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Research Article / Araştırma Makalesi

UTILIZATION OF SHIPBOARD AS A HEAT EXCHANGER IN SHIP CENTRAL COOLING SYSTEM: A NUMERICAL ANALYSIS

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ABSTRACT

A heat exchanger integrated to the central cooling system is used to cool the main engine in ships. Central cooling system serves a "sea water based" cooling option. The main focus of this study is to analyze the prospect of the shipboard to be used as a heat exchanger. 3-D CFD analysis (StarCCM+) results are compared with the results obtained by the SPH method implementing explicit time integration. The problem is also approached analytically by making certain assumptions and there is about 3% difference between the analytical and the numerical results.
Keywords: CFD, heat transfer, SPH, engine jacket cooling.

GEMİLERDE MERKEZİ SOĞUTMA SİSTEMİ İÇİN GEMİ BORDASININ BİR ISI DEĞİŞTİRİCİ OLARAK KULLANIMI: SAYISAL BİR ANALİZ

ÖZ

Gemilerde ana makine soğutulmasında merkezi soğutma sistemine entegre olmuş bir ısı değiştirici kullanılmaktadır. Merkezi soğutma sistemi deniz suyu kaynaklı bir soğutma seçeneği sunmaktadır. Bu çalışmada gemi dış kabuğunun deniz ile olan kontağından faydalanan bir ısı değiştirici kullanılması HAD (Hesaplamalı Akışkanlar Dinamiği) yazılımı (StarCCM+) ve SPH(Smoothed Particle Hydrodynamics) yöntemler yardımı ile deniz suyu sıcaklıklarında ısı transferi etkinlik analizleri yapılmıştır. Analitik olarak geometri incelenmiş elde edilen veriler sayısal sonuçlar ile %3 seviyelerinde bir yaklaşım göstermiştir.
Anahtar Sözcükler: HAD, Isı transferi, SPH, Ceket suyu soğutması.

1. INTRODUCTION

Sea water is used in today's ships both for cooling main engine jacket water, lubrication oil and meeting the cooling loads on central cooling line such as air-conditioning and cold room. Due to chemical and physical properties of seawater, maintenance and cleaning is frequently required for the lines and pumping equipment are used to put seawater into process in ships. Pollution due to the microorganisms and accumulation of salts contained in seawater also cause

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an additional thermal resistance in heat exchangers and besides performance loss [1]. This case create an effect that increase the maintenance and energy costs.

Sea water heat exchangers used in ships are required often to be taken to maintenance due to the characteristics of water. Because of the salinity rate and degree of acidity of seawater, heat exchangers are generally manufactured with stainless steel instead of normal steel, although their higher cost. In addition, because of the same reasons, maintenance and energy costs are occurred for circulating pumps that provide the circulation of seawater line [2]. Within this scope, removing the usage of seawater for the central cooling line means to reduce these costs. The wetted surfaces of the ships have a good degree of heat transfer capability, and it gives an idea to use the wetted surfaces as heat exchangers. In this study, by taking into consideration the heat transfer capability of the ship outer surface, shipboard, a heat exchanger model was developed and investigated by numerical and analytical approaches. Computational fluid dynamics and SPH methods was used in numerical analyses. Accuracy of the results were compared with developed analytic methods.

2. MATHEMATICAL MODEL

When the developed heat exchanger geometry is examined, it is consisted steel pipes welded to shipboard from inside with 45° welding seams and perpendicular to the direction of ship's length. Pipes working parallel and this provides to deactivate only the damaged pipe in case of any leakage and tearing. This geometry is not suitable to directly analytical solution. To get result, pipe flow and flat plate heat transfer theories were analyzed together.

The interaction occurring between the seawater and plate is similar to heat transfer arising from flow on a flat plate.

Experimental data for heat transfer can be written as, often adequate sensitivity;

$$Nu = C\,\mathrm{Re}_L^m\,\mathrm{Pr}^n \tag{1}$$

simple exponential equation, here m and n are constant exponent and the value of the constant C depends on geometry and flow.

If mean resistance and convection coefficients are known and resistance force is heat transfer speed from a surface (or to a surface) at constant temperature,

$$\dot{Q} = hA_s(T_s - T_\infty) \tag{2}$$

in Equation 2, T_s is the surface temperature, T_∞ is fluid temperature outside the boundary layer and A_s is the heat transfer surface.

The flow in the velocity boundary layer starts out as laminar, but if the plate is sufficiently long, the flow will become turbulent at a distance x_{cr} from the leading edge where the Reynolds number reaches its critical value for transition. The transition from laminar to turbulent flow depends on the surface geometry, surface roughness, upstream velocity, surface temperature, and the type of fluid, among other things, and is best characterized by the Reynolds number. The Reynolds number at a distance x from the leading edge of a flat plate is expressed as;

$$\mathrm{Re}_x = \frac{\rho V_x}{\mu} = \frac{V_x}{\nu} \tag{3}$$

For flow over a flat plate, transition from laminar to turbulent is usually taken to occur at $\mathrm{Re} \approx 1*10^5$. But the flow cannot be fully turbulent without Reynolds number reaches to $3*10^6$.

The local Nusselt number at a location x for laminar and turbulent flow over a flat plate is expressed as;

Laminar: $$Nu_x = 0.322\,\mathrm{Re}_x^{0.5}\,\mathrm{Pr}^{0.33} \qquad \mathrm{Pr} > 0.6 \tag{4}$$

Turbulen: $$Nu_x = 0.0296\,\mathrm{Re}_x^{0.8}\,\mathrm{Pr}^{0.33} \qquad 0.6 < \mathrm{Pr} < 60 \text{ and } 5.10^5 < \mathrm{Re} < 10^7 \tag{5}$$

In Equation 4 and 5, h_x is heat convection coefficient, k is thermal conductivity, Pr is Prandtl number and Re_x is Reynolds number at x point. It should be noted that to use these equations, the condition of Pr > 0.6 for Equation 4, 0.6 < Pr < 60 and $5.10^5 < Re < 10^7$ for Equation 5 should be satisfied.

The local friction and heat transfer coefficients are higher in turbulent flow than they are in laminar flow. Also, h_x reaches its highest values when the flow becomes fully turbulent.

The average Nusselt number over the entire plate is,

Laminar: $$Nu = 0.664\,\mathrm{Re}_L^{0.5}\,\mathrm{Pr}^{0.33} \qquad \mathrm{Re}_L < 5.10^5 \tag{6}$$

Turbulent $$Nu = 0.037\,\mathrm{Re}_L^{0.8}\,\mathrm{Pr}^{0.33} \qquad \frac{0.6 \le \mathrm{Pr} \le 60}{5.10^5 \le \mathrm{Re}_x \le 10^7} \tag{7}$$

In Equation 6 and 7, h is total heat convection coefficient, L is plate length and k is thermal conductivity.

Equation 6 gives the average heat transfer coefficient for the entire plate when the flow is laminar over the entire plate. Equation 7 gives the average heat transfer coefficient for the entire plate only when the flow is turbulent over the entire plate, or when the laminar flow region of the plate is too small relative to the turbulent flow region.

In developed geometry, heat transfer from hot water to shipboard firstly occur at the pipe welded to plate. Hot water heat in pipe is transferred through pipe thickness to weld seam and finally to plate.

Unlike the mean velocity, the mean temperature T_m will change in the flow direction whenever the fluid is heated or cooled. The energy transported by the fluid through a cross section in actual flow must be equal to the energy that would be transported through the same cross section if the fluid were at a constant temperature T_m.

$$E_{akışkan} = mc_pT_m = \int_m c_m.T.r.\partial m = \int_{A_c} \rho.c_p.T.r.u(r).V.dA_c \tag{9}$$

In Equation 9, m is the flow rate, c_p is specific heat and T_m is mean temperature.

For flow in a circular tube, the Reynolds number is defined as,

$$\mathrm{Re} = \frac{\rho V_{ort} D}{\mu} = \frac{V_{ort} D}{\nu} \tag{10}$$

The Nusselt number used to determine the heat transfer coefficient by convection for turbulent flow inside a pipe is defined as [3];

$$Nu_{in} = 0.023\,\mathrm{Re}_{in}^{0.8}\,\mathrm{Pr}^{0.33} \tag{11}$$

The temperature of a fluid flowing in a tube remains constant in the absence of any energy interactions through the wall of the tube. The thermal conditions at the surface can usually be approximated with reasonable accuracy to be constant surface temperature (T_s=constant) or constant surface heat flux (q_s=constant).

Surface heat flux is expressed as;

$$\dot{q}_s = h_x(T_s - T_m) \quad \text{(W/m2)} \tag{12}$$

where, h_x is the local heat transfer coefficient and T_s and T_m are the surface and mean fluid temperatures at that location. The mean fluid temperature T_m of a fluid flowing in a tube must change during heating or cooling. Therefore, when h_x=h=constant, the surface temperature T_s must change when q_s=constant, and the surface heat flux q_s must change when T_s =constant. Thus we may have either T_s =constant or q_s=constant at the surface of a tube, but not both. In the case of q_s=constant, the rate of heat transfer can also be expressed as;

$$\dot{Q} = q_s A_s = mc_p (T_e - T_i) \quad \text{(W)} \tag{13}$$

Then the mean fluid temperature at the tube exit becomes

$$T_e = T_i + \frac{q_s A_s}{mc_p} \tag{14}$$

Note that the mean fluid temperature increases linearly in the flow direction in the case of constant surface heat flux, since the surface area increases linearly in the flow direction (A_s is equal to the perimeter, which is constant, times the tube length) [4].

3. APPLICATION AND RESULTS

Underwater part of ship hull is always in contact with seawater. Because of sea is both a heat source and heat well, a certain heat load can be easily transferred to sea or from sea through ship hull. During the model setup it is important to use underwater part of ship hull to be able to obtain a permanent heat rejection capacity. Ships are floating at different drafts in loaded and unloaded situations and their wetted surface area change related to this. To use the hull surface more effective, designed system has been placed by groups. The model consist of $1^{1/2}$inch (DN 40) nominal diameter pipes, with same construction material of the ship, welded to ship hull from inside. Taking into consideration ship damages and corrosive effects, the situation that the fluid enter and exit to all pipes at the same time, also defined as parallel working principle, was accepted.

During the analyses, it was accepted that the ship inner surface did not make heat transfer with adiabatic conditions. The analyses were performed by using Finite Volume Method with turbulent flow conditions and k-ε turbulent model was selected.

The inlet condition of the water in pipes was entered as 65 C. Analyses were performed for different seaward water temperatures. Water in pipes, metal components and seawater were solved as integrated.

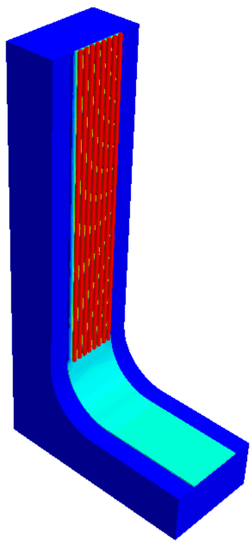

Figure 1. The position of the heat exchanger in ship mid-section

The heat exchanger components that welded to ship surface from inside were shown in Fig. 1 in mid-ship section to be clearly understood.

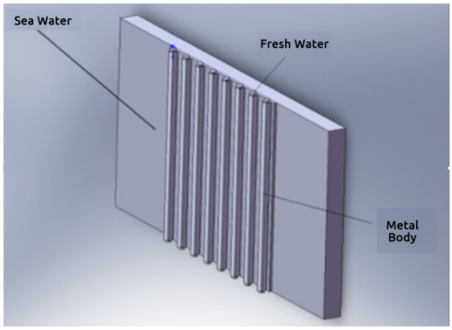

Figure 2. 3D numerical model) 1000mm x 1000mm

As shown in Figure 2, fresh water, sea water and metal surface was modeled discretely interpenetrating. Heat transfer between these three components was aimed to be solved with CFD analyses.

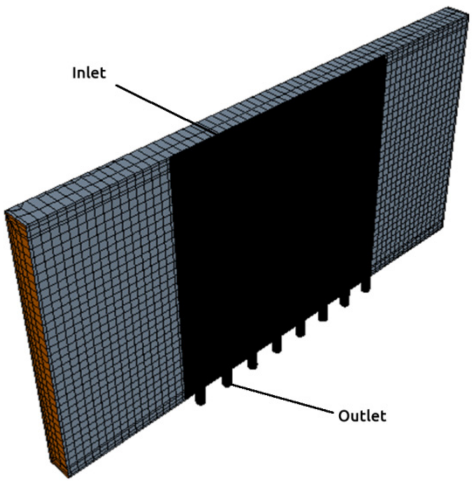

Figure 3. 3D model of computational domain

3D model of the computational domain is given in Figure 3. For each element a different mesh feature was defined. The control volume was created with adding virtual block geometries to where it was required more precise computational domain. The sub-features of the control volume, which were required to be interfered, was changed manually.

Table 1. Mesh independency cases

	Mesh Number
Case 1	3.139.000
Case 2	8.232.000
Case 3	17.349.000

The situations, which the solution is not change with the quality of mesh structure, are the cases that the mesh structure is converged and the solution is independent from the mesh structure. In table 1, mesh independency was provided and the variations of the model with mesh number between 3.139.000 - 17.349.000 cells were created. From these variations, it was found that the mesh independency was provided with 8.232.000 cells. It was also determined that the mesh quality should be higher at the places that the metal surface was close to the interfaces. Mesh quality of the seaward was decreased, by this means total mesh number was reduced and the performance of the analyses was increased.

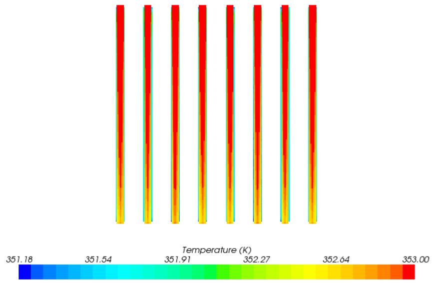

Figure 4. Temperature distribution at pipe inner cross-section

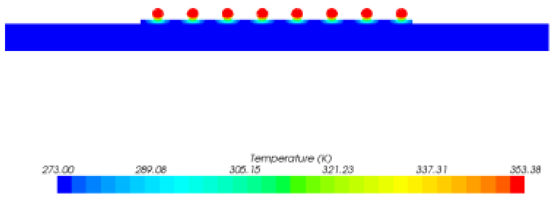

Figure 5. Temperature distribution at pipe upper cross-section

Temperature gradients on the plate at seawater side were also give information about heat transfer. Formation a straight line of pipe welding points caused to occur the temperature gradient at the highest point along the pipe welding lines.

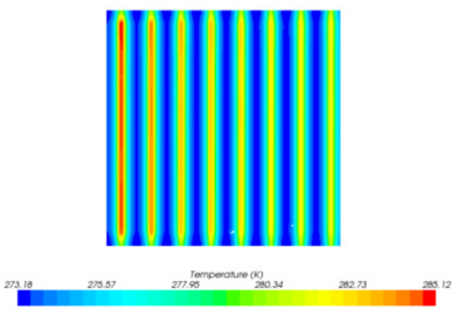

Figure 6. Temperature distribution of the plate at seawater side

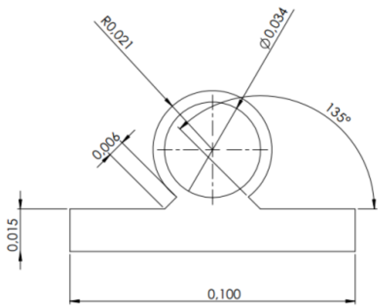

Figure 7. 2D sketch representing the pipe and welding seam

The pipe welded to the shipboard (Figure 7) was analyzed with SPH method and result were compiled. Geometry was represented with 2376 particles and h was accepted as $1.6*10^{-3}$. Material properties were entered and α_D was obtained. By this means Δt became to be obtained. It could be realized after finding m, mass of each particle, with dividing the footprint of geometry to particle number. Thus, temperature of all the particles can be found at each time step with a computer code [6,7].

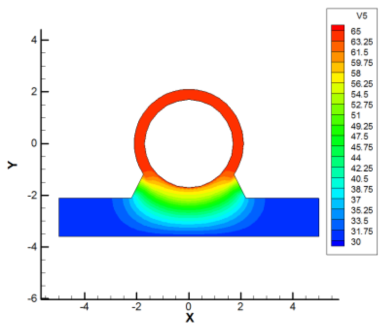

Figure 8. Time independent stable results for board and pipe that remain between internal and external flows

On a cruising ship, the jacket water temperatur e is not required to exceed 65 C. Therefore the flow of the cooling water in pipes was accepted as a constant temperature (65 C). The temperature distribution on shipboard and pipes, which remain between the flow passing through the pipe and outside seawater (30 C), has become time independent after a period of time. Accordingly, for the problem that was solved with computer code using SPH method, the results would be time independent and stable if the time step was kept sufficiently wide. Obtained time independent and stable result is given in Figure 8.

As shown in Figure 8, the pipe itself was under the influence of internal flow and got the temperature of pipe flow. The board side of the geometry was generally under the influence of external flow and results could be seen about 30 C. In the region, which the pipe and shipboard was welded each other, the effects of internal and external flows were mixed. In this region, the temperature was decreased as so drawing a distribution from inside to outside

It is not an analytical solution that directly identifies the existing geometry, so an approach was made by taking into consideration the pipes and plates.

This approach was progressed with accepting the surface length, which was used to transfer the heat taken from the pipe that the hot fluid flow to plate, as welding contact length. It was based on creating a pipe with accurate thickness by accepting inner wall length of an imaginary pipe section equal to contact point length (R_{inner}=X) and the length of plate that the heat was rejected to sea equal to outer wall length of an imaginary pipe section (R_{outer}=Y). If this acceptance made, the complex geometry turned into a pipe problem that hot water flows through and cold seawater flows outside. In this case, the solution could be achieved easily with pipe flow formulas.

Both in numerical and analytical solutions 3 different ship speeds and 9 different seawater temperatures were used. The effect of turbulence was increased with the increase of Reynolds number due to increase in ship speed. This effect also increased the convective heat transfer.

Observing the temperature values obtained from SPH solutions the contact point temperature between pipe and plate was calculated as 47.5° C. The same contact point temperature was found 48.23° C with CFD analyses. Deviation between CFD and SPH analyses was determined as 1.5%.

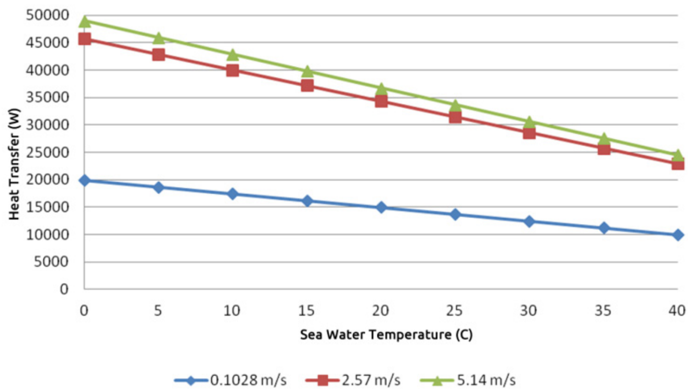

Figure 9. Heat transfer values obtained from numerical solutions for different seawater speeds and temperatures

The amount of heat transferred through shipboard for 3 different ship speeds and different seawater temperatures obtained from CFD analyses is given in Figure 9.

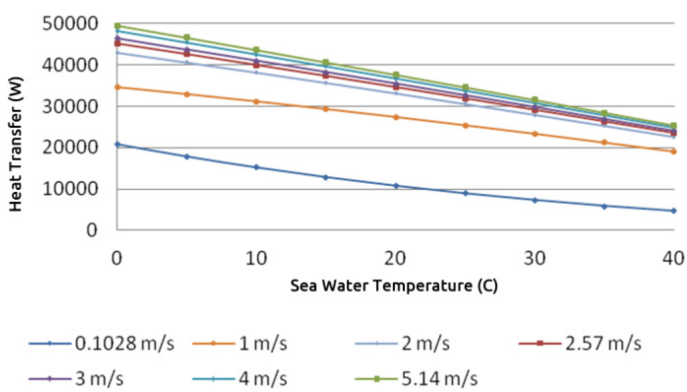

Figure 10. Heat transfer values obtained from analytical solutions

The amount of heat transferred through shipboard for 7 different ship speeds and different seawater temperatures obtained from analytical method, the details was given in previous sections, is given in Figure 10.

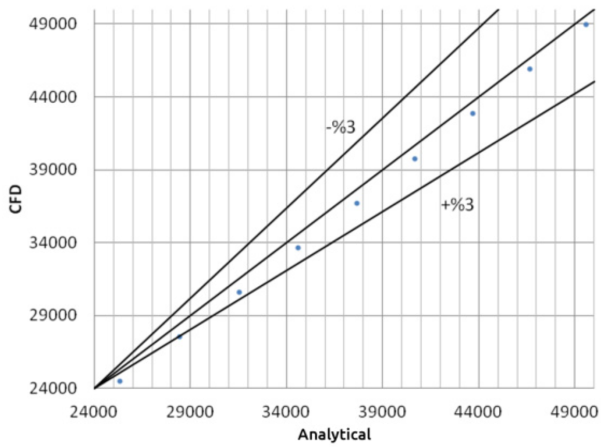

Figure 11. Analytical pipe expansion approach and weight distribution of numerical solution

In Figure 11, it is stated that the error distribution between analytical method and CFD is about 3%.

4. RESULTS AND SUGGESTIONS

Numerical and analytical analyses of the suggested system, which brings a new concept to jacket water cooling system used for main engine cooling, was completed as specified previous sections. In analyses, heat loads for poor operating conditions were found as 24479.05 W/m^2 for full speed, 22877.62 W/m^2 for half speed and 9958.67 W/m^2 for 0.1028 m/s (dead slow). It was selected an eight-cylinder MAN B&W S50ME-B8 main engine with total power of 13280 kW for an averaged cargo ship with 140m overall length, and the requirement of main engine jacket water cooling was determined as 2100 kW. According to suggested cooling model, an area of 210 m^2 was needed for poor operating conditions. With 4m length pipes, a distance of 52.5m in the direction of ship length was needed and this might be provided with considered sample ship.

To calculate the contribution of the viscosity change, as a result of increasing the temperature of sea film layer on ship outer surface, to ship resistance will reveal the significant increase in total benefit of system.

The solution of the model with unchanging fluid phase can be established with fluid phase change like classical cooling cycle. In this case heat exchange surfaces required for cooling systems can be supplied by model established central heat exchanger duty. In further studies, weight and volume calculations have importance to complete the suggested concept. Furthermore, a thermo-economic analysis should be carried out by means of comparing installation and operation costs of existing cooling system and suggested model.

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