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Simulation model of a ship diesel engine supercharging air cooler

SUMMARY

In the paper a simulation model of heat exchange in supercharging air cooler of ship diesel engine, is presented. The model makes it possible to calculate instantaneous values of heat flow from air to cooling medium, temperature distribution within the cooler as well as temperatures of media at outlet from the cooler.

Being a part of a general model of energy transformation in the diesel engine it can function in real time and within the entire range of engine load.

The model was verified by comparing the calculation results with the measured data from tests of a real cooler and by assessing correctness of time courses of selected parameters of the cooler.

INTRODUCTION

In the paper a part of the simulation model of energy transformation process in the ship main diesel engine of medium speed, is presented [2]. The fragment contains both general physical and mathematical models of the heat exchange process which occurs in a supercharging air cooler of ship diesel engine. Investigations of the model were performed for CPE 23/14/1 air cooler cooperating with 6ZA40S SULZER diesel engine.The occurring heat exchange process was simulated in real time mode.

NOMINAL MODEL

The supercharging air coolers are designed as cubicoid vesels with crossing flow of media (Fig.1). Sea water flows through pipes as cooling medium. Air, cooled medium, flows between lamellas (ribs) fixed paralelly to air flow direction. The lamellas made of thin copper plates are usually installed on each 2 or 3 pipes in regular spacing.

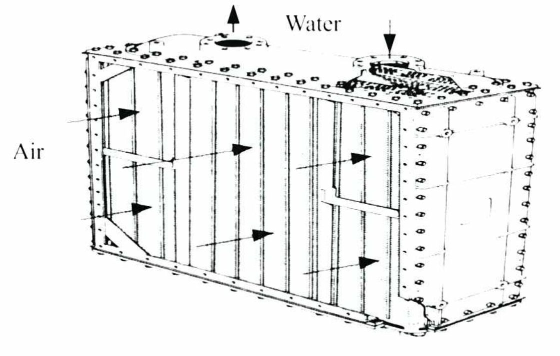

Fig. 1. A sketch of the considered supercharging air cooler

The following initial assumptions were adopted for determining the model :

* air flow through the cooler is guided by lamellas, and its rate and pressure is constant within the entire cooler volume
* velocity of water flow through pipes is uniform and constant over the entire length of the pipes
* no heat is transmitted through the pipe walls in the direction of water flow as well as between air layers and through air in the direction of air flow
* lamellas of uniform thickness and uniform physical properties are of the same thermal conductance value in the entire cooler
* heat amount exchanged with environment is negligible.

MODELLING METHOD

The finite element method (FEM) was applied to determine dynamic characteristics of the air cooler in question [1].The used finite element (FE) is a recurrent fragment of the pipe fitted with the lamella (Fig.2b), contained in the rectangular space of (Δx , Δy , Δz) dimensions (Fig.2a). Only a single two-dimensional area which contains one row of pipes in (ox , oy) plane, is considered due to identity of flow conditions in the pipe rows in (oz , oy) plane.

Values of the air temperature $T_a(x,\tau)$ and water temperature $T_w(y,\tau)$ in the $FE_{(i\ ,\ j)}$ are determined on the basis of values of the temperatures at inlet to the FE. The air temperature at outlet from the cooler is taken as the mean of the air temperature values at outlet from the FEs of the last pipe. And, the water temperature at outlet from the cooler is taken as the mean of the water temperature values at outlet from the last FEs of all pipes in a row.

A simplified transmittance of the FE is determined to be next used for determination of the dynamic characteristics of the entire cooler model.

The applied way of verification of the modelling

Statical characteristics of the cooler obtained on the basis of calculations of the model in question was compared with measurement data of a real cooler. However dynamic characteristics of the cooler was verified only qualitatively as no appropriate measurement data of a real object were at the author's disposal.

To the applied FE the following assumptions were adopted :

- the air flowing around the pipes between lamellas is perfectly mixed so its physical parameters (temperature, density, gas constant) are the same in any point of its volume
- water temperature in any pipe is constant in ox axis direction
- the lamellas are considered as disc ribs distant to each other halfway between neighbouring pipes
- heat flow from the lamellas is transferred through the butting contact area between the lamella and pipe wall, in the radial direction respective to the pipes
- the heat flow from lamella to pipe wall is inertialess due to good thermal conductance of the joint.

a) *b)*

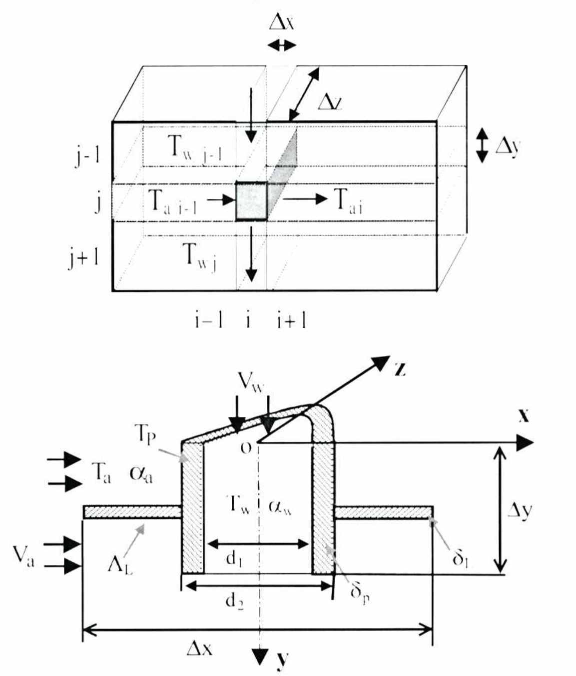

***Fig.2.** **a)** Scheme of division of the cooler into FEs of (Δx , Δy , Δz) dimensions*
***b)** The considered part of the pipe with lamella (disc rib)*

Notation :

T_a , T_w , T_p – air, water and pipe-wall temperature , respectively
V_a , V_w – air, water flow velocity, respectively
α_a , α_w – air, water surface film conductance, respectively
δ_p , δ_L – pipe, lamella wall thickness, respectively
A_l – lamella external surface area
d_1 , d_2 – internal and external diameter of pipe, respectively
i , j – successive index number of FEs

The applied model of heat flow through the finite element

Heat flow through the finite element was assumed to be consisted of :

- ✱ transfer of heat between cooled air and the pipe walls and lamellas
- ✱ heat transmission between the lamellas and pipe walls, and
- ✱ heat transfer between the pipe walls (their internal surface) and cooling water.

The surface film conductance coefficient from the side of air, α_a, was determined with the use of the following empirical relationship [4] :

$$\alpha_a = 0.33 \frac{\lambda_a^{0.67} c_a^{0.33} \rho_a^{0.33}}{d_2^{0.4} \upsilon_a^{0.27}} V_a^{0.6} \tag{1}$$

and, the analogous coefficient of conductance from the side of water, α_w, from the following formula [4] :

$$\alpha_w = 0.023 \left(\frac{V_w d_1}{\upsilon_w} \right)^{0.8} \left(\frac{c_w \rho_w \upsilon_w}{\lambda_w} \right)^{0.4} \frac{\lambda_w}{d_1} \tag{2}$$

Heat exchange between the lamella and pipe wall was determined on the basis of a model of constant thickness disc rib. As analytical solution of the problem is complicated and time consuming, a simplification was practically applied to the calculations by making use of diagrams for determination of efficiency of a constant thickness disc rib [3].

Therefore the following simplified formula was applied which expressed the steady-state heat flow rate through the contact surface of the lamella and pipe wall :

$$\dot{Q}_L = \eta_L A_L \alpha_a \left(T_a - T_p \right) \tag{3}$$

where :

A_L – area of the lamella external surface being in contact with air
η_L – lamella (rib) efficiency (defined as the ratio of the heat absorbed by the rib at real distribution of temperature and that absorbed at uniform temperature distribution along the rib's height).

Hence, heat balance equations for the FE in question obtained the following form :

for cooled air :

$$\begin{aligned} c_a \rho_a F_a \Delta x \frac{\partial T_a}{\partial \tau} + c_a \rho_a F_a V_a \Delta x \frac{\partial T_a}{\partial x} = \\ = \left(A_{p2} + \eta_L A_L \right) \alpha_a \left(T_p - T_a \right) \end{aligned} \tag{4}$$

for pipe wall :

$$\begin{aligned} c_p m_p \frac{\partial T_p}{\partial \tau} = \\ = \left(A_{p2} + \eta_L A_L \right) \alpha_a \left(T_a - T_p \right) + A_{p1} \alpha_w \left(T_w - T_p \right) \end{aligned} \tag{5}$$

for cooling water :

$$\begin{aligned} c_w \rho_w F_w \Delta y \frac{\partial T_w}{\partial \tau} + c_w \rho_w F_w V_w \Delta y \frac{\partial T_w}{\partial y} = \\ = A_{p1} \alpha_w \left(T_p - T_w \right) \end{aligned} \tag{6}$$

and the gas state law :

$$\rho_a = \frac{p_a}{R T_a} \tag{7}$$

where :

A_{p1} – pipe internal surface area [m^2]
A_{p2} – pipe external surface area [m^2]
F_a – air flow face area [m^2]
F_w – water flow face area [m^2]
η_L – lamella efficienty [–]

Operational equation of air temperature at outlet from the control volume (at x = Δx) was determined in the following form :

$$T_a(\Delta x, s) = \frac{\beta(s)}{\alpha(s)}\left[1 - e^{\frac{-\alpha(s)}{V_a}\Delta x}\right] T_w(s) + e^{\frac{-\alpha(s)}{V_a}\Delta x} T_a(o,s) \tag{8}$$

where :

$$\alpha(s) = \left(s + a_2 - \frac{a_2 a_3}{s + a_3 + a_4}\right)$$

$$\beta(s) = \frac{a_2 a_4}{s + a_3 + a_4}$$

$$a_2 = \frac{\alpha_a\left(A_{p2} + \eta_L A_L\right)}{\rho_a c_a F_a \Delta x}$$

$$a_3 = \frac{\alpha_a\left(A_{p2} + \eta_L A_L\right)}{c_p m_p}$$

$$a_4 = \frac{A_{p1}\alpha_w}{c_p m_p}$$

$$a_5 = \frac{\alpha_w A_{p1}}{\rho_w c_w F_w \Delta y}$$

The first component of (8) is a 1st order inertial element :

$$H_1(s) \approx \frac{\beta(s)}{V_a}\Delta x = \frac{\Delta x}{V_a} \cdot \frac{a_2 a_4}{s + a_3 + a_4}$$

and the second component - a delay element :

$$H_2(s) = e^{-\frac{\Delta x}{V_a}s} e^{-\frac{\Delta x}{V_a}a_2} e^{\frac{\Delta x}{V_a} \cdot \frac{a_2 a_3}{s + a_3 + a_4}}$$

Step response of the cooler's FE was determined in compliance with (8) as follows :

$$T_a(\Delta x, \tau) = h_1(\tau) T_{wst} + h_2(\tau) T_{ast} \tag{9}$$

where :

T_{wst}, T_{ast} – temperature step change values of water and air respectively, at inlet to the FE.

and :

$$h_1(\tau) = \frac{\Delta x}{V_a} \cdot \frac{a_2 a_4}{a_3 + a_4}[1 - e^{-(a_3+a_4)\tau}]\mathbf{1}(\tau)$$

$$h_2(\tau) = e^{-\frac{\Delta x}{V_a}a_2} e^{-(a_3+a_4)\left(\tau - \frac{\Delta x}{V_a}\right)} \sum_{k=0}^{\infty}(a_3 + a_4)^k \cdot \left(\frac{\Delta x}{V_a}a_3 a_2\right)^{-\frac{k}{2}}\left(\tau - \frac{\Delta x}{V_a}\right)^{\frac{k}{2}} \cdot \mathbf{J}_k\left(2\sqrt{\frac{\Delta x}{V_a}a_3 a_2\left(\tau - \frac{\Delta x}{V_a}\right)}\right)\mathbf{1}\left(\tau - \frac{\Delta x}{V_a}\right)$$

where : $\mathbf{J}_k$ – Bessel function of 1st kind and n-th order.

Analogically, from equations (4)-(6) the following operational transmittance of cooling water temperature at outlet from the FE, was obtained :

$$T_w(\Delta y, s) = H_3(s) T_a(s) + H_4(s) T_w(o,s) \tag{10}$$

where :

$$H_3(s) \approx \frac{\Delta y}{V_w} \cdot \frac{a_3 a_5}{s + a_3 + a_4}$$

$$H_4(s) = e^{-\frac{\Delta y}{V_w}s} e^{-\frac{\Delta y}{V_w}a_5} e^{\frac{\Delta y}{V_w} \cdot \frac{a_5 a_4}{s + a_3 + a_4}}$$

Response to step excitations from water and air temperatures at the FE's inlet obtained the following form :

$$T_w(\Delta x, \tau) = h_3(\tau) T_{ast} + h_4(\tau) T_{wst} \tag{11}$$

where :

$$h_3(\tau) = \frac{\Delta y}{V_w} \cdot \frac{a_5 a_4}{a_3 + a_4}\left[1 - e^{-(a_3+a_4)\tau}\right]\mathbf{1}(\tau)$$

$$h_4(\tau) = e^{-\frac{\Delta y}{V_w}a_5} e^{-(a_3+a_4)\left(\tau - \frac{\Delta y}{V_w}\right)} \sum_{k=0}^{\infty}(a_3 + a_4)^k \cdot \left(\frac{\Delta y}{V_w}a_3 a_5\right)^{-\frac{k}{2}}\left(\tau - \frac{\Delta y}{V_w}\right)^{\frac{k}{2}} \cdot \mathbf{J}_k\left(2\sqrt{\frac{\Delta y}{V_w}a_3 a_5\left(\tau - \frac{\Delta y}{V_w}\right)}\right)\mathbf{1}\left(\tau - \frac{\Delta y}{V_w}\right)$$

Steady-state values of the functions $h_1(\tau)$, $h_2(\tau)$, $h_3(\tau)$ and $h_4(\tau)$ were determined from the transmittances $H_1(s)$, $H_2(s)$, $H_3(s)$ and $H_4(s)$ by making use of the theorem on limiting values.

Time courses of air and water temperatures at the FE's outlet on response to the step excitation from air and water temperatures at the FE's inlet are presented in Fig.3 and 4 respectively.

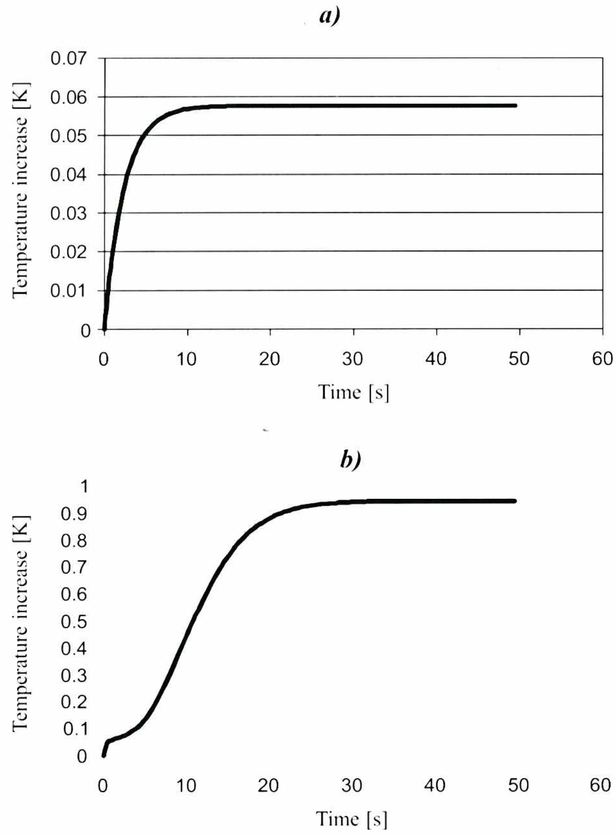

Fig.3. *Response of air temperature at outlet from the FE to the step excitation of 1 K value (at V_w = 0.2 m/s, V_a = 7.67 m/s) of :* ***a)*** *air temperature at inlet* ***b)*** *water temperature at inlet*

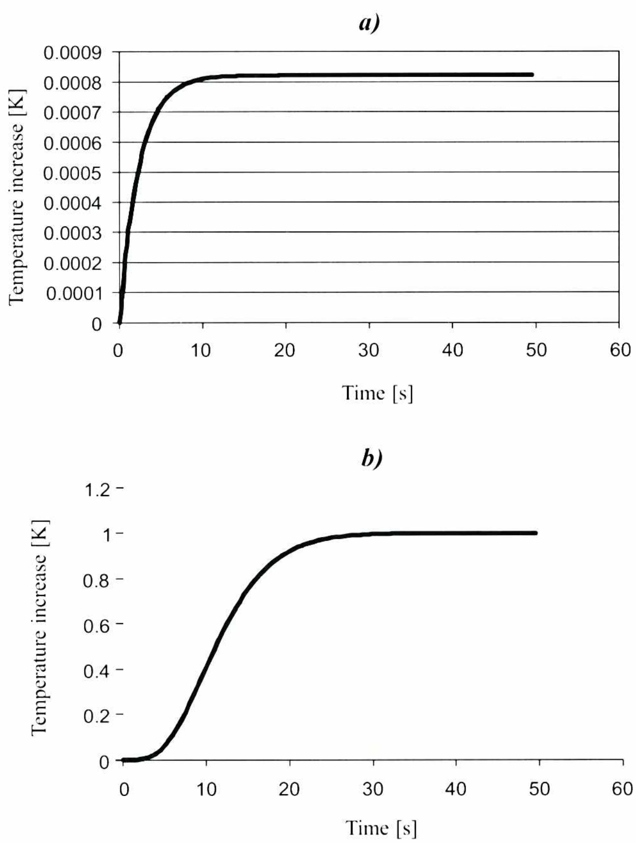

Fig.4. *Response of water temperature at outlet from the FE to the step excitation of 1 K value (at V_w = 0.2 m/s, V_a = 7.67 m/s) of :* ***a)*** *air temperature at inlet* ***b)*** *water temperature at inlet*

The FEs within the entire cooler were assumed mutually connected as shown in Fig.5

where :

$T_p(i\,,j\text{-}1)$ and $T_w(i\text{-}1,j)$ – temperature inputs to the FE (i , j)
$T_p(i\,,j)$ and $T_w(i\,,j)$ – temperature outputs from the FE (i , j) etc.

Hence the equations (9) and (10) for the FE (i , j) were transformed to the following matrix form :

$$\begin{bmatrix} T_a(i,j) \\ T_w(i,j) \end{bmatrix} = \begin{bmatrix} H_2(i,j) & H_1(i,j) \\ H_3(i,j) & H_4(i,j) \end{bmatrix} \begin{bmatrix} T_a(i,j-1) \\ T_w(i-1,j) \end{bmatrix} \tag{12}$$

where :

$$T_w(0,j) = T_{w1} \qquad T_a(i,0) = T_{a1}$$

T_{w1} – cooling water temperature at the cooler's inlet
T_{a1} – cooled air temperature at the cooler's inlet.

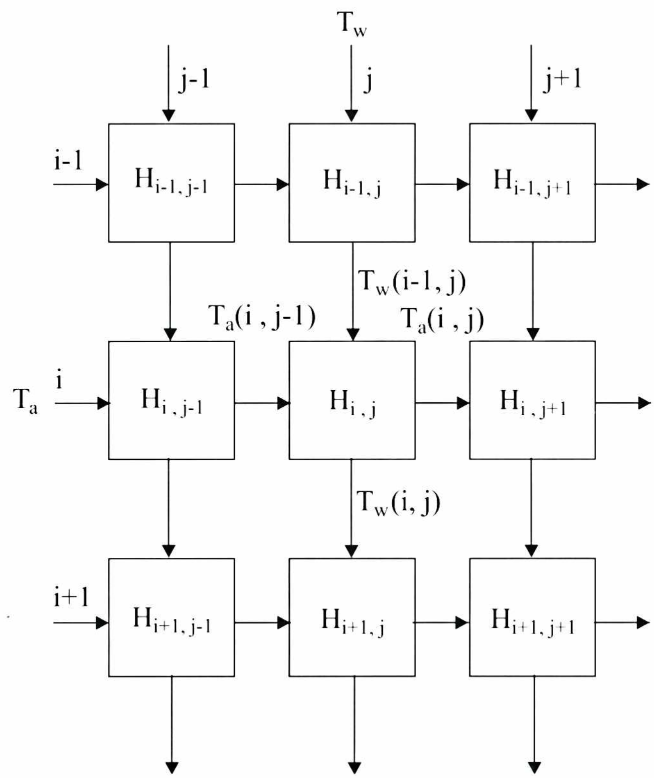

Fig.5. *Block scheme of FE connections within the considered air cooler model*

RESULTS OF INVESTIGATIONS OF THE SIMULATION MODEL

In order to exemplify the simulation model in question the CPE 23/14/I air cooler (of the technical data given in Tab.1) was selected.

Dynamic characteristics of the cooler was obtained by means of numerical calculations of its FE model, performed for assumed initial conditions with the use of a computer program written in *Borland Pascal* language and initiated by means of *Delphi 3* software. The FEs of the dimensions : Δx = 37.4 mm, Δy = 37.4 mm, Δz = 2.35 mm were applied.

The calculation results are presented in the form of dynamic time courses of the air temperature T_{a2} at outlet from the cooler in response to step change of the the air temperature T_{a1} and water temperature T_{w1} at inlet to the cooler.

Tab.1.
Technical data of the real air cooler

Technical data	CPE 23/14/I a one-circuit device
Gabarites	1150 mm x 587 mm x 574 mm
Pipe material	MNŻ-101 Cu-Ni alloy
Number of pipes / rows	322 / 14
Pipe diameter	20 mm
Pipe wall thickness	1.2 mm
Pipe spacing	15 mm
Lamella material	copper
Lammella thickness	0.25 mm
Lamella spacing	2.1 mm
Number of lamellas	501
Heat exchange surface area	338.2 m^2
Sea water flow rate	82 m^3/h
Air flow rate	7.67 m^3/h
Sea water temperature at inlet/outlet	37.4 / 35.7 °C
Air temperature at inlet/outlet	159 / 42 °C

Tab.2. Comparison of the calculated results
and measured data for the CPE 23/14/I supercharging air cooler

Measured/calculated results		Measurement number				
		11	10	9	3	8
Air temperature before cooler	°C	96.8	132.3	150.7	159.2	176.7
Air temperature behind cooler	°C	38.5	40	45	42.5	45.5
Air mass flow rate	kg/s	4.87	6.41	7.22	7.67	8.45
Water temperature before cooler	°C	30.7	27.9	31.4	27.4	27.1
Water temperature behind cooler	°C	33.6	33.2	38	35.7	37
Water mas flow rate	kg/s	23.09	23.11	22.94	22.69	23.11
Calculated air temperature behind cooler	°C	39.5	40.5	45.2	42.5	45.8
Relative calculation error	%	2.6	1.3	0.4	0.0	0.7
Calculated water temperature behind cooler	°C	34.6	34.1	39.2	35.8	36.4
Relative calculation error	%	3.0	2.7	3.2	0.3	-1.6

The statical characteristics (shown in Fig.6) was calculated by means of the same model for the steady-state conditions. The conditions were assumed the same as those given in Tab. 2 for the measurement 3 (at 100% engine load).

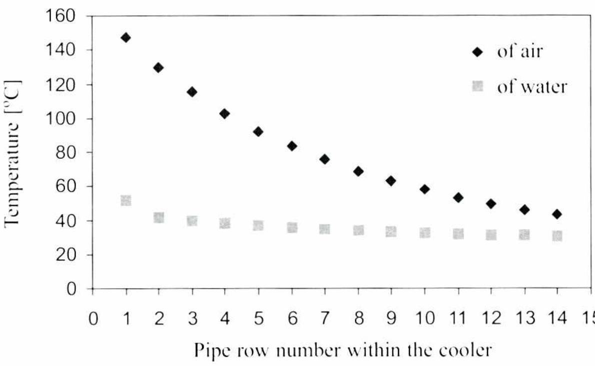

Fig.6. Calculated distribution of steady-state air and water temperatures along the air flow within the cooler

In Fig.7 time courses are presented of air and cooling water temperatures at outlet from the cooler in response to the step change of air temperature by 30 K (from 159 to 189 °C). For the flow rates : $\dot{m}_w$ = = 22.69 kg/s and $\dot{m}_a$ = 7.67 kg/s (Fig.7a) a time lag and air temperature change at outlet (of about 4 K) can be observed. A new steady-state temperature value (of 46 °C) is obtained. Change of cooling water temperature at outlet is small (abt. 0.1 K) hence hardly visible in the figure.

In Fig.7b and 7c time courses of air and cooling water temperatures at outlet from the cooler are presented for some other flow rate conditions of the media.

For the air flow rate decreased by 50% and the unchanged water flow velocity (Fig.7b), the time lag in reaching the outlet air temperature is greater by 7s and the temperature increase is smaller by 2.5 K. It means that the inertia time constant value is the same in both cases, and the time lag value is dependent on the medium flow velocity.

For the water flow rate decreased by 50 % and the unchanged air flow velocity, the time lag of in reaching the outlet air temperature is greater by 5s and the temperature increase greater by 2.5 K.

The simulation model of the cooler in question in steady-state conditions was verified by comparing the results of calculations with those measured on a real cooler by ZUT ZGODA Works [5]. In Tab.2. the calculated results and measured data for the CPE 23/14/I air cooler are compared.

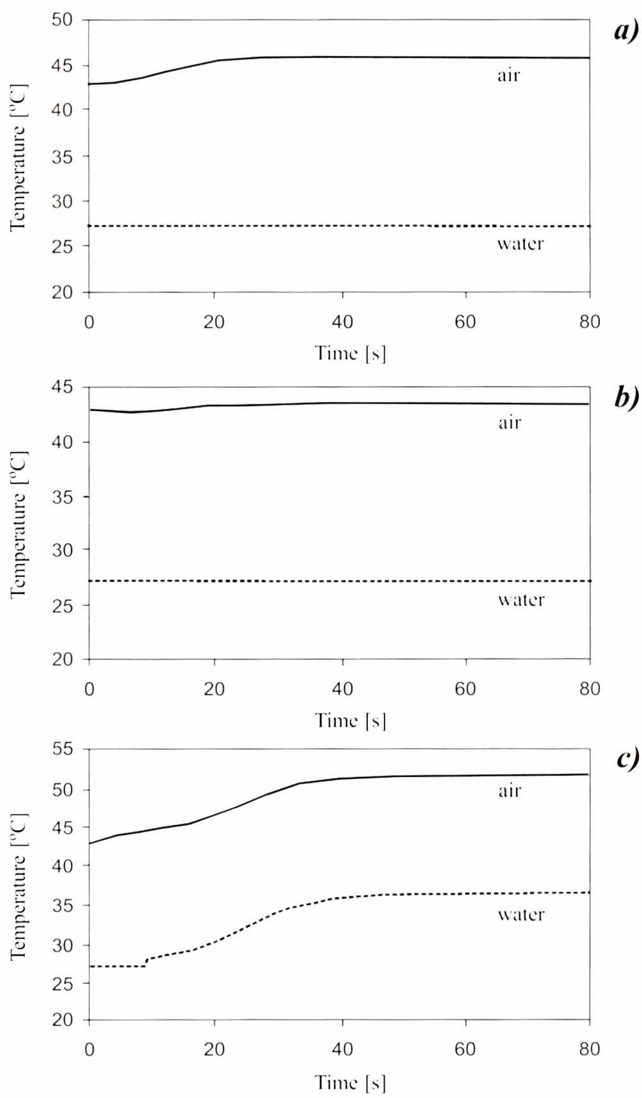

Fig.7. Time courses of air and cooling water temperatures at cooler's outlet in response to step changes of inlet air temperaturee from 159 to 189 °C :
***a)** at flow rates : $\dot{m}_a$ = 7.67 kg/s, $\dot{m}_w$ = 22.69 kg/s*
***b)** at flow rates : $\dot{m}_a$ = 3.84 kg/s, $\dot{m}_w$ = 22.69 kg/s*
***c)** at flow rates : $\dot{m}_a$ = 7.67 kg/s, $\dot{m}_w$ = 11.34 kg/s*

CONCLUSIONS

On the basis of the obtained results
the following conclusions can be offered :

- Comparison of the results of the calculation of the steady - state parameters of the simulation model, performed for different external conditions, with respective measured data of the cooler (Tab.2) showed that the relative error of the calculation was not greater than 4%; this is an acceptable result.
- The calculation results of the dynamic behaviour of the cooler revealed high variability of air temperature at outlet from the cooler, which resulted from the great difference between heat capacity of air and that of water. It means that it would be possible to easily control the steady-state temperature of air at outlet from the cooler by means of a small change of the water temperature at inlet to the cooler. It can be realized with the use of a sea water temperature automatic control unit equipped with PI controller.
- Dynamic properties of the supercharging air cooler of the ship main diesel engine were correctly – however only qualitatively – – represented by the simulation model in question.
- It demonstrated that thermal behaviour of air was more dynamical than that of water, which resulted from large flow velocity and small heat capacity of air. It also showed that some time lag of heat transport of the media ocurred in response to temperature input, as well as an inertia effect in heat exchange between the cooler elements.
- It was not possible to verify quantitatively the obtained calculation results due to lack of relevant data from dynamic tests of the cooler. It can be only presumed that real time-lag and time constants of the dynamic processes occurring in the cooler could be greater than those calculated due to the assumed non-inertial heat flow from the lamellas to pipe walls.

Appraised by Alfred Brandowski, Prof.,D.Sc.

NOMENCLATURE

a - constant
A - area [m^2]
c - specific heat capacity [J/kgK]
d_1 - internal diameter of pipe [m]
d_2 - external diameter of pipe [m]
F - flow face area [m^2]
H - operational transmittance
m - mass [kg]
$\dot{m}$ - mass flow rate [kg/s]
p - pressure [Pa]
$\dot{Q}$ - heat flow rate [m^3/s]
V - flow velocity [m/s]
R - universal gas constant [$J \cdot K^{-1} \cdot mol^{-1}$]
s - complex variable
T - temperature [K]

α - surface film conductance [W/mK]
δ - wall thickness [m]
Δ - increment, difference of a quantity
η_l - lamella efficiency [-]
λ - thermal conductance coefficient [W/mK]
υ - kinematic viscosity coefficient [m^2/s]
ρ - density [kg/m^3]
τ - time [s]

Indices

a - of air
L - of lamella
p - of pipe wall
st - step
w - of sea water

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Conference

KONES 2002

The International Scientific Conference on Combustion Engines, organized by the Institute of Aeronautics, Warsaw, and the Mechanical Faculty, Gdańsk University of Technology, was held already 28 times. The last one took place in Jurata on Hel Peninsula on 8-11 September 2002. The multi-year cycle of scientific meetings is devoted to the achievements in research and development of :

- compression-ignition and spark-ignition engines with special attention to ecology
- combustion processes
- mixture preparation
- exhaust aftertreatment
- heat exchange
- inspection and control
- diagnostics and maintenance
- durability and reliability
- new materials
- engine parts
- fuels and lubricants
- participation of Poland in the world production of engines.

Over 150 representatives of scientific and industrial circles participated in the conference in Jurata. They had an opportunity to listen to and discuss 112 papers presented during one plenary, 5 panel and 2 poster sessions. Their authors were guests from Belgium, Czech Republic, Japan, Canada, Lithuania, Germany, Slovenia and USA, and from Poland it was the representatives of 16 civil and 3 military technical universities as well as of 8 other scientific research and development centres. The greatest contribution to the Conference's program was brought by the following universities : of Poznań (with 16 papers), of Warszawa (with 11 papers), of Częstochowa (with 11 papers) and of Radom (with 10 papers). 33 papers were published in the conference's proceedings, and the remaining – in the Journal of KONES, vol. 9, No 1 to 4, edited by Permanent Committee and the conference organizers.

FOREIGN
conference

PORTS & MARINAS 2002

On 18-20 September 2002, Wessex Institute of Technology, UK, organized Third International Conference, held at Rhodes (Greece), devoted to management, operation, design and building of ports and marinas and other maritime works. Among authors of over 40 papers also three representatives of Maritime University of Szczecin participated in the conference, who presented two themes :

- *Methods of comparative plotting of the ship's position* by A. Stateczny
- *Implementation of marine regulations in decision support systems for vessel traffic services* – by Z. Pietrzykowski, M. Narkiewicz.