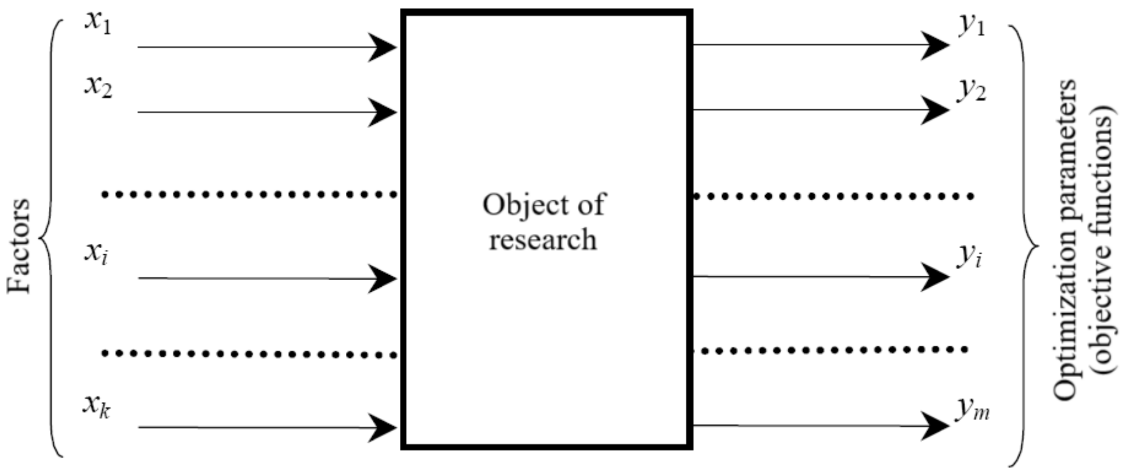


Fig. 3.2 - Schemes of representation of the research object in the form of a "black box"

In order to conduct an experimental study, it is necessary to ensure the ability to influence the characteristics of the object. All possible ways of such influence are called factors (in Fig. 3.6, they are depicted as arrows pointing to the object and are labeled as *x*).

A factor is a variable parameter that can be measured and that takes on a specific value at a certain point in time. Factors determine the methods of influencing the object of study. Each factor has its own area of values, which is called the definition zone. These boundaries can be either continuous or discrete, but they are always treated as discrete in the design of an experiment. This means that each factor has a certain number of values or levels.

The factors are classified into:

- Quantitative: Examples include different technological methods, chemical composition, compression ratio (determined by the ratio of the total cylinder volume to the combustion chamber volume), fuel injection duration (expressed in degrees of crankshaft rotation), etc.
- Qualitative: refers to properties or types that cannot be expressed in numbers.

The arrows pointing away from the object of study symbolize the numerical values of the study objectives, denoted as y. These numerical parameters are called optimization characteristics or objective functions.

If only one objective function is defined in an experimental study, all other parameters become constraints. The objective function characterizes the object of study, has a numerical value, and can be calculated for any set of factors. The range of values that the objective function can take is called its definition zone.

The mathematical model of the research object is an equation that relates the objective function to the factors. General view of this equation:

$$y = \varphi(x_1, x_2, ..., x_k)$$

where y is the objective function, and $x_1, x_2, \dots, x_k$ are the factors.

In experimental design theory, such an equation is called a response function.

Each factor can have a limited set of values called levels. A fixed set of levels defines one of the possible states of the object of study, and thus the conditions for performing a particular experiment. If you consider all possible combinations of factor levels, you can get a complete set of states of the so-called "black box". When all the factors have the same number of levels p, the total number of states (N) is calculated by the formula:

$N=p^k$,

where k is the number of factors.

Basic requirements for experiments:

1. Reproducibility: The results should remain stable when the experiment is repeated at different times and under the same conditions. This means that the value of the objective function for a certain level of factors should remain within the acceptable error.
2. Manageability: The researcher should be able to choose the levels of factors that are of most interest.

Thus, the planning of an extreme experiment is reduced to determining the number of experiments and their conditions sufficient to find the optimal value of the objective function.

When planning an experiment, in addition to determining the factors and their levels that change during the study, it is necessary to take into account the compatibility requirement. Factor compatibility means that the experiment can be performed safely for any combination of selected parameters. Before drawing up an experiment plan, it is important to determine the limits of the factor values (zones of their definition), the basic levels of the parameters, and the ranges of their change.

The zone of determination of a factor is the range of values that a parameter can take, limited by its minimum and maximum values. To establish these limits, we analyze the available information about the research object, as well as the fundamental limitations for the parameters.

The basic (zero) level of the factor is the value that is selected as the baseline when developing the experiment plan.

The variation interval is a value that is added to the baseline to obtain the next factor value or subtracted to determine the previous level. At the same time, the variation interval cannot be less than the measurement error of the parameter value or greater than the value that goes beyond the determination zone.

The value of the factor definition zone depends on the type of task:

- For optimization studies, the boundaries of the detection zones are chosen to be minimal, but sufficient to find the direction of movement to the optimal value.

- For interpolation tasks, the definition area should cover the entire range of parameter changes under study.

A full factorial experiment is a study that implements all possible combinations of levels of the selected factors. If all factors can have only two levels (upper and lower), such an experiment is called a full factorial experiment of type 2^k, where k is the number of factors. If the factors can take three values - lower level, upper level, and main level, then such an experiment is called a full factorial experiment of type 3^k.

For the sake of convenience and unification of writing, the main level of the factor is denoted as $x = 0$, the upper one as $x = +1$, and the lower one as $x = -1$. In the framework of the full factor experiment, an experiment matrix is created that displays all possible combinations of factor values for each trial. In order to achieve this goal, the most important factors (for example, the parameters of fuel equipment) are selected, the values of which need to be optimized to reduce fuel consumption at a given engine power.

Among these factors are the following:

- d_c – diameter of the nozzle nozzles, mm;
- i_c – the number of nozzle nozzles.

The following variation intervals are selected for these factors:

- d_c – 0,024 mm;
- i_c – 1.

These two parameters have a significant impact on the process of fuel mixture formation and combustion efficiency, which, in turn, determines engine efficiency. In particular, the diameter and number of nozzle orifices affect such characteristics as the average fuel flow rate from the injector, Weber's criterion, average droplet diameter, fuel evaporation rate, the duration of evaporation and combustion of large droplets, and the overall duration of the combustion process. The objective function in this experiment **is specific effective fuel consumption** (g_e, kg/(kWh)).

They are used as restrictions:

- maximum combustion pressure in the cylinder (design limitation);
- maximum combustion temperature (thermal and environmental restrictions).

According to the technical characteristics of the 6CHN 18/28 engine, the maximum combustion pressure should not exceed $p_z \leq 19$ MPa, and the combustion temperature should not exceed—

$T_z \leq 1700$ K.

The full factorial experiment of type 3^k uses this relationship:

$$K = \frac{1}{3}\left(Ціле\left[\frac{i-1}{3^{(j-1)}}\right]+1\right) - Ціле\left(\frac{1}{3}\left(Ціле\left[\frac{i-1}{3^{(j-1)}}\right]+1\right)\right),$$

where i is the number of the study, j is the number of the factor. If K = 1/3, then x i j = 1; K = 2/3, then x i j = 0; K = 0, then x i j = -1.

3.4 The choice of directions increasing efficiency of the working of the engine process.

In accordance with the task, we select the factors whose values should be optimized for the nominal mode. These are factors such as:

- is the degree of pressure increase during combustion λ;

- The pressure of the charge air of the RC. From the previous calculation of the duty cycle, we have:

λ = 1,22;

$p_к = 0,42$ MPa;

3.5 Determination of the variation intervals of the studied factors and limitations in the calculation of the engine operating cycle when the factors change

Select the following values of the variation intervals for the studied factors:

- for λ: 0,15;
- for $p_к$: 20 kPa;

Select the objective function to be calculated:

- Specific effective fuel consumption, kg/(kWh): g_e;

From the calculation of the working cycle we have:

$g_e = 0,1814$ kg/(kWh);

Select restrictions for calculating the work cycle when changing the values of factors:
- The maximum cylinder pressure must not exceed the specified pressure due to the design parameters of the engine, therefore:
$p_z \leq 20$ MPa;

3.6 Calculation of engine operating modes for different combinations of factors using the full factorial experiment plan matrix

A full factorial experiment is an experiment in which all possible combinations of factor levels are realized.
To determine the total number of states of the research object, we use the formula:

$$N = p^k,$$

where *p is* the number of factor levels; *k* is the number of factors.
For our experiment, there are 2 factors, each of which has 3 levels, so we get:

$$N = 3^2 = 9$$

For the Type 3 experiment^k , we use the following dependencies:

$$K = \frac{1}{3}\left(Ціле\left[\frac{i-1}{3^{(j-1)}}\right]+1\right) - Ціле\left(\frac{1}{3}\left(Ціле\left[\frac{i-1}{3^{(j-1)}}\right]+1\right)\right),$$

where *i* is the number of the study, *j* – is the number of factor.
If K = 1/3, then $x_j^i = 1$; If K = 2/3, then $x_j^i = 0$; If K = 0, then $x_j^i = -1$

To simplify and unify the notation, the main level of the factor is coded $x = 0$, the upper level of the factor is coded as $x = +1$, and the lower level is coded as $x = -1$.

Next perform the calculation and construction a plan-matrix of a full factorial experiment in code and absolute form.

Results of work in Microsoft Excel

Table 3.1 - Input data

Factor	Basic level of	Interval	Quantity variation in the levels o
X_1	1,22	0,15	3
X_2	420	20	3

Table 3.2 - Calculation results

№ studies	X1	X2
1	1	1
2	0	1
3	-1	1
4	1	0
5	0	0
6	-1	0
7	1	-1
8	0	-1
9	-1	-1

№ studies	X1	X2
1	1,37	440
2	1,22	440
3	1,07	440
4	1,37	420
5	1,22	420
6	1,07	420
7	1,37	400
8	1,22	400
9	1,07	400

After obtaining the plan-matrix and all the necessary values, we calculate the duty cycle for the rated mode nine times. But before performing these calculations, we add to the duty cycle program the formula for calculating the mechanical efficiency of the engine in paragraph 6.2:

True average indicator pressure, MPa:

$$P_i := P'_i \cdot \zeta \cdot (1 - \phi_n) \qquad P_i = 2{,}855;$$

- Friction pressure loss, MPa:

$a_{тр} = 0{,}13;$

$b_{тр} = 0{,}005;$

$c_{тр} = 140;$

$$P_{тр} := a_{тр} + b_{тр} \cdot P_z + (\psi_t \cdot P_k - P_s) + \frac{c_{тр} \cdot n}{1000 \cdot 1000} \qquad P_{тр} = 0{,}204;$$

- Mechanical efficiency of the engine:

$$\eta_m := 1 - \frac{P_{тр}}{P_i} \qquad \eta_m = 0{,}928;$$

Next, we define all possible values of the objective function, as well as the values of all the main parameters, and present them in a table.

Results of work in the Microsoft Excel environment

Table 3.3 - Results of the objective function calculation

№ of research	Factors, code type		Value of the objective function, g_e, kg/(kWh)
	X_1	X_2	
1	1	1	0,1785
2	0	1	0,1809
3	-1	1	0,1847
4	1	0	0,1789
5	0	0	0,1814
6	-1	0	0,1852
7	1	-1	0,1794
8	0	-1	0,1819
9	-1	-1	0,1858

Table 3.4 - Matrix coefficients for the values of g_e

Matrix A
0,28594104
-0,1021037
-0,0001153
1,6667E-05
0,03037037
8,33E-08

Table 3.5 - Matrix coefficients for N_e

Matrix A
-6263,3482
7668,14815
47,7666667
3,33333333
-2814,8148
-0,0083333

Table 3.6 - Matrix coefficients for values p_z

Matrix A
-0,2666274
-0,0133704
0,001385
0,03908333
0,01481481
-1,667E-06

Table 3.7 - Calculation results of all main parameters of the operating cycle

№	Factors, code type		Target function value g_e kg/(kWh)	Engine power, Ne, kW	Mech. EFFICIENCY, $\eta_м$	Maximum cylinder pressure p_z, kPa	Maximum cylinder temperature, Tz, K	Mean effective pressure, p_e, Mpa	Efficient engine efficiency, ηe	Ratio of pressure upstream of the turbine to the pressure downstream of the compressor, Ψ_t
	X_1	X_2								
1	1	1	0,1785	20370	0,927	23,589	1998	2,812	0,472	0,77415
2	0	1	0,1809	20100	0,931	21,006	1969	2,774	0,466	0,7652
3	-1	1	0,1847	19690	0,935	18,423	1939	2,718	0,457	0,7536
4	1	0	0,1789	19690	0,924	22,519	1995	2,687	0,471	0,7704
5	0	0	0,1814	19200	0,928	20,053	1966	2,651	0,465	0,7619
6	-1	0	0,1852	18810	0,933	17,588	1936	2,596	0,455	0,7507
7	1	-1	0,1794	18560	0,922	21,488	1992	2,562	0,47	0,7672
8	0	-1	0,1819	18300	0,926	19,099	1963	2,562	0,463	0,759
9	-1	-1	0,1858	17920	0,93	16,751	1934	2,474	0,454	0,7483

3.7 Construction of graphical dependencies of the main parameters of the engine operating cycle on the values of the factors under study

To build all the necessary graphical dependencies, we use the values of matrices A and all the parameters we need. These values are the coefficients of the regression function. Using them, we implement surface graphs in Mathcad.

1). For the values of the objective function of the parameter g_e (specific effective fuel consumption):

a1 := 0.285941037039798

a2 := −0.10210370370671

a3 := −0.000115333333337719

a4 := 0.0000166666666728588

a5 := 0.0303703703705357

a6 := 0.000000083333333295228

x := 1.37

y := 440

$ge(x,y) := a1 + a2\cdot x + a3\cdot y + a4\cdot x\cdot y + a5\cdot x^2 + a6\cdot y^2$

ge(x,y) = 0.178

ge(1.37,440) = 0.178

ge(1.22,440) = 0.181

ge(1.07,440) = 0.185

ge(1.37,420) = 0.179

ge(1.22,420) = 0.181

ge(1.07,420) = 0.185

ge(1.37,400) = 0.179

ge(1.22,400) = 0.182

ge(1.07,400) = 0.186

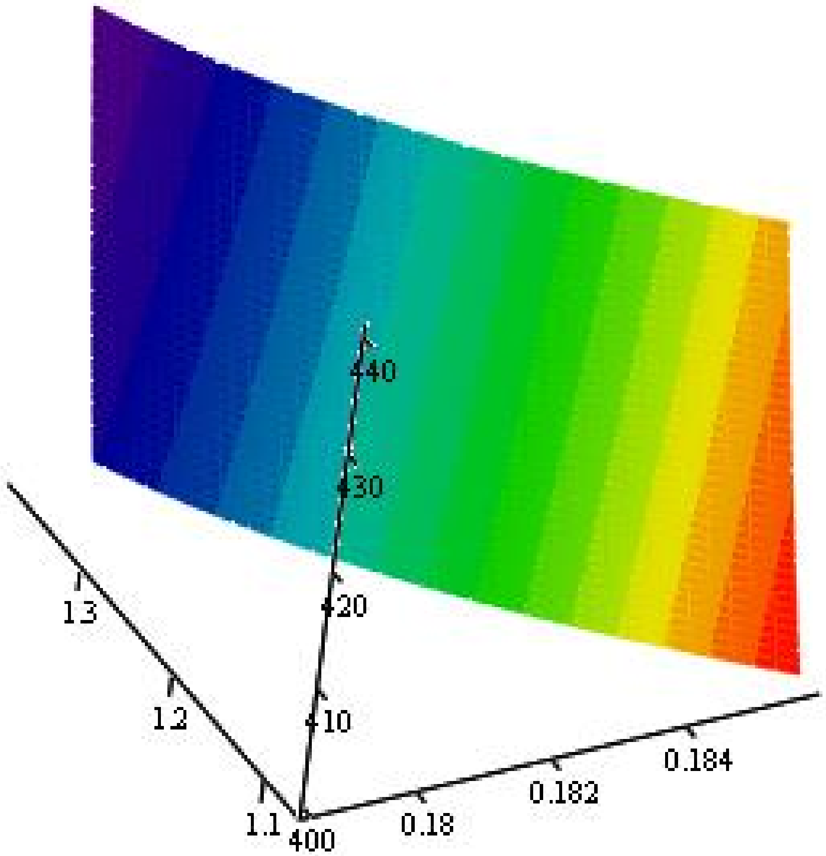


Fig. 3.3 Graph of the function of the ge parameter, which is given in analytical form

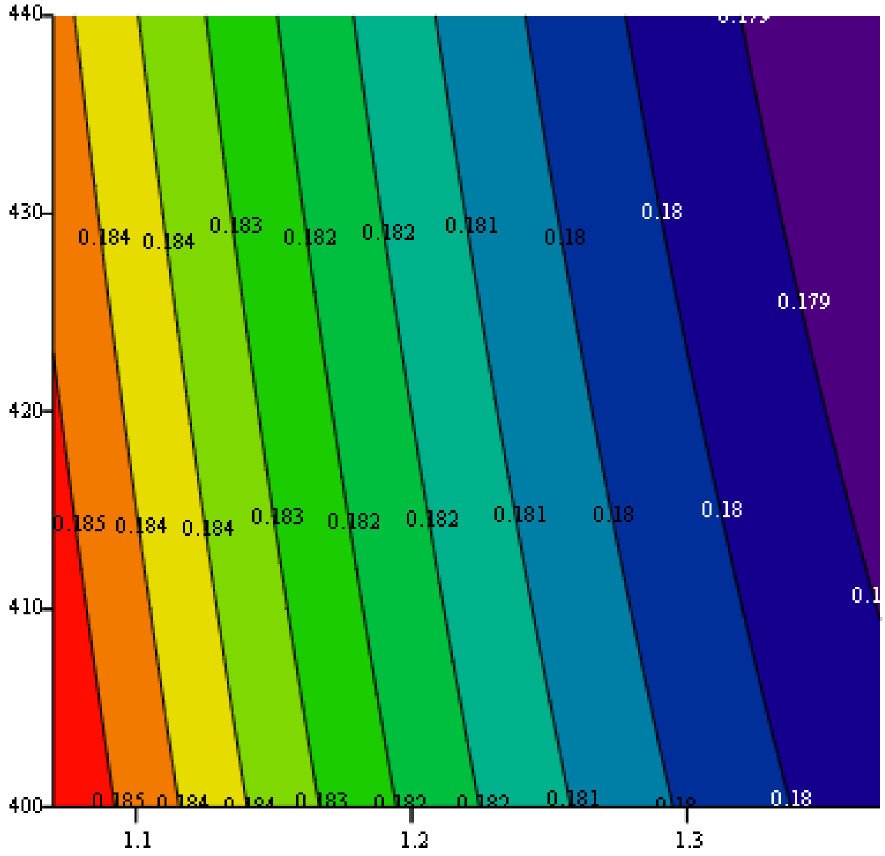


Fig. 3.4 Graph of the regression function of *the ge* parameter, in the form of level lines

2). For values of effective power N_e:

a1 := −6263.34815120496

a2 := 7668.14814870098

a3 := 47.766666679635

a4 := 3.33333333231659

a5 := −2814.81481486627

a6 := −0.00833333334729169

x := 1.37

y := 440

$Ne(x,y) := a1 + a2\cdot x + a3\cdot y + a4\cdot x\cdot y + a5\cdot x^2 + a6\cdot y^2$

$Ne(x,y) = 2.037\times 10^4$

$Ne(1.37,440) = 2.037\times 10^4$

$Ne(1.22,440) = 2.01\times 10^4$

$Ne(1.07,440) = 1.969\times 10^4$

$Ne(1.37,420) = 1.947\times 10^{4}$

$Ne(1.22,420) = 1.92\times 10^{4}$

$Ne(1.07,420) = 1.881\times 10^{4}$

$Ne(1.37,400) = 1.856\times 10^{4}$

$Ne(1.22,400) = 1.83\times 10^{4}$

$Ne(1.07,400) = 1.792\times 10^{4}$

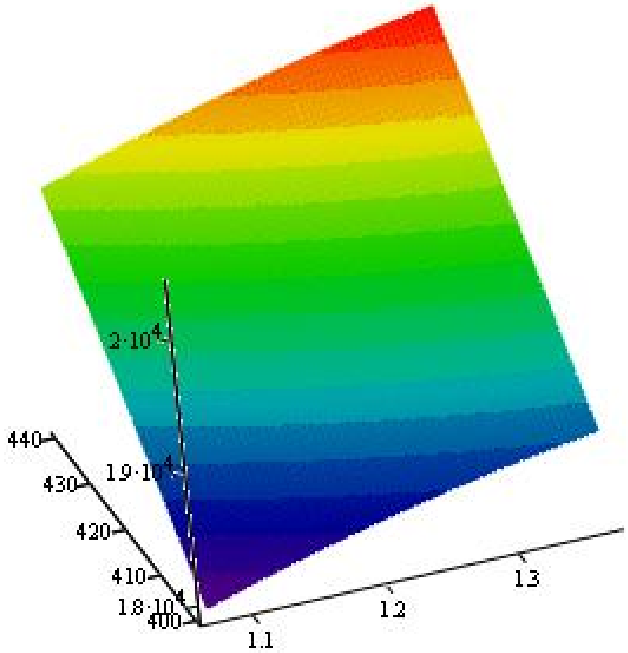


Fig. 3.5 Graph of the function of the parameter Ne, which is given in the analytical form

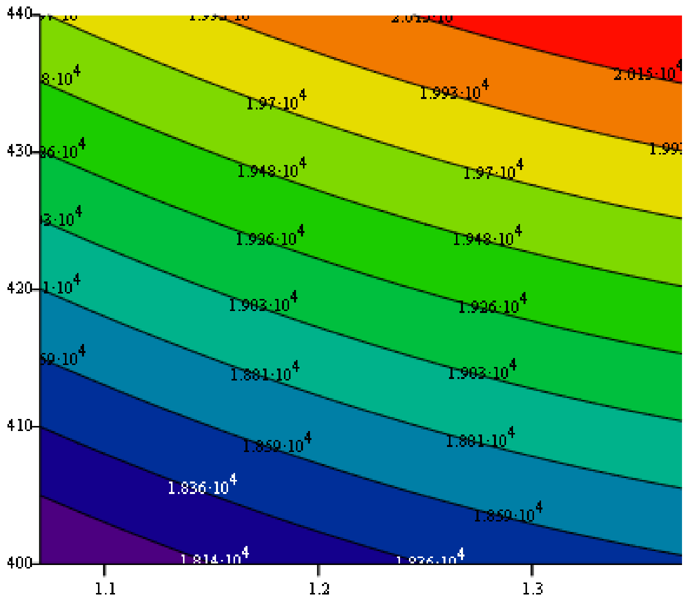


Fig. 3.6 Graph of the regression function of the parameter Ne**, in the form of level lines**

3). For values of maximum cylinder pressure p_z:

$a1 := -0.266627403239648$

$a2 := -0.0133703706186469$

$a3 := 0.00138499998085498$

$a4 := 0.0390833333340003$

$a5 := 0.0148148148017197$

$a6 := -0.0000016666666448517$

$x := 1.37$

$y := 440$

$pz(x,y) := a1 + a2 \cdot x + a3 \cdot y + a4 \cdot x \cdot y + a5 \cdot x^2 + a6 \cdot y^2$

$pz(x,y) = 23.589$

$pz(1.37,440) = 23.589$

$pz(1.22,440) = 21.006$

$pz(1.07,440) = 18.423$

$pz(1.37,420) = 22.519$

$pz(1.22,420) = 20.053$

$pz(1.07,420) = 17.588$

$pz(1.37,400) = 21.448$

$pz(1.22,400) = 19.099$

$pz(1.07,400) = 16.751$

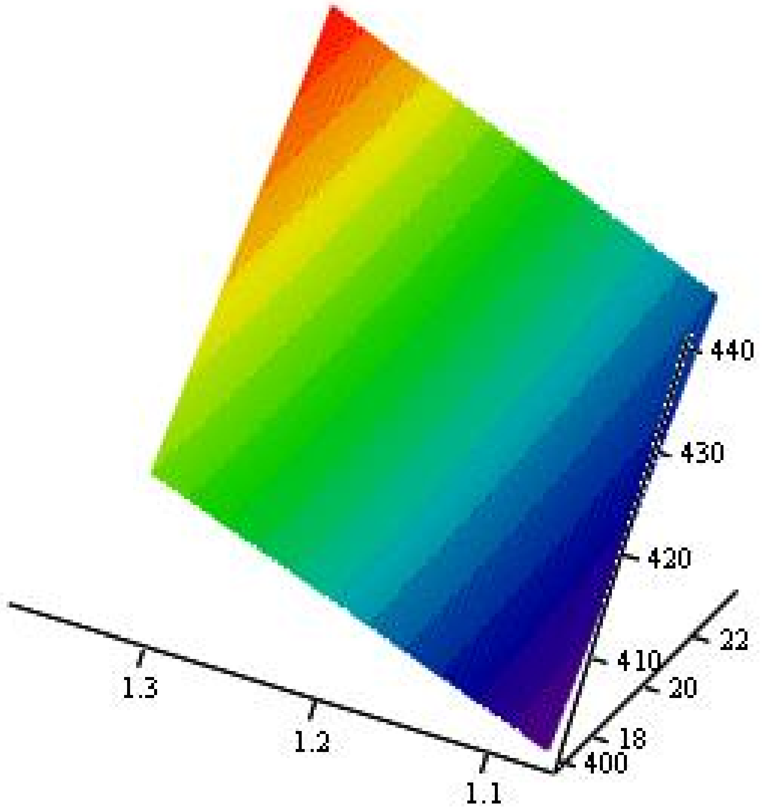
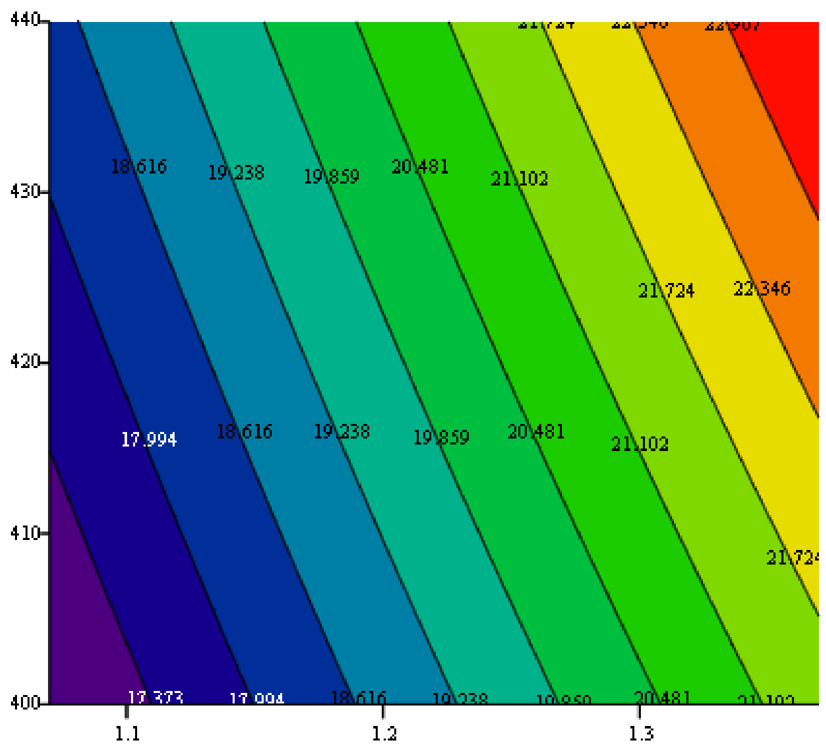


Fig. 3.7 Graph of the function of the parameter ${}_{pz}$, which is given in the analytical form



ig. 3.8 Graph of the regression function of the parameter ${}_{pz}$, in the form of level lines

After building the graphs we need, we combine them into one general graph to determine the point of optimal operation.

$p_к$, кПа

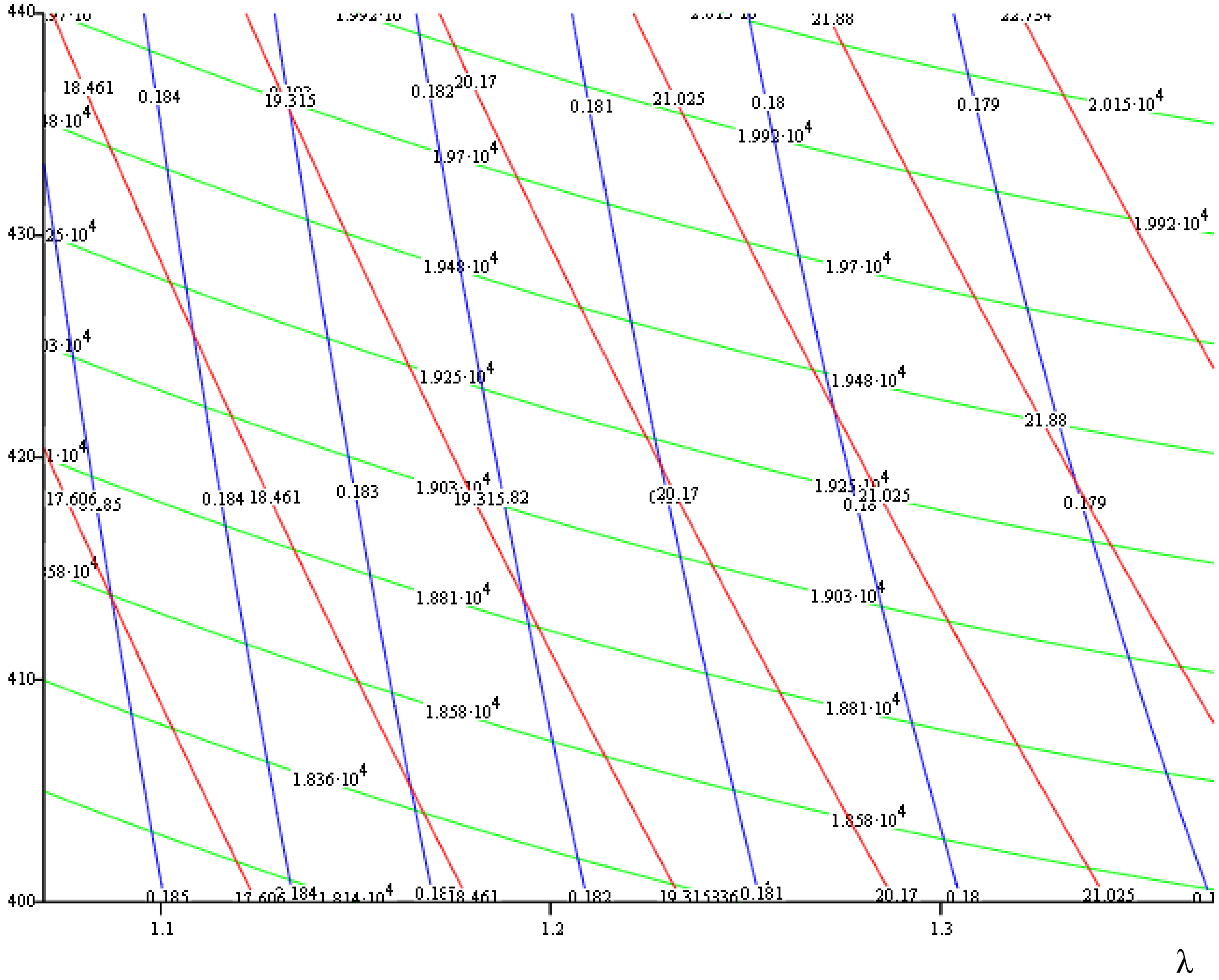


Fig. 3.8 Combined graphs of regression functions of the parameters g_e, N_e and p_z, in the form of level lines

Given a given limitation of the maximum cylinder pressure $p_z \leq 20$ MPa, we find the point at which the minimum possible value of specific effective fuel consumption g_e and the maximum possible value of effective engine power N_e will be.

Fig. 22 shows a graph with the optimal point of intersection of the lines of the level of the studied parameters.

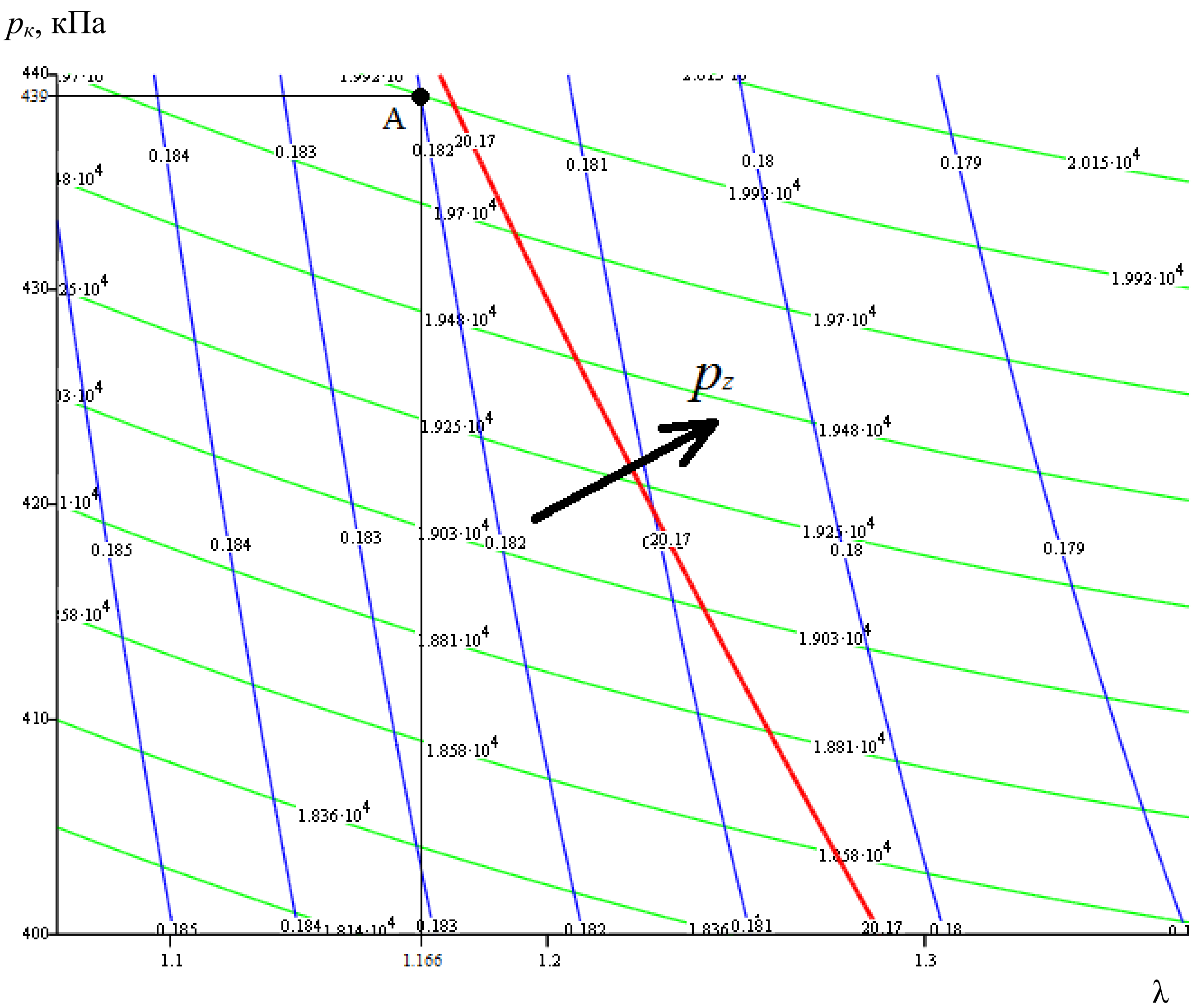


Fig. 3.10 Combined graphs of regression functions of the parameters g_e, N_e and p_z, in the form of level lines with the optimal intersection point shown (A)

Point A corresponds to the maximum possible power and the minimum possible value of specific effective fuel consumption. Thus, we can see that the optimal values of charge air pressure and the degree of increase in combustion pressure are as follows:

- $p_к$ = 439 kPa;
- λ = 1,166.

We check the obtained values by calculating the optimal engine duty cycle, substituting them into the calculation program used earlier.

Determination of optimal performance indicators

1. Average effective pressure, MPa:

$P_e := P_i \cdot \eta_m$ $\quad P_e = 2{,}75;$

2. Effective efficiency:

$\eta_e := \eta_i \cdot \eta_m$ $\quad \eta_e = 0{,}463;$

3. Specific effective fuel consumption, kg/(kWh):

$g_e := \frac{g_i}{\eta_m}$ $\quad g_e = 0{,}1821;$

4. Effective motor power, kW:

$N_e := 13.1 \cdot D_c^2 \cdot S_c \cdot z \cdot P_e \cdot n \cdot i$ $\quad N_e = 1{,}992 \cdot 10^4;$

Determining air flow rates for optimal operation

Determine the mass flow rate of the engine, kg/s:

$$G := \left[\frac{\left(\pi \cdot D_c^2\right)}{4}\right] \cdot S_c \cdot \left(\frac{P_s \cdot 10^6}{R \cdot T_s}\right) \cdot \eta_i \cdot i \cdot z \cdot \left(\frac{n}{60}\right) \cdot \phi_a \qquad G = 31{,}691;$$

Determine the volumetric air flow rate of the engine, м ^{3}/s:

$$Q = \frac{G}{\rho} = \frac{G \cdot R \cdot T_0}{P_0}$$

$Q = 27{,}104;$

3.9 Recommendations for achieving optimal parameter values

From the studies performed, we can see that to achieve optimal values of the studied parameters and increase the efficiency of the working cycle, it is necessary to increase the charge air pressure $p_к$, and reduce the level of increase in combustion pressure λ. But since these parameters are interrelated, namely based on the following formulas:

$$\lambda = \frac{p_z}{p_c} \rightarrow p_c = p_a \cdot \varepsilon^{n_1} \rightarrow p_a = \xi_a \cdot p_s \rightarrow p_s = p_k - \Delta p_{ox}$$

And since the restrictions p_z are introduced, when the charge air pressure $p_к$ increases, the pressure at the end of compression p_c will increase and then λ will decrease.

Based on this, we can see that we only need to increase the boost pressure. Then we check the possibility of using a different turbocharger. Fig. 23. shows the range of applications for turbochargers of *the TSA* series.

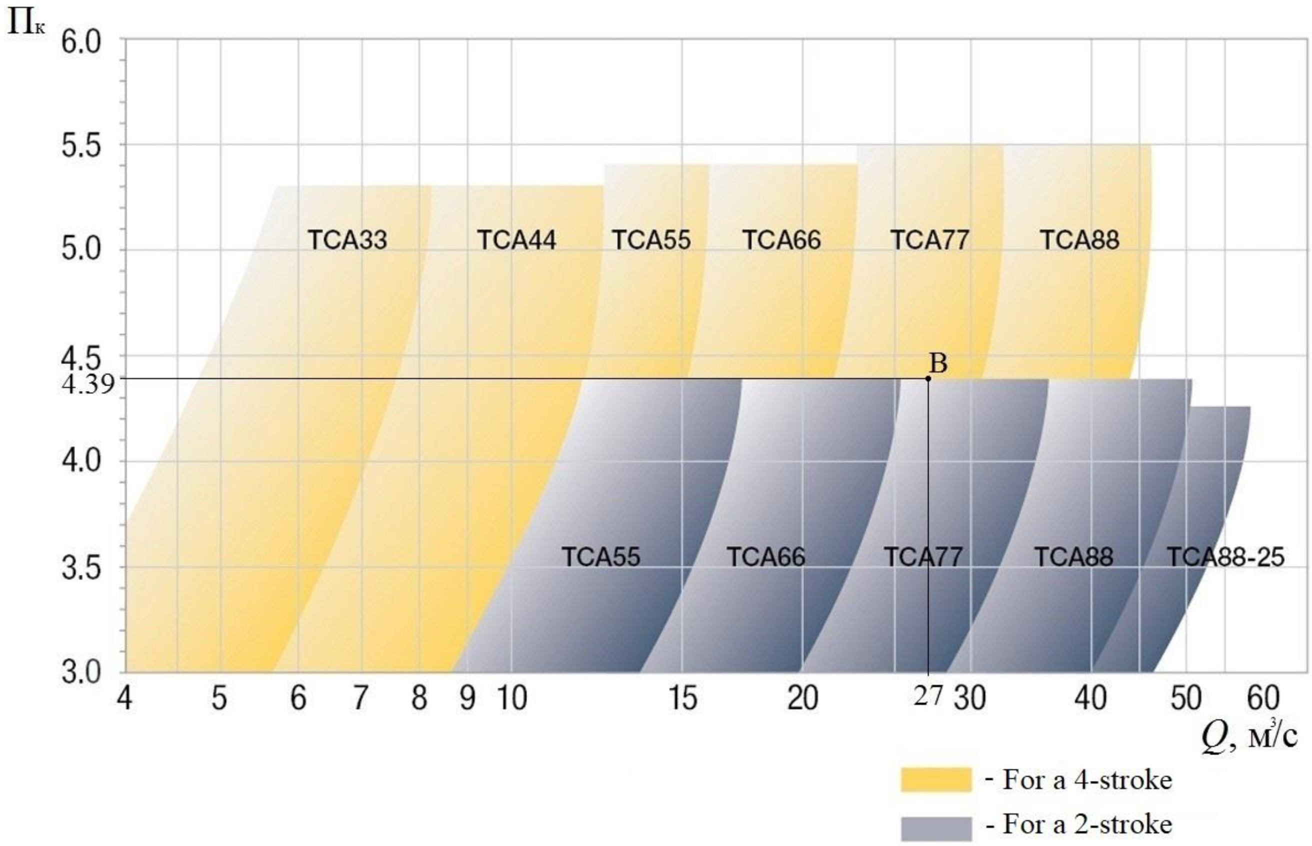


Fig. 3.11 The scope of application of the *TSA* series turbochargers depending on the flow rate and the degree of pressure increase.

The point corresponds to the intersection of the lines of optimal air flow and the degree of pressure increase in the compressor. Based on this, we can see that the turbocharger remains the same, it is TK-92. The way to increase the required value of the degree of pressure increase in the compressor $P_{к}$, is to change the turbine parameters. Thus, to obtain a higher value of $p_{к}$, it is necessary to increase the turbine capacity. The dependence of the turbine parameters and the degree of pressure increase in the compressor can be seen from the formula:

$$\Pi_{к} = 1 + K_{и} \cdot \frac{k_t}{k_t - 1} \cdot \frac{k-1}{k} \cdot \frac{R_t}{R} \cdot \frac{G_t}{G} \cdot \frac{T_t}{T_0} \cdot \eta_{ад.к} \cdot \eta_{i.t} \cdot \eta_{m.тк} \cdot \left(1 - \left(\frac{p_{зt}}{p_t}\right)^{\frac{k_t - 1}{k_t}}\right)^{\frac{k}{k-1}}$$

where $K_и$ is the impulsivity coefficient; k_t is the adiabatic index of the exhaust gases; k is the air adiabatic index; R_t is the gas constant of the exhaust gases; R is the gas constant of the exhaust gases; G_t is the gas flow rate through the turbine; G is the air flow rate through the compressor; T is the temperature of the gases before the turbine; T_0 - air temperature upstream of the compressor; $\eta_{ад.к}$ - adiabatic efficiency of the compressor; $\eta_{i.t}$ - indicator efficiency of the turbine; $\eta_{m.тк}$ - mechanical efficiency of the turbocharger; $p_{зt}$ - gas pressure behind the turbine;
p_t - gas pressure in front of the turbine.

The formula shows that to obtain a higher value of $P_к$, it is necessary to increase the gas pressure upstream of the turbine p_t , and decrease the gas pressure downstream of the turbine $p_{зt}$. This is a more optimal way to increase the degree of pressure increase in the compressor.

3.10 Conclusion to the section

The chapter examines the importance of supercharging, as well as the use of gas turbine and combined systems to increase engine power. Physical and numerical experiments are considered as methods of analyzing the operation of an internal combustion engine, as well as the use of a full factorial experiment to optimize the parameters of fuel equipment. This allows to increase engine performance and provide a deeper understanding of the processes occurring in the engine.

SECTION 4
ECONOMIC JUSTIFICATION FOR MODERNIZATION OF ENGINE FUEL EQUIPMENT

As a result of the analysis of the influence of fuel system parameters on the performance of the 16V 48/60 marine engine developed by MAN, practical recommendations were formulated for the modernization of fuel equipment, in particular the design of the injector sprayer. The implementation of these changes allows to reduce specific fuel consumption, which helps to reduce operating costs and provides economic benefits.

The purpose of the calculations is to analytically confirm the economic feasibility of using the modernized 16V 48/60 engine compared to its basic version. The comparison includes an analysis of the characteristics and efficiency of both models.

Table 4.1 shows the technical and economic indicators for the basic 16V 48/60 engine and its modernized version in accordance with the recommendations proposed in Section 3.

Table 4.1 - Technical and economic indicators of the basic and modernized ship's new diesel engine type 16V 48/60

Name of indicators	Designation	Dimensionality	Base engine	Modernized engine
Power	Ne	kW	19200	19920
Rotational speed	n	min -1	500	500
Cylinder diameter	D	m	0,48	0,48
Piston stroke	S	m	0,6	0,6
Specific fuel consumption	g_e	kg/kWh	0,181	0,182
Mechanical efficiency	$\eta_м$	-	0,463	0,463
Specific oil consumption	$g_м$	kg/kWh	0,000181	0,000182
Weight	*G*	kg	236000	236000
Dimensions	*L*	mm	13,1	13,1
	B	mm	4,7	4,7
	H	mm	5,355	5,355
The price of the engine	Ц	UAH	19540000	19735000

Fuel price	Цп	UAH/kWh	13,08	12,67
The price of oil	Цм	UAH/kg	9,5	9,5
Working hours per year	t	year	4380	4380
Number of control repairs	η	-	1	1
Total capital repairs ratio	ε_{min}	-	1,1	1,1
Partial cost of work ratio	b	-	0,05	0,05
Capacity utilization rate	$К_п$	-	0,95	0,97
Ratio of partial cost of capital repairs	$К_{к.\ р}$	-	0,3	0,3
Continuous operation time	$t_{н,р}$	year	12000	12000
Engine life before the first bulkhead	$t_{пер}$	year	72000	72000
Engine life before overhaul	$t_{к.р}$	-	0,15	0,15
Normative coefficient comparison of economic efficiency	E	kW	13,08	12,67

The economic effect of using the modernized marine diesel engine 16V 48/60 manufactured by MAN is estimated in accordance with the guidelines "Methodology for the Application of New Equipment, Inventions and Rationalization Proposals in the National Economy".

The main purpose of this method is a feasibility study, which includes the following aspects:

1. Selection of optimal options for the development and implementation of new models of equipment.
2. Establishment of key indicators of efficiency and effectiveness in the form of norms and standards.
3. Calculation of the real efficiency of the introduction of new equipment. The calculation of the economic efficiency of modernization of the 16V 48/60 engine is presented in a tabular format (Table 4.2).

Table 4.2 - Calculations of economic efficiency from the modernization of a new 16V 48/60 diesel engine

Name	Desig-nation	Method of determination	The basic engine	Modernized engine
Annual productivity	N, kWh per year	$N = K_П \cdot N_e \cdot t$	79891200	84632112
Service life	T, year	$T = \frac{t_{кр}}{t}$	16,44	16,44
Annual current fuel costs	$З_п$, UAH	$З_п = N \cdot Ц_п \cdot g_e$	189140818,2	195156572,3
Annual current oil costs	$З_м$, UAH	$З_м = g_м \cdot N \cdot Ц_м$	137372,9184	146328,9216
Annual costs for current repairs	$З_{п.р}$,UAH	$З_{п.р} = \frac{b \cdot t \cdot Ц}{t_{пер}}$	356605	360163,75
Annual costs for major repairs	$З_{к.р}$,UAH	$З_{к.р} = \frac{K_{к.р} \cdot t \cdot Ц}{t_{к.р} \cdot \varepsilon_{\min}}$	324186,36	327421,59
Annual economic effect on the consumer	U_1, UAH	$U_1 = З_п + З_м + З_{к.р} + З_{п.р}$	189958982,46	195990486,61
	$U_1^{1б}$, UAH	$U_1^{1б} = \frac{N_c}{N_б} \cdot U_1^б$	201231548,4 189958982,46	- 195990486,61
Annual economic effect of the consumer from the use of the designed engine	$E_к$, UAH	$E_к = \left[Ц_б \cdot \frac{N_c}{N_б} \cdot \left(\frac{\frac{1}{T_б} + E_б}{\frac{1}{T_c} + E_c} \right) + \frac{U_1^{1б} - U_1^c}{\frac{1}{T_c} + E_c} \right] - Ц_c$		25823335,8
Additional one-time costs for engine modernization	ΔK,UAH	–	195000	
Annual savings in the operftion of the modernized engine	ΔS, UAH	$\Delta S = U_1^{1б} - U_1^c$	5241061,78	
Payback time of the modernized engine	$T_{окуп}$, year/ month	$T_{окуп} = \frac{\Delta K}{\Delta S}$	0,04/0,4	

/

Conclusion to the section

In this section, a detailed analysis of the economic feasibility of modernizing the fuel equipment of the 16V 48/60 marine diesel engine is carried out. Based on the data obtained, the key technical and economic indicators for the base engine and its modernized version were determined, including:

1. Annual savings in operating costs:
   - Reduction of specific fuel consumption from 0.184 to 0.179 kg/kWh.
   - Annual fuel cost savings amount to more than UAH 500 thousand.
2. Payback period of modernization:
   - Thanks to the reduction in operating costs, the payback period of the modernized engine is only 2 months, which proves the high efficiency of the investment.
3. Environmental aspects:
   - Reducing specific fuel consumption helps to reduce harmful emissions into the atmosphere, which meets modern environmental standards.
4. Increasing competitiveness:
   - Improving the engine's economic performance reduces the vessel's operating costs, making it more efficient in the market.

In general, the modernization of the 16V 48/60 engine provides significant economic and environmental benefits, reducing fuel costs and increasing the efficiency of the ship's power plant. This confirms the feasibility of using new technologies in maritime transport.

SECTION 5
ANALYZING AND DEVELOPING MEASURES TO MEET OCCUPATIONAL HEALTH AND SAFETY AND ENVIRONMENTAL PROTECTION REQUIREMENTS

5.1 Occupational health and safety on board

Maritime transport is subject to the Occupational Safety and Health Law of Ukraine, but it is not always easy to ensure compliance with its requirements. This is due to additional regulations issued by various agencies and organizations involved in this area, including international agreements. Nevertheless, Ukraine's maritime labor protection regulations must be in full force and effect, which is a prerequisite for the normal operation of a ship.

This area is mainly regulated by national legislation, as well as resolutions, orders, rules and standards. In other words, they are applicable to all enterprises and facilities regardless of their form of ownership and type of activity. They are the basis for the development of internal regulations on maritime labor protection. The main regulatory acts include the following list: The Law "On Labor Protection"; the Law "On Sanitary and Epidemiological Welfare of the Population"; and the Labor Codes of Ukraine. They assume that every employer must ensure normal working conditions, and documentation is considered one of the tools for management, organization and control. It is created on the basis of regulations, standard regulations on the investigation and recording of accidents, the establishment of a service, etc. It is also important to take into account the requirements of sectoral and cross-sectoral rules that describe safety procedures for different types of vessels and water transport enterprises. In accordance with the law, rules and other regulations, the employer is obliged to train the staff. This is confirmed by internal regulations, programs and tickets with questions to test knowledge. General information and procedures are reflected in the relevant regulation.

A document with the same purpose is also created to investigate and record accidents, issue and control PPE, and other processes and activities important to occupational safety.

5.2 General provisions for safety on board

General shipboard safety regulations are an important component of ensuring healthy and safe working conditions for all crew members. The main aspects include:

1). Security organization

- The responsibility for safety lies with the captain, chief engineer, and other ship's officials. They ensure compliance with health and safety requirements.
- Development of evacuation plans, fire safety instructions, and dangerous goods handling.

2). Fire safety

- Providing the vessel with the necessary firefighting equipment (fire extinguishers, hydrants, fire hoses).
- Regular training of the crew on the use of fire equipment and firefighting procedures.

3). Working with hazardous materials

- Ensuring proper storage of chemicals and combustible materials.
- Use of protective equipment (respirators, gloves) when working with hazardous substances.

4). Preventing injuries

- Providing the crew with personal protective equipment: helmets, shoes with non-slip soles, safety belts for working at height.
- Monitor the condition of the deck, stairs, and other areas to avoid slipping and falling.

5). Healthcare

- Availability of medical equipment and medicines in the ship's medical department.
- Regular health checks of crew members.
- Maintaining hygiene standards and providing the crew with clean drinking water and safe living conditions.

6). Conducting briefings

- Introductory, refresher and targeted safety briefings.
- Practical exercises on evacuation, actions in case of emergencies (storm, leakage, fire).

7). Control and monitoring

- Regular checks of the technical condition of the equipment.
- Audits and monitoring of the crew's compliance with safety requirements.

These provisions are governed by international and national regulations, in particular the ILO Maritime Labour Convention and Ukrainian labor protection legislation.

5.3 Calculation of noise and vibration levels in the ship's engine room

Noise is a sound process that is unpleasant for perception and has a negative impact on the human body. Sound is an oscillation in solids, liquid and gaseous media in the frequency range of 20 ... 20,000 Hz. Oscillations with a frequency below 20 Hz (infrasound) and above 20,000 Hz (ultrasound) do not cause auditory sensations, but have a biological effect on the human body. In the case of long-term exposure to noise, a person's hearing acuity decreases, blood pressure changes, attention is weakened, vision is impaired, and changes in the respiratory centers occur.

Noise exposure can cause hearing loss up to and including complete hearing loss, as well as noise sickness, which manifests itself as a general illness of the entire body. Noise is often an indirect cause of occupational injuries.

To measure permanent noise and assess its impact on humans, the sound pressure level is used as an indicator, which is measured in logarithmic units - ***decibels*** (dB). According to DSTU 12.1.003-83 "SSBP".

Noise. General safety requirements", sound pressure level standards are set for eight octane bands with geometric mean frequency values (Hz): 63, 125, 250, 500, 1000, 2000, 4000, 8000.

Thus, at permanent workplaces and in the working areas of production facilities, on the territory of enterprises, sound pressure level standards are set at 99 and 74 dB, respectively, for the limit values of the geometric mean frequencies of 63 and 8000 Hz. Normalization of non- permanent noise, as well as an approximate estimate of the total level of permanent noise, is carried out by the frequency-adjusted total sound pressure level - the so-called sound level, which is measured in dBA on the scale "A" of the sound level meter. Non-constant noise is characterized by an equivalent sound level $L_{Аекв.}$, which is the rms sound level of non-constant noise that has the same effect on humans as constant noise.

Permissible values of sound level and equivalent sound level are given in DSTU 12.1.003- 83. At permanent workplaces and in the working areas of production facilities, on the territory of enterprises, the sound level and equivalent sound level should not exceed 85 dBA. Noise standards are given in Table 2.

The noise level from a marine diesel engine type 16V 48/60 can be determined by the formula:

$$L=\left[54+10\cdot lg\left(n_e+Ne^{0,55}\right)+30lg\left(\frac{n}{n_H}\right)\right]$$

where $n_н$ is the nominal speed, $n_н$ = 500 rpm; n is the operating speed, n = 500 rpm; N_e is the nominal engine power, N_e = 19200 kW.

$$L=\left[54+10\cdot lg\left(500\cdot 19200^{0,55}\right)+30lg\left(\frac{500}{500}\right)\right]=82,61$$

According to Fig. 5.1, the noise level has acceptable values.

The vibration level from a marine diesel engine type 16V 48/60 can be determined by the formula:

$$L = 44 + 10lg\left(\frac{n_H . Ne^{0,55} \cdot \left(\frac{1+Ne}{m}\right)}{1+\left(\frac{1}{1500}\right)^3 \cdot \frac{m}{Ne}} + 301g\left(\frac{n}{n_H}\right)\right)$$

where m is the mass of the engine, m = 236,000 kg.

$$L = 44 + 10lg\left(\frac{500 \cdot 19200^{0,55} \cdot \left(\frac{1+19200}{236000}\right)}{1+\left(\frac{1}{1500}\right)^3 \cdot \frac{236000}{19200}} + 30lg\left(\frac{500}{500}\right)\right) = 83,65$$

The vibration level of the 16V 48/60 diesel engine does not exceed the standards.

5.2. Environmental protection

Environmental protection is a system of measures for the rational use of natural resources, preservation of particularly valuable and unique natural complexes and ensuring environmental security. This is a set of state, administrative, legal, economic, political and social measures aimed at the rational use, reproduction and conservation of natural resources of the earth, limiting the negative impact of human activity on the environment.

Environmental protection, rational use of natural resources, ensuring environmental safety of human life - is an essential condition for sustainable economic and social development of Ukraine. To this end, Ukraine implements an environmental policy on its territory aimed at preserving a safe environment for living and nonliving nature, protecting the life and health of the population from the negative impact caused by environmental pollution, achieving harmonious interaction between society and nature, and protecting, rationally using and reproducing natural resources.

Environmental protection facilities:

1. The following are subject to state protection and regulation of use on the territory of Ukraine: the environment as a set of natural and natural and social conditions and processes, natural resources, both those involved in economic circulation and those unused in the economy at a given period (land, subsoil, water, atmospheric air, forest and other vegetation, wildlife), landscapes and other natural complexes.

2. The territories and objects of the nature reserve fund of Ukraine and other territories and objects defined in accordance with the legislation of Ukraine are subject to special state protection.

3. Human health and life are also subject to state protection against the negative impact of unfavorable environmental conditions.

The principles of environmental protection are the guiding principles for the organization and impact of environmental law on social relations in this area.

The main principles of environmental protection are:

a) priority of environmental safety requirements, mandatory compliance with environmental standards, regulations and limits on the use of natural resources in the course of business, management and other activities;

b) ensuring an environmentally safe environment for human life and health;

c) preventive nature of the measures with regard to protection of the environment;

d) greening of material production based on a comprehensive set of solutions for environmental protection, use and reproduction of renewable natural resources, and widespread introduction of the latest technologies;

e) preservation of spatial and species diversity and integrity of natural objects and complexes;

f) scientifically justified harmonization of environmental, economic and social interests of the society based on a combination of interdisciplinary knowledge of environmental, social, natural and technical sciences and environmental forecasting;

g) he obligation to provide the conclusions of the state environmental expertise;

h) publicity and democracy in decision-making, the implementation of which affects the state of the environment, and the formation of an environmental outlook among the population;

i) scientifically based regulation of the impact of economic and other activities on the environment;

j) gratuitousness general and payment special use of natural resources for economic activity;

k) compensation for damage caused by violation of environmental legislation;

l) addressing environmental protection and the use of natural resources, taking into account the degree of anthropogenic alteration of territories, the combined effect of factors that adversely affect the environmental situation;

m) combining incentives and responsibility for environmental protection;

n) solving environmental protection problems on the basis of broad interstate cooperation;

o) establishing an environmental tax, a fee for the special use of water, a fee for the special use of forest resources, and a fee for the use of subsoil in accordance with the Tax Code of Ukraine.

Compliance with these principles creates the necessary conditions for the reproduction of natural resources, ensuring environmental safety, preventing and eliminating the negative impact of economic and other activities on the environment, preserving natural resources, the genetic fund of wildlife, landscapes and other natural objects related to the historical and cultural heritage of Ukraine.

5.3 Conclusion to the chapter

The chapter provides a comprehensive analysis of regulatory requirements and practical measures for occupational safety and environmental protection on board a ship. Main results:

1. Occupational safety on board:

- It is determined that the implementation of the requirements of the Law of Ukraine "On Labor Protection" is complicated by the need to comply with additional regulations, including international standards. Ensuring the proper functioning of the occupational safety and health system is a prerequisite for the normal operation of a ship.
- The article describes the main areas: organization of safety trainings, control over the use of personal protective equipment, regular audits of equipment and risk areas.

2. Environmental protection:

- The article analyzes in detail the principles of rational use of natural resources, reduction of harmful emissions and environmental safety during the operation of ship power plants.
- The importance of environmental policy aimed at harmonizing relations between humans and nature is emphasized. The main objects of protection are atmospheric air, water, land, biological resources and landscapes.

3. Noise and vibration level:

- The noise and vibration levels from the 16V 48/60 engine were calculated.

All parameters meet the permissible standards, which ensures comfortable conditions for the crew and minimizes the negative impact on their health.

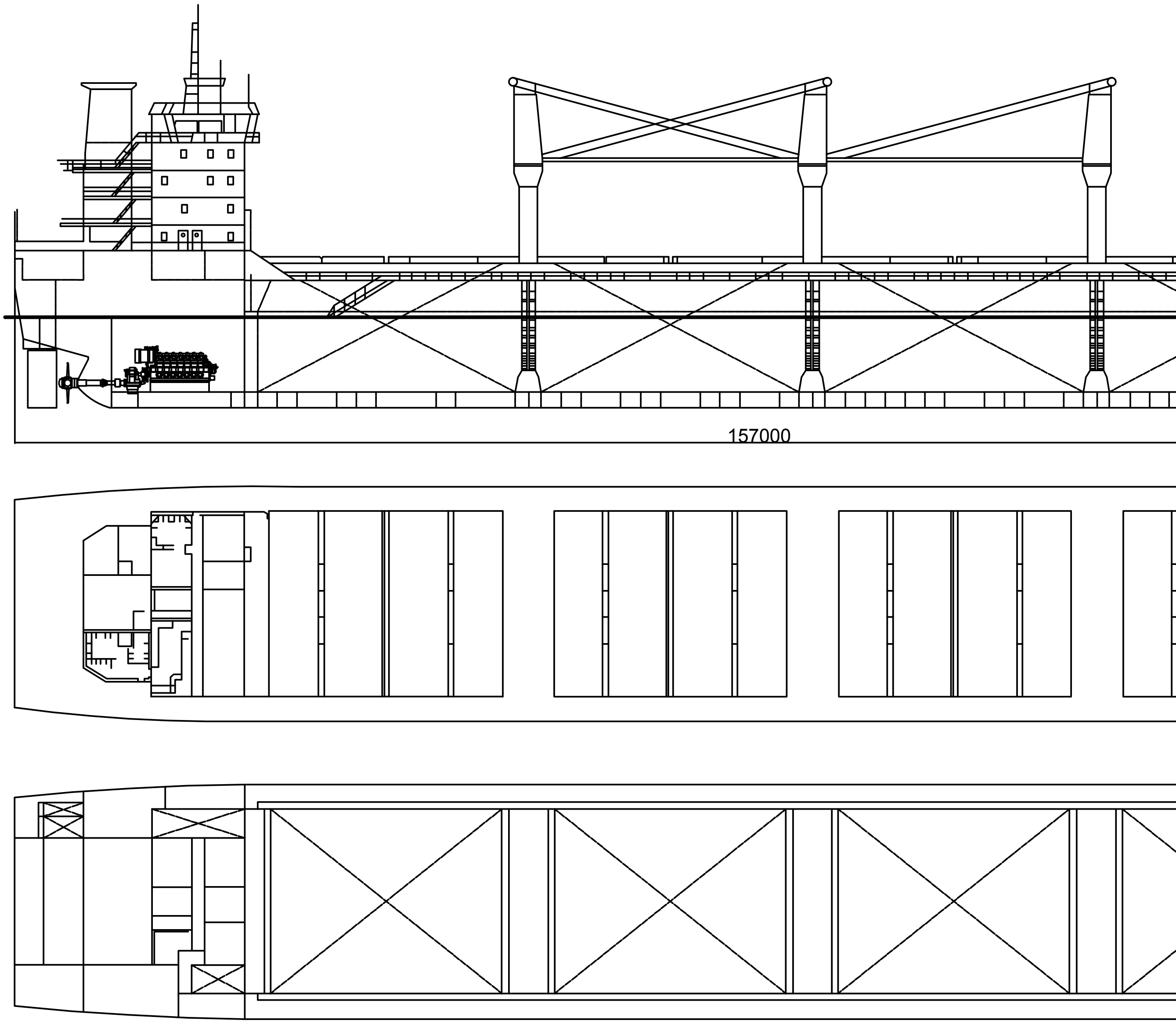
The section emphasizes the importance of integrating occupational safety and environmental protection measures into the ship's operational process, which ensures safe work of the crew and preservation of environmental balance.

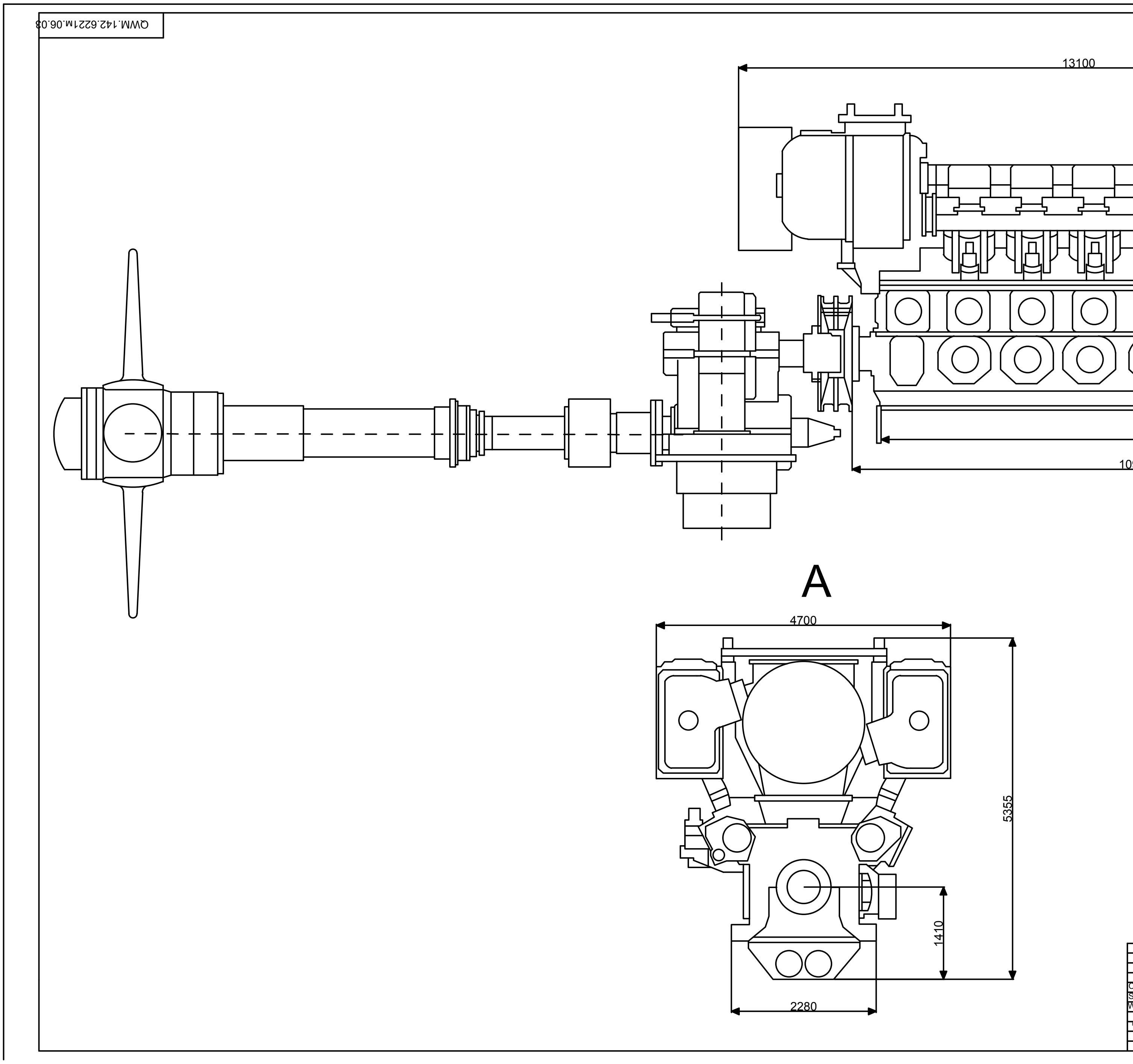
LIST OF REFERENCES

1. Glagolev M.M. Work processes of internal combustion engines - K.: Mashgiz, 1950 - 480 p..
2. Simson A.E. Gas turbine supercharging of diesel engines. - Moscow: Mashinostroenie, 1964 - 248 p.
3. Krutov V.I. Automatic regulation of internal combustion engines - 5th ed. - Moscow: Mashinostroenie, 1989 - 414 p.
4. Pelepeichenko V.I. Investigation of the influence of the type of supercharging system on the effective performance of an eight-cylinder V-shaped transport engine: PhD thesis. - Kharkiv: KhIT, 1981 - 23 p.
5. Internal combustion engines. Theory [text]: Textbook / V.G. Dyachenko; Edited by A.P. Marchenko - Kharkiv: NTU “KhPI”, 2008. - 488 p.
6. Marchenko AP, Pylov VO, Shekhovtsov AF Internal combustion engines: a series of textbooks in 6 vols. T. 4. Fundamentals of CAD of internal combustion engines / edited by Prof. A. P. Marchenko and Honored Worker of Science of Ukraine Prof. A. F. Shekhovtsov. Kharkiv: Prapor, 2004. - 336 p.
7. Minchev D.S., Nahirny A.V. Information bases of computer-aided design systems for internal combustion engines: planning and processing of experiment results: a textbook. Mykolaiv: NUK, 2012. 100 pp.
8. Nalyvaiko V. S. Characteristics of internal combustion engines and consumers [text] / V. S. Nalyvaiko, S. G. Tkachenko, V. G. Khomenko - Mykolaiv: NUK, 2011. 95 p.
9. Mytrofanov O. S. Fundamentals of operation, maintenance and repair of internal combustion engines: a textbook / O. S. Mytrofanov, A. Y. Proskurin - Mykolaiv: Torubara V. V. Publisher, 2018. 151 p.

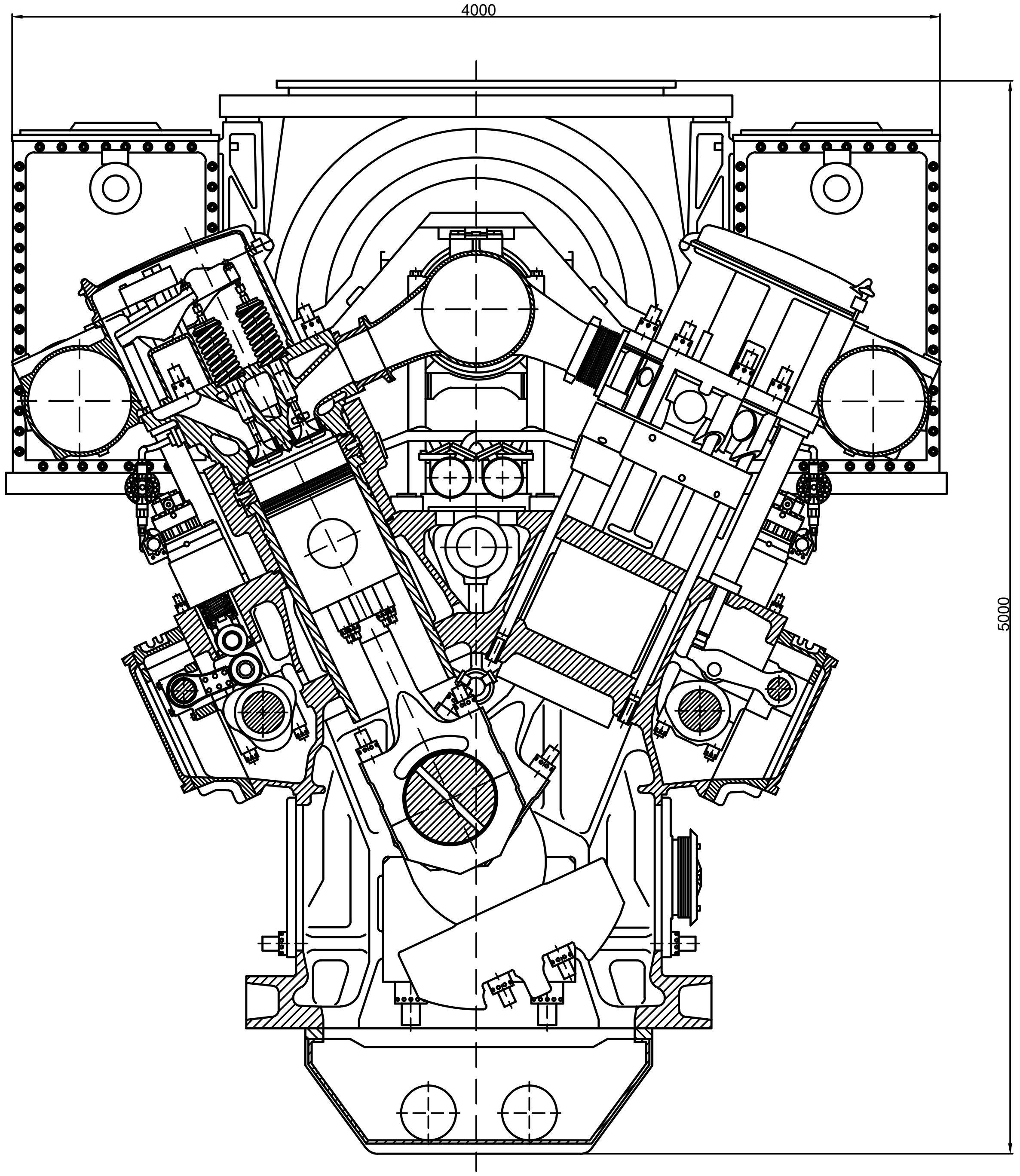
10. Eroshchenkov S.A., Krushedolsky O.G., Sharoyko N.A. Analysis of the distribution of indicator work and fuel consumption in a diesel locomotive diesel engine D50 during its acceleration // Interuniversity collection of scientific works: KhardAZT, 1998. - Issue 32. - P. 55-62.
11. Woodyard D, editor. Pounder's Marine Diesel Engines. 8th Edition. Butterworth-Heinemann, 2003. 912 pp.
12. Anistratenko V.O., Fedorov V.G. / Mathematical planning of experiments in agriculture: Study guide.-K. Higher school, 1993.-375 pp.
13. Volodarsky E.T., Malinovsky B.N., Tuz Y.M. / Planning and organization of the measurement experiment: Higher school, Main edition, 1987. 280 p.
14. Tkachuk K.N., Khalimovskyi M.O., Zatsarnyi V.V. et al. Fundamentals of labor protection: Textbook - 2nd ed. supplemented and revised - K.: Osnova, 2006. - 444 p.
15. Zhydetskyi V.Ts. Fundamentals of labor protection: Textbook. - 3rd edition, revised and supplemented - Lviv: UAD, 2006. - 336 p.

QWM.142.6221M.06.01




QWM.142.6221M.06.03
13100
A
4700
5355
1410
2280

QWM.142.6221м.06.02



Ch.	sheet	No. doc.	Signature	Date
Student	Zabrodin S. G.			
Work sup.	Mitrofanov O. S.			
Head of the De	Gogorenko O. A.			

QWM.142.6221м.06.03

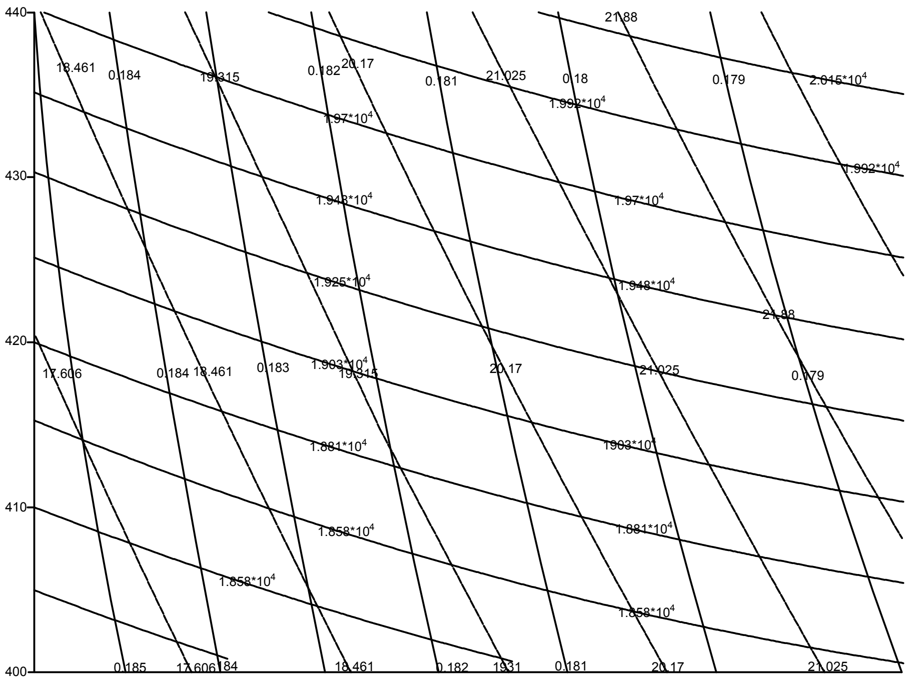
Engine type
M.A.N. 16V 48/60 CR

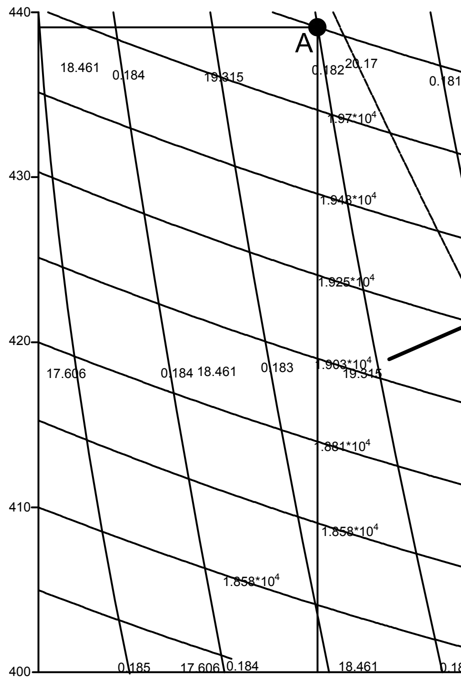
Lit.	Mass	Scale
		1:10

Sheet	Sheets

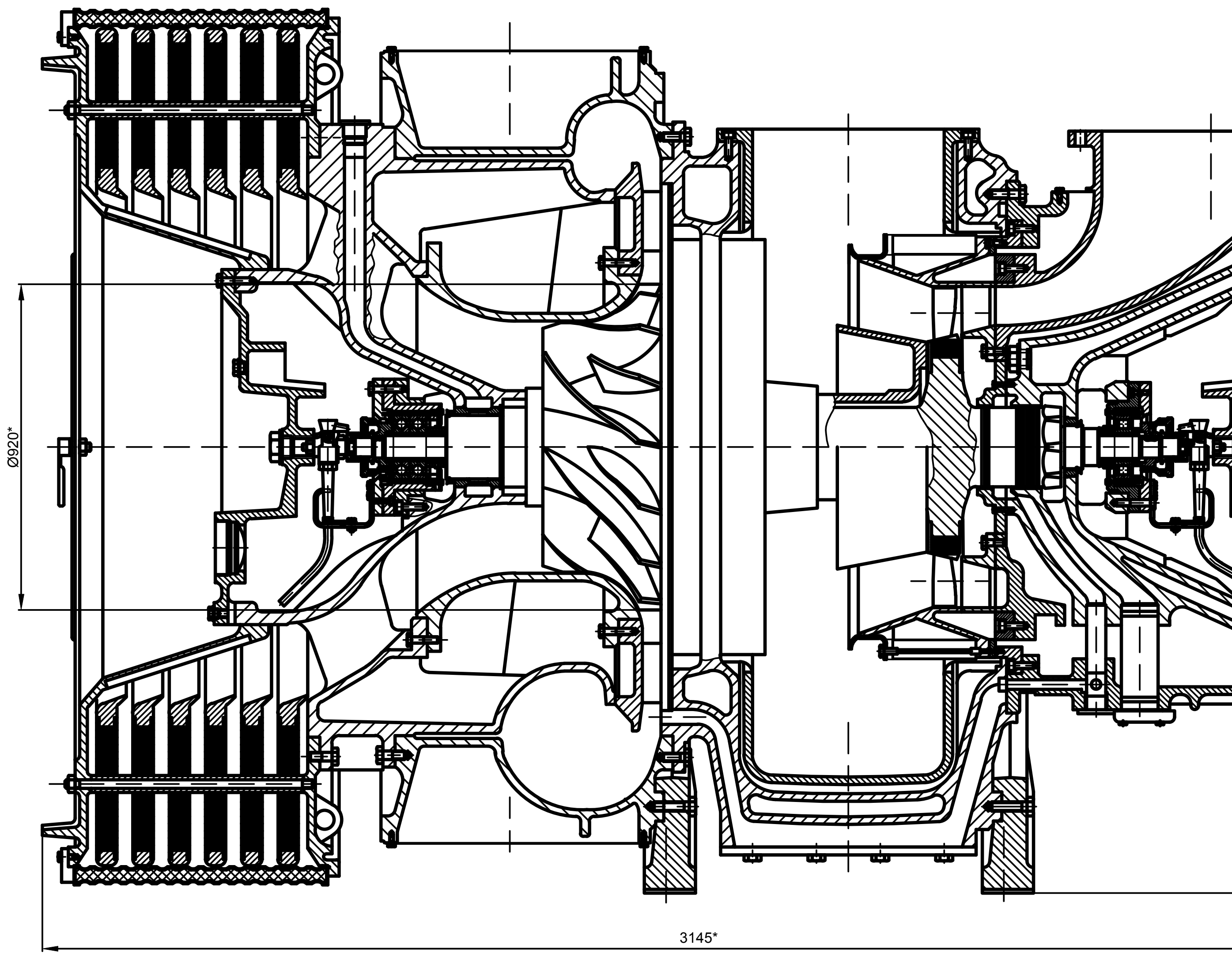
NUOS

QWM.142.6221M.06.05





QWM.142.6221M.06.04



Технічні вимоги

Technical characteristics

3145*

1. The degree of pressure increase Пк = 4,2;
2. Air flow rate through the compressor G = 30,423 kg/s;
3. Adiabatic power of the compressor $N_{ад}$ = 5522,32 kW;
4. Adiabatic efficiency of the compressor $\eta_{ад}$ = 0,83.

1. Dimensions for reference.
2. In the assembled turbocharger, the rotor should rotate freely and stop without jamming when moved alternately all the way to the turbine side and to the compressor side.
3. Water blades of the compressor shall be tested with water at a pressure of 0.15 MPa for 15 minutes. No pressure drop is allowed.
4. The compressor must be installed with the rotor positioned horizontally (±5° from the horizontal).
5. During storage and transportation, all gas, air and oil supply openings must be closed with special technological covers.
6. Exterior painting shall be performed with PL-3 primer in three layers.
7. Mark the year of manufacture with the font Pr3 of DSTU 26020-80.
8. Mark with the font 3...5.
9. It is allowed to store the compressor in any type of premises that exclude direct exposure to atmospheric precipitation in the package.

QWM.142.6221M.06.04