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

Department of Internal Combustion Engines, Installations and Technical Operation

"Admitted to the defence"
Head of the department
Oleksiy GOGORENKO
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« ___ » __________ 2025 р.

QUALIFICATION WORK

MODERNIZATION OF THE LUBRICATION SYSTEM OF A LOCOMOTIVE ENGINE

Speciality 142 – Power engineering

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

Head of work Boris TYMOSHEVSKY

Education applicant Dmytro SHALYA

Mykolaiv 2025

National University of Shipbuilding
Admiral Makarov

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

APPROVED
Head of the Department *ICE, I and TO*
Oleksiy GOGORENKO
«___» ______________ 20___ year

TASKS
FOR A QUALIFYING JOB FOR AN EDUCATION APPLICANT

Dmitry Alexandrovich Shalya
(surname, first name, patronymic)

1. Work topic ***Modernization of the Lubrication System of a Locomotive Engine***

2. Head of work *D., prof. B.G. Timoshevsky*,
(surname, name, patronymic, academic degree, academic title)
approved by the order of the higher education institution from "___" _____ 20___ year № ___
3. Deadline for submitting a job application ________________
4. Initial data for the work *engine power Ne = 2800 kW, rotational speed n = 850 rpm.*

5. Contents of the explanatory memorandum (list of questions to be developed)
1. Application of the research object in a mainline locomotive. 2. Calculation of the operating cycle for nominal operation of a diesel engine. 3. Modernization of the oil system of an internal combustion engine with the development of a new heat exchange surface for the oil cooler. 4. Economic justification for the modernization of the oil system. 5. Analysis and development of occupational health and safety and environmental protection measures.

6. List of graphic material (with an exact indication of the required drawings)
1. The TEP70 diesel locomotive. 2. Dimension drawing of a diesel generator with a 10DN20.7/2x25.4 engine. 3. The diesel locomotive engine 10DN20,7/2x25,4. 4 Engine 10DN20.7/2x25.4 cooling system. 5. Water-oil cooler.

7. Work section consultants

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

8. Date of issue of the task ____________________

CALENDAR PLAN

№ s/n	Name of the work stages	Timeframe for completion of work stages	Note
1.	*1. Application of the research object in a mainline locomotive.*		
2.	*Calculation of the operating cycle for nominal operation of a diesel*		
	engine.		
3.	*Modernization of the oil system of an internal combustion engine with*		
	the development of a new heat exchange surface for the oil cooler.		
4.	*Economic justification for the modernization of the oil system.*		
5.	*Analysis and development of occupational health and safety and*		
	environmental protection measures.		

Education applicant ________ (signature) *Dmitry SHALYA* (surname and initials)

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

АНОТАЦІЯ

Кваліфікаційна робота на тему: «***Модернізація масляної системи тепловозного двигуна»*** присвячена модернізації конструкції охолоджувача масла системи змащення двотактного дизельного двигуна 10DN20,7/2x25,4 магістрального локомотиву.

Представлено загальний устрій та особливості роботи магістрального локомотиву. Розглянуто основні характеристики роботи та конструкцію двотактного дизельного двигуна 10DN20,7/2x25,4. Виконано розрахунок робочого циклу для номінального режиму роботи.

Розглянуто можливі шляхи оптимізації теплообмінних апаратів системи охолодження двигунів внутрішнього згоряння. Розраховано геометричні параметри альтернативної поверхні теплообміну (отримано оптимальні геометричні параметри) у вигляді шахового пучка труб, оребрених накаткою.

Виконано розрахунок економічної доцільності впровадження альтернативної поверхні теплообміну, виконаної на базі оребреної трубки з накатним (стрічковим) оребренням, для системи охолоджування масла двигуна 10DN20.7/2×25.4, який застосовується в якості силової установка на тепловозі ТЕП70.

Розроблено загальні заходи з техніки безпеки для персоналу машинного відділення магістрального локомотиву ТЕР70, а також виконано аналіз шкідливого впливу на навколишнє середовище.

Кваліфікаційна робота виконана англійською мовою та містить 62 аркушів розрахунково-пояснювальної записки. Крім цього робота включає п'яти креслень формату А1.

Ключові слова: двотактний дизельний двигун; робочий цикл; масляна система; охолоджувач масла; поверхня теплообміну.

ABSTRACTS

Qualification work on the topic: ***“ Modenization of the Lubrication System of a Locomotive engine”*** is devoted to the modernization of the design of the oil cooler of the lubrication system of a two-stroke diesel engine 10DN20.7/2x25.4 of a mainline locomotive.

The general structure and features of the mainline locomotive are presented. The main characteristics of the operation and design of the 10DN20.7/2x25.4 two-stroke diesel engine are considered. The working cycle for the nominal operating mode is calculated.

Possible ways to optimize the heat exchange devices of the cooling system of internal combustion engines are considered. The geometric parameters of an alternative heat exchange surface (optimal geometric parameters obtained) in the form of a staggered bundle of roll-fin tubes are calculated.

The economic feasibility of introducing an alternative heat exchange surface based on a finned tube with rolled (strip) fins for the 10DN20.7/2×25.4 engine oil cooling system used as a power unit on the TEP70 diesel locomotive has been calculated.

General safety measures for the personnel of the engine room of the TEP70 mainline locomotive have been developed, and an analysis of the harmful impact on the environment has been performed.

The thesis is written in English and contains 62 pages of calculations and explanatory notes. In addition, the thesis includes five A1 format drawings.

Key words: two-stroke diesel engine; operating cycle; oil system; oil cooler; heat exchange surface.

CONTENT

INTRODUCTION

Modern mainline diesel locomotives are complex technical systems that ensure the reliable and efficient operation of rail transport. One of the key elements of their power plant is the diesel engine, which determines not only the power and efficiency of the locomotive, but also the environmental performance of its operation. The efficiency of a diesel engine is largely determined by the condition and sophistication of the cooling system, the main element of which is heat exchangers.

During the operation of diesel locomotives, heat exchangers are exposed to high thermal and mechanical loads, contamination, vibrations, and other factors, which leads to a decrease in their heat transfer, an increase in the engine temperature, and, as a result, increased wear on its components. Under these conditions, improving the efficiency of heat exchangers becomes one of the most important tasks in the modernization and maintenance of diesel locomotives.

The development and implementation of effective heat exchange systems can reduce specific fuel consumption, increase engine life, and improve the overall reliability of the locomotive. Increased efficiency can be achieved by improving the design of heat exchange equipment, using new materials, improving heat exchange organization, and implementing modern diagnostic and cleaning methods.

PART 1

DESCRIPTION OF THE 10DN20,7/2x25,4 ENGINE DESIGN

1.1. Application of the engine 10DN20,7/2x25,4 on the TEP70 diesel locomotive

The single-section TEP70 diesel locomotive with alternating-to-direct current electric transmission (Fig. 1.1) is designed for passenger train service and is equipped with two driver's cabs. Each cab features a headlight mounted below the front windshield.

The control console, integrating all monitoring and measuring instruments into a single unit, extends across the full width of the cab beneath the windshield. A locomotive signal light is mounted in the upper part of the front window. On the assistant driver's side of the console are several toggle switches, vigilance and horn buttons. The radio telephone is located closer to the driver's position. The speed indicator is mounted in the front right corner of the cab for clear visibility by both the driver and the assistant. A special safety key with a limit switch for emergency shutdown of the locomotive is installed in the center section of the control console.

The engine compartment is separated from the front cab by a high-voltage chamber and from the rear cab by a storage cabinet for crew equipment and personal items. As a result, the vestibules are formed by the rear walls of the cabs, the high-voltage chamber, and the storage cabinet. All equipment in the engine compartment is mounted on the locomotive frame and secured to the roof module.

The power unit, comprising a diesel engine and a traction generator, is positioned centrally on the locomotive atop the underframe. The two-stroke diesel engine features a gas-turbine supercharger with intercooling for both the intake air and the exhaust manifolds. Mounted on the synchronous six-phase AC generator with independent excitation and forced cooling are the starter-generator and the exciter, which receive torque through the diesel's distribution gearbox. The locomotive's underframe rests on rubber shock absorbers. The diesel engine's lubrication system components—including the oil cooler, coarse filters, centrifugal fine filters, and associated piping—are installed directly on the engine and its underframe.

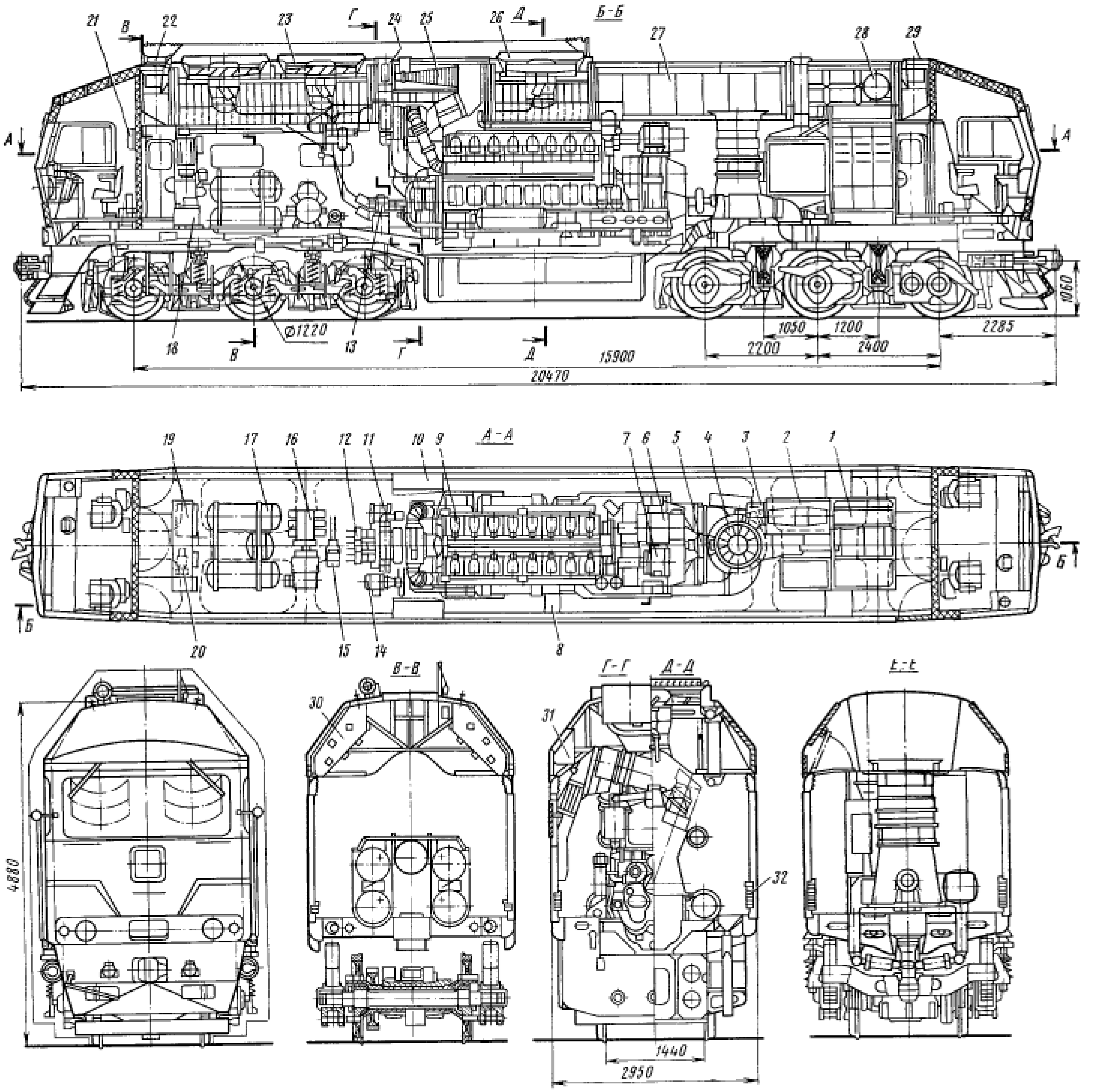

Figure 1.1 – Diesel locomotive TEP70:

1 – High-voltage compartment; 2 – Rectifier unit; 3 – Main generator excitation unit 4 – Axial fan of the centralized air supply system; 5 – Shell coupling; 6 – Starter-generator; 7 – Exciter; 8 – Sanitary room; 9 – Diesel generator; 10 – Diesel air cleaner 11 – Fuel preheater; 12 – Hydraulic pumps; 13 – Hydraulic pump reducer; 14 – Oil transfer pump; 15 – Fuel transfer pump; 16 – Brake system compressor; 17 – Air reservoirs; 18 – Power supply unit for the electro-pneumatic brake system; 19 – Clothing and utensil cabinet; 20 – Air distributor; 21 – Fire extinguisher; 22 – Horn; 23 – Double cooling unit; 24 – Water tank; 25 – Silencer; 26 – Single cooling unit; 27 – Air supply system filter block; 28 – Fire protection system reservoir; 29 – Roof hatch; 30 – Sandboxes; 31 – Oil system tank; 32 – Power cables

On the side of the traction generator, an axial fan of the centralized air-supply system is installed, which transmits torque from the generator shaft through a flexible shell-type coupling and a gearbox.

He removable block-type roof of the locomotive body (Fig. 1.2) consists of the following blocks and compartments: two cooling system units, a filter unit for the centralized air-supply system, an exhaust silencer unit at the diesel outlet, two cab roof covers, and a compartment above the high-voltage chamber, which accommodates the installation of the rheostatic brake unit.

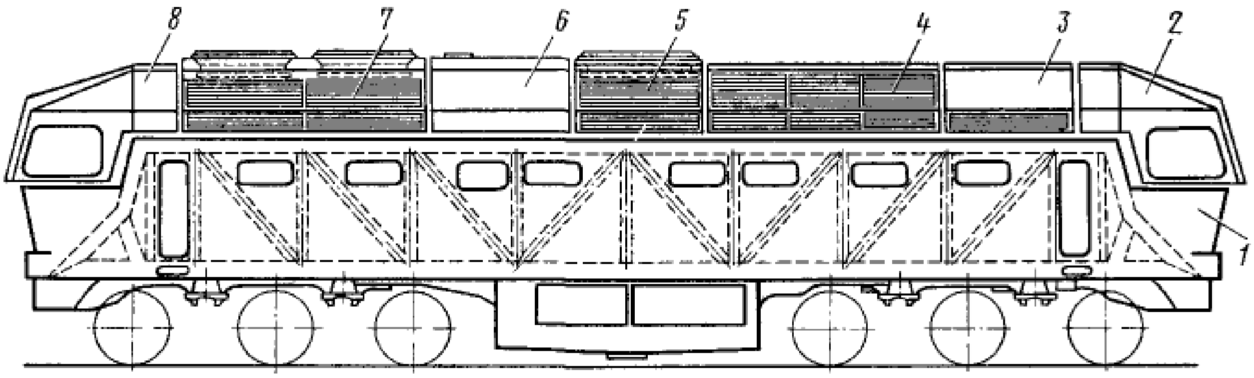

Figure 1.2 – Design scheme of a locomotive body with a removable block-type roof: *1* – load-bearing body frame; *2* – cab roof blocks; *3* – roof above the high-voltage compartment; *4* – centralized air-supply filter block; *5,7* – single and double cooling system blocks; *6* – exhaust silencer block

The modular arrangement of components and assemblies allows for their manufacture and maintenance in specialized workshops, simplifying the assembly and repair processes of the locomotive.

The axial fan draws air from both sides of the locomotive through the air intake filter block, which houses the fan’s inlet manifold. The filter block, composed of individual filter cassettes, is mounted on a special frame integrated into the removable roof section. The cassette mounting system allows for quick installation and removal through hatches located beneath the filter block above both internal walkways of the locomotive. If necessary, airflow for cooling the electrical machines can be directed from the diesel compartment by removing the hatches used for cassette maintenance.

The main ducts of the centralized air-supply system are welded to the locomotive frame, with air supplied via separate pipes to all consumers.

Air for the diesel engine passes through a two-stage air cleaner. The first stage is a continuous-duty filter, implemented as a rotating wheel filled with a set of corrugated screens, while the second stage is stationary and consists of a polyurethane foam packing.

The cooling system for diesel water, oil, and intake air is a dual-circuit design. The first circuit cools the engine water, while the second circuit handles cooling of the intake air in the intercooler and the engine oil in the heat exchanger. The cooling system is integrated into two roof-mounted blocks: a single block above the diesel engine and a double block near the rear cab vestibule. The coolers are constructed from flat-tube sections with a fin spacing of 2.3 mm, arranged in a single row. Twenty-two radiator sections of the first circuit are installed in a single cooler block, while forty sections of the second circuit are located in a double cooler block. A distinctive feature of the cooling system is the sequential connection of the section groups, which increases the water flow rate and enhances the heat transfer coefficient. In the double block, there is a shortened section specifically for cooling the hydraulic drive oil.

Due to the roof-mounted configuration of the cooler blocks, both circuits use shortened water-air sections with an effective length of 710 mm. Air cooling of the water is provided by three axial fans with a wheel diameter of 1400 mm, driven by a hydrostatic system. The hydraulic motors are powered by oil pressure generated by pumps mounted on the gearbox housing, which is installed on the locomotive frame at the pump end of the diesel engine. The operating mode of the hydraulic motors and pumps is regulated by a thermostatic controller, which automatically adjusts the fan speed and maintains the desired temperature range of the engine coolant and oil.

The rectifier unit and high-voltage compartment are centrally located in the body near the traction generator. The high-voltage compartment is mounted on the locomotive frame and divided into three sections: the power section with high-voltage devices (reverser, train contactors, field weakening contactors), a low-voltage section with relays and control units, and a section with regulating resistors. This layout ensures convenient access to any device for maintenance and adjustments.

The installed electrical equipment transmits power and torque from the diesel shaft to the driving wheels while automatically regulating traction force and speed. It also provides engine protection and locomotive shutdown during emergency conditions. Additionally, the electrical system supports diesel starting, operation of the brake compressor, fuel and oil pumps, cabin windshield heating, and other auxiliary functions.

The locomotive is equipped with automatic train signaling with an auto-stop feature, an electro-pneumatic brake, a fire suppression system with automatic signaling, and a radio station. To enhance operational reliability, the electrical system is designed to quickly detect inoperative devices in the diesel starting circuits and to engage the load efficiently.

1.2. Description of the design of the diesel locomotive engine 10DN20,7/2x25,4

The 10D20.7/2×25.4 and 10DN20.7/2×25.4 diesel engines are designed to drive stationary and marine generators, as well as traction generators on locomotives operating with DC–AC conversion systems. The 10GD20.7/2×25.4 gas engine is intended for driving stationary AC generators. The engine, generator, and exciter are mounted on a common frame and are supplied as a complete assembly.

The overall layout of the diesel-generator set equipped with the 10DN20.7/2×25.4 (10D100) engine is shown in Figure 1.3, and the cross-sectional view of the engine is presented in Figure 1.4.

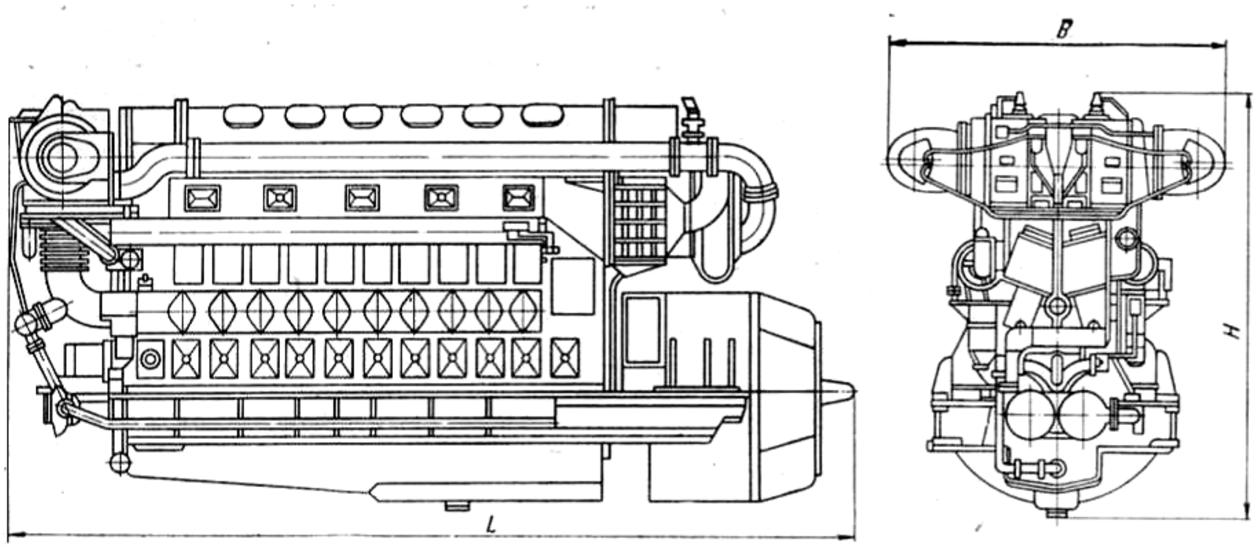

Figure 1.3 – General layout of the diesel-generator with the 10DN20.7/2×25.4 engine

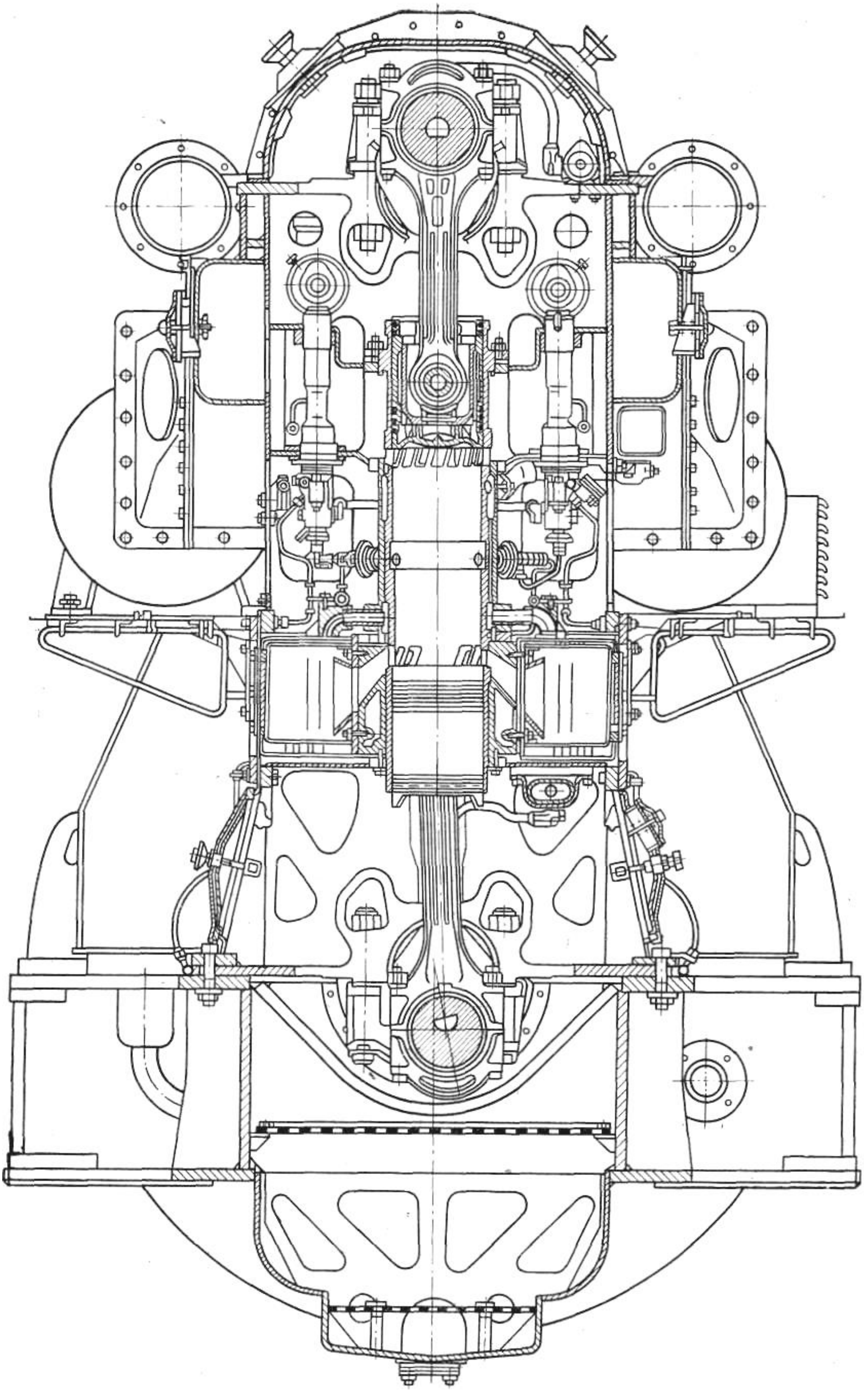

Figure 1.4 – General view of the diesel locomotive engine 10DN20,7/2x25,4

This engine features an in-line configuration with opposed-piston cylinders and two crankshafts. The cylinder block is a welded steel structure. The main bearing supports are welded to the vertical plates of the block, and the bearing shells are made of bronze with a babbitt lining. The top of the block is covered with a removable lid equipped with inspection hatches.

In locomotive and stationary versions, a sub-engine frame is attached to the lower part of the block, forming an oil sump. In marine versions, the block is fitted with an oil pan, from which the lubricating oil drains into a separate tank.

The cylinder liner is cast from a special alloyed cast iron. Around its midsection, a steel jacket is shrink-fitted to form a water-cooling cavity, sealed with rubber O-rings.

In the central part of each cylinder, two fuel injectors, an indicator valve, and a starting valve (for air-start engines) are installed.

The piston is a cast-iron component equipped with four compression rings and three oil-control rings, and it is oil-cooled. The floating hollow piston pin is made of case-hardened steel.

The connecting rod, featuring an I-beam cross-section, is forged from alloy steel. The upper end of the rod is fitted with a steel bushing lined with bronze, while the lower end contains bronze bearing shells coated with babbitt. In the 10D100M models, steel–aluminum bearings are used instead. The crankshafts are cast from special high-strength iron; the lower crankshaft is fitted with a pendulum-type vibration damper. The upper and lower crankshafts are connected by a vertical gear drive.

Two camshafts, positioned on both sides of the engine, drive the high-pressure fuel pumps and are mechanically linked to the upper crankshaft.

The governor is a multi-mode centrifugal, isochronous, indirect-action device equipped with a hydraulic servomotor. In locomotive versions, the governor includes an electropneumatic or electrohydraulic remote-control unit that provides 15 operating speed stages within the 270–850 rpm range. In marine versions, both manual and remote control systems are installed. The governor of AC diesel-generator sets also incorporates an adjustable speed-droop mechanism and a stepless electronic remote-speed control system. Each engine is fitted with an overspeed governor for limiting maximum revolutions.

The fuel system consists of a feed pump, a coarse mesh filter, a fine paper filter, high-pressure fuel pumps, injectors, and pipelines. The feed pump is driven by the crankshaft. Two plunger-type high-pressure pumps are installed per cylinder – one on each side of the engine – supplying fuel to two closed-type injectors located in each cylinder.

In gas-engine configurations, the gas-supply system includes gas valves with dosing units and pre-chambers with automatic valves for each cylinder, diaphragm-type automatic pressure regulators with pneumatic control heads for main and starting gas circuits, and a pneumatically controlled starting gas valve.

The ignition system is a dual, contactless, thyristor-based type with battery power supply. It consists of a distributor sensor, two electronic switches, ignition coils, and two spark plugs per cylinder. The system operates from a 12 V battery.

The lubrication system includes a circulating oil pump, a priming (pre-lubrication) pump, coarse and fine filters, a partial-flow centrifugal filter, and an oil cooler. The main oil pump is crankshaft-driven. The coarse filter is a plate-slot type (mesh type in stationary versions). The fine filtration system combines a centrifugal separator (centrifuge) and replaceable paper-element filters connected in parallel.

In marine engines, oil is cooled by seawater; in locomotives – by air; and in stationary engines – by fresh water. The auxiliary oil pump, driven by an electric motor, fills the lubrication system before engine start and circulates oil through the heater. Both marine and stationary engines use thermostatic regulators in their lubrication circuits.

The lubrication systems of diesel-generators 7D100M, 15D100, and 15D100M additionally include an independent electric-motor-driven oil pump and a heat exchanger for preheating oil and maintaining the unit in a “hot standby” condition. This standby system automatically activates after the diesel engine is shut down, ensuring continuous circulation of preheated oil and coolant through the system.

The marine diesel engine cooling system is dual-circuit: fresh water and oil are cooled by seawater. The seawater-cooling subsystem consists of a water pump, heat exchangers for both water and oil, and a thermostatic regulator. In automated diesel-generator units (7D100M, 15D100, 15D100M), the closed-loop cooling system provides forced circulation of fresh water, with automatic temperature regulation of both oil and water.

A rotary blower, driven by the upper crankshaft and mounted at the rear end of the engine, supplies scavenging air to the receiver. Engines equipped with turbocharging use a two-stage system: the first stage employs two parallel TK-34 turbochargers, while the second stage consists of a mechanically driven centrifugal or rotary compressor. After the second stage, air passes through intercoolers.

Locomotive diesel engines are started using storage batteries, utilizing the generator as a starter motor. Marine and automated versions (7D100M, 15D100, 15D100M) are started by compressed air at a pressure of 17.6 bar.

The automatic and remote start system enables starting of a preheated diesel engine (up to 50°C) and performs the following operations: oil and fuel pre-circulation, engine cranking and start, speed acceleration to nominal value, synchronization of generator sets with each other or with the external power grid, and load application. The diesel-generator set is controlled remotely.

The engine is equipped with monitoring instruments – pressure gauges for oil, fuel, and water, thermometers for oil and coolant temperature, and a tachometer.

The alarm and safety system monitors coolant temperature and oil pressure. In diesel-generators 7D100M, 15D100, and 15D100M, the automation system additionally provides protection in case of generator bearing overheating, excessive air temperature in the generator cooling system, crankcase overpressure, and disconnects the generator from the busbars during emergency shutdowns.

1.3. Conclusion to the part

This section examines the general design of the TEP70 mainline diesel locomotive, as well as the features of the 10DN20.7/2X25.4 two-stroke diesel engine used in it. The design, operating principle and structure of the 10DN20.7/2X25.4 two-stroke diesel engine systems are considered. A cross-section of this engine in operation is shown in an A1 format drawing.

PART 2
CALCULATION OF THE WORKING CYCLE OF THE NOMINAL OPERATING MODE OF THE 10DN20.7/2X25.4 TWO-STROKE LOCOMOTIVE ENGINE

2.1. Calculation of the working cycle of the nominal operating mode of a 10DN20.7/2x25.4 two-stroke diesel locomotive engine

In theory, *ideal (calculated) cycles* of heat engines are understood as a specific set of thermodynamic processes that approximately describe and represent the behavior of the actual working cycle. Each known theoretical cycle of a heat engine is based on a series of assumptions and simplifications. The results obtained and the conclusions drawn from the analysis of such a cycle are valid only within the limits of these assumptions and the initial conditions used.

The analysis of a theoretical heat engine cycle can be *single-factor*, meaning that it is possible to determine, by calculation, the influence of each individual factor while keeping all other influencing factors constant. However, when conducting theoretical studies based on real heat engines, it is not always possible to isolate and evaluate the effect of each selected factor independently.

The *thermodynamic theoretical cycles* of heat engines are defined as those that are constructed under a set of fundamental assumptions:

– the processes forming the cycle are reversible;

– the working fluid within the system remains constant in composition;

– the heat capacity of the working fluid does not depend on its state;

– there is no heat exchange between the working fluid and the surrounding environment.

In thermodynamic theoretical cycles of internal combustion engines, heat addition can occur at constant volume (isochoric process), at constant pressure (isobaric process), or as a combination of both—heat added partly at constant volume and partly at constant pressure. The most general form is the *combined heat addition thermodynamic cycle*, commonly known as the *Trinkler–Sabathé cycle*, named after the scientists who proposed it. The graphical representation of the Trinkler–Sabathé cycle is shown in Figure 2.1.

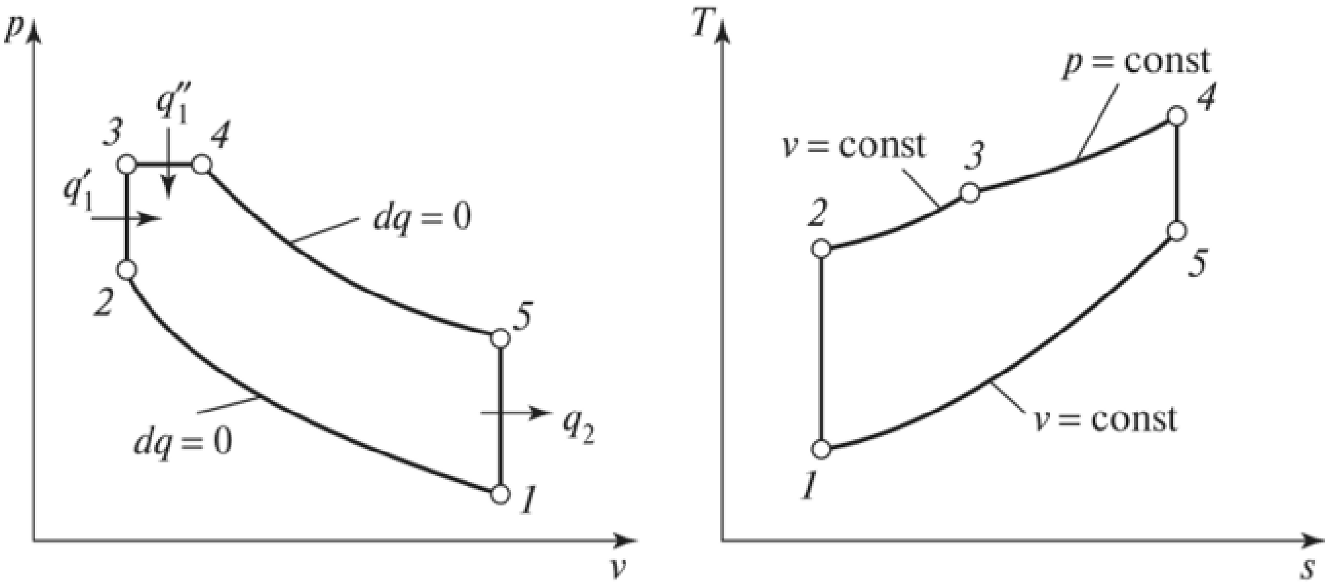

Figure 2.1 – Thermodynamic calculation cycle of internal combustion engines with mixed heat supply by Trinkler–Sabate:

Line 1-2 – adiabatic compression of air in the cylinder;

Line 2-3 – isochoric heat supply to the working body, rapid combustion of some of the fuel supplied in the prechamber or preliminary combustion chamber;

Line 3-4 – isobaric heat supply to the working body, slow combustion of the fuel supplied in the engine cylinder;

Line 3-5 – adiabatic expansion of combustion products, working stroke of the piston;

Line 5-1 – heat removal from the working body – exhaust into the atmosphere

With the advancement of technology particularly in engine design and the evolution of engineering thought, the combination and refinement of various calculation methods for internal combustion engine working processes led to the development of the highly reliable *Grynevetsky–Mazing method*, which became a unified and comprehensive system for engineering analysis.

Professor V. I. Grynevetsky, in his thermal calculation approach, proposed that the temperature of residual gases (T_r) and the cylinder filling coefficient (η_n) should be set in advance. He also introduced formulas for calculating the temperature at the start of compression (T_a) and the residual gas coefficient (γ_r). His follower, E. K. Mazing, suggested a modification: to assume given values for T_r and γ_r, and to determine Ta and ηn by introducing into Grynevetsky's system of equations an additional parameter – the temperature of air heated by the cylinder walls (T'_0).

Today, there exists a wide range of software–analytical systems that differ in capability (number of implemented functions), purpose (analysis, diagnostics, verification, design, etc.), and the fundamental calculation methodology that underpins their logical framework.

Thus, the Grynevetsky-Mazing method, which has undergone continual refinement and enhancement over the past century, serves as the foundation for calculating the performance parameters of the working cycle of the 10DN20.7/2x25.4 two-stroke diesel locomotive engine.

The results and calculations of the working cycle for the two-stroke diesel locomotive engine 10DN20.7/2x25.4 are presented in tabular form (Tables 2.1–2.6).

Table 2.1 – Initial data for calculating the operating cycle parameters of a 10DN20.7/2x25.4 two-stroke diesel engine and calculation results

№ s/n	Parameter	Dimension	Value
1.	Crankshaft speed	n, rpm	850.00
2.	Power	N_e, kW	2762.40
3.	Average effective pressure	P_e, MPa	1.1406
4.	Torque	M_e, Nm	31036.00
5.	Cycle fuel consumption	q_c, g	1.0677
6.	Specific fuel consumption	g_e, kg/(kW)	197.120
7.	Effective efficiency	η_e	0.42972
8.	Average gauge pressure	P_i, MPa	1.3113
9.	Gauge efficiency	η_i	0.49404
10.	Friction pressure	P_{fric}, MPa	0.17073
11.	Mechanical efficiency	η_m	0.8698
Environmental parameters			
12.	Ambient pressure	P_0, bar	1.00
13.	Ambient temperature	T_0, K	288.00
14.	Static pressure behind the turbine	P_{0t}, bar	1.040

15.	Inhibited flow pressure behind the filter	P_{oin}, bar	0.9800
Supercharging and gas exchange			
16.	Pressure upstream of the exhaust manifold	P_k, bar	2.4000
17.	Temperature upstream of the exhaust manifold	T_k, K	315.41
18.	Air flow rate through the ICE cylinders	G_{air}, kg/s	4.5982
19.	Efficiency of the supercharging unit	η_t	0.67274
20.	Average pressure before the turbine	P_t, MPa	0.20904
21.	Average temperature before the turbine	T_t, K	814.5701
22.	Gas flow rate through the ICE cylinders	G_{gas}, kg/s	4.75110
23.	Excess air coefficient	α_{sum}	2.09760
24.	Filling coefficient	η_v	0.74387
25.	Residual gas coefficient	γ_r	0.06714
Inlet manifold			
26.	Average pressure in the intake manifold	P_s, MPa	0.239950
27.	Average temperature in the intake manifold	T_s, K	321.34001
28.	Average wall temperature of the intake manifold	T_{ws}, K	351.35011
29.	Heat transfer coefficient in the intake manifold	α_{ws}, W/(m^2·K)	160.0001
30.	Heat transfer coefficient in the valve channel	α_{wsc}, W/(m^2·K)	143.52101
Exhaust manifold			
31.	Average static pressure of the gas	P_r, MPa	0.2090401
32.	Average static gas temperature	T_r, K	814.56001
33.	Average gas velocity	W_r, m/s	74.855102
34.	Strouhal number	Sh	2.46500111
35.	Average wall temperature of the exhaust manifold	T_{wr}, K	639.210021
36.	Heat transfer coefficient in the exhaust manifold	α_{wr}, W/(m^2·K)	153.551123
37.	Heat transfer coefficient in the valve channel	α_{wcr}, W/(m^2·K)	317.960237
Combustion			
38.	Excess air coefficient	α	2.200000
39.	Maximum combustion pressure	P_z, MPa	105610010
40.	Maximum combustion temperature	T_z, K	1698.7001
41.	Maximum pressure angle	φ_{i_pz}, degrees above UDC	2.0000
42.	Maximum temperature angle	φ_{i_tz}, degrees above UDC	16.0000
43.	Maximum pressure rise rate	$dp/d\varphi_i$,	4.330210

		bar/grad	
Injection			
44.	Maximum injection pressure	$P_{inject\max}$, MPa	88.524012
45.	Average droplet diameter	d_{32}, μm	19.319
46.	Injection advance	θ_{ad}, degrees to UDC	18,000
47.	Duration of fuel injection	φ_{i_in}, degrees	26.334
48.	Ignition delay period	φ_{i_delay}, degrees	2.6194
49.	The proportion of fuel that evaporated during the delay period	*Sig*	0.00412
50.	Combustion duration	φ_z, degrees	69.400
51.	Vortex number (ratio) in the combustion chamber at TDC	H_{TDC}	2.4700
52.	Vortex number in the cylinder at the start of compression	H_{LDC}	1.8589
53.	Maximum vortex velocity in the combustion chamber at radius R=46	W_{swirl}, m/c	13.597
Environmental indicators			
54.	Hartridge smoke emission	Hartridge	2.5770010
55.	Smoke emission on the Bosch scale	Bosch	0.2822400
56.	Absolute light absorption coefficient of gas according to ECE	K, m^{-1}	0.0613600
57.	Particulate matter emission	PM, g/(kW·h)	0.0433300
58.	Carbon dioxide emission	CO_2, g/(kW·h)	620.4201001
59.	NO_x emissions reduced to NO_2	NO_2, g/(kW·h)	3.62690010
60.	Complex of total NO_x and PM emissions	SE	0.6625600
61.	SO_2 emissions	SO_2, g/kW·h	0.000000
Duty cycle parameters at typical points			
62.	Initial compression pressure	P_a, MPa	0.272750010
63.	Compression start temperature	T_a, K	373.0501023
64.	End of compression pressure	P_c, MPa	8.8663001
65.	End of compression temperature	T_c, K	952.06001
66.	Pressure at the beginning of release	P_b, MPa	0.78690100
67.	Temperature at the beginning of the release	T_b, K	1017. 20010
Parameters that determine the working process			
68.	Compression ratio (for oppositely moving pistons with both pistons at TDC)	ε_1	18.600
69.	Compression ratio (actual, taking into account	ε_2	17.548

	shaft jamming)		
70.	Compression ratio taking into account the proportion of lost stroke	ε_3	13.429
71.	Number of nozzle nozzles	$i_{nozzles}$	4.000
72.	Nozzle nozzle diameter	d_{nozzle}, mm	0.37000
73.	Injection duration for a given injection characteristic	φ_{inject}, degrees	25.0000
74.	Start of exhaust (Exhaust crankshaft)	$\varphi_{s.d}$, deg. to the LDC	62.000
75.	End of exhaust (Exhaust crankshaft)	$\varphi_{e.e}$, deg. behind LDC	62.000
76.	Exhaust crankshaft advance angle relative to intake crankshaft	$\varphi_{\text{exh.}}^{\text{crank.}}$, deg.	12.0000
77.	Inlet start (intake crankshaft)	$\varphi_{i.s}$, deg. to TDC	45.000
78.	End of the inlet (intake crankshaft)	$\varphi_{i.e}$, deg. behind LDC	59.000
Cylinder heat transfer parameters			
79.	Average equivalent cycle temperature	T_{avg}, K	1119.7
80.	Average heat transfer coefficient from gas to walls	α_w, Вт/m^2/K	553.53
81.	Average temperature of the piston crown	T_w^{piston}, K	580.00
82.	Average temperature of the bushing combustion surface	$T_w^{bushing}$, K	430.00
83.	Average temperature on the cooling side of the cylinder bushing	$T_w^{cooling}$, K	381.2700
84.	Boiling temperature in the cooling fluid system	T_{boil}, K	391.3700
85.	Average heat transfer coefficient from the cylinder liner wall to the cooling medium	α_w^{cool}, W/(m^2·K)	12291.000
86.	Heat flux into the piston	q_{piston}, J/s	10054.000
87.	Heat flux into the cylinder liner	$q_{cylinder}$, J/s	6242.100
Compressor parameters (high pressure stages)			
88.	Compressor rotor speed	n, min^{-1}	8500.0000
89.	Compressor power	N, kW	506.7201
90.	Adiabatic compressor efficiency	η_{ad}	0.78600
91.	Air flow through the compressor	G_{khp}, kg/s	4.598200
92.	Air flow through the KHP	G_{giv}^{khp}	79.62600
93.	Air flow through the KHP (corrected)	G_{cor}^{khp}, kg/s	4.612600

94.	KHP rotor rotation frequency (reduced)	n_{giv}^{khp}	500.87000
95.	KHP rotor rotation frequency (corrected)	n_{cor}^{khp}, min^{-1}	8646.3000
96.	Degree of pressure increase in the HP compressor	P_{hp}	2.50000
97.	Total pressure at the KHP inlet	P_o^{khp}, MPa	0.098001
98.	Brake temperature at the KHP inlet	T_o^{khp}, K	288.0000
99.	Total pressure after the HP compressor	P_k, MPa	0.2450001
100.	Brake temperature after the HP compressor	T_k, K	397.6500
101.	Thermal efficiency of the HP ONP	η_{Ecool}^{hp}	0.7500100
102.	Cooling water temperature in the HP EPC	T_{cool}^{vt}, K	288.00000
103.	Boost pressure at the KHP	P_k^{khp}, MPa	0.2400001
104.	Boost air temperature at the KHP	T_k^{khp}, K	315.41001
Turbine parameters (high pressure stages)			
105.	Rotor rotation frequency	n, min^{-1}	8500.0000
106.	THP power taking into account mechanical efficiency	N_{thp}, kBt	575.8200
107.	Internal efficiency of the HP turbine	η_t	0.84400
108.	Mechanical efficiency of the HP turbine	η_m	0.9820001
109.	Gas flow through the THP	G_{thp}, kg/s	4.7511012
110.	Gas flow through the THP (reduced)	G_{giv}^{thp}	64.86600
111.	THP rotor rotation speed (reduced)	n_{giv}^{thp},	297.8200
112.	Degree of pressure reduction in the HP turbine	P_{thp}	2.01000
113.	Relative work of the THP	B_{thp}	15.8860007
114.	Total pressure upstream of the HP turbine	P_t^{thp}, MPa	0.20904
115.	Braking temperature at the inlet to the THP	T_t^{thp}, K	814.570100
116.	Back pressure behind the HP turbine	P_o^{thp}, MPa	0.1040001
117.	Gas temperature behind the HP turbine	T_o^{thp}, K	703.780010

During the working cycle in the cylinder of the 10DN20.7/2x25.4 diesel locomotive engine, the pressure and temperature constantly change. The pressure change can be measured using a special sensor (indicator). Such measurements are performed during experimental studies and when testing the engine. Such experimental diagrams are called indicator diagrams. Mathematical modeling also produces an indicator diagram. Indicator diagrams are constructed in different coordinate systems. The most common and informative are diagrams in p-V and p-φ coordinates. The main dependencies are shown in Figs. 2.1–2.3.

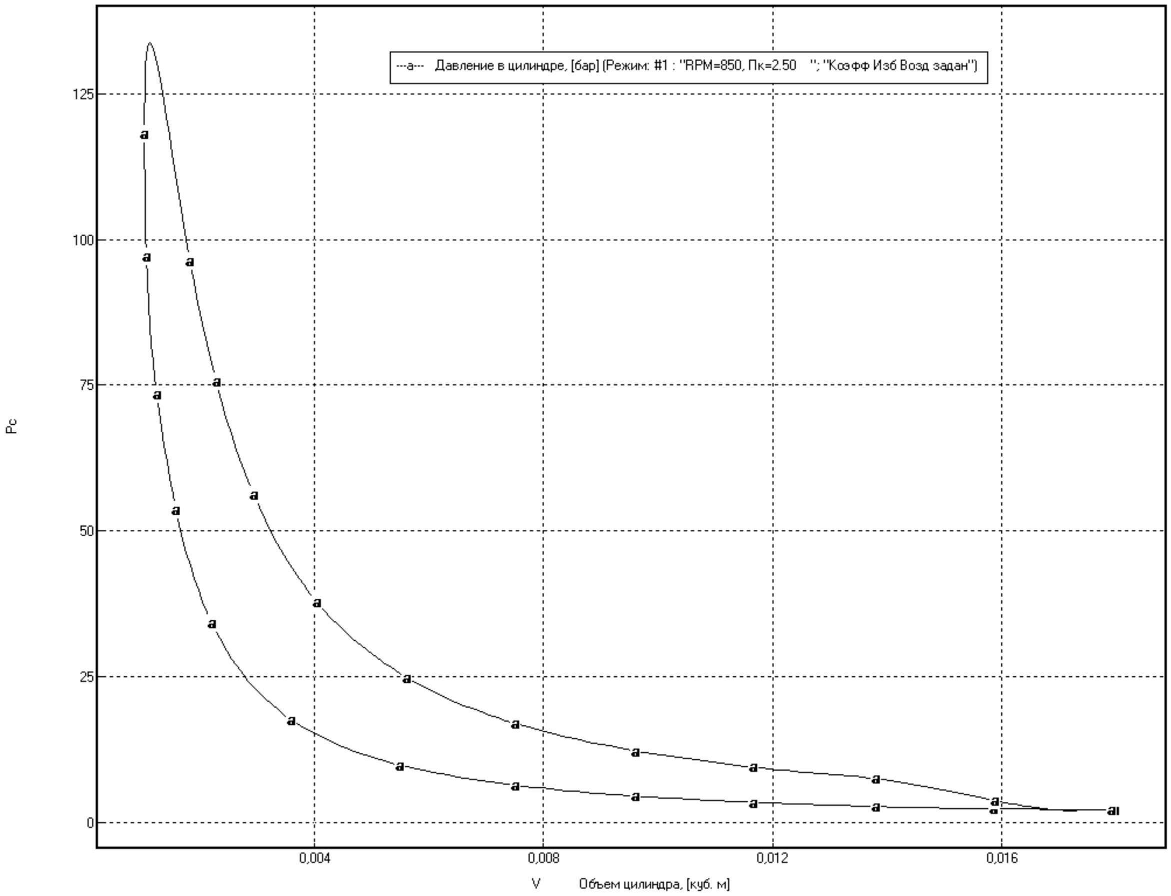

Figure 2.1 – Indicator diagram of a 10DN20.7/2x25.4 diesel locomotive engine

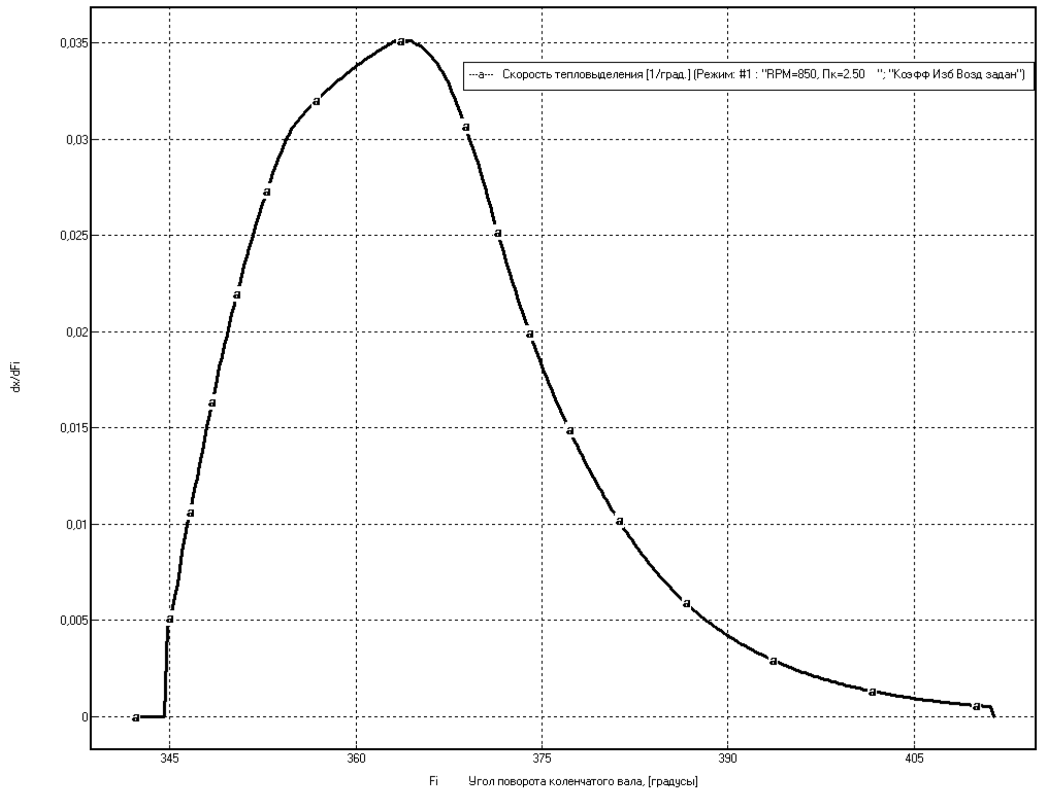

Figure 2.2 – Dependence of heat release rate on crankshaft rotation angle of a 10DN20.7/2x25.4 diesel locomotive engine

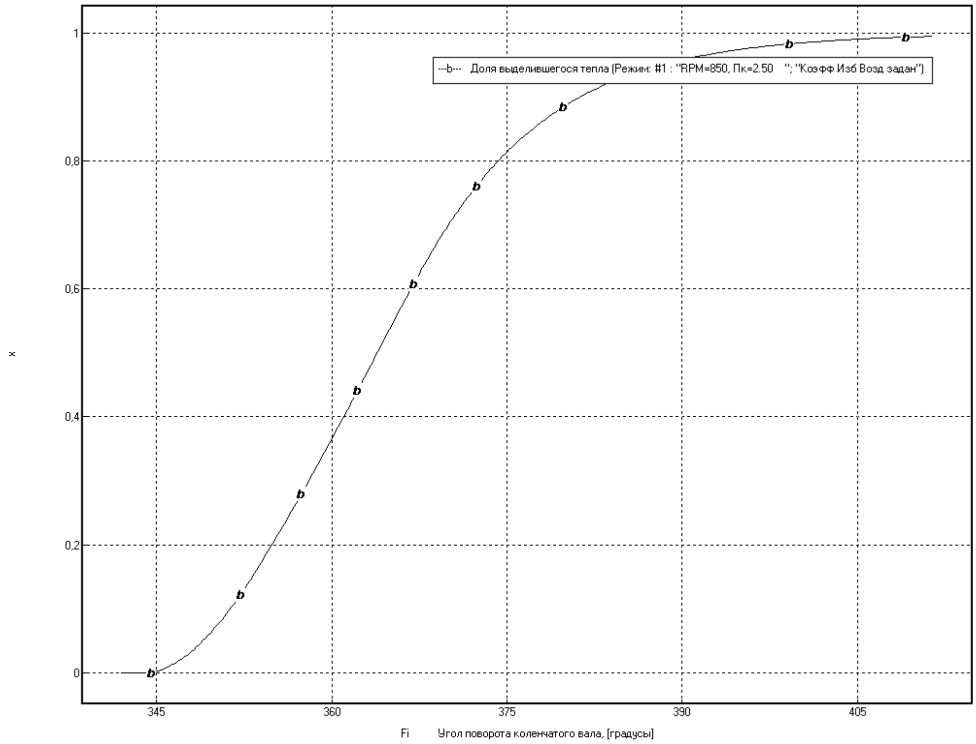

Figure 2.3 – Dependence of the proportion of burned fuel on the crankshaft rotation angle of a 10DN20.7/2x25.4 diesel locomotive engine

2.2. Conclusions to the section

This section contains calculations of the working cycle of a 10DN20.7/2x25.4 two-stroke diesel locomotive engine for nominal operating mode, as well as the corresponding dependencies. The calculation results form the basis for further locomotive engine modernization.

PART 3

MODERNIZATION OF THE OIL SYSTEM OF THE TWO-STROKE LOCOMOTIVE DIESEL ENGINE 10DN20.7/2X25.4

3.1. Optimization of the heat exchange surface of the water-oil cooler of a mainline locomotive TEP70

It is necessary to determine the geometric parameters of a finned tube with knurled (strip) fins (Fig. 3.1), as well as the arrangement parameters of the tubes in a staggered bundle, so as to obtain a tube bundle for a locomotive oil cooler that ensures the required air resistance and the highest possible efficiency under fixed overall bundle dimensions.

The standard oil cooler of the 10DN20.7/2x25.4 engine cooling system is designed as a shell-and-tube water–oil heat exchanger (Fig. 3.2). The shell of this type of heat exchanger is made in the form of a large-diameter tube containing a bundle of small-diameter tubes. The ends of the tubes are fastened in tube sheets.

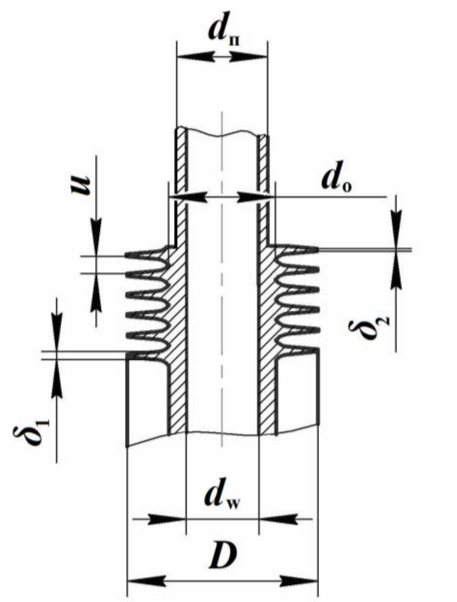

Figure 3.1 – Tube with rolled fins

The oil coolers are based on a staggered tube bundle with knurled fins. For tubes with the same external diameter, the tube pitch in the oil cooler bundle is designed in a specific way (see Fig. 3.3). Although varying these pitches could, in principle, optimize the heat exchange surface, in this case such a change would alter the shell diameter, which is undesirable due to standardization requirements.

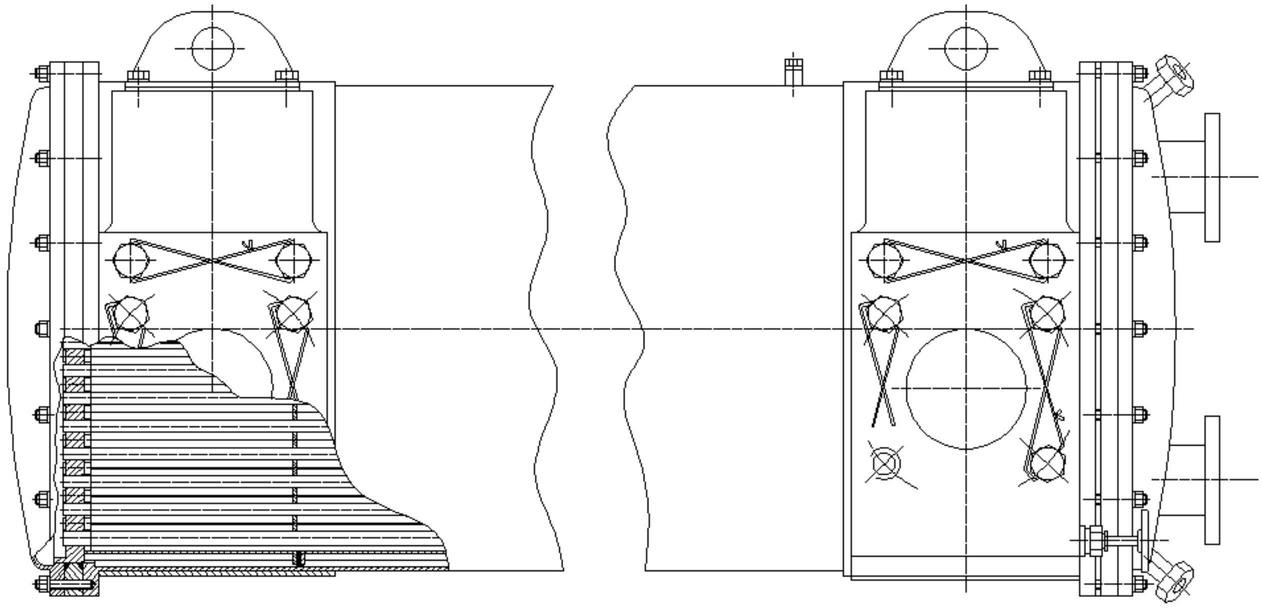

Figure 3.2 – General view of a shell-and-tube oil cooler

Therefore, in designing this heat exchanger, it was decided to introduce only a new type of tube while keeping the tube pitches in the bundles and their overall arrangement on the tube sheet unchanged.

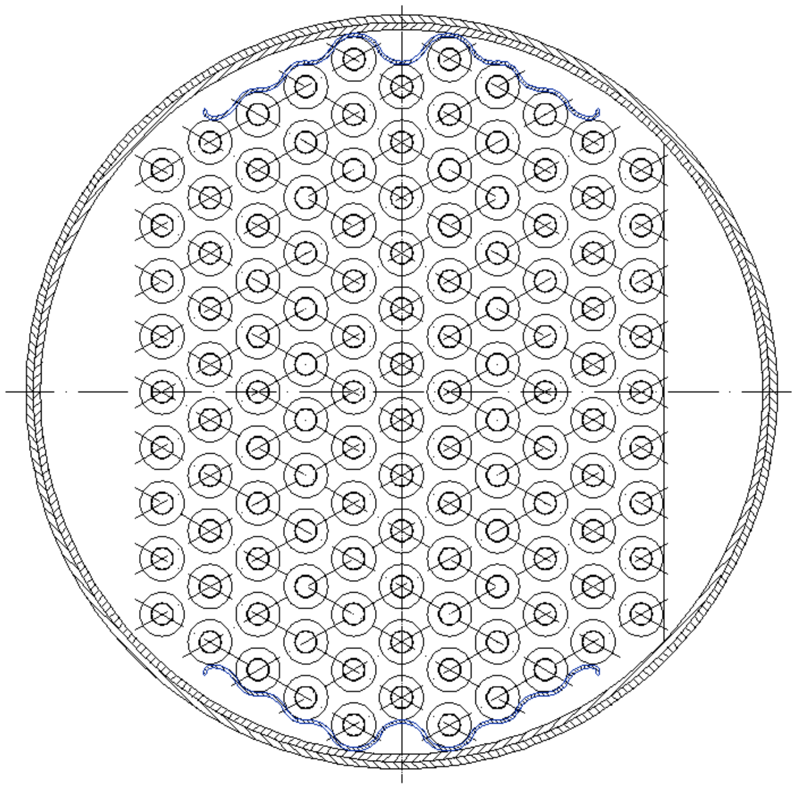

Figure 3.3 – Pipe arrangement diagram in a bundle

The oil cooler operates with a reversible flow of the working fluids. The oil cooler currently used in locomotives is characterized by the following thermal and mass-dimensional parameters:

– oil flow resistance, ΔP_{oil} = 28.75 kPa;

– oil outlet temperature, $t_{oil}{}^2$ = 84.7 °C;

– water flow resistance, ΔP_w = 75.94 kPa;

– heat exchanger efficiency, η_h = 0.197;

– bundle mass, M_b = 275.94 kg.

The oil cooler is built on a heat exchange surface with geometric characteristics presented in Table 3.1.

Table 3.1 – Geometric dimensions of the prototype heat exchange surface oil cooler

№ s/n	Parameter	Dimension	Value
1.	Outer diameter of fins	D, m	0,0245
2.	Diameter of supporting tube	d_0, m	0,0156
3.	Inner diameter of tube	d_w, m	0,01
4.	Pitch between fins	u, m	0,003
5.	Fin thickness at base	δ_1, m	0,0007
6.	Fin thickness at top	δ_2, m	0,0003
7.	Pitch between tubes in transverse row	S_1, m	0,027
8.	Pitch between transverse rows of tubes	S_2, m	0,0234

Conditions for Performing the Assigned Task

1. To ensure competitiveness, the alternative oil cooler must, at minimum, have a lower bundle mass, higher thermal efficiency, and the same hydraulic resistance on the water side.

2. The coolers must be manufactured using tubes with an internal diameter of 10 mm. The following dimensional limits are established for the tubes: fin thickness at the base $\delta_1 \geq 0.35$ mm, fin thickness at the tip $\delta_2 \geq 0.15$ mm. The fin pitch u should be within the range of 1.7–3.0 mm. The external finned diameter D must be between 20.5 and 24.5 mm.

3.2. Possible approaches to the optimization of heat exchangers

Before proceeding with the practical part of this work, let us consider existing approaches to optimizing heat exchangers and their components in order to select the most appropriate one.

The problem of optimizing heat transfer surfaces remains unresolved in the field of heat exchanger design, as there is still no unified approach to its solution. Both in international and domestic research, a large number of studies have been devoted to this topic, each presenting different methods and approaches, none of which fully capture the complexity and multifaceted nature of the problem.

Some researchers base optimization on the amount of heat transferred in the heat exchanger (HX), others on its mass, volume, or various other parameters. However, the most effective approach is optimization based simultaneously on several key performance criteria.

A wide range of performance indicators and methods for optimizing and comparing heat exchangers has been proposed. In this work, we will focus on the most common and widely used among them.

M. V. Kirpichev proposed using the so-called energy coefficient to evaluate the effectiveness of a heat transfer surface.

$$E = \frac{Q}{AN}, \quad (1)$$

which represents the ratio of the amount of heat transferred, Q, to the work $A \cdot N$ expended in overcoming the resistance to coolant flow, where A = 1/427 kcal/kg·m is the thermal equivalent of mechanical energy. Somewhat later, this coefficient was also recommended for use by Glaser.

It is evident that the higher the value of the energy coefficient E, the more efficient the heat transfer surface is in terms of energy consumption. At the same time, this coefficient does not take into account the geometric dimensions of the surface, its weight, or similar characteristics.

From the structure of equation (1), it follows that when optimizing different heat exchange surfaces, the amount of heat transferred Q must be determined based on the temperature head – that is, the mean temperature difference between the coolant and the surface. As is well known, this temperature difference depends on the flow arrangement within the heat exchanger and reflects the specific operating conditions of the heat exchange unit as a whole.

Although the efficiency indicator proposed by M. V. Kirpichev provides a qualitative energy-based assessment of the heat transfer process, it does not allow for a quantitative comparison of processes occurring at different temperature levels.

To address this limitation, Antufiev proposed evaluating the thermal performance of a heat exchange surface using the heat transfer coefficient α, while relating the energy expenditure required to overcome the flow resistance to a unit area of the heat transfer surface. In this case, the energy coefficient represents the amount of heat transferred per unit time through a unit area for a temperature difference of 1 °C per unit of energy consumed for moving the coolant, that is

$$E = \frac{\alpha}{AN_0}, \tag{2}$$

where **α** is the heat transfer coefficient, kcal/(m²·h·°C); and $A{\cdot}N_0$ is the amount of energy (expressed in thermal units, kcal/m²·h) expended per hour to move the coolant over 1 m² of surface area.

To optimize and comparatively evaluate two heat transfer surfaces, it is necessary to compare their *E* values over the relevant range of flow velocities: the surface with the higher *E* value is considered more energy-efficient. Therefore, the task reduces to finding the maximum of the function *E*.

When optimization is performed using the energy coefficient *E*, the analysis typically considers heat transfer only on one side of the surface. The conclusions obtained are valid under the assumption that the heat transfer coefficient on the opposite side is infinitely large – that is, the thermal resistance on that side is negligible. At the same time, factors such as the relative dimensions and weight of the heat transfer surfaces are not taken into account.

However, analyzing and optimizing heat exchangers solely on the basis of thermodynamic criteria – according to the methods described above and others – does not always reflect the actual technical and economic efficiency of the device. In many cases, energy costs represent only a small portion of the total process expenses. Therefore, thermodynamic analysis alone cannot yield accurate conclusions about ways to improve process efficiency.

The primary criterion determining the efficiency of any process, including heat exchange, is techno-economic efficiency. The most effective design, from this standpoint, is the one that ensures the desired process performance with the minimum expenditure of materials and labor.

A large number of different criteria have been proposed for the optimization of heat exchangers. In addition to the thermodynamic criterion Q/AN introduced by M. V. Kirpichev and discussed above, various other performance indicators have also been repeatedly suggested and applied, such as Q/F, Q/G, Q/v, Q/w, and others. Here: Q – amount of heat transferred; G – mass of the heat exchanger, kg; v – volume of the heat exchanger, m^3; w – mass of the heat transfer fluid, kg.

N. I. Gelperin and S. D. Zaitsev proposed evaluating the operational feasibility of a heat exchanger using the following function:

$$M = aQ(t) - bF(t)$$

where a – cost of 1 J of heat or cold, UAH/J; b – operating cost per 1 m^2 of heat transfer surface, UAH/(m^2·h); $Q(t)$ and $F(t)$ – analytical dependencies of the heat load and heat transfer surface area on the temperature difference, respectively.

The optimal (temperature-based) operating mode of the apparatus is determined from the condition $dM/dt=0$.

A. S. Nevsky developed a methodology for determining the optimal gas velocity in heat exchanger ducts for both longitudinal and transverse flow across tube surfaces. He proposed expressing capital investment and operating costs E as functions of

$$\text{К}=\varphi_1(w_к);\ \text{Е} = \varphi_2(w_э).$$

The optimal values of the velocities w_k and w_E are determined using standard methods by solving the equations $d\text{К}/dw_к= 0$ and $d\text{Е}/d\,w_E$ with respect to the unknowns w_k and w_E . The optimal velocity is usually selected by iteration within a relatively narrow range between w_k^{opt} and w_E^{opt}. Eigevson L.S. derived an equation for the function C

C =φ(K; E), which relates capital investment and operating costs to the flow velocity www and the design parameters – the tube diameter d and the number of rows z. The task of determining the optimal values of w_2, d and z is thus reduced to solving a system of equations

$$\frac{\partial C}{\partial w^2} = 0; \ \frac{\partial C}{\partial d} = 0; \ \frac{\partial C}{\partial z} = 0.$$

Shpakiv S.A. proposed a graph–analytical method for solving certain specific cases of this problem, taking into account the payback period of capital investments.

Mykhailivskyi B.I. developed a graphical method for determining the optimal flow velocities in a heat exchanger. For various practical conditions, methods for determining optimal velocities, temperatures, and pressure drops have been discussed in the works of Z.F. Chukhanov, I.M. Razumov, V.Z. Zhadan, G.A. Vikhorev, Ts.A. Bakhshiyan, A.V. Zarubina, L.A. Richter, I.K. Shtiper, and others.

The main material costs associated with the heat exchange process include:

– the cost of materials and labor for the manufacture, installation, repair, and maintenance of the heat exchanger;

– the cost of purchasing and replenishing losses of the heat-transfer medium;

– the energy costs required to circulate the heat-transfer medium through the heat exchanger;

– the cost of materials and labor for the manufacture, installation, repair, and maintenance of the devices used for fluid circulation.

The key techno-economic performance indicators of the heat exchange process are:

– capital investment K;

– operating costs E;

– payback period T.

These indicators are functions of the technological parameters of the process and the design characteristics of the heat exchanger. The objective of designing a custom heat exchanger or selecting one from a standardized series is to determine the structural and process parameters that, under given conditions, minimize operating costs and capital investments within a specified payback period. In some cases, the problem may be formulated differently –for example, minimizing only the operating costs for a fixed capital investment, or ensuring a predetermined payback period.

Establishing a functional relationship between the economic, technological, and design characteristics of a heat exchanger makes it possible to determine optimal values based not on the designer's intuition but on analytical calculations that, in most cases, yield unique solutions.

From a techno-economic standpoint, an optimal heat exchanger is one for which the sum of the annual operating costs E and the capital investment K, distributed over the normative payback period $t_{n.o.}$, is minimized.

Thus,

$$P=E+K/t_{n.o}.$$

The capital investments K and annual operating costs E include not only the expenses directly related to the heat exchanger itself, but also the corresponding share of the capital and operating costs of the auxiliary equipment (pumps, compressors, fans). This share is determined in proportion to the ratio of the hydraulic resistance power of the heat exchanger to the total power of the driver unit. Certain other related costs are also included.

Let the quantity P be referred to as the optimality criterion of the apparatus. The mathematical optimization problem then reduces to finding the global minimum of the function P.

The total capital investment K includes those expenditures that directly affect the choice of the optimal design variant:

$$K=K_t+K_{m.t}+K_n+K_{m.n}$$

where K_t – capital investment in the heat exchanger itself (UAH); $K_{m.t}$ – installation cost of the heat exchanger (UAH); Kn – capital investment in the driver unit (pump, compressor, or fan) providing the design flow rate of the heat carrier (UAH); $K_{m.n}$ – installation cost of the driver unit (UAH).

The capital cost of the heat exchanger is assumed to be proportional to the heat transfer area and can be expressed as:

$$K_t=K_{1F}\cdot F$$

where F is the calculated (or catalog) heat transfer surface area, m² and K_{1F} is the manufacturing cost per unit area of the heat transfer surface (UAH/m²).

The installation cost of the heat exchanger is defined as:

$$K_{m.t}=K_{2F}\cdot F$$

where K_{2F} is the installation cost per unit area (UAH/m²).

The capital investment in the driver unit is given by:

$$K_n=K_{1N}\cdot N_{h.r.}$$

where K_{1N} is the manufacturing cost per unit power of the driver (including the drive motor), UAH/kW, and $N_{h.r.}$ is the power required to overcome the hydraulic resistance within the heat exchanger circuit.

The installation cost of the driver unit is:

$$K_{m.n}=K_{2N}\cdot N_{h.r.}$$

where K_{2N} is the installation cost per unit installed power (UAH/kW).

A common drawback of conventional analytical and graphical-analytical approaches is their inability to account for the mutual influence of a large number of independent variables – such as flow velocities, temperatures, pressure drops, and geometric characteristics of the apparatus (tube diameters and lengths, grid geometry, number, shape, and dimensions of baffles, etc.).

This problem can be addressed by setting the first-order partial derivatives of the objective function to zero, resulting in a system of equations. Solving this system yields the values of the independent variables corresponding to the extrema (minimum or maximum) of the objective function. However, the resulting system typically consists of transcendental equations, which can only be solved by iterative methods. Furthermore, additional analysis is required to determine whether the obtained solution corresponds to a global minimum or maximum.

The above-described techno-economic criteria (and even more so the thermodynamic ones) cannot always be taken as decisive when selecting or designing a heat exchanger. In many cases, constructive, technological, or operational criteria become determining factors.

Constructive criteria include constraints on the weight, shape, overall dimensions, and layout of the heat exchange surface, as well as limitations imposed by material properties (e.g., operation at high or low temperatures, exposure to aggressive media).

Technological criteria, which can significantly affect the dimensions, shape, and design of the apparatus, include: absolute pressure and pressure drop across the unit, temperatures of the working fluids and walls, physical properties (viscosity, specific heat, thermal conductivity, density) and their dependence on temperature, as well as the type of heat transfer process involved (condensation, evaporation, or chemical reactions between components of the heat carrier, etc.).

Since a heat exchanger is always an element of a larger technological or energy system—typically controlled and regulated automatically – the dynamic characteristics of the unit also acquire particular importance. These characteristics may be classified as both technical and constructive design criteria.

Operational characteristics of heat exchangers may include factors such as the rate of fouling on heat transfer surfaces, corrosion intensity, and time-dependent changes in the properties of the heat transfer medium, among others.

The methods used to determine optimal designs, dimensions, and operating conditions of heat exchangers under the aforementioned constraints may differ somewhat from the optimization method based solely on minimizing the optimality criterion. The following sections discuss the specific features of selecting optimal heat exchanger designs using computers (ECS) when constructive, technological, or operational limitations are present.

Designing a minimum-weight heat exchanger is an independent and practically important task. Such optimization is especially relevant for heat exchangers used in transport systems – on ships, aircraft, locomotives, and mobile technological units (e.g., oxygen, refrigeration, or gas-fuel systems). In some cases, weight reduction is necessary to improve the dynamic performance of the heat exchanger.

The relationship between the weight of a heat exchanger and its design and technological parameters can be obtained through standard transformations. The total weight of the heat exchanger can be represented as the sum of the weight of the active surface (tube bundle) G_p and the shell (including tube sheets):

$$G_{to}=G_p+G_k$$

To a first approximation, the dependence between the mass of the active surface and the heat transfer area can be expressed by an equation of the form:

$$G_p=F\cdot\delta_{mp}\rho_m$$

where F is the heat transfer surface area (determined by the geometry of the unit).

Because of the large number of independent variables and their complex interrelationships, determining the minimum possible mass of a heat exchanger – even for relatively simple configurations – can only be achieved using computer-based numerical methods.

From the governing relationships, it follows that reducing the weight of a heat exchanger can be achieved by:

– decreasing the tube diameter d_p;

– increasing flow velocities in both the tube w_p and shell $w_{bet.p}$ sides;

– reducing the operating pressures in the tube p_p and shell $p_{bet.p}$.

However, it should be noted that while lowering the pressure and wall thickness decreases the weight, it also reduces the heat transfer coefficient, thereby requiring a larger heat transfer area and, consequently, increasing the mass.

The mass of the heat exchanger is directly proportional to the ratio of the material density ρ_m to the allowable stress or ultimate strength σ_{lim} . Therefore, materials with a minimum ratio ρ_m/σ_{lim} are preferred for lightweight construction.

In cases where there is a large difference between the heat transfer coefficients on the hot and cold sides, finned tubes provide an effective way to reduce the overall mass. By formulating an equation for the total weight of a finned-tube heat exchanger – where the independent variables include fin geometry (height, thickness, spacing) – and minimizing this equation using computational methods, the optimal fin parameters can be determined.

Another means of weight reduction in process systems is changing the type of heat transfer fluid**.** The use of high-temperature liquid organic heat carriers (instead of water or steam) allows for significantly lower operating pressures while maintaining high temperature differentials. Both factors contribute to reducing the overall mass, even if the numerical values of the heat transfer coefficients decrease.

Further reduction in weight can be achieved through constructive measures, such as minimizing the mass of the shell, tube sheets, and covers. In some cases, even changing the flow arrangement (e.g., shell-side vs. tube-side circulation) can yield noticeable benefits in terms of weight or cost.

Designing a minimum-volume heat exchanger may be of particular importance for specialized applications. The approach to solving this problem is almost identical to that used for minimum-mass design: an explicit relationship is formulated between the exchanger volume and the independent variables, followed by numerical minimization with respect to volume. The corresponding variable values define the operating regime that ensures minimum volume.

One of the most effective optimization approaches is based on the relationship between the heat transfer factor of a finned surface and its airflow resistance factor – a method widely used in the United States. For air, this relationship can be expressed in the form:

$$Nu = f(Eu \cdot \mathrm{Re}^3).$$

The distinctive feature of this approach is its generality: it enables the determination of an optimal geometry for the heat transfer surface. The process of finding the optimum is easier to formalize mathematically and thus can be more efficiently handled using established computational techniques. The resulting optimum can be regarded as a theoretical limit of possible geometric optimization. In practical design, however, the real geometry may deviate slightly from this mathematical optimum in either direction.

3.3. Calculation of parameters for the prototype oil cooler of the diesel locomotive engine 10DN20.7/2×25.4

As a prototype, an oil cooler that is mass-produced and used in the diesel engine model 10DN20.7/2×25.4 was considered. The initial data for the calculation were determined based on the results of tests conducted on the oil cooling system of the TEP-70 type locomotive. All calculation parameters are presented in Table 3.2**.**

The prototype oil cooler, with all its known geometric characteristics, was analyzed using the so-called reverse calculation method. In this approach, the flow rates and initial parameters of the heat carriers, as well as the geometric parameters of the heat exchanger, are known, while the final parameters of the heat carriers – such as the efficiency and hydraulic resistances on both sides – are determined.

Table 3.2 – Initial data for the calculation of the prototype oil cooler of the diesel locomotive engine 10DN20.7/2×25.4

№ s/n	**Parameter**	**Dimension**	**Value**
1.	Oil consumption	G, kg/s	21.400
2.	Water consumption	G_w, kg/s	22.200
3.	Inlet oil temperature	t_{oil1}, °C	87.140
4.	Inlet water temperature	t_{w1}, °C	74.700
5.	Inlet oil pressure	p_{oil1}, kPa	850.00
6.	Inlet water pressure	P_w, kPa	392.00

To perform the reverse calculation, a software library developed by the ***Department of Internal Combustion Engines, Power Plants, and Technical Operation*** was used. The obtained results showed close agreement with experimental data, which confirms the reliability and accuracy of the applied calculation methodology.

According to the results of the computation, the main parameters of the oil cooler (OC) are as follows:

– oil outlet temperature: t_{w2}=84.7∘Ct;

– oil-side pressure drop: ΔP=28.75 kPa;

– tube bundle mass: M_p=276 kg.

3.4. Determination of the geometrical dimensions of a finned round tube

To determine the geometrical dimensions of the tube that satisfy the conditions formulated in Section 3.1, a parameter sweep method can be applied. This involves systematically varying the main geometric parameters to achieve the desired performance criteria.

In this case, the solution can be obtained using a reverse problem algorithm. A predeveloped computer program, provided by the department, was used for this purpose. Here, only the calculation results and their analysis are presented.

When finding the optimal solution in the reverse problem, four parameters are varied sequentially in the initial dataset: *D*, *u*, δ_1**,** and δ_2. To obtain a solution, all possible combinations of these parameters within their respective ranges must be tested. If *m* is the number of variables and *n* the number of gradations (steps) per variable, the total number of combinations is C_m^n. Thus, for four parameters and five gradations each, the total number of variants is 1024.

Since such a large computational volume is impractical, even with automation, a simplified approach – previously tested by the department in solving similar problems – is applied. This method proceeds as follows:

1. The solution is first sought as a function of one parameter (most conveniently, the diameter *D*).
2. At the point of this solution, the required heat exchanger performance is then examined as a function of another parameter.
3. The process continues similarly for the third and fourth parameters.

This approach allows for automatic parameter scanning, generating ready-made tables and even graphical representations of the results.

To organize this automation efficiently, it is advisable to select a composite function of the heat transfer surface, which simultaneously relates all parameters of interest. In this case, the goal is to minimize the tube bundle mass M_p and maximize the heat exchanger efficiency**.** If a single combined (dimensionless) parameter can be defined to represent these relationships, then by tracking its behavior, one can determine the optimal combination of geometrical dimensions corresponding to its extremum.

The proposed approach, therefore, can be formulated as a coordinate descent method, which is one of the most reliable optimization techniques – provided the solution domain does not contain significant local extrema.

As such a composite dimensionless function, one of two possible nondimensional complexes may be used:

$$EuRe^3 \text{ or } \frac{Nu}{Eu\,\mathrm{Re}^3}.$$

Such composite parameters can be analyzed as functions of dimensional quantities. The resulting dependencies can then be used to find a solution according to the approach described above.

At the same time, an alternative way to obtain a possible solution to the given problem is to use relationships of the form:

$$EuRe^3 = f(Nu).$$

In this case, the algorithm for solving the problem can be significantly simplified, since it requires operating only with boundary conditions. However, the obtained solution will not be directly linked to the specific geometric dimensions of the possible heat exchanger design.

In the following section, a search for the solution to the stated problem is carried out, resulting in specific recommendations for the optimal geometry of the heat transfer surface under the given conditions.

Determination of the optimal external fin diameter

To determine the optimal value of the external fin diameter D, a series of reverse calculations for the oil cooler was performed with a sequential variation of this parameter. Based on the obtained data, the optimal diameter value was identified under the condition that the oil outlet temperature from the cooler (t_{m2}) does not exceed 84.7 °C, while the pressure drop is minimized at this temperature.

The calculations were performed using the initial conditions listed in Table 3.2**.** As a result, a recommended value of D = 21.5 mm was obtained. This value is very close to the diameter that provides the absolute minimum oil temperature, being only slightly smaller. It should be noted that any further increase in D, although it may slightly reduce the temperature, leads to a rapid rise in flow resistance.

Determination of the Optimal Fin Pitch

To find the optimal fin pitch uuu, a tube with the previously determined optimal diameter *D* was used, and a similar series of reverse calculations was performed with stepwise variation of *u*. According to the results, the optimum was defined based on maintaining the specified oil outlet temperature. The corresponding optimal fin pitch was found to be 2 mm.

An increase in fin pitch results in higher oil outlet temperatures, while a decrease in pitch lowers the temperature but simultaneously increases flow resistance. Conversely, reducing resistance leads to an increase in outlet temperature. Thus, the chosen value of 2 mm should be considered a compromise optimum for this task.

Determination of the optimal fin thickness

To determine the optimal fin thickness, the dependencies of temperature and flow resistance on the thickness at both the fin base and tip were analyzed. The minimum allowable thicknesses are constrained by the manufacturing capabilities of the production facility. In this case, the limits were set as follows:

– fin base thickness**:** not less than 0.35 mm

– fin tip thickness**:** not less than 0.15 mm

It was established that the compromise optimum corresponds to a fin tip thickness of 0.15 mm and a fin base thickness of 0.35 mm. Fin thickness has a strong influence on the mass and therefore on the cost of the heat exchanger. Its influence on the thermal performance of the unit, within the permissible range of variation, is relatively small. Therefore, reducing fin thickness is advantageous, as it leads to a lower overall mass of the heat exchanger and simultaneously reduces flow resistance.

3.5. Results of selecting the optimal heat transfer surface for the oil cooler of the diesel locomotive engine 10DN20.7/2×25.4

As a result of the performed analysis, the optimal geometric parameters of a finned tube with rolled (strip-type) fins were determined (see Fig. 3.1). The obtained results are summarized in Table 3.3.

Table 3.3 – Geometric parameters of the rolled-fin tube

№ s/n	Parameter	Dimension	Value
1.	Outer diameter of fins	D, m	0,0215
2.	Diameter of carrier tube	d_0, m	0,0109
3.	Inner diameter of tube	d_w, m	0,0102
4.	Pitch between fins	u, m	0,0020
5.	Thickness of fin at base	δ_1, m	0,00035
6.	Thickness of fin at apex	δ_2, m	0,00015

The resulting geometric parameters of the heat transfer surface must satisfy the specified constraints regarding pressure drop**,** oil outlet temperature (t_{oil2}), and the mass of the heat transfer surface. When all these constraints are met, the obtained solution can be regarded as a compromise optimum, since a change in one of the geometric parameters leads to an improvement in one criterion while simultaneously deteriorating another. For example:

– a reduction in the external fin diameter D decreases both the pressure drop and the tube bundle mass M_p, but results in an increase in oil outlet temperature;

– conversely, reducing the fin pitch u lowers the oil outlet temperature t_{oil2}, but increases both the pressure drop and the mass of the heat transfer surface.

3.6. Results of thermal performance calculations for the oil cooler of the diesel locomotive engine 10DN20.7/2×25.4

Table 3.4 presents the results of the thermal performance calculations for the oil cooler based on staggered tube bundles with rolled fins. The results were obtained by solving the inverse heat transfer problem using the previously established recommendations for selecting the optimal heat transfer surface geometry.

Table 3.4 – Results of calculations for the alternative oil cooler based on a staggered tube bundle with rolled fins (using modified fin dimensions)

№ s/n	Parameter	Dimension	Value
Thermal parameters of the oil cooler and coolants			
1.	Oil consumption	G, kg/s	21,400
2.	Oil temperature at the inlet	t_{oil1}, °C	87,140

3.	Oil pressure at the inlet	p_{oil1}, kPa	850,000
4.	Oil temperature after the cooler	t_{oil2}, °C	84,700
5.	Oil cooler efficiency	η	0,196
6.	Average oil speed	w_{oil}, m/s	1,250
7.	Oil resistance	ΔP_{oil}, kPa	20,880
8.	Water consumption	G_w, kg/s	22,200
9.	Water temperature at the inlet	t_{w1}, °C	74,700
10.	Water temperature at the outlet	t_{w2},°C	75,920
11.	Water pressure at the inlet	P_w, kPa	392,000
12.	Water velocity in the tubes	w_w, m/s	4,590
13.	Water resistance	ΔP_w, kPa	272,900
14.	Tube bundle mass utilization factor	k_g, W/kg·K	144,300
15.	Amount of heat transferred to the oil cooler	Q, W	113589,200
16.	Thermal resistance of the wall and the layer of contaminants	R_{wall}, m^2·K/W	0,00030
Tube and tube grid parameters			
17.	Outer diameter of fins	D, m	0,0215
18.	Diameter of carrier tube	d_0, m	0,0109
19.	Inner diameter of tube	d_w, m	0,0102
20.	Pitch between fins	u, m	0,0020
21.	Thickness of fin at base	δ_1, m	0,0004
22.	Thickness of fin at apex	δ_2, m	0,0002
23.	Pitch between tubes perpendicular to oil flow	S_1, m	0,0270
24.	Pitch between tubes in depth	S_2, m	0,0234
Design parameters of the oil cooler			
25.	Number of oil passes in a single-pass water-cooled oil cooler or number of series-connected reversing elements installed countercurrently in the water flow of a multi-pass water-cooled oil cooler	b	1.000
26.	Number of heat exchanger passes in water	b_w	2.000
27.	Inner diameter of the casing	D_p, m	0,380
28.	Diameter of the circle circumscribed around the bundle	D_o, m	0,349
29.	Total length of heat exchanger tubes	L_p, m	2,300
30.	Total number of tubes in the bundle	Z	122.000
31.	Actual number of tubes in a transverse row in the center of the cross section	$Z_д$	12.000
32.	Number of transverse rows in the compartment	Z_2	11.000
33.	Total number of diaphragms	$Z_{дф}$	11.000
34.	Distance between diaphragms in the middle of the bundle	L_1, m	0,118

35.	Distance between the diaphragm and the board in the receiving compartment	L_2, m	0,147
36.	Distance between the diaphragm and the board in the discharge compartment	L_3, m	0,147
37.	Thickness of the diaphragm	S_d, m	0,007
38.	Height of the segment cutout on the diameter D_t	H_{seg}, m	0,060
39.	Weight of the tube bundle	M_p, kg	78.000
40.	Total area of the heat exchange surface on the fin side	F, m^2	57,300

A comparison of the standard and alternative oil cooler is given in Table 3.5.

Table 3.5 – Comparison of the standard and alternative oil cooler of the diesel locomotive engine 10DN20.7/2×25.4

№ s/n	**Parameter**	**Dimension**	**Value**	
			Standard oil cooler	**Alternative oil cooler**
1.	Oil temperature after oil cooler	t_{oil2}, ^oC	84.700	84,700
2.	Oil resistance	ΔP_{oil}, kPa	28,750	20.880
3.	Estimated bundle mass	M_p, kg	275.940	78.000
4.	Oil cooler efficiency	η	0.197	0.196

3.7. Conclusions to the part

This section discusses possible ways to optimize heat exchange devices in internal combustion engine cooling systems.

The geometric parameters of an alternative heat exchange surface (optimal geometric parameters obtained) in the form of a staggered bundle of roll-fin pipes used in the oil cooler of the 10DN20.7/2x25.4 diesel locomotive engine have been calculated.

The resulting alternative design of the oil cooler for the 10DN20.7/2x25.4 diesel engine at the same oil temperature at the heat exchanger outlet provides lower energy consumption for pumping the coolant (i.e., the oil resistance is 7.87 kPa lower). In addition, the alternative design of the oil cooler has a 71.7% lower heat exchange surface mass (78 kg) and, therefore, a significantly lower cost.

PART 4

ECONOMIC JUSTIFICATION FOR THE IMPLEMENTATION OF AN ALTERNATIVE HEAT EXCHANGE SURFACE FOR THE OIL COOLER OF A MAINLINE DIESEL LOCOMOTIVE

4.1. Technical and economic justification for modernizing the engine oil system of the locomotive engine 10DN20,7/2x25,4

The objective of this study is to provide an analytical justification of the economic efficiency of implementing an oil cooler with an alternative heat transfer surface, designed on the basis of a finned tube with rolled (strip-type) fins, for the cooling system of the 10DN20.7/2×25.4 diesel engine, which serves as the power unit of the TEP-70 locomotive, in comparison with the currently mass-produced (standard) oil cooler.

To determine the economic effect of introducing the oil cooler with the alternative heat transfer surface into operation, it is necessary to compare the thermal performance parameters of the working fluids in both the standard and alternative heat exchangers, as well as to analyze the engine parameters before and after the implementation of the new surface design.

The initial data were selected from the technical specifications of the diesel-generator set with the 10DN20.7/2×25.4 engine and from manufacturer data.

Table 4.1 presents the technical and economic indicators of the standard power unit (SPU).

The fuel and oil prices correspond to the 2025 market rates. The cost of the standard oil cooler was obtained from the official website of the manufacturer, while the cost of the alternative unit was estimated considering the lower price of its oil-cooling heat exchanger. The mass and cost of the tube bundles for both the standard and alternative oil coolers are given in Table 3.2.

The cost of the heat exchangers was evaluated based solely on their mass, assuming that the manufacturing expenses per unit mass are identical. The metal cost was taken as the average market value.

Table 4.1 – Technical and economic characteristics of the standard power unit of the locomotive engine 10DN20,7/2x25,4

№ s/n	Name of indicators	Dimension	Value
1.	Power	N_e, kW	2800,000
2.	Rotation speed	n, min^{-1}	850,000
3.	Diameter of the cylinder	D, m	0.207
4.	Stroke of the piston	S, m	0.254
5.	Specific fuel consumption	g_e, kg/kW·h	0,197
6.	Mechanical efficiency	η_m	0,86
7.	Specific oil consumption	g_o, kg/kW·h	0.00170
8.	Weight	G, kg	16000
9.	Dimensions	L, m	6,096
		B, m	1,405
		H, m	3,55
10.	Price of the engine	P_{engine}, UAH	6000000
11.	Price of fuel	P_{fuel},UAH/t	50000
12.	Price of oil	P_{oil}, UAH/t	43000
13.	Work hours for the year	T, год	4000
14.	Normative cost-effectiveness comparison coefficient	E	0.15
15.	Number of control repairs	k	1
16.	Coefficient of part of the cost of capital repairs	$K_{c.r}$	0.05
17.	Engine life before overhaul	$t_{o.h}$ h	64000
18.	Service personnel, people	l	1
19.	Salary of service personnel,	S, UAH	25000

Table 4.2 – Weight and cost of heat exchanger bundles for serial and alternative oil coolers of the diesel locomotive engine 10DN20.7/2×25.4

№ s/n	Parameter	Dimension	Value	
			Standard oil cooler	Alternative oil cooler
1.	Estimated bundle mass	M_p, кг	275.940	78.000
2.	Cost of metal	C, UAH/kg	300	300
3.	Cost of oil cooler	C_{cooler}, UAH	77382	23400

The calculation is carried out according to the methodology described in Korchagin V.A. et al., “Calculation of the Economic Efficiency of New Technology in Machine-Building Enterprises.”

The power unit of the mainline locomotive, in which an alternative heat transfer surface based on a finned round tube with rolled (strip-type) fins is used, will hereafter be referred to as the alternative power unit (APU).

Next, we will estimate the payback period for replacing the standard oil cooler with the alternative one in existing power units already in service.

The cost of the new oil cooler is estimated taking into account the deduction of the value of the metal recovered from the decommissioned standard oil cooler.

The approximate cost of the new oil cooler is therefore estimated to be:

$$C_A^{cooler} = 2 \cdot C_{A.Cost}^{cooler} - C_{S.Cost}^{cooler} = 2 \cdot 23400 - 77382 = -30582 \text{ UAH}.$$

Cost of alternative power unit

$$C_{\text{APU}} = P_{engine} + C_A^{cooler} = 6000000 - 30582 = 5969418 \text{ UAH}.$$

Amount of electricity generated during one year of diesel generator 10DN20.7/2×25.4 operation:

$$W = N_e \cdot T$$

where T – work hours for the year.

Standard power unit – $W_S = 2800 \cdot 4000 = 11200000$ kW·h;

Alternative power unit – $W_A = 2800 \cdot 4000 = 11200000$ kW·h.

Operating costs of the standard power unit and alternative power unit:

– fuel consumption determined by the formula – $G_f = 1,02\ W \cdot g_e$, kg;

$G_f^S = 1,02 \cdot 11200000 \cdot 0,197 = 2250528$ kg;

$G_f^A = 1,02 \cdot 11200000 \cdot 0,197 = 2250528$ kg.

– the cost of fuel required to operate the engine during the year is determined by the formula – $C_f = 1,1 \cdot G_f \cdot P_{fuel}$, UAH

$C_f^S = 1,1\ 2250528 \cdot 50000 \cdot 10^{-3} = 123779040$ UAH;

$C_f^A = 1,1 \cdot 2250528 \cdot 50000 \cdot 10^{-3} = 123779040$ UAH.

– oil consumption is determined by the formula – $G_{oil} = 1,02 \cdot W \cdot g_e$, kg;

$G_{oil}^S = 1,02 \cdot 11200000 \cdot 0,0017 = 19420.8$ kg;

$G_{oil}^A = 1,02 \cdot 11200000 \cdot 0,0017 = 19420.8$ kg;

– the cost of oil required for the engine to operate throughout the year is determined by the formula – $C_{oil} = 1,1 \cdot G_{oil} \cdot P_{oil}$, UAH;

$C_{oil}{}^{S}$ = 1,1 · 19420.8 · 43000 · 10^{-3} = 918603.84 UAH;

$C_{oil}{}^{S}$ = 1,1 · 19420.8 · 43000 · 10^{-3} = 918603.8 UAH.

– the total cost of fuel and lubricants for the year is determined by the formula –

$C_{fl} = C_f + C_{oil}$, UAH;

$C_{fl}{}^{S}$ = 123779040 + 918603.84 = 124697644 UAH;

$C_{fl}{}^{A}$ = 123779040 + 918603.84 = 124697644 UAH.

Cost of spare parts for the entire period of engine operation

$$C_{SP} = 1{,}05 \cdot K_{c.r} \cdot k \cdot P_{engine}, \text{ UAH;}$$

$C_{SP}{}^{S}$ = 1,05 · 0,05 · 1 · 6000000 = 315000 UAH;

$C_{SP}{}^{A}$ = 1,05 · 0,05 · 1 · 5969418 = 313394.45 UAH.

– the annual engine maintenance cost is calculated using the following formula –

$C_m = 12 \cdot l \cdot S$, UAH;

$C_m{}^{S=}$ 12 · 1 · 25000 = 300000 UAH;

$C_m{}^{A}$ = 12 · 1 · 25000 = 300000 UAH.

– overall costs – $B_{\Sigma} = C_{fl} + C_A^{cooler} + C_{SP} + C_m$, UAH;

$B_{\Sigma}{}^{S}$= 124697644 + 0 + 315000 + 300000 = 125312644 UAH;

$B_{\Sigma}{}^{A}$= 124697644 – 30582 + 313394.45 + 300000 = 125280456 UAH.

Cost of one kW·h of electricity:

– cost of a kilowatt of electricity standard power unit

$$U^S = \frac{B_{\Sigma}^S}{W^S} = 125312644 / 11200000 = 11.188629 \text{ UAH/(kW} \cdot \text{year)}$$

– cost of a kilowatt of electricity alternative power unit

$$U^A = \frac{B_{\Sigma}^A}{W^A} = 125280456/11200000 = 11.185755 \text{ UAH /(kW} \cdot \text{year)}$$

Payback period:

$$T_0 = \frac{C_A^{cooler}}{B_{\Sigma}^S - B_{\Sigma}^A} = –30582/(125312644 –125280456) = 0{,}9501 \text{ year.}$$

Engine operating time – $T = \frac{t_{кр}}{T}$, *рік*

$$T^S = \frac{64000}{4000} = 16 \text{ year}; \quad T^A = \frac{64000}{4000} = 16 \text{ year}.$$

Annual economic effect for consumers from using alternative power unit:

$$E_к = \left[P_{engine} \cdot \frac{W^A}{W^S} \cdot \left(\frac{\frac{1}{T^S} + E_S}{\frac{1}{T^A} + E_A} \right) + \frac{B_\Sigma^S - B_\Sigma^A}{\frac{1}{T^A} + E_A} \right] - C_A^{cooler}, UAH$$

$$E_к = \left[6000000 \cdot \frac{11200000}{11200000} \cdot \left(\frac{\frac{1}{16} + 0{,}15}{\frac{1}{16} + 0{,}15} \right) + \frac{125312644 - 125280456}{\frac{1}{16} + 0{,}15} \right] + 30582 = 182052.8 \text{ UAH.}$$

4.2. Conclusions to the section

The calculation proved the economic feasibility of introducing an alternative heat exchange surface based on a finned tube with rolled (strip) fins for the 10DN20.7/2×25.4 engine oil cooling system used as a power unit on the TEP70 diesel locomotive. The economic effect was achieved due to the significantly lower weight of the designed heat exchanger. According to calculations, the payback period is approximately 0.95 years, and the annual economic effect of using an alternative heat exchange surface in the oil cooler is 182052.8 UAH.

PART 5
ANALYSING AND DEVELOPING MEASURES TO ENSURE LABOUR AND ENVIRONMENTAL PROTECTION REQUIREMENTS FOR TWO-STROKE LOCOMOTIVE ENGINE 10DN20.7/2X25.4

5.1. General occupational safety requirements for crews servicing mainline locomotives

The general occupational safety requirements establish safety standards for locomotive crews during the operation of electric locomotives, diesel locomotives, and diesel trains (hereinafter referred to as “locomotives”), as well as for multiple-unit rolling stock (hereinafter referred to as “MURS”) and their maintenance.

Individuals allowed to perform maintenance and operation of diesel locomotives and MURS must be at least 18 years old, have undergone professional selection and training, a preliminary medical examination, and passed qualification exams granting them the right to operate locomotives and MURS. They must also complete induction, on-the-job, and safety training, as well as examinations on occupational health and fire safety.

During their duties, the locomotive crew – consisting of a driver and an assistant – must regularly undergo refresher, unscheduled, and targeted safety briefings, as well as periodic medical check-ups.

The driver and assistant must be familiar with:

- the general design of diesel locomotives and MURS, and methods for identifying and correcting possible equipment malfunctions;
- the impact of hazardous and harmful production factors that may occur during operation, the rules for providing first aid, the location of first-aid kits or medical bags, and the purpose and dosage of their contents;
- the arrangement of electrical machines, wiring, devices, and equipment operating under high voltage on locomotives and MURS;
- the occupational safety, industrial hygiene, and fire safety requirements applicable to diesel locomotives.

The driver and assistant are required to:

- carry all necessary documents while on duty, including an electrical safety group certificate and an official ID confirming their qualification to operate a locomotive or MURS (or, in the case of the assistant, a valid assistant driver certificate);
- face the locomotive and use handrails when boarding or disembarking;
- wear designated protective clothing and footwear during work and technical inspections of locomotives and MURS.

During work, the driver and assistant may be exposed to the following hazardous and harmful factors:

- moving parts of traction rolling stock (TRS) equipment;
- high levels of dust and gas in the workplace air;
- elevated or reduced temperatures of equipment surfaces and materials;
- excessively high or low ambient air temperature in the workplace;
- increased noise and vibration levels;
- exposure to infrasound;
- high or low humidity and air movement in the workplace;
- high voltage in electrical circuits that could pass through the human body;
- static electricity accumulation;
- electromagnetic radiation from specific devices;
- insufficient or absent natural lighting;
- poor workplace equipment or layout;
- formation of flammable atmospheres due to leaks or discharges of oil, diesel fuel, or heated gases under pressure, and the potential for ignition sources leading to fires;
- exposure to toxic, irritating, or sensitizing substances entering through the respiratory system or skin;
- neuropsychological stress caused by mental fatigue, monotonous work, and periodic emotional strain.

The locomotive crew must be provided with all necessary personal protective equipment (PPE) according to established standards.The train driver is personally responsible for ensuring fire safety and the proper functioning of the fire suppression system on the assigned locomotive and MURS.

The diesel locomotive crew must strictly observe the following fire safety rules:

- it is prohibited to store or transport any foreign objects in the driver's cab, high-voltage compartment, or diesel engine room;
- all service areas and driver's cabs of diesel locomotives and multiple-unit rolling stock (MURS) must be kept clean and tidy at all times;
- lubricants must be stored exclusively in metal containers (such as cans, jerrycans, or oilers) with narrow necks and tightly fitting lids. Clean and used wiping materials must be stored separately in metal boxes or covered buckets;
- all protective devices of electrical equipment must remain in proper working condition;
- smoking and the use of open flames in the engine compartment or anywhere on locomotives and MURS are strictly prohibited;
- the crew must regularly check the operability and integrity of fire suppression systems, including all primary firefighting equipment.

The diesel locomotive crew must adhere to the following personal hygiene practices:

- to prevent potential skin diseases caused by contact with diesel fuel, lubricants, or cooling water, protective creams, pastes, and "biological" gloves must be used. The selection of these protective agents should follow the recommendations of the depot's medical office;
- after completing work that required the use of protective creams or pastes, hands should be washed with soap and water, then treated with boric petroleum jelly or lanolin cream, gently rubbing it into the skin;
- if any signs of skin irritation appear, the affected person must report to the medical office for treatment.

Compliance with and understanding of safety instructions is a professional duty of every driver and assistant driver. Any violation of these safety regulations constitutes a breach of labor discipline and may result in disciplinary or other liability in accordance with the current legislation of Ukraine.

Before commencing work, the diesel locomotive crew must:

1. Undergo a medical examination.

2. Receive keys and control handle: Obtain an inventory set of keys and a reversible control handle from the depot duty officer or the outgoing driver at the crew change point. Verify that the key numbers correspond to the assigned locomotive or MURS. The use of non-inventory control handles, interlock devices, switch keys, or any improvised substitutes is strictly prohibited.

3. Ensure immobilization: Before accepting the locomotive or MURS, confirm that it is secured with a handbrake or that wheel chocks are placed under the front axle from both sides to prevent unintended movement.

4. Inspect the cab: Conduct an initial inspection of the driver's cab, paying particular attention to the condition of the control panel and the shielding of the heating system.

5. Check protective and safety systems: Test the operation of all special interlocking devices and electrical contacts, and verify the presence and condition of grounding systems and protective enclosures. Operation of a locomotive or MURS with defective or missing grounding, interlocking devices, enclosures, or voltage indication systems in the high-voltage (apparatus) compartment is strictly prohibited.

6. Inspect protective equipment: Verify the availability and proper condition of the following personal protective devices:

– dielectric gloves;

– dielectric mats;

– grounding rods for the primary winding of traction transformers (for AC-powered rolling stock);

– insulating rods for disconnecting traction motor switches;

– gas masks (for locomotives equipped with gas-based fire suppression systems);

– noise-insulating headphones (for diesel locomotives).

It is strictly prohibited to use any protective equipment that has exceeded its expiration date. When inspecting a gas mask, the crew must ensure that it has no visible external damage and that all valves, the filter box, and the hose are in good working condition. During the preliminary inspection of insulated-handled mechanic or assembly tools, verify that the insulation is free from pits, chips, blisters, or any other defects. Dielectric mats must not have any mechanical damage. The presence of punctures in dielectric gloves should be checked by rolling the glove toward the fingers to detect air leaks.

7. Troubleshooting and Compressed Air Systems. Before repairing any equipment or eliminating air leaks in joints, reservoirs, or pressurized devices, these systems must be deactivated and completely depressurized. It is strictly prohibited to open or close valves or taps by striking them with a hammer or any other object. Before performing brake tests, the locomotive driver must notify the assistant, ensure that there are no people near or under the locomotive or MURS, and confirm that the braking system is engaged. It is forbidden to use portable lamps without protective wire guards, or those with damaged wire insulation or plugs. When connecting portable lamps to a power source, they must be held securely or firmly fixed to prevent accidental falls.

8. Battery Inspection Procedures. Before inspecting the batteries, the main switch must be turned off and fuses removed. It is strictly prohibited to place tools directly on top of batteries. All inspections must be conducted using a portable lamp. Battery jumpers must be removed, tightened, and installed only with socket wrenches equipped with insulated handles.

9. Entering and Leaving the Depot or Maintenance Facility. Before entering or leaving a diesel locomotive depot or a locomotive maintenance point (hereinafter – LMPS), the crew must ensure that:

– the gates are fully open and securely fixed;

– there are no people standing on the steps, ladders, platforms, or roof of the locomotive, or in the inspection pit of the track where the locomotive will be positioned.

10. Movement Within the Depot. Locomotive entry to or exit from the depot or LMPS must be carried out according to the local signal indications of the specific track

and only on the command of one designated person — the duty officer, their assistant, or the senior foreman — under the supervision of the shift foreman (or crew leader). The locomotive's speed during entry must not exceed 3 km/h, and jerky or impact movements are strictly prohibited.

11. Locomotive Entry into the Depot. Locomotives must enter the depot with the diesel engine shut down. Movement should be carried out by supplying power to the traction motors from an external energy source or by coupling with another locomotive. When using another locomotive for towing, protective railway platforms must be attached in front to prevent it from entering the building directly.

Basic safety requirements for the maintenance of diesel locomotives and diesel trains:

1. Before starting the diesel engine, the condition of all its components and auxiliary equipment must be inspected. All tools and accessories must be removed, the floor panels secured, and a warning signal given to the assistant driver.

2. It is strictly prohibited to start the diesel engine on multi-section locomotives if the "Fuel Pump" buttons on the control panels of both sections are activated simultaneously.

3. It is strictly prohibited to perform any assembly or installation work on rotating parts of the locomotive while the engine is running.

4. Entering the diesel compartment while the engine is operating above the 10th throttle position is strictly forbidden.

5. Crankcase hatches must only be opened 10–15 minutes after the diesel engine has stopped to prevent exposure to hot gases and components.

6. Before inspecting the diesel engine, electrical equipment, or auxiliary systems, all electrical circuits must be de-energized, and the battery switch must be turned off.

7. It is strictly prohibited to open or repair any electrical devices or apparatus that are energized. In diesel trains lacking automatic circuit breakers in power supply circuits, special insulated handles must be used when replacing fuses. The installation and connection of wires to automatic switch terminals, electrical machines, and apparatus – as well as inspection and replacement of fuses – must only be performed when the cir-

cuit is de-energized. Only fuses with calibrated inserts and properly color-coded cartridges should be used.

8. Climbing onto the roof of diesel locomotives or diesel trains in electrified areas is strictly prohibited if the overhead contact wire is live or if its de-energized state cannot be verified.

9. If one section of a multi-section locomotive or diesel train has a non-operational diesel engine and is being moved by another connected section, the following safety measures must be observed on the inactive section:

– the reverser handle and main controller handle must be set to the neutral position;

– all control panel buttons, except for the lighting switch, must be turned off;

– the traction motors must remain disconnected.

10. When performing any work on the locomotive’s friction clutch, the air line supplying its drive must be closed.

11. During external inspection of the radiator section, the louver drive must be secured with a locking device. It is forbidden to open inspection hatches or enter the radiator shaft while the fan is running.

12. Inspection or removal of covers from traction motors, the main generator, or any auxiliary electrical machines is permitted only when the diesel engine and locomotive are completely stopped.

13. Fuel accumulated in the compartments of diesel fuel pumps must be removed using special vacuum extraction equipment. After refueling, tank caps must be tightly closed.

14. Prevent oil, fuel, or water from spilling onto the floor of the locomotive’s engine room.

15. When leaving the driver’s cab to enter the diesel compartment while the engine is running, noise-isolating headphones must be worn.

5.2. Environmental protection

Rail transport plays a significant role in the economic development of a country. In addition to the undoubtedly positive effects, such as the transportation of people and goods, there are also many negative consequences, particularly environmental pollution caused by harmful emissions from exhaust gases, as well as noise generated by diesel engine operations.

Noise is an inherent part of modern life, connected to the use of various forms of transport and the operation of industrial facilities, among other things. Continuous exposure to noise can have serious effects on the human body, leading to a range of health problems and an overall decline in well-being.

Recent studies show that noise exposure contributes to the development of various diseases and disorders, especially when a person is subjected to it for extended periods. One of the most common consequences of prolonged noise exposure is an increase in stress levels. Constant noise creates tension, which increases the likelihood of developing stress-related conditions, psychological disorders, and other issues.

As a result, individuals may experience irritability, fatigue, difficulty sleeping, and challenges in performing their professional duties. In addition to stress and physical fatigue, constant noise exposure can lead to hearing problems, including reduced hearing or even partial hearing loss.

In the workplace, such as in the locomotive engine room, prolonged exposure to high levels of noise can lead to noise intolerance for the driver and their assistant, and even hearing loss. Therefore, it is essential to take measures to protect the hearing of personnel working with locomotives.

There are many methods for noise protection on locomotives, which can significantly reduce the impact of noise on the human body. One of the most common and simplest solutions is the use of specialized ear protection. These headphones help dampen the sounds of the diesel engine and protect the ears from the harmful effects of noise.

To develop appropriate measures and strategies for noise reduction, detailed calculations must be made. These calculations will form the basis for the implementation of noise and vibration reduction measures.

Calculations of noise and vibration levels for the modernized locomotive 10DN20.7/2x25.4 two-stroke diesel engine

The noise level of the modernized locomotive 10DN20.7/2x25.4 two-stroke diesel engine can be determined by the following expression:

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

where n_n is the nominal rotation speed, n_n = 850 rpm; n is the operating rotation speed, n = 850 rpm; N_e is the nominal effective power, N_e = 2800 kW.

$$L = \left[54 + 10 \cdot \lg\left(850 + 2800^{0,55}\right) + 30\lg\left(\frac{850}{850}\right)\right] = 84{,}81 \text{ dB}$$

The noise level of the modernized locomotive 10DN20.7/2x25.4 two-stroke diesel engine, as shown in Fig. 5.1, falls within the permissible values.

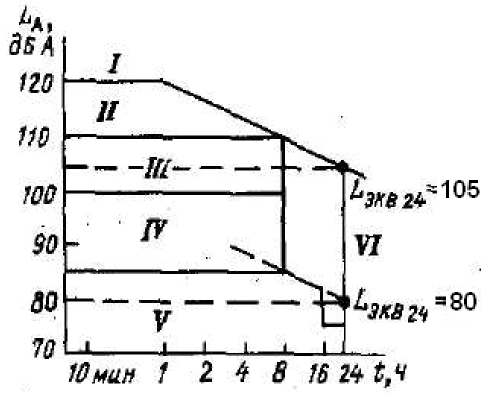

Figure 5.1 – Zones of permissible noise levels

The vibration level of the modernized 10DN20.7/2x25.4 two-stroke locomotive engine can be determined by the following expression:

$$L = 44 + 10\lg\left(\frac{n_n \cdot Ne^{0,55} \cdot \left(\frac{1+Ne}{m}\right)}{1 + \left(\frac{1}{1500}\right)^3 \cdot \frac{m}{Ne}} + 30\lg\left(\frac{n}{n_n}\right)\right)$$

where m – diesel weight 10DN20.7/2x25.4, m = 16000 kg.

$$L = 44 + 10\lg\left(\frac{850 \cdot 2800^{0,55} \cdot \left(\frac{1+2800}{16000}\right)}{1 + \left(\frac{1}{1500}\right)^3 \cdot \frac{16000}{2800}} + 30\lg\left(\frac{850}{850}\right)\right) = 84{,}14 \text{ dB}$$

The vibration level of the upgraded 10DN20.7/2x25.4 two-stroke locomotive engine does not exceed the established standards.

5.3. Conclusions to the part

This section outlines general safety measures for the engine room personnel of the TEP 70 mainline locomotive with a 10DN20.7/2x25.4 two-stroke engine. It has been determined that the vibration and noise levels of the modernized 10DN20.7/2x25.4 two-stroke engine used on the mainline locomotive do not exceed the established standards and are 84.81 and 84.14 dB, respectively.

CONCLUSION ON THE WORK

The master's thesis on the topic: ***“Modernization of the oil system of a diesel locomotive engine”*** is devoted to the modernization of the design of the oil cooler of the lubrication system of a 10DN20.7/2x25.4 two-stroke diesel engine of a mainline locomotive. The thesis contains all the necessary sections and drawings.

The first section presents the general structure and features of the mainline locomotive. The main characteristics of the operation and design of the 10DN20.7/2x25.4 two-stroke diesel engine are considered.

The second section calculates the operating cycle for the nominal operating mode of the 10DN20.7/2x25.4 two-stroke diesel locomotive engine.

The third section considers possible ways to optimize the heat exchange devices of the cooling system of internal combustion engines. The geometric parameters of an alternative heat exchange surface (optimal geometric parameters obtained) in the form of a staggered bundle of roll-fin pipes used in the oil cooler of the 10DN20.7/2x25.4 locomotive engine are calculated. It has been established that the obtained alternative design of the oil cooler for the 10DN20.7/2x25.4 diesel engine, at the same oil temperature at the outlet of the heat exchanger, provides lower energy consumption for pumping the coolant (i.e., the oil resistance is lower by 7.87 kPa). In addition, the alternative design of the oil cooler has a 71.7% lower heat exchange surface mass (78 kg) and, therefore, a significantly lower cost.

The fourth section proves the economic feasibility of introducing an alternative heat exchange surface based on a finned tube with rolled (strip) fins for the 10DN20.7/2×25.4 engine oil cooling system. The economic effect is achieved due to the significantly lower weight of the designed heat exchanger. According to calculations, the payback period is approximately 0.95 years, and the annual economic effect of using an alternative heat exchange surface in the oil cooler is 182,052.8 UAH.

The fifth chapter develops general safety measures for the personnel of the engine room of the TEP70 main locomotive and analyzes the harmful impact on the environment.

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