Cargo Tank Heating Using Vertically Arranged Heating Coils
Grijanje tankova tereta vertikalno raspoređenim ogrjevnim cijevima

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Abstract

This paper proposes and analyzes a novel heating coil design for cargo oil tanks, characterized by a series of heating coil bundles asymmetrically extended along the middle part of the tank. Each heating coil bundle is a single heating body comprising several horizontal heating tubes, stacked vertically. During heating, each bundle generates a large-scale circulation that cross-flows through the empty spaces between the tubes; thus upgrading a heat transfer mechanism from less effective natural convection to more effective mixed convection. The proposed heating coil design is numerically analyzed with *OpenFOAM* software using a finite volume method. Simulated results indicate that for fuel oil with 585 mm^2/s viscosity heated by 50A-size steel coils driven by 8 bar steam, a 66,9% increase in the heat transfer coefficient may be achieved.
Keywords: circulation, coil, heating, tank, mixed convection

Sažetak

U radu je predložena i analizirana nova konstrukcija ogrjevnih cijevi za grijanje tekućih tereta na tankerima. Značajka predložene konstrukcije je niz snopova ogrjevnih cijevi asimetrično položenih duž središnjega dijela tanka. Svaki snop ogrjevnih cijevi jedinstveno je ogrjevno tijelo sazdano od niza vertikalno raspoređenih horizontalnih ogrjevnih cijevi. Tijekom grijanja snop ogrjevnih cijevi stvara cirkulaciju grijanoga tereta koja poprečno struji kroz praznine između ogrjevnih cijevi, potičući promjenu mehanizma prijenosa topline od manje učinkovite prirodne konvekcije do učinkovitije miješane konvekcije. Predložena konstrukcija ogrjevnih cijevi analizirana je metodom konačnih volumena primjenom programa *OpenFOAM*. Rezultati simulacije ukazuju da se u slučaju grijanja teškoga ulja viskoznosti 585 mm^2/s, čeličnim ogrjevnim cijevima veličine 50A, kroz koje struji vodena para tlaka 8 bara, može ostvariti porast koeficijenta prijelaza topline od 66,9%.
Ključne riječi: cirkulacija, grijanje, miješana konvekcija, ogrjevna cijev, tank

1. Introduction

The first modern oil tanker, with the tanks carrying cargo oil, was built in the late nineteenth century. The systems for heating cargo oil, despite great progress in design and manufacturing, have remained basically unchanged in design since then: a grid of steam-driven heating coils evenly distributed close to the bottom of the tank [1].

Numerical simulation of heat transfer within a tank was introduced in the work of Akagi et al. [2, 3], which used a finite differences method to simulate thermal and fluid flow processes within a two-dimensional rectangular cavity under different wall temperatures. Also, in [4], a numerical simulation of a single heating tube within a small test tank used a finite volume method to assess the coil's heat transfer coefficient and the distribution of oil temperatures within a tank. More recently [5], the results of a simulation of actual tanker heating were published, as performed by commercial computational fluid dynamics (CFD) software.

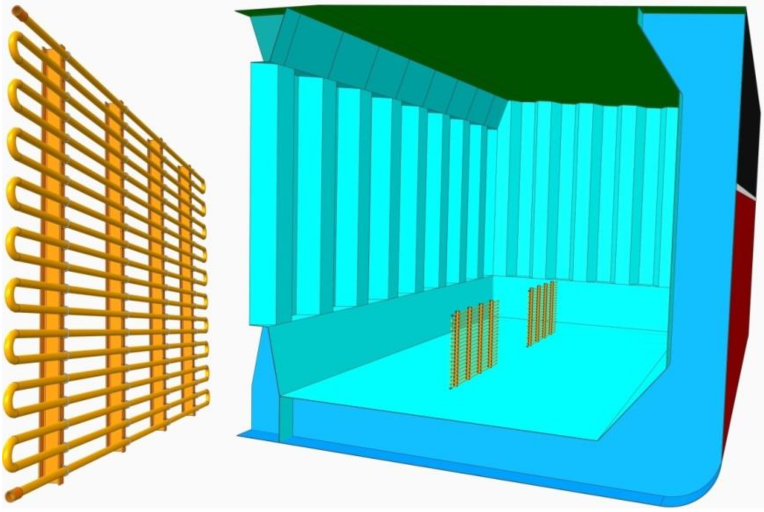

Fig. 43 Proposed heating coil bundle (left) and their placement within a cargo tank (right)
Slika 1. Predloženi snop ogrjevnih cijevi (lijevo) i smještaj unutar tanka (desno)

The present paper proposes and analyzes a novel heating coil design for cargo oil tanks. The vertical heating coils (Fig. 1) are asymmetrically extended in a row along the middle part of the tank, at the bottom. Each heating coil bundle is a single heating body comprising several horizontal heating tubes, stacked vertically. During heating, the coil bundle generates large-scale circulation through the heated cargo [6] that flows across the empty spaces between the tubes. This changes the heat transfer mechanism from less effective natural convection to more effective mixed convection.

2. Problem statement

The dimensions of a real tanker's cargo hold are taken as a basis for numerical simulation. A 7360 m^3 tank has 17,5 m breadth, 16,8 m height, 26,4 m length, and a 15,4 m filling level [5]. The tank is filled with Bunker C heavy fuel oil at a 55ºC temperature and with 585 mm^2/s nominal kinematic viscosity. The environment is defined by 2ºC air and 5ºC sea temperatures (Fig. 2).

A conventional heating coil, arranged as a horizontal grid of 24 50A-sized mild steel heating tubes driven by 8 bar steam, was chosen as a baseline design. As documented in a paper submitted for publication elsewhere, the average heat transfer coefficient of this conventional heating coil is estimated as 120,2 W/m^2K.

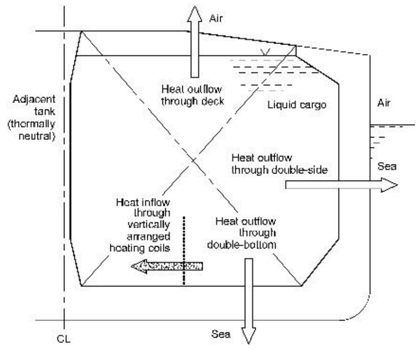

Fig. 2 Problem domain and main processes
Slika 2. Domena zadatka i glavni procesi

The present study investigates the possibility of replacing the conventional heating coils with a series of vertical heating coil bundles (as in Fig. 1), each comprising 20 horizontal, mild steel heating tubes of 50A size, with 180 mm transverse pitch between heating tubes. If six-meter-long heating tubes are used to construct the bundles, the bundle's overall tube length totals 126 m. The bundles are situated asymmetrically within the tank, 7,59 m off the tanker's longitudinal plane of symmetry. The heating coils are driven by 8 bar steam, the same as in the baseline case.

The following describes the setup of the simulation and the corresponding results.

3. Numerical simulation

In evaluating the proposed arrangement of heating coils, three-hour tank heating was simulated. Simulating longer heating would incur prohibitively high computational costs. For the same reason, the problem domain was treated as two-dimensional, even though a real tank is three-dimensional.

Transient simulation of laminar fluid flow and heat transfer was performed with an *OpenFOAM* [7] CFD tool that utilizes the finite volume method [8]. A set of governing equations, comprising Eq. (1) for mass conservation, Eq. (2) for momentum conservation, and Eq. (3) for energy conservation, must be fulfilled for any part of the simulation domain and any time step during a time-marching procedure:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{U}) = 0, \tag{1}$$

$$\frac{\partial (\rho \mathbf{U})}{\partial t} + \nabla \cdot (\rho \mathbf{U}\mathbf{U}) = \nabla \cdot \left\{ \mu_{\text{eff}} \left[\nabla \mathbf{U} + (\nabla \mathbf{U})^T \right] \right\} - \nabla \left[\frac{2}{3} \mu_{\text{eff}} (\nabla \cdot \mathbf{U}) \right] - \nabla p + \rho \mathbf{g}, \tag{2}$$

$$\frac{\partial (\rho h)}{\partial t} + \nabla \cdot (\rho \mathbf{U} h) + \frac{\partial (\rho K)}{\partial t} + \nabla \cdot (\rho \mathbf{U} K) - \frac{\partial p}{\partial t} = \nabla \cdot (\alpha_{\text{eff}} \nabla h) + \rho \mathbf{U} \cdot \mathbf{g}, \tag{3}$$

where ρ is density, t is time, $\mathbf{U}$ is the velocity vector, μ_{eff} is the effective viscosity, p is pressure, $\mathbf{g}$ is the gravitational acceleration vector, h is the enthalpy, K is the specific kinetic energy, and α_{eff} is the effective thermal diffusivity.

The thermophysical properties of the cargo oil are provided in Table 1.

Table 29 Cargo properties at 55ºC and 157ºC
Tablica 1. Svojstva tereta pri 55ºC i 157ºC

Temperatute, ºC	55	157	Model
Density, kg/m^3	964	893	linear
Dynamic viscosity, Pa·s	0,405	0,0085	polynomial
Thermal conductivity, W/m·K	0,130	0,106	linear
Specific heat capacity, J/kgK	1870	2260	linear

A spatial domain filled with liquid cargo (Fig. 2) was discretized into roughly 187.000 mostly hexahedral cells, with the outer boundary sliced into four boundary layers and the heating coil boundaries meshed into 29 layers, 120 cells per layer, with the first layer thickness measuring only 0,05 mm.

The boundary conditions were set as follows. The top of the domain is exposed to zero total pressure, while the remaining wall boundaries are subject to the hydrostatic pressure. Similarly, the fluid velocity at the top of the domain is bounded by conditions of free atmospheric flow, while the solid wall boundaries are constrained by a magnitude of zero velocity. Temperature boundary conditions for all outer walls, including the tank top but excluding the thermally neutral longitudinal bulkhead, were determined by convective heat outflows. The tank bottom and top overall heat transfer coefficients were selected as 3,2 W/m^{2}K, whereas a value of 3,3 W/m^{2}K was used for the outer tank wall. To model 8 bar saturated steam, a 170,4ºC heating fluid was used as the energy source for heating the cargo. This study assumes a maximum coil surface temperature of 165ºC, with a 15ºC difference between the upper and lower coil surfaces, as was measured in [9]. Consequently, coil surface temperature is linearly distributed from the top (165ºC) to the bottom (150ºC), with a mean of 157,5ºC.

A *buoyantPimpleFoam* transient solver for compressible fluids was used for numerical simulation, with the following settings. Euler method time-marching was limited by a maximum Courant number of 0,5. Each time step assumes at most five Pimple (Piso and Simple combined) iterations, with each iteration comprising two corrector and two non-orthogonal corrector steps. A preconditioned conjugate gradient (PCG) pressure solver, enhanced with a diagonal-based incomplete Cholesky (DIC) preconditioner, uses a tolerance of 10^{-8}, while the remaining solvers use a diagonal-based incomplete lower and upper (DILU) preconditioned bi-conjugate gradient (PBiCG) algorithm and a tolerance of 10^{-6}. The time-step convergence criteria were 10^{-4}, and the conventional under-relaxation factors were chosen of 0,3 for pressure and 0,7 for enthalpy and velocity. Applied numerical discretization schemes comprised a Gauss linear corrected scheme for the diffusion term and Gauss vanLeer and Gaus vanLeerV schemes for the advection terms. The heat fluxes at the boundaries were determined using the *wallHeatFlux* utility. The simulation took 314 CPU hours using six cores of a server with a 28-core, dual Intel Xeon E5-2680v4 processor.

4. Results

Simulating three hours, the complete domain state vectors were updated with the most current results after every five seconds of heating. After simulation, those values were collected for further processing. The most important results are presented in Figs. 3 to 8.

Figure 3 shows the cargo velocity distribution six minutes after heating has started. In the upper-half of the tank, a wide plume has developed, while the lower-half of the tank remains nearly unaffected. Over the next 39 minutes, this plume further expands, gradually transforming into a single circulation at the size of the tank, Fig. 4. These figures depict two characteristic phases of tank heating. During the first phase, heated cargo flows out from the heating coils vertically (Fig. 3), while, during the second phase, heated cargo flows across the bundle in a nearly horizontal direction (Figs. 4 and 5).

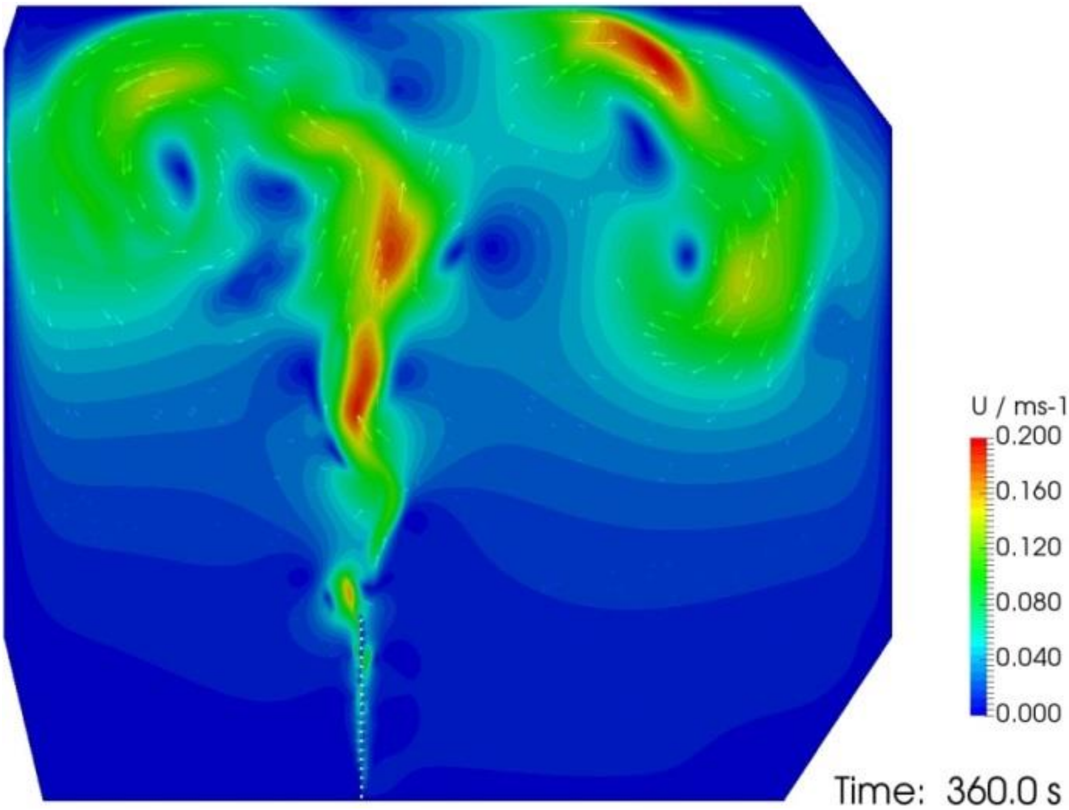

Fig. 3 Cargo velocity distribution at t = 360 s (pre-circulation phase)
Slika 3. Razdioba brzina tereta pri t = 360 s (predcirkulacijska faza)

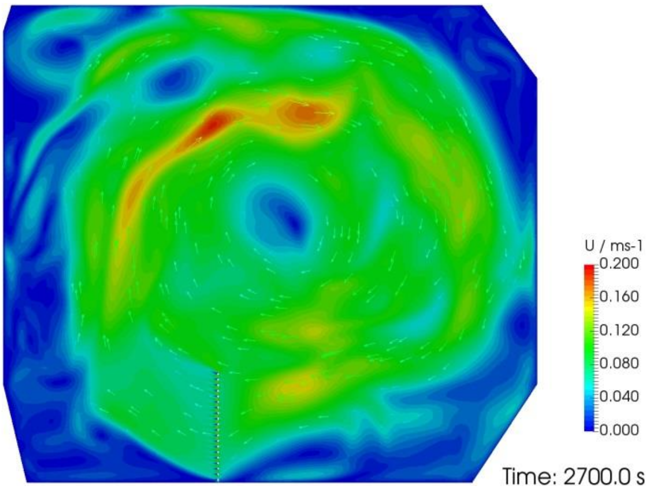

Fig. 4 Cargo velocity distribution at t = 2700 s (circulation phase)
Slika 4. Razdioba brzina tereta pri t = 2700 s (cirkulacijska faza)

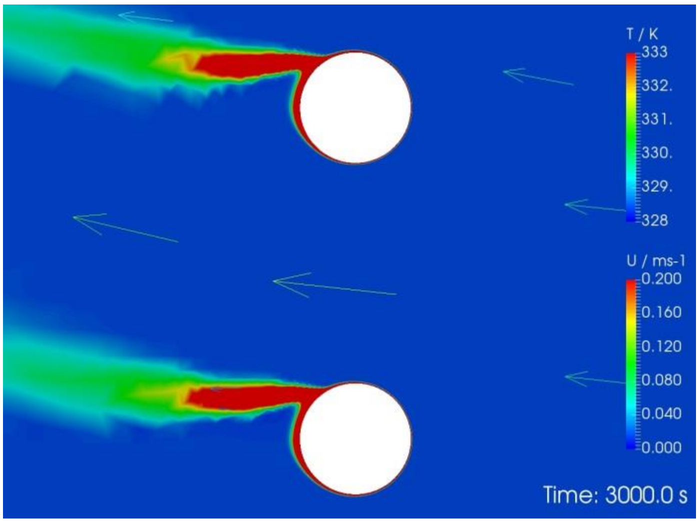

Fig. 5 Cargo temperature and velocity distribution at t = 3000 s (circulation phase)
Slika 5. Razdioba temperatura i brzina tereta pri t = 3000 s (cirkulacijska faza)

4.1. Coil surface heat transfer coefficient

The heat transfer coefficient of the coil surface is determined based on the heat flux data generated by the *wallHeatFlux* utility, according to the equation:

$$h_{\mathrm{c}} = \frac{q_{\mathrm{c}}}{T_{\mathrm{c}} - T_{\mathrm{b}}}, \tag{4}$$

where q_{c} is the sum of the heat fluxes generated at the heating tubes, T_{c} is the temperature at the coil surface, and T_{b} is the temperature of the bulk cargo.

Variation in the coil heat transfer coefficient over time, area-averaged over all heating tubes, is shown in Fig. 6. Due to the unsteady nature of the fluid flow, the surface heat transfer coefficient fluctuates within a range that also changes over time, with the flow structure within the tank. Therefore, the heat transfer coefficient h_{c} must be time-averaged to determine the average heat transfer coefficient $\bar{h}_{\mathrm{c}}$.

Neglecting the results for the first 2700 s of heating, the time-averaged surface heat transfer coefficient over the remaining 8100 s is $\bar{h}_{\mathrm{c}}$ = 200,6 W/m^{2}K. This value is 66,9% higher than the heat transfer coefficient of the conventional coil.

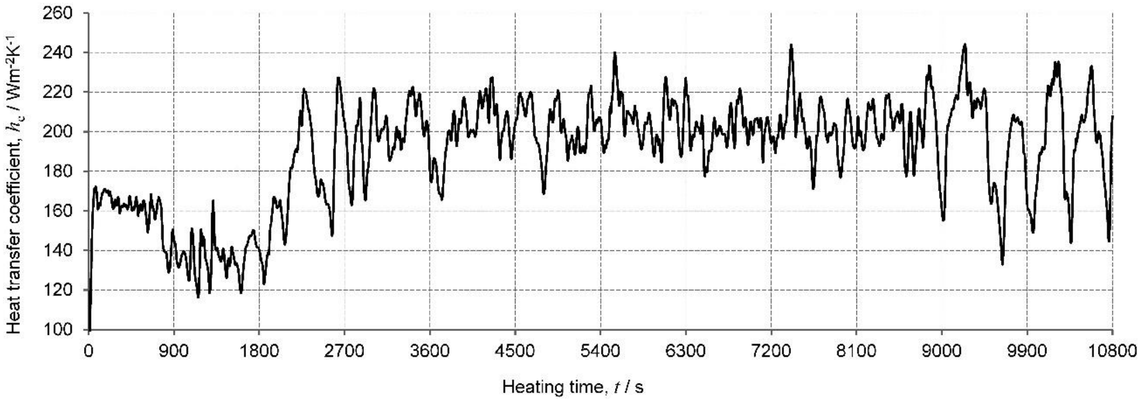

Fig. 6 Average surface heat transfer coefficient
Slika 6. Prosječni koeficijent prijelaza topline

4.2. Cargo temperature

Concerning cargo temperature, two distinct quantities have special importance: (1) bulk cargo temperature and (2) the temperature at the bottom of the tank. Both are shown in Fig. 7.

Over three hours of heating, the temperature 25 mm above the tank bottom rose from an initial 55ºC to a final 56,4ºC. This temperature indicates the temperature of fluid at the level of cargo pump suction. However, it is also an indicator of bulk cargo temperature, since these two quantities differ by less than ±0,1ºC.

Cargo temperature at the bottom of the tank is an important indicator for assessing the probability of cargo component solidification. Higher temperatures raise thermal losses but lower the solidification probability and vice versa. In the present study, after three hours of heating, the simulated cargo temperature at the bottom of the tank fell to 32,8ºC.

Simulated thermal losses from the bottom of the tank equaled 1,34 kW per meter of tank length. The corresponding thermal losses at the outer side wall and top of the tank were 2,18 kW and 2,24 kW, respectively. The remaining tank walls are thermally neutral, thus incurring zero losses.

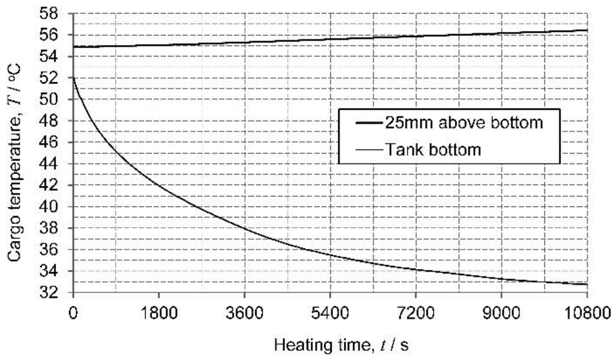

Fig. 7 Average cargo temperature
Slika 7. Srednja temperatura tereta

4.3. Cross-flow velocity

The change in the heat transfer mechanism may be indirectly observed by tracking the cross-flow velocity of cargo between the horizontal heating tubes (Fig. 8). During the pre-circulation phase, the cargo velocity is mostly directed upwards, parallel to the plane comprising the heating tubes, with an average magnitude of around 0.05 m/s. After establishing circulation, the cargo velocity is mostly directed perpendicular to the plane comprising the heating tubes, with cross-flow velocity averaging significantly higher magnitude (Table 2). Thus, the rising cross-flow velocity is the key feature responsible for the greater heat flux.

Table 2 Heating coil bundle, cross-flow velocity
Tablica 2. Prolazna brzina kroz snop ogrjevnih cijevi

Point height, mm	Peak, m/s	Average, m/s
200	0,112	0,070
2000	0,195	0,119
3800(*)	0,201	0,116

(*)The point is 280 mm above the uppermost tube

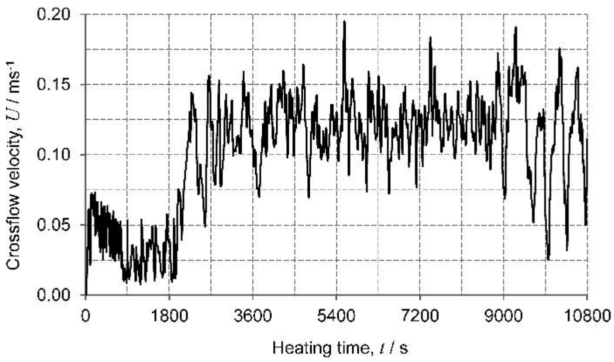

Fig. 8 Peak cross-flow velocity at 2000 mm height
Slika 8. Vršna prolazna brzina na 2000 mm visine

5. Conclusions

This paper investigated a new design of heating coil characterized by the vertical arrangement of multiple horizontal heating tubes. Due to the asymmetric position of the heating coil bundle within the tank, a powerful circulation of heated cargo is generated that superimposes circulation-based forced convection on buoyancy-generated natural convection. The coil surface consequently achieves a 66,9% higher heat transfer coefficient.

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