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OPTIMIZING THE VALUES OF THE SELECTED SCALAR QUANTITIES IN DESIGNING A CONDENSER OF MARINE STEAM TURBINE ACCORDING TO THE ADOPTED CRITERIA IN THIS DESIGNING

Krzysztof Łukaszewski

Gdynia Maritime University
Faculty of Marine Mechanical Engineering
Morska Street 81-87, 81-225 Gdynia, Poland
tel.: +48 58 6901304, fax: +48 58 6901399
e-mail: klukasz@am.gdynia.pl

Abstract

The paper presents designing activities concerning the rating of the values of scalar quantities optimized in accordance with the accepted criteria in designing a condenser of marine steam turbine, which result from the forecasted characteristic, most frequently occurring set points of steam turbine power appearing during the operation of a sea ship. It refers to the method of designing heat exchangers contained in the paper [1] and to the issues raised in the publication [2]. The paper also presents exemplary computational results of the above-mentioned optimized scalar quantities, which refer to designing a shell and tube condenser of marine steam turbine. It was assumed in the paper that the values of mass the mass flow rate of the steam: one $\dot{m}_p$, *0.75* $\dot{m}_p$, *0.5* $\dot{m}_p$, *which result from the assumed characteristic set points of marine steam turbine, which occur during the operation of a sea ship. Then, the selected designing activities referring to the above-mentioned optimization were exemplified. On the basis of the computational results, it was shown that the appropriate division of heat exchange area into the definite numbers of condensers, which results from the operation of steam turbine of a definite sea ship, increases operation effectiveness of these condensers in relation to one marine steam turbine condenser. This is a result of optimal values retention: velocity of cooling water of the definite condenser or condensers and temperature of this water on the output of this condenser or condensers during characteristic set points in the operation conditions defined in designing. Next, economic advantages were pointed out, which resulted from the described in the paper, technical solution. These advantages constitute the designing criterion that must be taken into consideration while making a decision about the division of heat exchange area into the definite number of condensers of marine steam turbine. Finally, the conclusions resulting from the paper's contents are presented.*

Keywords: *heat exchangers, optimization of condenser, designing of condenser*

1. Introduction

The aim of the paper is to indicate the definite successive designing activities, which enable designing the optimally functioning condenser or the definite set of condensers of marine steam turbine for characteristic set points in the operational conditions defined in designing. The optimized scalar quantities values were computed: velocity w_2 of the cooling fluid flow and temperature T_2'' of this fluid on the output of the condenser according to the adopted criteria for optimization in designing this condenser. The computational examples refer to a shell and tube condenser of marine steam turbine with the water installation cooling this condenser.

In designing a definite condenser, in order to ensure its functionality in the technical energy system, in which it occurs, it is necessary to rate heat exchange area for the maximum heat flux resulting from the operation of this system. Then, the excess heat exchange area for the forecasted fouling thickness is rated.

During the operation of marine steam turbine condensers, there often appears a considerable decrease of the amount of heat to be exchanged in comparison to the amount assumed in designing and an increase in the difference of temperature on the input of this condenser. In such a case the operation effectiveness of condensers decreases, which in turn, affects the effectiveness of marine steam turbine operation and leads to the increased operation costs of both: a turbine and a condenser. The following problem appears how to create optimization activities of a heat exchanger on its designing stage, so that it is possible to minimize the operation costs of this exchanger in its different operational conditions?

In the cooling systems of steam turbines, where pumps of non-controllable efficiency operate, an increase in hydraulic resistance due to the fouling settlements on the heat exchange area of a condenser causes a decrease in the amount of the cooling water flow through a condenser, and then an increase in the pressure of a condenser. When there is a possibility to control the water flow through a condenser, an increase in the amount of the cooling water flow results in an increase in the power taken by pumps. In turn, when the turbo set load is lower than the nominal load and significantly lower temperatures of the cooling water on the output of this condenser occur in autumn and winter periods, in the condenser appears pressure, which is lower than the optimal pressure for the last steps of low-pressure turbine. This situation leads to losses in the turbo set power. The mentioned above operational cases of condensers of marine steam turbines were described in the paper [3]. The paper [4] emphasized that the level of dynamic stresses of wicket gates of the last steps of the part of low-pressure turbine depend, among others, on pressure existing in the condenser, and in the case of, among others, an excessive drop in this pressure there appears an increase in static and dynamic stresses of these wicket gates.

The need to adapt the efficiency of water pumps, which cool the marine steam turbine condenser, to the atmospheric conditions because of their considerable power consumption was pointed out in the work [5]. In the article [6], the presented analysis proved that it is advisable to control the cooling water mass flow rate in steam turbine power plants to follow turbine load changes.

2. Optimization of values of the selected scalar quantities in designing a condenser of marine steam turbine according to the accepted criteria in this designing process

Optimization of the selected designing scalar quantities: velocity of the flow w_2 of water cooling the condenser, temperature T''_2 on the input of this water from the condenser, was broken down into the following activities, which were presented in the general diagram in Fig. 1. The adopted criterion for optimizing values of the indicated designing scalar quantities is the minimum sum of the overall costs $min\sum_{i=1}^{n} K_i(\tau)$ in time τ. Time τ is the set value τ_z and it constitutes the period, in which optimization is considered. It results from the adopted criterion in the optimization of the technical energy system, in which the definite heat exchanger occurs.
First of all, the general equation of the overall costs $K_C(\tau_z)$ of a condenser of marine steam turbine together with the installation of the water-cooling this condenser is determined [1]:

$$K_C(\tau_z)=\left(\frac{d_w}{d_z\alpha_1}+\frac{d_w\delta_s}{d_{sr}\lambda_s}+\frac{d_w^{0.2}\nu_2^{0.8}}{0.023\,\mathrm{Pr}_2^{0.4}\,\lambda_2}\frac{1}{w_2^{0.8}}\right)\dot{Q}_2(KC_A)\frac{\ln\frac{T_1-T'_2}{T_1-T''_2}}{T''_2-T'_2}+\sum_{i=1}^{m}K_{i,elwc}(\tau_z)+K_{mwc}(\tau_z)+K_{czwc}(\tau_z)+K_{pwc}(\tau_z)+$$

$$+K_{rwc}(\tau_z)+\sum_{i=1}^{m}K_{i,elinst}(\tau_z)+K_{\mathrm{minst}}(\tau_z)+\frac{\dot{Q}_2\tau_z(KC_e)w_2^2}{2c_{p2}(T''_2-T'_2)\eta_{p2}}\left(\frac{l}{d_w}\lambda_{str2}+\xi'+\xi''\right)+\frac{\dot{Q}_2w_{2k}^2\tau_z(KC_e)}{2c_{p2}(T''_2-T'_2)\eta_{p2}}\left(2\xi_k+\frac{l_{inst}}{d_{winst}}\lambda_{str2inst}+\sum_{i=1}^{m}\xi_{i,elinst}\right)+$$

$$+K_{pinst}(\tau_z)+K_{rinst}(\tau_z), \qquad (1)$$

where:

$c_{p.2}$ – specific heat at constant pressure p, J/kgK,
d_w – inside diameter of the condenser pipes, m,
d_{winst} – inside diameter of the installation pipes of the water-cooling the condenser, m,

d_z – outside diameter of the condenser pipes, m,
T_1 – temperature of steam condensation in the condenser, K,
T'_2 – temperature of the cooling water on the input of the condenser, K,
T''_2 – temperature of the cooling water on the output of the condenser, K,
l – length of the condenser pipes, m,
l_{inst} – length of the installation of the water cooling the condenser, m,
$\dot{Q}_2$ – heat flow of the water cooling the condenser, W,
α_1 – heat transfer coefficient taking heat from the condensing steam to the pipes walls of the condenser, W/m^{2}K,
w_{2k} – velocity of the cooling water flow through the input stub pipes to the condenser, m/s,
δ_s – thickness of the pipe wall of the condenser element, m,
η_{p2} – efficiency of the water pump cooling the condenser,
λ_s – thermal conductivity of the element wall of the heat exchange area, W/mK,
$\lambda_{str2inst}$ – friction resistance factor, at the flow of the cooling water, of the inside surfaces of the condenser installation,
λ_2 – thermal conductivity of fluid of the water cooling the condenser, W/mK,
$\lambda_{str\ 2}$ – friction resistance factor, at the flow of the cooling water, of the inside surfaces of the condenser pipes,
ν_2 – coefficient of kinematic viscosity of fluid of the water cooling the condenser, the value of the mean temperature in the core of the flowing water, m^2/s,
$\xi_{i,elinst}$ – local resistance coefficient of *i*-number of elements of the water installation cooling the condenser,
ξ_k – local resistance coefficient of the input stub pipe, through which the cooling water flows to the condenser,
ξ' – local resistance coefficient on the input of the cooling water to the condenser pipes,
ξ'' – local resistance coefficient on the output of the cooling water from the condenser pipes,
τ_z – set time of the condenser operation in the engine room of a ship, h,
$K_{czwc}(\tau_z)$ – the costs of cleaning heat exchange areas of the condenser in the definite time, PLN,
$K_{i,elwc}(\tau_z)$ – the costs of *i*-number of elements of the condenser, PLN,
$K_{mwc}(\tau_z)$ – the costs of assembling this condenser, PLN,
$K_{pinst}(\tau_z)$ – the costs of diagnosing the installations during the operation of the technical energy system, PLN,
$K_{pwc}(\tau_z)$ – the costs of diagnosing the condenser during its operation in the technical energy system, PLN,
$K_{rwc}(\tau_z)$ – the costs of preventive replacements of definite elements of the condenser, the costs of surveys and repairs of the condenser, PLN,
$K_{rinst}(\tau_z)$ – the costs of surveys and repairs of technical installations, PLN.

Next, the optimal values are rated: flow velocity $w_{2,opt}$ of the water cooling the condenser and temperature $T''_{2,opt}$ on the output of this water from the condenser with the use of models (2) [1] and (3) [1]. The obtained computational values $w_{2,opt}$ and $T''_{2,opt}$ are checked if they are within the range of accessible results $[w_{2,min}, w_{2,max}]$ and $[T''_{2,min}, T''_{2,max}]$. In the case when $w_{2,opt}$ and $T''_{2,opt}$ are not within this range, the appropriate values are selected.

Limitations in the velocity w_2 of the definite fluid, in this optimization, are the maximal and minimal values of the quantity $[w_{2,min}, w_{2,max}]$. The minimum value $w_{2,min}$ results from the relationship between the velocity of the definite fluid flow and fouling settlements on the heat exchange area, the maximal value $w_{2,max}$ results from the relationship between the velocity of the definite fluid flow as well as erosion and cavitation [1].

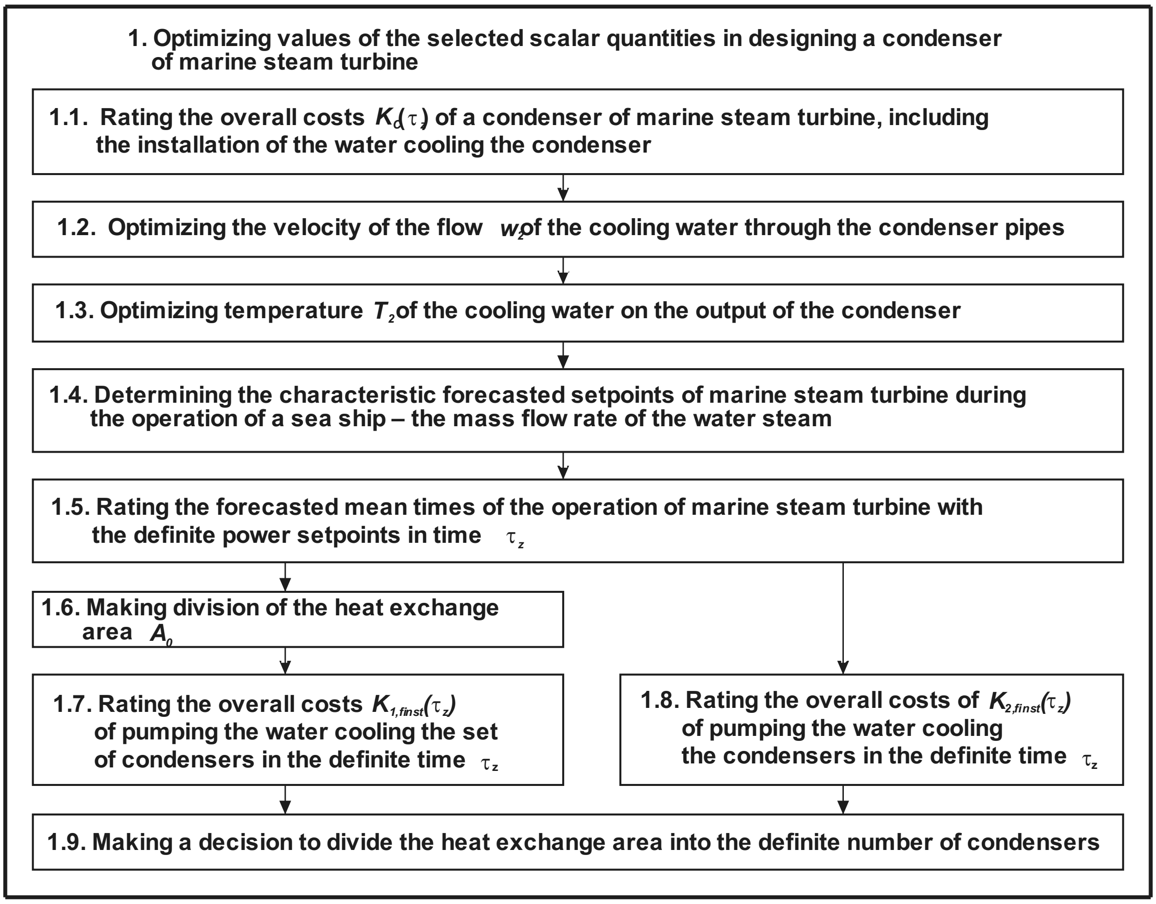

Fig. 1. Algorithm for optimizing values of the selected scalar quantities in designing a condenser of marine steam turbine

Limitations in temperature T''_2 of the cooling fluid on the output of the heat exchanger are the maximal and minimal values of this quantity $\left[T''_{2,min}, T''_{2,max}\right]$. The minimal value $T''_{2,min}$ is limited by the value of temperature on the input of the heat exchanger, whereas the maximal value $T''_{2,max}$ results from the relationship between the cooling fluid temperature and precipitations of fouling (salt) on the heat exchange area [1].

$$w_{2,opt} = \exp\left[\frac{1}{2.8}\ln\left(\frac{0.8 d_w^{0.2} \nu_2^{0.8} (KC_A) c_{p2} (T''_2 - T'_2) \eta_{p2}}{\Delta T_{sr}\, 0.023\, \mathrm{Pr}_2^{0.4}\, \lambda_2 \tau_z (KC_e)\left(\frac{l}{d_w}\lambda_{str2} + \xi' + \xi'' + 2\xi_k + \frac{l_{inst}}{d_{winst}}\lambda_{str2inst} + \sum_{i=1}^{m}\xi_{i,elinst}\right)}\right)\right], \quad (2)$$

$$T''_{2,opt} = T_1 - \exp\left[-1 - \frac{\tau_z (KC_e) w_2^2 \left(\frac{l}{d_w}\lambda_{str2} + \xi' + \xi'' + 2\xi_k + \frac{l_{inst}}{d_{winst}}\lambda_{str2inst} + \sum_{i=1}^{m}\xi_{i,elinst}\right)}{2 c_{p2} \eta_{p2} \left(\frac{1}{d_z \alpha_1} + \frac{\delta_s}{d_{sr}\lambda_s} + \frac{\nu_2^{0.8}}{0{,}023 d_w^{0.8} \mathrm{Pr}_2^{0.4} \lambda_2 w_2^{0.8}}\right) d_w (KC_A)}\right]. \quad (3)$$

With regard to the type of a ship and the function of marine steam turbine in the room engine, the characteristic forecasted power set points of marine steam turbine are determined, which occur in the operation of a sea ship. On the basis of this, one rates the *i*-number of most frequently occurring intervals of heat flows $\dot{Q}_i$ replaced in the definite condenser. The values $\dot{Q}_i$ are given by designers of this ship. The forecasted *i*-number of intervals of time $\tau_{i,z}$ is rated according to the occurrence of these *i*-number of $\dot{Q}_i$ in the set time τ_z of the condenser operation in the engine room of a ship. On the basis of this, the values adopted in the condenser optimization are determined. For instance, for computations included in the further part of this paper *j*-number of the values $\dot{Q}_{\max}$ and τ_z, $\dot{Q}_1 = 1\dot{Q}_{\max}$, $\dot{Q}_2 = 0.75\dot{Q}_{\max}$, $\dot{Q}_3 = 0.5\dot{Q}_{\max}$ and $\tau_{1,z} = 0.6\tau_z$, $\tau_{2,z} = 0.25\tau_z$, $\tau_{3,z} = 0.15\tau_z$, respectively.

In the earlier stage of condenser designing – before the optimization, the heat exchange area A_0 for the definite values of designing scalar quantities, including the value $1\dot{Q}_{\max}$, is rated. On this basis, taking into account the rated values $w_{2,opt}$ and $T''_{2,opt}$, the division of the heat exchange area A_0 into the definite number is made. This number results from the adopted characteristic values of $\dot{Q}_j$, e.g. 3, which means two condensers exchanging the heat flow in the amount $0.25\dot{Q}_{\max}$ and one in the amount $0.5\dot{Q}_{\max}$.

Then, the overall costs $K_{i,finst}(\tau_z)$ of pumping the water cooling the condenser $K_{1,finst}(\tau_z)$ and the set of condensers $K_{2,finst}(\tau_z)$ together with the installation of the water cooling the condenser (the set of condensers) in time τ_z with the use of the model (4) are rated:

$$K_{i,finst}(\tau_z) = \sum_{j=1}^{m} \frac{\dot{Q}_j \tau_{j,z} (KC_e) w_{2,opt}^2}{2c_{p2}\left(T''_{2,opt} - T'_2\right)\eta_{p2}} \left(\frac{l}{d_w}\lambda_{str2} + \xi' + \xi'' + 2\xi_k\right) + \sum_{j=1}^{m} \frac{\dot{Q}_j \tau_{j,z} (KC_e) w_{2,j}^2}{2c_{p2}\left(T''_{2,opt} - T'_2\right)\eta_{p2}} \left(\frac{l_{inst}}{d_{winst}}\lambda_{str2inst} + \sum_{i=1}^{m} \xi_{i,elinst}\right). \quad (4)$$

A preliminary decision about the division of the heat exchange area into a definite number of condensers or the introduction of one condenser to the installation is made on the basis of the comparison of the overall costs of pumping cooling water of both designing cases.

Then, repeat designing computations are carried out of the definite set of condensers or one condenser and their (its) overall costs in a given time τ_z are rated. Finally, the values of optimized scalar quantities and the division of the heat exchange area into the definite number of condensers or leaving one condenser are adopted in designing after meeting the distinguished designing conditions, which refer to the definite features of the condenser, including the desired reliability value $R_{wc}(t_z)$ of the condenser, in the set time (life) t_z, the definite maximal dimensions $L_{wc\max,i}$ of the condenser and its maximal weight $G_{wc\,max}$ together with fluids exchanging heat.

3. Exemplification of the optimization of values of the selected scalar quantities in designing a condenser of marine steam turbine according to the criteria adopted in this designing

The following values of definite scalar quantities resulting from the dependence (1) were assumed in order to show exemplary divergences in the computational values of the overall costs of the condenser and the set of condensers.

The optimal values were computed: velocity $w_{2.opt}$ of the cooling water flow thorough the condenser pipes and temperature $T''_{2.opt}$ of the cooling water on the output of the condenser with the use of equations (2) and (3).

Computational assumptions: d_z=0.02 m, d_w=0.015 m, d_{winst}=1.28 m, kind of material of pipes Cu Zn20 AI2 [7], λ_s=100.38 W/mK, l=4.7 m, l_{inst}=80 m, T_1=302 K, T'_2=295 K, on the basis of [8] ξ'=15, ξ''=18, ξ_k=16, λ_{str2}=0.026, $\lambda_{str2inst}$=0,05, $\sum_{i=1}^{m} \xi_{i,elinst} = 54$, then [9] $c_{p,2}$=3973 J/kgK, Pr_2=6.38, ν_2 =0.904*10^{-6} m^2/s, λ_2=0.571 W/mK, τ_z=8640 h, on the basis of [10] KC_e=0.0001678 [PLN/Wh], then [11] KC_A=828.5 [PLN/m^2], η_{p2}=0.75.

The following results were obtained: $w_{2.opt}$= 1.287 [m/s] and $T''_{2.opt}$= 301.8 [K].

Referring to the literature [9, 7], the following limitations of velocity of the flow of the cooling water $w_{2.opt}$= [1. 3] and temperature [12] $T''_{2.opt}$= [296.313] were assumed.

The computational assumptions: $w_{2.opt}$= 1.287 m/s and $T''_{2.opt}$= 299 K.

Next, the following assumptions were made: the j-number of heat flows $\dot{Q}_j$ exchanged between the fluids (fluids: steam, cooling water) in the condenser and the set time $\tau_{j,z}$ referring to the characteristic power set points of marine steam turbine in the set time τ_z=8640 h: $\dot{Q}_1$=26.755 MW, $\dot{Q}_2 = 20.066$ MW, $\dot{Q}_3$=13.378 MW and $\tau_{1,z}$ =5184 h, $\tau_{2,z}$ =2160 h, $\tau_{3,z}$ =1296 h, respectively.

The value of the heat exchange area A_0 is introduced from the computations preceding the considered optimization, and then its division into the appropriate number resulting from the assumed j-number values of heat flows $\dot{Q}_j$ is made. In this paper, the heat exchange area A_0 is computed and divided into 3 parts, which means two areas $A_{1,2}$ exchanging heat in the amount 0.25 $\dot{Q}_{\max}$ and one area A_3 in the amount 0.5 $\dot{Q}_{\max}$.

In order to illustrate the computational example the following models were used:

– to calculate the Nusselt number Nu_2:

$$Nu_2 = 0.023\,\mathrm{Re}_2^{\;0.8}\,\mathrm{Pr}_2^{\;0.4}\text{, where: } \mathrm{Re}_2 > 10\,000\text{, } 0.7 < \mathrm{Pr}_2 < 100\text{, } \frac{l}{d_w} > 60 \text{ [13],} \tag{5}$$

– to calculate the mean coefficient of heat consumption $\alpha_{1,sr}$ from condensing steam to the whole circumference of the vertical pipe of the condenser:

$$\alpha_{1,sr} = 0.728\left[\frac{(\rho_k-\rho_1)\,g\,\lambda_k^3\,r}{\nu_k\,(T_1-T_{sk})\,d_z}\right]^{\frac{1}{4}} \text{[14],} \tag{6}$$

where:

ρ_k – condensate density, kg/m^3,
ρ_1 – water steam density, kg/m^3,
g – gravity acceleration, m/s^2,
λ_k – heat conductivity of condensate, W/mK,
r – heat (enthalpy) of evaporation, J/kg,
ν_k – kinematic viscosity of condensate, m^2/s,
T_{sk} – average temperature of outside walls of the condenser pipes, K,

– to calculate the heat permeability coefficient k:

$$k = \frac{1}{\dfrac{1}{\dfrac{d_z}{d_w}\alpha_1} + \dfrac{1}{\dfrac{d_{sr}}{d_w}\dfrac{\lambda_s}{\delta_s}} + \dfrac{1}{\alpha_2}}\text{, where: } \alpha_1 > \alpha_2\text{,} \tag{7}$$

– to calculate i-number of heat exchange areas:

$$A_i = \frac{\dot{Q}_j}{k\,\Delta T_{sr}}. \tag{8}$$

The computational assumptions are as follows: ρ_k=996 kg/m^3, ρ_1=0.02909 kg/m^3, g=9.81 m/s^2, λ_k=0.6132 W/mK, r=2432.3 kJ/kg, ν_k=0.825*10^{-6} m^2/s.

Having performed the appropriate computations, the following values were adopted: T_{sk}=300.651K, α_1=9729 W/m^{2}K, taking into account the adopted correction coefficient α_1=0.6$\alpha_{1,sr}$ on the basis of [9], α_2=5345 W/m^{2}K, k=3500 W/m^{2}K, A_0=1619 m^2, the number of pipes n_0=7314, mass concentration of the flow of the water cooling the condenser $\dot{m}_{2,0}$=1685 kg/s, velocity of the water flow through the installation pipes of the water cooling the condenser $w_{2,0}$=$w_{2,opt}$=1.287 m/s, then $A_{1,2}$=404.8 m^2, $n_{1,2}$=1828, $\dot{m}_{1,2} = 422$kg/s, A_3=809.5 m^2, n=3657, $\dot{m}_3 = 843$kg/s. Velocities of the water flow through the installation pipes of the water cooling the set of condensers $A_{1,2}$ and A_3 are respectively: exchanging heat in the amount 20 MW, w_2=0.97 m/s, and exchanging heat in the amount 13 MW, w_3=0.647 m/s.

The overall costs of pumping the cooling water in time τ_z through the condenser are $K_{1,finst}(\tau_z)$=351397 PLN, including the installation of the water-cooling the condenser, whereas in case of the set of condensers they are: $K_{2,finst}(\tau_z)$=271016 PLN, including the installation of the water-cooling the condensers. The profit resulting from the division of the heat exchange area A_0 in the computational case is 22.9% in one year of the operation of the set of condensers.

4. Conclusions

The division of the area A_0 into the definite number of condensers reduces the costs of pumping the cooling water in the forecasted time τ_z, in the case when periodical changes of set points of marine steam turbine power occur during the operation of a sea ship there.

The designing decision to divide the heat exchange area A_0 and to construct thereby the set of condensers results from the analysis of periodical, forecasted changes in power of marine steam turbine during the operation of a sea ship.

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