ANDRZEJ MIELEWCZYK, D.Sc.,M.E.
Gdynia Maritime Academy
Department of Basic Engineering Sciences
Gdynia

A simulation model of plate cooler

SUMMARY

Models of the controled objects are necessary for digital simulation of the industrial automatic control systems. The paper deals with three computational models of the plate heat exchanger, applicable to digital simulation at different accuracy levels, i.e. analytical, simplified and difference model.

Calculation results of the analytical model were compared with the producer's data of Alfa–Laval plate coolers.

INTRODUCTION

This is the second paper devoted to simulation of control processes of ship power plant systems. Basic assumptions and principles of the simulation were presented in [9]. The plate cooler is an object to which, due to its dynamic behaviour, special modelling method should be applied.

Its steady-state heat exchange process can be described by means of simple mathematical relationships [2 , 13]. However, dynamic features of the process depend on many data and require solving differential equations for any considered case.

Recognizing the features is necessary to form a model of ship diesel engine auxiliary systems. The object of modelling are static and dynamic heat exchange phenomena occurring within a plate heat exchanger applicable to ship power plant systems. Principles of the mathematical modelling of heat exchangers, equation solving methods and their general analysis can be found in [4 , 5 , 6 , 7]. However, in each case which describes that exchange it is necessary to know the occurring physical phenomenon and design features of the exchanger, as well as to adopt simplifying assumptions and a method of solving the model.

A typical heat exchanger is the pipe-within-pipe exchanger whose heat exchange process was described in [10 , 13].

In the case of *Alfa–Laval* plate coolers of wide application on ships, data dealing with their steady-state parameters are provided by the producer himself who uses his own original software to this aim. It was not possible to obtain any information on dynamic characteristics of the coolers from that firm.

Therefore an attempt was undertaken on one's own to model the dynamic phenomena in question with taking into account control of the heat exchange process. It was assumed that the model should be solved by means of the methods which allow for using a computer in real time mode.

PLATE COOLER MODEL

The plate cooler consists of parallel plates. Heat exchange within it occurs over the entire plate area. The plate surface is deliberately folded to better utilize its heat exchange area and to trigger turbulent flow of liquid. A scheme of the plate cooler is shown in Fig.1.

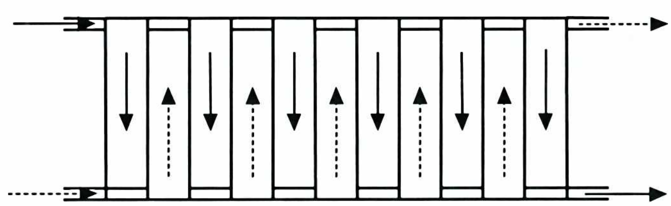

Fig.1. Scheme of plate cooler cross-section
——► fresh water flow ------► sea water flow

The cooler is so designed as to obtain the same flow parameters and heat exchange conditions in every inter-plate space because all of them are directly supplied from the inlet collector. After flowing through the inter-plate space the liquid is collected in the outlet collector.

Therefore in order to form a mathematical model of the plate cooler it is sufficient to consider the system of a two-plate exchanger with co-current or counter-current flow, shown in Fig.2.

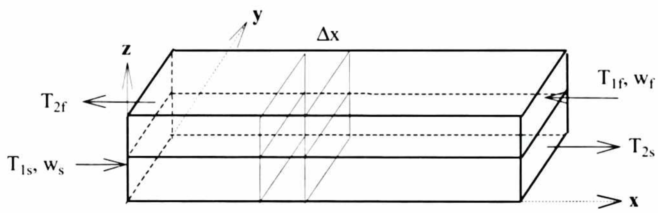

Fig.2. Scheme of plate cooler segment of counter-current flow

The following assumptions were adopted to simplify the analytical description of heat exchange in the plate cooler :

- physical parameters of liquid are temperature-independent within the range of temperature changes occurring in the cooler
- the average flow velocity profile is uniform (turbulent flow)
- no temperature gradient occurs in y-direction
- heat exchange coefficient in steady and dynamic states is constant and of the same value
- heat capacity of the plate partition is neglected due to its thinness (g = 0.4 mm , Ti-alloy)

Taking into account the above stated assumptions one can formulate energy balance of the cooled liquid segment of Δx in length [10] :

$$c_f \rho_f \Delta X\, A\left[(T_f)_{\tau+\Delta\tau} - (T_f)_\tau\right] = c_f \rho_f w_f \Delta\tau\, A\left[(T_f)_x - (T_f)_{x+\Delta x}\right] + \\ -2kF\Delta\tau(T_f - T_s) \qquad (1)$$

After several transformations and substitution of $A=ZY$ and $F=\Delta XY$ the following equation is obtained :

$$\frac{\partial T_f}{\partial \tau} + w_f \frac{\partial T_f}{\partial x} = -a(T_f - T_s) \qquad (2)$$

where :

$$T_f = f(x,\tau) \qquad T_s = f(x,\tau) \qquad a = \frac{2k}{Zc_f\rho_f}$$

By using the similar procedure the following equation for the cooling liquid is obtained :

$$\frac{\partial T_s}{\partial \tau} + w_s \frac{\partial T_s}{\partial x} = b(T_f - T_s) \qquad (3)$$

where : $$b = \frac{2k}{Zc_s\rho_s}$$

The sign at the right-hand side of the equation (2) is opposite to that of (3) as the cooling liquid absorbs heat. The coefficients a and b account for the inter-plate distance Z, coefficient k and the liquid physical constants. They are always positive, and in the case of the water-water cooler equal to each other : $a = b$.

Dynamic processes occurring in the plate cooler are described by the set of differential equations (2) and (3). In order to solve it the Laplace transformation was applied and in result the following expressions were obtained :

$$\begin{cases} T_f(x,s)\left(1+\dfrac{s}{a}\right) + \dfrac{w_f}{a}\dfrac{dT_f(x,s)}{dx} - \dfrac{1}{a}T_f(x,0) = T_s(x,s) \\ T_s(x,s)\left(1+\dfrac{s}{a}\right) + \dfrac{w_s}{b}\dfrac{dT_s(x,s)}{dx} - \dfrac{1}{b}T_s(x,0) = T_f(x,s) \end{cases} \qquad (4)$$

And, a constant non-zero value of temperature along the plate was assumed, at the time instant $\tau = 0$:

$$\begin{aligned} T_f(x,0) &= T_{fx}(x) = \text{const} \\ T_s(x,0) &= T_{sx}(x) = \text{const} \end{aligned} \qquad (5)$$

The general solution of (4) for the zero initial conditions are the following functions :

$$\begin{cases} T_f(x,s) = Ae^{r_1 x} + Be^{r_2 x} & (6) \\ T_s(x,s) = \left(1+\dfrac{s}{a}+\dfrac{w_f}{a}r_1\right)Ae^{r_1 x} + \left(1+\dfrac{s}{a}+\dfrac{w_f}{a}r_2\right)Be^{r_2 x} \end{cases}$$

where :

$$r_1 = \lambda\left(s+\alpha-\sqrt{(s+\alpha)^2+\beta^2}\right) - \frac{s+a}{w_f} \qquad (7)$$

$$r_2 = -\lambda\left(s+\alpha-\sqrt{(s+\alpha)^2+\beta^2}\right) - \frac{s+b}{w_s} \qquad (8)$$

and :

$$\alpha = \frac{w_f b - w_s a}{w_f - w_s} \qquad (9)$$

$$\beta = 2\sqrt{ab}\,\frac{\sqrt{w_s w_f}}{w_s - w_f} \qquad (10)$$

$$\lambda = \frac{w_s - w_f}{2w_s w_f} \qquad (11)$$

For the non-zero initial conditions (5) the general solution of (4) are the functions as follows :

$$\begin{cases} T_f(x,s) = Ae^{r_1 x} + Be^{r_2 x} + \dfrac{1}{s}T_{sf} \\ T_s(x,s) = \left(1+\dfrac{s}{a}+\dfrac{w_f}{a}r_1\right)Ae^{r_1 x} + \left(1+\dfrac{s}{a}+\dfrac{w_f}{a}r_2\right)Be^{r_2 x} + \dfrac{1}{s}T_{sf} \end{cases} \qquad (12)$$

where : $$T_{sf} = \frac{aT_{sx} + (s+b)T_{fx}}{s+a+b} \qquad (13)$$

The function T_{sf} is a particular integral of the set (4). The complete solution can be obtained if initial conditions are known. Practically, the counter-current flow is applied to the plate cooler, due to its better thermal effectiveness. A stepwise temperature change is assumed as the initial conditions at the plate cooler inlet on the side of the cooled as well as the cooling medium, Fig.2 :

$$T_f(x=L,s) = Ae^{r_1 L} + Be^{r_2 L} + \frac{1}{s}T_{sf} = \frac{1}{s}T_{0f} \qquad (14)$$

$$T_s(x=0,s) = \left(1+\frac{s}{a}+\frac{w_f}{a}r_1\right)A + \\ +\left(1+\frac{s}{a}+\frac{w_f}{a}r_2\right)B + \frac{1}{s}T_{sf} = \frac{1}{s}T_{0s} \qquad (15)$$

After determination of the integration constants A and B from the equations (14) i (15) and substitution of them into (12) the following solution was obtained :

$$T_{2f}(x=0,s)=\frac{T_{0f}-T_{sf}}{s}\times$$

$$\times\frac{2e^{-r_1L}\sqrt{(s+\alpha)^2+\beta^2}}{(s+\alpha)\left(1-e^{(r_2-r_1)L}\right)+\sqrt{(s+\alpha)^2+\beta^2}\left(1+e^{(r_2-r_1)L}\right)}+$$

$$+2a\frac{T_{0s}-T_{sf}}{s}\times\frac{w_s}{w_s-w_f}\times \tag{16}$$

$$\times\frac{1-e^{(r_2-r_1)L}}{(s+\alpha)\left(1-e^{(r_2-r_1)L}\right)+\sqrt{(s+\alpha)^2+\beta^2}\left(1+e^{(r_2-r_1)L}\right)}+\frac{1}{s}T_{sf}$$

$$T_{2s}(x=y,s)=\frac{T_{0s}-T_{sf}}{s}\times$$

$$\times\frac{2e^{r_2L}\sqrt{(s+\alpha)^2+\beta^2}}{(s+\alpha)\left(1-e^{(r_2-r_1)L}\right)+\sqrt{(s+\alpha)^2+\beta^2}\left(1+e^{(r_2-r_1)L}\right)}+$$

$$+2b\frac{T_{0f}-T_{sf}}{s}\times\frac{w_f}{w_s-w_f}\times \tag{17}$$

$$\times\frac{e^{(r_2-r_1)L}-1}{(s+\alpha)\left(1-e^{(r_2-r_1)L}\right)+\sqrt{(s+\alpha)^2+\beta^2}\left(1+e^{(r_2-r_1)L}\right)}+\frac{1}{s}T_{sf}$$

Values of the steady-state temperatures T_{2f} and T_{2s} were determined from (16) and (17) at $s\rightarrow 0$:

$$T_{2f}(0,\infty)=\left(T_{0f}-T_{sf}\right)\frac{w_f b+w_s a}{w_f b+w_s a e^{-L\left(\frac{a}{w_f}+\frac{b}{w_s}\right)}}+$$

$$+(T_{0s}-T_{sf})\frac{w_s a\left(e^{-L\left(\frac{a}{w_f}+\frac{b}{w_s}\right)}-1\right)}{w_f b+w_s a e^{-L\left(\frac{a}{w_f}+\frac{b}{w_s}\right)}}+T_{sf} \tag{18}$$

$$T_{2s}(L,\infty)=\left(T_{0s}-T_{sf}\right)\frac{w_s a+w_f b}{w_s a+w_f b e^{L\left(\frac{a}{w_f}+\frac{b}{w_s}\right)}}+$$

$$+(T_{0f}-T_{sf})\frac{w_f b\left(e^{L\left(\frac{a}{w_f}+\frac{b}{w_s}\right)}-1\right)}{w_s a+w_f b e^{L\left(\frac{a}{w_f}+\frac{b}{w_s}\right)}}+T_{sf} \tag{19}$$

where :

$$T_{sf}=\frac{aT_{sx}+bT_{fx}}{a+b} \tag{20}$$

On this basis, the course of the temperatures T_{2s} and T_{2f} was determined as the response to the stepwise input at the plate cooler's inlet on its cooled medium side. The Laplace reverse transformation of the functions T_{2s} and T_{2f} have not been available in the tables [11]. Hence the responses to the stepwise functions were determined numerically [1,3]. Calculation results for the below given data are presented in Fig.3 and 4.

Calculation input data :

cooled side	**cooling side**
fresh water	sea water
$T_{fx}(x,0) = 1$ deg	$T_{sx}(x,0) = 1$ deg
$\rho_f = 980$ kg/m³	$\rho_s = 1015$ kg/m³
$c_f = 4180$ J/(kg·deg)	$c_s = 4170$ J/(kg·deg)
Z = 0.005 m	Z = 0.005 m
$T_f(L,0) = T_{0f} = 2$ deg	$T_s(0,0) = T_{0s} = 1$deg

k=5000 W/m²deg

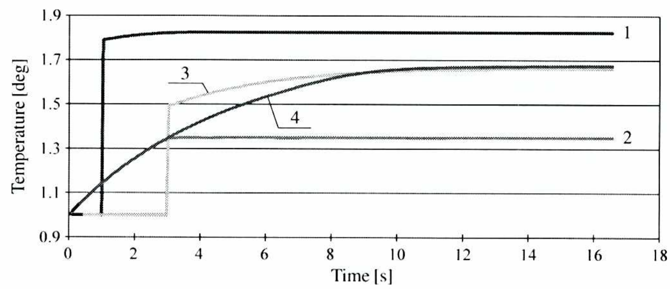

Fig.3. *Course of medium temperatures at plate cooler outlet*
w_f=2m/s, w_s=1m/s, step input T_{0f} = 2 deg
Notations : *1 – fresh water, L = 2m; 2 – sea water, L = 2m*
3 – fresh water, L = 6m; 4 – sea water, L = 6m

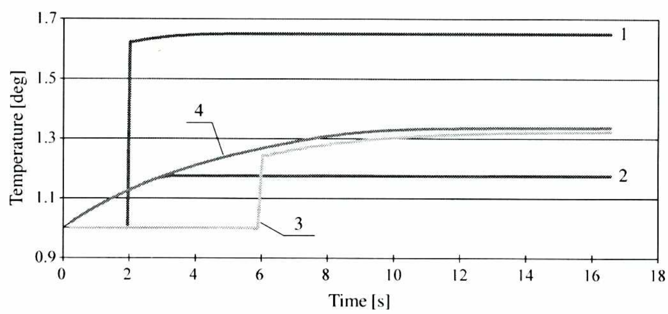

Fig.4. *Course of medium temperatures at plate cooler outlet*
w_f=1m/s, w_s=2m/s, step input T_{0f} = 2 deg
Notations : *1 – fresh water, L = 2m; 2 – sea water, L = 2m*
3 – fresh water, L = 6m; 4 – sea water, L = 6m

The fresh water temperature (curve 1 and 3) shows a transport lag connected with the time of water flow along the plate. The sea water temperature (curve 2 and 4) has no transport lag as the flow is counter-current and the jump of fresh water temperature increases the sea water temperature immediately. With longer plates the heat exchange time is longer and the cooling process more effective.

By appropriate increasing the plate length of the counter-current cooler a lower temperature of the cooled liquid is obtained than that of the cooling liquid at outlet, $T_{2f}<T_{2s}$ (curve 3 and 4).

The cooler, as an automated object, can be presented as a unit of two inputs and two outputs (Fig.5). Velocities of flow of both liquids through the cooler are the parameters of the unit.

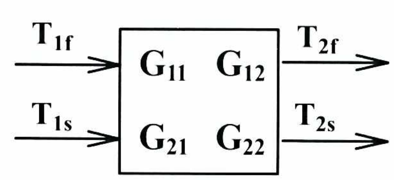

Fig.5. *Plate cooler as an automated object*
Notations :
T_{1f} - cooled liquid temperature at inlet, T_{1s} - cooling liquid temperature at inlet,
T_{2f} - cooled liquid temperature at outlet, T_{2s} - cooling liquid temperature at outlet,
G_{ij} - operator transmittance

On the basis of (16) and (17) the particular transmittances for the plate counter-current cooler yield the following forms :

$$G_{11} = \frac{2e^{-r_1 L}\sqrt{(s+\alpha)^2+\beta^2}}{(s+\alpha)\left(1-e^{(r_2-r_1)L}\right)+\sqrt{(s+\alpha)^2+\beta^2}\left(1+e^{(r_2-r_1)L}\right)} \tag{21}$$

$$G_{22} = \frac{2e^{r_2 L}\sqrt{(s+\alpha)^2+\beta^2}}{(s+\alpha)\left(1-e^{(r_2-r_1)L}\right)+\sqrt{(s+\alpha)^2+\beta^2}\left(1+e^{(r_2-r_1)L}\right)} \tag{22}$$

$$G_{12} = 2a\frac{w_s}{w_s-w_f}\times \frac{1-e^{(r_2-r_1)L}}{(s+\alpha)\left(1-e^{(r_2-r_1)L}\right)+\sqrt{(s+\alpha)^2+\beta^2}\left(1+e^{(r_2-r_1)L}\right)} \tag{23}$$

$$G_{21} = 2b\frac{w_f}{w_s-w_f}\times \frac{e^{(r_2-r_1)L}-1}{(s+\alpha)\left(1-e^{(r_2-r_1)L}\right)+\sqrt{(s+\alpha)^2+\beta^2}\left(1+e^{(r_2-r_1)L}\right)} \tag{24}$$

It is always possible to increase the heat effectiveness of the plate cooler by increasing the number of plates. Real lengths of the plates do not exceed 2 m.

The transmittances G_{11} and G_{22} can be replaced, on the basis of the course of the response curve 1 of Fig. 3 and 4, by the transmittance of the simple delaying unit, multiplied by the steady-state temperature, as follows :

$$G_{11} = T_{U11}e^{-\tau_f s} \tag{25}$$

$$G_{22} = T_{U22}e^{-\tau_s s} \tag{26}$$

where :

$$T_{U11} = \frac{w_f b + w_s a}{w_f b + w_s a e^{-L\left(\frac{a}{w_f}+\frac{b}{w_s}\right)}} \qquad \tau_f = \frac{L}{w_f}$$

$$T_{U22} = \frac{w_s a + w_f b}{w_s a + w_f b e^{L\left(\frac{a}{w_f}+\frac{b}{w_s}\right)}} \qquad \tau_s = \frac{L}{w_s}$$

Basing on the course of the response curve 2 of Fig. 3 and 4 one can replace the transmittances G_{21} and G_{12} by the transmittance of the integrating and proportioning unit, respectively as follows :

$$G_{21} = \begin{cases} \dfrac{T_{(2)1}}{s} & \text{for } \tau<\tau_1 \\ T_{(2)1}\tau_1 + \dfrac{T_{(2)2}}{s} & \text{for } \tau_1 \le \tau < \tau_2 \\ T_{U21} & \text{for } \tau \ge \tau_2 \end{cases} \tag{27}$$

$$G_{12} = \begin{cases} \dfrac{T_{(1)1}}{s} & \text{for } \tau<\tau_1 \\ T_{(1)1}\tau_1 + \dfrac{T_{(1)2}}{s} & \text{for } \tau_1 \le \tau < \tau_2 \\ T_{U12} & \text{for } \tau \ge \tau_2 \end{cases} \tag{28}$$

where :

$$\tau_1 = \frac{L}{w_f} \quad \tau_2 = \frac{L}{w_s} \quad \text{for} \quad w_f > w_s$$

$$\tau_1 = \frac{L}{w_s} \quad \tau_2 = \frac{L}{w_f} \quad \text{for} \quad w_f < w_s$$

$$T_{U21} = \frac{w_f b\left(e^{L\left(\frac{a}{w_f}+\frac{b}{w_s}\right)}-1\right)}{w_s a + w_f b e^{L\left(\frac{a}{w_f}+\frac{b}{w_s}\right)}}$$

$$T_{(2)1} = -\frac{w_f b}{w_s - w_f}$$

$$T_{(2)2} = \frac{T_{U21}-T_{(2)1}\tau_1}{\tau_2}$$

$$T_{U12} = \frac{w_s a\left(e^{-L\left(\frac{a}{w_f}+\frac{b}{w_s}\right)}-1\right)}{w_f b + w_s a e^{-L\left(\frac{a}{w_f}+\frac{b}{w_s}\right)}}$$

$$T_{(1)1} = \frac{w_s a}{w_s - w_f}$$

$$T_{(1)2} = \frac{T_{U21}-T_{(1)1}\tau_1}{\tau_2}$$

By using the two above presented analytical solutions of the mathematical model of dynamic behaviour of the plate cooler, its responses to different input functions can be determined as well as its frequency analyses performed.

The responses can be obtained for the entire range of working media parameters and of plate dimensions of the cooler. The responses obtained by means of both presented solutions are given in Fig.6.

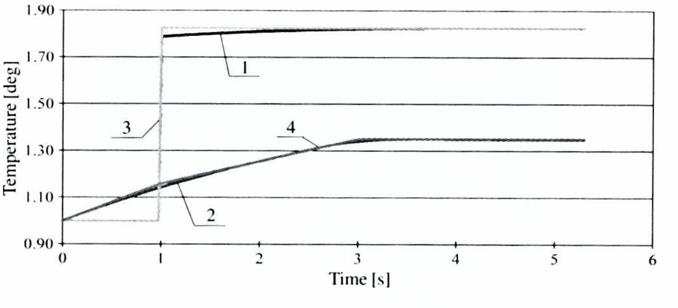

Fig.6. *Comparison of the stepwise characteristics of plate cooler according to transmittances (21) and (24) (curves 1 and 2) and transmittances (25) and (27) (curves 3 and 4) for counter-current flow, L=2m, w_f=2m/s, w_s=1m/s*

The achieved solutions not always are appropriate for digital programming. The models have to operate in real time mode, which implicates application of simple algorithms making it possible to obtain short computation times.

The first model achieved from the solution of the differential equations requires longer computations which is connected with inverse numerical Laplace transformation.

The second model transmittances contain variable coefficients. For example the delaying unit is of variable delay value, and the integrating unit is used only within a variable time interval dependent on medium flow velocity. Digital simulation of such units requires re--recording the memory content continuously in successive computation steps till reaching the steady state. It would be better to have a solution which provides response in one computation step on the basis of simple arithmetical operations.

An approximate solution of the differential equations (2) and (3) are the difference equations. The input temperature values in successive computation steps are then determined as follows :

$$T_{fi,j+1} = -\Delta\tau\, a\left(T_{fi,j} - T_{si,j}\right) + \\ -\frac{\Delta\tau}{\Delta x}\left(w_f - \frac{\Delta x}{\Delta\tau}\right)\left(T_{fi+1,j} - T_{fi,j}\right) + T_{fi,j} \qquad (29)$$

$$T_{si,j+1} = \Delta\tau\, b\left(T_{fi,j} - T_{si,j}\right) + \\ -\frac{\Delta\tau}{\Delta x}\left(w_s - \frac{\Delta x}{\Delta\tau}\right)\left(T_{si+1,j} - T_{si,j}\right) + T_{si,j} \qquad (30)$$

This solution corresponds to the plate cooler model of the sectionally concentrated parameters, described in [13].

The cooler is divided into a definite number of discrete elements (sections) in which the full mixing of the liquid is assumed.

A comparison of the results obtained from the analytical and difference models is presented in Fig.7.

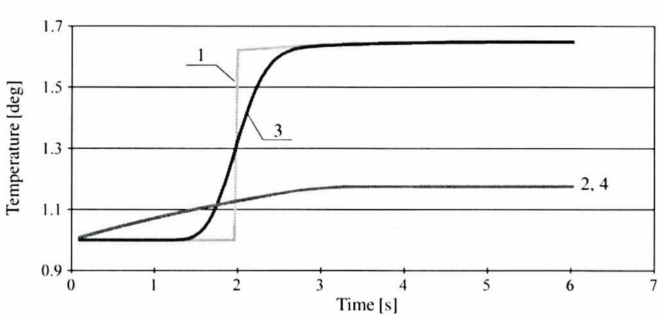

Fig.7. *Comparison of the plate cooler temperature courses computed by means of the first analytical model (curves 1 and 2) and the difference model (curves 3 and 4) for counter-current flow, L=2m, w_f=1m/s, w_s=2m/s, fresh water – curves 1 and 3, sea water – curves 2 and 4.*

From Fig.7 it can be observed that the difference model correctly reproduces the continuous courses of temperature (curve 4).The discontuity point of curve 1 is represented by the continuous course 3, and it makes a difference between the two characteristics.The tangent at the discontuity point in the difference model course approaches the stepwise characteristics as the time step $\Delta\tau$ decreases. It makes the result difference decrease and the real computation time increase. In the remaining points the difference model representation does not cause any changes of the characteristics. In real working conditions the stepwise change of the input temperatures does not occur. Therefore the difference model which provides the effective algorithm for digital simulation seems more suitable for simulation of the plate cooler operation.

VERIFICATION OF THE PLATE COOLER MODEL

In order to verify the above presented results a simulation of operation of the typical plate cooler was performed in accordance with the data from *Alfa–Laval*. Its results are as follows (see also Fig.8 and 9) :

M10 – MFM cooler of fresh water – sea water type

k = 6404 W/m²deg

X = 0.675 m Y = 0.326 m Z = 0.00415 m

Initial distribution of temperature along cooler plate :

$t_{fx}(x,0)$ = 39.3°C $t_{sx}(x,0)$ = 39.3°C

Signal set at cooler inlet :

$t_{0f}(\tau)$ = 75.0°C	$t_{0s}(\tau)$ = 39.3°C
ρ_f = 977 kg/m³	ρ_s = 988.5 kg/m³
c_f = 4180 J/(kg·deg)	c_s = 4170 J/(kg·deg)

Simulated temperatures :

t_{2f} = 64.9°C t_{2s} = 52.2°C

Calculated temperatures :

t_{2f} = 64.99°C t_{2s} = 52.24°C

M10 – MFM cooler

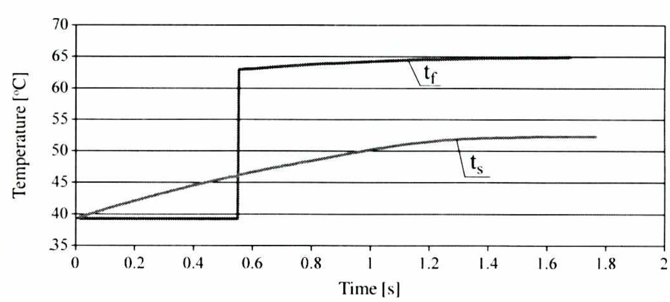

Fig.8. *Dynamic characteristics of M10-MFM Alfa–Laval plate cooler determined by means of the analytical model on the basis of the producer's data*
Notations : *t_f – fresh water, t_s – sea water*

M10 – BFM cooler of SAE 40 oil – sea water type

k = 445.1W/m²deg

X = 0.740 m Y = 0.324 m Z = 0.00257 m

Initial distribution of temperature along cooler plate :

$t_{ox}(x,0)$ = 32.0°C $t_{sx}(x,0)$ = 32.0°C

Signal set at cooler inlet :

$t_{0o}(\tau)$ = 65.0°C	$t_{0s}(\tau)$ = 32.0°C
ρ_o = 983.4 kg/m³	ρ_s = 1010 kg/m³
c_o = 1980 J/(kg·deg)	c_s = 4070 J/(kg·deg)

Simulated temperatures :

t_{2o} = 55.0°C t_{2s} = 39.0°C

Calculated temperatures :

t_{2o} = 54.98°C t_{2s} = 39.04°C

M10 – BFM cooler

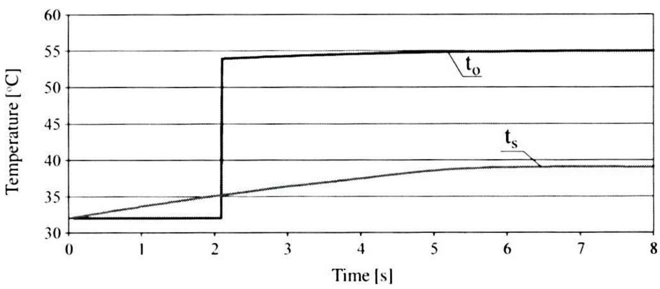

Fig. 9. *Dynamic characteristics of M10-BFM Alfa–Laval plate cooler determined by means of the analytical model on the basis of the producer's data*
Notations : *t_o – oil, t_s – sea water*

FINAL REMARKS, CONCLUSIONS

The elaborated theoretical models of the plate cooler provide good conformity with the simulation results of the real cooler operation. The courses of cooler model responses to the stepwise input function can be justified physically, and the model static characteristics are in compliance with experimental data. Each of the models can be used for digital simulation, depending on the available simulation time. The cooler model makes it possible to simulate the influence of failure, e.g. increase of thermal resistance of a cooler.

On the basis of the performed cooler model analyses the following can be concluded :

- The obtained static characteristics are entirely adequate within full range of load which can occur in service.
- The transient-state characteristics can be justified physically, and their asymptotic values are in compliance with real data.
- The model in question correctly reacts to change of flow velocity of working media.
- It is possible to analyze thoroughly the simulated phenomena and processes. It concerns both control and technical-state diagnosing processes of the cooler.

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

NOMENCLATURE

a, b – plate cooler heat exchange constant for cooled and cooling side, respectively [s^{-1}]
c – specific heat capacity of liquid [J/(kg K)]
k – heat transfer coefficient [W/(m^2 K)]
w – flow velocity of liquid [m/s]
A – inter-plate cross section area, integration constant
B – integration constant
F – heat exchange area [m^2]
L – height of plate [m]
Q – heat [J]
S – argument of complex variable
T, t – temperature [K,°C]
X,Y – plate dimension along x – and y - axis, respectively [m]
Z – distance between plates [m]
ρ – density of liquid [kg/m^3]
τ – time [s]

Indices

0 – initial value
1 – input boundary value
2 – output boundary value
i – location along plate
j – time instant
f – fresh water
s – sea water
o – oil

BIBLIOGRAPHY

1. Baron B.: *Numerical methods in Turbo Pascal* (in Polish). Wydawnictwo Helion. Gliwice, 1995
2. Brodowicz K.: *Theory of heat and mass exchangers* (in Polish). PWN. Warszawa, 1982
3. Doetsch G.: *Practice of Laplace transformations* (in Polish). PWN. Warszawa, 1964
4. Douglas J. M.: *Dynamic processes and their control. Analysis of dynamic systems* (in Polish). WNT. Warszawa, 1976
5. Friedly J. C.: *Analysis of dynamic processes* (in Polish). WNT. Warszawa, 1975
6. Gdula S.J.ed.: *Heat transfer* (in Polish). PWN. Warszawa, 1984
7. Hobler T.: *Heat transfer and exchangers* (in Polish). WNT. Warszawa, 1979
8. Mielewczyk A.: *Simulation model of control process of cooling and lubricating systems of medium speed diesel engine* (in Polish) Doctorate thesis. Technical University of Gdańsk, Faculty of Ocean Engineering and Ship Technology. Gdańsk, 1998
9. Mielewczyk A.: *Computer simulation of control processes of ship power plant auxiliary systems*. Polish Maritime Research, 2000, Vol.7, No 3 (25)
10. Mikielewicz J.:*Modeling of heat-flow processes* (in Polish). Institute of Fluid Flow Machinery. PAN. Wrocław, 1995
11. Mikusiński J.: *Calculus of operators* (in Polish). PWN. Warszawa, 1957
12. Petela R.: *Heat flow* (in Polish). PWN. Warszawa, 1983
13. Piekarski M., Poniewski M.: *Dynamics and control of heat and mass exchange processes* (in Polish). WNT. Warszawa, 1994
14. Staniszewski B.: *Thermodynamics* (in Polish). PWN. Warszawa, 1986
15. International Electrotechnical Commission: *Analysis techniques for system reliability - Procedure for failure mode and effects analysis (FMEA)*, Technical Committee, 1985, No 56

Conference

KONBIN 2001

On 22÷25 May the International Conference on

Safety and Reliability

was held at Szczyrk. 125 papers were prepared for the conference, read during four topical groups as follows :

- I – General theoretical problems of reliability and safety (21 papers)
- II – Theoretical background of research on reliability and safety (40 papers)
- III – Investigation methods of reliability and safety (29 papers)
- IV – Selected application examples of reliability and safety theory (35 papers)

The main organizers of the conference, Polish Safety and Reliability Association and Air Force Institute of Technology drew interest of 23 foreign scientific centres to the conference, and thus also experts from Brasil, China, Czech Republic, Finland, Spain, Canada, Germany, Portugal, Romania, Slovakia, Sweden, Ukraine and Italy presented their papers.

The Polish authors represented 37 universities, academies, scientific institutes and research centres as well as central institutions.

Gdynia Maritime Academy (with its 20 papers mainly of topical group II), Air Force Institute of Technology (17 papers mainly of topical group IV), Warszawa University of Technology (10 papers) and Technical University of Wrocław (9 papers) contributed most to the content-related preparation of the conference.