Dr. sc. Dean Bernečić / ***Ph. D.***
Prof. dr. sc. Ivica Šegulja / ***Ph. D.***
Sveučilište u Rijeci / *University of Rijeka*
Pomorski fakultet u Rijeci /
Faculty of Maritime Studies Rijeka
Studentska 2, 51000 Rijeka
Hrvatska / *Croatia*

Izvorni znanstveni rad
Original scientific paper

UDK / *UDC*:
536.24
621.436.13

Primljeno / *Received:*
3. travnja 2013. / *3rd April 2013*
Odobreno / *Accepted:*
15. svibnja 2013. / *15th May 2013*

ANALIZA UTJECAJA PRIJELAZA TOPLINE NA TLAK IZGARANJA U DVOTAKTNOM SPOROOKRETNOM BRODSKOM DIZELSKOM MOTORU

HEAT TRANSFER INFLUENCE ANALYSIS ON COMBUSTION PRESURE IN TWO – STROKE SLOW – SPEED MARINE DIESEL ENGINES

SAŽETAK

U priloženom radu je analiziran utjecaj koeficijenta prijelaza topline na tlak izgaranja u cilindru. Analiza je izvršena primjenom simulacijskog modela koji se koristio za potrebe istraživanja utjecaja višestrukog ubrizgavanja na procese i produkte izgaranja u cilindru sporookretnog brodskog dizelskog motora. U svrhu analize primijenjen je nuldimenzionalni matematički model izgaranja, koji je vrednovan usporedbom stvarnih podataka s broda i rezultata dobivenih simulacijama. U radu je napravljena komparacija utjecaja prijelaza topline primjenjujući izraze za prijelaz topline prema Annandu, Eichelbergu i Woschniju. Rezultati su analizirani i prezentirani grafički te statistički obrađeni. Objašnjen je i predložen najbolji izbor.

Ključne riječi: *dvotaktni sporookretni brodski dizelski motori, koeficijent prijelaza topline, tlak izgaranja, Woschni, Eichelberg, Annand*

SUMMARY

In the presented paper, the effect of the heat transfer coefficient on the cylinder combustion pressure has been analyzed. The analysis has been performed by using a simulation model that has been used for the research into the influence of multiple injections on the combustion processes and products in the cylinder of slow-speed marine diesel engines. For the analysis purpose, the combustion nul-dimensional mathematical model has been applied, which was validated by comparing the actual data measured on board a ship and the results obtained from simulations. The paper presents the results of the heat transfer influence comparison obtained by applying formulas for the heat transfer used by Annand, Eichelberg and Woschni. The results are statistically analyzed and graphically presented. The best choice is explained and suggested.

Key words: *two-stroke slow-speed marine diesel engines, heat transfer coefficient, combustion pressure, Woschni, Eichelberg, Annand*

1. UVOD

Toplina prelazi s toplijeg na hladnije tijelo ili okolinu na tri načina:

- kondukcijom ili vođenjem – prijelaz topline između dvaju tijela u dodiru
- konvekcijom ili strujanjem – usmjereno gibanje odnosno strujanje fluida (tekućina i plinova), u kojem se topliji fluid giba prema hladnijem i predaje toplinu okolini
- radijacijom ili zračenjem – prijelaz topline koji se odvija putem elektromagnetskog zračenja.

1.1. Kondukcija (vođenje)

Prijenos topline ***vođenjem*** nastaje uslijed gibanja molekula i njihove interakcije. Kod krutih tijela prijenos topline vođenjem nastaje uslijed molekularnih vibracija. Fourier je utvrdio da je odnos Q/A [W/m^2] proporcionalan prirastu temperature dT/dx.

Fourierov izraz: $\frac{Q}{A} = -k\frac{dT}{dx}\left[\frac{W}{m^2}\right]$.

Konstanta proporcionalnosti ***k*** se naziva **koeficijent vodljivosti** [W/mK] i ovisi o materijalu (Tablica 1.).

Npr. za cilindarsku košuljicu od ljevanog željeza debljine 0,05 m i stacionarno stanje:

$$\frac{Q}{A} = -k\frac{dT}{dx} = \frac{k(T_1\text{-}T_2)}{\Delta x} = \frac{80(300\text{-}100)}{0{,}05} = 0{,}32\ MW/m^2.$$

1.2. Konvekcija (strujanje)

Newton je odredio da je odnos Q/A, kod prijenosa toplinske energije konvekcijom, proporcionalan razlici između temperature krutog tijela (T_k) s kojeg prelazi toplina na tekućinu i temperature same tekućine (T_f). Za T_k se uzima

1 INTRODUCTION

Heat transfers from a warmer to a cooler body or to the environment in three ways:

- by conduction – the heat transfer between two bodies in contact,
- by convection – heat transfer due to bulk fluid flow (liquids or gases), where the warmer fluid flows toward a cooler one and transmits the heat to the environment,
- by radiation – heat transfer due to the emission of electromagnetic waves.

1.1 Conduction

Conduction heat transfer is the transfer of energy due to the molecular motion and their interaction. Conduction heat transfer through solids is due to molecular vibration. Fourier determined that Q/A [W/m^2], the heat transfer per unit area is proportional to the temperature gradient dT/dx.

Fourier's equation: $\frac{Q}{A} = -k\frac{dT}{dx}\left[\frac{W}{m^2}\right]$

The thermal conductivity ***k*** [W/mK] depends on the materials and their temperature (table 1).

For example: for the cast iron cylinder liner of 0.05 m thickness at a steady state, it follows that

$$\frac{Q}{A} = -k\frac{dT}{dx} = \frac{k(T_1\text{-}T_2)}{\Delta x} = \frac{80(300\text{-}100)}{0.05} = 0.32\ MW/m^2$$

1.2 Convection

Newton determined that the heat transfer/area, Q/A, is proportional to the fluid solid temperature difference (T_k-T_f). T_k is the temperature of the fluid thin boundary layer at the solid

Tablica 1. Koeficijenti vodljivosti za pojedine materijale i medije [1]
Table 1 *Thermal Conductives of Common Materials* [1]

Materijal *Materials*	*k* [W/mK]
Bakar (*Copper*)	400
Aluminij (*Aluminum*)	240
Ljevano željezo (*Cast Iron*)	80
Voda (*Water*)	0.61
Zrak (A*ir*)	0.026

Tablica 2. Vrijednosti koeficijenta provodljivosti konvekcijom – *h* [1]
Table 2 *Convective Heat Transfer Coefficients* [1]

Vrsta strujanja *Convection type*	**Opis** *Description*	**Tipična vrijednost *h*** ***Typical values of h*** **(W/m^2K)**
Prirodno *Natural*	Strujanje zbog razlike u gustoći *Fluid motion induced by density differences*	10 (plin – *gas*) 100 (tekućina – *liquid*)
Prisilno *Forced*	Strujanje uzrokovano razlikom tlaka stvorenim pumpom ili ventilatorom *Fluid motion induced by pressure differences from a fan or pump*	100 (plin – *gas*) 1000 (tekućina – *liquid*)
Vrenje *Boiling*	Strujanje uzrokovano pretvaranjem tekuće faze u parnu fazu *Fluid motion induced by a change of phase from liquid to vapor*	20 000
Kondenzacija *Condensation*	Strujanje uzrokovano pretvaranjem parne faze u tekuću fazu *Fluid motion induced by a change of phase from vapor to liquid*	20 000

temperatura tankog (graničnog) sloja fluida uz samu stijenku s koje toplina prelazi:

$$\frac{Q}{A} = h(T_k - T_f)\left[\frac{W}{m^2}\right].$$

Konstanta proporcionalnosti ***h*** naziva se koeficijent provodljivosti ***konvekcijom***, a ovisi o vrsti tekućine (fluida) i njezinoj brzini. Toplinski fluks[1] ovisi o lokalnoj površini i lokalnoj razlici temperatura te se kao takav može uzeti u razmatranje ili se može uzeti kao prosječan za ukupnu površinu. Koeficijenti *h,* ovisno o vrsti strujanja vide se iz tablice 2.

Npr. za cilindarski blok s prisilnom cirkulacijom rashladne vode ($h = 1000$ W/(m²K)), temperature stijenke od 100 °C i temperature rashladnog sredstva od 80 °C:

$$\frac{Q}{A} = h(T_k - T_f) = 1000(100-80) = 20\ 000\left[\frac{W}{m^2}\right].$$

1.3. Radijacija (zračenje)

Prijenos topline ***zračenjem – radijacijom*** odvija se elektromagnetskim valovima ili fotonima koji se emitiraju s toplije površine ili s toplijeg volumena. Ovakav oblik prijenosa topline ne zahtijeva medij i može se odvijati u vakuumu. Koeficijent prijelaza topline zračenjem σ je Stefan-Boltzmanova konstanta jednaka 5,67 x 10^{-8} W/m^2K^4 [1]. Prijelaz topline zračenjem također ovisi o svojstvima materijala, izraženom preko svojstva isijavanja ε: $\frac{Q}{A} = \varepsilon\sigma T^4\left[\frac{W}{m^2}\right]$, npr.

$$\frac{Q}{A} = \varepsilon\sigma T^4 = (0,8)(5,67x10^{-8})(373)^4 = 878\left[\frac{W}{m^2}\right].$$

[1] Gustoća prijenosa toplinske energije konvekcijom.

surface from where the heat energy is transferred:

$$\frac{Q}{A} = h(T_k - T_f)\left[\frac{W}{m^2}\right].$$

The convective heat transfer coefficient ***h*** depends on the type of fluid and its velocity. The heat flux[1] depends on the local area of interest and on the local temperature differences and as such can be considered, or can be taken as the average of the total area. The coefficients *h,* depending on the fluid flow type can be seen from table 2.

For example: for a cylinder block with forced cooling water circulation ($h = 1000$ W/(m²K)), surface temperature of 100 °C and coolant temperature of 80 °C, it follows that

$$\frac{Q}{A} = h(T_k - T_f) = 1000(100-80) = 20\ 000\left[\frac{W}{m^2}\right].$$

1.3 Radiation

The radiation heat transfer is the transfer of energy due to electromagnetic waves or photons emitted from a warmer surface or volume. This type of heat transfer does not require a heat transfer medium and can occur in vacuum. The radiation coefficient σ (proportionality constant) is the Stefan Boltzman constant equal to 5.67 x 10^{-8} W/m^2K^4 [1]. The radiation heat transfer also depends on the material properties expressed by the emissivity of the material ε:

$$\frac{Q}{A} = \varepsilon\sigma T^4\left[\frac{W}{m^2}\right], \text{ for example:}$$

$$\frac{Q}{A} = \varepsilon\sigma T^4 = (0.8)(5.67x10^{-8})(373)^4 = 878\left[\frac{W}{m^2}\right].$$

[1] Convection heat transfer density.

Treba napomenuti da su za srednje razlike u temperaturama (manje od 100 ^{o}C) prijenos topline zračenjem i prijenos topline prirodnom konvekcijom približno isti.

2. PRIJELAZ TOPLINE U MOTORIMA S UNUTARNJIM IZGARANJEM

Maksimalna temperatura[2] u prostoru izgaranja (cilindru) motora s unutarnjim izgaranjem (MSUI) se danas kreće do 2500 ^{o}C, dok su maksimalne temperature metala koji su u doticaju s takvim temperaturama ograničene na znatno manje vrijednosti[3] i zato je potrebno hlađenje. Uslijed tako visokih temperaturnih razlika se javljaju vrlo veliki toplinski tokovi (fluksevi), koji mogu u vremenu izgaranja doseći i 10 MW/m^2 [2]. Tijekom ostalih procesa toplinski tok je mali ili blizu nuli. Dakle toplinski tok varira po intenzitetu, smjeru, prostoru i vremenu.

Toplinski tok je najveći u dijelu cilindra gdje su najveće temperature i brzine plinova izgaranja. Upravo na tim dijelovima potrebno je hlađenjem održavati toplinska opterećenja materijala u dozvoljenim granicama. Temperatura stijenki plinske strane košuljice se mora održavati ispod 180 ^{o}C, kako bi se zadržala dovoljna debljina uljnog sloja.[4]

Prijelaz topline utječe na radne značajke motora, iskoristivost i emisije. Za istu količinu goriva dovedenu u cilindar, veći prijelaz topline na stijenke cilindra (jače hlađenje), znači pad tlaka i prosječne temperature plinova izgaranja, smanjujući tako rad, odnosno iskoristivost.

Toplinu oduzima ili predaje voda i ulje ovisno o kojem dijelu procesa se radi. U procesu ispiranja, stijenke košuljica su obično toplije od ispirnog zraka, čija se temperatura kreće oko 40 ^{o}C i relativno je velike brzine, pa nastaje prijelaz topline na ispirni zrak. To smanjuje iskori-

[2] Kratkotrajne temperatura procesa dok se srednja temperatura procesa kreće oko 800 ^{o}C.

[3] Granične temperature za ljevano željezo (gizu) se kreću oko 400 ^{o}C, a za aluminij oko 300 ^{o}C.

[4] U teškom gorivu postoje ostaci katalizatora – katalitičke čestice (catalytic fines) koje prođu separator i filtre goriva. Vrlo su tvrde te abrazivno djeluju na košuljicu i klipne prstenove te je neophodno da plivaju u uljnom sloju čija debljina mora biti veća od njihove veličine, a smanjuje se povećanjem temperature. Samo ulje bi počelo stradavati na temperaturama iznad 270 ^{o}C kad dolazi do koksiranja. Temperatura ulja kod brzohodnih motora se kreće iznad 200 ^{o}C.

It should be noted that for the mean difference in temperature (less then 100 ^{o}C), radiation and natural convection heat transfer are about the same.

2 HEAT TRANSFER IN INTERNAL COMBUSTION ENGINES

The maximum temperature[2] in the combustion chamber (cylinder) of internal combustion engines is now moving around 2500 ^{o}C, while the maximum temperatures of metals in contact with such temperatures are limited to much lower values[3] and, therefore, cooling is necessary. Due to such a high temperature differences, very large heat fluxes occurred, which, during combustion, can reach around 10 MW/m^2 [2]. During the other processes, the heat flux is small or close to zero. So heat flux varies in intensity, direction, time and space.

The greatest cylinder heat flow is in the area where the highest combustion gases temperatures and velocities are. On these areas, it is necessary to maintain the cooling thermal load within acceptable limits. The gas side cylinder liner wall temperature should be maintained below 180 ^{o}C, in order to maintain a sufficient oil film thickness[4].

The heat transfer affects the engine performance, namely, the efficiency and the emissions. For the same fuel amount brought into the cylinder, a greater heat transfer to the cylinder liner wall (cooling increased) means the pressure and average combustion gas temperature drop, thus reducing the performance and the efficiency.

The heat is transferred to cooling water and oil depending on which part of the process is in question. In the scavenging process, the cylinder liner walls are usually warmer than the scavenging air that has a temperature of around 40 ^{0}C and has a relatively high speed so there is a heat transfer from the cylinder liner to the

[2] The temperature duration for a very short period while the process mean temperature is around 800 ^{o}C.

[3] The temperature limits for cast iron are around 400 ^{o}C and for aluminium around 300 ^{o}C.

[4] Heavy fuel oil contains catalization processed fine particles (catalytic fines) that pass the separator and the fuel filters. They are very hard and have an abrazive effect on the cylinder liner and piston rings and it is, therfore, necessary for them to flow in oil film which thickness must be greater than their size. The oil film thickness decreases with the oil temperature increasing. The lubeoil itself will start to die above 270 ^{o}C when it turns into coke.

stivost punjenja motora što opet utječe na radne značajke.

U taktu kompresije temperatura ispirnog zraka se penje te u određenom trenutku nadilazi temperaturu stijenki košuljice, glave i stapa (temperatura pladnja ispušnog ventila je i dalje veća jer se ne stigne ohladiti), brzina ispirnog zraka opada, pa nastaje prijelaz sa zraka na stijenke.

U taktu izgaranja i ekspanzije plinovi izgaranja dostižu najveće temperature procesa, brzine gibanja su velike i turbulentne te u tom vremenu postoji najveći toplinski tok s radnog medija na stijenke cilindra. Kako odmiče ekspanzija, brzine i temperature padaju pa se smanjuje i toplinski tok.

U trenutku otvaranja ispušnog ventila ponovno se povećava brzina strujanja zbog razlike tlakova u cilindru i ispušnom kolektoru te postoji nagli pad tlaka i temperature, a veliki dio topline koju sadrže plinovi izgaranja se odvodi u ispušni sustav. Dio te topline se prenosi na stijenke ispušnog ventila i ispušnog kanala koji je hlađen zbog održavanja temperaturnih granica, a preostali dio „hrani" plinsku stranu turbopuhala. Ispušni kolektor je izoliran i o kvaliteti izolacije ovisi gubitak (prijelaz) topline na stijenke ispušnog kolektora. On treba biti što manji kako bi se što više topline iskoristilo u turbopuhalu.

Promjena temperature plinova izgaranja, uzrokovana prijelazom topline, utječe na proces nastajanja polutanata, kako u samom prostoru izgaranja tako i kasnije u ispušnom sustavu.[5] Prijelaz topline utječe i na iskoristivost toplinske energije ispušnih plinova u turbopuhalu što ima značajan utjecaj na snagu, iskoristivost i radne značajke motora. Trenje između stapa, prstenova i košuljice također utječe na stvaranje dodatnog toplinskog opterećenja, a koje je rashladnim sredstvima potrebno održavati u dozvoljenim granicama.

U cilindru motora prijelaz topline se odvija konvekcijom i zračenjem. Kod dizelskih motora, za razliku od benzinskih, zračenje može imati značajan udio u ukupnom toplinskom toku, te ga treba uzeti u razmatranje.

U praktičnim proračunima i simulacijama se najčešće koriste koeficijenti prijelaza topline dobiveni eksperimentalnim putem za pojedini

[5] U ispušnom sustavu dogorijevaju CO i CH.

scavenging air. This reduces the engine turbocharging efficiency that influences again the engine performances.

In the compression stroke the scavenging air temperature rises and at some points is beyond the temperatures of the cylinder liner wall, cylinder head and piston (the exhaust valve seat temperature is still higher because there is no time to cool it), the scavenging air velocity decreases and the heat transfer from air to wall occurs.

In the combustion and expansion strokes, the combustion gases reach the highest temperatures of the process, gas velocities are high and turbulent and at that time there is the largest heat transfer (flux) from the working fluid to the cylinder walls. As the expansion progresses, the gas speed and the temperatures are dropping thus reducing the heat flux.

When the exhaust valve opens, due to the pressure difference in the cylinder and exhaust gas manifold, the gas velocity is increasing again and there is a sudden cylinder pressure and temperature drop, and a large part of heat, that contains combustion gases, is discharged into the exhaust gas manifold. A part of this heat is transferred to the exhaust valve walls and to the exhaust valve channel which is cooled to maintain the temperature limits, and the remaining part of the heat drives the rotor of the turbo-charger. The exhaust gas manifold is isolated and the heat transfer (lost) through it depends on the insulation material quality. It should be as small as possible in order to get more heat to be used in the turbo-charger.

The combustion gases temperature change, caused by heat transfer, affects the pollutants formation process, both in the area of combustion and later in the exhaust gas system[5]. The heat transfer affects the turbo-charger exhaust gas thermal energy efficiency that has a significant impact on the engine power, efficiency and performance. The friction between the piston, piston rings and cylinder liner also contributes to the generation of additional heat load, and the coolant is needed to maintain the permitted limits.

The heat transfer into the cylinder is applied by convection and radiation. For diesel engines, unlike gasoline, radiation can have a significant part in the total heat transfer, and it should be taken into consideration.

[5] CO and CH are burning in the exhaust system.

tip motora, koji u sebi sadrže konvektivni udio prijelaza topline tako i udio zračenja.

Količina topline, koja se konvekcijom preko stijenki predaje rashladnoj vodi, kreće se od 1/3 do 1/4 ukupne kemijske energije (topline) dovedene gorivom. Otprilike polovica te topline se prenosi na stijenke unutar cilindra, dok se veći dio preostale polovice prenosi na stijenke ispušnog kanala, u slučaju kada kanal nije izoliran.

Za vrijeme radnog ciklusa stvarni toplinski tok varira u vremenu i prostoru. Način i izbor matematičkog opisa, a time i kompleksnost simulacija ovisi o predmetu istraživanja.

Može se reći da postoje dvije krajnosti. Prva, jednostavnija, je proračun ukupnog prijelaza topline na rashladni medij pomoću srednjeg koeficijenta prijelaza topline na cijeloj površini u vremenu i prostoru. U tu svrhu se najčešće koriste empirijske ili poluempirijske jednadžbe.

Druga, za proračun znatno teži izbor, je slučaj kada se ispituju termička opterećenja pojedinih dijelova motora.[6] U takvim slučajevima, procjena prostorne raspodjele i vremenske promjene toplinskog toka mora biti što preciznija, pa se stoga koriste i znatno kompliciraniji algoritmi koristeći višedimenzionalne modele.

Najčešći slučajevi predviđanja snage, iskoristivosti i emisija „padaju" unutar ove dvije krajnosti, tj. QD modeli daju zadovoljavajuće rezultate.

U ovome radu se ne promatra detaljan utjecaj na toplinska opterećenja pojedinih dijelova motora, pa se koristio srednji koeficijent prijelaza topline.

Vremenska procjena toplinskog toka unutar cilindra je važna, ali prostorna raspodjela zna biti puno važnija. Zahtijevana točnost procjene nije velika jer pogreška od 10 % u procjeni ukupnog prijelaza topline za vrijeme ciklusa obično generira pogrešku unutar ± 1 % kod proračuna snage i iskoristivosti. Za istu vrijednost pogreške od 10 %, a kod predviđanja CO, utjecaj je zanemariv, za NOx je oko 1 %, dok je za CH oko 2 % [3].

[6] Stap, glava, ispušni ventil ili slično.

In practical calculation and simulations, the heat transfer coefficients, experimentally obtained for each type of the engine and which contain the convective and radiation part of the heat transfer, are commonly used.

The heat amount, which is transferred to the cooling water through the cylinder liner wall, ranges from 1/3 to 1/4 of the total chemical energy brought into the cylinder by the fuel.

Approximately half of this heat is transferred to the walls inside the cylinder while most of the remaining half goes to the wall of the exhaust gas channel, in a case when the channel is not isolated.

During the working cycle the heat flux varies in time and space. The method and selection of the mathematical description, and thus the complexity of the simulation depends on the research matter.

It can be said that there are two extremes. The first, the simpler one, applies the total cooling water heat transfer calculation by using the medium heat transfer coefficient over the entire surface in time and space. For this purpose empirical or semi-empirical equations are the most common in use.

The second, for the calculation a significantly heavier choice, is the case when certain engine parts[6] thermal loads have to be examined.[6] In such cases, the assessment of the heat flux spatial distribution and time change should be as accurate as possible and, therefore, much more complicated algorithms with multidimensional models are used.

The most common cases of the power prediction, efficiency and emissions fall within the above-mentioned two extremes, i.e. the QD (quasidimensional models) give satisfactory results.

A detailed effect on certain engine parts thermal loads is not observed in this paper, so that the medium heat transfer coefficient has been used.

The heat time change estimation within the cylinder is important but the surface heat distribution can be much more important. The required accuracy of the assessment is not great because of the 10% error in estimating the total heat transfer during a cycle usually generates an error within ±1% when power and efficiency has to be calculated. For the same error val-

[6] Piston, head, exhaust valve or the like.

3. KOEFICIJENTI PRIJELAZA TOPLINE

Istraživanjima [4] i [5] je utvrđeno da je temperatura stijenki košuljice, klipa i glave u stacionarnom režimu rada konstantna, stoga se može računati sa srednjom temperaturom površina. Razlika koeficijenata prijelaza topline po površini cilindra, a u svrhu istraživanja u ovome radu, može se zanemariti te se prihvaća srednji koeficijent prijelaza topline – α_κ.

Brzina prijelaza topline konvekcijom može se izraziti kao:

$$\frac{dQ_{st}}{d\varphi} = \sum_{i=1}^{n} \alpha_\kappa \cdot A_{st} \left(T_{st,i} - T_C\right) \frac{dT}{d\varphi}.$$

Površina stijenke na kojoj se odvija prijelaz topline jednaka je izloženoj površini stijenke cilindra ($A_{C,i}$) uvećanoj za dio površine stapa do prvog stapnog prstena:

$$A_{st} = \sum_i A_{C,i} + 2 \cdot d_C \cdot \pi \frac{h_K}{3},$$

gdje je h_K visina boka stapa od čela do prvog stapnog prstena.

U gornjem izrazu najveća nepoznanica je α_κ. Kako se o ponašanju (gibanju) plina unutar cilindra malo zna, najčešće se za izračunavanje α_κ koriste empirijske formule dobivene eksperimentalnim mjerenjima za razne slučajeve.

Sve jednadžbe za izračunavanje koeficijenta prijelaza topline polaze od Nusseltove teorije prijelaza topline. Taj bezdimenzionalni broj definiran je izrazom:

$Nu = \frac{\alpha \cdot b}{K}$, gdje je $\alpha \left[\frac{\mathrm{W}}{\mathrm{m^2K}}\right]$ koeficijent prijelaza topline, b [m] je debljina stijenke, a $K\left[\frac{\mathrm{W}}{\mathrm{mK}}\right]$ je koeficijent toplinske vodljivosti.

Na osnovi Nusseltove značajke i eksperimentalnih mjerenja, Woschni [4], [6], je predložio jednadžbu za srednji koeficijent prijelaza topline α_κ koju koriste i citiraju mnogi autori.

U svrhu istraživanja prezentiranom u ovome radu, u simulacijskom programu razvijenom i primijenjenom za potrebe doktorske disertacije [7], također se koristio Woschnijev izraz, budući da je za istraživanu temu davao najbolje rezultate.

ue of 10%, the influence on the CO production prediction is negligible, for NOx it is around 1%, while for CH it is around 2% [3].

3 HEAT TRANSFER COEFFICIENTS

Studies [4] and [5] have found that the cylinder liner walls, piston and cylinder head temperature in a stationary mode is a constant one, so that the mean surface temperature can be counted on. The heat transfer coefficients difference over the cylinder liner surface can be ignored, for the purpose of this study, and the mean heat transfer coefficient accepted.

The convection heat transfer speed can be expressed as:

$$\frac{dQ_{st}}{d\varphi} = \sum_{i=1}^{n} \alpha_\kappa \cdot A_{st} \left(T_{st,i} - T_C\right) \frac{dT}{d\varphi}.$$

The wall surface where the heat transfer takes place is equal to the cylinder liner wall exposed surface ($A_{C,i}$), increased for a portion of the piston surface up to the top piston ring:

$$A_{st} = \sum_i A_{C,i} + 2 \cdot d_C \cdot \pi \frac{h_K}{3},$$

where α_κ is the height from the piston top up to the top piston ring.

In the above expression the greatest unknown is α_κ. Since the gas behaviour (flow) inside the cylinder is a little known, therefore, for calculating α_κ empirical formulas obtained by the experimental measurements of various cases, are commonly in use.

All equations for heat transfer coefficients calculation are based on the Nusselt's heat transfer theory. This dimensionless number is defined by the term:

$Nu = \frac{\alpha \cdot b}{K}$, where $\alpha \left[\frac{\mathrm{W}}{\mathrm{m^2K}}\right]$ is the heat transfer coefficient, b [m] is the wall thickness, and $K\left[\frac{\mathrm{W}}{\mathrm{mK}}\right]$ is the thermal conductivity coefficient.

Based on the Nusselt's expression and experimental measurements, Woschni [4], [6], has proposed an equation for the mean heat transfer coefficient α_κ, which is used and cited by many authors.

Izbor α_κ ovisi o brzini plina na površini cilindra, odnosno izmjena topline između radnog medija i unutarnje površine cilindra vrši se najvećim dijelom prisilnom konvekcijom. Tako u literaturi nalazimo veliki broj izraza za α_κ vezanih za ovaj slučaj, a neki od najpoznatijih su:

3.1. Woschni

$$\alpha_\kappa = 130,5 \cdot D_C^{-0,2} \cdot p_C^{0,8} \cdot T_C^{-0,53} \cdot w^{0,8} \left[\frac{\mathrm{W}}{\mathrm{m^2\ K}}\right],$$ gdje je:

D_C – promjer cilindra [m]

p_C – tlak [bar]

T_C – temperatura cilindra [K]

$$w = C_1 \cdot c_m + C_2 \cdot (p_C - p_{C,K}) \cdot \frac{V_S \cdot T_{C,UZ}}{p_{C,UZ} \cdot V_{C,UZ}},$$ gdje je:

c_m – srednja stapna brzina [m/s]

V_S – stapajni volumen [m³]

$p_{C,UZ}$, $T_{C,UZ}$, $V_{C,UZ}$ – tlak, temperatura i volumen u trenutku zatvaranja usisa (kod sporookretnog dvotaktnog brodskog motora je to trenutak zatvaranja ispušnog ventila)

$C_1 = 6,18 + 0,417 \cdot c_{vr} / c_m$ – tijekom izmjene radnog medija

$C_1 = 2,28 + 0,308 \cdot c_{vr} / c_m$ – tijekom kompresije ili ekspanzije

$C_2 = 0,00324\ ms^{-1}K^{-1}$ – za dizelske motore s direktnim ubrizgavanjem

$C_2 = 0,00622\ ms^{-1}K^{-1}$ – za dizelske motore s pretkomorom

c_{vr} / c_m – omjer brzine vrtloga i srednje stapne brzine.

3.2. Eichelberg

$$\alpha_\kappa = 2,83 \cdot c_m^{0,33} \left(p_C T_C\right)^{0.5} \left[\frac{\mathrm{W}}{\mathrm{m^2\ K}}\right].$$

3.3. Annand

$$\alpha_\kappa = a \cdot \left(\frac{\lambda}{D_C}\right) \cdot \mathrm{Re}^{0,7} + \frac{C}{T_C - T_{st}} \cdot \left[\left(\frac{T_C}{100}\right)^4 - \left(\frac{T_{st}}{100}\right)^4\right] \left[\frac{\mathrm{W}}{\mathrm{m^2\ K}}\right],$$

For the purpose of the research presented in this paper, the Woschni's expression is also used into the simulation programme specially developed for the doctoral thesis [7], since it gives the best results for the researched topics.

The choice of α_κ depends on the gas velocity at the cylinder liner surface, i.e. the heat exchange between the working fluid and the cylinder liner inner surface is carried out mainly by forced convection. Thus, related to this case, in the literature a large number of expressions for α_κ can be found in literature, and some of the best known are as the following ones:

3.1 Woschni

$$\alpha_\kappa = 130.5 \cdot D_C^{-0.2} \cdot p_C^{0.8} \cdot T_C^{-0.53} \cdot w^{0.8} \left[\frac{\mathrm{W}}{\mathrm{m^2\ K}}\right],$$ where

D_C – is the cylinder diameter [m],

p_C – is the cylinder pressure [bar],

T_C – is the cylinder temperature [K],

$$w = C_1 \cdot c_m + C_2 \cdot (p_C - p_{C,K}) \cdot \frac{V_S \cdot T_{C,UZ}}{p_{C,UZ} \cdot V_{C,UZ}},$$ where

c_m – is the mean piston speed [m/s],

V_S – is the stroke volume [m³],

$p_{C,UZ}$, $T_{C,UZ}$, $V_{C,UZ}$ – pressure, temperature and volume at the inlet valve closing time (in the two-stroke slow-speed marine diesel engine that is the exhaust valve closing time), where

$C_1 = 6.18 + 0.417 \cdot c_{vr} / c_m$ – during the working fluid exchange,

$C_1 = 2.28 + 0.308 \cdot c_{vr} / c_m$ – during the compression or expansion,

$C_2 = 0.00324\ ms^{-1}K^{-1}$ – for direct injection diesel engines,

$C_2 = 0.00622\ ms^{-1}K^{-1}$ – for combustion chamber diesel engines,

c_{vr} / c_m – vortices – secondary piston speed ratio.

3.2 Eichelberg

$$\alpha_\kappa = 2.83 \cdot c_m^{0.33} \left(p_C T_C\right)^{0.5} \left[\frac{\mathrm{W}}{\mathrm{m^2\ K}}\right].$$

gdje je:

a – koeficijent koji se povećava povećanjem brzine [0,17 – 0,93]

C – koeficijent koji ovisi o taktovima i tipu motora:

C = 0 za takt kompresije kod benzinskih motora

C = 0,43 za ostale taktove

C = 3,27 za dizelske motore.

λ – toplinska provodljivost u $\left[\frac{\mathrm{W}}{\mathrm{m}^2\ \mathrm{K}}\right]$,

D_C – promjer cilindra [m]

Re – Reynoldsov broj, $\mathrm{Re}=\frac{\rho \cdot D \cdot c_m}{\mu}$, gdje je ρ gustoća, a μ kinematički viskozitet.

3.4. Nusselt

$$\alpha_\kappa = 1{,}15 \cdot \left(1+24 \cdot c_m\right) \cdot \left(p_C^2 \cdot T_C\right)^{2/3} \quad \left[\frac{\mathrm{W}}{\mathrm{m}^2\ \mathrm{K}}\right].$$

3.5. Brilling

$$\alpha_\kappa = 1{,}15 \cdot \left(3{,}5+0{,}185 \cdot c_m\right) \cdot \left(p_C^2 \cdot T_C\right)^{2/3} \quad \left[\frac{\mathrm{W}}{\mathrm{m}^2\ \mathrm{K}}\right].$$

Modifikacije Woschnijevog izraza, a ovisno o tome radi li se o benzinskim ili dizelskim motorima, brzookretnima ili srednjookretnima, ovisno o vrsti procesa i ostalim uvjetima ispitivanja, predložili su razni autori. Neki od njih su Hohenberg, Asley-Campbell, Kolesa, Schwarz, Huber, Vogel i Gerstle.

3.6. Hohenberg

$$\alpha_\kappa = 130 \cdot V_C^{-0{,}06} p_C^{0{,}8} T_C^{-0{,}4} \left(c_m + 1{,}4\right)^{0{,}8} \left[\frac{\mathrm{W}}{\mathrm{m}^2\ \mathrm{K}}\right],$$ gdje je:

V_C – trenutni volumen cilindra [m^3]

p_C – tlak u cilindru [bar]

T_C – temperatura u cilindru [K]

vrijednost 130 i 1,4 su koeficijenti C_1 i C_2 koji mogu varirati od slučaja do slučaja.

3.3 Annand

$$\alpha_\kappa = a \cdot \left(\frac{\lambda}{D_C}\right) \cdot \mathrm{Re}^{0.7} + \frac{C}{T_C - T_{st}} \cdot \left[\left(\frac{T_C}{100}\right)^4 - \left(\frac{T_{st}}{100}\right)^4\right] \quad \left[\frac{\mathrm{W}}{\mathrm{m}^2\ \mathrm{K}}\right],$$

where

a – is the coefficient increased by increasing the speed [0.17 up to 0.93],

C – is the coefficient that depends on strokes and the engine type:

C = 0 for the compression stroke of gasoline engines,

C = 0.43 for other strokes,

C = 3.27 for diesel engines.

λ – is the thermal conductivity $\left[\frac{\mathrm{W}}{\mathrm{m}^2\ \mathrm{K}}\right]$,

D_C – is the cylinder diameter [m],

Re – is the Reynold’s number, $\mathrm{Re}=\frac{\rho \cdot D \cdot c_m}{\mu}$, where ρ is the density, and μ is the kinematic viscosity.

3.4 Nusselt

$$\alpha_\kappa = 1.15 \cdot \left(1+24 \cdot c_m\right) \cdot \left(p_C^2 \cdot T_C\right)^{2/3} \quad \left[\frac{\mathrm{W}}{\mathrm{m}^2\ \mathrm{K}}\right].$$

3.5 Brilling

$$\alpha_\kappa = 1.15 \cdot \left(3.5 + 0.185 \cdot c_m\right) \cdot \left(p_C^2 \cdot T_C\right)^{2/3} \quad \left[\frac{\mathrm{W}}{\mathrm{m}^2\ \mathrm{K}}\right].$$

The Woschni’s expression modifications have been proposed by various authors, depending on whether it is the case of petrol or diesel engines, of high or middle speed engines, and depending on the type of process and other conditions as well. To mention only some of them: Hohenberg, Asley-Campbell, Kolesa. Schwarz, Huber, Vogel and Gerstle.

3.6 Hohenberg

$$\alpha_\kappa = 130 \cdot V_C^{-0.06} p_C^{0.8} T_C^{-0.4} \left(c_m + 1{,}4\right)^{0.8} \left[\frac{\mathrm{W}}{\mathrm{m}^2\ \mathrm{K}}\right],$$

where;

V_C – is the current cylinder volume [m^3],

p_C – is the cylinder pressure [bar],

3.7. Asley-Campbell

$\alpha_\kappa = 0{,}13 \cdot D_C^{0{,}12} \cdot p_C^{0{,}8} \cdot T_C^{-0{,}5} \cdot Z^{0{,}8} \left[\frac{\mathrm{W}}{\mathrm{m}^2\ \mathrm{K}}\right]$, gdje je Z brzina radnog medija [m/s].

3.8. Schwarz

Kolesa (1987.) dolazi do rezultata koji pokazuju da vrijednosti koeficijenta prijelaza topline znatno rastu na temperaturama iznad 600 K, dok Schwarz (1993.) razvija konstantnu funkciju za koeficijent C_2 na višim temperaturama [8]:

$C_2^* = C_2 + 23 \cdot 10^{-6}(T_{st} - 525)$, za $T_{st} \geq 525\ ^0C$.

3.9. Huber

Huber (1990.) [8] dokazuje da korigirani Woschnijev izraz za brzinu daje dobre rezultate kod izgaranja, ali na malim opterećenjima su rezultati premali, pa izraz za w korigira u:

$$w = C_1 \cdot c_m \cdot \left[1 + 2\left(\frac{V_C}{V}\right)^2 \cdot p_{sr,ind}^{-0{,}2}\right] \ za\ 2 \cdot C_1 \cdot c_m \left[\frac{V_C}{V(\varphi)}\right]^2 \cdot$$

$$\cdot\, p_{sr,ind}^{-0{,}2} \geq C_2 \cdot \frac{V_S T_1}{p_1 V_1} \cdot (p - p_0),$$

gdje je p tlak izgaranja, a p_0 tlak kompresije. Vrijedi također kad je $p_{r,ind} \leq 1 \rightarrow p_{sr,ind} = 1$.

3.10. Vogel

Vogel (1995.) [8] nastavlja Huberova istraživanja te ispituje ponašanje koeficijenata prijelaza topline kod stvaranja naslaga ulja, čađe i sl. na unutrašnjoj strani prostora izgaranja. Uvodi takozvanu Vogelovu konstantu C_3, pa Huberova jednadžba prelazi u:

$$w = C_1 \cdot c_m \cdot \left[1 + 2\left(\frac{V_C}{V}\right)^2 \cdot C_3\right], \text{ za}$$

$$2 \cdot C_1 \cdot c_m \left[\frac{V_C}{V(\varphi)}\right]^2 \cdot C_3 \geq C_2 \cdot \frac{V_S T_1}{p_1 V_1} \cdot (p - p_0),$$

gdje je za dizelske motore:

$$C_2 = 3{,}24 \cdot 10^{-3} \left[\frac{m}{s\ K}\right]$$

$C_3 = 1 - 1{,}2 \cdot e^{-0{,}65 \cdot \lambda}$, gdje je λ toplinska vodljivost [W/(mK)].

T_C – is the cylinder temperature [K]

values 130 i 1.4 are the coefficients C_1 i C_2, which may vary from case to case.

3.7 Asley-Campbell

$\alpha_\kappa = 0.13 \cdot D_C^{0.12} \cdot p_C^{0.8} \cdot T_C^{-0.5} \cdot Z^{0.8} \left[\frac{\mathrm{W}}{\mathrm{m}^2\ \mathrm{K}}\right]$, where Z is the working fluid speed [m/s].

3.8 Schwarz

The Kolesa's results (1987) have shown that the heat transfer coefficient value significantly grow at temperatures above 600 K, till Schwarz (1993) has developed a constant function for the coefficient C_2 at higher temperatures [8];

$C_2^* = C_2 + 23 \cdot 10^{-6}(T_{st} - 525)$, for $T_{st} \geq 525\ ^0C$.

3.9 Huber

Huber (1990) [8] has proved that the modified Woschni's term for speed gives good combustion results, since the results at low loads have been to small, so that the expression for w are corrected to:

$$w = C_1 \cdot c_m \cdot \left[1 + 2\left(\frac{V_C}{V}\right)^2 \cdot p_{sr,ind}^{-0{,}2}\right] \ za\ 2 \cdot C_1 \cdot c_m \left[\frac{V_C}{V(\varphi)}\right]^2 \cdot$$

$$\cdot\, p_{sr,ind}^{-0{,}2} \geq C_2 \cdot \frac{V_S T_1}{p_1 V_1} \cdot (p - p_0),$$

where p is the combustion pressure, and p_0 is the compression pressure. This also applies when $p_{r,ind} \leq 1 \rightarrow p_{sr,ind} = 1$.

3.10 Vogel

Vogel (1995) [8] has continued with the Huber's research, looking into the heat transfer coefficient behaviour in case of oil deposits formation, soot formation, etc. on the inner side of the combustion chamber. He has introduced the so-called Vogel's constant C_3, so the Huber's equation turns into as:

$$w = C_1 \cdot c_m \cdot \left[1 + 2\left(\frac{V_C}{V}\right)^2 \cdot C_3\right], \text{ for}$$

$$2 \cdot C_1 \cdot c_m \left[\frac{V_C}{V(\varphi)}\right]^2 \cdot C_3 \geq C_2 \cdot \frac{V_S T_1}{p_1 V_1} \cdot (p - p_0),$$

Kod velikih srednjookretnih 4T i sporookretnih 2T motora primjena gornjih jednadžbi daje odstupanja između izmjerenih i proračunatih vrijednosti temperature ispušnih plinova od nekih 20 K [8]. To se u proračunu manifestira kao dovod plinova manje entalpije na turbinu što kao posljedicu ima manji tlak ispirnog zraka. Za velike motore je to važno zbog toga što su oni najčešće optimirani na stacionarnu radnu točku na kojoj najčešće rade.

3.11. Gerstle

Gerstle je 1999. modificirao Woschnijevu jednadžbu za slučaj ispiranja i punjenja svježeg zraka. Dakle, stari izraz za C_1 se uvećava za konstantu $k = 6{,}5 \div 7{,}2$, a važi za vrijeme od otvaranja ispušnog ventila pa do zatvaranja ispirnih otvora:

$$C_1 = k\left(2{,}28 + 0{,}308\frac{c_{vr}}{c_m}\right),\ k = 6{,}5 \text{ do } 7{,}2.$$

4. UTJECAJ KOEFICIJENATA PRIJELAZA TOPLINE NA TLAKOVE IZGARANJA

Za potrebe istraživanja analizirana su tri najčešće spominjana izraza za koeficijente prijelaza topline, Annandov, Eichelbergov i Woschnijev te je izvršena analiza dobivenih rezultata.

Na slici 1. je prikazana usporedba tlakova dobivenih mjerenjem i onih dobivenih simulacijom uz korištenje ova tri izraza za koeficijente prijelaza topline.

Iz slike je vidljivo da se najbolja preklapanja izmjerenih i stvarnih vrijednosti, za ispitivani sporookretni motor tipa Wärtsilä 7 RT Flex 50, dobivaju korištenjem upravo Woschnijevog izraza. Izrazi Annanda i Eichelberga rezultiraju premalim prijelazom topline pa su temperature i tlakovi kompresije, izgaranja i ekspanzije preveliki u odnosu na stvarne, izmjerene vrijednosti.

Statistički pokazatelji (koeficijent determinacija (R^2) i drugi korijen sredine kvadrata odstupanja promatranih od procijenjenih vrijednosti (RMSE)), dani su u tablici 3.

where for diesel engines:

$$C_2 = 3.24 \cdot 10^{-3}\ \left[\frac{m}{s\ K}\right],$$

$C_3 = 1 - 1.2 \cdot e^{-0.65 \cdot \lambda}$, where λ is the thermal conductivity [W/(mK)].

In large middle-speed four stroke and slow-speed two stroke diesel engines the above-mentioned equation application gives the deviation between the measured and calculated values of the exhaust gas temperature of about 20 K [8]. In the simulation results, this is presented as less enthalpy gas supply to turbo charger which results in a lower scavenging air pressure. For large engines this is important because they are usually optimized to stationary operating point at which they most often operate.

3.11 Gerstle

In 1999, Gerstle modified the Woschni's equation for the case of scavenging and charging of fresh air into the cylinder. The old term for C_1 is increased for the constant $k = 6.5 \div 7.2$, and is valid for a period from the exhaust valve opening till the scavenging ports closing;

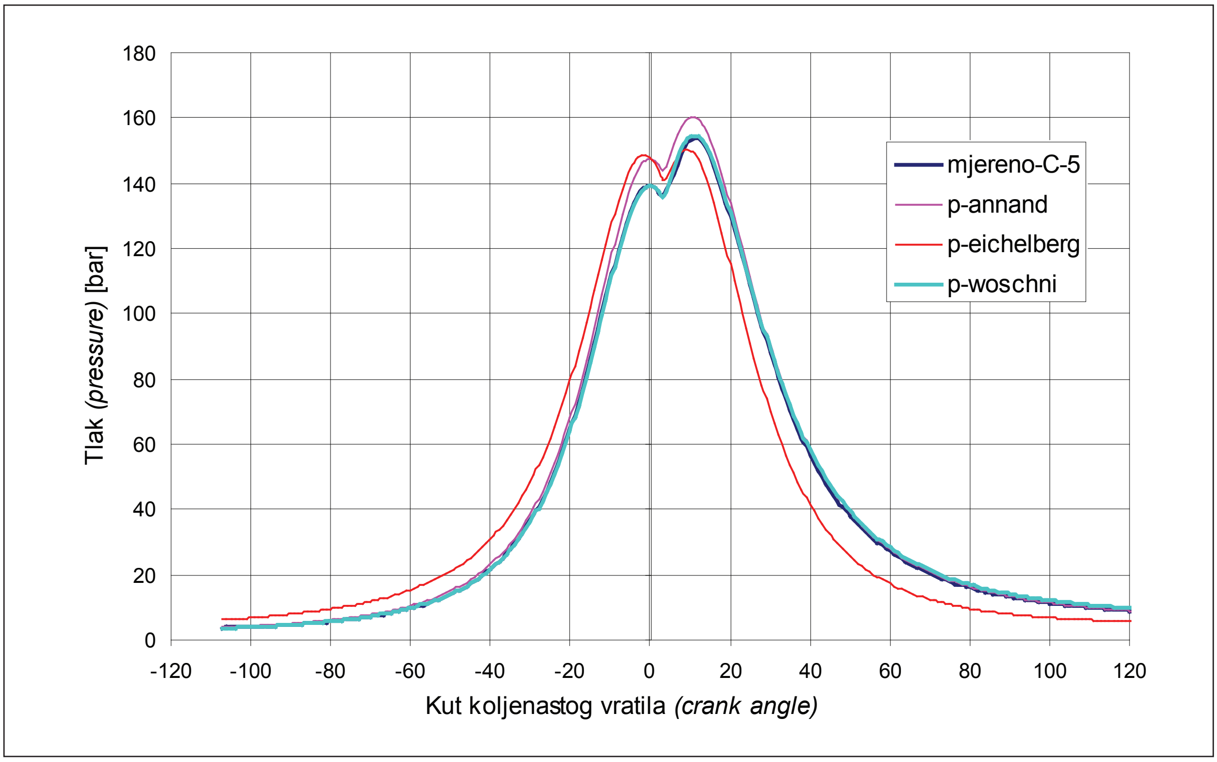
$$C_1 = k\left(2.28 + 0.308\frac{c_{vr}}{c_m}\right),\ k = 6.5 \text{ to } 7.2.$$

4 HEAT TRANSFER COEFFICIENTS IMPACT TO COMBUSTION PRESSURE

For the research purpose, the three most frequently mentioned terms of heat transfer coefficients, Annand, Eichelberg and Woschni, are taken into consideration and the results analysis has been made.

The comparison of the measured values and of the values obtained by simulation and based on the three heat transfer coefficient expressions are given in figure 1.

The figure shows that the best overlap of the measured and simulated values for the tested two-stroke slow-speed marine diesel engine Wärtsilä 7 RT Flex 50 has been obtained by using the Woschni's formula. The formula by Annand and Eichelberg gives little heat transfer so at the end the combustion temperature and pressure are too high as compared to the measured values.



Slika 1. Usporedba stvarnih vrijednosti tlaka i simulacijom dobivenih tlakova na motoru Wärtsilä 7 RT Flex 50 pri 100 % maksimalne snage koristeći različite izraze za prijelaz topline
Figure 1 *Simulated indicated pressures comparission for Wärtsilä 7 RT Flex 50 on 100 % MCR using different heat transfer formulas*

Tablica 3. R^2 i RMSE za Woschnija, Eichelberga i Annanda
Table 3 *R^2 and RMSE for Woschni, Eichelberg andi Annan*

	Annand	Eichelberg	Woschni
Koeficijent determinacije (R^2)	0,997884	0,976772	0,999822
RMSE	2,904208	9,621429	0,842647

5. ZAKLJUČAK

Koeficijent prijelaza topline ima veliki utjecaj na rezultate simulacija procesa izgaranja te je potrebno odabrati najbolji. Važno je napomenuti da određivanje najpogodnijeg izraza za prijelaz topline nije jednostavno jer ovisi o velikom broju parametara. Isti izrazi za različite tipove motora ne daju uvijek najbolje rezultate.

Razni autori su se bavili istraživanjima ove teme, većinom na motorima manjih snaga i malih promjera cilindara te manjih stapnih brzina, što ima znatan utjecaj na ispitivanu temu. Sporookretni motori imaju svoje specifičnosti koje utječu, kako na sam proces izgaranja, tako i na proces prijelaza topline. Stoga, prilikom modeliranja rada motora treba biti pažljiv kod odabira izraza za koeficijent(e) prijelaza topline, kao i cijelog izraza za prijelaz topline.

Statistical indicators R^2 and RMSE are given in table 3.

5 CONCLUSION

The heat transfer coefficient has a major effect on the combustion process simulation results and there is a need to choose the best one for a particular research. It should be noted that the determination of the most appropriate term for the heat transfer is not easy because it depends on a number of parameters. The same heat transfer equations for different types of engines do not always provide the best results.

Various authors have dealt with this research, mostly with the low-power and small diameter IC engines and low piston speed which has a significant impact on the research topics. Large

Ovim radom su prikazani rezultati istraživanja provedeni na dva najčešća tipa sporookretnih motora nove generacije MAN B&W ME 60 i Wärtsilä RT Flex 50. Uz napomenu da nisu uzimane u obzir lokalne temperature, već se model pojednostavio uzimanjem srednjih temperatura ispušnih plinova, srednje temperature košuljice i srednje temperature rashladne vode, na oba tipa motora, najbolji rezultati simulacija radnih procesa dobiveni su primjenom Woschnijevog izraza.

slow-speed diesel engines have their own peculiarities that affect, both the combustion and the heat transfer process. Therefore, when modeling the engine processes, a careful attention should be paid in choosing the right heat transfer coefficient(s), as well as the entire heat transfer formula.

This paper aims at presenting the results of the research carried out on the two most common types of slow-speed marine diesel engines of the MAN B&W ME 60 and Wärtsilä RT Flex 50 new generation. It should be also noted that the local cylinder temperature are not taken into consideration, but the model is simplified by taking mean exhaust gas temperatures, mean cylinder liner wall temperatures and mean cooling water temperature for one combustion process. The best results were the ones obtained by using the Woschni's fomula.

LITERATURA / *REFERENCES*

[1] *Modelling of turbocharged engines in transient operation, Part 1*: *Insigt into the Relevant Physical Phenomena*, Proceedings of the Institution of Mechanical Engineers: Journals of Automobile Engineering, Professional Engineering Publishing, Volume 216 (2002), 5. http://www.engr.colostate.edu

[2] Heywood, J. B., *Internal Combustion Engine Fundamentals*, New York, McGraw-Hill, 1989.

[3] Benson, R. S., *The Thermodinamics and Gas Dinamics of Internal-Combustion Engines*, Volume II, Clarendon Press, 1986.

[4] Woschni, G., *Die Berechtung der Wandverluste und thermischen Belastung der Bauteile von Dieselmotoren*, MTZ 31 (1970), 12, 491-499.

[5] Löhner, K., Döhring, E., Chore, G., *Temperaturschwingungen an der Innenwand von Verbrennungskraftmaschinen*, MTZ, 1956., 12.

[6] Woschni, G., *Einfluß von Rußablagerungen auf den Wärmeübergang zwischen Arbeitsgas und Wand im Dieselmotor*, in Proceedings to Der Arbeitsprozeß des Verbrennungsmotors, Graz 1991.

[7] Bernečić, D., *Analiza utjecaja višestrukog ubrizgavanja na procese i produkte izgaranja u sporookretnom brodskom dizelskom motoru*, doktorski rad, Pomorski fakultet, Sveučilište u Rijeci, 2011.

[8] Merker, G. P., Schwarz, C., Stiesch, Otto, G. F., *Simulating Combustion*, Springer – Verlag Berlin Heidelberg, 2006.