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ANALYSIS OF THE TRANSIENT COOLDOWN OF SUB-SEA PIPELINES

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ABSTRACT

A precise analysis of the transient cooldown of subsea pipelines is crucial for offshore flow assurance, to avoid, for example, hydrate formation or wax deposition. Flow assurance in transportation lines, where column separation can occur due to large temperature drop coupled with large pressure drop, must also be addressed. Usually, pipeline thermal insulation is designed for steady state conditions. However, during shutdowns, the temperature drop experienced by the stagnant fluid is more pronounced, eventually reaching some critical temperature level, that can lead to flow line blockage and flow re-start problems. Thus, the determination of the temperature and pressure distributions along the pipeline under transient conditions is important to maintain the line operating safely.

To determine the transient heat transfer in pipelines, the fluid flow conservation equations coupled with the heat conduction equation applied to the pipeline wall were numerically solved. The heat loss from the pipeline was determined by solving the transient heat conduction equation for the pipewall layers, utilizing a simple one-dimensional model in the radial direction. The coupled system was solved numerically employing the finite difference method.

Transient analyses were performed for two scenarios. In the first one, the cooldown process of oil in a subsea pipeline was evaluated, with the effect of variable thermal properties on the temperature profile being investigated. The dependence of the temperature on the thermal conductivity and specific heat capacity was analyzed. In the second scenario, gas flows inside the pipeline, and the effect of temperature variation on a stagnant fluid is presented. Tests for different values of thermal diffusivity corresponding to new and old thermal insulations were performed.

INTRODUCTION

As drilling and production operations expand into deepwater environments, seawater temperature decreases to a few degrees above freezing, generating a variety of problems, such as hydrate formation and wax deposition in well bores, subsea pipelines and subsea equipment. In pipelines, hydrates can restrict and even block the flow. Partial blockages can have adverse effects including reduced production. Hydrates are crystalline solids formed from a combination of water and one or more hydrocarbon gases such as methane, ethane, or propane which are stable at high pressure and low temperatures; predominant conditions in deepwater operations. In physical appearance, hydrates resemble packed snow or ice. Controlling heat losses is usually the best choice for preventing hydrate formation. The technology available for the control of heat losses and the prevention of hydrate formation and wax deposition is well known in the industry and includes pipe-in-pipe systems, single insulated pipe, and pipe burial.

Often the temperature in the flowline will be above the hydrate formation envelope under steady state operating conditions. However, during a shutdown or interruption in flow, stagnant fluid in the pipeline loses heat to the surroundings and cools to local seabed temperatures, increasing the risk of hydrate formation. Thermal insulation is necessary in these cases to keep the produced fluids above the hydrate temperature long enough for the operator to introduce hydrate inhibitors or until flow can be reestablished. Typically, operators require eight to twelve hours of temperature levels above the hydrate formation temperature at no-flow conditions (Dwight J. et al, 2004).

The revision of some available commercial flow simulation softwares reveals that a majority of them carry out thermal calculations. The *Pipeline Studio* software *(TGNET* and *TLNET*) (Energy Solutions Inc, 2004) was developed to determine single phase flow (with empirical correlations adjusted for natural gas) inside the pipeline. It allows the user to consider or not the thermal capacitance of the pipe. In this former case (available only for gas), it performs a calculation of the radial temperature distribution within the wall at each node along the pipe (Licenergy Inc, 2000).

The commercial software *OLGA* (Scandpower Petroleum Technology, 2000) was developed to solve multiphase and single phase flows. It also allows the user to select two distinct conditions to analyze a thermal problem. With the first option, the thermal capacitance of pipe layers is neglected and an overall heat transfer coefficient must be prescribed. The second option, called 3D Thermal module, takes into account the thermal capacitance of the pipe layers. It solves the thermal transient in two steps: in the first step the energy equation is solved for the fluid and pipe wall, yielding the temperature profile of both, the fluid and the pipe wall along the pipeline. In the following step, a 2D conduction equation is solved for the solid media surrounding the pipe, giving the temperature distribution over the cross-section along the pipeline. A combination of the solutions of steps one and two result in a 3D temperature field.

A literature survey shows few papers dealing with transient thermal analysis. Brown et al (1996) developed a computational model to describe transient heat transfer in pipeline bundles; this model was coupled to the transient, multiphase flow simulator *OLGA*. Pipeline walls and insulation layers were not explicitly accounted for in the bundle model. To handle their thermal resistances, equivalent heat transfer coefficients were defined. This approach neglects the effects of the thermal capacity of the pipe walls and insulation layers during transients. The lines containing the multiphase production fluids were modeled by *OLGA*, and the heat transfer between the internal lines, carrier pipe and surrounding was handled by the bundle model. Zabaras and Zhang (1998) presented a finite element heat transfer calculation of the transient cooldown for different bundle and pipe-in-pipe configurations. Danielson and Brown (1999) developed two models to predict the behavior of pipeline bundles, the first one is similar to the above model, and the second model is an analytic approach. According to the authors, the second model is not grid sensitive; it is extremely fast and easy to set up.

According to Hight (2000), conditions in gas flowlines are not strongly affected by insulation. Due to the low heat capacity of gases, the heat content of fluid entering the flowline is low and the fluid temperature is affected mainly by the cooling from expansion as pressure is reduced. Therefore, the main consideration is to minimize frictional pressure losses. Temperatures below ambient can be avoided by sizing flowlines with enough capacity to avoid excessive pressure losses. However, this point should be questioned for pressurized gas lines, since compressed gases present a thermal behavior similar to liquids and quite different from gases at atmospheric pressure.

Jackson et al. (2005) discussed the design of subsea thermal insulation systems, by addressing the influence of the addition of water on insulating layers, as well as their compression after 20 years of operation, which affects their properties. The influence of the thermal capacity of the pipeline was investigated by Barrera et al. (2005, 2006) for different type of fluids flowing inside the pipeline.

The aim of the present paper is to perform an analysis of the influence of the pipe wall thermal capacitance on the transient behavior of heavily insulated lines. Different parameters were analyzed to represent new and old thermal insulations. The main concern is to address the importance of the energy stored in the insulation layers; therefore, a simplified solution was obtained assuming the flow to be single phase. The influence of the Nusselt number correlations to determine the gas cooldown time is also addressed. The solutions obtained with the models developed in this work were compared with the commercial softwares *OLGA,* and *Pipeline Studio* (*TLNET* and *TGNET*).

MATHEMATICAL MODEL

The present model couples the solution of the flow field inside the pipeline and the conduction equation at the pipe wall and insulation. The wall temperature was calculated taking into account the full thermal capacitance of the pipe wall, which may be made up of several layers of different materials.

Since typical pipelines are very long in relation to their diameter, the flow can be considered as one-dimensional, with uniform velocity along the cross section. Additionally, the axial diffusion heat flux can be neglected in relation to the convective heat flux, which is dominant. Also, due to the large length of the pipelines, the longitudinal conduction within the pipe wall can be neglected in relation to the radial heat flux.

Heat transfer between the pipe and the fluid is dependent on the flow characteristics inside the pipe and it was determined from convective heat transfer correlations, as it will be shown in the following sub-sections. A convective boundary condition was also specified at the external pipe boundary, determining the heat flux from the pipe to the ambient.

The flow inside the pipeline is governed by the time-dependent continuity, momentum and energy equations, and an equation of state.

In the present analysis, the oil density ρ was considered as a function of pressure P and temperature T, being given by the following relation

$$\rho = \rho_{ref}\,[1 - \beta\,(T - T_{ref})] + \frac{(P - P_{ref})}{a^2} \tag{1}$$

where a and β are the isothermal speed of sound and the thermal expansion coefficient, respectively, both assumed constant. ρ_{ref} is the reference density evaluated at the reference pressure p_{ref} and reference temperature T_{ref}.

The gas was considered as a quasi-ideal gas, thus the equation of state is

$$\rho = \frac{P}{a^2} \qquad ; \qquad a^2 = \frac{z\,\Re\,T}{\overline{M}} \tag{2}$$

where $\Re$ = 8314 Nm/(kgmol K) is the universal gas constant and $\overline{M}$ is the average molecular weight of the gas. z is the compressibility factor, which for natural gas can be determined from (Stoner Pipeline Simulator, 2003):

$$\frac{1}{z} = 1 + \frac{(3.44 \times 10^5\; P'\, 10^{(1.785\; SG)})}{\tilde{T}^{\,3.825}} \tag{3}$$

where P' is the bulk pressure in psig, $\tilde{T}$ is the fluid temperature in ºR and SG the specific gravity.

To simplify the problem and to evaluate only the influence of the pipeline thermal properties on the transient heat transfer, all other fluid properties, like the absolute viscosity μ and the thermal properties, such as specific heat at constant pressure c_p and thermal conductivity k were considered constant.

In order to allow the representation of hilly terrains, the duct centerline can be inclined with respect to the horizontal direction at an angle α. Further, due to significant pressure variations, its effect on the pipe deformation was considered.

The mass conservation equation can be written as (Wylie and Streeter, 1978)

$$\frac{dP}{dt}+\frac{\rho a^2}{\xi}\left[\left(\frac{\partial V}{\partial x}+\frac{V}{A}\frac{\partial A}{\partial x}\right)-\beta\frac{dT}{dt}\right]=0 \tag{4}$$

where d/dt is the material derivative defined as

$$\frac{d}{dt}=\frac{\partial}{\partial t}+V\frac{\partial}{\partial x}. \tag{5}$$

V is the velocity, A is the cross section area and β is the coefficient of thermal expansion. ξ is given by

$$\xi = 1 + \rho a^2\, 2\, C_D \frac{D}{D_{ref}} \quad ; \quad C_D = (1-\upsilon^2)\frac{D_{ref}}{2\,e\,E} \tag{6}$$

where D and D_{ref} are the inner pipe diameter and the reference diameter determined at atmospheric pressure p_{atm}. The coefficient C_D accounts for the pipe deformation due to pressure, where e is the pipe wall thickness, E is the Young's modulus of elasticity of the pipe material, and ν is the Poisson's ratio.

The linear momentum equation can be written as

$$\frac{dV}{dt}=-\frac{1}{\rho}\frac{\partial P}{\partial x}-\frac{f}{2}\frac{|V|V}{D}-g\,\sin\alpha \tag{7}$$

where g is gravity and f is the hydrodynamic friction coefficient factor, which depends on the Reynolds number $\mathbf{Re} = \rho\,|V|\,D/\mu$, where μ is the absolute viscosity. In the turbulent regime the friction factor is also a function of the relative pipe roughness ε/D. To simplify the solution, the friction factor is approximated by its fully developed expression. For a laminar regime, $\mathbf{Re} < 2000$, it was specified as

$$f=\frac{64}{\mathbf{Re}}. \tag{8}$$

For the turbulent regime, $\mathbf{Re} > 2500$, the friction factor was approximated by Miller's correlation (Fox and McDonald, 2001),

$$f=0.25\left\{\log\left[\frac{\varepsilon/D}{3.7}+\frac{5.74}{\mathbf{Re}^{0.9}}\right]\right\}^{-2}. \tag{9}$$

Between $\mathbf{Re}$= 2000 and 2500, a linear variation of the friction factor with the Reynolds number was assumed, to avoid numerical instabilities due to abrupt changes in the friction factor from the laminar to the turbulent value.

The energy conservation equation can be written as

$$\frac{dT}{dt}=\frac{\beta T}{\rho c_p}\frac{dP}{dt}+\frac{f}{2c_p}\frac{V^2|V|}{D}-\frac{4U_e}{\rho c_p D}\left(T-T_{ref}\right) \tag{10}$$

where c_p is the specific heat at constant pressure, U_e is a heat transfer coefficient and T_{ref} is a reference temperature.

At the present work, two models were considered to account for the heat flux through the pipeline wall. The first one neglects the energy stored at the pipeline wall and insulation layers, and will be referred to as *Simdut_U*, while the second one, called *Simdut_W*, considers the energy stored in the pipe wall layers by solving the transient heat conduction equation. For each model a different reference heat transfer coefficient U_e and temperature T_{ref} were employed, as will be explained in the following sections.

Simdut_U

The reference temperature T_{ref} for *Simdut_U* is the external ambient temperature T_∞, which at the present case is the subsea water temperature. This model is based on the steady state overall heat transfer coefficient U_w (based on the inner diameter of the pipe), which was calculated ignoring the heat capacity of the pipeline:

$$U_e=\left[\frac{1}{h_{in}}+\frac{1}{U_w}\right]^{-1} \; ; \; U_w A_i=\left[\sum_{i=1}^{N} Res_i + Res_o\right]^{-1} \tag{11}$$

where Res_i and Res_o are the equivalent thermal resistance for the insulation layers and outer wall to the ambient resistance, defined in the following way

$$Res_i=\frac{D}{2k_i}\mathbf{ln}\left(\frac{D_{ei}}{D_i}\right) \quad ; \quad Res_o=\frac{D}{D_o}\frac{1}{h_o} \tag{12}$$

where D is the inner pipe diameter, D_e is the exterior diameter of the wall pipe. Each layer i has a thickness $e_i = D_{ei} - D_i$, where D_{ei} and D_i are the exterior and interior diameter of each insulating layer, respectively. $k_{,i}$ is the thermal conductivity of the pipeline layers. h_{in} and h_o are, respectively, the inner and outer convective heat transfer coefficients.

Simdut_W

In this approach U_e in Eq. (8) is the inner convective heat transfer h_{in}, and T_{ref} is T_{is}, the inner wall temperature, which was determined by solving the transient heat conduction equation, taking into account the thermal capacity of the materials. As already mentioned, to simplify the problem, the axial diffusion was neglected, and at each axial location, the radial conduction equation was solved

$$\rho_i\, c_{pi}\frac{\partial T_i}{\partial t}=\frac{1}{r}\frac{\partial}{\partial r}\left(r\,k_i\frac{\partial T_i}{\partial r}\right) \qquad 1\le i\le N \tag{13}$$

where T_i is the wall layer temperature, ρ_i, c_{pi} and k_i are the density, heat capacity, and thermal conduction of the material, respectively, and N is the number of layers including the pipe wall. These thermal properties were considered constant for each layer and were evaluated at the mean temperature between ambient and inlet temperature. The boundary conditions at the inner and outer walls of the main pipe are:

$$-k_1 \frac{\partial T_1}{\partial r} = h_{in}(T - T_{is}) \tag{14}$$

$$-k_N \frac{\partial T_N}{\partial r} = h_o(T_{os} - T_\infty) \tag{15}$$

where h_{in} and h_o are, respectively, the inner and outer convective heat transfer coefficients, T_{is} is the inner wall temperature, T_{os} is the outer wall temperature, T_∞ is the ambient temperature, and r is the radial coordinate.

Heat Transfer Coefficient

The outer convective heat transfer coefficient is independent of the pipeline flow condition, while the inner heat transfer coefficient h_{in}, between the gas or oil flowing in the pipeline and the inner pipe wall surface is computed during the solution of the flow field. It is a function of flow regime, flow properties (Prandtl number, $\mathbf{Pr}=\mu_f\, c_p /k$) and flow velocities (Reynolds number, **Re**). Both heat transfer coefficients were determined from the Nusselt number defined as $\mathbf{Nu_{in}} = h_{in}\, D / k$ and $\mathbf{Nu_o} = h_o\, D_o / k_o$ where the subscripts *in* and *o* refer to inside and outside of the pipeline, respectively.

Internal Nusselt number correlations for both forced and mixed (forced and natural) convection were employed to determine the temperature cool down profile. The correlations implement in the *Simdut* software, developed in the present work, were specified assuming fully developed flow for both laminar and turbulent regimes.

For laminar regime, $\mathbf{Re} \le 2000$, the forced convection Nusselt number corresponding to a constant surface temperature condition was defined as

$$\mathbf{Nu_{in}} = 3.66 \tag{16}$$

A limiting velocity was defined to neglect convection, and an equivalent Nusselt number ($\mathbf{Nu_{in}}$ =2) due only to conduction was prescribed.

To consider the effects of natural convection combined with the laminar forced convection, the Nusselt Number was approximated by the Olivier's correlation (Holman, 1983), for $\mathbf{Gr/Re^2} > 1$

$$\mathbf{Nu_{in}} = 1.75\,[\mathbf{Re\,Pr}\,(D/L) + 0.0083(\mathbf{Gr\,Pr})^{0.75}]^{1/3} \tag{17}$$

where *D/L* is the ratio between the pipe diameter and the pipeline length, being negligible for long pipes, and **Gr** is the Grashof Number, $\mathbf{Gr} = D^3\rho^2 g\beta|\Delta T|/\mu^2$, where ΔT is the temperature differential between fluid and pipe wall.

For turbulent regime, the correlation due to Gnielinski (Incropera and DeWitt, 1998) was utilized. This correlation is valid for $0.5 < \mathbf{Pr} < 2000$ and $2300 < \mathbf{Re} < 5\times10^6$.

$$\mathbf{Nu_{in}} = \frac{\mathbf{Pr}\,(f/8)\,(\mathbf{Re} - 1000)}{[1 + 12.7\sqrt{(f/8)}\,(\mathbf{Pr}^{2/3} - 1)]} \tag{18}$$

For the transition zone ($2000 < \mathbf{Re} < 2500$) the following linear profile between the laminar and turbulent Nusselt number was utilized: $\mathbf{Nu_{in}} = 0.01056\,\mathbf{Re} - 17.46$.

The outer convective heat transfer coefficient was calculated by correlations for forced convection over a pipe laying on the seafloor. At the present work the following correlation was employed (Knudsen and Katz, 1958)

$$\mathbf{Nu_o} = 0.0266\,\mathbf{Pr}_o^{1/3}\,\mathbf{Re}_o^{0.805} \tag{19}$$

where $\mathbf{Re}_o$ is the Reynolds Number based on seawater velocity and properties and diameter D_o.

NUMERICAL METHOD

The solution of the equation governing the flow field inside the pipeline was determined by a finite difference scheme, while the pipe wall temperature distribution was determined by a finite volume technique (Patankar, 1980). The spatial derivatives were approximated by a central difference scheme, and a fully implicit approach was adopted for the time integration. In this code, a full thermal model was considered, in which the coupled continuity, linear momentum and energy equations were solved simultaneously inside the pipeline employing a direct heptadiagonal algorithm. At each axial location, the transient radial conduction equation was solved along the pipeline wall (duct wall and insulation layers) by a tri-diagonal algorithm.

CASES STUDIED

Two cases were studied: *Case 1* consisted of oil flowing in a short pipe (L = 1 km) with diameter D = 6 in, while for *Case 2* gas flows in a L = 20 km pipeline with diameter D = 10 in. Both cases were studied considering as initial condition a steady state flow entering the pipeline at 60 ^{o}C, losing heat to the sea water at 5 ^{o}C, and discharging to a reservoir at P_{out}. The fluid was supplied to the line by a supply reservoir at P_s through a full open inlet valve as shown in Fig. 1. Cool down starts once the flow rate is cut down by closing the inlet valve. Due to the different pipeline sizes, the supply and discharging pressures were also different. For the oil, the following pressure were defined: P_s = 7.2 kgf/cm^2 and P_{out} = 1 kgf/cm^2, with initial mass flow rate equal to 44 kg/s, while for the gas flow, the line was pressurized, with P_s = 72 kgf/cm^2 and P_{out} = 49 kgf/cm^2, and initial mass rate of 23 kg/s. For both cases, the pipeline parameters were: thickness e = 12.7 mm, roughness $\varepsilon = 0.01$ mm, Young's modulus of elasticity of the pipe material E = 2.1×10^5 MPa, and Poisson's ratio elasticity modulus, ν = 0.3. The insulation thickness was set as 200 mm. The pipewall density, thermal conductivity and specific heat at constant pressure were defined as ρ_1=7800 kg/m^3, k_1=50 W/(m K), c_{p1}=500 J/(kg K), respectively, while the insulation properties were: $\rho_2 = 52$ kg/m^3, $k_2 = 0.38$ W/(m K), $c_{p2} = 657$ J/(kg K).

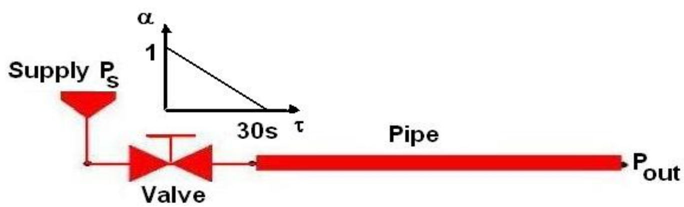


Figure 1. Pipeline configuration.

The oil reference density was set as $\rho = 883$ kg/m^3, speed of sound a = 1286 m/s and thermal expansion coefficient $\beta = 7.88 \times 10^{-4}$ K^{-1}, while the gas specific gravity SG = 0.6 and average molecular weight $\overline{M}$ = 17.4 kg/kgmol. The other fluid properties are listed in Table (1).

Table 1. Fluid properties

	Absolute viscosity (cP)	Heat capacity [J/(kg K)]	Thermal conductivity [W/(m K)]
oil	25.7	1831	0.1313
gas	1.05×10^{-2}	2244	0.0301

The seawater parameters utilized were: stream velocity of 1 m/s, thermal conductivity of 0.59W/(m K), density of 1025 kg/m^3, viscosity of 1.08×10^{-3} kg/(m s) and $\mathbf{Pr}_o$ = 8.81 resulting in an outer heat transfer coefficient of 2000 W/(m^2 K) and an overall heat transfer coefficient U_w equal to 3.36 W/(m^2 K).

Based on a grid test, so that temperature differences smaller than 1% were obtained, a 10s time step, and 100 nodes were defined with only one control volume specified at the pipe wall and 5 control volumes at the insulation layer. In all analyses, the simulation time was 12 hours long. To evaluate the cooldown time with the different softwares, it was considered that the critical temperature was equal to 15 ^{o}C.

For comparison, the cases were simulated with the models described in previous section and with the commercial softwares *Pipeline Studio (TLNET* and *TGNET)* and *OLGA*.

Figure 2 illustrates the initial temperature profile along the pipeline, corresponding to the steady state condition obtained with all softwares. It can be seen that the oil results present an excellent agreement. The gas results obtained by *Simdut* and *Olga* also agree very well, and a small discrepancy can be seen with the *TGNET*, which has a lower order of approximation scheme. As expected, both formulations *W* and *U* do not differ for the steady state condition.

Axial temperature profiles obtained 4 hours after closing the valve are shown at Fig. 3, for both cases. Analyzing the axial temperature distribution in Figs. 3a, it can be seen that all models predicted a very small oil temperature drop along the pipeline. The results obtained using *TLNET_U*, *Simdut_U* and *Olga_U* were once again almost coincident. It can also be seen that the model *OLGA_W* predicted a slower cooldown than all others models. Figure 3b corresponds to the gas results, and it can be seen that after the flow rate was cut down, the temperature profiles show an abrupt drop near the entrance of the pipeline. Analyzing Fig. 3b, a large discrepancy among the softwares can be seen. Even for the case where the heat capacity is neglected. For that case *Simdut_U* predicts higher temperatures near the entrance. Barrera (2005) showed that the difference between the softwares is related to the correlation employed to determine the heat loss to the ambient and the limit of validity of each regime. It should be mentioned here that after a very long time interval, all softwares predict the same temperature distribution (i.e., equal to the sea water temperature, T_∞).

Figure 4 shows the temperature variation with time in a section located in the middle of the pipe obtained with the models described here and with the two commercial softwares. The solutions obtained with *Simdut_U*, *TGNET_U* and *Olga_U* for both oil and gas showed a very good agreement since the thermal capacity of the pipe and insulating layers were neglected. At the mid section of the pipelines, they predicted less than six hours for the oil to reach the critical temperature, while the gas took less than one hour. An analysis of Fig. 4 shows that the heat capacity of the pipe wall and insulation (*Simdut_W*) slows down the cool down process, substantially increasing the time to reach 15^{o}C in relation to the solution of *Simdut_U*, which neglects this contribution. As already mentioned, there is a time delay between the responses of the 3 softwares. The solution obtained with *Olga_W* is the least conservative of all, with a much higher time interval to reach the critical temperature, regardless the fluid.

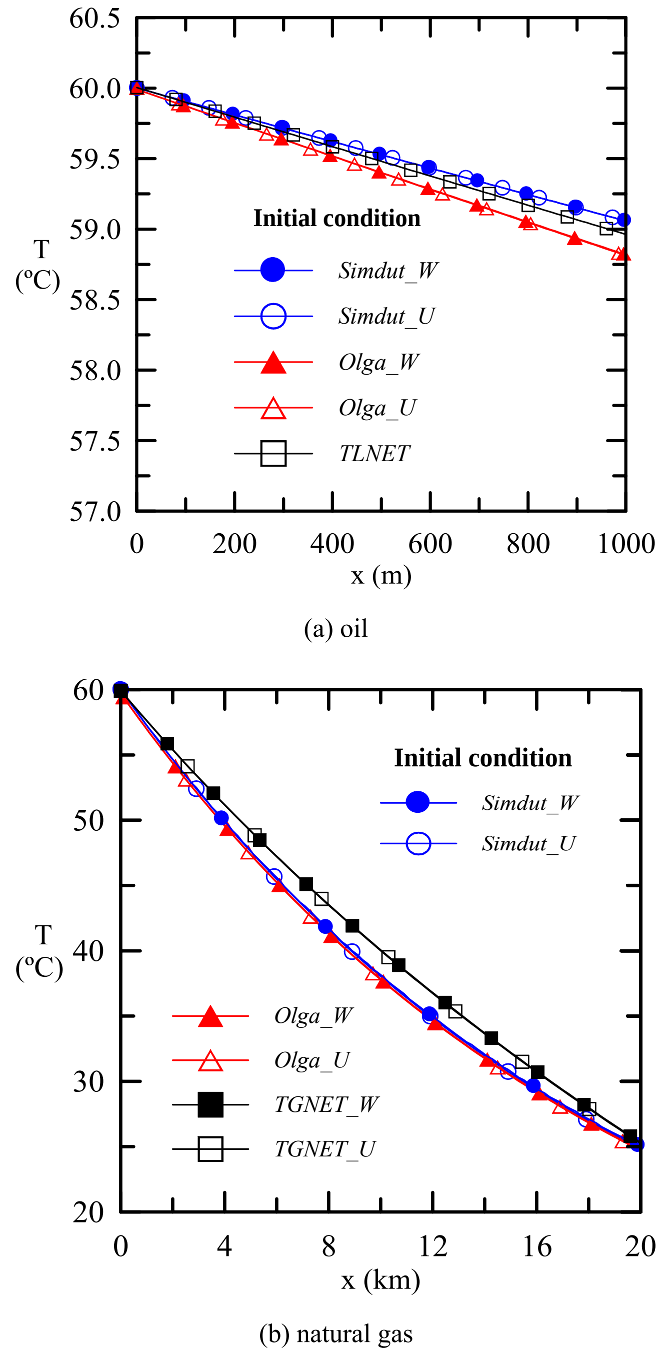
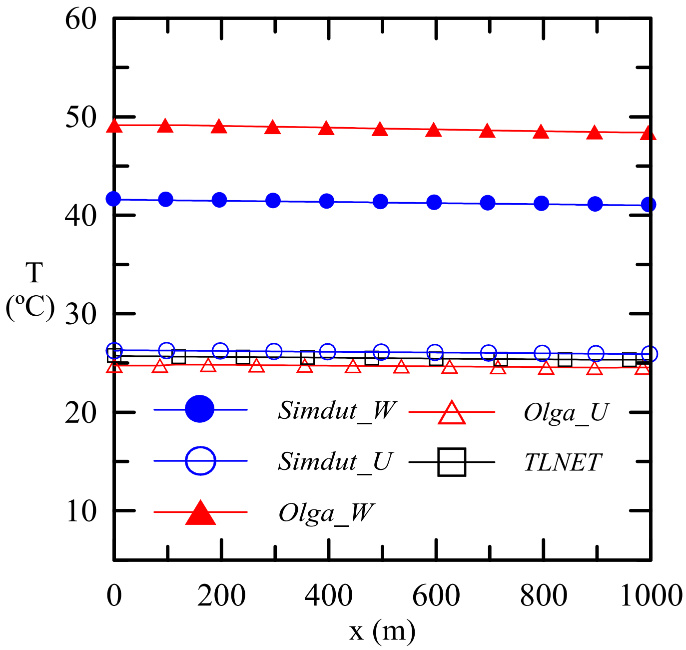
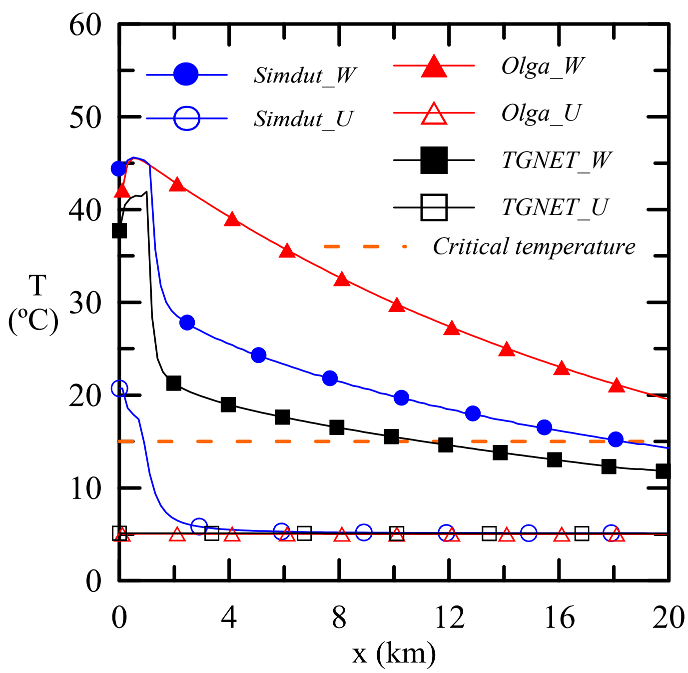


Figure 2 – Axial temperature distribution. Initial condition

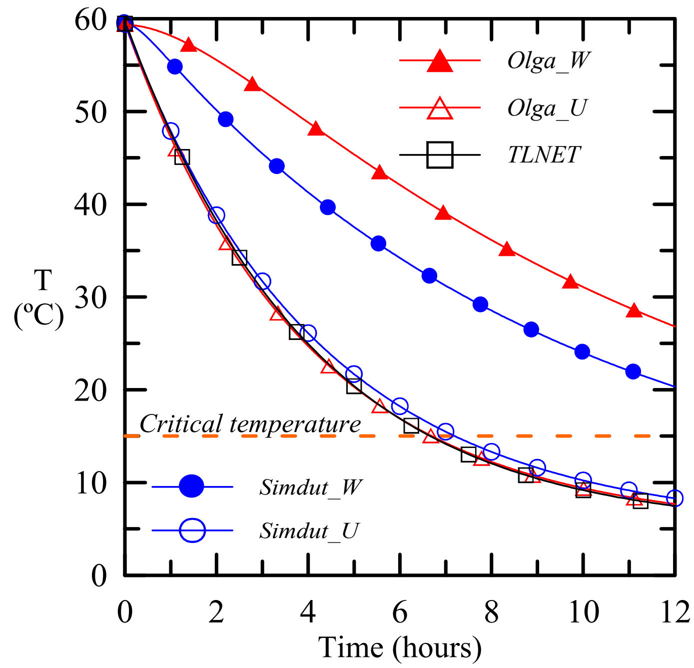


(a) oil

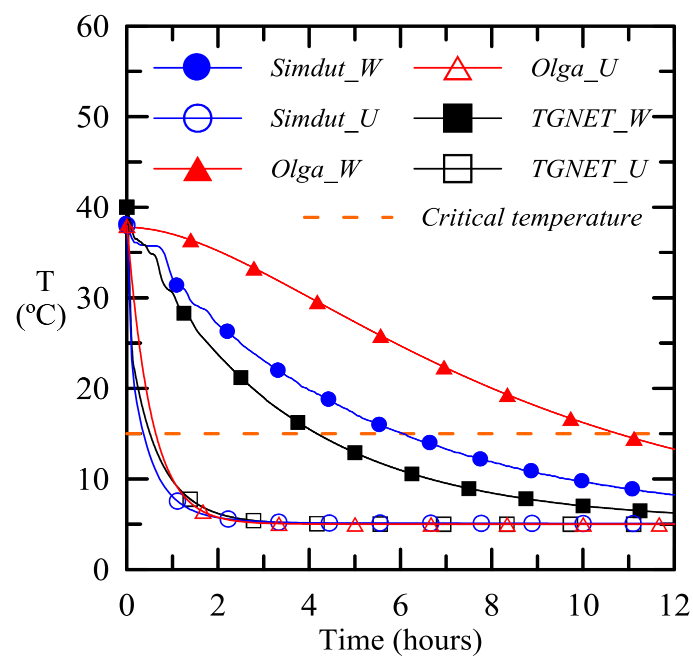


(b) natural gas

Figure 3 – Axial temperature distribution, after 4 hours.



(a) oil



(b) natural gas

Figure 4 – Time temperature evolution at center of pipeline.

To illustrate the influence of the internal Nusselt correlation on the temperature distribution inside the pipeline, the same problem was solved for the gas, employing different limit velocity to neglect convection. Figure 5 shows that the limit velocity to employ the pure conduction approximation plays a significant role in the temperature drop. Since the “conduction” Nusselt number is almost half of the forced convection $\mathbf{Nu}_{\mathbf{in}}$, a smaller temperature drop is obtained. Further, it can be seen that the *Simdut_W* solution agrees with the *Olga_W* solution when *V_limit* = 0.1 m/s, explaining why *Olga_W* presented a smaller temperature drop.

The influence of the insulation parameters in the flow can be appreciated in Figs. 6 and 7 for the oil and in Fig. 8 for the gas, where the time temperature evolution at the center of the pipeline is shown. The solution was obtained with *Simdut_W* and *Simdut_U*, maintaining the same fluid, pipeline, initial and boundary conditions, but varying the insulation parameters, to represent new and old thermal insulations. The entrainment of water in the insulation as time passes leads to a great increase in the thermal conductivity, while only a moderate increase is observed for the density and heat capacity (Jackson et al., 2005).

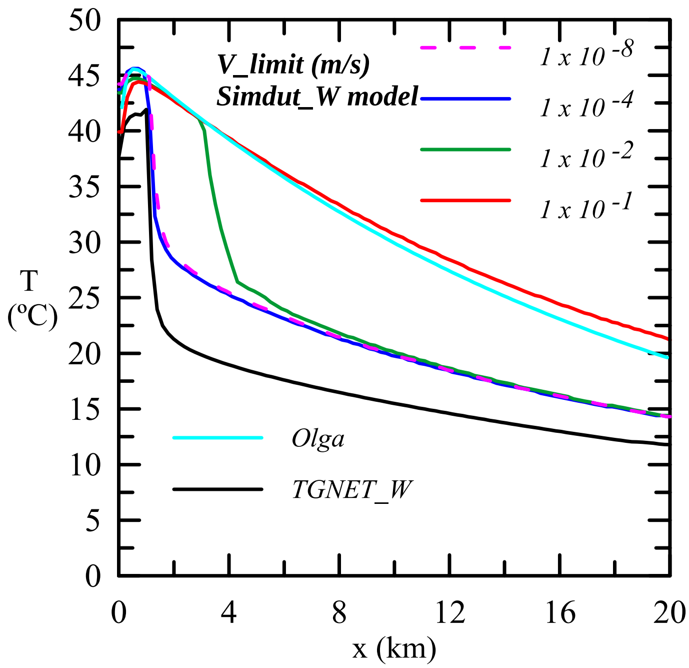


Figure 5 – Gas axial temperature distribution, after 4 hours. Influence of correlations.

For the first test shown in Fig. 6, the thermal diffusivity α was kept constant and the conductivity k was varied. The reference parameters for the base case are: k_{iso}/k =2.89 and α_{iso}/α= 136. The reference value k_{iso}/k was divided and multiplied by 10. It can be seen in Fig. 6 that, as expected, there is a smaller temperature drop when the conductivity decreases.

Figure 7 illustrates the effect of the thermal diffusivity on the temperature variation with time at x= 500 m. The thermal conductivity ratio was kept constant. The reference α was divided by 20, and since the temperature variation was small with larger α values, a very large value (equivalent to zero $\rho\, c_p$) was specified. The influence of the thermal capacity ($\rho\, c_p$) can be observed in Fig. 7, where a smaller temperature drop can also be observed when α decreases. In Fig. 6, k was varied maintaining α constant; therefore, $\rho\, c_p$ was also varied. Since α is the ratio of these two quantities and both have the same influence on the temperature, there is an optimum value of k for a given α, as one can see by the reduction of the influence of the stored energy as k increases and decreases from the reference case.

Still analyzing Fig. 7, one can compare the prediction obtained with *Simdut_U,* which employs a steady state over-all heat transfer coefficient and it is independent of α (since it neglects the thermal capacity of the pipe and insulation) and the prediction of *Simdut_W* neglecting the thermal capacity of the pipe and insulation. It can be observed that the solution obtained neglecting the thermal capacity of the pipe wall and insulation ($\alpha_{iso}/\alpha = \alpha_w/\alpha = \infty$) and the *Simdut_U* prediction are similar. The models predict a temperature distribution with a difference of approximately 2 °C. These curves should coincide and the difference can be attributed to the radial mesh distribution employed, since when the mesh was refined in the radial direction, the difference was reduced.

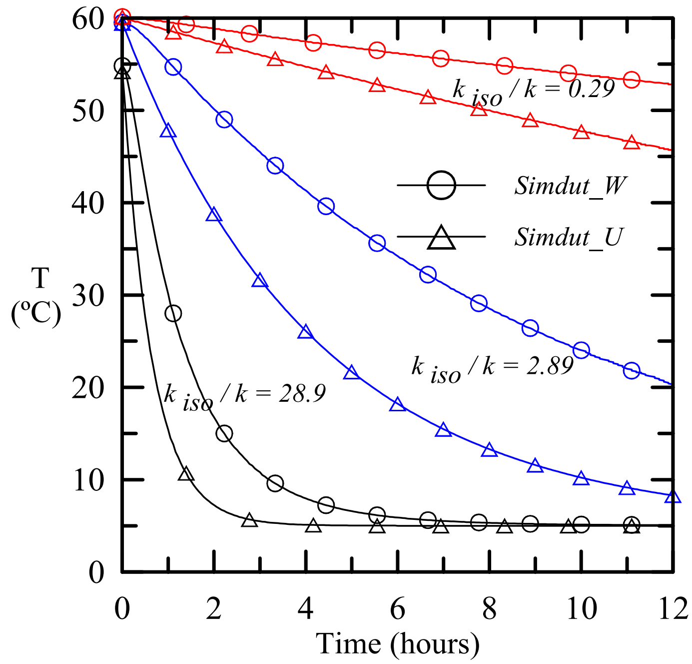


Figure 6 – Oil. Time temperature evolution at center of pipeline Influence of insulation thermal conductivity.

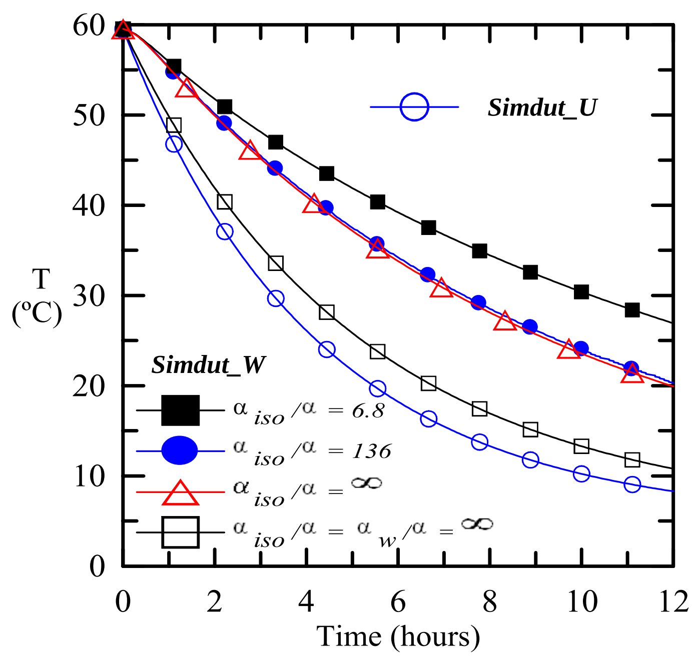


Figure 7 – Oil. Time temperature evolution at center of pipeline Influence of insulation thermal diffusivity

An additional curve was also included in Fig. 7. corresponding to zero thermal capacity not only for the insulation, but also for the pipe. It can be observed that since the thermal capacity of the insulation is already small, neglecting it does not influence the temperature distribution. Surprisingly, the influence of the thermal capacity of the pipe wall is significant even though its thickness is small. This can be explained by its large product $(\rho\, c_p)_w$. At the mid section of the pipe, a temperature difference of 10 ^{o}C can be observed after 12 hours of cooling due to the thermal capacity of the pipe wall.

The influence of the thermal diffusivity of the insulation on the gas flow can be seen in Fig. 8, where the time evolution of the temperature at the center of the pipeline is shown. The reference parameter for this case is $\alpha_{iso}/\alpha = 35.4$. The insulation was varied keeping the gas α constant, and as in the previous case, the ratio k_{iso}/k was also kept constant. Again, as in the oil case, a smaller temperature drop was obtained as α_{iso} diminishes, i.e., as $(\rho\, c_p)_{iso}$ increases. The reference α_{iso} value was divided by 20 (α_{iso}/α= 1.77), and since the temperature variation was smaller with larges α_{iso}, a very large value was tested (corresponding to a negligible $\rho\, c_p$). The results obtained with *Simdut_U,* based on the overall heat transfer coefficient, were independent of α (since it neglects the thermal capacity of the pipe and insulation). The solution neglecting the thermal capacity of the insulation as well as the pipe was also added in the graph. It can be seen that the solution with the overall heat transfer coefficient agreed perfectly with the solution obtained neglecting the thermal capacity of the insulation and pipewall. This result indicated that the energy stored at the pipewall can be more significant than that stored at the insulating layer.

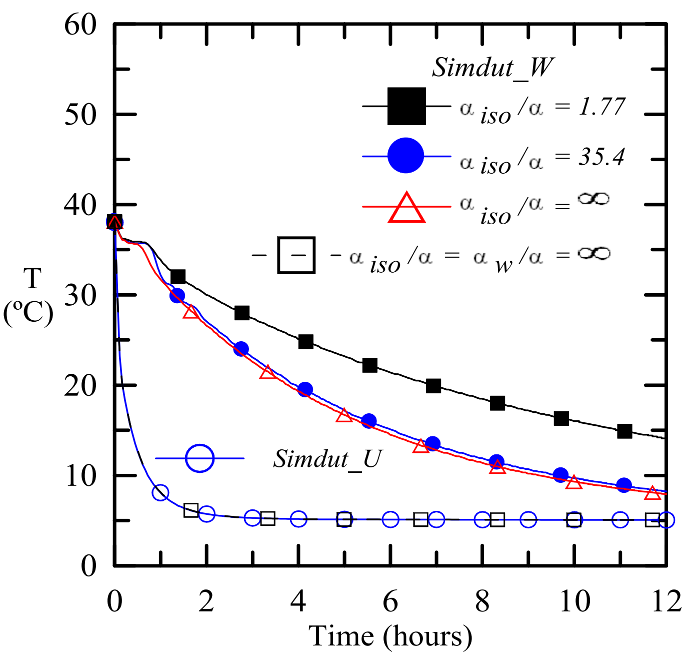


Figure 8 – Gas. Time temperature evolution at center of pipeline Influence of insulation thermal diffusivity

FINAL REMARKS

A transient thermal model has been developed to solve the temperature distribution within the wall of a pipeline. The thermal performance of insulated pipelines was investigated, considering both a detailed model for the temperature distribution within the wall and using an overall heat transfer coefficient ignoring the thermal capacity of the wall.

Observations show that a smaller temperature drop is obtained when the energy stored at the pipe wall and insulation is considered, regardless if the fluid flowing inside the peipeline is liquid or gas. These results illustrate the importance of accounting for the transient thermal properties in the cooldown calculations. As a conclusion, it can be stated that it is essential to account for the thermal capacity of the wall layers that compose the pipeline since they significantly influence the cooldown time.

The relative influence of the insulation for different operating conditions of the line remains to be evaluated. However, even though a simple test was performed, it is already clear that it is essential to address correctly the energy stored in the insulation and in the pipe wall, and that the use of a steady state overall heat transfer coefficient should be avoided.

It is clear that not only the correlations employed to determine the Nusselt number are crucial to the temperature prediction, but their limit of validity. These results indicate that there is a need to better determine these correlations, which are based on steady state and fully developed regime. Either experimental measurements must be obtained, or more precise simulations, with reduced number of hypothesis must be performed, in order to determine better approximations for the Nusselt number.

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