



Design Principles and Specifics of Air Supply Systems Used in the Engines Produced by JSC Kolomensky Zavod

V. F. Knelts and A. B. Matisen^(✉)

Moscow Polytech Kolomna Institute (Branch), 408, Oktyabr'skoy Revolyutsii Street,
Kolomna 140402, Russia
mab5-5@yandex.ru

Abstract. This paper presents the development stages of the basic principles to design and implement air supply systems produced by the engineering school of the JSC Kolomensky zavod. The unique character of technical solutions was due to the necessity to insure a high level of the air charge pressure in a wide range of loads not only for the transport engines produced by the Kolomensky Zavod, but also for specialised engines working with high suction values at the intake and back pressure values of the exhaust gases. To meet the requirements set up by the customer, a wide range of design solutions was developed systematically and introduced into serial production. These solutions were aimed at the improvement of methods and techniques to control supercharging. Along with fulfilling constantly growing requirements, the engineering team paid special attention to providing the necessary engine reliability and service life. In the course of actions to achieve the goals mentioned above, the Kolomensky Zavod, one of the first in the Russian diesel engineering, designed and implemented variable blades in the flow parts, as well as air and gas bypass systems that allowed for a sufficient rise in the degree of efficiency. The article also deals with advantages and disadvantages of each of the implemented engineering solutions to design air supply systems. It considers different design measures developed and undertaken by the plant's design bureau to provide operating capacity and durability of turbocharge systems.

Keywords: JSC Kolomensky zavod · Air supply system · Diesel engine · Engine specifications · Supercharge pressure · Supercharger efficiency

1 Introduction

It is known that boosting combined vehicle engines by increasing the supercharge pressure and fuel injection rate results in the increased mean effective pressure, as well as the increased mechanical and thermal tension of diesel components. In this case, maximum pressure and cycle temperature act as limiting factors, as well as standard harmful emission values for exhaust fumes. A perfect diesel engine reaches the highest permissible mean effective pressure at the rated power output mode and maintains it throughout the

entire range of operating crankshaft speeds. This can facilitate a constant torque value at the power output flange and thus, the high adaptability of the engine and the significant simplification of the transmission for the devices driven by the diesel engine.

2 Problem Definition

Assuring constant torque is very complicated due to the limited capacities of air supply systems that prevent them from maintaining the best excess air ratio for a wide range of speeds and loads. The problem is that the existing turbocharge systems with free unadjustable turbocompressors fail to provide sufficient air charging pressure under partial diesel operation modes.

To expand the capacities of turbocharge systems, JSC Kolomensky zavod developed, tested, and experimented with various types of air supply systems featuring different components, designs, and adjustment methods [1]. Early turbocharge systems (installed on such two-stroke diesel engines as 10D100; 14D40, and 11D45) used two-stage compressors [1] where the first stage was a volumetric air blower driven by the crankshaft. The second stage was driven by the turbine with a gas link to the piston block of the engine. These systems provided high throttle response and torques at medium and low crankshaft speeds, yet they were not developed further due to their large sizes and complexity, as well as the low total efficiency.

It should be noted that the control of air supply to the engine cylinders becomes most relevant in such cases, as along with a wide change in load, external conditions are imposed on the engine operation, associated with a change in the back pressure on the gases exhaust and air vacuum at the suction. Most of the above mentioned problems of the considered turbocharging systems can be eliminated by using turbochargers with rotary blades of nozzle apparatuses [2]. In this case, it is possible to provide optimum flow inlet angles to the rotor blades and maintain the turbine power in a wide range of flow rates. To solve this problem, the factory's design department developed a whole range of guide vanes of inlet vanes, air diffusers and nozzle apparatuses designs. According to the conducted researches, the rotation of the compressor blades could increase the flow rate range more than twice.

At the same time, these technical solutions were not devoid of shortcomings. They were convinced of the need for increased clearances to ensure smooth blades rotation, that increased the losses associated with gas overflows. In addition, in the operation of turbochargers with rotor blades, behind-coking of the blades with a loss of their mobility was often observed.

Thus, despite the high control efficiency of pressurization with the power of rotary blades installed in the flow sections of the compressor and turbines, in general these structures turned out to be not statically reliable.

To overcome these problems, JSC Kolomensky zavod developed, tested, and experimented with relatively simpler adjustable turbocharge options with air blow-off to compressor input and gas blow-off behind the turbine [3–6].

Air blow-off, as well as gas blow-off, helps increase the pressure of the air charged into the engine receiver under partial modes due to the initial adjustment of the turbocompressor and turbine air-gas channels to accommodate a higher supercharge pressure than

is required for the rated engine power. The excessive supercharge pressure at rated power output is reduced through the air (gas) blow-off using dedicated controlled discharge valves to prevent unacceptably high combustion pressure in cylinders and the surge under extremely cold conditions. From the viewpoint of loss reduction, gas blow-off bypassing the turbine is more desirable because in this case there are no energy losses due to the compression of the discharged air. The efficiency of such control systems increases with growth of its performance coefficient [7].

Apart from the mentioned air and gas blow-off systems, we researched and put into production a system featuring air blow-off at the turbine inlet. The main advantage of blowing off some of the compressed air (~10%) at the turbine inlet is getting extra mechanical energy to drive the compressor wheel through increasing the efficiency of the turbine by facilitating its operation in the rated duty mode under a wider range of diesel operating modes. The blow-off of some of the compressed air at the turbine inlet increases engine output by 30–40% under medium loads. At the same time, the mixing of rather cool air with exhaust fumes results in the reduction of gas temperature before the turbine, which means it is necessary to increase the pressure to maintain the required stage heat drop to be activated [8]. Increasing the pressure before the turbine leads to poorer cylinder drain and increased pumping losses. Another drawback of this method is that the adjustment range is determined by the range of the operating loads, under which the air charging pressure exceeds the pressure of the fumes before the turbine.

The integration of a dedicated heating device (heat exchanger) using the heat of the gases leaving the turbine to heat the blow-off air into the air blow-off channel is a logical improvement of the reviewed supercharge system. The use of the heat exchanger increases the energy of the exhaust fume and blown-off air mixture, as well as the supercharge pressure rate. This system is referred to as the recuperation system, and was first used on the 16ChN26/26 engine. The recuperation system was tested on a high-power marine diesel and proved to be quite efficient within a relatively narrow power range (60–100% of the rated output). The narrowness of the efficiency range can be explained by the higher resistance of air and gas channels. This fact, as well as the low temperatures of the blown-off air under partial modes, prevents the system from producing the mentioned effect in a wider load range.

Studies show that so-called register turbocharge systems with two or more turbocompressors [4] some of which can be turned off under partial modes are the most desirable option in terms of expanding the operating limits of diesel engines [9]. The first of such systems in Russia was developed for the high-power 16D49 diesel engine. This system comprised two turbocompressors, and the air and gas flows in one of them are shut off when the diesel engine operates at 50–60% load to channel all the exhaust fumes to the turbine of the active turbocompressor. Increasing the gas flow through the active turbocompressor facilitates acceptable gas flow parameters on turbine blades in terms of efficiency, which results in increased rotor speed and, thus, higher supercharge pressure. This engineering solution helps significantly extend the limits, which is crucial for the acceleration of the ship and its operation with a heavy screw. Apart from the advantages of this solution, it was noticed that, when the engine was operated at the upper load limit and the second turbocompressor went on, the power reduced to a certain degree and the engine produced more smoke due to the short-term drop in the air charging

pressure. This pressure drop can be attributed to the delayed response of the rotor and the activated turbocompressor, as well as the reduction of the compressor efficiency due to the operating point entering their lock zone under the abrupt pressure drop in the air and gas channel.

To reduce these negative factors in the operation of the register turbocharged system, we implemented the following solutions:

- Using turbocompressors with high efficiency ratings and reduced rotor response delay.
- Using turbocompressors with different passage areas of nozzle sets. The full-time turbocompressor has a greater passage area of the nozzle set.
- Using controlled chokes in air and gas channels for air and gas blow-off to prevent surging when one of the turbocompressors shuts off and improve the throttle response when increasing the power after the preliminary acceleration of the activated compressor rotor.

The implementation of these requirements helped significantly increase the dynamic and the operating limits of the diesel engine.

Further improvement of the register turbocharge system's efficiency can be achieved by increasing the number of activated turbocompressors. Yet it must be noted that increasing the number of turbocompressors results in a greater number of gas and air ducts, complicates control procedures, and increases the weight and dimensions of the turbocharge system.

Overall, the use of register systems may result in a significant complication of the engine design, their efficiency can compensate for the drawbacks discussed, which can justify a wide practical use of such systems.

The majority of the abovementioned problems related to turbocharge systems can be solved by using turbocompressors with turning vane nozzle sets [2]. In this case, it is possible to achieve the best flow inlet angles for moving vanes and maintain the turbine output under a wide range of flow rates. This solution, however, has some drawbacks as well. One of them is the need to keep greater gaps to accommodate for the free spinning of the vanes, which results in increased losses associated with gas leakage. Besides, turning vanes of such turbocompressors often carburize and stop moving during operation.

Installing a power turbine behind the turbine driving the compressor is an efficient solution that extends the supercharge stability range when the engine power decreases. This method is based on the following: when gas consumption reduces in a series of turbines, the overall stage heat drop also decreases. The greatest reduction of the available heat drop is observed in the last turbine in the series [4]. As a result, the power output of the first turbine driving the compressor remains nearly constant, thus assuring the required air supply in a wide range of diesel operating modes. The power turbine consumes the excessive exhaust fume energy by transferring the power over a mechanical link to the diesel crankshaft.

The drawback of this system is the dependency of the power turbine rotor speed and the crankshaft speed, which affects the efficiency of the turbine. This problem can be solved by using a two-stage turbocharge system where a series of two high and low-pressure turbines drive the compressor wheels of the second and first stage respectively. The absence of a rigid connection between the rotors and the crankshaft facilitates the

self-adjustment of the best proportion of the inflow gas velocity and the circumferential velocity of the turbine vane tip.

An elegant example of using this scheme was the development and implementation by the factory of a two-stage unit on special-purpose engines with a large range of back pressure and suction changes [10, 11]. As already noted, since changes in the heat transition are reflected only in the second stage, the first stage of the turbine part, when the load changes, continues to operate in the design mode with high efficiency, which provides a stable air flow through the engine. In addition, relatively low pressure increases in the compressor stages and, accordingly, low air and gas flow rates contribute to the expansion of the flow range. The advantage of using a two-stage turbine unit for the implementation of controlled pressurization is also that this unit does not contain rotary blades, bypass valves, other regulating bodies that are susceptible to coking, which has a beneficial effect on reliability. Two-stage turbine units were used in the construction of diesel engines' prototypes D56 (CHN32/32). In these units, the rotor of the low pressure stage was connected through a gear train to the crankshaft to provide the necessary excess air coefficient at low frequency modes. Such a scheme made it possible to unwind the rotor at low load modes with the provision of the necessary pressurization and, conversely, from giving extra power from the turbine to the co-roll at high diesel loads. To standardize and reduce the costs of production, we developed a standard series of 2-stage turbine sets [12] that was used by the JSC Kolomensky zavod to design and launch the production of turbocharge sets for 30DG diesel generators, as well as some experimental D49, D56 diesel engines. The use of 2-stage turbine sets in air supply systems helps significantly increase their efficiency (up to 10%), as well as the operating range of internal combustion engines [13].

The designs of two-stage turbine units manufactured by Kolomensky Zavod JSC have a characteristic feature, namely: the stages of low and high pressure turbocompressors are connected by turbine parts in one case. The arrangement of turbocompressors in one housing with the joint of turbine parts allows to minimize the length of the gas path of turbines, eliminating track losses. The sequential connection of the turbine unit stages imposes a number of features when designing it. In particular, due to the higher number of seals compared to a single stage turbocharger, the provision of gas density becomes important. To reduce leaks, spring rings (such as piston rings) are installed on the rotor shaft following labyrinth seals. High pressure differences on the sealing units can lead to rapid wear of the sealing rings and the walls of the streams. To reduce the drop, special drainage is organized with air and gas bleeding. The operability of the seals depends largely on the relaxation resistance of the spring rings. To increase relaxation resistance in turbine units, shutoff air is supplied to rings, thereby reducing the temperature of the ring material and increasing their resistance to deformations. For the same purpose, an insulating material (such as zirconia) is sprayed onto the shielding sleeves. To increase the allowable pressure difference, a double installation of rings in one groove is used. Theoretically, the allowable difference can increase by 100%. For the manufacture of spring sealing rings, special materials with increased relaxation resistance are used at the factory. Due to the fact that in the low pressure stage there is a summation to the rotor from the side of air and gas flows, the axial load on the thrust bearing with an increase in flow rate can increase markedly. In order to reduce the forces, special cavities are

made, which have drainage with the compressor suction cavity. Besides, thrust bearings have special thrust heel on elastic pack of plates providing uniform load on each thrust cushion. The cantilever arrangement of the impellers, as well as the need to maintain minimum axial dimensions, predefine the small distances between the rotor supports, which leads to an increase in loads from unbalance and the manifestation of edge effects in the bearings. Reliability of bearing units in these conditions is ensured by the use of special vibration-resistant bearings with self-aligning inserts that exclude edge wear [14]. In such bearings, a hydrostatic damping layer is created on the occipital part of the self-aligning inserts, which dissipates the vibration energy and provides a good heat removal. The possibility of self-installation of inserts paralyzes the occurrence of self-oscillations on the oil layer and provides a long engine resource. The use of two-stage pressurization at the modern stage of the diesel engineering development is due to the availability of a second compressor stage in such units provides unlimited opportunities for creating extra-high air charging pressure (up to 10 bar). The availability of extra-high air charging pressure helps further increase the power output of engines using mean effective pressure and reduce harmful emissions, including by the implementation of Miller's cycle [15–20].

3 Theoretical Part

Miller's cycle, stipulating early closing of inlet valves, can be efficiently used in combination with a 2-stage turbocharge because of the following:

- Intercooling reduces air temperatures at the beginning of the compression phase. The reduction of air temperatures at the beginning of the compression phase results in a lower maximum cycle temperature, which significantly reduces the formation of harmful emissions of nitrogen oxides NO_x
- Due to their higher efficiency, 2-stage turbochargers have a higher positive pressure differential during intake, which results in higher mean efficient pressure values.
- The longer expansion of the actuating medium in two turbines instead of one results in higher cycle efficiency.

The selection of the best air pressure for the developed Miller cycle diesel with a 2-stage supercharge is somewhat different from the strategy used in designing new 1-stage supercharge engines. The difference between these approaches is that 1-stage supercharge engines are subject to a dependency between the mean efficient pressure and the required air charging pressure. This dependency can be used when designing a new engine taking into account the operating limits of 1-stage turbocompressors.

As for designing Miller cycle engines with 2-stage turbocharge, it is incorrect to use the said dependency between the mean efficient pressure and the required air charging pressure. This fact can be explained by the significant impact of the inlet valve closing angle on the hydraulic parameters of the diesel and the operating parameters.

To assess the impact of the inlet valve closing angle on the parameters of the new 12ChN26,5/31 Miller cycle engine developed at JSC Kolomensky zavod featuring 2-stage turbocharge, we performed calculations with constant values of power output

and maximum combustion pressure [5]. The Miller cycle inlet valve closing angle was initially assumed to be $M = 60^\circ$ before bottom center. (The calculations were carried out using the DizeL-RK software package). Some of the calculation results are shown in Fig. 1. As we can see from the figure, for this inlet valve closing angle $M = 60^\circ$, there is a clear fuel rate minimum with a specific degree of pressure increase (in our case, $\pi_k = 6.5$). At the same time, nitrogen oxide emissions do not have such minimum when π_k is changing, and their value reduces almost linearly as π_k increases.

The data obtained make us assume that each inlet valve closing angle value should have a specific corresponding pressure increase degree facilitating the fuel consumption minimum. Therefore, we calculated the best combination of inlet valve closing angle value and the degree of pressure increase in the compressor that facilitates the lowest fuel consumption possible along with acceptable nitrogen oxide emission values for the diesel engine in question.

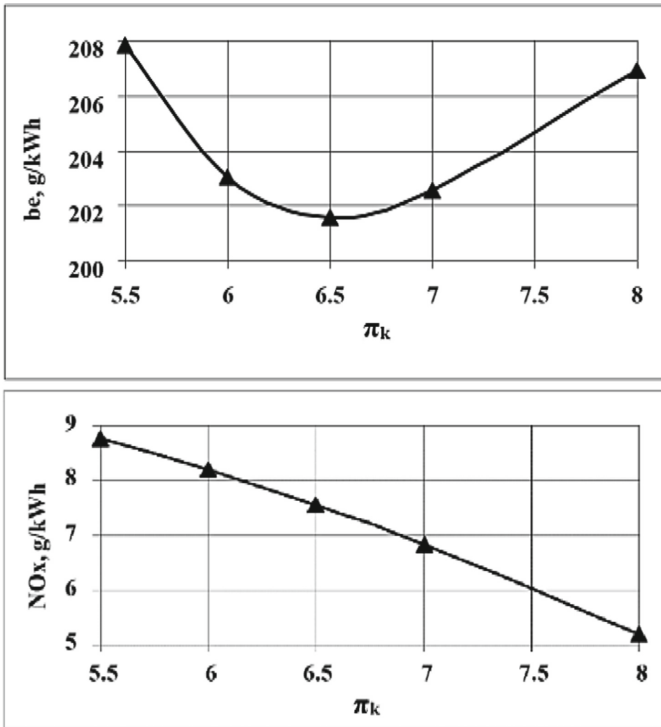


Fig. 1. The relationship of the specific fuel rate b_e and the emissions of nitrogen oxides NO_x and the degree of pressure increase π_k for Miller’s cycle with inlet valve closing angle $M = 60^\circ$ before bottom center.

Figure 2 shows the results of the calculations for inlet valve closing angles $M = 30, 40, 50,$ and 60° before bottom center over the change range of the pressure increase degree $\pi_k = 4 \div 8$.

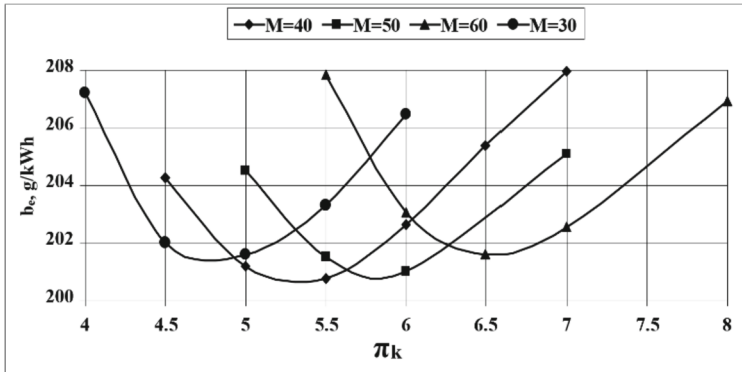


Fig. 2. The relationship of the specific fuel consumption b_e and the emissions of nitrogen oxides NO_x for various combinations of inlet valve closing angle M and pressure increase degree π_k

The analysis of the results showed the following:

- The pressure increase degree $\pi_k = 5.3 \div 5.5$ and the inlet valve closing angle $M = 40^\circ$ before bottom center facilitate the lowest specific fuel consumption possible.
- The pressure increase degree $\pi_k = 5.8 \div 6.0$ and the inlet valve closing angle $M = 40^\circ$ before bottom center facilitate the greatest reduction of NO_x emissions with a moderate increase in fuel consumption.

4 Practical Significance of Proposals and Implementation Results

Our calculations show that using a 2-stage turbocharge with Miller's cycle on a 12ChN26,5/31 diesel at $M = 40^\circ$ and $\pi_k = 6.0$ compared to a 1-stage turbocharge 12ChN26,5/31 diesel may reduce the emissions of nitrogen oxides as per GOST 31967-2012 and simultaneously reduce fuel consumption by 1.5 g/kWh while operating at 52% load.

Figure 3 shows the arrangement of a 12ChN26,5/31 diesel with a 2-stage turbocharge system.

According to the image shown, the air enters the low-pressure compressors running in parallel at 1, and then it is fed through nozzles 5 to intercoolers 4. After cooling, the air is channeled to the inlet of the high-pressure turbocompressor 2. After secondary compression, the air enters the aftercooler 7, after which the cooled air is fed into the diesel receiver. The fumes from the exhaust manifolds of the diesel are fed over the connector nozzle 8 to the high-pressure compressor turbine and then to low-pressure turbines over gas pipes 9 where it is discharged after operation into the exhaust system. The connector nozzle 8 has a bypass option for the fumes from the manifolds to be channeled directly to the inlets of the low-pressure turbines instead of the high-pressure turbine.

By analogy with the algorithm of optimum diesel settings determination given above, calculations for optimization of a two-level air supply system with Miller's cycle for the 3275 kW diesel 16ChN26/28 were executed.

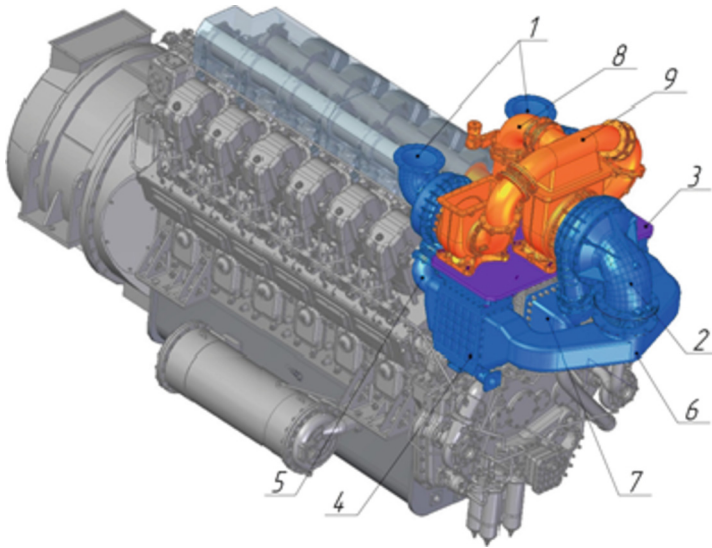


Fig. 3. The arrangement of a 12ChN26,5/31 diesel generator with a 2-stage turbocharge system.

As the calculations showed, the use of a 2-stage turbocharge with Miller's cycle instead of a single-stage turbocharger on this diesel engine allows reducing the emission of nitrogen oxides NO_x from 12.26 g/(kWh) to 7.14 g/(kWh), thereby ensuring

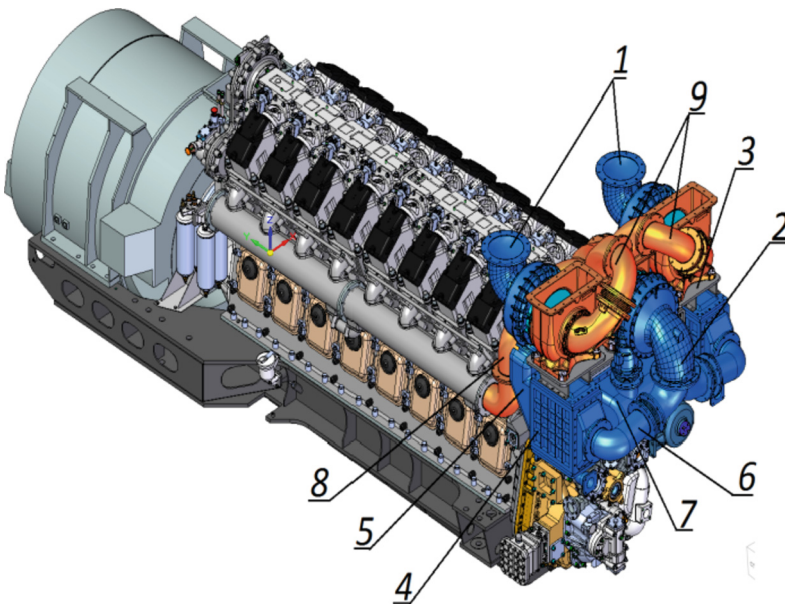


Fig. 4. The arrangement of a 16ChN26/28 diesel generator with a 2-stage turbocharge system.

the requirements of GOST 31967-2012 (7.4 g/(kWh). At the same time, it should be noted that due to the selection of optimal diesel settings and higher efficiency of the turbocharging system, it was possible to reduce specific fuel consumption by ~1 g/kWh at the rated power mode.

Figure 4 shows the arrangement of a 16ChN26/28 diesel with a 2-stage turbocharge system.

5 Conclusions

As the theoretical studies have shown, the fulfillment of the GOST 31967-2012 requirements to ensure an acceptable level of harmful emissions in diesel engines can be effectively fulfilled through the use of the Miller cycle. In this case, the boost required for this cycle can be realized by using double-stage units. This technical solution allows not only to fulfill the requirements for harmful emissions, but also significantly improve the efficiency of diesel engines due to the better efficiency of two-stage turbocharging. The above layout solutions for placing two-stage supercharging on diesel engines show sufficient compactness of these engines for the placement in the engine room of various objects (on ships, diesel locomotives, etc.).

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