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Thermodynamic Analysis of Marine Two Stroke Diesel Engine Combustion Process

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Preliminary note

This report deals with a method of working fluid availability analysis during combustion process in a cylinder of a marine two-stroke compression the ignition engine. The model, used in this report, is able to detect exact positions of working fluid availability destruction, due new possibilities to prevent such destruction and to increase efficiency of the thermodynamic combustion process.
A unique thermodynamic approach to combustion process in compresion ignition engine is obtained and described by a zero – dimensional two – zone mathematical model.
There is also the possibility, using this method, to calculate all thermodynamic values including entropy, and based on this possibility realistic indicated work values can be obtained. Also availability destruction can be detected, which is caused by irreversibility in the combustion process.
A two-stroke compresion ignition engine for ship propulsion is analyzed by this method, with a 84 % load. The calculated values are shown in following diagrams: *u-s, u-φ, s-φ* and ΔW_{indic},ΔQ-φ and availability destruction positions are located.
A computer simulation program shown in this thesis can be used for two stroke marine compression ignition engine optimization and for further development and thermodynamic analysis of combustion process.

Termodinamička analiza procesa izgaranja u brodskom dvotaktnom dizelskom motoru

Prethodno priopćenje

U ovom radu razmatran je model analize radne sposobnosti radnog medija tijekom procesa izgaranja u cilindru dvotaktnog brodskog dizelskog motora. Cilj modela je otkrivanje mjesta i uzroka destrukcije radne sposobnosti radnog medija da bi se stvorile mogućnosti otklanjanja takvih destrukcija i povećanja efikasnosti termodinamičkog procesa izgaranja.
Također u radu je formuliran jedinstveni termodinamički pristup procesu izgaranja u dizelskom motoru, te izrađen nultodimenzijski dvozonalni matematički model za navedenu metodu analize.
Metoda koja je razvijena daje i proračun svih termodinamičkih veličina stanja uključujući i entropiju, te na osnovu ovih podataka dobivaju se vrijednosti realnog indiciranog rada i gubitaka koje uzrokuje nepovrativost procesa izgaranja.
Metodom je obrađen dvotaktni dizelski motor veće snage za brodski pogon, pri približnom opterećenu od 84 %. Dobiveni rezultati su prikazani u nizu dijagrama i to: u-s, u-φ, s-φ i ΔWindic,ΔQ-φ, te su locirana mjesta gdje se zbivaju najveći gubici radne sposobnosti, odnosno gdje je najveći gubitak indiciranog rada.
Razvijeni simulacijski program koji je predstavljen u ovome radu može se primjeniti za optimizaciju brodskih dizelskih motora a i za daljnju nadogradnju i daljnju termodinamičku analizu procesa izgaranja u svrhu povećanja efikasnosti i smanjenja potroška pogonskog goriva.

Keywords
Availability
Entropy
Thermodynamic analysis
Two-stroke diesel engine

Ključne riječi
Dvotaktni dizelski motor
Entropija
Radna sposobnost
Termodinamička analiza

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1. Introduction

Compression ignition engines are today the most commonly used marine propulsion engines. These engines achieve thermodynamic efficiency of 50% and due to such high efficiency, specific oil consumption and emissions are relatively low. But due to the high oil prices and ecology regulations, higher efficiency and lower emissions are important.

Today´s marine propulsion engines are very sophisticated and technically developed heat engines; thus any significant increase in thermal efficiency or emissions decrease cannot be expected. Already the high oil prices mentioned will shift the engine combustion

Symbols/Oznake

c_V - specific heat capacity at constant volume, V = const., J/kgK
- specifična toplina pri konstantnom volumenu

h - specific enthalpy, J/kg
- specifična entalpija

h_i - specific enthalpy of inlet air, J/kg
- specifična entalpija ulaznog zraka

h_{pr} - enthalpy of working fluid loss due to leaking process, J/kg
- entalpija radnog fluida izgubljenog kroz procjepe

H - enthalpy, J
- entalpija

H_d - lower heating value, J/kg
- donja ogrijevna vrijednost

m - mass, kg
- masa

m_g - mass of injected fuel, kg
- masa ubrizganog goriva

m_{gi} - mass fuel burned in *i*-th zone, kg
- masa goriva izgorjelog u *i*-toj zoni

m_i - mass of outlet combustion products, kg
- masa produkata izgaranja

$m_{IP\ MIX}$ - mass of combustion products mixing with fresh air, kg
- masa produkata izgaranja pomješanih s svježim zrakom

m_{MIX} - mass of fresh air mixing with combustion products, kg
- masa svježeg zraka pomješanog s produktima izgaranja

m_{pr} - mass of working fluid loss due to the leaking process, kg
- masa radnog fluida izgubljenog kroz procjepe

m_a - mass of inlet air, kg
- masa ulaznog zraka

m_{zi} - mass of air in *i*-th zone, kg
- masa zraka u *i*-toj zoni

m_{zsi} - mass of flow air for stoichometric combustion, kg
- masa tijeka zraka za stehiometrijsko izgaranje

$m^0{}_1$ - mass of fresh air in zone 1, kg
- masa svježeg zraka u zoni 1

$m^0{}_2$ - mass of combustion products in zone 1, kg
- masa produkata izgaranja u zoni 1

$m^1{}_1$ - mass of fresh air in zone 2, kg
- masa svježeg zraka u zoni 2

$m^1{}_2$ - mass of combustion products in zone 2, kg
- masa produkata izgaranja u zoni 2

Nu - Nusselts number
- Nusseltov broj

p - pressure, Pa
- tlak

q - specific heat, J/kg
- specifična toplina

Q - heat, J
- toplina

Q_{gor} - fuel chemical energy release, J
- toplina oslobođena izgaranjem goriva

Q_{gubit} - heat loss, J
- gubitak topline

Q_{pozit} - heat release, J
- oslobođena toplina

Q_{st} - heat loss through cylinder wall, J
- gubitak topline kroz stijenke cilindra

R - gas constant, J/kgK
- plinska konstanta

Re - Reynolds number
- Reynoldsov broj

s - specific entropy, J/kgK
- specifična entropija

S - entropy, J/K
- entropija

S_{gor} - entropy due to the fuel burn, J/K
- entropija uslijed izgaranja goriva

S_{st} - entropy due to the heat exchange process through to the cylinder wall, J/K
- entropija uslijed procesa izmjene topline kroz stijenke cilindra

t - time, s
- vrijeme

T - temperature, K
- temperatura

T_{gor} - fuel temperature, K
- temperature goriva

T_{st} - cylinder wall temperature, K
- temperature stijenke cilindra

u - specific internal energy, J/kg
- specifična unutarnja energija

U - internal energy, J
- unutarnje energija

V	- volume, m^3 - volumen	
η_{izg}	- combustion efficiency - efikasnost izgaranja	
φ	- crank angle - kut zakreta koljenastog vratila	
λ	- equivalence ratio - pretičak zraka	
π	- Ludolfs number - pretičak zraka	

Indices/ indeksi

cyl.	- cylinder - valjak
ind	- indicated - indicirano
mix	- mixture - mješavina
ST	- cylinder wall - stijenka cilindra

process research toward increasing thermal efficiency with already achieved emission results.

A standard approach to compression ignition cylinder process analysis is based on the following assumptions:

- Process in a cylinder is not assumed to be closed-cycle: a gas mixture flows through the engine with variable composition.
- It does not suppose «heat addition» during combustion at p = const or at v = const, but mass fraction burned rate at certain real conditions.
- Processes of compression and expansion are not adiabatic, but with the heat exchange to surrounding Q_L plus the heat added by combustion Q_R.
- It is accepted that lower fuel heating value is independent of combustion temperature and type of cmbustion process. These simplifications are based on the relatively small change in the number of moles during combustion, and the relatively small difference in specific heat capacities between the fuel-air mixture and combustion products.
- The equilibrium composition of combustion products during the process at given conditions of pressure, temperature and equivalence ratio Φ is supposed.

Therefore, for process calculation, taking into account the above assumptions it is necessary to know distribution of ΔQ_R during process, i.e. heat release rate data as a function of piston displacement, or crankshaft angle position. To get the curve of heat released during the process it is necessary to analyze the indicated pressure diagram, and additional knowledge of heat transfer to the cylinder walls and coolant – in a standard process marked as $\Delta Q_{L,}$ and where:

$$\Delta Q = \Delta Q_L + \Delta Q_R \ . \tag{1}$$

For the calculation of heat loss ΔQ_L it the relation which connects Nusselt and Reynolds numbers can be used:

$$Nu = \mathrm{A} \cdot \mathrm{Re}^{0.7} \ . \tag{2}$$

The described procedure to obtain the curve of mass fraction burned rate from indicated pressure diagram allows its use in the opposite direction for design improving optimization (pressure change through influence on combustion process in design procedure).

The physical model which has been used for this standard method is not accurate enough mostly because of the simplifying assumption of fuel chemical energy release ΔQ_R as external heat addition. This eliminates the possibility to calculate gas mixture entropy change in a cylinder. Such standard calculation method, even though it gives the approximate indicated work, eliminates the possibility of study of availability losses during the cylinder process. Essentially a less important inaccuracy is neglecting the influence of the process type and temperature on the fuel heating value.

A more accurate modeling of heat release rate, used in this paper, is the preassumption that the thermodynamic combustion process starts when the first part of reactants are burned, and not due to fuel injection starting.

The complete cylinder combustion process model begins with fuel injection to cylinder and such presentation shows four periods of fuel burning: ignition delay, rapid burning period, controlled burning period and afterburning.

Applying the second law of thermodynamics, all thermodynamic values can be calculated including entropy and availability loss during, the combustion process.

In this paper, two – zone, zerodimensional combustion process model is applied on the compression ignition engine, and the described model is implemented in computer simulation program developed by Medica [1].

Important simplifications and assumptions are used due to combustion process modeling:

- combustion is described as incomplete combustion process with thermodynamic equilibrium,
- dissociation process, which occurs on higher temperatures, is neglected,
- a real heavy fuel diesel oil is burning in the cylinder,
- turbulent heat transfer to cylinder wall is implemented in model.

2. Mathematical model of compression ignition engine process

A marine compression ignition engine model is shown in Fig. 1. The model also contains all thermodynamic values which will be observed during the combustion process, and directions of heat and mass transfer.

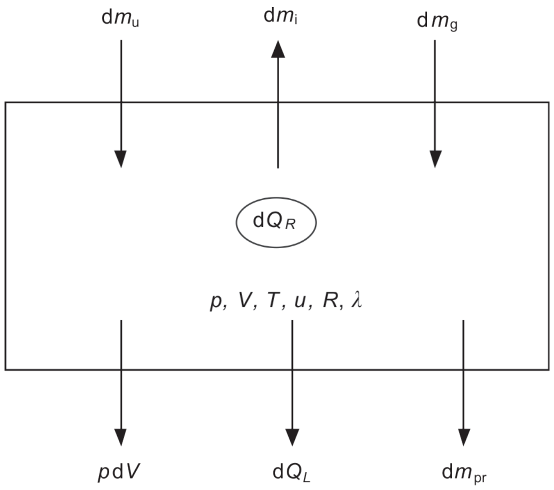

Figure 1. Compression ignition engine cylinder model

Slika 1. Model cilindra dizelskog motora

For proposed model, control volume is defined with boundaries: cylinder head, cylinder wall and piston head.

2.1. Fuel chemical energy release

The first law of thermodynamics gives the main equation for internal energy change:

$$\Delta Q_L = \Delta U + p\Delta V \,. \tag{3}$$

Assuming the cylinder fuel burning process as a process with heat added by combustion ΔQ_R, the first law equation becomes:

$$\Delta Q_L + \Delta Q_R = \Delta U^* + p\Delta V \,, \tag{4}$$

where: ΔU^* - internal energy change of working fluid model.

For better understanding of value ΔU^*, diagram u – s is shown in Figure 2. The diagram contains one realistic process 1-2, which is replaced with two equivalent processes 1-1* and 1*-2, according to Ninić [2].

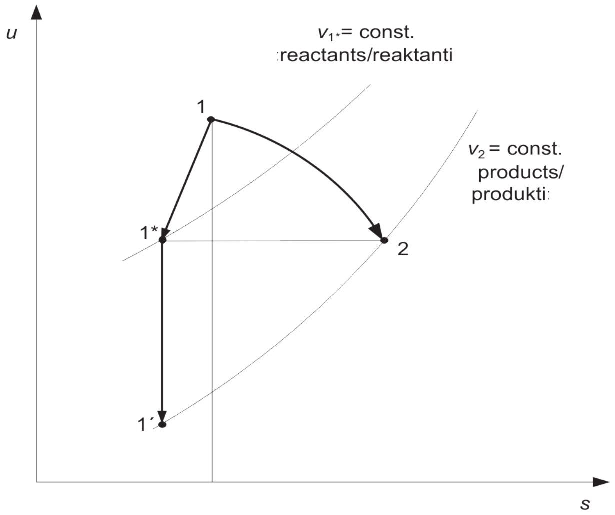

Figure 2. Alternative description of real process 1-2

Slika 2. Alternativni prikaz relanog procesa 1-2

First process 1-1* is cylinder volume change with equilibrium reactant atate (no combustion), with real heat loss through cylinder wall and real work done:

$$\Delta W = p(V_2 - V_1) = p(V_{1*} - V_1) \,. \tag{5}$$

Process 1*-2 is non-equilibrium isochoric combustion process. This second non-equilibrium process can also be replaced by two equilibrium processes which give the same change as process 1*-2.

Those two equilibrium processes are 1*-1´ (work done) and 1´-2 (heat added, identical by the value to the work done).

Due to this model, for the process 1-1*-1´-2 the First law of thermodynamic can be written:

$$\Delta q_L + \Delta q_R = u_{1*} - u_1 + p_1(V_2 - V_1) + u_{1*} - u_{1'} \,, \tag{6}$$

where: $u_{1*} - u_{1'} = u_2 - u_1$, and $u_{1*} - u_{1'}$ is additional work by $p_1(V_2 - V_1)$.

The following equation is:

$$\Delta q_L + \Delta q_R = u_{1*} - u_1 + u_2 - u_{1'} + p_1(V_2 - V_1) \,, \tag{7}$$

where:

$$\Delta q_L + \Delta q_R = c_{V,reakt}(T_{1*} - T_1) + \\ + c_{V,produk}(T_2 - T_{1'}) + p_1(V_2 - V_1) \,. \tag{8}$$

The following equation can be written:

$$\Delta u^* = \left[c_{V,reakt} \frac{T_{1^*} - T_1}{T_2 - T_1} + c_{V,produk} \frac{T_2 - T_{1'}}{T_2 - T_1} \right] \cdot (T_2 - T_1), \tag{9}$$

where:

$$\Delta u^* = \overline{c_V} (T_2 - T_1). \tag{10}$$

Combining the first and the second law of thermodynamics for equilibrium process in open thermodynamic system shown in Fig. 1, the following form is:

$$\Delta U = T\Delta S - p\Delta V + \sum \mu_i \Delta m_i, \tag{11}$$

where: μ_i - chemical potentials of mass flow

For a real process equation is:

$$\Delta U = T\Delta S - \Delta Q_R - p\Delta V + \mu_g \cdot \Delta m_g + \\ + \mu_u \cdot \Delta m_u - \mu_i \cdot \Delta m_i - \mu_{pr} \cdot \Delta m_{pr}. \tag{12}$$

The first two members on the right side in equation (12), together with entropy members in chemical potentials, are representing real value of heat loss $\Delta Q_{L.}$

The value ΔU in equation (11) is, in fact, a value ΔU^* of ideal gaseous model in cylinder and the value of mass flow entropy rise should be removed from calculation in $T\Delta S$, to get the complete value of heat exchanged ΔQ_R and $\Delta Q_{L:}$

$$\Delta U^* = T\Delta S - p\Delta V + \mu_g \cdot \Delta m_g + \\ + \mu_u \cdot \Delta m_u - \mu_i \cdot \Delta m_i - \mu_{pr} \cdot \Delta m_{pr}. \tag{13}$$

Expanding equation (13) with (10), the following form is:

$$m \cdot \overline{c_V} \cdot \Delta T = T\Delta S - p\Delta V + \mu_g \cdot \Delta m_g + \\ + \mu_u \cdot \Delta m_u - \mu_i \cdot \Delta m_i - \mu_{pr} \cdot \Delta m_{pr}, \tag{14}$$

where:

$$\frac{\Delta T}{\Delta \phi} = \frac{T \begin{bmatrix} \frac{\Delta S}{\Delta \phi} - p \cdot \frac{\Delta V}{\Delta \phi} + \mu_g \cdot \frac{\Delta m_g}{\Delta \phi} + \mu_u \cdot \frac{\Delta m_u}{\Delta \phi} - \\ -\mu_i \cdot \frac{\Delta m_i}{\Delta \phi} - \mu_{pr} \cdot \frac{\Delta m_{pr}}{\Delta \phi} \end{bmatrix}}{m \cdot \overline{c_V}}. \tag{15}$$

Value $T\Delta S$ includes entropy change in cylinder due to: heat loss through cylinder wall (ΔQ_L), heat added by combustion (ΔQ_R), fuel and air injection (ΔS_g, ΔS_u), combustion products outlet (ΔS_i) and fluid loss due to leaking process (ΔS_{pr}).

The complete heat value in combustion process is represented by sum of $T\Delta S$ and all other members which contains entropy value.

Equation members with chemical potentials are left only with enthalpies and mass change values.

Equation (15) is changed and can be written:

$$\frac{\Delta T}{\Delta \phi} = \frac{\frac{T(\Delta S_L + \Delta S_R)}{\Delta \phi} - p \cdot \frac{\Delta V}{\Delta \phi} + h_g \cdot \frac{\Delta m_g}{\Delta \phi} + \\ + h_u \cdot \frac{\Delta m_u}{\Delta \phi} - h_i \cdot \frac{\Delta m_i}{\Delta \phi} - h_{pr} \cdot \frac{\Delta m_{pr}}{\Delta \phi}}{m \cdot c_V}, \tag{16}$$

where:

$$T\Delta S_R = \Delta Q_R, \quad T\Delta S_L = \Delta Q_L. \tag{17}$$

According to Škifić [4], equation (16) can be modified as follows:

$$\frac{\Delta T}{\Delta \phi} = \frac{T \begin{bmatrix} \frac{\Delta S}{\Delta \phi} - \frac{u}{T} \cdot \frac{\Delta m}{\Delta \phi} - \frac{p}{T} \cdot \frac{\Delta V}{\Delta \phi} - \\ -\frac{m}{T} \cdot \frac{\partial u}{\partial \lambda} \cdot \frac{\Delta \lambda}{\Delta \phi} \end{bmatrix}}{m \cdot c_V}, \tag{18}$$

where ΔS is represented only with ΔQ_R and $\Delta Q_{L.}$

For zero – dimensional combustion process model, entropy change due to combustion process is described as follows:

$$dS_{gor} = f(\phi) = \left(x_{gor} m_{gor} H_d \eta_{izgar} \right) \cdot \frac{1}{T_{gor}}. \tag{19}$$

Derivative of equation (19) as function of crank angle gives:

$$\frac{dS_{gor}}{d\phi} = \left(\frac{dx_{gor}}{d\phi} \cdot m_{gor} H_d \eta_{izgar} \right) \cdot \frac{1}{T_{gor}}, \tag{20}$$

where: $dS_{gor} = \dfrac{dQ_{gor}}{T_{gor}}$.

2.2. Two-zone combustion model

This paper shows a simplified thermodynamic model of combustion volume, divided in to two zones. The process in the cylinder is divided into 1800 steps, and lasts from 90 ° before TDC to 90 °after TDC. The mixture is determined due to use of real heavy diesel fuel oil as burning fuel.

The first part of analyzed process is compression and combustion process starts a few degrees before TDC. In that moment, the first zone 2 (with combustion products) is formed, and during the combustion process combustion volume contains two zones, zone 1 with reactants and zone 2 with combustion products.

At combustion ends the cylinder volume contains only zone 2.

Each zone is described as zero – dimensional homogeneous space due to temperature and fluid mixture, and can be shown as open thermodynamic system with mass (fluid flow, fuel injection) and heat (heat exchange to cylinder wall, mechanical work) transfer. Also entropy change is involved in each step.

The following assumptions and approximations are used during, the combustion process:

- combustion volume is divided into two zones: zone 1 with reactants, zone 2 with combustion products,
- physical shape of each zone is not considered, but only volume,
- the thermodynamic properties (pressure and temperature) vary only with time (crank angle) and are spatially uniform in each zone,
- before fuel injection, the cylinder contains only reactants which are spatially homogeneous and occupy one zone (1),
- after fuel injection and burning, the second zone was started (2).

In Figure 3 the thermodynamic model of combustion volume as a function of crank angle φ and $\varphi + \Delta\varphi$ is shown.

At the beginning of the combustion process, combustion volume is divided into two zones. Zone 1 contains reactants and zone 2 contains combustion products. As the combustion process varies with time (crank angle), as function of φ, the combustion volume is changed. Zone 2 spreads and flame front changes its position. Also during combustion, there is a mixing process of fresh air which is partially transferred from zone 1 to zone 2. Such process is continous during the combustion process from step 1 to step 1800.

The pressure is equal in both zones:

$$p_1 = p_2 = p\,. \tag{21}$$

The combustion volume is equal to sum of both zones volume:

$$V = V_1 + V_2\,. \tag{22}$$

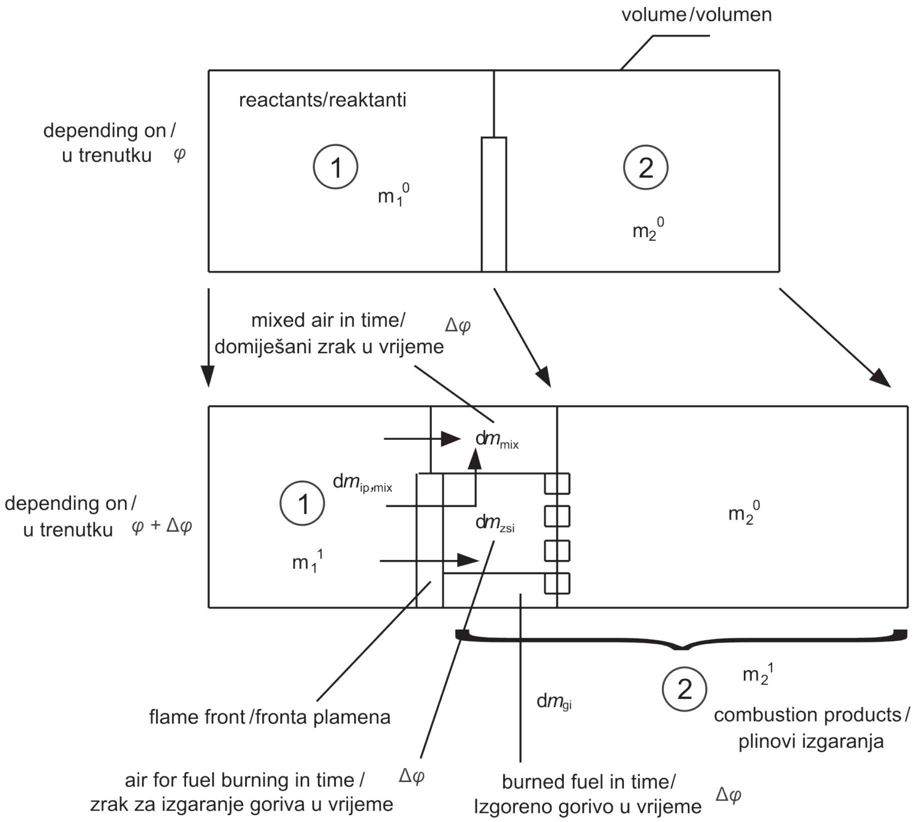

Figure 3. Thermodynamic model of combustion volume as function of crank angle φ and $\varphi + \Delta\varphi$.

Slika 3. Termodinamički model prostora izgaranja u trenutku φ i $\varphi + \Delta\varphi$

Each zone contains pure stochiometric air which is chemically connected to burned fuel, and the rest is free air. Also each zone contains mass fuel burned, which is chemically connected to combustion products.

Complete mass for each zone is:

$$m_i = m_{z,i} + m_{g,i} \,, \tag{23}$$

where: m_i - complete mass of each zone,
$m_{z,i}$ - mass flow air for i - zone,
$m_{g,i}$ - mass fuel burned for i – zone.

Ideal gaseous equations for both zones are:

$$pV_1 = m_1 R_1 T_1 \,, \tag{24}$$

$$pV_2 = m_2 R_2 T_2 \,. \tag{25}$$

Sum of equations (24) and (25) and introducing equation (22) gives:

$$\begin{aligned} pV_1 + pV_2 &= p\left(V_1 + V_2\right) = pV = \\ &= m_1 R_1 T_1 + m_2 R_2 T_2 \,. \end{aligned} \tag{26}$$

The crank angle change is a function of time change using equation:

$$d\phi = \phi \cdot dt \,. \tag{27}$$

Derivative of equations (24) and (25) as function of time change is:

$$\begin{aligned} & p\frac{dV_1}{d\phi} + V_1 \frac{dp}{d\phi} = \\ & = m_1 R_1 \frac{dT_1}{d\phi} + m_1 T_1 \frac{dR_1}{d\phi} + R_1 T_1 \frac{dm_1}{d\phi} \,, \end{aligned} \tag{28}$$

$$\begin{aligned} & p\frac{dV_2}{d\phi} + V_2 \frac{dp}{d\phi} = \\ & = m_2 R_2 \frac{dT_2}{d\phi} + m_2 T_2 \frac{dR_2}{d\phi} + R_2 T_2 \frac{dm_2}{d\phi} \,, \end{aligned} \tag{29}$$

$$\begin{aligned} & p\frac{dV}{d\phi} + V\frac{dp}{d\phi} = m_1 R_1 \frac{dT_1}{d\phi} + m_1 T_1 \frac{dR_1}{d\phi} + \\ & + R_1 T_1 \frac{dm_1}{d\phi} + m_2 R_2 \frac{dT_2}{d\phi} + m_2 T_2 \frac{dR_2}{d\phi} + R_2 T_2 \frac{dm_2}{d\phi} \,, \end{aligned} \tag{30}$$

The mass conservation equations and energy conservation equations are calculated for each zone separately with assumptions of equations (21) and (22).

Equation for specific heat is:

$$c_V = \frac{\partial u}{\partial T} \,. \tag{31}$$

Energy conservation equations for both zones, expanded with ideal gaseous equation for realistic process, and including equation (31) are:

$$\frac{dT_1}{d\phi} = \frac{T_1}{m_1 c_{V,1}} \begin{bmatrix} \frac{dS_1}{d\phi} - \frac{u_1}{T_1}\cdot\frac{dm_1}{d\phi} - \frac{p}{T_1}\cdot\frac{dV_1}{d\phi} - \\ -\frac{m_1}{T_1}\cdot\frac{\partial u_1}{\partial \lambda_1}\cdot\frac{d\lambda_1}{d\phi} \end{bmatrix} , \tag{32}$$

$$\frac{dT_2}{d\phi} = \frac{T_2}{m_2 c_{V,2}} \begin{bmatrix} \frac{dS_2}{d\phi} - \frac{u_2}{T_2}\cdot\frac{dm_2}{d\phi} - \frac{p}{T_2}\cdot\frac{dV_2}{d\phi} - \\ -\frac{m_2}{T_2}\cdot\frac{\partial u_2}{\partial \lambda_2}\cdot\frac{d\lambda_2}{d\phi} \end{bmatrix} . \tag{33}$$

Mass conservation equations for both zones are:

$$\frac{dm_1}{d\phi} = \frac{dm_{zsi,1}}{d\phi} + \frac{dm_{mix,1}}{d\phi} + \frac{dm_{ipm,1}}{d\phi} \,, \tag{34}$$

...

$$\frac{dm_2}{d\phi} = \frac{dm_{g,i,2}}{d\phi} + \frac{dm_{zsi,2}}{d\phi} + \frac{dm_{mix,2}}{d\phi} + \frac{dm_{ipm,2}}{d\phi} \,, \tag{35}$$

...

where:

m_{zsi} - mass flow air for stoichometric combustion,
m_{mix} - mass of reactants mixing with combustion products,
m_{ipm} - mass of working fluid in zone 1,
$m_{g,i}$ - mass fuel burned in *i* – zone.

Gaseous constant change as a function of crank angle in equations (21), (22) and (23) is given:

$$\frac{dR}{d\phi} = \frac{\partial R}{\partial \lambda}\cdot\frac{d\lambda}{d\phi} \,. \tag{36}$$

Entropy change, shown in equations (25) and (26), can be described due to energy conservation equation for each zone, expanded with second law of thermodynamics:

$$\begin{aligned} \frac{dS_1}{d\phi} &= \frac{T_{st,1}}{T_1}\cdot\frac{dS_{st,1}}{d\phi} + \frac{h_{zsi,n}}{T_1}\cdot\frac{dm_{zsi,1}}{d\phi} + \\ &+ \frac{h_{mix,n}}{T_1}\cdot\frac{dm_{mix,1}}{d\phi} + \frac{h_{irm,n}}{T_1}\cdot\frac{dm_{irm,1}}{d\phi} \,, \end{aligned} \tag{37}$$

$$\begin{aligned} \frac{dS_2}{d\phi} &= \frac{1}{T_2}\cdot\frac{dm_{g,i,2}}{d\phi}\cdot H_d \cdot \eta_{i,g} + \frac{T_{st,2}}{T_2}\cdot\frac{dS_{st,2}}{d\phi} + \frac{h_{zsi,2}}{T_2}\cdot\frac{dm_{zsi,2}}{d\phi} + \\ &+ \frac{h_{mix,n}}{T_2}\cdot\frac{dm_{mix,2}}{d\phi} + \frac{h_{zsi,n}}{T_2}\cdot\frac{dm_{zsi,2}}{d\phi} \,. \end{aligned} \tag{38}$$

3. Marine two-stroke compression igniton engine analysis

A two-stroke marine diesel engine is used for complete analysis of all thermodynamic values during the combustion process. Specifications of two-stroke diesel engine WARTSILA SULZER 6RTA72 are:

- number of cylinders: 6,
- bore: 0,720 m,
- stroke: 2,5 m,
- connecting rod: 3,468 m,
- compression ratio: 16,2
- power: 17,96 MW,
- engine speed: 1,618 s^{-1},
- mean effective pressure: 1,816 Mpa,
- inlet valves open: 38,7 °CA before BDC,
- inlet valves closes: 38,7 °CA after BDC,
- exhaust valve open: 121 °CA after TDC,
- exhaust valve closes: 246 °CA after TDC,
- fuel LHV: 42,490 MJ/kg,
- turbocharger type: VTR 564,
- cylinders are cooled with fresh water,
- piston is cooled with lubricating oil,
- camshaft is lubricated with separated lubricating oil.

The change of all thermodynamic values is observed at the beginning of the process, which starts 90° CA before TDC, and ends 90° CA after TDC. Average load of analysed marine diesel engine is 84 %.

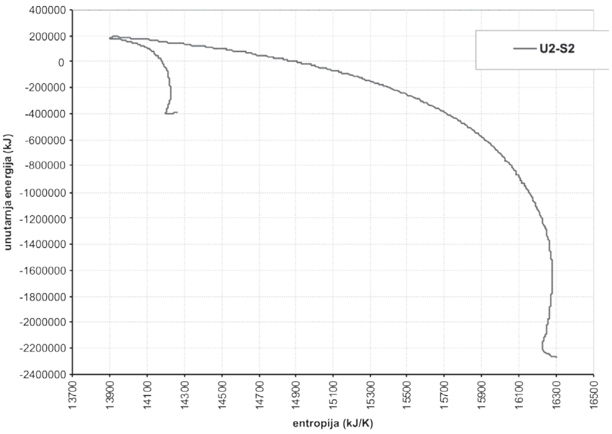

Figure 4. Internal energy change as function of entropy change

Slika 4. Dijagram ovisnosti promjene unutarnje energije o promjeni entropije

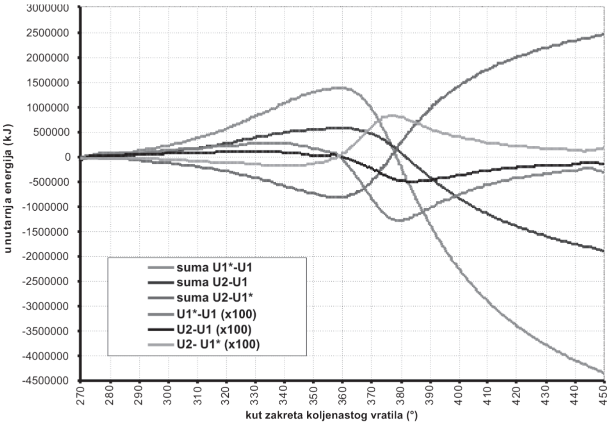

Figure 5. Internal energy and sum of internal energy changes as function of crank angle

Slika 5. Dijagram promjena unutarnjih energija i suma unutarnjih energija

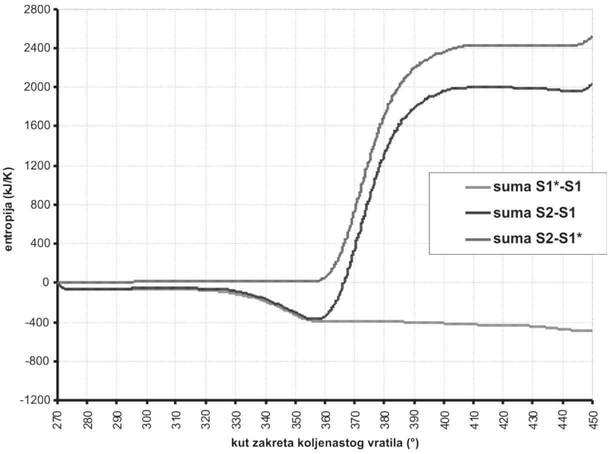

Figure 6. Sum of entropy changes as function of crank angle

Slika 6. Dijagram promjene suma entropija

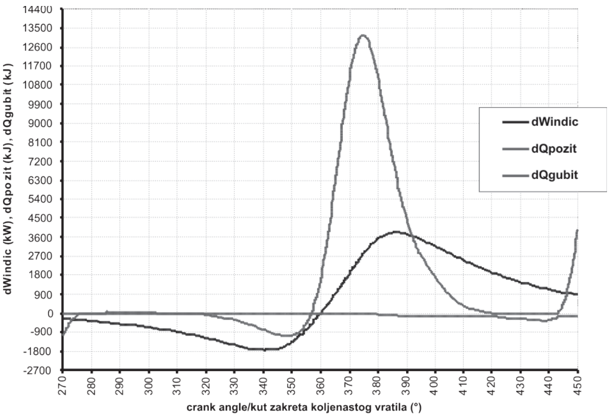

Figure 7. Indicated work, heat release rate and heat loss as function of crank angle

Slika 7. Dijagram indiciranog rada, pozitivne oslobođene topline i ukupnih gubitaka izgaranja

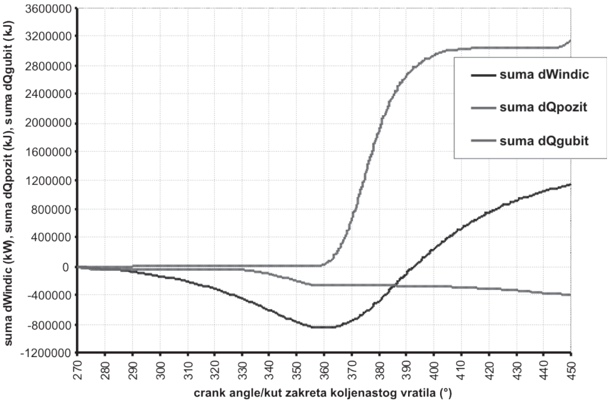

Figure 8. Sum of indicated work, heat release rate and heat loss as function of crank angle

Slika 8. Dijagram suma indiciranog rada, pozitivne oslobođene topline i ukupnih gubitaka

4. Conclusion

The global world is faced to high oil prices, so it is very important to achieve better results in compression ignition engine optimization. The main efforts are focused on lower fuel oil consumption and on higher thermal efficiency.

Analyzing thermodynamic data for SULZER marine diesel engine, it can be observed that a significant amount of energy is transferred from the system by heat lost rate to the liner wall. Maximum values of heat loss can be detected a few degrees before TDC. Also summarized values of heat loss show a strong increase on the same positions during the combustion process.

It is important to define a heat loss as summarized values of heat transferred through to the cylinder wall and heat used to prepare fuel mixture.

A reduction of heat loss is possible by decreasing the cylinder wall and cylinder head cooling. A drop in cooling results in lower energy transfer through to the cylinder wall and thus lower heat loss rates. Also it increases combustion temperature and such increase requires better construction material for the cylinder head and for the piston head. Better materials which can withstand very high temperatures are special steels and ceramic materials. Ceramic materials can withstand high temperatures but have a great disadvantage due to a significant ability to accumulate heat.

An increase of combustion temperature triggers on an increase of difference U_2-U_1, and an increase of U_2-U_1 triggers on the value decrease of U_2-U_{1*}, which represents direct work loss.

Entropy change is also connected to the heat loss value change, and there is a high increase of entropy values a few degrees after TDC. It also can be expected that entropy rise will fall by reducing the direct work loss values, i.e. difference U_2-U_{1*}.

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