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# HEAT RELEASE MODELS - A COMPARATIVE ANALYSIS FOR ENGINE PERFORMANCE PREDICTIONS

## *MODELI OSLOBAĐANJA TOPLINE - KOMPARATIVNA ANALIZA ZA PROGNOZU PERFORMANSA*

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*Abstract*

*This paper deals with a theoretical investigation of the influence of different heat release rates on the performance curves of a diesel engine. The study was further extended to determine the theoretical optimum heat release rate diagram without exceeding the construction constraints of the engine.*

*A marine diesel engine of 16 cylinders, 4 valve per cylinder, direct injection supercharged type was considered. A well proven computer programme for the complete simulation of diesel engines developed by the research team at ITU was employed in the study. The computer code is capable of simulating the unsteady gas flow in the manifolds, in cylinder processes and the turbocharger. The maximum combustion pressure, specific fuel consumption and effective power of the engine cylinders were compared for the different heat release rate cases. The time resolved pressure curves for the intake and exhaust manifolds were also included and the effects of different heat release rates on these diagrams were analysed.*

*Sažetak*

*Ovaj članak se bavi teorijskim ispitivanjem utjecaja različitih stupnjeva ispuštanja topline na krivulje performansi dieselskog motora. Proučavanje je dalje prošireno da bi se odredio diagram optimalnog stupnja oslobađanja topline ne premašujući konstruktivna ograničenja stroja.*

*Razmatra se dieselski motor od 16 cilindara s prednabijanjem sa 4 ventila po cilindru, tip direktnog ubrizgavanja. U studiji je korišten dobro prokušani kompjutorski program za cjelovitu simulaciju dieselskih motora razrađen od istraživačkog tima na ITU. Kompjutorski kod je u stanju simulirati nestabilno dotjecanje plina u kolektor, u proces unutar cilindra i u turbopuhalo. Maksimalni tlak izgaranja i specifična potrošnja goriva i efektivna snaga cilindara motora uspoređena je s različitim stupnjevima oslobađanja topline.Vremenski određene krivulje tlaka za usisni i ispušni kolektor također su uključene i analizirani su utjecaji različitih stupnjeva oslobađanja topline na ovim dijagramima.*

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## INTRODUCTION
### *UVOD*

A steady improvement has been recorded in engine design, operation, and manufacturing over the years directed by market forces of the desire to produce higher performance units.

The classical design approach to marine diesel engines by employing theoretical calculations, prototype manufacturing and testing is expensive in terms of time and money. Thus, the application of the digital computer utilising CAD and manufacturing investigation programmes to the problem has accelerated the engine research, especially the engine performance analysis, and influenced the development of the marine diesel engines.

During the combustion process in a CI engine, liquid fuel is injected towards the end of the compression stroke so that the fuel is distributed in a jet or in several jets throughout all or part of the combustion chamber. Distribution of fuel, mixing with air, evaporation and diffusion to produce a gaseous mixture and chemical reactions to burn the fuel have to be accomplished in an extremely short period of time if the engine is to be efficient.

From the commencement of injection, combustion process may be divided into several stages, which are the ignition delay, rapid combustion (uncontrolled), lower rate of combustion, and a very low rate of combustion.

During the delay period a considerable amount of fuel may have entered to the combustion chamber. Some of this fuel will have become heated and evaporated and mixed with air so that it is ready for rapid combustion that it can take part in chemical reaction. Ignition in one place will be followed by ignition elsewhere, so rapid combustion of prepared fuel follows the first ignition. The rate and quantity of combustion in the second phase is thus dependent on the duration of the delay period and the rate of preparation of fuel during this period, which all these are supported by experimental evidence by Lyn and Valdamis [1]. Once the prepared fuel accumulated during the delay period has been consumed, the rate of consumption falls to a value that can be maintained by preparation of fresh fuel. Finally, after all the fuel has been injected, combustion continues at a diminishing rate as the fuel and oxygen are each consumed. This last stage referred to as the tail of combustion and the previous one are both characterised by diffusion combustion with production and combustion of carbon particles and high rate of heat transfer by radiation.

During all the process in combustion modelling the major factor in making power calculations is the heat-release rate which is usually plotted as net heat-release rate that is the change of sensible internal energy of the cylinder gasses and the work done on the piston. It differs from the rate of fuel energy released by combustion by the heat transferred to the combustion chamber walls. The heat loss to the walls is 10 to 25 percent of the fuel heating value in smaller engines which is less in larger engine sizes.

In modelling the CI combustion calculations, generally single-zone combustion models are used which needs less computing time and they are also reliable. The simplest single-zone model specifies the heat release or combustion pattern in advance so that cycle calculations merely involves adding energy to the cylinder contents in accordance with this pattern at the appropriate points in this calculation. The pattern used is based upon experience with similar engines, having been obtained by analysing experimental cylinder pressure diagrams. Typical heat release diagrams have been published by early workers in this field, notably by Austen and Lyn [2].

The present work gives much more attention to the study and identification of heat-release models utilising computer simulation program that simulates a certain make marine diesel engine [3]. The objectives are not only to provide a clearer understanding of the effects of different heat-release models on the operational values of the engine (power produced, mean effective pressure, fuel consumption) and the pressure variations in the inlet manifold but also to find out the optimum heat release-rate model without exceeding the construction parameters of the engine.

## COMBUSTION MODELLING
## *MODEL IZGARANJA*

For compression ignition combustion calculations, the major limitation is that the combustion process is taken to be a heat addition process and in almost all cases the heat addition is considered to take place unnaturally and at very high rates. The use of modern computers for engine cycle calculations makes more realistic complex cycle calculations possible and these in turn require mathematical modelling of the combustion processes. Thus an early and still major purpose is to estimate the overall engine performance by calculating a cylinder pressure diagram and this has been achieved by a modest improvement on the air standard cycle using small steps, spatially uniform cylinder conditions of composition, pressure and temperature and considering the combustion as a "heat release" process. This may be called a single zone model. In some cases the model has been made more complex by considering the way in which the fuel and air mix inside the cylinder and expressing the result as if the cylinder contents were consisted of a volume of air, fuel and combustion products. Although two-zone models give a zone of comparatively hot gas which might be thought to be more realistic than the mean value obtained from a single zone. The main advantage is found to be the control on heat release due to oxygen availability or unavailability rather than for calculation of chemical species whether by equilibrium or reaction kinetic methods.

During combustion period the pressure rises rapidly for a few crank angle degrees. then more slowly to a peak value about few degrees after TDC. Injection continues after the start of combustion. A rate-of-heat release diagram corresponding to the injection and cylinder pressure data is shown in the Figure 1, [4]

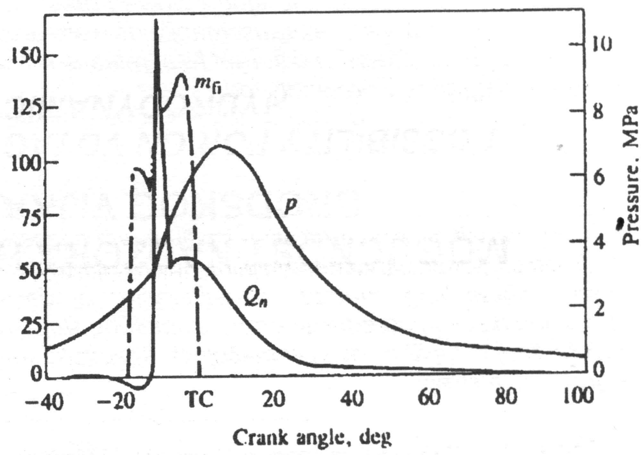


**Figure 1. Rate-of-heat release diagram**
***Slika 1. Dijagram stupnja ispusta topline***

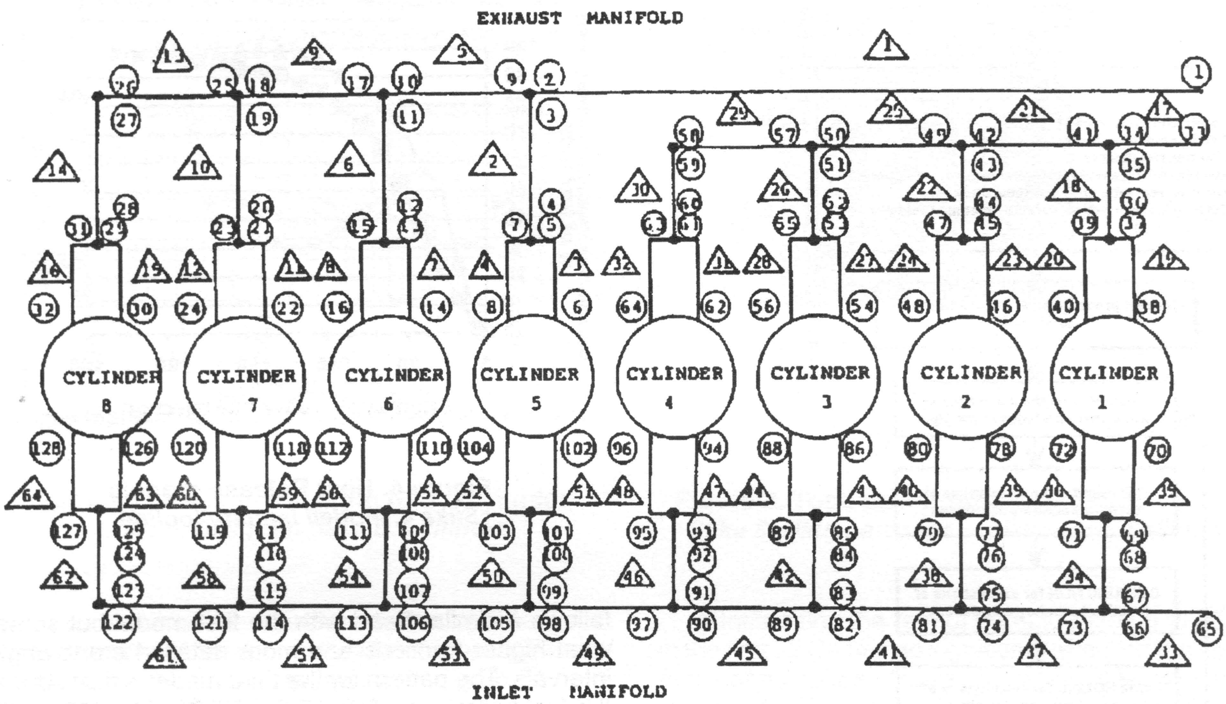


**Figure 2. The engine configuration sketch**
*Slika 2. Shema konfiguracije stroja*

The heat-release-rate diagram shows negligible heat release until towards the end of compression when a slight loss of heat during the delay period (which is due to heat transfer to the walls and to fuel vaporisation and heating) is evident. During the combustion process the burning proceeds in three distinguishable stages. In the fist stage the rate of burning is generally very high and lasts for only few crank angle degrees it corresponds to the period of rapid cylinder pressure rise. The second stage corresponds to a period of gradually decreasing heat-release rate. This is the main heat-release period and lasts about 40 degrees. Normally about 80 percent of fuel energy is released in the first two periods. The third stage is the stage in which a small but distinguishable rate of heat release persists throughout much of the expansion stroke. The heat release during this period usually amounts to about 20 percent of the total fuel energy.

## COMPUTER SIMULATION PROGRAM AND PARAMETRIC STUDY

### *PROGRAM KOMPJUTORSKE SIMULACIJE I STUDIJA PARAMETARA*

To constitute a baseline for the investigation, the studies have been carried out by simulating a marine diesel engine with 16 cylinders, V type, 4 valves (2 inlet and 2 exhaust), direct injection, single piston, supercharged, having inlet and exhaust manifolds. The configuration sketch is shown in Figure 2.

In the engine, each cylinder blocks has two manifolds. The exhaust manifold consists of 32 short pipes, 64 pipe ends, 16 junctions and an inlet nozzle. The compression ratio of the engine is 12.3 and the ratio of stroke to bore is 1.1212. The fuel that is used in the engine has calorific value of 42800 kj/kg and its carbon fraction ratio is 0.8672. During the calculations the engine speed was 1900 rpm.

In the computer simulation program, one dimensional time dependent gas flow throughout the intake and exhaust manifolds of a marine diesel engine is successfully modelled by the non-homentropic gas flow theory and some gas dynamics and thermodynamics characteristics like pressure, temperature, mass flow rate etc. at the end of the time step are determined by means of boundary conditions which are generally classified as active and passive boundary conditions. Active boundary conditions are described as the reason of the time dependent gas flow in control volume like cylinder, turbine compressor etc. Passive ones are the factors absorbing and reflecting the waves inside the pipes with many kinds of waves and they represent junctions, nozzles valves etc. Though the basic equations are not difficult to formulate, the logical organisation of the sequence of calculations of the program involving thermodynamics and the engine kinematics makes the program very complex. The thermodynamic calculations use the first law of thermodynamics as stated above to balance all the energy terms and the state equations of gas and the gas properties in the cylinder are assumed uniform throughout the cylinder at any instant of time. The

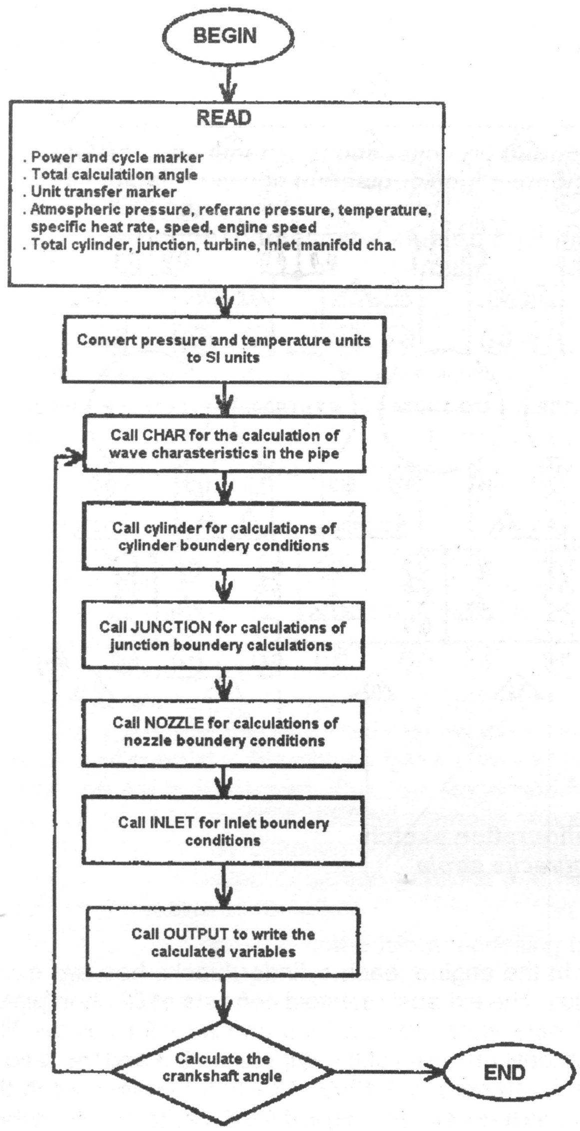


**Figure 3. Flow chart of main programme segment of the simulation programme**
***Slika 3. Dijagram toka glavnih programskih segmenata simulacije***

computer simulation program consists of 1 main program, 34 subroutines, in total 121748 byte without being compiled [3,5]. A flow chart of the computer simulation programme is given in Figure 3.

## DISCUSSION AND CONCLUSIONS
## *DISKUSIJA I ZAKLJUČCI*

For the reported work in this paper, a series of heat release models was tried, but only the most appropriate patterns are discussed here [3].

The first model considered in the base line heat release model and the calculations are based on the original engine data. In the base line model 21% of the fuel is burnt in the first 12 Crank angle degrees (Ca) after the fuel injection followed by the 54% at 22 Ca, 85% at 42 Ca and 95% at 72 Ca. The second model

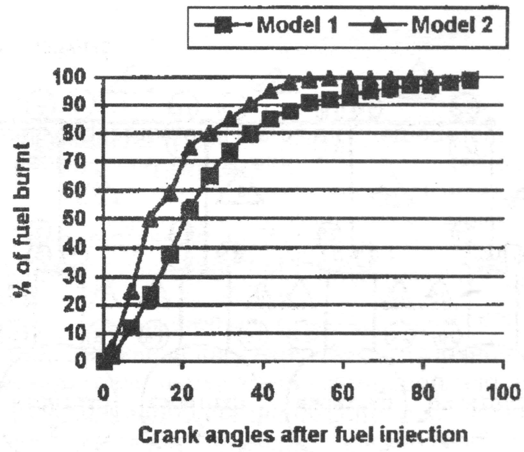


**Figure 4. Heat Release models**
***Slika 4. Modeli ispusta topline***

follows a similar trend with the first model but somewhat higher fractions and more detailed crank angle intervals. The pattern for the third model is that 50% of the fuel burnt in the first 12 Ca, followed by 75% at 22 Ca, 95% at 42 Ca and 99,5% at 72 Ca (Figure 4).

When the maximum cylinder pressures are examined as shown in Figure 5, the peak cylinder pressure o 187.3 bar is observed at 9 Ca after TDC for the third model. There is no significant change for the peak pressured calculated for the base line and the second models.

In some other models considered, although 98% of the fuel is burnt at the first few degrees of crank angle after the fuel injection, the total power produced is less that of the third model which to be optimum model regarding the total power output and specific fuel consumption per cylinder.

The significant point observed from the pressure diagrams is that in the base line model the pressure curve is steeper than that of the third model which produced the maximum power output. The time resolved gas pressures in the manifolds are nominally

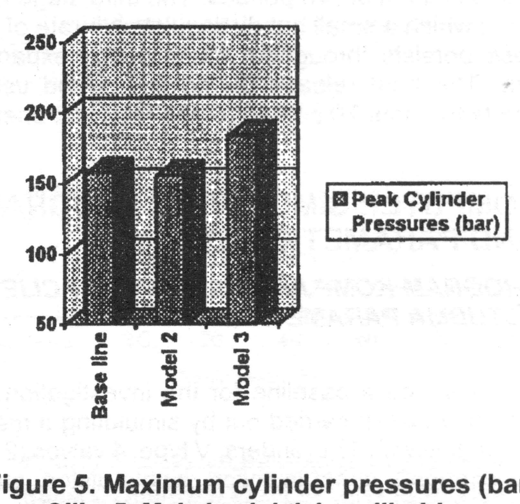


**Figure 5. Maximum cylinder pressures (bar)**
***Slika 5. Maksimalni tlak u cilindrima***

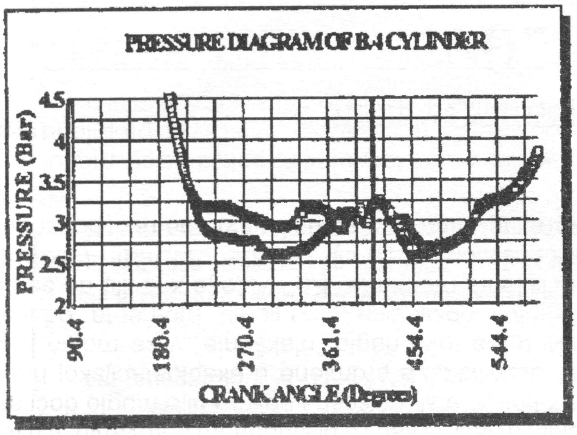


**Figure 6. Pressure diagram of B.4 cylinder**
***Slika 6. Dijagram tlaka B.4 cilindra***

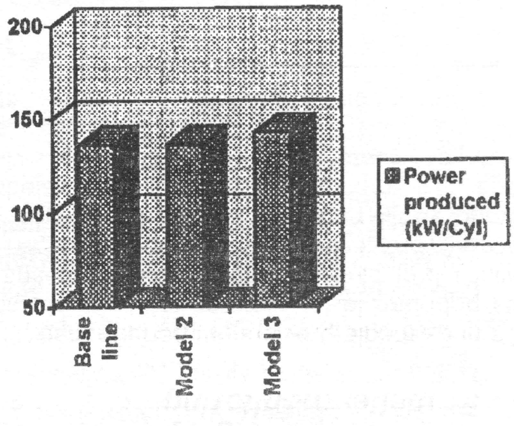


**Figure 8. Total power produced in the cylinders**
***Slika 8. Ukupna snaga proizvedena u cilindrima***

same for the models considered. The cylinder pressure diagrams of the gas period for the base line and the optimum models are compared in Figure 6.

It can be observed in Figure 7 that the specific fuel consumption decreases by 8 g/kWh for the third model with respect to the base line model. The rest of the trials increasing the rate of burning in the first 30 degrees of fuel injection beyond the third model did not result any decrease in the specific fuel consumption.

The corresponding total power outputs per cylinders are illustrate id Figure 8. A 5-6% increase in the total power produces is observed with respect to the base line mode. Beyond the third model, although the rate of burning was increased up to 85-95% in the first stage of combustion, the total power produced per cylinder was found to be decreasing.

The main conclusion for the study may be summarised as follows:

1. In the first stage of combustion up to 50% rate of burning ratio has no effect on the maximum pressures reached in the cylinder. But 80-90% rate of burning ratio in the first 30 Ca of the fuel injection has a definite effect.

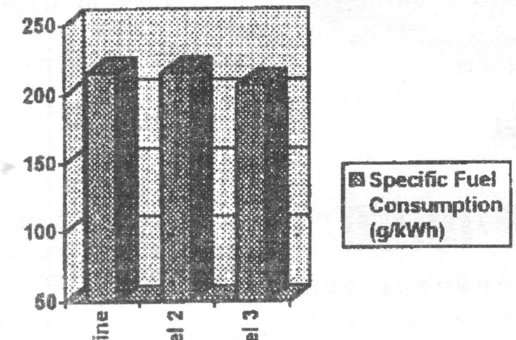


**Figure 7. Specific fuel consumption**
***Slika 7. Specifična potrošnja goriva***

2. Increasing the rate of burning ratio up to 50-55% in the first 22 Ca has no significant effect on the specific fuel consumption. However, when this percentage reaches to 70-85%, the specific fuel consumption decreases. Further increase in the rate of burning ratio has an adverse effect.

3. The optimum result obtained with the rate of burning ratio as follows: 75% of the fuel burnt at the first stage up to 22 Ca, 95% at the second stage up to 42 Ca and the rest at the third stage. Thus, replacing the first stage as the main heat release stage for the second stage will increase the total output per cylinder.

4. The optimum model determined by the results indicated a 4.5-5% economy in the specific fuel consumption as well as a 4.8-5.6% increase in the total power output.

5. With the optimum heat release model, although the total power output per cylinder has been increased and the specific fuel consumption decreased, no significant pressured rise has been observed in the cylinder and at the inlet manifold side of the cylinder.

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