Performance Simulation of Ship Propulsion System with Slow Speed Diesel Engine under Aggravated Conditions

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The doctoral thesis *Simulation of performance of the ship propulsion system with slow speed diesel engine under aggravated conditions* has resulted from years-long research and development work in the field of mathematical and computer-aided simulation model for a ship propulsion diesel engine. Writing and publishing scientific papers and reports have contributed to guidelines for creating a computer-aided model using the software application MATLAB 7.0 – SIMULINK. In this way, the possibility of connecting with already formulated and accessible models for control system elements has been made as well as usage of numerical procedures for forming and optimising inlet and outlet numerical data.

The model has been applied to the propulsion system of a tanker built in the *Brodosplit* shipyard, powered with the diesel engine MAN B&W 6S50MC. Enquiry into technical documentation and all testing that was essential for this paper were carried out in the Split diesel engine plant *Tvornica dizelskih motora – Split* during the testing of the engine at the test bed and during the ship's trial trips.

The model is applied in the analysis of static and dynamic operating conditions of a turbocharged slow-speed diesel propulsion engine with a fixed blade propeller. Special attention has been paid to examining the stability and availability under aggravated operating conditions such as load reduction in one of the cylinders or difficulties in scavenging and the turbocharging system. The analysis of the results has identified ultimate limits of affecting features and has defined safe propulsion conditions, in particular those related to the speed regulation and turbo charging systems.

Simulacija rada brodskog propulzijskog sustava sa sporohodnim dizelskim motorom u otežanim uvjetima

Izvornoznanstveni članak

Doktorska disertacija *Simulacija rada brodskog propulzijskog sustava sa sporohodnim dizelskim motorom u otežanim uvjetima* nastala je kao rezultat višegodišnjeg istraživanja, te rada na razvijanju matematičkog i računalno-simulacijskog modela brodskog propulzijskog dizelskog motora. Izradom i publiciranjem znanstvenih članaka i referata, postavljene su smjernice za izradu računalnog modela koristeći računalnu aplikaciju MATLAB 7.0 – SIMULINK. Na taj način postavljena je mogućnost povezivanja s već izrađenim i dostupnim modelima elemenata regulacijskih sustava, te korištenje numeričkih procedura za oblikovanje i optimiranje ulaznih i izlaznih numeričkih podataka.

Model je primijenjen na propulzijskom sustavu jednog tankera izgrađenog u brodogradilištu *Brodosplit* s pogonskim dizelskim motorom MAN B&W 6S50MC.

Uvid u tehničku dokumentaciju, te sva ispitivanja, koja su bila potrebna za izradu ovoga rada obavljena su u *Tvornici dizelskih motora-Split*, tijekom ispitivanja motora na ispitnom stolu, te tijekom probnih vožnji broda.

Model je primijenjen za analizu stacionarnih i dinamičkih uvjeta rada, sporohodnog propulzijskog dizelskog motora s prednabijanjem i brodskog vijčanog propulzora s nepomičnim krilima. Posebno je ispitana stabilnost i raspoloživost u otežanim uvjetima rada, kao što je smanjenje opterećenja na nekom od cilindara ili poteškoće u radu sustava ispiranja i prednabijanja cilindara. Analizom rezultata utvrđene su krajnje granice utjecajnih značajki, određena su sigurna pogonska stanja, posebno sustava regulacije brzine vrtnje i sustava turbopuhala.

Symbols/Oznake			**Indices/Indeksi**		
m	-	mass, kg	ex	-	external
	-	masa		-	vanjski
M	-	torque, N·m	g	-	fuel
	-	moment		-	gorivo
n	-	revolution speed, min^{-1}	ik	-	exhaust receiver
	-	brzina vrtnje		-	ispušni kolektor
P	-	power, W	k	-	compresor
	-	snaga		-	kompresor
p	-	pressure, Pa (bar)	proc.	-	process
	-	tlak		-	proces
$Sfoc$	-	specific fuel consumption, g/(kW·h)	r	-	rack
	-	specifična potrošnja goriva		-	poluga
t	-	time, s	t	-	turbine
	-	vrijeme		-	turbina
T	-	temperature, K	tp	-	turbocharger
	-	temperatura		-	turbopuhalo
V	-	volume, m^3	uk	-	suction receiver
	-	zapremina		-	usisni kolektor
η	-	efficiency			
	-	efikasnost			
λ	-	air excess ratio			
	-	pretičak zraka			
π	-	pressure ratio			
	-	omjer tlakova			
X	-	shift, mm			
	-	pomak			
u/c	-	speed ratio			
	-	omjer brzina			

1. Introduction

Transient phenomena caused by significant load changes followed by considerable oscillations, which can take place under aggravated sailing conditions, result in exposure of engine parts to heat load and/or incomplete combustion. Soot can be expected as well. Besides its adverse impact on the environment, soot can settle in exhaust gas passages and lead to fire.

When sailing in heavy seas the propulsion engine may experience a sudden load reduction, which may activate the overspeed shut-down system in order to prevent mechanical damage to the system. This would actually imply a brief loss of ship propulsion and, consequently, a safety hazard. In practice this is prevented by reducing revolution speed so that the ship loses speed, as it is under considerably increased loads; rolling increases and, in some instances, all these elements may lead to the master's decision to alter the course and find a most suitable one. In addition to the aggravated external operational conditions affecting the ship's propulsion, failures of certain sub-

systems are likely to occur. Reduced loads in some of the cylinders result from combustion anomalies, reduced or interrupted fuel supply, difficulties in scavenging and turbocharging system, etc.

In the master thesis [1] the author presented the simulation model for the ship's propulsion system with slow speed diesel engine. The model is based on mathematical engine models presented by Jankov [2] and Medica [3]. The author's doctoral thesis [4] provides a detailed description of the model which has been extended and adapted to the ship propulsion system with slow speed diesel engine. The simulation model was thoroughly tested through comparison with measurements performed in the real system. Impacts of load changes in some of the cylinders on the engine performance were analysed in detail on the tested model. The load change was simulated by interrupting fuel supply to one of the cylinders at various engine loads. Impacts of step engine load change were examined as well. Analyses were carried out for various control system designs in order to select the most suitable one. In the simulation model

featuring the control system selected as the most suitable, we have carefully analysed the engine performance at external load changes occurring when sailing in heavy seas. In addition to the engine performance, we have also analysed the turbocharger performance in order to determine, as accurately as possible, the availability of the ship`s propulsion system under aggravated conditions. We have performed a detailed analysis of impacts of various air temperatures at the compressor suction as well as impacts of the turbine contamination. This paper presents an outline of research results from the defended doctoral dissertation [4].

2. Application of the numerical simulation model

The simulation model that was discussed in detail in the thesis [4], was applied to the propulsion system of a tanker conveying chemicals. The propulsion engine is a reversible turbocharged slow speed diesel engine with a very long stroke driving a fixed pitch propeller. The engine bearing the code type 6S50MC was manufactured in the diesel engine factory in Split under MAN B&W licence. The paper presented all features of the selected propulsion system which were essential for the simulation model, as well as the course of the examination of the model's validity, comparing the obtained values with the ones measured in the real engine. Besides examination at the stationary operating points at 25 %, 50 %, 75 %, 93,5 %, 100 % and 110 % of the load, we performed analyses during transient phenomena related to load change up to the boundary conditions.
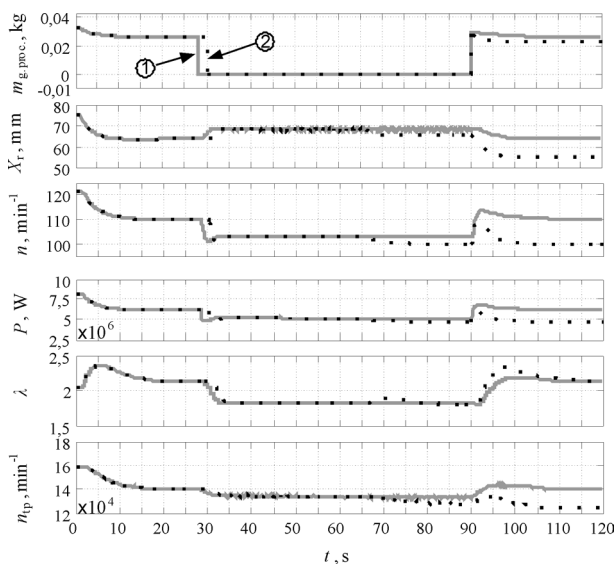
2.1. Impact of interruption of fuel supply to a cylinder on engine performance

The interruption of fuel supply to a cylinder or an uneven fuel supply to cylinders may cause overloading of the remaining cylinders and vibration. The engine performance at various loads has been analysed in order to determine safe operating regimes under the aggravated conditions resulting from uneven fuel supply.

In the event of interrupting the fuel supply to one of the cylinders running at 100 % load, a considerable reduction of engine speed and power occurs. Overloading does not occur due to the fuel rack limit - the engine runs under maximum continued load on the remaining cylinders. The maximum revolution speed could be reached only if the fuel rack limits were increased, but this would make the load on the remaining cylinders unacceptable. The interruption of fuel supply to one of the cylinders should be done without undoing or increasing the fuel rack limit. Also, when re-establishing fuel supply, the overspeed shut-down safety system might be activated due to the

sudden increase of power, i.e. revolution speed. Therefore it is necessary to determine the safe operating limits in the event of uneven fuel supply to the cylinders.

The following example describes the interruption of fuel supply to a cylinder running under 75 % load, i.e. at the revolution speed of 110,4 min^{-1}. To begin with, the engine load is reduced from 100 % to 75 %. According to Figure 1, at the twenty-eighth second, the fuel supply to one of the cylinders is cut off (shown by the line 1), i.e. the fuel mass is reduced to zero by step function. The interruption of fuel supply to a cylinder results in a sudden reduction of engine power. Due to the difference between the propeller load and the engine load, the revolution speed decreases. The control system is activated sending a signal requiring an increased fuel supply. Due to the lower amount of exhaust gases, the scavenging air pressure decreases, and the fuel rack shifts as shown in Figure 2. It stops at the limit, i.e. at the position of 68,5 mm, and the scavenging air pressure drops to 2,68 bar. Figure 3 shows that the rack has moved to the position at the limit of the maximum torque allowed for the achieved revolution speed. At the sixty-fifth second, the given revolution speed drops to 100 min^{-1}. The observed characteristics are presented by intermittent lines 2 in Figure 1.

Figure 1. Change in the engine performance due to the interruption of fuel supply to a cylinder under 75 % engine load

Slika 1. Promjena značajki motora prekidanjem dobave goriva jednom cilindru pri 75 % opterećenja motora

The control system moves the fuel rack towards the lower supply, the new stationary point is at 57 % of the engine load. The fuel rack stops at the position of 65,5 mm. By following the rack's move in Figure 2, it can be

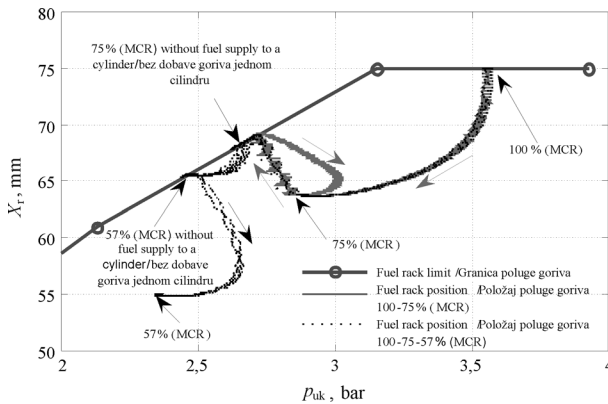noticed that the rack is constrained by the scavenge air pressure.

Figure 2. Movement of the fuel rack against scavenge air pressure

Slika 2. Pomak poluge goriva, prema tlaku zraka za ispiranje cilindara

Figure 3 shows that the engine has not reached the maximum torque limit. At the ninetieth second, in both observed cases, the fuel supply is re-established to all the cylinders, and the revolution speed suddenly increases and stabilises at the set values.

According to the described example, in the event of decrease or interruption of the fuel supply to one of the cylinders, reduction of the engine load is recommended by decreasing the set revolution speed. This should reduce the possibility of difficulties related to the engine operation at boundary torques which are unevenly distributed by the cylinders in case of an uneven fuel supply.
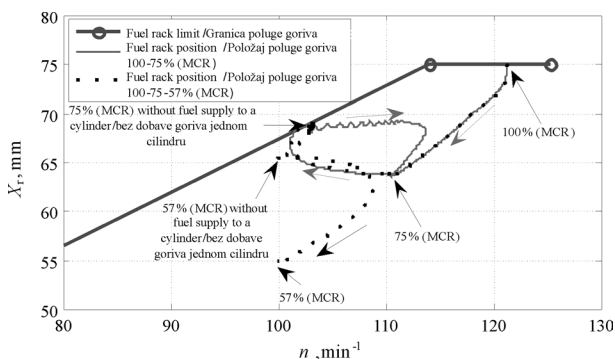

Figure 3. Movement of the fuel rack against engine revolution speed

Slika 3. Pomak poluge goriva, prema brzini vrtnje motora

2.2. Analysis of engine performance under varying loads due to external impacts

Considerable load changes may occur at various sailing regimes. The most demanding sailing regimes with regard to sudden load changes include manoeuvring and sailing in heavy seas. A number of parameters affect the load changes in these conditions. The calculation of the changing loads on the ship's propeller requires modelling of the hull structure, the propeller, and the sailing conditions affecting the entire ship system. Reports to the manufacturer of the engine MAN B&W [5], Day et al. [6], and Nakamura et al. [7], describe the loss of speed of the container ship in various sea conditions. They assert that when the sea state is 5 and the ship is heaving to, her speed is reduced by 8 % to 10 %, and the reduction may be double when the sea state is 7. In even worse sea conditions, the speed is expected to be reduced by 40 % to 50 %. The speed reduction results not only from the increased resistance to the ship's movement, but also from the vertical acceleration of aft part of the ship. Such changes directly lead to sudden load changes where overspeed may occur. In order to be as fast as possible under the conditions, the ship requires a safe engine operation at high loads. Keeping the engine at high loads under such conditions requires fast and accurate control systems. Figure 4 shows the external load based on the results of examining load torque change on a container ship in heavy seas conditions [8].
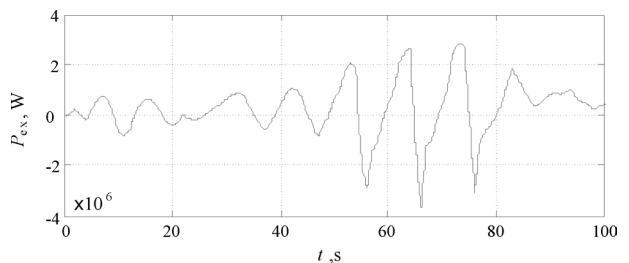

Figure 4. Change in the external load in heavy seas conditions

Slika 4. Funkcija promjene vanjskog opterećenja motora u uvjetima teškog mora

The ultimate engine unloading would mean an almost instantaneous unloading of the engine due to loss of the propeller or the crankshaft fracture. Such changes can be considered as instantaneous and described by step functions. In such cases the overspeed limit is overrun. The limit is set by the manufacturer and usually ranges from 109 % to 110 % of the revolution speed at the maximum continuous load. The insurance societies regulations provide terms for incorporating the modules which will completely cut off the fuel supply in such a case.

Most of today's slow speed medium power marine engines are fitted with electronic revolution speed governors featuring a number of upgraded functions. These are PID regulators which form the control signal, taking into consideration the difference between the set revolution speed and the feedback revolution speed, and acting upon the occurrence of deviation.

### 2.2.1.　Example of an engine load step change

Changes of the propulsion system features during sudden changes of the propeller load depend on the system's inertia and on the rate of torque which is developed from the energy released by the fuel combustion in the engine cylinders. Common revolution speed control systems act only upon the occurrence of deviation between the signals of the actual and given revolution speed. This means that, in case of sudden load changes, oscillations in the revolution speed will inevitably occur. Also, an engine with fewer cylinders and a lower revolution speed will have greater amplitudes, and will require more time to re-establish steady operation.

In the model presented by this paper, the governor has been modelled in line with the basic principles of PID regulation, and fitted with additional functions enhancing the propulsion system response.

The following example, provided through the simulation model, examines the propeller's step unloading and loading by 50 % of the nominal load. The model includes only a basic acting governor, with invariable constants. The overspeed protection has been switched off. Upon the first second of the engine's operation, at the revolution speed of 121,4 min⁻¹ and 100 % load, there is a step withdrawal of the propeller's power by 50 %. Due to unloading, the revolution speed suddenly increases, and the speed activating the overspeed protection of 132,4 min⁻¹ (109 % n_{MCR}) is overrun. As shown in Figure 5, due to the sudden increase of the revolution speed, the engine power rises and so does the engine load. The engine torque instantly increases as well because of the sudden unloading and the power obtained from the cylinders which are still burning the nominal amount of the fuel injected just before the supply reduction. After 15 seconds, the revolution speed is stabilised again at 121,4 min⁻¹, with power reduced by 50 %. In the twentieth second, there is a step increase of the load by 50 %.

Figure 5 shows the instantaneous rise of the propeller power which abruptly decreases due to the reduction of the engine revolution speed. The control system moves the fuel rack towards greater supply, but at the position of 60 mm the fuel rack reaches the scavenging pressure limiter, and it slowly moves along the limit up to the position of 75 mm, which is the position at 100 % of the engine load. During the load test, the engine was controlled by the governor with a switched-off module for programmed operation. Under real conditions of the

ship exploitation, it is desirable that the engine runs close to its maximum continuous power. In order to enable the engine withstand abrupt unloading, without activating the overspeed protection system, various enhancements of the control system have been introduced.
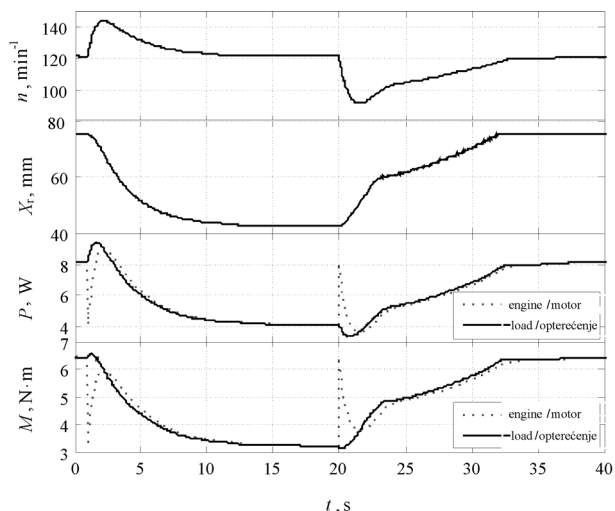
Figure 5. Change in the engine performance during step loading and unloading by 50 %

Slika 5. Promjena značajki motora pri skokovitom opterećenju i rasterećenju od 50 %

Figure 6 shows the results of the simulation for five ways of the response enhancement, by using additional functions in the engine control system. The figure shows the performance characteristics at the response of the engine experiencing the step unloading from 100 % to 50 % and then re-loading to 100 %. The characteristics are recorded in the intervals of 1 second, in order to appear more clearly in the figure. Lines 5 and 1 mark the performance with the activated high speed module. The module's function is to rapidly decrease the fuel rack movement. The high speed module is activated in the event of overrunning the set speed limit, the limit for the observed engine being 126,3 min⁻¹ if the given speed is up to 121,4 min⁻¹; if the given speed is higher, then the high speed module is activated at 129,9 min⁻¹. The high speed module is shut off at 100 % of the revolution speed under maximum continuous load. There are several ways to carry out the supply reduction when overrunning the set triggering limit. The programme defines the scope and speed of the fuel rack movement after activating the module.

In Figure 6, lines 5 and 1 show the engine performance when the high speed module is combined with various programmes. In the first example, line 5 shows the sudden movement of the fuel rack towards reduction of the supply. In comparison with Figure 5, we can notice the decrease of the maximum revolution speed. The programme is adjusted in such a way that,

when the function direction of the revolution speed is changed, the high speed module returns the rack position to the position defined by the basic control system. In the second case (line 1), the programme is set to switch off the high speed module when the speed drops to 100 % of the revolution speed under maximum continuous load. The programme is adjusted to return the position of the fuel rack to the position defined by the basic control system when crossing the value of the activating limit (126,3 min^{-1}).

While observing the response, the oscillation of the fuel rack can be noticed as well as the revolution speed oscillations, i.e. *hearting*. This phenomenon can be avoided with the help of the programme, in a number of ways. When switching off the high speed module, the oscillations could be avoided if the basic control system followed the rack movement as defined by the high speed module.

Line 2 presents the engine performance when the high speed module affects the input signal of the basic regulator's integrating unit. Observing the fuel rack movement, an almost instantaneous movement of the fuel rack towards the start the engine position can be noticed. In this way, at the step unloading by 50 % the engine accelerates, but only up to the safety limit of the overspeed system. After 10 seconds, the governor obtains the given revolution speed of 121,4 min^{-1}. After 29 seconds, there is a step increase of the load by 50 %. Observing the responses in the three cases, the same performance change in all of them can be perceived, i.e. the engine is controlled by a governor, and the high speed module is switched off. After 4 seconds, the acceleration is decreased as it is constrained by the scavenge air pressure. In order to make the period of transient phenomena shorter, amplification of the inlet signal is introduced into the control system, according to the size of the interval between the actual and the given value of the revolution speed. Line 3 presents the engine performance during the response of the control system programmed in such a manner. During loading in the thirty-second second, a faster movement of the fuel rack and a decrease in the revolution speed oscillation amplitudes can be noticed. Line 4 presents the engine performance obtained by combining all the described ways of the response enhancement during step changes of the engine load. The characteristics obtained by the regulator programmed in such a manner imply that the activation of the overspeed safety system would not take place. The protection is set by the manufacturer at 109 % of the revolution speed under maximum continuous power. Step load changes are likely in practice, e.g. when the shaft breaks or the propeller falls out.
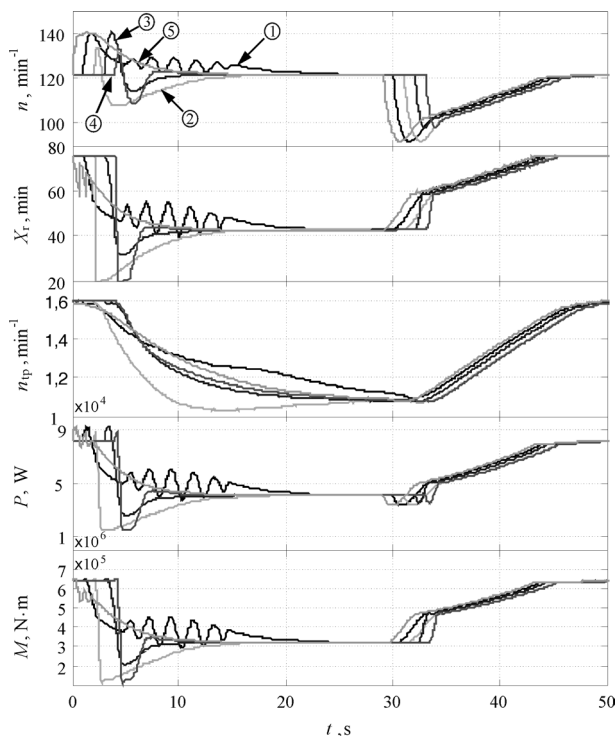
Figure 6. Change in the engine performance during step loading and unloading, with various enhancements of the control system
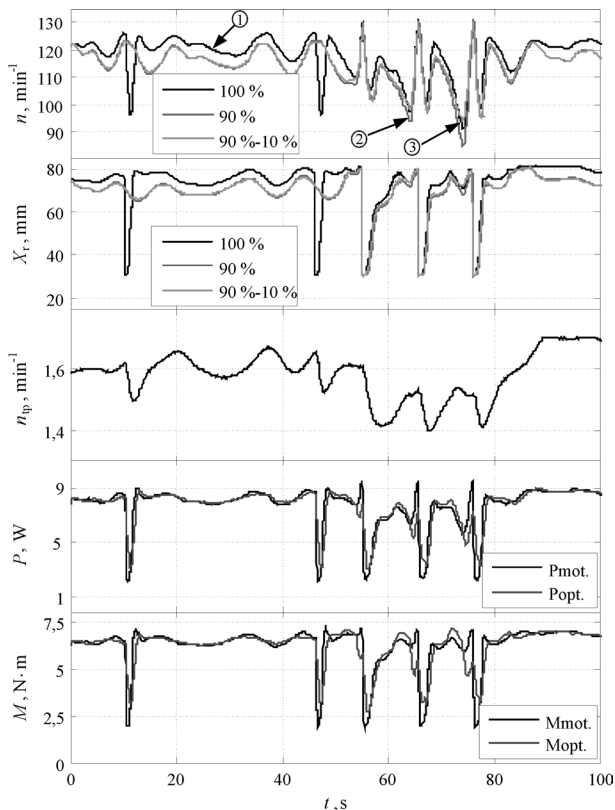
Slika 6. Promjena značajki motora pri skokovitom opterećenju i rasterećenju, pri različitim poboljšanjima regulacijskoga sustava

2.2.2. Example of load change when sailing in heavy seas

When sailing in heavy seas, some of the most common phenomena include rapid alternating loading and unloading of the engine at various intervals of time, which depend on a number of elements. A function is taken as an example of the load change, as shown in Figure 4.

The example presents a field of an extreme load change. The area presenting the time when the interference between movements of the ship and the waves takes place can be perceived. The load amplitudes reach up to 40 % with reference to 100 % of the engine load. Moreover, the changes are consecutive and occurring within short intervals of time.

Figure 7 shows the engine performance obtained through the simulation model. In the first case (line 1), the engine load is 100 %, the revolution speed being 121,4 min^{-1}. An increase of the revolution speed over 126,3 min^{-1} and the activation of the highspeed module is noticeable, even at moderate load changes, e.g. in the tenth and forty-seventh second.
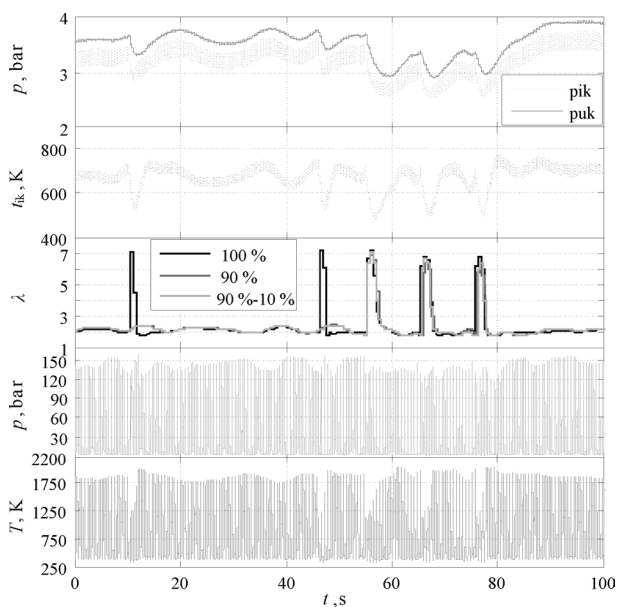
Figure 7. Change in the engine performance under influence of the varying external load

Slika 7. Promjena značajki motora pri djelovanju promjenjivoga vanjskoga opterećenja

Line 2 presents the change in the revolution speed at a load decreased by 10 %. The given revolution speed is reduced to 117,2 min⁻¹, which increases the interval until the activation of the highspeed module. In this case, the revolution speed will not reach the activating limit of the highspeed module in the tenth and forty-seventh second. Obviously, the revolution speed oscillations at the time of the module's activation are greater than in the first example, as the load change function has remained the same, and the power developed by the engine decreases by 10 %. In practice, the engine load decreased by 10 %, under varying loads affected by the external effects, would be followed by a decrease in the external load amplitudes. Line 3 presents the revolution speed response obtained through the simulation model, for an operating point at 90 % of the nominal engine load and the changes in external load reduced by 10 %. This example implies that under heavy seas sailing conditions, it is beneficial to select an engine operating point at a lower load in order to avoid the activation of high-speed modul at smaller oscillations of the external load.

Figure 7 also shows changes related to the engine power, torque, and load. A very good response to the load changes can be noticed, even in the extreme

events when the high speed system is activated. Then the rapid decrease in fuel supply results in the sudden engine power reduction and the revolution speed drop. The period needed for returning to the given operating point amounts to only a few seconds, owing to a special module that amplifies the input signal to the governor, according to the size of the interval between the actual and the given value of the revolution speed. By analysing the turbocharger revolution speed, it can be noticed that the turbocharger revolution speed is retarded in the observed example. Slow speed two-stroke engines use the turbocharging system with constant pressure, where the capacity of the exhaust gas receiver is big enough to fairly compensate for the pressure pulsation affecting the torque of the turbine. The gas temperature in the exhaust gas receiver oscillates very much, due to the combustion of the varying fuel amount as determined by the control system. This combustion of the varying fuel amount in the engine cylinders directly affects temperature changes in the cylinders, whereas the pressure change is considerably affected by the scavenge air pressure, as shown in Figure 8. The figure also shows the change in the air excess ratio in the engine cylinders. When unloading the engine, the fuel supply is reduced and the air excess ratio rapidly increases. The reverse phenomenon occurs during rapid loading: the fuel supply is increased and the air excess is reduced to compensate for the loss of the revolution speed. In the observed example, the lowest point of the air excess ratio is 1,66 in the seventy-ninth second, so that it may be assumed that the fuel rack movement limits, given the scavenge air pressure, are set correctly.

Figure 8. Change in the engine performance under influence of the varying external load

Slika 8. Promjena značajki motora pri djelovanju promjenjivoga vanjskoga opterećenja
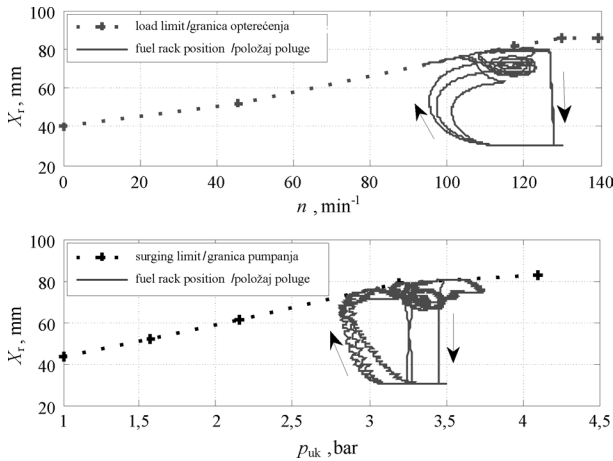
Figure 9. Fuel rack movement against the revolution speed and scavenge air pressure under influence of the varying external engine load

Slika 9. Pomak poluge goriva prema brzini vrtnje i tlaku zraka za ispiranje cilindara pri djelovanju promjenjivoga vanjskoga opterećenja motora

Figure 9 shows the fuel rack movement in relation to the scavenge air pressure and the engine revolution speed. It is estimated that, in this example, the air excess ratio is high enough, so that there will not be any incomplete combustion resulting in contamination of the environment by solid pollutants. Also, in case of incomplete combustion, soot can settle in exhaust gas receiver and increase the risk of the turbine contamination.

The change in the compressor's operating point for the three observed cases is shown in Figure 10. When responding to the load, the fuel supply increases, making the pressure and temperature in the exhaust gas receiver higher. After that, the shaft torque rises and the turbine accelerates the turbocharger rotor, and the compressor supplies a larger amount of air. Accordingly, the compressor's operating point, on its way towards a larger supply of air, passes closer to the surging limit. The movement of the compressor's operating point towards the surging limit is restricted by setting the limits to the fuel rack movement against the scavenge air pressure. Figure 11 shows the change in the turbine's operating point during the simulation of the engine operation under varying external effects. In the event of a sudden unloading of the engine, the engine revolution speed increases, and the governor reduces the fuel supply. There is a fall in pressure in the exhaust gas receiver, decreasing the pressure ratio before and after the turbine, whereas the speed ratio u/c increases.
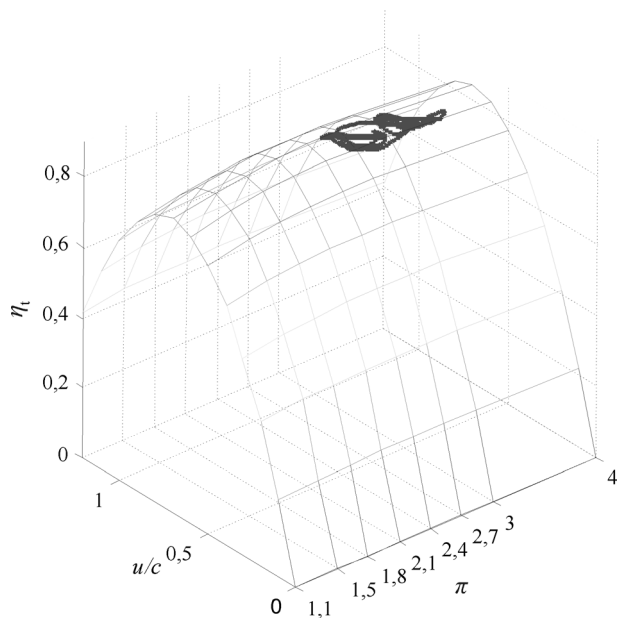
The turbine's efficiency at the reference point, within the turbine performance area, is below the maximum value at the right side. In this example, in the event of a sudden load increase, the efficiency first rises to the maximum, then decreases at the left side. When loading the engine,

the turbine's operating points reach the referential point in reverse order.
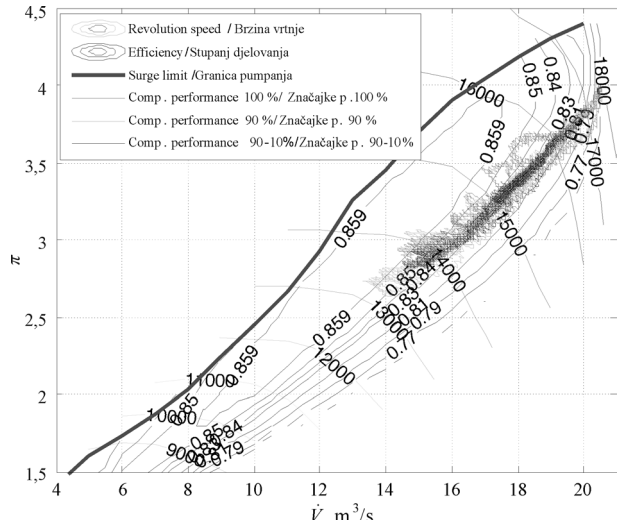
Figure 10. Change in the compressor's operating point during the effects of the varying external engine load

Slika 10. Promjena radne točke puhala za vrijeme djelovanju promjenjivoga vanjskoga opterećenja motora

Figure 11. Change in the turbine operating point during the effects of the varying external engine load

Slika 11. Promjena radne točke turbine za vrijeme djelovanja promjenjivoga vanjskoga opterećenja motora

The analysis of the engine and turbocharger performance during sudden load changes has been carried out assuming the referential environment conditions and immaculate technical condition of all engine systems. The environment condition values change while sailing, and it is common that contamination affects the turbine

nozzle and blades, suction filter before the turbine, air filter, air cooler, exhaust gas steam generator, exhaust duct, etc. It is particularly important to study the operating parameters under that kind of aggravated conditions, and determine the exploitation limits.

2.3. Analysis of engine and turbocharger performance under aggravated conditions

The ventilation system of an engine room featuring slow speed engine needs to ensure enough air for combustion in the main engine, auxiliary engines, and for oil-fired steam generators. The capacity of the ventilation system should be 1,75 to 2 times higher than the amount of air necessary for the operation of all engines under maximum continuous load. The air temperature at the compressor suction highly affects the engine performance. Change in air temperature at the compressor suction is influenced by the environment conditions. According to research of MAN B&W manufacturer [9], it has been found that the maximum air temperature at the inlet to the engine room ventilation system is 1 to 3 ºC higher than the sea surface temperature, whereas the temperature in the engine room is 10 to 12 ºC higher than the inlet temperature. External air is supplied to the engines immediately before the compressor filter, so that the temperature at that point is additionally higher by 1 to 5 ºC than the air temperature at the inlet to the engine room. The minimum air temperature in the engine room is practically maintained above 5 ºC in such a way that, in case of a lower temperature, the air supply is decreased to sufficient for engine operation, and reduced for engine room ventilation.

Standard referential environment conditions have been introduced for determining referential fuel consumption and exhaust gas temperature, in line with ISO 3046-1:2002 (E) and ISO 15550:2002 (E) requirements, whereas maximum environment temperature conditions *tropical conditions* have been defined according to IACS[1] M28 (1978) requirements. In tropical environment conditions, ISO standards define the unlimited availability of the engine under maximum continuous load at the air temperature at the compressor suction amounting to 45 ºC. Under extreme conditions, the sea temperature can be even higher, e.g. in ports or shallow bay with poor water circulation, so that the high scavenge air temperature alarm is set at 55 ºC. In practice, such cases rarely occur because ships sail in those areas under reduced load, e.g. at manoeuvring speed, hence the scavenge air temperature is lower.

In case of a lower temperature at the compressor suction, air flow mass through the compressor is greater, so that the mass in the scavenge air receiver is greater as well. The temperature is increased after the cooler.
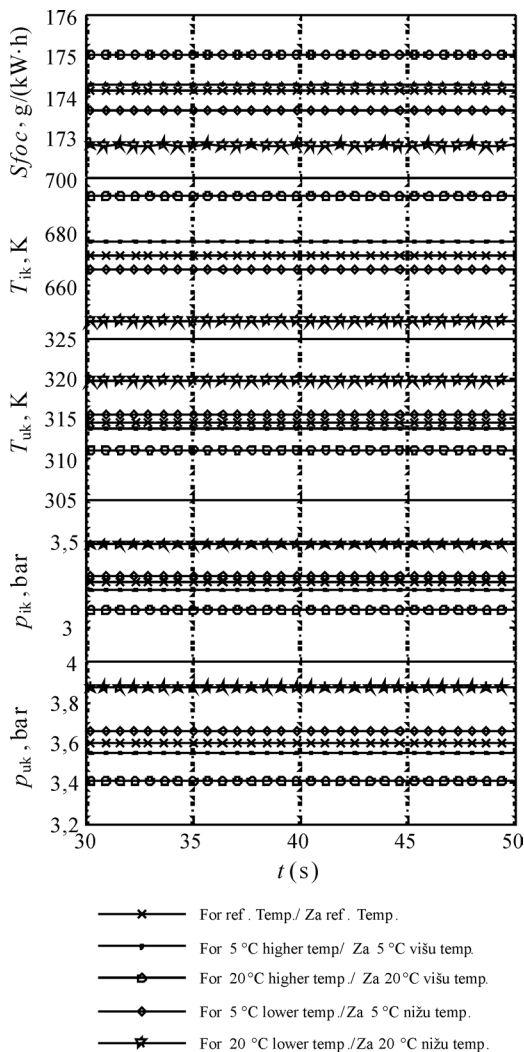
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[1] IACS – International Association of Classification Societies

Figure 12 shows that, if the temperature at the compressor suction is decreased by 20 ºC, the scavenge air pressure is increased by 0,265 bar. Likewise, the temperature is higher, precisely by 4,81 ºC. Due to a higher scavenge air pressure, the compression pressure is higher as well. The increased compression pressure results in higher positive performance per cycle, so that the fuel consumption is lower by 0,79 % under the same engine load. The temperature of cylinder gases is decreased with regard to the referential one, due to combustion of the lower amount of fuel. The lower temperature of the cylinder process results in exhaust gas temperature reduced by 24 ºC. The pressure in the exhaust receiver is higher because of an increased amount of exhaust gases, which is the result of a higher pressure during the process of scavenging the engine cylinders. If the air temperature at the compressor inlet is higher than the referential one, we can notice similar changes as in the described example of the reduced temperature, but the sign is reversed.

Figure 13 shows the compressor operating points when engine load changes from 100 % to 30 %, with reduced revolution speed and at varying air temperature at the compressor suction. Line 1 presents the simulation results for referential conditions. Line 2 presents the change of the compressor operating point, under decreased load, and at the compressor inlet air temperature increased by 20 ºC. The operating points presented with line 3 are obtained by simulation at the compressor inlet air temperature reduced by 20 ºC with regard to the referential temperature. The simulation results indicate the approach of the compressor operating range to the surging limit. This is much more emphasised at higher engine loads, i.e. at higher capacity air flow through the compressor. According to the above example, impacts of the temperature change at the compressor suction do not threaten the safe engine operation. In practice, the engine performance may change owing to a number of other circumstances resulting from the technical and environmental conditions. For instance, periodic contamination of the turbine, filter, etc. is a common phenomenon. These circumstances may have a multiplier effect on the engine performance and may lead to boundary values.

Various solid and liquid dirt in the exhaust gases may cause corrosion, erosion, and deposits on the turbine and compressor elements. This reduces the engine's life and efficiency. Deposits reduce the flow area through which the gases pass and the turbine's efficiency. The turbocharger characteristics affect, to a certain extent, the fuel consumption, exhaust gas temperature, and emission of NO_x gases and solid dirt. Slow speed two-stroke engines tend to have a growing mean effective pressure. Such trends require a higher pressure ratio at the turbocharger flanges. Meanwhile, owing to modern technical innovations, the turbocharger efficiency

grows, so that the sufficient amount of air is obtained from a smaller amount of exhaust gas. When selecting a turbocharger, the operating range is determined with some reserve regarding the surging limit (usually 15 %). During the engine exploitation, the turbocharger operating point may approach the surging limit. This may be caused by the fuel injector wear, surplus of lubricating oil, failure of scavenge air no-return valve, but also a fault related to the turbocharger system. The turbocharger performance is also affected by the changes resulting from contamination at various places. It often occurs that the engine room pressure falls due to insufficient ventilation or contamination of the compressor air filter, air cooler, suction filter before the turbine, turbine nozzle and blades, exhaust gas steam generator, exhaust duct, etc.
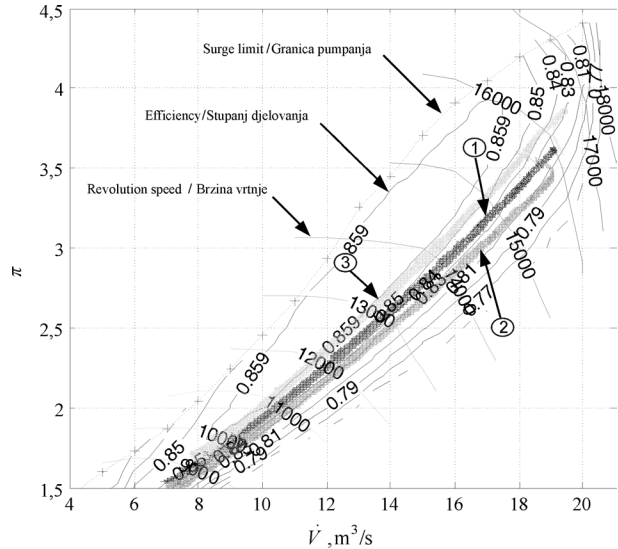
Figure 13. Change in the turbocharger operating point during load reduction, due to the change in the engine revolution speed, at varying compressor inlet air temperature

Slika 13. Promjena radne točke puhala za vrijeme smanjenja opterećenja, mijenjanjem brzine vrtnje motora, u odnosu na različite temperature zraka na usisu u turbopuhalo

The turbine and compressor performance areas have been obtained from the manufacturer and presented in the thesis [4]. Data referring to the reduced performance due to the turbine contamination have been obtained from the turbocharger manufacturer. The data have been obtained by measurement and are in the function of the pressure ratio before and after the turbine.

Figures 14 shows the results of the simulation of the turbocharged engine with the flow area of the nozzle ring reduced by 15 % with regard to the referential one, and with the efficiency additionally reduced by 10 % with regard to the referential one. The pressure increase in the exhaust gas receiver which has resulted from the reduced flow area, is attenuated as it is affected by the reduced efficiency.

The increased losses lead to the decrease of the torque at the turbocharger shaft, which results in the reduced revolution speed, making the air capacity flow lower than the referential one by 11 %. A smaller air flow through the cooler leads to a lower air temperature in the scavenge air receiver. This makes the air mass larger, which results in an almost unchanged pressure in the scavenge air receiver. The increased pressure and temperature in the exhaust gas receiver leads to the increase in temperature during the process of the working fluid exchange *scavenging*, so that the entire process occurs at a somewhat higher temperature. In addition, the specific fuel consumption is increased and a higher amount of heat energy is released in the engine cylinder. Contamination of the turbocharger adversely affects the

Figure 12. Specific fuel consumption, temperature, and pressure in the scavenge air and exhaust gas receivers, at various air temperatures at the compressor suction

Slika 12. Specifična potrošnja goriva, temperatura i tlak u kolektorima ispušnih plinova i ispirnog zraka, u odnosu na različite temperature zraka na usisu turbopuhala

engine's performance. The specific fuel consumption is higher and the engine is more exposed to heat loads. It operates with a lower air excess ratio in the cylinders, which increases the possibility of incomplete combustion and accelerates the process of further contamination of the turbine with soot. The contamination of the turbine significantly affects the compressor's operation.
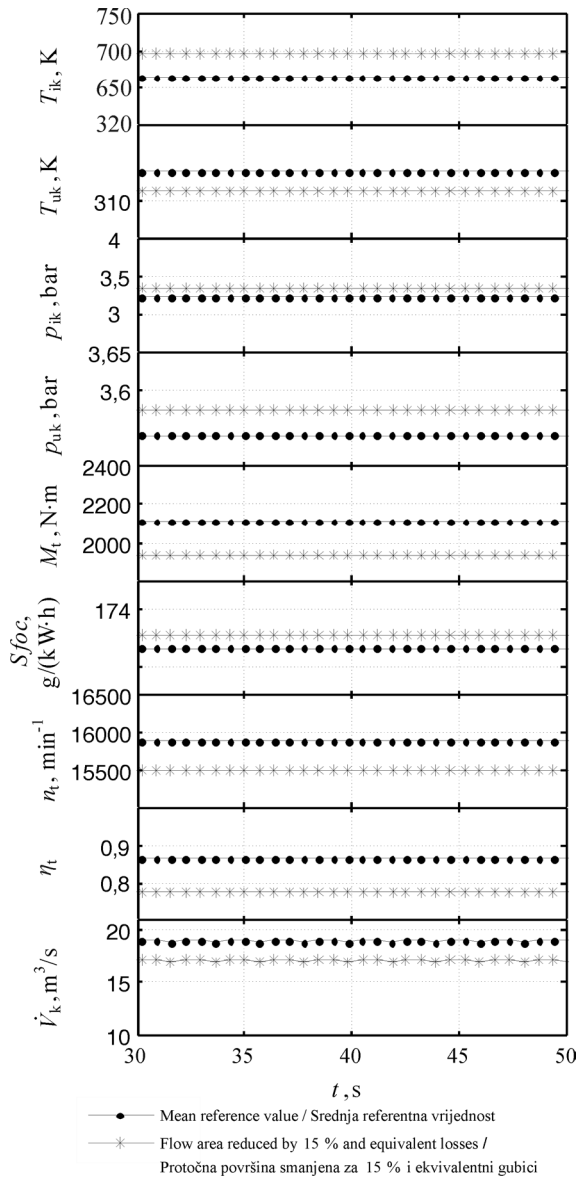

Figure 14. Engine performances at 100 % load, for referential parameters and for the contaminated turbine with the flow area reduced by 15 % and equivalent losses

Slika 14. Značajke motora pri 100 % opterećenju, za referentne parametre i za onečišćenu turbinu sa smanjenom protočnom površinom za 15 % i ekvivalentnim gubitcima

Figure 15 shows the turbocharger operating points during the change in the engine load from 100 % to 30 %, due to the reduced revolution speed and for various

cases of the turbine contamination. Line 1 presents the simulation results for reference conditions (clean turbine). Line 2 presents the change in the compressor operating point for the turbine whose efficiency has been reduced by 10 %. The operating points presented with line 3 have been obtained by simulation, with the flow area of the nozzle ring reduced by 15 %. Line 4 presents the simulation results for the contaminated turbine and the flow area reduced by 15 %, with efficiency reduced by 10 %, owing to the increased losses. This is likely to occur in practice. The occurrence has the most adverse impact on the turbocharger, it moves the operating points towards the surging limit, and reduces the air supply backup by half with regard to the referential conditions (clean turbine).
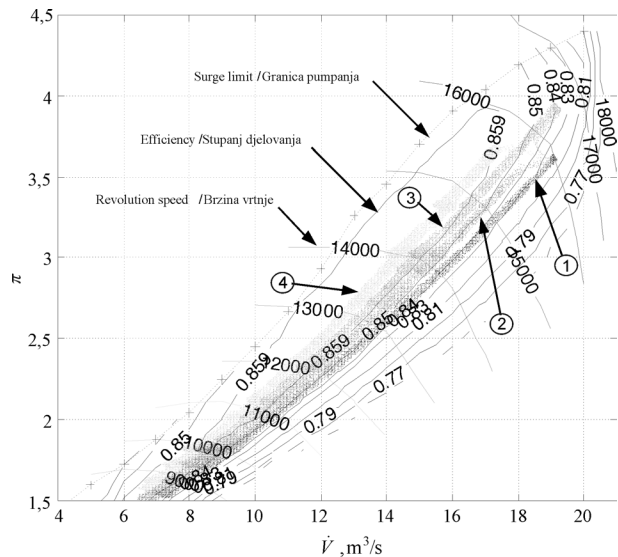
Figure 15. Change in the compressor operating point during load reduction, due to the change in the engine revolution speed, for various flow areas and turbine performance

Slika 15. Promjena radne točke puhala za vrijeme smanjenja opterećenja, mijenjanjem brzine vrtnje motora, u odnosu na različite protočne površine i efikasnosti turbine

3. Conclusion

In this paper, the simulation model is applied to an analysis of the diesel engine propulsion system featuring a two-stroke slow speed turbocharged diesel engine, and a fixed blade propeller.

The analyses of interruption of the fuel supply to a cylinder show that in the event of the nominal limitation of the fuel rack movement, there is a considerable reduction in power and engine revolution speed, while the remaining cylinders run at the maximum continuous load. When rising the fuel rack limit, the engine achieves the given revolution speed, but the load on the remaining

cylinders is higher than the acceptable load. It may be concluded that the reduction or the interruption of the fuel supply to one of the cylinders should be carried out without undoing or raising the limit of the fuel rack movement. It is advisable to decrease the engine load by decreasing the given revolution speed; this in turn reduces the possibility of problems related to engine operation at the boundary torque which is unevenly distributed by cylinders in case of uneven fuel supply.

We have analysed five ways of the response enhancement at a sudden load change, using the additional functions of the engine control system. Through of tests performed on the model, we have come up with the most favourable solution, where the overspeed safety system is not activated at a sudden unloading by 50 %.

The analysis of the engine and turbocharger performance at sudden load changes has been carried out assuming the referential environment conditions and the immaculate technical condition of all engine systems. The results obtained on the simulation model indicate that when sailing in heavy seas, it is beneficial to select an engine operating point at a lower load. This increases the interval before the limit when the overspeed safety module is activated, and the engine runs with less oscillation in the revolution speed. In the observed example, we have determined that the lowest point of the air excess ratio is 1,66, and we may conclude that the fuel rack limits are set correctly, given the scavenge air pressure and that incomplete combustion will not occur.

In addition to the referential environment conditions, the analysis has included the engine and turbocharger performance at various compressor inlet air temperatures under the maximum continuous engine load. If the temperature at the compressor suction is decreased by 20 ˚C, the scavenge air pressure is increased by 0,265 bar, and the temperature is increased by 4,81 ºC. The compression pressure is raised by 11,3 bar, and a higher positive performance per cycle is achieved, whereas the fuel consumption is lower by 0,79 %. During the process, the maximum temperature is lower by 63 ºC with regard to the referential one, so that the exhaust gas temperature is lower as well, precisely by 24 ºC. If the air temperature at the compressor suction is decreased by 20 ºC, the air supply rises in particular, and the compressor runs more efficiently, but the operating point approaches the surging limit. If the air temperature at the compressor inlet is higher than the referential one, we can notice similar changes, but the sign is reversed. The application of the model described in this paper has enabled examination of impacts of the turbine contamination on the engine and turbocharger performance. The analysis of the results indicates that, owing to the turbine contamination, the specific fuel consumption is higher and the engine is more exposed to heat loads, whereas the air excess ratio is

decreased. The contamination of the turbine particularly affects the compressor performance. When the flow area is reduced by 15 %, and efficiency is reduced by 10 %, the turbocharger operating point approaches the surging limit, and the air supply backup is reduced by half with regard to the referential conditions (clean turbine). During analysis of the simulation of the engine running in heavy seas, the contamination of the turbine moves the compressor operating point towards the surging limit.

The performed analyses have proved that the simulation model may be applied for the design purpose. Furthermore, the results obtained through the examination of the performance under aggravated conditions lead to the conclusion that the model is accurate enough to be used for predicting various propulsion conditions. We have particularly emphasised the possibilities for predicting the marine propulsion system performance under aggravated conditions, indentifying performance limits, defining safe propulsion conditions, and performing simple research activities aimed at finding plausible enhancements.

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