Modelling Heat Transfer in an Air Natural Convection Evaporator for Liquefied Natural Gas

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Abstract. *The paper presents work carried out to characterise a natural air convection heat exchanger to vaporise liquified natural gas (LNG). The vaporiser is made of a parallel set of vertical tube's serpentines with fins both inside and outside. The natural gas is vaporised and heated up by the surrounding air by natural convection. A model was build to calculate the evolution of natural gas inside the pipes based on the heat transfer from the surrounding air to the tubes. For the natural convection process, different heat trasnfer correlations are tested. Alternatively a numerical algorithm is used to describe the air circulation between tubes calculating the air properties along height and balancing the pressure gradients with the variation in air density. The results of this algorithm lead to higher convection heat transfer coefficients than correlations, but allow for a description of the properties of the moist air. Measurements of air temperature were carried out, showing temperature gradients between the tubes and the fins. The measured surface tube temperature presented errors that were corrected and show values in one side of the heat exchanger that is corroborated by the observation of snow and ice formation. The temperature profiles show also that during the phase change process, temperature increased which can be explained by a model for the mixture. Comparisons between measured and calculated values suggest that the correlations for heat trasnfer in channels are the most appropriate..*

Keywords*. Natural convection, Liquefied Natural gas, Vaporisation, Numerical Mode, Experimental characterization.*

1. Introduction

Natural gas (NG) utilization has increased considerably in the recent past as it is the cleanest fossil fuel and is convenient to use in different sectors including the domestic sector. In Portugal the natural gas network was started in 1997 and in 2004 a main maritime port for natural gas (Sines) went in operation. For areas far from the main pipeline local networks can be fed from a local liquefied natural gas (LNG) storage tank filled up by trucks. LNG is vaporised and heated up in air natural convection heat exchangers and distributed in the local area networks. The first unit of this type started in 2000 and there were 7 of these units in 2002. The natural gas distributed by pipe line in Portugal is imported from Hassi R'Mel in Argelia. LNG was distributed up to the end of 2003 from Cadiz in Spain and after that from Sines port, where LNG is delivered by boat from multiple sources.

The storage tanks and vaporisers are designated by autonomous as they only require man assistance to load the LNG. Heat transfer to the vaporisers occurs by natural convection with the surrounding air and therefore the air cooling leads to condensation and ice formation over the tubes. The usual design of the units for Portugal includes three sets of heat exchangers operated in shifts allowing ice melt when they are not being used. The ice formation depends on the ambient conditions and on the gas consumption as well as on the geometry of the heat exchanger. Knowledge of the heat transfer conditions and ice formation allows a better dimensioning of the heat exchanger and the selection of the time intervals for operation with individual units.

This paper presents a numerical model and results from measurements in a typical vaporiser for summer and winter conditions. The numerical model calculates the evolution of the natural gas properties within the tubes based on the heat transfer from the surrounding ambient air as presented in section 2. Heat transfer and pressure drop within the tubes is calculated based on internal pipe flow and properties of natural gas as explained in section 2.1. Two different approaches were considered for the air heat transfer as presented in section 2.2, one based on heat transfer correlations and the other based on the estimated air flow rate between the tubes. The measurements carried out for the air and the tube surface temperature with thermocouples are presented in section 3. Section 4 presents a comparison between the model results and experimental values corrected with estimated errors. Section 5 presents some conclusions from the present work and defines possible improvements.

2. Numerical model

This section presents the numerical model prepared to calculate the natural gas properties within the tubes of the heat exchanger in section 2.1 that is coupled with the calculation of heat transfer from the air side from correlations in or from a natural circulation model presented in section 2.2. The calculation of the average heat transfer coefficient around the tubes can be calculated from heat transfer correlations as presented in 2.2.1, but in this case no indication is obtained for the air temperature around the tubes. Therefore a second approach presented in 2.2.2 was implemented to calculate the air flow rate passing along the tubes. The calculation procedure is based on the pressure losses in the flow around the tubes and the variation of the air density. Heat transfer between the air and the tube is calculated, allowing an estimative of the air temperature profile.

2.1. Natural gas model

2.1.1. Natural gas properties

LNG is a mixture of methane and ethane that have different properties. The properties of each substance can be calculated using appropriate models (Reid et al, 1987). Due to the large difference between their boiling points the mixture requires the consideration of equilibrium considerations to accurately describe the evolution of the composition in each phase (Rojey et al, 1997). In the present work the methane-ethane mixture was considered as an equivalent substance with critical properties calculated from the critical properties of the individual components according to Kays rule (Reid et al, 1987; Rojey et al, 1997). Based on the LNG composition, 91% methane and 9% ethane, the critical properties of the equivalent mixture are: $P_{c,m}$=46.25 bar; $T_{c,m}$=-71.1ºC and the acentric factor is ω_m=0.0198.

The mixture saturation pressure $P_{VP,m}$ is expressed as a function of temperature by the Hankinson-Brobst-Thomson (HBT) technique recommended by Reid et al (1987):

$$P_{VP,m} = P_{c,m} P_{r,m} \tag{1}$$

where $P_{r,m}$ is a mixture reduced pressure, calculated from the generalized Riedel vapour pressure equation:

$$\log_{10} P_{r,m} = P_{r,m}^{(0)} + \omega_{SRK,m} P_{r,m}^{(1)} \tag{2}$$

that is expressed as a function of the mixture temperature, the pseudocritical temperature $T_{c,m}$ and acentric factor. It should be noted here that the critical properties of the mixture for the application of the HBT technique are different from the values from the Kays rule. The mixture saturation temperature is obtained from the mixture saturation pressure equation by an iterative procedure using the previous Eq. (1) and (2).

The enthalpy of the mixture is calculated as for a single component based on the specific heat at constant pressure for the liquid and vapor zones. For the liquid zone, specific heat $c_{pL,m}$ is calculated from the Rowlinson-Bondi modification of the corresponding states method as a function of the ideal gas specific heat $c^0_{p,m}$, the reduced temperature of the mixture $T_{r,m}$ and Pitzers's acentric factor by (Reid et al, 1989):

$$\frac{c_{pL,m} - c^o_{p,m}}{R} = 1.45 + 0.45\left(1 - T_{r,m}\right)^{-1} + 0.25\omega_m \left[17.11 + 25.2\left(1 - T_{r,m}\right)^{\frac{1}{3}} T_{r,m}^{-1} + 1.742\left(1 - T_{r,m}\right)^{-1}\right] \tag{3}$$

The ideal gas specific heat is expressed as a function of temperature by a fourth power law from Kee et al (1989). For real gas specific heat capacity a residual heat capacity is added to the ideal gas specific heat according to the Lee-Kesler method suggested by Reid et al (1979).

$$c_{p,m} = c^o_{p,m} + \left(\Delta c_{p,m}\right)^{(0)} + \omega_m \left(\Delta c_{p,m}\right)^{(1)} \tag{4}$$

where $\Delta c_{p,m}^{(0)}$ and $^{(1)}$ are respectively the simple fluid and the deviation functions presented in tabular form and considered for values of the reduced pressure above 0.01 up to 5. The calculation of the critical properties for Eq. (3) and (4) are based on the Lee-Kesler (Reid et al; 1979).

The mixture vaporization enthalpy $\Delta H_{LG,m}$ is determined as a function of the reduced temperature by:

$$\Delta H_{LG,m} = RT_{c,m}\left(7{,}08\left(1 - T_{r,m}\right)^{0{,}354} + 10{,}95\omega_m\left(1 - T_{r,m}\right)^{0{,}456}\right) \text{ for } 0.6 < T_r \leq 1.0 \tag{5}$$

The vapour specific volume is obtained from the Soave-Redlich-Kwong cubic equation of state (SRK) as a function of the critical properties and the acentric factor of the components of the mixture. The vapour specific volume of the mixture includes further a correction due to Peneloux and Rauzy expressed as a function of Rackett compressibility factor for the components and their critical properties. The mixture liquid molar volume is determined using the Thomson method that is an extension of the HBT technique recommended by Reid et al (1979). All the details about the model are available in Azevedo and Toporova (2004), including correlations for the transport properties.

2.1.2. Natural gas flow and heat transfer

The natural gas flow inside the tubes is promoted by the pressure in the storage tank. This pressure is regulated vaporizing some LNG in an auxiliary natural convection heat exchanger directly connected to the storage tank. The numerical model was prepared based on an assumed natural gas flow rate taken from the measured values. The objective of the model is the calculation of the evolution of the natural gas state (temperature and pressure) that change due to heat transfer and friction. The calculations are based on the division of the total pipe length in small tube elements along the natural gas flow direction. The variation of pressure along each tube element is calculated based on the modification of kinetic and potential energy and from localized and friction losses using the following equation:

$$P_i = P_{i-1} + \frac{\rho_{i-1} u_{i-1}^2}{2} - \frac{\rho_i u_i^2}{2} + \rho_{i-1} g h - \frac{4 f h}{D_h}\left(\frac{\rho_{i,i-1} u^2{}_{i,i-1}}{2}\right) - \Delta P_{bend} \tag{6}$$

The effect of gravity changes in consecutive tubes as the flow is alternating upwards and downwards. Pressure losses in the bends among consecutive tubes are considered further to the friction pressure loss following the recommendations of Hewitt et al (1994) including the properties for the two phase region.

The state of the natural gas and temperature at each tube element is calculated in each location from the enthalpy in the previous element from an energy balance:

$$h_i = h_{i-1} + Q_i / \dot{m}_{NG} \tag{7}$$

The heat transferred at each tube element is calculated based on the external wall and the natural gas temperature:

$$Q_i = \left(T_W - T_{NG}\right) \Big/ \left[\frac{1}{h_{mix}\left(\eta_{fin} A_{fin} + A_w\right)_{In,i}} + \frac{\ln\left(D_e / D_i\right)}{2\pi \Delta Z k_{Al}} \right] \tag{8}$$

The natural gas heat transfer coefficient inside the tube is calculated from single phase flow for the liquid and vapour based on correlations recommended by Gnielinski (1998), while for two-phase flow the heat transfer coefficient is calculated from the correlation of Steiner and Taborek (1992). The external wall temperature is calculated from the heat transfer in the air side as explained in the next section.

2.2. Air heat transfer model

2.2.1. Natural convection heat transfer correlations

The natural convection outside the tubes results from the air flow induced by the air cooled by the fins and tubes. There are no correlations in the literature exactly for the situation under consideration, so approximations were introduced to select appropriate heat transfer correlations. Two types of correlations were considered, corresponding respectively to the heat transfer around vertical surfaces and heat transfer within open ended channels. For the heat transfer over external surfaces no consideration is given to the neighbouring tubes and there are correlations for isolated vertical plates as well as for cylinders or other vertical surfaces with various forms (Churchill, 1998). All these correlations use the total height as the characteristic dimension for the Rayleigh and Nusselt numbers and produce results within 10%.

To use the correlations for channels a hydraulic diameter was defined from the area and perimeter in a rectangle formed between four tubes although in reality the fins do not contact each other, so there is no closed channel. Two heat transfer correlations were considered, one for the natural convection inside tubes using the hydraulic diameter as the characteristic dimension (Churchill ,1998) and another for the heat transfer between two parallel plates (Bar-Cohen and Rohsenow, 1984). The correlation for parallel plates uses the space between the walls as the characteristic dimension which is equivalent to half of the hydraulic diameter. This correlation presents as assymptotics the cases for completely developed flow in channels and the case of isolated vertical surfaces.

All the heat transfer correlations were evaluated using an average external wall temperature that is also used to calculate the rate of heat transfer from the tube to the air:

$$Q_i = (T_{Air} - T_W)\times(R_{Ice} + R_{Fins})/(R_{Ice}R_{Fins}) \tag{9}$$

where R_{Ice} and R_{Fins} are the heat transfer resistances across the ice and fins including air convection:

$$R_{Ice} = \frac{1}{h_{Air}A_{Ice}} + \frac{\ln(D_{Ice}/D_e)}{2\pi\Delta Z k_{Ice}} \text{ and } R_{Fins} = \frac{1}{h_{Air}\eta_{Fin}A'_{Fin}} + \frac{L_{Ice}}{k_{Al}N_{Fin}A_{t,Fin}} \tag{10 a,b}$$

The ice diameter is an input to the model, although one of the aims of the present work was the evaluation of the ice formation using the model presented in the following section. The value of the external convection heat transfer coefficient is a function of the wall temperature and therefore is obtained iteratively using Eq. (8) and (9). Constant heat transfer coefficients were also considered to evaluate the experimental results more directly.

2.2.2. Heat transfer coefficient from air circulation calculation

The use of a heat transfer coefficient has indicated above does not allow the estimative of the air temperature circulating between the tubes. One of the motivations of the study being carried out is the formation of ice over the tubes, so a simple model was prepared to estimate the circulation of air between the tubes. The basis for this model is the calculation of the flow rate that allows the value of total pressure drop along the vertical channel to be similar to the difference in weight between the air within the channel and the undisturbed air far from the heat exchanger. The static pressure at the inlet of the tubes at a height H is calculated from the atmospheric pressure minus the weight of the air column at the atmospheric conditions and the dynamic head due to conversion in kinetic energy minus the pressure loss due to the inlet contraction:

$$P_{inlet} = (P_{atm} - \rho_{atm}gH) - (K_c + 1)\frac{\rho_{atm}u_0^2}{2} \tag{11}$$

Based on an assumed air flow rate the pressure at the outlet of the channel is calculated from the pressure drop calculation in individual tube elements, using local values of temperature, until reaching the exit of the channel.

$$\Delta P = \sum_i P_i - P_{i-1} = \frac{\rho_{i-1}u_{i-1}^2}{2} - \frac{\rho_i u_i^2}{2} + \rho_{i-1}gh - \frac{4fh}{D_h}\left(\frac{\rho_{i,i-1}u^2{}_{i,i-1}}{2}\right) \tag{12}$$

The pressure after the exit from the channel decreases further due to expansion losses and recovers the dynamic head. Taking this into account the pressure loss from the above equation should be similar to:

$$\Delta P = \rho_{atm}gH + (K_c + 1)\frac{\rho_{atm}u_0^2}{2} + (K_e - 1)\frac{\rho_n u_n^2}{2} \tag{13}$$

Assuming a value for the air flow rate in each channel, the previous equation can be used to define an error that is used to correct the air flow rate until convergence. This iterative procedure is performed in parallel with the calculation of the air heat transfer which is used to calculate the wall temperature. The air temperature profile from energy balances in channel elements equivalent to the tube elements considered for the natural gas:

$$h_i = h_{i-1} + Q_i/\dot{m}_{Air} \tag{14}$$

In this model variant the convection heat transfer coefficient between the air and the fins or ice layer are calculated using correlations for heat transfer in forced convection inside equivalent tubes using the Hausen correlation for laminar flow, Gnielinski for the transition region and Petukov and Popov for fully turbulent conditions (e.g. Gnielinski, 1998).

To test the model described above, calculations were performed for different constant wall temperatures changing the height and hydraulic diameter. The hydraulic diameter D_h was also used to define the average space between the wall channels $S=D_h/2$ to apply the correlations for channels which produces similar results as the correlation for the flow within a tube, so only one correlation is represented in the following figure. Figure (1) presents a comparison of the heat transfer coefficient in dimensionless form as a function of the Raleigh number defined with the channel thickness multiplied by S/L that is the channel thickness to length ratio as indicated by the correlation of Bar-Cohen and Rohsenow (1984).

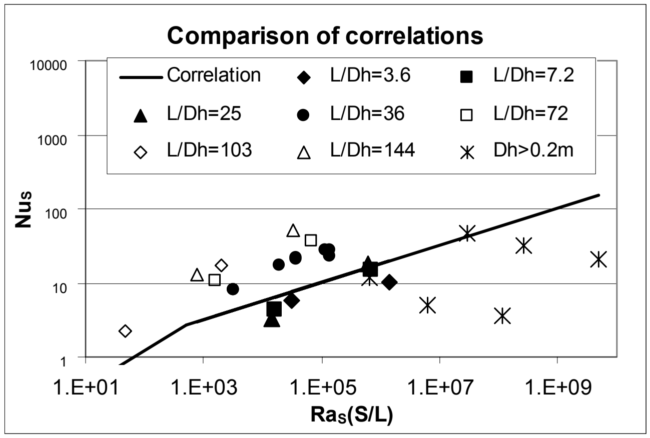


Figure 1. Calculated values of the Nusselt number as a function of the modified Rayleigh number for channels.

It can be observed that for a given height to hydraulic diameter the model predicts the correct trend for the heat transfer coefficient as a function of the modified Rayleigh number. The model however is very sensitive to the hydraulic diameter. For values of the hydraulic diameter from 0.6 to 2 m the calculated values are within or smaller than the correlations but they present a large variation despite the value of L/Dh ranges from 2 to 10. The hydraulic diameter has the larger influence on the results and therefore comparisons of model results changing the hydraulic diameter should be analysed with care. The results for hydraulic diameter from 0.14 m to 0.2 m corresponding to values of L/Dh from 3.6 to 36 are closer to the values calculated from the correlations and for a given value of L/Dh the trend follows the trend displayed by the correlations. The actual dimensions of the tubes considered in this study correspond to the value L/Dh=36 so in this case the model over-predicts the convection coefficient compared with the correlations. For smaller hydraulic diameters and correspondingly larger values of L/Dh the over-prediction increases.

As discussed previously, this simple model was prepared to calculate the air temperature distribution and from this is used to predict the occurrence of condensation and ice formation as detailed in Azevedo and Toporova (2004). For heat transfer however the differences observed in Fig. (1) favour the use of the heat transfer correlation. Possible adjustments to the model parameters could improve the model sensitivity to the hydraulic diameter or the calculation method for the air flow rate resulting from equation (13).

3. Experimental characterization

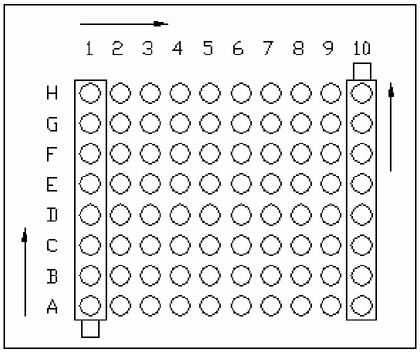
3.1. Heat exchanger and measurements performed.

The installation characterised in this study is located in Evora that has a continental climate. This unit has a LNG storage tank of 120m^3 and three sets of two natural convection heat exchangers in parallel. Each pair is operated during 8 hours allowing for the formation of ice deposits that can be melt during the other periods. For the particular unit considered and the natural gas consumption rate, only a small amount of ice was observed in the first tubes of the serpentines and therefore the time of the observation within the eight hour period was not important. The realization of the measurements was conditioned to the periods when LNG was delivered to the storage tanks and therefore on average each set of measurements was done within 3 to 4 hours in 10 days mainly during the morning (once 1-3 pm).

Measurements were performed during summer and winter corresponding also to an increase of the total flow rate with mean values respectively of 209 and 338 Nm^3/h. The flow rate consumption changes daily and within each day but the values observed have variations within $\pm$ 10%. The average air temperature and relative humidity for the summer and winter periods were 33ºC/24% and 12ºC/76%.General data from the plant included readings of pressure in the storage tank and temperature in the line from the tank to the vaporisers. This line was normally covered with an ice thickness of about 4 cm that was removed when the tank was loaded.

Each heat exchanger is made of 8 serpentines with 10 tubes each one, where the natural gas flows alternating upwards and downwards. Figure (2) presents a sketch of the heat exchanger with the identification of the tubes. Measurements of air temperature were performed using a HygroClip probe with 15 mm diameter that was located between the tubes as indicated by position (T) in Fig. (2). The measurement of air temperature with the HygroClip required longer exposure time so limited measurements were done with this probe. A 2 mm thermocouple was also used to measure air temperature between consecutive fins in position (A) as indicated in Fig. (2). The location of the thermocouple between the fins required some care to avoid any contact with snow in contact with the fins.

The measurement of tube wall temperature is difficult as this value is expected to be close to the LNG temperature and therefore much lower than the ambient air temperature. The use of an infrared thermometer could not be done with success due to the low emissivity of the aluminium. A thermocouple (K) 0.5 mm in diameter mounted with a spring was used to measure the wall temperature. Before the measurement at a given position, the surface was cleaned to remove any snow or ice formed and than the probe was applied with a conducting paste to minimize the contact heat transfer resistance. The values observed from the thermocouple during each measurement typically decreased to a minimum due to the transient cooling of the probe, but than increased again due to the formation of some snow close to the thermocouples. In some cases values close to 0ºC were observed after some time. From this observation it was concluded that significant errors affect the measured values so it was adopted to register the absolute minimum value observed at each location rather than average values from different measurements.



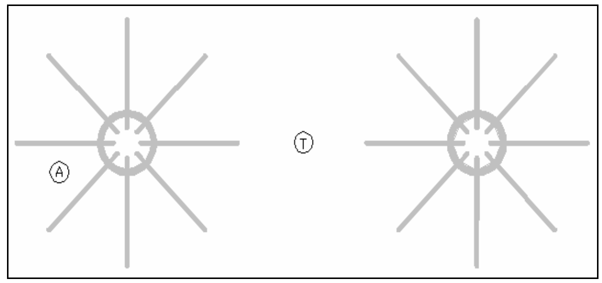


Figure 2. Geometry of the heat exchangers. a) Layout and nomenclature to identify the tubes, b) Sketch of the tube with fins and location for measurement of air temperature.

3.2. Measured values in the heat exchanger

Figure (3) presents several sets of air temperature observed between the fins during the summer campaigns. When a large thickness of snow is deposited at the fin's tip a closed channel was formed for a certain height between the fins and in this case the air temperature is very low. The measured values of air temperature could not be reproduced closely between different times so the values in Fig. (3) have a range of 15ºC for each location. During the summer conditions the air temperature as mentioned was on average 33ºC so the air temperature at maximum was observed to cool about 50ºC. Air temperature between the fins of the following tubes did not present significant gradients with values close to 20ºC for the second, 27ºC for the third and values close to the ambient air temperature for the fourth row tube. The air temperature between the tubes in summer conditions was not recorded but was much closer to the ambient temperature.

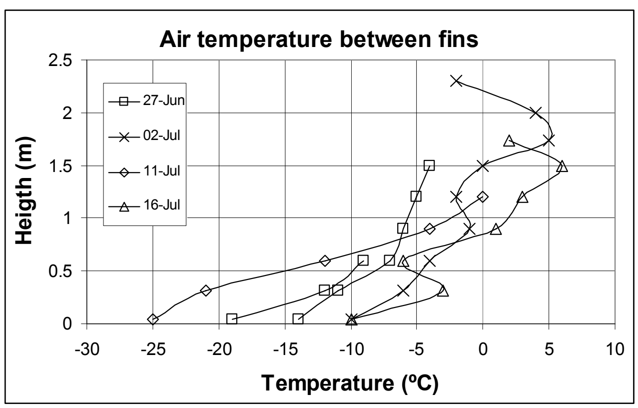


Figure 3. Air temperature distribution between the fins for the first tube in summer conditions.

Figure (4) presents a sample of the measured wall temperature in one day at the first row of tubes for all serpentines. It can be observed that there is a substantial difference between the first tubes in all the serpentines. Lower values are observed in the tubes closer to the inlet to the distributor that were also the tubes with higher tendency to fix snow in summer and form an ice layer in winter. This can be a consequence of a flow mal-distribution and the different evolution of the temperature profiles may be a consequence of the equilibrium conditions during phase change that occurs slowly in the first tube where the flow rate may be higher.

The wall tube temperature is expected to be closer to the natural gas temperature than the air temperature before and during phase change. In the horizontal pipe from the storage tank to the vaporisers there is some heat transfer that was estimated using a correlation for natural convection, leading to the conclusion that the vapour mass fraction should be less than 7% in winter and 16% in summer. At these early stages of phase change the temperature should be close to the equivalent mixture phase change temperature which for the pressure under consideration is about -132ºC, which is a value much lower than those measured and presented in Fig. (4).

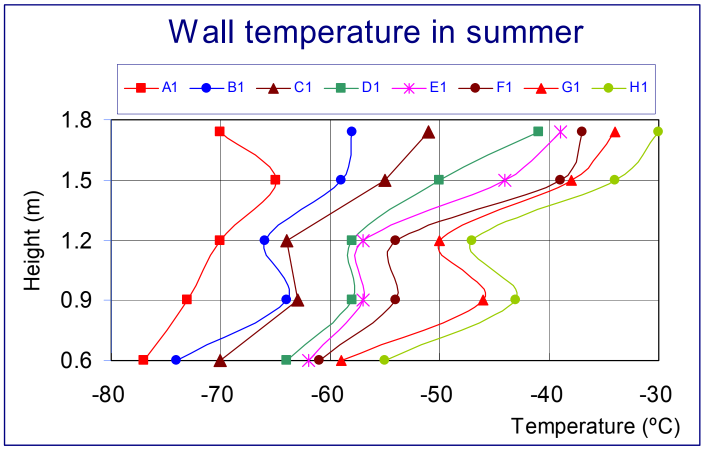


Figure 4. Measured wall temperature during summer at the first tube of each serpentine (See Fig. (2)).

Based on the lowest measured value and the expected minimum wall temperature (-130ºC) an estimate was made for an error factor that is represented in Table (1)

$$E_{Temp} = 1 - \left(T_{Min,Observed} - T_{Wall}\right) / \left(T_{Air} - T_{Wall}\right) \tag{15}$$

Table 1. Estimative of the error in temperature.

	T_{air} (ºC)	T_{Wall} (ºC)	$T_{Min,\ Observed}$ (ºC)	E_{Temp}
Summer	35	-130	-84	0.28
Winter	11	-130	-90	0.28

A similar value was obtained for summer and winter conditions showing some consistency for the deviations observed. Measurements for the wall temperature were also performed at the middle of the horizontal pipe from the storage tank leading to comparable error factors (0.31 and 0.22). The error in reality can be lower has the natural gas temperature was derived from an equivalent mixture properties, while considering the equilibrium conditions the temperature should increase during phase change. To compare the experimental results with the numerical model, the values in the next section were all corrected with the error factor defined by Eq. (15).

4. Comparisons and analysis of the results

This section presents a comparison between calculated and measured results. Figure (5) presents a comparison for the air temperature obtained using the numerical model described in section 2.2.2., for winter conditions. In this period measurements of air temperature were performed between the tubes in position T from Fig. (2), showing minimum values around -5ºC between the first and second tube row, when the ambient air temperature was about 10ºC. The values of temperature observed in similar positions later on the same day or in another day were about 1ºC although temperature was below 13ºC, so the negative values could not be observed as were obtained early in the morning. Figure (5) includes the calculated values of the air temperature showing a reasonable agreement. The flow rate of the natural gas was changed by $\pm$ 20% but this has a small influence on the air temperature that is higher than observed. Due to the approximation of the phase change in the model, the wall temperature should in general be under-predicted. Another factor that was already mentioned is the difference from the air temperature between the fins and the tubes, which for another day at winter was observed to be 8ºC. Due to the higher calculated temperature in the model it seems that the flow circulation is over-predicted leading to higher values of the heat transfer coefficient, which corroborates the observations made in Fig. (2).

Figure (6) presents a comparison between the calculated wall temperature and the corrected values for the second tube row. Results are presented using the different heat transfer correlations: 1) is the heat transfer model from section 2.2.2; 2-3) are correlations for heat transfer within channels; 4-6) are correlations for heat transfer around external vertical surfaces and 7) corresponds to the specification of constant convection heat transfer coefficients with values from 1 to 3 W/m^{2}K. The experimental results for the first tube show some gradient as indicated in Fig. (4), while the model results show an almost constant value, followed by an increase when phase change is complete.

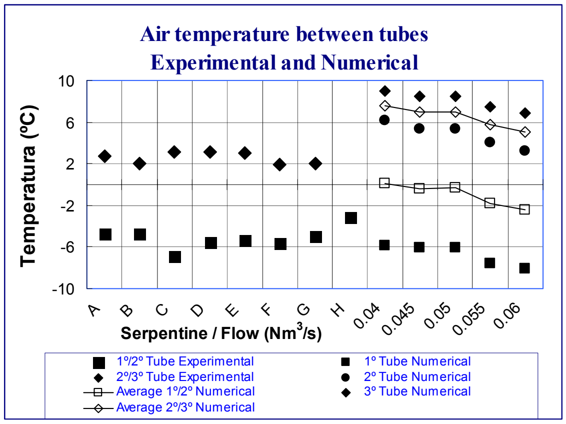


Figure 5. Comparison between measured and calculated air temperature at the bottom of all tubes.

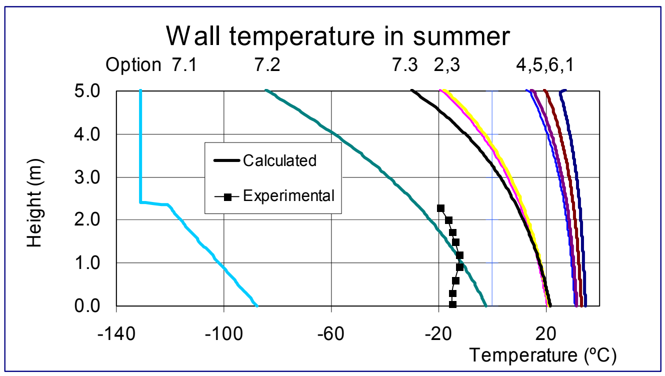


Figure 6. Comparison of corrected wall temperature and calculated values with different heat transfer coefficients.

From the comparison of the calculated values it can be concluded that the best agreement is obtained for the constant convection heat transfer coefficient of 2 W/m^2K. The correlations for channels produce values close to 3 while the correlations for convection in vertical surfaces produce values between 5 and 6 W/m^2K, which is also the result of the calculated value based on the numerical model presented in section 2.2.2. For the smaller value of the heat transfer coefficient (1 W/m^2K) the phase change is calculated to finish only close to the bottom of the second tube as indicated in Fig. (6). The heat transfer to the natural gas is over estimated when considering phase change at constant temperature and therefore due to this fact it can be said that the real values of heat transfer coefficient should be higher. The values obtained from the correlations in channels and in particular the one from Bar-Cohen and Roshenow presents the correct asymptotic when the space between the walls increases and therefore is the correlation suggested for further use.

A similar comparison between the temperature calculated by the different options and the corrected measured values for winter was performed for four consecutive tubes in the first serpentine (tubes A1 to A4, see Fig. (2)). The comparisons made indicated that the corrected values were in closer agreement with a constant heat transfer coefficient of 3W/m^2K as presented in Figure (7).

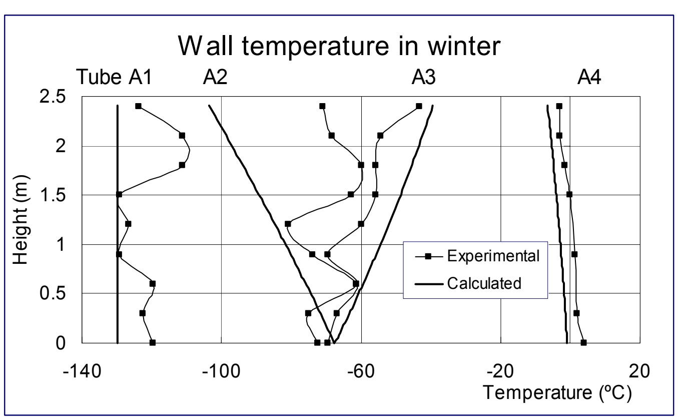


Figure 7. Comparison between measured and calculated air temperature at successive tubes A1- A4.

The correlations for the heat transfer in channels produced convection coefficients between 3 and 4 W/m^2K. Again the correlations for isolated vertical surfaces led to larger values between 6 and 7 W/m^2K and the values from the model led to even larger values 7.3 W/m^2K. Despite the lower air temperature, the values for the air side heat transfer coefficients were higher due to the lower wall temperature in this period. Of course the comments made earlier to Fig. (6) regarding the approximation of the phase change of an equivalent substance apply here and again the convection heat transfer correlations for channels are the best selection for calculations.

5. Conclusions

The paper presents a numerical model for the calculation of natural gas evolution in a natural air convection heat exchanger. Different approximations were assumed to build the model, namely considering an equivalent substance allowing for the evaluation of the temperature distribution in natural gas vaporisers. A model for the calculation of the air temperature distribution was implemented and comparison of this model with heat transfer correlations lead to the conclusion that it tends to over-predict convection heat transfer.

Measurements were obtained in a representative natural gas vaporiser. Although the wall temperature is subject to significant errors, it shows that the temperature distributions in parallel serpentines are not similar, possibly due to a flow mal-distribution in the serpentines. The measured values show a temperature increase since the start that can only be explained by a further detailed model for the phase change.

Air temperature measurements show that there are significant differences between the fins and between the tubes. The values calculated by the model which over-estimates heat transfer, lead to larger values of air temperature which is consistent with the previous observation. The comparison of wall temperature with results from the model, suggests that the most appropriate heat transfer correlations are those for heat transfer within channels.

Improvements of the model are possible. For the natural gas properties a model to account for the mixture phase change is necessary. The natural air circulation model can also be improved to allow for a good reproduction of the heat transfer correlations within channels, as this is the situation that is simulated.

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