MARIUSZ GIERNALCZYK, D.Sc.,M.E.
Gdynia Maritime Academy
Marine Engineering Faculty

ANDRZEJ BALCERSKI, Prof.,D.Sc.,M.E.
Technical University of Gdańsk
Polish Naval Academy, Gdynia

Probabilistic models for designing the sea--water pump systems of ship power plant

Part I

Estimation of probabilistic characteristics of operational condition parameters of the sea-water cooling systems of ship power plant

SUMMARY

In this paper the original method is presented which makes it possible to select a pump arrangement by applying the statistical distribution of real demand for cooling sea water, obtained with taking into account the statistical distributions of sea-water temperature and the total cooling heat flow of power plant during ship operation.

INTRODUCTION

The cooling heat of the ship power plant machines and equipment is transfered to the sea water forced through coolers by the main sea-water pump (-s).

To provide operation reliability of the installation an additional stand-by pump is required of the same output as that of the main pump, by the classification society rules for such systems.

Various sea-water pump arrangements may be applied, e.g.:

- two identical pumps, one - continuous operating, the other - stand-by pump, (i.e.together providing 2 x 100% of the required pump output)
- more than two identical pumps (e.g. 3 x 50%, 4 x 33% or 5 x 25% of the required pump output) one of which serves as the stand-by pump for each of the remaining ones
- several pumps of different outputs, selected on the condition of providing the stand-by pump for the largest one out of the continuous operating pumps (e.g. 2 x 80% + 1 x 20% of the required pump output).

In the present methods of ship power plant design it is assumed that the heat flows received from the main engine and auxiliary engines are determined on the conditions of their nominal power outputs, shut-down state of the evaporator, nominal outputs of : the surplus condenser, drip cooler, compressors etc. The so determined heat flow must be carried away at the possible highest temperature of sea water (32÷35°C). In service conditions the main and auxiliary engines and other ship power plant devices operate at real service loads; the devices are often put out of the operation or work periodically, and sea-water temperature values are lower during prevailing time of ship service than those met in tropical conditions.

If that is not taken into account the so selected pumps operate for a long time at the outputs exceeding the really demanded ones, as well as at lower efficiencies.

In this paper the original method [13] is presented which makes it possible to select a pump arrangement by applying the statistical distribution of real demand for cooling sea water, obtained with taking into account the statistical distributions of sea-water temperature and the total cooling heat flow of power plant during ship operation (Fig.1).

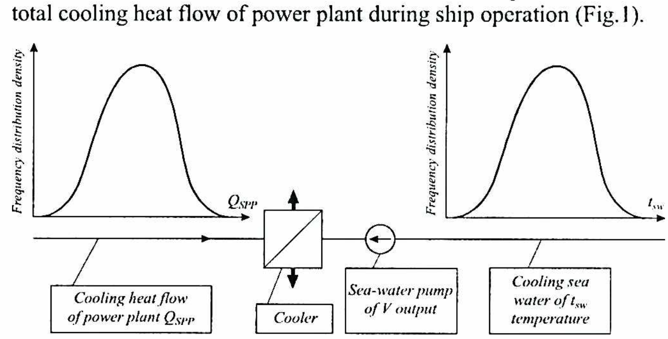

***Fig.1.** Schematic diagram highlighting the idea of the presented method*

ESTIMATION OF THE STATISTICAL DISTRIBUTION OF SEA-WATER TEMPERATURE VALUES ON ASSUMED SHIPPING ROUTES

The sea-water temperature distributions during ship service at a given shipping route can be predicted already in ship design stage. To this end the following data are required :

- sea-water temperature values over appropriate sea regions
- ship's routes and service speeds.

Sea-water temperature characteristics can be determined on the sea chart drawn with the use of the Mercator's cylindrical, right-angle mapping method, divided into the regions formed by :

★ the meridians placed at every 30^0 of geographic longitude
★ the parallels placed at every 10^0 of geographic latitude.

For each of the so established water regions crossed by the selected shipping route the sea-water temperature characteristics can be estimated by using the data published in [1].

In the publication the mean temperature lines (isotherms) for a given month, up to 10 m depth of water are contained. It is possible to determine sea-water temperature characteristics for each of the regions by superimposing the sea region network and temperature maps. An exemplary set of isotherms for a selected water region and month is presented in Fig.2.

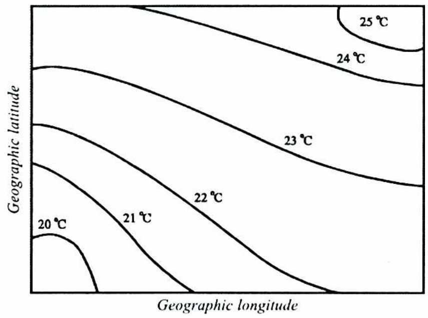

Fig.2. An exemplary sea region with superimposed mean sea-water isothermal lines for a given month

For the given shipping route on which several (a dozen or so) ships operate one cannot precisely predict all dates of arrivals to and departures from the ports taking into account disturbances due to storms, failures etc. Therefore for a longer time interval, e.g. several years, the real sea-water temperature characteristics is usually assumed as defined by :

t_i^{mv} - the arithmetic mean annual value of water temperature over the entire i-th sea region

t_i^{min}, t_i^{max} - the minimum and maximum water temperature occurring within one year over a given region, respectively.

The t_i^{mv} value can be determined by calculating the arithmetic mean of the monthly mean temperatures of i-th region. The mean temperature values of the region in successive months can be easily determined by using the isothermal lines for the considered water region.

Assuming a route between successive ports for the ship in question and drawing it on the chart of the recommended routes, and assuming an average service speed of the ship, one can determine the staying time in a given sea region by the ship, as well as its share in the total voyage time. If share of the staying time in i-th sea region is defined as :

$$\lambda_i = \frac{T_i}{T_{tv}} \tag{1}$$

where : T_i and T_{tv} - the staying time in i-th sea region and total voyage time (with excluded stop-over time in ports), respectively.

then for the entire voyage the following relationship holds :

$$\sum_{i=1}^{n} \lambda_i = 1 \tag{2}$$

The mean value of the sea-water temperature distribution for a given shipping route, t_{sr}^{mv} can be expressed as follows :

$$t_{sr}^{mv} = \sum_{i=1}^{n} t_i^{mv} \lambda_i \tag{3}$$

The standard deviation value of the temperature distribution can be determined on the assumption that the greatest water temperature difference Δt_{sr}^{max} which occurs during one year on the sea regions crossed by a given shipping route corresponds to 3 σ-statistical range (of p = 0.999); therefore :

$$\sigma_{sr} = \frac{\Delta t_{sr}^{max}}{3} \tag{4}$$

In the case of a truncated normal distribution (e.g. of the temperature range : $0 \div 32^0$C) appropriate frequency density values have to be corrected to get the cummulative distribution function value equal to 1 within that range. The presented method was proved correct on the basis of such calculations performed for some shipping routes (see e.g. Fig.3).

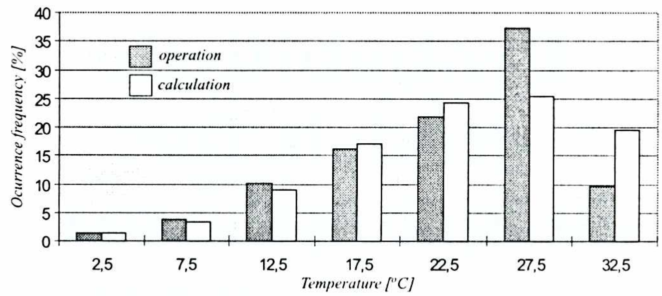

Fig.3. Distribution histograms of sea-water temperature on Hamburg - Yokohama shipping route

ESTIMATION OF THE TOTAL HEAT FLOW DUE TO SEA-WATER COOLING PROCESSES IN THE SHIP POWER PLANT

To estimate the total cooling heat flow in the ship power plant it is necessary to determine in advance the heat flow characteristics of cooling processes of particular engines and devices. The characteristics for every k-th device contain three parameters :

Q_k^{mv} - mean value of cooling heat flow
σ_k^{mv} - mean value of standard deviation of that flow
λ_k - operation-time share factor of the device during ship operation at sea.

In Fig.4 an overall structure is presented of the cooling heat flows of the ship power plant, carried away by sea water.

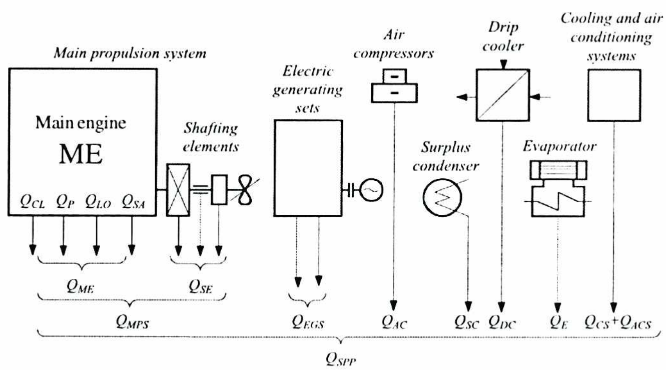

Fig.4. Overall structure of the cooling heat flows of the ship power plant, carried away by sea water

Therefore :

$$Q_{SPP} = Q_{MPS} + Q_{EGS} + Q_{AC} + Q_{SC} + Q_{DC} + Q_E + Q_{CS} + Q_{ACS} \tag{5}$$

The cooling heat flow Q_{MPS} connected with operation of the main propulsion system comprises the main engine cooling heat flow Q_{ME} consisting of :

- ✓ the fresh-water cooling heat flow Q_{CL} of the cylinder liners, exhaust valves and turbocharger bodies
- ✓ the fresh-water cooling heat flow Q_P of the pistons (if such system is applied)
- ✓ the heat flow Q_{LO} of circulating lubrication oil
- ✓ a part of or the entire supercharging air cooling heat flow Q_{SA},

and the cooling heat flow Q_{SE} of the shafting elements.

Hence :

$$Q_{MPS} = Q_{CL} + Q_P + Q_{LO} + Q_{SA} + Q_{SE} \tag{6}$$

Producers of ship diesel engines publish the data for estimating the relative values of particular cooling heat flows in function of the engine load, e.g. [11,12,16].

In the case of utilizing a part of the heat flow Q_{CL} to heat the sea water in the evaporator, and using a separate sea-water pump for exclusive cooling the evaporator's condenser, relevant values of the cooling heat flow of the cylinders, Q^*_{CL} can be obtained from the following expression :

$$Q^*_{CL} = (1-\chi)Q_{CL} \tag{7}$$

where :

χ - the utilization coefficient of the heat of cooling the cylinders to produce fresh water [14,15,17].

The evaporator condenser, if cooled by means of the fresh water, low-temperature circulation system, can be disregarded in calculations as the evaporation heat of the sea water within the evaporator heater, absorbed from the cylinder cooling water, returns to the cooling system in the form of the condensation heat.

The cooling heat flow of the shafting elements can be estimated with the use of the shaftline efficiency :

$$Q_{SE} = (1-\eta_{SE})N_{ME} \tag{8}$$

where :

η_{SE} - the shaftline efficiency accounting for the heat losses occurring in the cooled shafting elements such as in general : the reduction gear, journal bearings, thrust bearing, CPP setting mechanism etc.

Using the above specified data one can determine the relationship between the total (absolute or relative) heat flow due to cooling the main propulsion system, and its loading (Fig.5.1).

If the relationship is known and the main engine relative loading during sea voyage is expressed in the form of a frequency histogram, the distribution of the relative total heat flow due to cooling can be estimated [3,5]. In [2] the values are contained of : the mean, relative main engine loading $\overline{N}^{mv}_{ME}$, relative standard deviation of that loading, σ_{ME}, and operation-time share factor λ_{ME}. Making use of these data one can determine the distribution histogram of the main propulsion system relative loading (Fig.5.2).

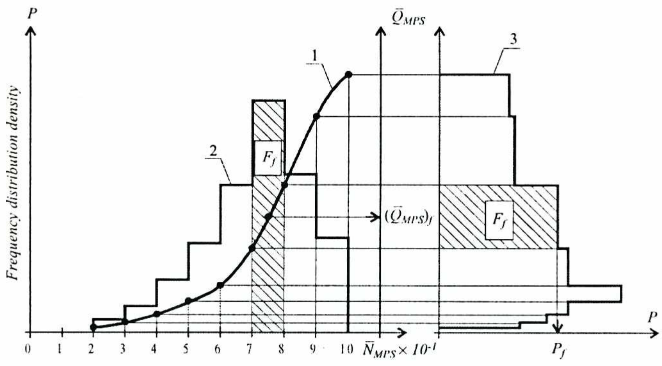

Fig.5. Scheme for determining the distribution histogram of the relative, total cooling heat flow of the main propulsion system :
1 - the relative, total cooling heat flow versus relative loading
2 - the relative loading distribution histogram
3 - the distribution histogram of the relative, total cooling heat flow

The relevant parameters concerning the main propulsion system are defined as follows :

$$\overline{N}^{mv}_{MPS} = \frac{N^{mv}_{MPS}}{N^{nom}_{MPS}}$$

$$\overline{\sigma}_{MPS} = \frac{\sigma_{MPS}}{N^{mv}_{MPS}} \tag{9}$$

$$\lambda_{MPS} = \frac{T_{MPS}}{T}$$

where :

T_{MPS}, T - the main propulsion system time of operation and annual sea voyage time of operation, respectively.

By using the histogram presented in Fig.3 the mean relative value of the total cooling heat flow of the main propulsion system can be estimated as follows :

$$\overline{Q}^{mv}_{MPS} = \sum_{f=1}^{p} (\overline{Q}_{MPS})_f \cdot p_f \tag{10}$$

where :

$(\overline{Q}_{MPS})_f$ - the mean relative value of the total cooling heat flow in the middle of f-th interval of the histogram

p_f - f-th interval frequency of the distribution histogram of total cooling heat values.

The mean value of the **cooling total heat of the main propulsion system** is determined by :

$$Q^{mv}_{MPS} = Q^{nom}_{MPS} \cdot \overline{Q}^{mv}_{MPS} \tag{11}$$

where :

$$Q^{nom}_{MPS} = Q^{nom}_{CL} + Q^{nom}_P + Q^{nom}_{LO} + Q^{nom}_{SA}$$

Using the data determined by means of the histogram in Fig.3 one can calculate the mean value of the standard deviation of the total cooling heat flow distribution, σ_{MPS}.

The above presented considerations deal with the situation of lack of appropriate data from exploitation of the ship in question. However the basic data ($\overline{N}^{mv}_{ME}, \overline{\sigma}_{ME}, \lambda_{ME}$) should be derived from relevant service data in the case when modernization of a ship is considered or if advisability of introduction of changes to a successive ship of those built in series is investigated.

Estimation of the mean value of **cooling heat flow of the auxiliary engines**, Q^{mv}_{EGS}, and the mean value of standard deviation of that flow, σ_{EGS}, as well as the operation-time share factor of the generating sets, λ_{EGS}, can be performed in the same way as in the case of the main propulsion system.

The distribution histograms of relative loading of the auxiliary engines can be prepared on the following assumptions :

- ✧ value of the electric energy demanded during sea voyage, predicted by the energy balance, is taken as the mean loading value of the electric power plant
- ✧ electric generator efficiency (about 0.95) is accounted for in the loading of the generating set engine
- ✧ if the electric energy demand is satisfied by the generating sets working in parallel, relative loading of the engines is equal
- ✧ values of the relative standard deviation of generating set loading , $\overline{\sigma}_{EGS}$, are contained within the range of 0.12÷0.24 [2,3,7].
- ✧ in the case of taking over the entire electric load or a part of it by a shaft generator during sea voyage, course of calculations remains unchanged. The reason is that in such loading case the cooling heat flows of both the main engine and generating set, are nearly of the same value and usually serviced by a common, central cooling system.

Moreover during sea voyage $\lambda_{EGS} = 1$ as the ship must be continuously supplied with electric energy in every condition.

Characteristics of the **cooling heat flow of the air compressors** can be estimated if the nominal electric power N_{AC}^{nom} absorbed from the electric network to drive the compressors at the nominal outlet pressure of 3 MPa, is known.

From an analysis of energy balances of the compressor aggregates it results that the cooling heat flow of the compressors, transfered to sea water is [10] :

$$Q_{AC} \cong 0.8 N_{AC} \tag{12}$$

This makes it possible to determine, at the limiting pressure values (of about 2.5 MPa and 3 MPa, respectively), the extreme values of the flow in question, $Q_{AC}^{min}, Q_{AC}^{max}$ and their mean value :

$$Q_{AC}^{mv} = \frac{Q_{AC}^{max} + Q_{AC}^{min}}{2} \tag{13}$$

as well as the standard deviation value (at the assumed range of 3 σ) :

$$\sigma_{AC} = \frac{Q_{AC}^{max} - Q_{AC}^{min}}{3} \tag{14}$$

The operation-time share factor of the compressors at sea, λ_{AC} is of the range of 0.25÷0.8 [2].

The cooling heat of the surplus condenser depends on the steam flow rate which is connected with a type and size of the ship as well as her engine power, fuel oil viscosity, engine service loading, filling degree of the fuel spare tanks, and sea-water temperature [4].

By using technical data and steam balances of the modern container ships [8] the regression equations which determine surplus values of the steam flow passing through the surplus condenser in summer conditions, ΔD_s , and shortage values in winter conditions, ΔD_w , were expressed in function of :

N_{ME}^{nom} - main engine rated power, [kW]
v - fuel oil kinematic viscosity, [cSt/50°C].

All so determined regression relationships were statistically verified by testing their significance with the use of F test. Some of the relationships are presented in Tab.1.

Tab.1. Selected multi-parameter regression equations determining the surplus ΔD_s of the steam flow passing through the surplus condenser in summer and its shortage ΔD_w in winter [kg/h]

Relationship	R Correlation coefficient	σ Standard deviation	F test value	F_{kr} Critical test value
$\Delta D_s = 0.00233\, N_{ME}^{nom} + 1.70156\, v$	0.819	563.34	11.299	6.09
$\Delta D_w = 0.171\, N_{ME}^{nom} - 2.156\, v$	0.906	597.86	33.940	4.10

Having the so determined values of ΔD_w and ΔD_s and assuming a linear relationship between the rate of steam flow through the surplus condenser, as well as sea-water temperature, one can determine the temperature range within which the steam flow through the condenser occurs (Fig.6).

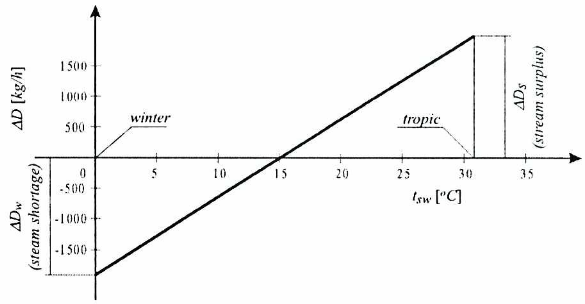

Fig.6. The linear relationship between the rate of steam flow through the surplus condenser, and sea-water temperature (for a container ship)

The distribution characteristics of the steam flow through the surplus condenser, D_{SC}^{mv}, σ_{SC} , versus the distribution characteristics of sea-water temperature, can be estimated in a similar way as that applied for the main engine.

Therefore the mean **heat flow transfered to sea water within the surplus condenser** is as follows :

$$Q_{SC}^{mv} = \Delta D_{SC}^{mv} \cdot r \tag{15}$$

where :
r [kJ/kg] - steam condensation heat at a given pressure.

The operation-time share factor of the surplus condenser at sea can be expressed as follows :

$$\lambda_{SC} = \lambda_{ME} \lambda_{ME}^{UB} \lambda_{UB}^{SC} \tag{16}$$

where :

λ_{SC} - share factor of operation-time of the surplus condenser at sea
λ_{ME} - share factor of operation-time of the main engine at sea
λ_{ME}^{UB} - share factor of operation-time of the exhaust gas utilization boiler during work of the main engine
λ_{UB}^{SC} - share factor of the surplus condenser operation-time during work of the exhaust gas utilization boiler.

The factor λ_{SC} represents operation-time share of the ship in sea regions at that water temperature when the steam flow through the condenser occurs.

To estimate the **heat flow carried away to the sea water within the drip cooler** it is necessary to know the steam flow rate through the cooler, which is equivalent to knowing the delivery rate of the steam produced for heating purposes.

The delivery rate of the steam produced in summer, D_s [kg/h], and that in winter, D_w [kg/h], was estimated with the use of the regression equations presented in Tab.2 and containing the following parameters :

$L \times B \times d$ - product of ship main dimensions, [m³]
N_{ME}^{nom} - main engine rated power , [kW]
v - fuel oil kinematic viscosity, [cSt/50°C]
z - number of crew members, [persons]
Z - ship operation range, [naut. miles].

Tab.2. Selected multi-parameter regression equations for determining the delivery rate of heating steam demanded for the containership power plant in summer, D_s, and in winter, D_w [kg/h]

Relationship	R Correlation coefficient	σ Standard deviation	F test value	F_{kr} Critical test value
$D_s = 1.16\, Z + 0.369_z - 0.53\, N_{ME}^{nom} + -1.4\, v + 1.4\, LBd$	0.997	179.4	41.31	9.01
$D_w = 0.149\, z - 0.44\, N_{M}^{nom} + -0.01\, v + 1.28\, LBd$	0.985	904.9	23.12	6.39

Having the so determined values of D_w and D_s , assuming a linear relationship between the rate of steam flow through the drip cooler, D_{DC} and the sea-water temperature, and proceeding in the same way as previously, one can determine the distribution characteristics of the rate of steam flow through the drip cooler, (D_{DC}^{mv}, σ_{DC}).

The mean heat flow carried away to the sea water within the drip cooler can be calculated as follows :

$$Q_{DC}^{mv} = D_{DC}^{mv} \cdot \Delta e \tag{17}$$

where :

Δe - drop of drip specific enthalpy in the drip cooler, [kJ/kg] (usually abt. 100 kJ/kg).

The appropriate value of the operation-time share factor of the drip cooler at sea, λ_{DC} is equal to 1.

Characteristics of the **cooling heat flow absorbed from the refrigerating devices** concern only the devices whose condensers are cooled by the sea water delivered by main sea-water pumps, or by the fresh water, low temperature system.

The heat flow carried away by sea water from every refrigerating device contains :

- the heat flow equivalent to the refrigerating capacity of the refrigerating and air conditioning devices, Q_{CS}, Q_{ACS}, respectively
- the heat flow corresponding with the driving power of the compressors of such devices, N_{CS}, N_{ACS}, respectively,

and it can be described by the following relationship :

$$\left.\begin{aligned} Q_{CS} &= Q_{CS}^{nom} + N_{CS} \\ Q_{ACS} &= Q_{ACS}^{nom} + N_{ACS} \end{aligned}\right\} \tag{18}$$

The mean values of these flows are expressed as follows :

$$\left.\begin{aligned} Q_{CS}^{mv} &= \beta_{CS} \cdot Q_{CS}^{nom} \\ Q_{ACS}^{mv} &= \beta_{ACS} \cdot Q_{ACS}^{nom} \end{aligned}\right\} \tag{19}$$

where :

β_{CS}, β_{ACS} - loading factors of the refrigerating and air conditioning systems, respectively

Values of the factors can be assumed as follows [2,3,10] :

- for cargo refrigerated compartment : $\beta_{CS} = 0.6 \div 0.8$
- for refrigerated provision store : $\beta_{CS} = 0.85 \div 0.95$
- for accommodation air conditioning system : $\beta_{ACS} = 0.6 \div 0.8$
- for Central Control Station (CCS) air conditioning system : $\beta_{ACS} = 0.7 \div 0.9$;

and : $\sigma_{CS} = \sigma_{ACS} = 0.05 \div 0.1$

The operation-time usage factors of the refrigerating devices, λ_{CS} and λ_{ASC} should be determined separately, in accordance with the kind of their operation and influence level of external conditions :

- for the refrigerated provision stores usually placed far away from the external hull plating, i.e. practically independent of ambient temperature, λ_{CS} value can be determined if a scheduled operation-time per day is known, usually 14 h/day, hence $\lambda_{CS} \cong 0.58$
- for the refrigerated cargo compartments and accommodation air conditioning systems relevant values of the factors can be assumed the same as for the surplus condenser, i.e. : $\lambda_{CS} = \lambda_{ACS} \cong \lambda_{SC}$ as the ambient conditions influence their time of operation
- for CCS air conditioning systems λ_{ACS} value can be assumed close to 1 as CCS space is connected to the engine room where rather high temperature prevails.

Having the values $Q_k^{mv}, \sigma_k, \lambda_k$ which characterize the total cooling heat flows of the power plant machines and systems in question one can estimate the distribution parameters of the **total, sea-water cooling heat flow of the power plant**, $Q_{SPP}^{mv}, \sigma_{SPP}$. To this aim it is necessary to assume that :

- ⇨ the cooling heat flow distributions of all machines and devices in question are of normal form [2,3]
- ⇨ isochronic values of the cooling heat flows of all machines and devices in question are independent of each other.

Investigations relating the main and auxiliary engines showed that values of their correlation coefficients R are smaller than 0.44, usually placed within the range : 0.025÷0.16 [2,3,7].

Lack of statistical dependence may be assumed between random variable values of the following cooling heat flows :

- ✓ of the main engine and surplus condenser
- ✓ of the main engine and air compressors
- ✓ of the auxiliary engines and drip cooler etc,

as there are no functional relationships between these devices.

Hence in further considerations mutual independence is assumed of all random variables of the component cooling heat flows of power plant.

On this assumption the normal distribution parameters of the sea-water cooling heat flow of the power plant can be calculated as follows [9] :

$$\left.\begin{aligned} Q_{SPP}^{mv} &= \sum_{k=1}^{l} \lambda_k Q_k^{mv} \\ \sigma_{SPP} &= \sqrt{\sum_{k=1}^{l} (\lambda_k)^2 (\sigma_k)^2} \end{aligned}\right\} \tag{20}$$

It may be assumed that $\lambda_{SPP} = 1$ as some receivers must be continuously supplied with electric energy in all ship operation states.

To estimate the distribution of the total cooling heat flow of the power plant, its extreme values can be assumed as follows :

$$\left.\begin{aligned} Q_{SPP}^{min} &= Q_{SPP}^{mv} - 3\sigma_{SPP} \\ Q_{SPP}^{max} &= Q_{SPP}^{mv} + 3\sigma_{SPP} \end{aligned}\right\} \tag{21}$$

To be continued

NOMENCLATURE

N - loading (power)
Q - cooling heat flow
T - time
t - sea-water temperature

AC - air compressors
ACS - air conditioning system
CL - cylinder liners
CS - cooling system
DC - drip cooler
E - evaporator
EGS - electric generating sets
LO - lubricating oil
ME - main engine
MPS - main propulsion system
P - pistons
SA - supercharging air
SC - surplus condenser
SE - shafting elements
SPP - ship propulsion plant
SWP - sea-water pump
UB - exhaust gas utilization boiler

Indices

e - specific enthalpy
mv - mean value
nom - nominal (rated) or required
s - summer
sr - shipping route
sw - sea water
tv - total voyage
w - winter
– - relative value

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Conference

SCIENCE AND RESEARCH STANDARDS

On 7 December 1999 a seminar under this heading was held, organized and hosted by the Head Office of Polish Register of Shipping in Gdańsk. Five interesting papers were presented to the seminar participants representing maritime universities, scientific research centres, shipyards and maritime administration. The papers were prepared by employees of Polish Register of Shipping (PRS), Ship Design & Research Centre (CTO) and Technical University of Gdańsk (PG) and dealt with the following topics :

- „ Motion of liquid in the partly filled tank – theory, computer algorithms and simulation"– by Jan Jankowski, (of PRS)
- „ Motion of liquid in the partly filled tank – a model experiment" – by Leszek Konieczny, (of CTO)
- „ Wave impact load acting on the ship bow – a safety standard verified by model experiment" – by Jan Jankowski, (of PRS)
- „ Strength assessment of bow visor fittings – by Marian Bogdaniak (of PRS)
- „ Vibration of the ship propulsion systems" – by Edmund Wittbrodt, and Stefan Sawiak, (of PG)

The papers were highly appreciated by the seminar participants mainly due to the fact they referred to the latest research results.

Current reports

Maritime University of Szczecin
Marine Traffic Engineering Institute

Pre-design research on the Sea Ferry Terminal at Świnoujście

Building the Sea Ferry Terminal at Świnoujście was one of the biggest port investments over last several years. It is located on the east side of the Świna between 3.3 and 4.7 km of the Świnoujście-Szczecin fairway. The Marine Traffic Engineering Institute of the Maritime University of Szczecin, headed by Wiesław Galor, D.Sc., effectively contributed to that huge and economically important undertaking.

The Institute was asked to determine:

- location of the ferry berths
- behind-propeller stream velocities close to sea bed to make designing of the sea bed strengthening possible
- water area breath required for the ferries manoeuvring along the quays
- conditions of approaching and putting off the ferry berths, to minimize quay and berth slope failures, to verify influence of the berth 6 on the traffic at the Municipal River Crossing, under the appropriate icing conditions to be assumed in the research.

In the first stage of design and building five ferry berths along the Świna's bed were assumed. The research aim was accomplished by applying a computer simulation model of ship motion over the water area in question. The water area model was elaborated on the basis of the navigation charts, geodesy maps and hydro-meteorological data (on icing, wind, current distribution).

In the ship motion model the characteristics of the following ferries : m/f „JAN ŚNIADECKI", POMERANIA and a designed special ferry (of ro-ro type) were taken into account. The ferry models were verified on the basis of full-scale investigations as well as an own database of ship manoeuvring characteristics.

The ferry's position on the water area was presented on the monitor screen by means of the panoramic imaging technique with optional scale changing.

The investigations were arranged as a series of the short voyages executed by the experienced ship masters of the ferries in question, carried out in the conditions determined by the ranges of the investigated variables. The real-time voyage quality parameters were recorded for further statistical processing and analyzing with the use of a special computer software.

280 simulation voyages were executed. The final report comprised the following results for each of the ferry berths :

- impact energy of the ferry (of its first contact with the quay)
- close-to-bed velocity distribution of the behind-propeller stream due to main propulsion
- over-quay velocity distribution of the behind-propeller stream due to thrusters.

The research results were applied to design the Sea Ferry Terminal.