Numerično modeliranje hladilnega sistema na motorju z notranjim zgorevanjem

Numerical Modelling of an Engine-Cooling System

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V prispevku je predstavljen simulacijski model za analizo prenosa toplote in snovi v hladilnem sistemu motorja v prehodnem režimu delovanja. Simulacijski model ima modulno sestavo in je sestavljen iz različnih podmodelov hladilnega sistema, ki so med seboj povezani s toplotnimi povezavami. Analizo je mogoče izvesti učinkovito le, če so povezave med posameznimi modeli upoštevane sočasno. V prispevku je zaradi svoje zahtevnosti in različnosti vgrajenih naprav predstavljen model hladilnega sistema za plovila, ki vsebuje dva ali več hladilnih krožnih tokov. Raven temperatur v posameznih delih hladilnega sistema se vzdržuje z uporabo termostatskih ventilov in z mešanjem različnih tokov hladilne tekočine. Različne oblike hladilnega sistema je mogoče doseči s spreminjanjem toka skozi termostatske ventile, s spreminjanjem toplotnih obremenitev ali s priklapljanjem in odklapljanjem različnih hladilnih naprav. Zaradi različnih možnih izvedb pomeni jedro simulacijskega modela prilagodljiv algoritem za izračun tokov in tlačnih padcev v hladilnem sistemu. Iz rezultatov ovrednotenja simulacijskega modela je mogoče ugotoviti dobro ujemanje numerično ocenjenih in izmerjenih vrednosti. Predstavljena je tudi uporaba tega modela na primeru motorja vozil.
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(Ključne besede: sistemi hladilni, motorji dizelski, modeliranje numerično, modeli simulacijski)

This paper presents a simulation model for a heat and mass transfer analysis applied to an engine-cooling system under transient conditions. The simulation model has a modular structure, composed of various cooling-system submodels that are interconnected by links. An efficient analysis is possible only when the interactions between the submodels are considered simultaneously. Because of its complexity and the variety of installed devices, a marine cooling system consisting of two or more interconnected cooling circuits is presented. The temperatures in particular parts of the cooling system are maintained through thermostatic valves by mixing or diverting streams of cooling fluid. Due to the change of flow rate through the thermostatic valves, by varying the heat load and connecting or disconnecting any cooled device, a different pipeline configuration occurs. For this reason, a self-adapting algorithm for the flow and pressure-drop calculation across the cooling system is the core of the simulation model. The validation of the presented simulation model indicates good matching of the numerical predictions with the measured values. The possible application of such a model to the cooling systems of vehicle engines is presented too.
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(Keywords: cooling systems, diesel engines, numerical modelling, simulation models)

0 UVOD

Namen hlajenja motorjev z notranjim zgorevanjem je ohranjanje ustreznega temperaturnega stanja in s tem učinkovitega delovanja motorja. Glede na vedno večje zahteve tržišča po boljši gospodarnosti in zanesljivosti motorja postaja tudi hladilni sistem motorja tehnično vedno zahtevnejši. Pri zasnovi sodobnih hladilnih sistemov imajo načrtovalci v smislu izbire različnih konstrukcijskih rešitev široke možnosti.

Pri hladilnih sistemih ladijskih motorjev je zelo pomembno, da so cevovodi , skozi katere teče morska

0 INTRODUCTION

The purpose of an engine-cooling system is to keep the engine at its most efficient operating temperature. A consideration of the economic aspects and demands for better performance are leading to increasingly complex cooling systems. Engine-cooling systems have a variety of possible arrangements and devices, which leads to a demanding design task.

In marine cooling systems, the length of the pipeline and the number of components in contact

voda, čim krajši, število krmilnih organov pa čim manjše. Sodobne hladilne naprave imajo zato zelo kratke cevi z morsko vodo in osrednji zelo učinkoviti in prostorsko prilagodljivi prenosnik toplote. Po drugi strani pa na strani hlajenja s sladko vodo običajno naletimo na vsaj dva ali več krožnih tokov, po katerih se pretaka voda z različnimi temperaturami. Temperature različnih krožnih tokov ohranjamo nespremenjene z mešanjem hladnejše ali toplejše vode z uporabo termostatskih ventilov. Obsežnost in konstrukcijske možnosti izbire sestavnih delov, npr. različne črpalke in drugi gradniki obsežnega hladilnega sistema večjih ladijskih motorjev pomenijo za inženirja – oblikovalca velik izziv. Čeudi je hladilni sistem ustrezno zasnovan in preverjen v razmerah ustaljenega delovanja, pa je njegovo delovanje v prehodnih, spremenljivih pogojih delovanja popolna neznanka. K temu znatno prispeva spremenljivo delovanje termostatskega ventila, ki v prehodnem režimu delovanja močno spreminja delovne parametre celotnega hladilnega sistema. Podoben učinek povzročata tudi vklop ali izklop katerekoli komponente hladilnega sistema.

Osnovo pri načrtovanju hladilnega sistema pomeni ustrezna določitev značilnosti cevovodov in tlačne črpalke, ki mora zagotavljati ustrezen tok vode pri določeni tlačni višini. Če se povrnemo na osnovne zahteve omenjene v začetku, je treba na podlagi potrebnih tokovnih zahtev: masnih tokov, uporov itn. in vgradbenih – izmernih zahtev najprej določiti geometrijske izmere cevovodov in drugih sestavnih delov hladilnega sistema. Izračun izmernega in delovnega prilagajanja je iterativen in ga moramo ponavljati, dokler z rezultati nismo zadovoljni. Takšen način je zelo primeren za preproste cevne odseke, ni pa primeren za zahtevne, razvejane cevne sisteme. V razpoložljivi literaturi najdemo mnogo uporabnih metod, s katerimi lahko izračunamo zamotane cevne sisteme [1]. V prispevku opisan hladilni sistem je izračunan na podlagi Hardy-Crossove metode, ki je bila izboljšana z uporabo Newton-Raphsonove tehnike.

1 OPIS NUMERIČNEGA MODELA HLADILNEGA SISTEMA

Hladilni sistem motorja sestavljajo cevi, črpalke, ventili in porabniki toplote, ki jih moramo ohladiti. Vse omenjene sestavne dele lahko zložimo skupaj na različne načine, zato mora biti numerični model zelo gibek in prilagodljiv najrazličnejšim tehničnim izvedbam. Sestavne dele hladilnega sistema obravnavajo posebni, neodvisni modeli, ki jih lahko pri različnih izvedbah hladilnega sistema poljubno izberemo in med seboj povežemo. Vsak posamezni sestavni model obdeluje probleme prenosa snovi in toplote za obravnavano komponento, kot vhodni podatek pa je namenjen masni tok tekočine. Delovni parametri hladilnega sistema se stalno spreminjajo, zato je v raziskavah vpeljan algoritem masnega toka tekočine in kombiniran s posameznimi

with seawater are kept as low as possible. A modern cooling system has a very short seawater pipeline, using only one central heat exchanger, cooled with seawater, that offers good heat management combined with high flexibility. As parts of the freshwater cooling system are not operating within the same temperature range, at least two or more interconnected cooling circuits are used. The temperature in a particular cooling circuit is maintained by mixing the flow through a thermostatic valve. The complex structure and the variety of possible arrangements of the cooling-system pipeline represent design challenge in which appropriate pumps and other equipment have to be chosen. If that task is successfully completed for steady-state operation, system behaviour in the transients often remains unknown. In addition, it should be mentioned that the operation of thermostatic valves during transients constantly causes a change of the system's working point. The same happens when any unit is connected or disconnected from the cooling system.

The design of the cooling system involves a determination of the pipeline characteristics and the selection of the pump, which is able to satisfy the required flow rate and pressure specifications. Based on a previously established pipeline layout, the design process consists of defining the section dimensions, taking into account the recommended fluid velocities and the system constraints. This iterative process is repeated until the cooling system satisfies the design requirements. It is possible to apply the described process to a simple pipeline, but its application to the design of complex systems is impracticable. Several methods for the analysis of complex pipe networks are described in [1]. The analysis of cooling system pipe network presented here is based on the Hardy-Cross approach, improved with the Newton-Raphson technique.

1 DESCRIPPTION OF THE NUMERICAL MODEL OF THE COOLING SYSTEM

The engine-cooling system consists of pipes, heat exchangers, pumps, valves and units that need to be cooled. All these components can be arranged in many different ways, so the numerical model must be flexible in adapting to any possible cooling-system configuration. The above-mentioned components are represented through separate submodels, which are interconnected according to a given cooling-system layout. The component submodels are focused on heat and mass transfer within the component itself, taking flow rates as the input data. As the changes of the cooling system's working point occur constantly, an algorithm for

modeli gradnikov hladilnega sistema, ti pa so povezani v celotni numerični model hladilnega sistema. V prispevku so na kratko opisani numerični modeli – podsklopi celotnega numeričnega modela.

1.1 Cevovodi

Numerični model, ki opisuje cevovode, temelji na zakonih ohranitve snovi in energije. Pri tem je uporabljen enorazsežen model, ker je dolžina cevi znatno večja od njenega premera. Kinetična in potencialna energija sta v primerjavi s preneseno toploto neznatni in ju zato v tem primeru zanemarimo, medtem ko so viskozne sile upoštevane pri izračunu tlačnih izgub. Predstavljeni model sestavlja sistem enačb, ki opisuje hladilno sredstvo oziroma steno cevovoda, skozi katerega se to sredstvo pretaka (sl. 1).

flow determination is combined with component submodels in a common numerical model. In the following text, numerical submodels are briefly presented.

1.1 Pipes

The numerical model of the pipes is based on equations of mass and energy conservation. As the length of the pipe is much larger than the cross-sectional diameter, a one-dimensional model is applied for the pipes. The potential and kinetic energies are neglected in comparison to the thermal energy, whereas viscous dissipation is taken into account during the calculation of the pressure drops. This model consists of an equations system related to a cooling fluid and an equations system related to a pipe wall (Fig.1).

Sl. 1. *Fizikalni model prenosa snovi in toplote v cevi*
Fig. 1. *Physical model for heat and mass transfer in a pipe*

Enačbo, ki obravnava ohranitev energije v hladilu, lahko zapišemo v obliki:

The energy conservation for a coolant can be expressed as follows:

$$\rho_f c_f A_f \frac{\partial T_f}{\partial t} + \rho_f c_f v_f A_f \frac{\partial T_f}{\partial x} = \lambda_f A_f \frac{\partial^2 T_f}{\partial x^2} + \alpha_f O_f \left(T_w - T_f \right) \tag{1}$$

medtem ko je ustrezni zapis zakona o ohranitvi snovi:

while mass conservation is:

$$\frac{\partial}{\partial x} \left(\rho_f v_f \right) = 0 \tag{2}$$

Ravnotežna energijska enačba za cevno steno se glasi:

The energy conservation for a pipe wall can be written as:

$$\rho_w A_w c_w \frac{\partial T_w}{\partial t} = \lambda_w A_w \frac{\partial^2 T_w}{\partial x^2} + \alpha_0 O_0 \left(T_0 - T_w \right) - \alpha_f O_f \left(T_w - T_f \right) \tag{3}$$

V enačbah (1) do (3) označujejo indeksi: f - hladilo, w – steno cevi in 0 – okolico. Spremenljivke so: gostota tekočine ρ, specifična toplota c, ploščina prečnega prereza cevi A, temperatura T, kinematična viskoznost v, toplotna prevodnost λ, toplotna prestopnost α in površina opazovanega mejnega elementa O.

In Eq.(1-3) f is the index for the fluid, w for the pipe wall and 0 for the environment. The variables used in Eq.(1-3) are: ρ for the density, c for specific heat, A for the cross-sectional area, T for the temperature, v for the fluid velocity, λ for the conduction coefficient, α for the heat-transfer coefficient and O for the perimeter of the control volume boundary.

1.2 Prenosnik toplote

Pri sodobnih hladilnih sistemih motorjev z notranjim zgorevanjem najpogosteje srečamo ploščate

1.2 Heat exchanger

A plate heat exchanger (PHE) is the most frequently used heat exchanger in modern engine-

prenosnike toplote. Odlikuje jih zelo dober učinek prenosa toplote, poleg tega pa so v primerjavi s cevnimi prenosniki manj občutljivi na zamašitev z nečistočami. Zelo lahko jih očistimo, po zasnovi so zelo zgoščeni in prilagodljivi na spremembe delovanja. Žal pa je njihova uporaba zaradi uporabljenega gradiva tesnil temperaturno in glede na kemično sestavo hladiva omejena. Delovni tlak hladiva je omejen na 15 bar, tlačne izgube – upori pa so sorazmerno veliki. V motorjih z notranjim zgorevanjem so tlaki hladiva navadno nižji od 15 bar, pa tudi sestava hladiva ni takšna, da bi ogrozila zanesljivost delovanja hladilnika; ploščati hladilniki so zato zelo primerna oblika hladilnikov za motorje (sl. 2).

cooling systems. PHEs have high heat-transfer rates and produce less fouling than shell-and-tube exchangers. They are easy to clean, require little space and they are very flexible if the process conditions are changed. However, the choice of fluids is limited by the chemical resistance and the temperature limits of the gaskets. The working pressures of PHEs are limited to 15 bar, and they produce a relatively high pressure drop. In normal conditions, the working pressures in engine cooling systems do not exceed 15 bar and the cooling fluids are not aggressive, so there are practically no limits for the applications of PHEs (Fig.2).

legenda:	legend:
1 – nosilna prečka	1 – carrying bar
2 – cevna povezava	2 – connection
3 – trdno ogrodje	3 – fixed frame
4 – sidrni vijaki	4 – tightening bolts
5 – sklepna plošča	5 – end plate
6 – plošča prenosnika	6 – channel plate
7 – sklepna plošča	7 – end plate
8 – pritisna plošča	8 – pressure plate
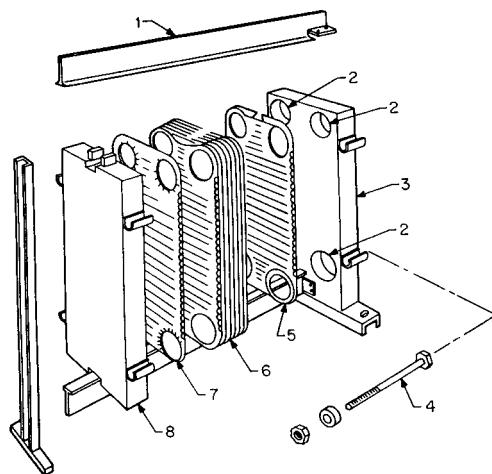
Sl. 2. *Prikaz sestavnih delov ploščatega prenosnika toplote*
Fig. 2. *Exploded view of a plate heat exchanger*

Zaradi sorazmerno zamotane oblike sestavnih delov in posebnosti v različnih razmerah delovanja ploščatih prenosnikov toplote, je bil razvit ustrezen prilagojen numerični model. V hladilniku hladivo teče skozi zaporedne ploščate prehode; numerični model obravnava le en prehod skozi ploščati kanal. Pri tem predpostavimo, da se tokovi hladiva v preostalih kanalih ne spremenijo in da se prenos toplote izvede le v dejavnih ploščah prenosnika (sl. 2, plošča 6).

Zaradi zgoraj omenjenih predpostavk lahko kanal ploščatega prenosnika (slika 3) obravnavamo kot model cevi:

Due to the complexity of the geometry and the specific behaviour of the PHE in different operating conditions, a more detailed numerical model had to be developed. The fluids within a PHE flow in alternate spaces between the plates. The numerical model of a PHE can be reduced to a single channel between the plates, with the assumption that flows of the same fluid are equal in all channels. Also, it is assumed that the heat exchange is limited only to the plates.

With these assumptions, the physical model of a PHE channel (Fig.3) is similar to a pipe model:

$$\rho_f c_f A_f \frac{\partial T_f}{\partial t} + \rho_f c_f v_f A_f \frac{\partial T_f}{\partial x} = \lambda_f A_f \frac{\partial^2 T_f}{\partial x^2} + \alpha_f A_{pl} \left(T_{pl} - T_f \right) \tag{4},$$

kjer se indeks *f* nanaša na hladivo, indeks *pl* pa na dejavno ploščo prenosnika

Zaradi specifične oblike in sestave plošč prenosnika ima masni tok hladiva močan vpliv na učinek ploščatega hladilnika. Zato je v numeričnem modelu upoštevan vpliv masnega toka hladiva na toplotno prestopnost in tlačne izgube in omogoča izračun obeh. Različne oblike površin prenosnikov toplote seveda določajo različne toplotne prestopnosti, zato, žal splošno veljavne odvisnosti med obliko prenosnih površin in prestopom toplote

where index *f* represents the cooling fluid and the index *pl* is the heat-exchanger plate.

Due to the specific geometry of the plates, the flow rate has a strong influence on the performance of a PHE. For this reason, the numerical model of the heat exchanger takes into account the flow rate, which is the basis for the heat-transfer coefficient and the pressure-drop calculation. Every type of plate has its own heat-transfer coefficient. There is no universal correlation for the calculation of the heat-transfer coefficient. An extensive

Plošča *n*-2
Plate *n*-2

Plošča *n*-1
Plate *n*-1

Plošča *n*
Plate *n*

Plošča *n*+1
Plate *n*+1

$T_{f1} + dT_{f1}$

Q_{n-1}

Q_n

T_{f1}

w_{pl}

v_{f2}

b

v_{f1}

v_{f2}
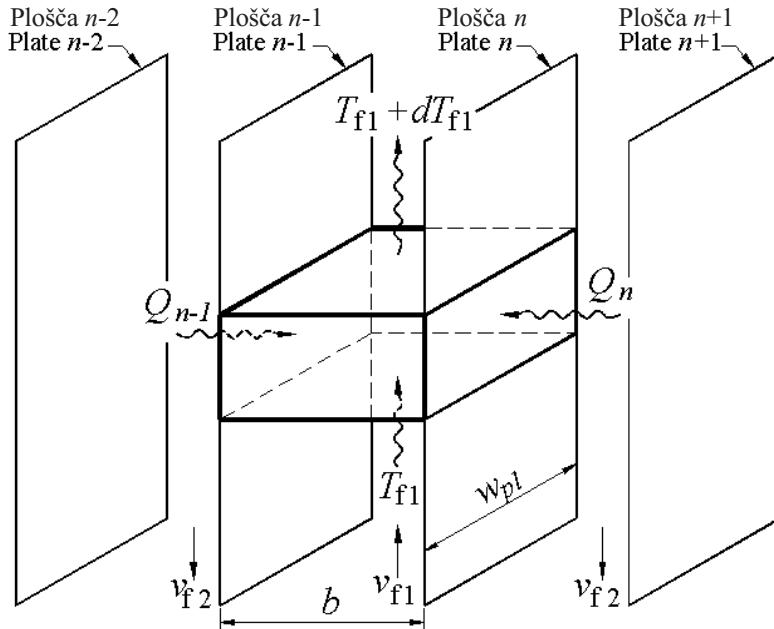
Sl. 3. *Fizikalni model kanala v ploščatem prenosniku toplote*
Fig. 3. *Physical model of a PHE channel*

ne moremo določiti. Izčrpno razpravo o izrazih, ki popisujejo toplotno prestopnost, lahko najdemo v delu avtorjev Raja in Bensala [2]. V tem prispevku je uporabljen izračun toplotne prestopnosti v skladu z virom [3], medtem ko je koeficient tlačnih izgub izračunan z izrazom:

discussion of the equations for the heat-transfer coefficient is given in a paper by Raju and Bansal [2]. In this paper, a procedure for the heat-transfer coefficient calculation presented in [3] is used; the friction coefficient is calculated from the following equation:

$$\frac{1}{\sqrt{\xi}} = \frac{\cos\varphi}{\sqrt{b\tan\varphi + c\sin\varphi + \xi_0\left(\mathrm{Re}\right)/\cos\varphi}} + \frac{1-\cos\varphi}{\sqrt{a\xi_0\left(\mathrm{Re}\right)}} \tag{5},$$

kjer pomenijo φ kot tokovnice delca toka na ploščo prenosnika, a, b in c pa so eksperimentalno določene stalnice z vrednostmi:

where φ represents the angle of the chevron pattern on the plates, while a, b and c are experimentally determined constant whose values are:

$$a = 1,6 \; ; \; b = 0,40; \; c = 0,36 \tag{6}.$$

Koeficient trenja ξ_0(Re) se nanaša na kot tokovnice delca toka na ploščo prenosnika pri kotu φ = 0 in ga določimo z uporabo znanih izrazov:

The friction factor ξ_0(Re) refers to a chevron pattern with angle φ = 0, and is obtained from:

$$\xi_0 = 53,39/\mathrm{Re} \; \text{ za/for } \; \mathrm{Re} < 2000 \; \text{(laminarni tok / laminar flow)} \tag{7}$$

$$\xi_0 = \left(1,8 \; \lg \; \mathrm{Re} \; -1,5\right)^2 \; \text{ za/for } \; \mathrm{Re} \geq 2000 \; \text{(turtbulentni tok / turbulent flow)} \tag{8}.$$

Toplotno prestopnost vsebuje Nusseltovo število, ki ga izračunamo z izrazom:

The heat-transfer coefficient is expressed with the Nusselt number calculated from the following equation:

$$\mathrm{Nu} = 0,122\,\mathrm{Pr}^{1/3}\left(\frac{\eta}{\eta_{st}}\right)^{1/6}\left[\xi\,\mathrm{Re}^2\sin\left(2\varphi\right)\right]^{0.374} \tag{9}.$$

Diagrami na slikah 4 in 5 prikazujejo spremembo koeficienta trenja, vrednosti ξ, Nusseltovega števila in njegove odvisnosti od Reynoldsovega števila za različne vrednosti kota φ tokovnice delca toka na ploščo prenosnika. Veliki koti φ povzročajo večje tlačne izgube in zagotavljajo boljši prenos toplote.

The diagrams in Fig.4 and Fig.5 show the change of friction factor, ξ, and Nusselt number, Nu, depending on the Reynolds number and the angle of the chevron pattern, φ. It is obvious that a larger angle φ produces a higher pressure drop and a better heat-transfer coefficient.
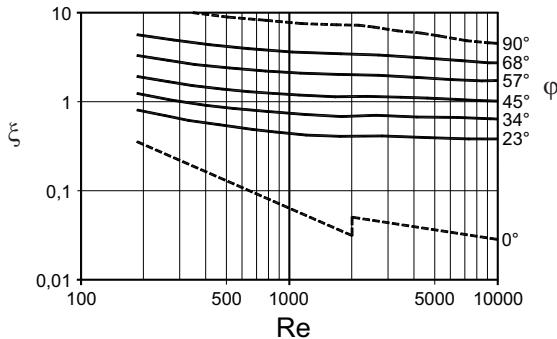
Sl. 4. *Koeficient tlačnih izgub v odvisnosti od Re števila in kota φ*
Fig. 4. *Friction factor as a function of Re and the chevron angle φ*

Sl. 5. *Odvisnost vrednosti Nu števila od Re števila in kota φ*
Fig. 5. *Nu number as a function of Re and the chevron angle φ*

1.3 Termostatski ventil

Termostatski ventil vzdržuje temperaturo tekočine v ozkem območju želene temperature. PI krmilnik uravnava – v odvisnosti od razlike med želeno in izmerjeno temperaturo – lego termostatskega ventila tako, da je nihanje temperatur okoli pričakovane vrednosti čim manjše. Vsako temperaturno zaznavalo, ki meri lastno temperaturo na različnih mestih toplotnega prenosnika, ima zaradi lastne mase določeno zakasnitev, ki jo opredeljuje časovna stalnica t_k. V predstavljenem numeričnem modelu termostatskega ventila je upoštevana časovna zakasnitev spremembe temperature v časovnem intervalu Δt z naslednjo eksponencialno odvisnostjo [4]:

1.3 Thermostatic valve

A thermostatic valve maintains the desired temperature within narrow limits for a particular cooling circuit. Depending on the difference between the measured and the desired temperature, a PI controller positions the thermostatic valve trying to eliminate temperature deviation. The temperature at which a particular point in the cooling system is measured by a temperature sensor that has time constant t_k. The numerical model of the thermostatic valve includes a transient response to temperature changes through time interval Δt according to the following exponential function [4]:

$$T - T_0 = \left(T_f - T_0\right)\left(1 - e^{-\Delta t/t_k}\right) \tag{9}$$

V enačbi (9) se indeks *0* nanaša na začetno stanje, indeks *f* pa na zadnje doseženo stanje.

In Eq.(9), the index *0* represents the state before the change, whereas index *f* represents the last achieved state.

1.4 Določitev masnih tokov in tlačnih izgub

Rezultati izračunov vseh predloženih numeričnih modelov temeljijo na znanih masnih tokovih tekočin. Značilnici cevovoda in črpalke določata določeno skupno delovno točko. Izračun masnega toka tekočine temelji na upoštevanju zakonov o ohranitvi mase in energije. Kontinuitetna enačba mora biti izpolnjena v vsaki stični točki cevovoda, in sicer:

1.4 Flow rates and pressure drops

All the presented numerical models assume known flow rates through the cooling system. As mentioned before, the pipe characteristics associated with the selected pump will meet a certain working point. The flow-rate calculation is based on mass and heat conservation equations, too. The continuity equation must be satisfied at each junction, which can be written as:

$$\sum_{k=1}^{n} \lambda_{j,k} Q_k = \sum_{k=1}^{n} \lambda_{j,k} A_k v_k = 0 \tag{10}$$

kjer koeficient λ nakazuje povezavo med *k*-to cevjo in *j*-tim spojem. Energijska enačba za ustaljen tok nestisljive tekočine v *k*-ti cevi lahko zapišemo v obliki:

where λ is an indicator representing the existence of the connection between pipe *k* and junction *j*. The energy equation for steady, incompressible fluid flow in pipe *k* can be written as:

$$\left(\Delta E\right)_k = \left(\Delta p\right)_k + \rho g \left(\Delta z\right)_k + \frac{1}{2}\rho\left(\Delta\left(v^2\right)\right)_k + \left(\Delta p_{fr}\right)_k - \left(\Delta p_P\right)_k = 0 \tag{11}$$

kjer pomenijo: p vrednost statičnega tlaka, z višinsko razliko (lego) cevnega spoja, v hitrost tekočine, Δp_{fr} celotni padec tlaka, Δp_{P} pa tlačilki dovedeno energijo [5].

Celotni padec tlaka Δp_{fr} upošteva dolžinske in lokalne izgube tlaka v cevovodu. Dolžinske izgube so izračunane z uporabo znanih obrazcev za laminarni in turbulentni tok. Energija, ki je potrebna za pogon črpalke Δp_{P}, je izračunana na podlagi podatkov meritev, ki jih je podal izdelovalec črpalke. Podana je v obliki polinoma druge stopnje v odvisnosti od toka tekočine oziroma hitrosti tekočine:

where p is the static pressure, z is the height level of the junction, v is the flow velocity, Δp_{fr} is the total pressure drop and Δp_{P} is the energy added by the pump [5].

Total pressure drop Δp_{fr} consists of head loss for straight pipes and of local pressure drops. The head loss for straight pipes is calculated according to known relations for laminar or turbulent flow. The pressure energy, Δp_{P}, produced by the pump is calculated from characteristics obtained from the pump producer. The pump's characteristic curve is written in the form of a second-order polynomial as a function of flow rate or velocity:

$$p_P = f(Q) = a_0 + a_1 Q + a_2 Q^2 \qquad \text{ali/or} \qquad p_P = f(v) = a_0 + a_1 v + a_2 v^2 \qquad (12).$$

Nelinearni sistem enačb je rešljiv z Newton-Raphsonovo metodo, ki je zelo stabilna, hitro konvergira, rešitev pa ni odvisna od izbranih začetnih vrednosti ([1] in [6]).

V tem prispevku so glavni pogonski motor in pomožni motorji večjega postroja prikazani kot točkovni viri toplote. Ocenimo, da je dovod hladilne toplote motorja bodisi stalen, ali pa časovno odvisna funkcija. Podobno so obravnavani tudi drugi viri toplote: hladilnik polnilnega zraka, destilacijska naprava, kompresorji, generator električnega toka , hidravlični motorji, glavni ležaji itn. Hladilnik mazalnega olja je modeliran podobno kakor osrednji hladilnik motorske (sladke) hladilne vode, pri čemer so bili predpostavljeni podatki: stalni masni tok olja, vstopna temperatura mazalnega olja v hladilnik z izračunano gostoto in viskoznostjo.

The obtained nonlinear system of equations is solved by the Newton-Raphson method. This method is very stable, it has a fast convergence and the solution is insensitive to the starting values ([1] and [6]).

In this study, the main engine and the auxiliary engines in a power plant are presented as nodal heat sources. The heat inflow can be assumed to be a constant value or as time-dependent function. The same method is used for the other units, like the scavenge air cooler, the distillation unit, the compressors, the A/C plant, the hydraulic plant, the line shaft bearing, etc. The lube-oil cooler is modelled in the same way as the fresh-water central cooler, with a constant lube-oil flow rate, given lube oil inlet temperature and corresponding viscosity and density values.

2 RAČUNSKI PRIMER SIMULACIJE IN VREDNOTENJE REZULTATOV

2 APPLICATION EXAMPLE AND RESULTS VALIDATION

Numerični model je bil uporabljen za določitev sedanjega ladijskega hladilnega sistema (sl. 6). Hladilni sistem sestoji iz dveh krožnih hladilnih tokov: visokotemperaturnega (oznaka VT), ki hladi glavni pogonski motor in nizkotemperaturnega (oznaka NT), ki hladi pomožne stroje in naprave. Pri hladilnem krogu VT vzdržujemo temperaturo vode pri 85 °C, medtem ko je temperatura vode na izstopu iz osrednjega hladilnika za sladko vodo omejena na 36 °C. Predpostavljeno je, da je vstopna temperatura hladilne morske vode pri delovanju v tropih omejena na 32 °C. Hladilni krožni tok VT ne teče skozi prenosnik toplote; želeno temperaturo vzdržujemo z neposrednim mešanjem s hladnejšo vodo iz hladilnega kroga NT. Izmerjena temperatura hladilne vode na izstopu iz motorja krmili termostatski ventil in določa delež primešane hladilne vode iz hladilnega kroga NT.

Podatki za izračun so dobljeni z meritvami na dejanskem ladijskem hladilnem sistemu. Rezultati izračunov se dobro ujemajo z izmerjenimi podatki; pri

The numerical model was applied to the prediction of an existing marine cooling system (Fig.6). This cooling system consists of a high-temperature (HT) circuit for the main engine cooling and a low-temperature (LT) circuit for auxiliary engines and other equipment. The high-temperature cooling circuit is maintained at 85°C on the main engine outlet, while the low-temperature cooling circuit should be 36°C on the fresh- water central cooler outlet. It is assumed that seawater cooling of the central cooler has an inlet temperature of 32°C (in tropical conditions). The high-temperature circuit has no heat exchanger or cooler. Its temperature is maintained by mixing with cooling fluid from the low-temperature circuit. According to the temperature measured at the engine outlet, a thermostatic valve determines the portion of fluid that has to be taken from the low-temperature circuit.

All the data for the cooling system are taken from an existing ship-propulsion plant. The results of the numerical simulation showed good matching with the measured data obtained during a sea trial. The measured values are the flow rate and pressure
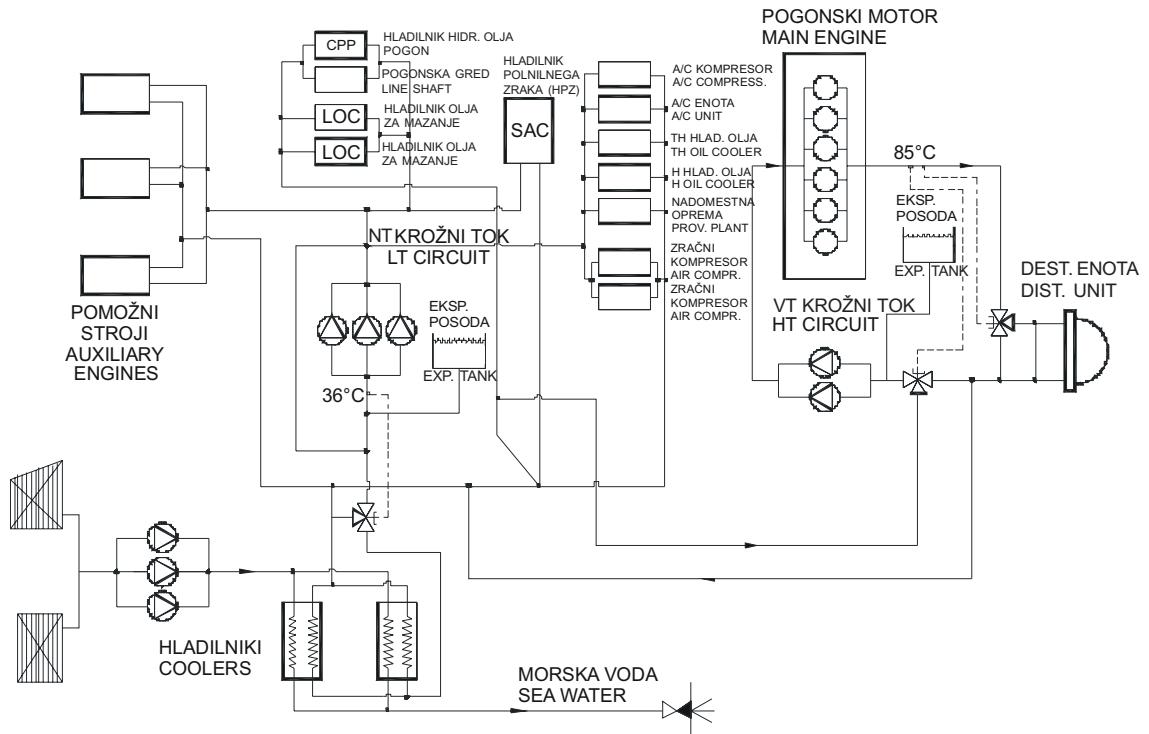
Sl. 6. *Shema hladilnega sistema pogonskega sistema na ladji*
Fig. 6. *Example of the layout of a marine cooling system*

tem mislimo predvsem na vrednosti masnih tokov in ustreznih tlakov v nadzornih točkah hladilnega sistema. Na sliki 7 so prikazani rezultati primerjav med izračunanimi in izmerjenimi rezultati preizkusov na ladji. Pri preizkušanjih so bili podatki zajeti šele, ko je bilo doseženo ustaljeno temperaturno stanje pri določeni izbrani moči pogonskega motorja oziroma celotnega postroja.

Med delovanjem motorja je bilo opazovano tudi nihanje masnih tokov vode v ceveh, ki povezujejo oba hladilna kroga. Posebej težavna naloga je bila določitev časovnega poteka lege povezovalnih členov med obema hladilnima krogoma, še posebej zaradi različnih tlačnih ravni obeh krogov. Zato je zelo pomembno, da je tlačna razlika na obeh koncih povezovalnega voda ustrezna, saj tako zagotavlja ustrezni tok hladilne vode in s tem tudi ustrezno količino odvedene toplote. Rezultati numeričnega modeliranja so pokazali, da je lega cevne povezave med obema hladilnima krožnima tokovoma pravilno izbrana, da pa je za izrazita prenihanja kriva predvsem nastavitev krmilnika, ki upravlja termostatski ventil. Opravljeni so bili še različni primeri izračunov za različne nastavitve krmilnika termostatskega ventila, rezultati pa so prikazani na sliki 8. Izbira - nastavitev temperatur v obeh hladilnih krogih je bila enaka tisti, ki smo jo navedli v prvem primeru. Začetek simulacije upošteva hladni zagon pogonskega motorja in temperaturo "vroče vode" (VT) 65 °C. Zgornji diagram prikazuje premočni vpliv sorazmernega in prešibek vpliv integracijskega člena krmilnika termostatskega ventila. Spodnji in tretji diagram kažeta vpliv popravka krmilnih parametrov in doseganje želenega stanja v hladilnem krogu VT.

in the characteristic points of the cooling system. Fig.7 shows a comparison between the numerical model results and the measured data during one sea trial. During the sea trial the values are measured after the temperatures and other parameters have reached steady values for a given power output of the main engine, or in other words, for steady-state conditions of the propulsion plant.

A fluctuation of the flow rates in the pipes that connect two cooling circuits was observed during the measurement. It is not an easy task to determine the right positions for the connections between two cooling circuits, especially when they work at different pressure levels. It is essential that the pressure difference between two points of the connecting pipe must be sufficient to provide adequate fluid exchange between the two cooling circuits. In this case, the numerical model showed that the connecting pipes are well positioned and that the cause of the fluctuations was in the controller adjustments, which is responsible for the thermostatic valve behaviour. Further analysis with different controller parameters was performed and the obtained results are shown in Fig.8. The temperatures of the cooling circuits are set as described earlier. The simulation started with a cold engine and the high-temperature (HT) coolant at 65°C. The first diagram shows a too strong proportional and a weak integral component of the HT thermostatic valve controller. The second and third diagrams show a correction of the controller parameters and achieving the desired behaviour of the HT cooling circuit.
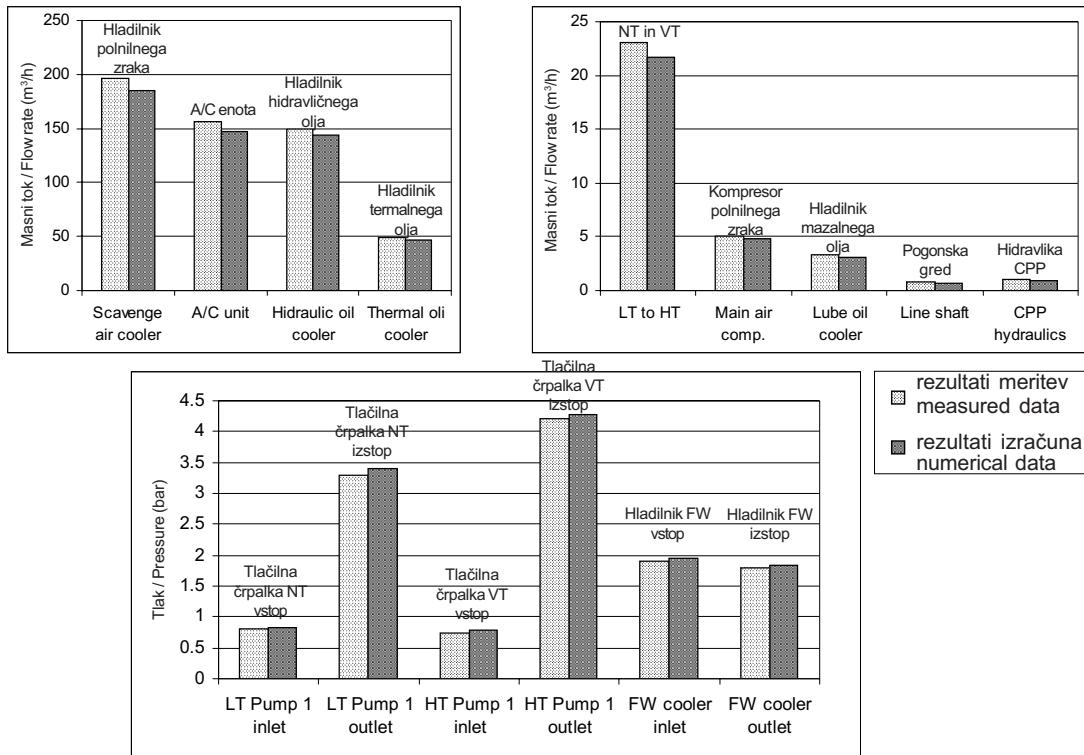
Sl. 7. *Primerjava med izračunanimi in izmerjenimi rezultati*
Fig. 7. *Comparison of numerical and measured data*
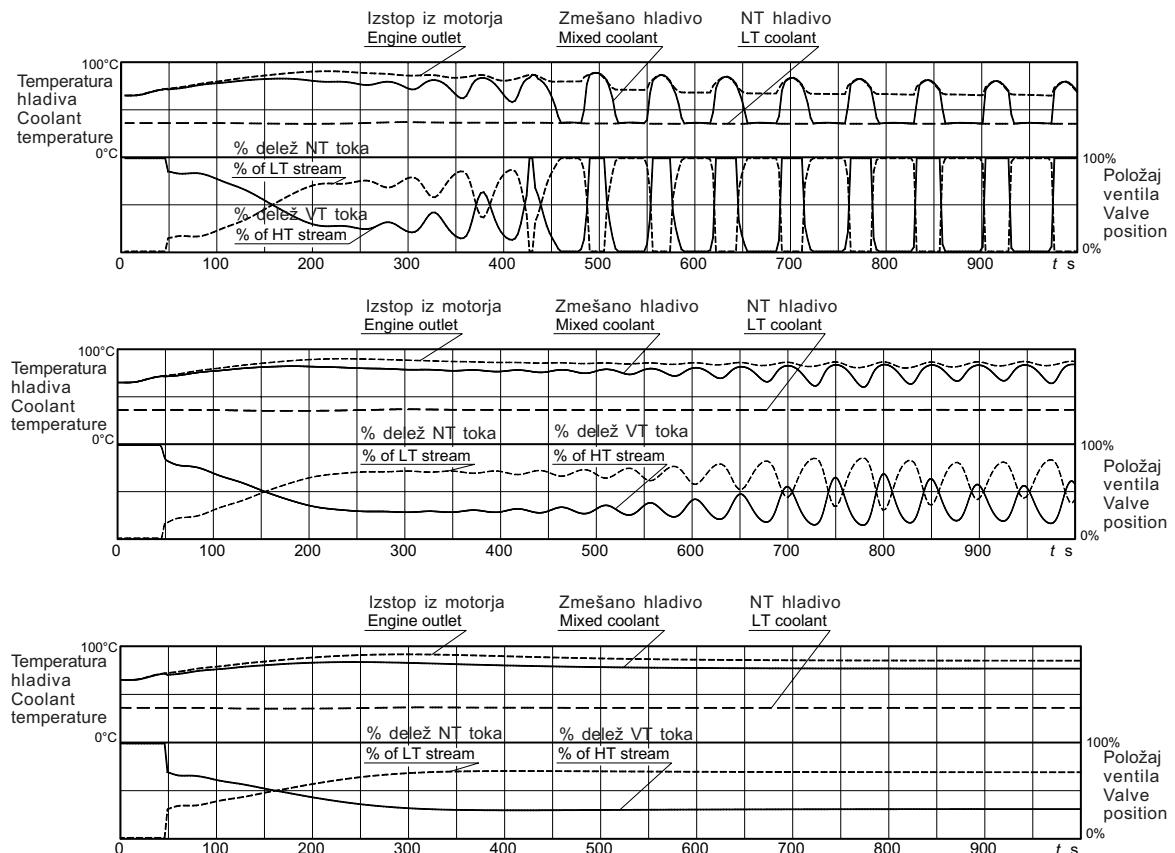
Sl. 8. *Vpliv krmilnih parametrov na odzivnost hladilnega sistema*
Fig. 8. *Influence of controller parameters on cooling-system behaviour*

3 UPORABA NUMERIČNEGA MODELA ZA DOLOČITEV HLADILNIH SISTEMOV CESTNIH VOZIL

Prilagodljivost predstavljenega numeričnega modela omogoča uspešno uporabo tudi pri določanju hladilnih sistemov cestnih vozil. Seveda je treba uvesti določene značilnosti in razviti ustrezne dodatne sestavne modele, npr.:
- model prenosnika toplote zrak - voda
- model obtočne, mehansko gnane črpalke za vodo
- model termostata

Cevi, ki povezujejo osnovne dele hladilnega sistema vozil, so opazno krajše, tokovi hladilne tekočine so majhni, zato je odzivnost takšnega sistema na spremembe sorazmerno hitra. Sistem ima tudi manjšo toplotno vztrajnost, zato je treba določiti bolj podroben sistem odvoda toplote iz motorja. Način določanja masnih tokov, tlačnih izgub v cevovodih ostaja nespremenjen, dodatno pa je treba upoštevati še dejstvo, da sta motor in hladilnik glavna vira toplotnih izgub.

Na podlagi zgoraj opisanih dejstev lahko povzamemo, da lahko opisani numerični model uspešno uporabimo pri določitvi ladijskih in drugih hladilnih sistemih za hlajenje motorjev, le ustrezne dodatne modele je treba dodati osnovnemu modelu.

4 SKLEPI

Brez predhodne uporabe rezultatov numeričnega modeliranja v začetni fazi načrtovanja hladilnega sistema motorja bi za določitev sestavnih delov in njihovega skupnega delovanja porabili zelo veliko časa. Zaradi varnosti se v približnih rešitvah pogosto uporabi prevelike agregate, ki sicer zagotavljajo varno delovanje, vendar se kasneje v fazi prilagajanja dejanskim razmeram s postopki dušenja in dodatnih obvodov vnašajo nepotrebne velike izgube energije.

Prednost predstavljene metode je v natančni določitvi masnih tokov in tlačnih padcev v vsaki točki hladilnega sistema za ustaljene in prehodne načine delovanja. Zelo preprosto je tudi mogoče ugotoviti vpliv posameznih komponent in njihovih parametrov na delovanje celotnega hladilnega sistema. To je zelo pomembno v primerih, ko je treba določiti delovanje hladilnega sistema pri hitrem izklapljanju obtočne črpalke, hladilnika, ali celo pri ustavitvi motorja.

Z uporabo predstavljenega numeričnega modela lahko spremljamo časovno spremembo hladilnih parametrov tudi v prehodnih režimih delovanja. V prispevku obravnavani računski primer je pokazal, da je mogoče s pomočjo izračunov izbrati in optimirati celo krmilne parametre termostatskega ventila.

Prikazani model je skratka vsestransko uporaben za snovanje in preverjanje različnih hladilnih sitemov, ki so povezani z motorji z notranjim zgorevanjem.

3 APPLICATION OF THE DEVELOPED NUMERICAL MODEL TO THE COOLING SYSTEMS OF VEHICLE ENGINES

The flexibility of the presented numerical model allows it to be applied to the cooling systems of vehicle engines. However, there are some differences that require minor changes to the presented model and require the creation of additional numerical submodels for the following components:
- air-to-fluid heat exchangers,
- engine-driven centrifugal pump,
- thermostat.

The pipes between the vehicle's cooling-system components are short and there is less cooling fluid than in a marine cooling system, so the response of such a system is relatively fast. Due to the smaller thermal inertia of the whole cooling system, a more detailed engine-heat release model should be used. The method and the principle of the flow rate and pressure drop calculations in the pipeline remain the same, but in this case the engine and the radiator generate the majority of the heat losses.

From these facts it can be concluded that the developed numerical model can be applied to marine, vehicle or any other engine-cooling system, including adequate numerical submodels of the system components.

4 CONCLUSION

If the initial design of a cooling system is performed without the help of an appropriate numerical model and computer, a lot of time will be spent for extensive calculations due to iterative nature of this problem. As a result, an approximate solution is obtained and the designer usually chooses over-dimensioned pumps to the satisfy design requirements with a safety margin. In this case, the required flow rates and pressures are obtained with some damping device or bypass giving high energy losses.

The advantage of the presented numerical model is the precise determination of flow rates and pressure drops at any desired point of the cooling system during steady-state and transient operation. This is very helpful in primary designs. Also, it is easy to investigate how the change of any design parameter influences the behaviour of the whole cooling system. This is especially important for simulation of emergency situations when pump, heat exchanger or even the main engine must be shut down.

Transients that occur during any change of stationary working point are easily monitored. It was shown in the paper that such a numerical model can even be used for setting the thermostatic valve controller.

The presented numerical model can also be used for different optimisations of the cooling system.

5 LITERATURA
5 REFERENCES

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