Increasing the accuracy of a marine diesel engine operation limit by thermal factor

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Abstract. The experience of using forced marine diesel engines revealed the problem of their operation in a wide range of high-speed operating modes with the prevention of thermal stress overloads: when setting partial speed modes of the main diesel engine with gas turbine charging, an increase in thermal loads and incomplete combustion with increased exhaust gas opacity are observed. This occurs often due to the diesel load limiters being set to the universal limit constraint that has been used in less boosted designs. Often, to reduce the likelihood of thermal overloads, the universal limiting dependence is shifted in parallel towards the "wider" margin, but, in fact, this leads to underutilisation of the main engine power. To evaluate the efficiency of the linear limiting dependence used in the limiter by numerical modelling at partial modes of the main diesel engine HYUNDAI-MAN B&W 6G70ME-C9.2-TII of the large-capacity tanker "GOLDWAY", the permissible fuel supply was determined at five steady running modes of the engine. The permissible values were meant to be those at which the temperatures of the gases on the expansion line did not exceed the design threshold values. The scientific novelty is determined by the fact that the temperature of the gases on the expansion line was taken as an integral criterion for limiting the fuel supply by thermal stress. The practical significance of the study is conditioned by the fact that with the help of the obtained limiting characteristic in terms of thermal stress, the power range of the diesel engine in steady modes without overloading was expanded.

Keywords: maritime transport, internal combustion engine, numerical simulation, thermal stress, accuracy.

1. Introduction

The main engines (ME) of vessels are operating in a wide range of propulsion modes, the loads on which change not only in proportion to the square of the shaft speed, but also due to changes in the draft and trim of the vessel, the depth under the keel, the degree of fouling of the hull and propeller, sea roughness, direction and strength of wind. This means that for each given shaft speed, the permissible fuel supply must be determined, at which diesel engine overloads due to thermal stress do not occur. The dependence of the permissible fuel supply on the shaft speed is called thermal stress limiting. On vessels with fixed pitch propellers (FPP) this limiting dependence is implemented by a limiter in the shaft speed governor, on ships with controllable pitch propellers (CPP) – by a load governor acting on the propeller pitch. The more precisely the dependence of the permissible fuel supply on the diesel engine speed is determined and, accordingly, maintained, the wider range of operating modes (without overloads) of the main marine engines. Expansion of the range of non-overload operation modes of main diesel engines improves the economic performance of vessels.

On modern vessels, propulsion units are used, the working processes of which are more "sensitive" to operating conditions, including changes in air temperature in the engine room, sea water in the cooling system, and coking of the exhaust gas tract. Despite the use of modern load limiters, the problem of operating the main marine diesel engines in a wide range of high-speed operating modes with the prevention of thermal stress overloads has emerged. As evidenced by practice, with a decrease in the shaft speed of the main diesel engine with gas turbine charging, an increase in thermal loads occurs, incompleteness of combustion with a pronounced increase in exhaust smoke. This is often conditioned by the fact that the diesel load limiters are tuned to the universal constraint obtained for structures with less boost. Often, in order to reduce the likelihood of thermal overloads, the universal limiting dependence is shifted in parallel towards the "wider" margin, but, in fact, this leads to underutilisation of the main engine power.

With an increase in the aggregate power of diesel engines, due to the forcing of the working process, the loads on the parts of the cylinder-piston group (CPG) and turbocharging units increase significantly. The resource indicators of which significantly decrease with an increase in the temperatures of the working fluid [1]. This causes an increase in the temperature of rubbing surfaces in the mating parts, change in the temperature regime of the oil film, and, consequently, deterioration in the quality of the lubricant [2]. Therefore, it becomes necessary to analyse the features of the stress-strain state of components [3]. The thermal stress of forced diesel engines can be reduced by decreasing heat removal to the walls of the combustion chamber [4], which is often achieved by changing the closing period of the exhaust valves, increasing the excess air ratio, intermediate air cooling or thermal insulation of the walls of the combustion chamber [5].

For turbocharging diesel engines, the power limitation requires a comprehensive account of the peculiarities of the operation and the tension of engine parts [6]. Therefore, the stresses due to high temperatures of the working fluid and parts of the CPG become decisive when assigning the range of permissible operating modes of the diesel engine. The limits of the range of permissible operating modes of modern diesel engines are determined by the temperature of uncooled units and engine parts [7]. However, in practice, universal limiting dependencies are used for the main marine diesel engines, which do not always coincide with the actual ones. The latter can be estimated using well-known mathematical models that provide a good adequacy of the reproduction of engine cycles. Clarification of the limiting characteristics will expand the permissible operating modes of diesel engines [8].

Therefore, the purpose of the study is formulated: to improve the accuracy of limiting the operation of the main marine diesel engine. The implementation of this objective should improve the economic performance of vessels by expanding the range of non-overload operation modes of the main diesel engines.

2. Materials and Methods

To solve the problems of predicting various operating modes and characteristics, taking into account changes in the conditions of the operating processes of a diesel engine, it is possible to use methods of mathematical modelling, where the elements of an internal combustion engine can be represented as open gas-dynamic systems exchanging mass and energy [9]. The gas parameters in these systems are described by the differential equations of conservation of mass, energy, momentum and the equation of state. Using a mathematical model with a 0-dimensional problem statement, i.e., with neglect of the pressure difference in the cylinder by volume, greatly simplifies the calculation and does not introduce noticeable errors in the results [10]. With this thermodynamic approach, the engine is considered as an open thermodynamic system, or as a collection of several thermodynamic systems. The velocity field is not considered in it, the pressure and temperature inside each thermodynamic system are considered independent of coordinates, but dependent only on the angle of rotation of the crankshaft [11]. The diesel engines operating on FPP are in the most difficult modes of operation, since in partial modes the propeller requires torques that significantly exceed the torques of the propeller characteristic [12; 13].

The study was carried out using mathematical modelling according to the method [14] of numerical modelling of the Central Research Diesel Institute (CNIDI)

[15]. In an improved form, namely in the part of the description of the working processes occurring in the diesel manifolds and the gas turbine supercharger, as well as in unsteady operating modes, it is published in [16]. With regard to vessel propulsion system, the modelling technique was adapted from [17; 18], which differs from the previously used ones in the following:

– working processes in the diesel cylinder and collectors are described by the differential equations of the first law of thermodynamics, mass balance and state, which are solved at each step of the crank rotation (the step when the crank is rotated is automatically adjusted in the range from 0.1 to 4 degrees, depending on the intensity of the processes);

– the mixture of gases in the diesel cylinder and in the manifolds is considered as consisting of two components – air and "clean" combustion products, the ratio of which changes at each step of the calculation;

– when calculating the purge process, a two-stage nature of the interaction of the purge air with gases in the cylinder is assumed. That is, it is believed that at the beginning of the purge, the interaction of air with gases is in the form of layer-by-layer displacement of gases, and complete mixing at the end of the purge. Moreover, the law of transition from layer-by-layer displacement to complete mixing is taken exponential;

– the calculation takes into account the change in hydraulic losses (depending on the flow rate) at the compressor inlet, in the air cooler and behind the turbine. The change in heat losses (depending on the air flow rate) in the air cooler is also taken into account. The conversion of kinetic energy of gases into potential is taken into account in the exhaust manifold. The cylinder takes into account the heat exchange of gases with the cylinder walls.

The latest revision of the technique is presented in [19] with improvements in terms of:

– description of the operating process of a diesel engine, which is considered as a set of interrelated processes by universal balance ratios that are valid at all stages of the working cycle and in all volumes of the gas-air tract;

– determination of arrays of values of flow rates and enthalpies of gases leaving the cylinder, depending on the angle of crank rotation, which allows clarifying the calculation of exhaust processes from adjacent cylinders, united by an exhaust manifold (exhaust processes from adjacent cylinders are calculated by phase displacement from the design cylinder by the angle of crank rotation, equal to the quotient of dividing the crank angle per cycle by the number of cylinders combined into a manifold);

– description of the purge process, which is calculated using a three-zone submodel that adopts the individual provisions of the methodology [20], in which it is assumed that when the cylinder is purged, three zones arise: purge air, mixing of purge air with residual gases, and residual gases. It is also assumed that the purge air enters the first and second zones in a predetermined proportion, and the zone of residual gases is first displaced through the exhaust bodies, and then simultaneously the first and second zones.

The foregoing allowed minimising the use of empirical and semi-empirical dependencies. They are used only to describe the law of fuel combustion, to determine the autoignition delay period of the fuel and the coefficient of heat transfer from gases to the wall as well as when modelling modes different from the original one to calculate the change in fuel combustion indicators, the efficiency of the compressor and turbine.

To simulate and adjust the mathematical model, the design parameters [21], the data of sea trial, as well as the following parameters, measured specifically for modelling using standard control and measuring instruments, were used:

– maximum pressure in the cylinder was determined using the Kistler piezoelectric sensor of the PMI Auto tuning 1410-v1.42.0.11 stationary diagnostic system;

– angle of crankshaft rotation was determined by a magnetic type sensor;

– position of the exhaust valve was measured by the ZT-4111 position sensor;

– torque and effective shaft power measured by the stationary Sea Performer System (CE-HW-01-I5) from Enamor Ltd. with ETM-E-Torque meter;

– fuel supply index was determined using the remote control unit (RCU) K-Chief, R/N 98530, S\N 992117;

– air boost pressure in the receiver and constant outlet backpressure were measured by a pressure gauge with a combined input type EN 837-1 manufactured by WIKA, Poland, with a measurement range of 0…6 kgf /cm^2;

– turbocharger speed was determined by a 73RS-DM digital tachometer type (DC24V, 190 mA) from Applied Electronics Corporation, Japan;

– temperature of the gases in front of and behind the turbine, at the outlet from the cylinders, was measured with Tempress thermometers from Keumyang Ind. Co., Ltd., Denmark, measuring range 50…650 ^{0} C;

– temperature of the water cooling the outlet manifold and the cylinder at the outlet, measured with thermometers of type 5/1 manufactured by Sinwoo Co., Ltd., with a measurement range of 0…200 ^{0} C;

– temperature of the charge air before and after the cooler was measured by thermocouples R/B HR 101 (R2), type PT100, manufactured by Heriana Co., Ltd., with a measurement range of 0…200 ^{0} C;

– temperature of the cooling water at the inlet and outlet to the air cooler was measured with a 2/1 thermometer manufactured by Sinwoo Co., Ltd., with a measurement range of 0…120 ^{0} C;

– pressure loss in the air cooler was measured by a U-tube manometer of the HMA 10 type from Hyundai Instruments Co., Ltd., with a measurement range of ± 220 mm Aq;

– pressure loss at the compressor inlet and behind the turbocharger was measured by U-tube pressure gauges DY MANO 150 made by Daeyoug Instrument, measuring range of ± 150 mm Aq.

3. Results and Discussion

One or several parameters can be selected as criteria for assessing the thermal stress, including the average indicated or average effective pressure, torque on the engine shaft, maximum combustion pressure, exhaust gas temperature, and excess air ratio. In practice, the temperature of the exhaust gases is often used due to the fact that it can be easily measured. However, the temperature of the exhaust gases does not always objectively characterise the thermal stress. It happens that at a normal value of the exhaust gas temperatures, the engine is thermally overloaded. This occurs when the engine under operating conditions is operating on an external characteristic. For engines with increased degrees of boost, operation according to the external characteristic is allowed only in an area close to the nominal speed.

According to [22], in diesel engines, if an increase in temperature is detected at the end of fuel supply or after by a certain number of degrees, then the tendency to increase by this number of degrees remains until the start of release. This means that it is sufficient to control the temperature at any selected crank angle after the end of the injection. For this, the maximum cycle temperature can also be used, since the wall temperature, temperature difference; temperature stresses and the heat load of the CPG depend on it. Taking into account the above, the temperature of the gases on the expansion line was taken as a criterion for limiting the fuel supply by thermal stress. In the shipboard conditions it is rather difficult to obtain accurate values of the gas temperature on the expansion line, even if modern systems for diagnosing the working process are used. Thus, the numerical simulation of the diesel engine working processes was carried out, but with the maximum adjustment of the calculation results to the actual operating parameters. The coincidence is about 20 calculated parameters of diesel engine with experimental will provide an adequate determination of the temperatures of gases on the expansion line.

The object of the study was the main engine HYUNDAI-MAN B&W 6G70ME-C9.2-TII of the tanker "GOLDWAY", equipped with a MITSUBISHI HEAVY INDUSTRIES turbocharger of the MET 60MB series. This engine develops a rated power of 15,088 kW at a rated shaft speed of 71.8 min^{-1}, is a low-speed, two-stroke, crosshead, reversible and direct drive to a FPP. The measurement data are shown in Table 1, presented in the format recommended by [23].

Table 1. Experimental indicators of working processes of the main marine diesel engine HYUNDAI-MAN B&W 6G70ME-C9.2-TII

Sr. No.	Indicator		Experimental value at diesel power				
			25 %	50 %	75 %	85 %	100 %
1	Main engine speed, min^{-1}		45.2	57.0	65.0	68.0	71.8
2	Fuel supply index, conventional units		43.0	64.0	84.0	90.0	100.0
3	Turbocharge rotation, min^{-1}		7049	11251	13397	13556	14343
4	Air pressure at the cooler inlet, bar		0.43	1.60	2.64	2.69	3.08
5	Air pressure at the cooler outlet, bar		0.50	1.62	2.68	2.73	3.13
6	Air temperature at the cooler inlet, ºC		52	110	156	163	176
7	Air temperature at the cooler outlet, ºC		19	24	26	29	32
8	Pressure drop across the air cooler, mm WG		40	105	156	160	180
9	Turbine inlet temperature, ºC		253	319	355	381	421
10	Turbine outlet temperature, ºC		182	180	189	208	247
11	Maximum pressure in cylinder No. 6, kgf/cm^2		104.9	141.5	167.3	184.0	185.7
12	Gas temperature at the cylinder outlet, ºC	No. 1	161	238	289	321	370
		No. 2	168	228	273	310	359
		No. 3	191	257	301	331	380
		No. 4	189	251	293	321	369
		No. 5	190	256	309	332	378
		No. 6	191	257	308	329	375

The measurements were carried out in the following conditions: the draft of the bow and stern – 17.28 m each; air temperature in the engine room – 16.2 °C; air pressure in the engine room – 101.3 kPa; seawater temperature – 21.1 °C; sea agitation – 3 points; IFO – 120 fuel (a mixture of 11.88 % Gas Oil Marino and 88.12 % Fuel Oil 380). During the operation of the main marine engine HYUNDAI-MAN B&W 6G70ME-C9.2-TII, a smoky exhaust was detected at low speeds of the vessel, which indicates thermal overloads. As a result of the analysis, the improper performance of the KONGSBERG AutoChief 600 (AC 600) limiter with a standard linear limiting dependence was noted. The limiting performance can only be adjusted in ECS (Engine Control System), such as ME-C-ECS from MAN DIESEL & TURBO SE, version 1312-2.1. Table 2, presented in the format recommended in [23], shows the simulation data for five steady engine running modes.

Table 2. Estimated indicators of working processes of the main marine diesel engine HYUNDAI-MAN B&W 6G70ME-C9.2-TII

Sr. No.	Indicator	Estimated value at diesel power				
		25 %	50 %	75 %	85 %	100 %
1	Maximum cycle pressure, kgf /cm^2	105.1	141.4	167.4	183.9	185.7
2	Maximum cycle temperature, K	1746.6	1710.6	1688.9	1681.2	1678.9
3	Temperature in the cylinder at the start of release, K	620.3	620.7	638.1	655.6	663.5
4	The share of heat removed to the cylinder walls per cycle,%	20.5	17.0	15.3	14.3	13.9
5	Average conditional temperature of the cylinder walls, K	501.6	537.1	565.0	584.9	593.0
6	Actual excess air ratio	2.133	2.313	2.457	2.493	2.562
7	Total excess air ratio	2.372	2.577	2.792	2.824	2.845
8	Average turbine inlet temperature, K	536.2	602.7	637.6	657.7	695.3
9	Total indicated efficiency	0.497	0.497	0.499	0.500	0. 499

To evaluate the efficiency of the linear limiting dependence used in the limiter by numerical modelling in partial modes, the permissible fuel supply (fuel pump indices) in the range of shaft speed 45.2…71.8 min^{-1} were determined. The permissible values were understood to be those at which the temperatures of the gases on the expansion line (respectively, the temperatures in the exhaust manifold) did not exceed the values at the nominal mode.

Figure 1 shows the limiting relationships used in the AutoChief 600 (AC 600) limiter is linear and calculated at a constant maximum cycle temperature by numerical simulation. Figure 2 compares the indices of the fuel pumps with the used and calculated limiting characteristics and in percentage terms it is determined that an excess amount of fuel is supplied to the engine at modes: 25% of power – 2 %; 50 % of power – 7 %; 75-85 % of power – 3 %.

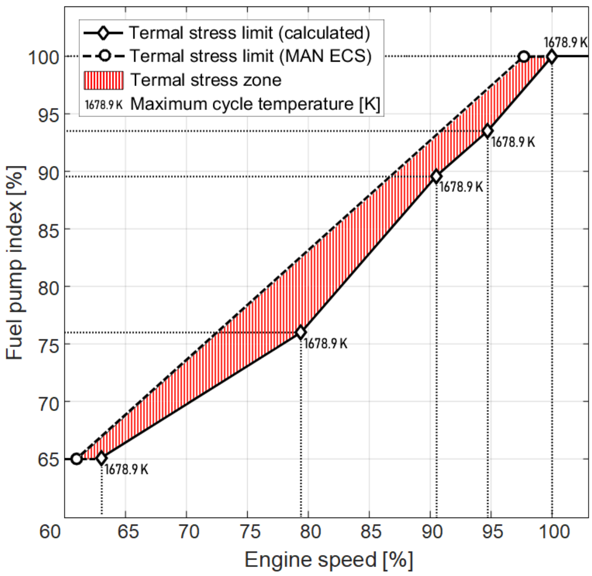

Fig. 1. The used and calculated limiting characteristics of a diesel engine at a constant maximum cycle temperature in terms of thermal stress

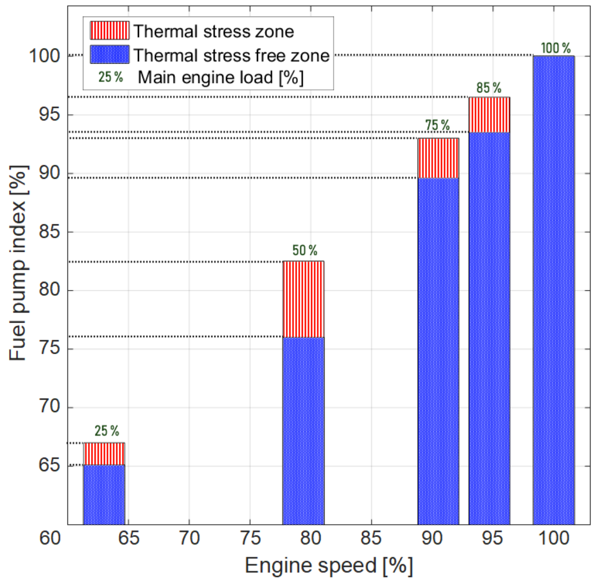

Fig. 2. Comparison of fuel pumps indices with the used and calculated limiting characteristics of a diesel engine in terms of thermal stress

The standard load limiter, when using a linear dependence, is adjusted according to the diesel thermal intensity at the nominal mode (i.e., at the nominal speed, it limits the nominal fuel supply by adjusting the shaft speed). If the second point of the linear dependence is chosen with a decrease in the speed of rotation to 50-60 % of the nominal and to ensure there is no overload at this point, then in the interval between these points the real limiting dependence with a deflection turns out to be below linear, which means the diesel engine is overloaded in this interval. The vessel's crew will inevitably notice that temperatures rise excessively during operating conditions (within the specified interval) and will set the load limiter indicator not to 100 %, but to 80-85 %. By this action, as shown in Fig. 3, the linear dependence will move in parallel downward, which will significantly narrow the area of permissible operation of the diesel engine.

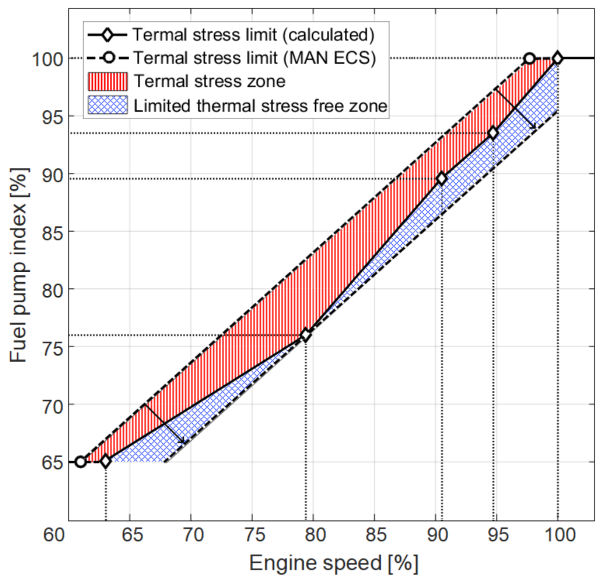

Fig. 3. Calculated and used linear limiting characteristic moved parallel downward

Figure 4 compares the indices of the fuel pumps with the calculated and used linear limiting characteristic moved in parallel downward. As a percentage, it is determined that an excess amount of fuel is supplied to the engine in the following modes: 25 % of power – 4 %; 75-85 % of power –3 %; 100 % of power – 5 %. That is, the use of the universal limiting characteristic used in the limiter narrows the ranges of operating modes due to the fact that it is necessary to "reinsure" against motor overloads and limit more than necessary.

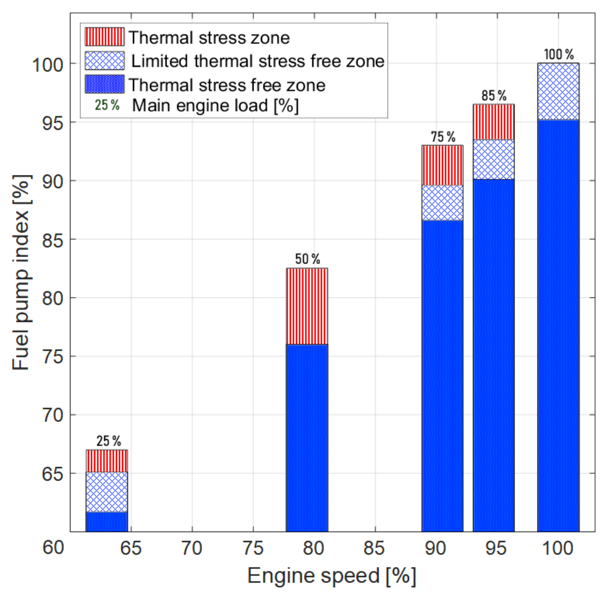

Fig. 4. Comparison of the indices of fuel pumps with a calculated and used linear limiting characteristic moved in parallel downward

The second option for correcting the situation by the crew is to tilt the restrictive dependence more, until the pointer remains at values close to 100 %. But the range of possible modes of the diesel engine will narrow again and the engine will operate with a poor load-to-speed ratio, which means excessive fuel consumption. In Fig. 5, a variant with a modified slope of the used linear limiting characteristic is considered.

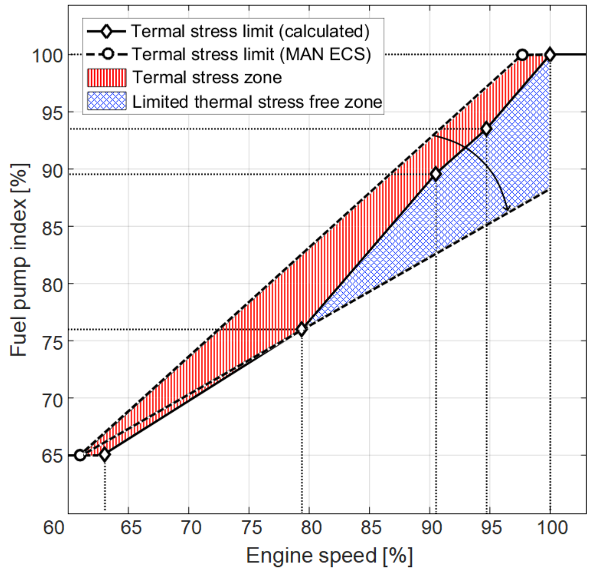

Fig. 5. Calculated and sloped linear characteristic for thermal stress limit

In Fig. 6 compares the indices of the fuel pumps with the calculated and used linear limiting characteristic, which is sloped. As a percentage, it was determined that the ranges of engine operating modes narrowed according to the fuel pump indices when operating at the following modes: 75 % of power – 7 %; 85 % of power – 8 %, 100 % of power – 12 %.

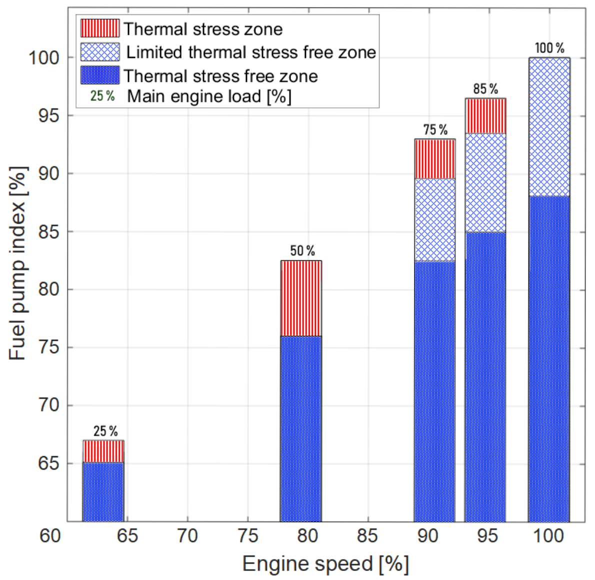

Fig. 6. Comparison of the fuel pumps indices with the calculated and used linear limiting characteristic with a modified slope

Since the standard load limiter can only implement linear dependences, it is proposed to approximate the dependence recommended for adjustment by four linear sections. The use of this dependence will expand the range of partial operating modes of the diesel engine without overloading it.

Conclusions

As an integral criterion for limiting the fuel supply in terms of thermal stress, it is recommended that the gas temperature on the expansion line, which should not exceed the nominal value in all partial operating modes. Using this condition, modelling can determine the permissible fuel supply in the entire range of speed modes of the main diesel engine. Based on the finding of the study, it was concluded that the standard linear limiting dependence at any of its slopes either does not prevent diesel overloading, or (if the crew introduces an additional limitation) narrows the range of operating modes of the diesel engine, which is accompanied by excess fuel consumption. The study recommends the calculated limiting characteristic for use in the load limiter AutoChief 600 (AC 600), the propulsion complex of the large-capacity tanker "GOLDWAY" with the HYUNDAI-MAN B&W of the 6G70ME-C9.2-TII series main marine diesel engine.

According to the findings, it is proposed to approximate the calculated limiting dependence in terms of thermal stress by at least four linear sections, which will significantly expand the range of operating modes of the main diesel engine compared to the standard setting of the AutoChief 600 (AC 600) load limiter. Operational displacement of the standard linear limiting dependence of the AutoChief 600 (AC 600) load limiter, both in parallel and by changing its slope, either does not prevent diesel overload, or unreasonably narrows the range of operating modes of the main diesel engine.

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None.

Conflicts of interest

The authors have no conflicts of interest to disclose.

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