USING STEAM COILS IN MARINE APPLICATIONS

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It is rather common practice for the marine industry to use water heating coils to heat heavy fuel oil. Obviously, the efficiency of the heat transfer process is strongly dependent on the dimensions of the coil (length, thickness, and diameter) as well as on the operational parameters (oil temperature, steam temperature, and steam pressure). In the present work, the heat transfer from superheated water at high temperature ($\sim$ 424 K) and fixed pressure (5 bar) to fuel oil tanks of specific dimensions was theoretically investigated by both a macroscopic thermo-dynamical approach and microscopic simulations (using the commercial CFD-ACE$^+$ software package). Since the scope of this study was to estimate the necessary size and length under the assumption of an insulated tank, a parametric analysis was also performed in order to identify the relative influence of each parameter on the process performance.

KEY WORDS: *heavy fuel oil, thermodynamics, heat transfer, steam*

1. INTRODUCTION

Currently, numerous merchant ships still use low-round diesel engines that are fueled by relatively inexpensive heavy fuel oil (HFO). The efficiency of these marine diesel engines approaches 48%–51%, while a huge amount of heat is wasted, mainly through flue gases (Dzida, 2009). To prepare HFO for propulsion, an increment of its temperature is necessary in order to lower its viscosity and obtain a smooth and continuous flow. Hence, a large amount of heat is necessary to increase the tank temperature, which is frequently provided by superheated water (MAN Diesel & Turbo, 1985). This concept of heat transfer between a heated liquid flux and a colder volume of stationary HFO is mainly encountered in applications other than a ship's supply, such as oil drainage (Pang et al., 2017), recovery in wellbores (Cheng et al., 2017), etc. Since the amount of necessary heat is quite large, efficient energy waste management is often absolutely crucial.

Computational fluid dynamics (CFD) can be a very powerful tool toward this aim. CFD is actually the use of computer-based simulation to mathematically analyze complex systems involving fluid flow, heat transfer, and associated phenomena such as chemical reactions. The development and application of CFD has recently increased and become a powerful tool in the design and analysis stages of several engineering and industrial systems and processes (Rafiee and Sadeghiazad, 2017; Dbouk, 2017). On the other hand, the heat transfer process can be sufficiently described through normal thermodynamic analysis, at least from a macroscopic point of view.

The main scope of the present work is to identify the appropriate geometrical characteristics and the length of the carbon–steel tube (also known as a coil) through which the superheated water flows and exchanges heat with the HFO. This application has been studied from both thermodynamical and process/simulation points of view. Therefore, a thermodynamic analysis of heat exchange was performed based on heat balances throughout control volumes. In addition, detailed simulations were carried out, allowing for a deep understanding of the heat transfer process. The results of both approaches have been evaluated against the geometrical characteristics (coil length and diameter) and the operational parameters (tank insulation and steam pressure).

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NOMENCLATURE

A	area used in the integration of Eq. (5) (m^2)
[$A_1, \ldots, A_5$]	coefficients used in Eq. (4)
[$C_1, \ldots, C_4$]	coefficients used in Eq. (3)
C_P	isobaric specific heat capacity ($J/mol^{-1} \cdot K^{-1}$)
D	cylindrical diameter (mm)
H	enthalpy (J)
I	internal energy (J)
K	thermal conductivity ($W/m^{-1} \cdot K^{-1}$)
L	coil length (m)
m	mass (kg)
P	pressure (atm)
Q	thermal energy (J)
R	gas constant value equal to 8.1344 ($J/mol^{-1} \cdot K^{-1}$)
r	cylindrical radius (mm)
$\hat{r}$	radial coordinate
T	temperature (K)
t	time (hours)
$\vec{u}$	velocity (m/sec^{-1})

Greek Symbols

ΔT	temperature difference (K)
$\hat{\theta}$	angular coordinate
μ	dynamic viscosity ($kg/m^{-1} \cdot sec^{-1}$)
ν	kinematic viscosity (m^2/sec^{-1})
ρ	density ($kmol/m^{-3}$)

Subscripts

fin	final
in	inner
init	initial
oil	heavy fuel oil
out	outer
sup/ted water	fluid inside the tube

2. THE SYSTEM

Consider a tank, with dimensions of 3.2 m $\times$ 7.6 m $\times$ 12.56 m and volume of 306 m^3, filled up to 85% with heavy fuel oil. This settling tank receives the HFO from onshore sources at an average temperature of 303 K and has to heat it up to the average temperature of 333 K. Toward this aim, heating coils are used, where the provided superheated water is maintained at a temperature of 424 K and constant pressure of 5 bar. The coils are manufactured by a carbon–steel boiler tube, 50 mm in external diameter and 4 mm thick, having a serpentine shape, as depicted in Fig. 1. The main scope of the present study is to investigate the appropriate size and length of the heating coil over specific time periods.

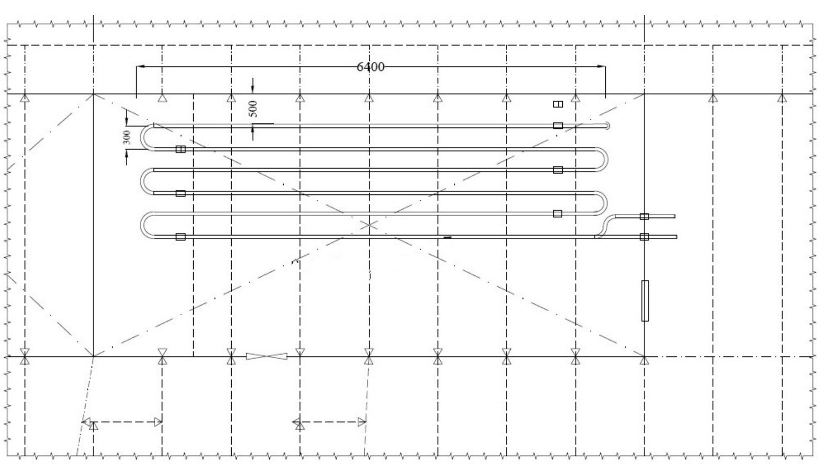

FIG. 1: Heating steam coils

3. THERMODYNAMICS OF HEATING

3.1 Theory

The thermodynamics of such a system are usually based on the energy (heat) balance between the heating fluid and the heated fluid, namely, between superheated water and HFO. If the only heat source is the heated steam, then by considering the total insulation of the apparatus the aforementioned balance is obtained as follows:

$$Q_{\mathrm{sup/ted\ water}} = Q_{\mathrm{oil}} \tag{1}$$

where Q denotes the thermal energy transferred from the coil to the HFO.

The total amount of necessary thermal energy that is appropriate to supply to the HFO, Q_{oil}, can be written as follows (Sato, 2004):

$$Q_{\mathrm{oil}} = m\left[C_P\left(T_{\mathrm{fin}}\right)T_{\mathrm{fin}} - C_P\left(T_{\mathrm{init}}\right)T_{\mathrm{init}}\right] \tag{2}$$

where C_P is the specific heat capacity, which can be calculated using the National Aeronautics and Space Administration (NASA) polynomials and the appropriate specific coefficients for each chemical element (McBride et al., 1993); and m is the total mass of the HFO (kg) calculated for a given tank volume through the temperature-dependent density, ρ, of the HFO, which is given as follows (Perry and Green, 1997):

$$\rho = \frac{\mathrm{C}_1}{\mathrm{C}_2^{\left\{1+\left[1-(T/\mathrm{C}_3)\right]^{\mathrm{C}_4}\right\}}} \tag{3}$$

where $\mathrm{C}_1 = 0.5373$, $\mathrm{C}_2 = 0.2612$, $\mathrm{C}_3 = 568.7$, and $\mathrm{C}_4 = 0.2803$ are the coefficients; T is the temperature (K); and ρ is the density (kmol/m^{-3}). Given the volume of the tank as well as the filling level, the initial total mass of HFO is 1.81×10^5 kg at a temperature equal to 303 K.

In Eq. (2), $C_P\left(T_{\mathrm{init}}\right)$ and $C_P\left(T_{\mathrm{fin}}\right)$ are the specific heat capacity values (J/mol$^{-1}\cdot$ K^{-1}) and T_{init} and T_{fin} are the temperatures (i.e., 303 and 333 K) at the initial and final stages of the heating process, respectively. The specific heat capacity for each chemical substance can be expressed through the NASA polynomial as follows (McBride et al., 1993):

$$\frac{C_P\left(T\right)}{R} = \mathrm{A}_1 + \mathrm{A}_2 T + \mathrm{A}_3 T^2 + \mathrm{A}_4 T^3 + \mathrm{A}_5 T^4 \tag{4}$$

where $R = 8.3144$ is the gas constant value (J mol^{-1} K^{-1}); $C_P\left(T\right)$ is the calculated specific heat capacity (J/mol$^{-1}\cdot$ K^{-1}); and A_1–A_5 are the coefficients specific to each chemical substance. For HFO, these coefficients are the following (Perry and Green, 1997): $\mathrm{A}_1 = 1.25 \times 10^1$, $\mathrm{A}_2 = -1.01 \times 10^{-2}$, $\mathrm{A}_3 = 2.22 \times 10^{-4}$, $\mathrm{A}_4 = -2.85 \times 10^{-7}$, and $\mathrm{A}_5 = 1.12 \times 10^{-10}$.

The heat transfer takes place between the carbon–steel coil and the HFO and can be described through the integral form of Fourier's law (Holman, 1990):

$$\frac{dQ}{dt} = -k\int_A \vec{\nabla} T\, dA \tag{5}$$

where dQ/dt is the amount of heat transferred per unit time (J/sec^{-1}); k is the thermal conductivity of the carbon–steel coil (W/m$^{-1}\cdot$ K^{-1}); and $\vec{\nabla} T$ is the temperature gradient over the radial dimension of the system, where the integral is closed over the specific surface A of the cylindrical hot coil through which the heat transfer occurs.

By considering the aforementioned parameters, geometry Eq. (5) can be written as follows (Bejan, 1993):

$$\frac{dQ}{dt} = -\frac{2\pi L k\left(T_{\mathrm{out}} - T_{\mathrm{in}}\right)}{\ln\left(r_{\mathrm{out}}/r_{\mathrm{in}}\right)} \tag{6}$$

where L is the necessary length of the coil; T_{out} and T_{in} are the temperatures on the outer and inner cylindrical surfaces, respectively; and r_{out} and r_{in} are the outer and inner radii of the coil (thickness), where the necessary length is given as follows:

$$L = \frac{Q_{\mathrm{oil}}}{\left(dQ/dt\right)t} \tag{7}$$

where t is the time at which the heat transfer process occurs (hours).

3.2 Thermodynamics Results

Figure 2 clarifies the influence of time on the coil length. As expected, the shorter the time at which heat transfer occurs, the longer is the coil in order to assure the appropriate transfer surface. This behavior is not linear since the amounts of heat that must be transferred to the oil are not linear over time. In particular, the heat transfer substantially occurs in two dimensions (cylindrical coordinates: $\hat{r}$ and $\hat{\theta}$), where the radial component is the only important parameter, developing on several imaginary coaxial isothermal cylindrical surfaces through the total mass of the HFO in the banker tank. Over long time periods, the temperature of the oil increases to such a level that the gradient is low enough for considerable heat fluxes. Furthermore, given a constant thickness of the steel tube ($d = 4$ mm), the diameter is a favorable parameter for heat transfer, i.e., larger diameters assure higher available surfaces for heat transfer. It is important to underline the restrictions on the coil's length, which are imposed by the tank's dimensions: the area covered by the coil's serpentine shape should not exceed 95.46 m^2, which corresponds to a maximum length of approximately 154 m, depending on the frame space (see Fig. 1).

The importance of the geometrical characteristics of the coil is further depicted in Fig. 3, where the same behavior of the necessary length over time is also observed. What is important here is that the length increases with the thickness for a given constant diameter (here, it is 50 mm) because of the consequent increment of the heat capacity of the steam coil: by increasing the thickness, the steel mass is also increased and so is the amount of energy that is wasted to heat the solid steel.

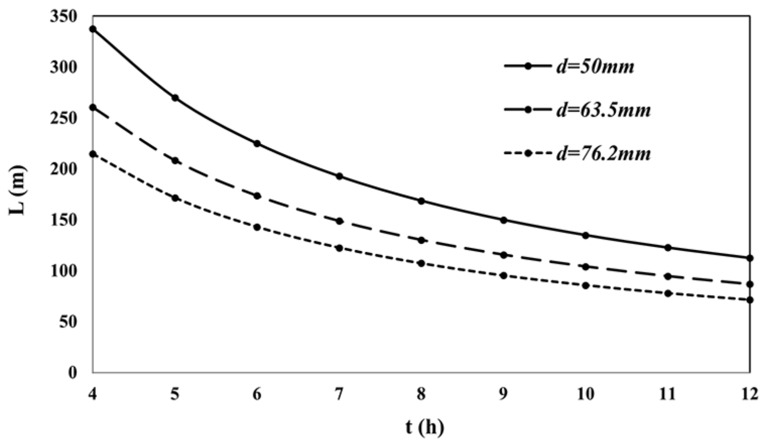

FIG. 2: Coil length as a function of time

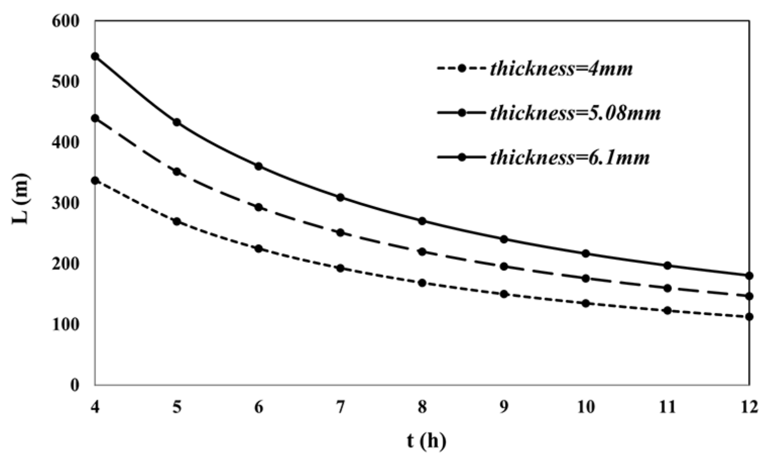

FIG. 3: Effect of the coil's thickness on its length

The results, as depicted in Figs. 2 and 3, were obtained by considering a constant temperature difference equal to to ~ 2 K between the inner and outer surfaces of the steam coil (in contact with the HFO). Figure 4 further investigates the effect of the temperature difference between the flowing superheated water inside the carbon–steel tube and its outer surface on the heat transfer for various periods of time, where it was considered that the inlet temperature was 424 K (superheated water). It was found that the necessary length was lower for the higher temperature differences because the latter corresponds to higher amounts of heat that are actually transferred to the HFO in the tank.

As Figs. 2–4 indicate, the dependence of the length on time is not linear due to the character of the heat transfer process that occurs. To be more precise, there is only one heat source (coils), which has to offer heat in order to: (a) initially increase the temperature of the coil itself and then keep this temperature constant, and (b) increase the HFO's temperature up to the desirable level. Since the relationship of these two heat consumptions is not linear, the length of the coil has to follow a nonlinear time evolution in order to satisfy this requirement.

4. MICROSCOPIC SIMULATIONS

4.1 Heat Transfer in Detail

The fundamental transport phenomena occurring in the system under consideration are the superheated water flow and the heat transfer from the coil to the HFO. Since laminar conditions are considered, the flow can be well described by the Navier–Stokes equations, which for incompressible fluids are given as follows:

$$\frac{\partial \vec{u}}{\partial t} + \left(\vec{u} \cdot \vec{\nabla}\right) \vec{u} = -\frac{1}{\rho} \vec{\nabla} P + \nu \left(\nabla^2 \cdot \vec{u}\right) \tag{8}$$

where $\vec{u}$ is the velocity vector; ρ is the density; P is the pressure; and ν is the kinematic viscosity. The previous equation has to be considered along with the following continuity equation:

$$\frac{\partial \rho}{\partial t} + \vec{\nabla} \cdot (\rho \vec{u}) = 0 \tag{9}$$

By neglecting radiation, the energy balance in the system can be written as follows:

$$\frac{\partial}{\partial t} (\rho T) + \vec{\nabla} \cdot (\rho \vec{u} H) \vec{u} = \vec{\nabla} \cdot \left(k \vec{\nabla} T\right) + \frac{\partial P}{\partial t} \tag{10}$$

where T is the temperature; k is the thermal conductivity; and H is the total enthalpy of the fluid, which is given as follows:

$$H = i + \frac{P}{\rho} + \|\vec{u}\|^2 \tag{11}$$

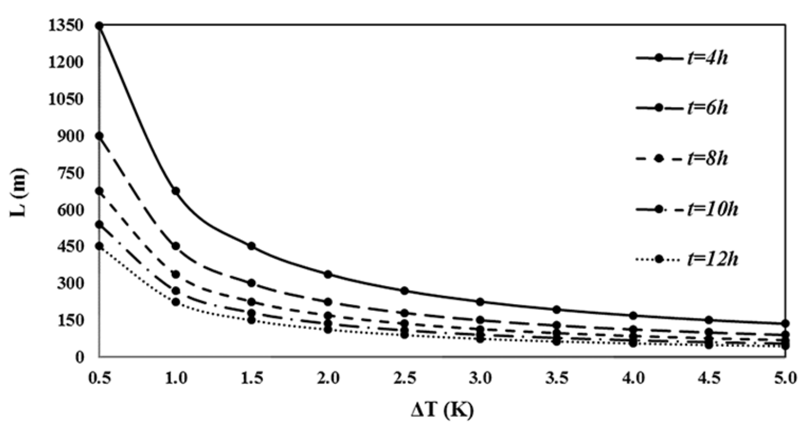

FIG. 4: Length as a function of ΔT

where i is the internal energy as a function of state variables ρ and T, and $\|\vec{u}\|$ denotes the Euclidean norm of the velocity vector.

The boundary conditions in the problem were as follows: the flow rate in the steam coil was set to 0.024 m/s^{-1}, while the pressure was set accordingly at the inlet and outlet in order to assure that the fluid was in a liquid phase in the coil tube (actually, $0.002 < P < 0.035$). Also, fixed heat flux ($0.15 < J_T < 0.33$) and adiabatic conditions were set for the coil and tank boundaries, respectively, in each time step. Finally, a no-slip condition ($u = 0$) was assumed for the steam flow on inner tube surfaces.

The previous transient equations were considered to be strongly coupled; therefore, numerical solutions were obtained using the commercial CFD-ACE$^+$ package, which is based on the finite-volume method, in order to achieve residual values of less than 10^{-4} for all calculated quantities. A three-dimensional (3D) tank with a coil was discretized in space by an unstructured grid consisting of approximately 2–5.2 $\times$ 10^5 cells, where the amount depended on the coil length. The values of the parameters used as well as the properties of the materials are listed in Table 1.

To exclude the influence of the grid size, a full grid dependency test was carried out as follows: the spatial discretization (grid size) was increased by a factor of 0.1 in all dimensions until the main outcome, which was defined as the average temperature alongside all of the steam coils, was constant up to six significant digits. Regarding time averaging, the results were found to be independent of the selected dt value since different time intervals always corresponded to the same steady-state results, which were assumed to be constant in all of the other remaining physical and numerical parameters.

4.2 CFD-ACE$^+$ Simulations

Typical temperature spatial distributions are presented in Figs. 5(a) and 5(b) for a perpendicular two-dimensional cut and the footprint of the banker tank, respectively. The coil length was 140 m and the time passed was 12.5 hours, which was the maximum sufficient time for successful integration of the heating process for this specific layout. The black line in Fig. 5(a) indicates the level of the HFO, which shows that the majority of the volume occupied by the HFO is indeed heated to 333 K. Areas of lower temperature (seventh wavy line and fifth wavy line) are attributed to isothermal conditions ($T = 303$ K), and are presented at less than 1 m length from the walls of the tank and less than 0.30 m from the surface of the HFO. These areas correspond to 20% of the total volume of the HFO for a 140-m-long tube and obviously can be minimized up to 14% by considering a longer layout coil ($\sim$ 160 m), as will be discussed subsequently. Also, by duplicating the established overall coil length ($\sim$ 300 m)—i.e., two horizontal rows, 150 m in length—this percentage can reach 2%–3% over a 5-hour time interval. During real life heating processes, HFO presents better homogeneity than this due to physical stirring at its entry point in the banker tank.

In order to quantify the HFO's heating process, the oil temperature was spatially averaged for various coil lengths, as presented in Fig. 6. These microscopic results indicated that the tank approached the desirable temperature of 333 K in the period of the 14.5-hour time limit for the shortest coil ($\sim$ 120 m). Considering longer coils, the time period can been calculated to be approximately between 12.5 and 10 hours, a result that is in an acceptable agreement with thermodynamic analysis. This approach allows for a more detailed calculation of the temperature profile, thus some areas of significantly lower temperature can be recognized in the tank's volume. Furthermore, these results are

TABLE 1: Parameters and properties used based on data from Wagner and Kretzschmar (1998)

Parameter	Liquid Phase		Carbon–Steel (Solid Phase)
	Superheated Water	Heavy Fuel Oil	
ρ (kg m^{-3})	917.02	695.23	7850
μ (kg/m$^{-1}\cdot$ sec^{-1})	1.80×10^{-4}	1.02×10^{-3} (at T_{env})	—
C_p (J/kg$^{-1}\cdot$ K^{-1})	2.41×10^3	1.67×10^3 (at T_{in})*; 1.80×10^3 (at T_{fin})†	470
k (W/m$^{-1}\cdot$ K^{-1})	6.82×10^{-1}	1.44E-01	48

†Through NASA polynomials (McBride et al., 1993).

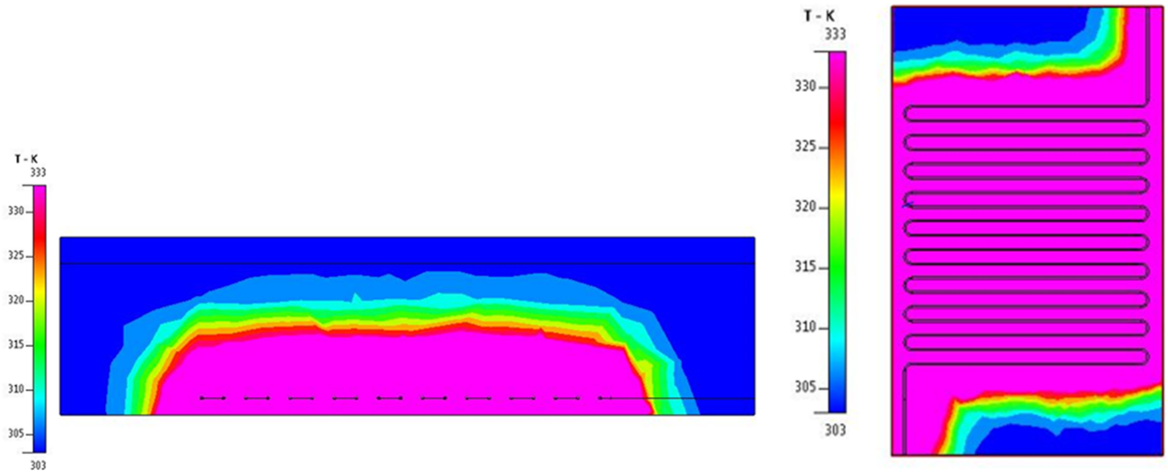

FIG. 5: Typical microscopic results: (a) perpendicular cut; (b) footprint

consistent with those shown in Figs. 2 and 3: the longer the coil, the shorter is the time needed for the oil to reach the desirable temperature of 333 K.

Also, by keeping as constant the overall length of the carbon–steel tube ($\sim$ 160 m)—the longer coil in the aformentioned scenarios as depicted in Fig. 6—two more different layout scenarios were considered in the establishment of the coil (one established at the half height of the tank and one installed in two different vertical rows, 80 m each, in the middle) in order to investigate the effect of the different placements over the time intervals during the heating process. More precisely, Fig. 7 reveals that the change in the coil's placement does not significantly influence the results on the time interval of 10 hours, which is essential for the fuel oil to reach the desirable temperature of 333 K.

It should be noted that by considering two vertical rows with a total length of 160 m the HFO slightly increases its temperature in a shorter period of time compared to the other layouts at the beginning of the process; however, at the end of the process the temperature drops and the estimated time increases by about 1.5 hours, which constitutes an important result for process optimization. Moreover, the main disadvantages of the half height and two vertical rows

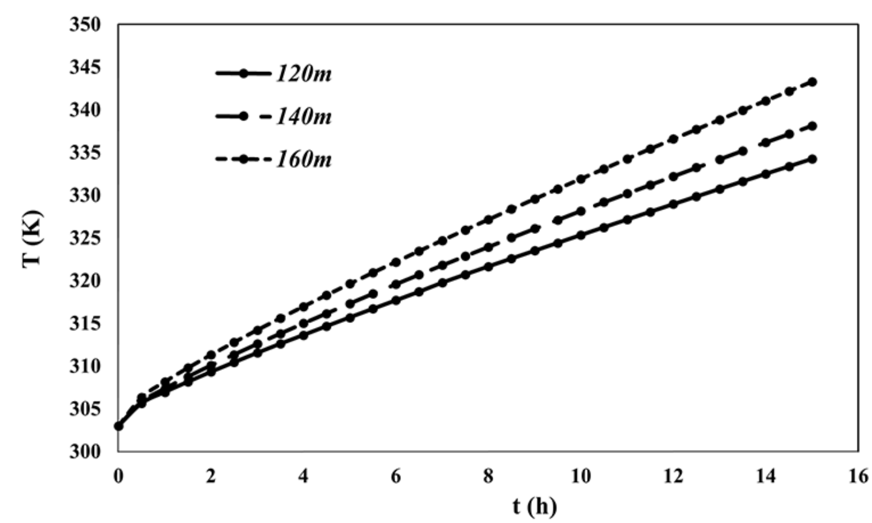

FIG. 6: Average temperature as a function of time for various coil lengths

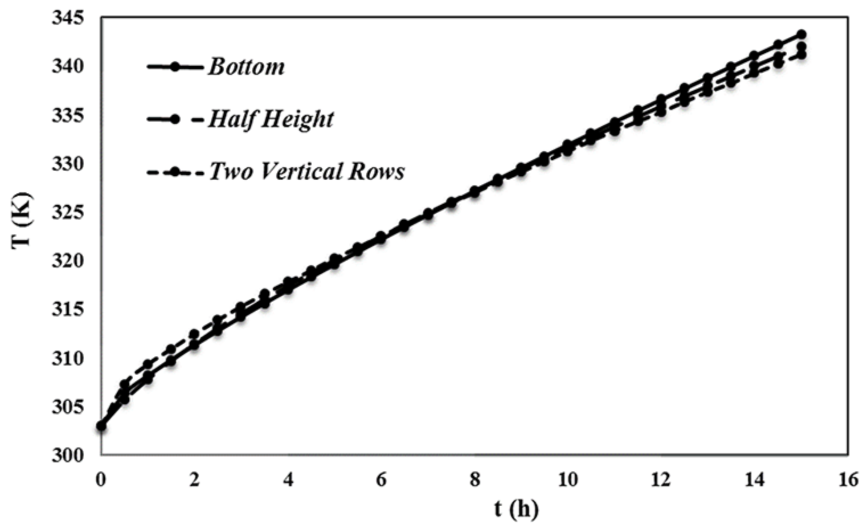

FIG. 7: Effect of different placements on the average temperature

layouts are the stability as well as the complexity of the established construction due to the high pressure ($\sim$ 5 bar) flowing the superheated water through the coil.

5. CONCLUSIONS

The heat transfer problem was considered for a tank of specific dimensions filled at about 85% with HFO and heated from 303 to 333 K by exploiting the flow of superheated water through a coil of serpentine design and various lengths, diameters, and thicknesses. To obtain the length values, both thermodynamical analysis and detailed 3D simulations were carried out, and the effect of the most crucial parameters on the results was investigated. It was found that the optimal length is approximately 160 m for the shortest time period that the process takes place. More specifically, the maximum length that can be established due to limitations on the geometrical characteristics of the banker tank cannot exceed 160 m for one layer of carbon–steel coil of 50 mm diameter and 4 mm thickness, and this can increase the HFO temperature to 333 K in about 9.5 hours according to the thermodynamic analysis and 10 hours through the detailed 3D and CFD-ACE^{+} simulations, which constitutes the most favorable result for the feeding process to the engines.

The most efficient scenario can be presented through the installation of a double layer of carbon–steel coil, which increases the overall length by about 300 m and limits the time needed to an interval of 5 hours. The main disadvantages of such an established scenario can be observed on the homogeneity of the temperature of the total HFO and the increased initial establishment costs for this construction. Finally, it should be mentioned that the temperature of the HFO inside the banker tank was not found to be consistently uniform in lower time interval processes due to the stirring that physically occurs when the oil enters the tank and the variations in the viscosity during several of the time steps.

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REFERENCES

Bejan, A., *Heat Transfer*, New York: Wiley, 1993.

Cheng, L., Gu, H., and Huang, S., A Comprehensive Mathematical Model for Estimating Oil Drainage Rate in SAGD Process Considering Wellbore/Formation Coupling Effect, *Heat Mass Transf.*, vol. **53**, pp. 1777–1795, 2017.

Dbouk, T., A Review about the Engineering Design of Optimal Heat Transfer Systems Using Topology Optimization, *Appl. Therm. Eng.*, vol. **112**, pp. 841–854, 2017.

Dzida, M., On the Possible Increasing of Efficiency of Ship Power Plant with the System Combined of Marine Diesel Engine, Gas Turbine and Steam Turbine, at the Main Engine–Steam Turbine Mode of Cooperation Possible to be Further Utilized, is Emitted from the Engines, *Pol. Marit. Res.*, vol. **16**, pp. 47–52, 2009.

Holman, J.P. Ed., *Heat Transfer*, New York: McGraw-Hill, 1990.

MAN Diesel & Turbo, *The MC Engine. Exhaust Gas Date. Waste Heat Recovery System. Total Economy*, Copenhagen, Denmark: MAN B&W Publications S.A., 1985.

Mcbride, B.J., Gordon, S., and Reno, M.A., Coefficient for Calculating Thermodynamic and Transport Properties of Individual Species, NASA Technical Memo TM-4513, Oct. 1993.

Pang, Z., Wu, Z., and Zhao, M., A Novel Method to Calculate Consumption of Non-Condensate Gas during Steam Assistant Gravity Drainage in Heavy Oil Reservoirs, *Energy*, vol. **130**, pp. 76–85, 2017.

Perry, R.H. and Green, D.W., *Perry's Chemical Engineers' Handbook*, New York: McGraw-Hill, 1997.

Rafiee, S.E. and Sadeghiazad, M.M., Experimental and 3D CFD Investigation on Heat Transfer and Energy Separation inside a Counter Flow Vortex Tube Using Different Shapes of Hot Control Valves, *Appl. Therm. Eng.*, vol. **110**, pp. 648–664, 2017.

Sato, N. Ed., *Chemical Energy and Exergy: An Introduction to Chemical Thermodynamics for Engineers*, Amsterdam, Netherlands: Elsevier, 2004.

Wagner, W. and Kretzschmar, H.-J. Eds., *International Steam Tables, Properties of Water and Steam based on the Industrial Formulation*, Berlin, Germany: Springer, 1998.