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NUMERICAL ANALYSIS OF LINEAR THERMAL EXPANSION OF PIPES DURING THE THERMAL TRANSIENT

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Abstract. *Thermal expansion is a problem the pipe engineers often have to deal with, since it is important to control the total linear thermal expansion and minimize stresses and forces associated with this physical phenomenon. The design of pipe systems is guided by standards, being ASME B31.3 (2010) certainly the most used in Brazil and in the United States. For this standard the pipe system, from the thermal standpoint, is evaluated based on a constant and uniform design temperature, being this normally referenced by the steady state value. This criterion is sufficient to ensure the pipe structural integrity due to the way the admissible and acting stresses are established; however, the standard is silent regarding the thermal transient and the pipe linear thermal expansion during this period. Such thermal expansion may be associated with forces transmitted by the pipe to equipment and structures, neglected by ASME B.31.3 and by the literature. This paper presents the equations involved in the pipe thermal transient problem. Based on the results of numerical simulations and the procedures normally used by Pipeline Engineering to compute forces, a discussion is made about the forces associated with the thermal expansion in the period of the thermal transient. It was found that the higher the flow velocity, the Nusselt number and the thermal diffusivity of the pipe material, the higher the pipe heating rate. Besides, it was found that the higher the heating rate and the thermal expansion coefficient, the higher the linear thermal expansion rate of the pipe. Moreover, the force associated with the linear thermal expansion starts to be transmitted to the pipe fixed point (anchor or stop) in an abrupt way and increases intermittently until its maximum value, then falls to its minimum value in an extremely abrupt way, being this minimum value the load obtained in the usual pipe load analysis.*

Keywords: *thermal transient, pipe, thermal expansion, loads.*

1. INTRODUCTION

A given pipe whose service is to transport a fluid at a temperature different from the ambient will expand (or contract) due to the temperature gradients in its wall, being this gradient consequence of temperature and characteristics of the internal flow and the boundary conditions external to the pipe. The thermal expansion, which is a phenomenon explained by the Thermodynamics and the Molecular Physics, is directly related to body temperature.

In the context of Pipeline Engineering, the thermal expansion is a problem the engineers often have to deal with, as part of the role of these professionals is to control the total linear thermal expansion and minimize stresses and forces associated with this physical phenomenon. However such professionals, due a gap in the literature and the standards that govern the design of pipe systems, which traditionally addressed the problem only in steady state, often have doubts about how the pipe behaves during thermal transient, driving them to associate, for example, high design factors for the loads related to thermal expansion or even ignore them.

This paper investigates the transient heat transfer problem in pipes due to internal flow and external convective boundary conditions. It presents the suitable continuity and the momentum and energy conservation equations that were solved numerically based on a physical model that simulates a real installation where, after $50\,m$ from its entrance, a process pipe is routed through a pipe rack. Knowing the pipe temperature as a function of position and time, it is possible to associate the thermal transient to the pipe linear thermal expansion in order to discuss the influence of the main parameters and properties and, based on the methodology classically used by the Pipeline Engineering to compute loads, how the thermal origin loads are transferred to anchors and stops.

2. THERMAL TRANSIENT IMPORTANCE FOR PIPE DESIGN

The design of pipe systems are guided by standards set by associations officially linked to a given country, which can have also a global reach, and design criteria developed by engineering offices.

ASME B31.3 (2010) standard is certainly the most used in Brazil and the United States, being US the country where this standard was originated. It aims to establish best practices honed over decades – the first publication of ASME B31 series is 1935 – to minimize the risks associated with the design and the operation of piping systems.

In the case of the structural design of the pipe system, ASME B31.3 fundamentally is concerned with protecting the pipe laying the allowable stresses and the manner how the stresses acting on piping system must be calculated, not caring about the forces that the piping system apply to equipment and structures due to thermal expansion even if the restrictions imposed by these ones are considered in the calculations. From ASME B31.3 point of view, the piping system design can be perfectly approved, even if the forces that the pipe applies on equipment and structures are unacceptable. The system, from the thermal standpoint, is evaluated based on a constant and uniform design temperature, being this normally referenced by the steady state value. This criterion is sufficient to ensure the pipe structural integrity due to the way the admissible and acting stresses are established by this standard.

It will be demonstrated below, by illustrating simulations performed in the software Caesar II (2015), how the load applied by the pipe to the structure can be influenced by the thermal transient. Caesar II is in compliance with ASME B31.3, among others codes, and is one of the most used worldwide software in piping design.

Figure 1. Modeled pipe in Caesar II.

The model of Fig. 1 is a representation of a 200 meter length straight pipe with a stop located at the middle of this pipe represented by the node 110. The other nodes have simple supports. The stop, which is a pipe support, is physically mounted as shown in Fig. 2, where it is seen a sectioned pipe supported by a beam and metallic profiles welded to the pipe, constituting the stop support that restricts the pipe axial displacement. Stops and anchors are indispensable pipe supports, as they direct the thermal expansion by fixing the pipe in a specific point.

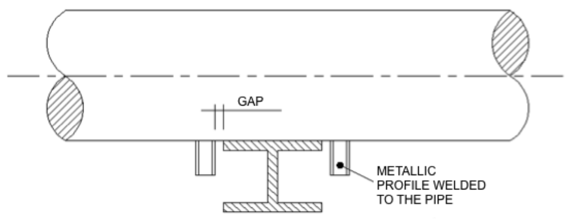


Figure 2. Representation of the stop from the elevation view.

Silva Telles (1999) shows the classical methodology used in Pipeline Engineering to compute loads and stresses for 3D layouts. The pipe is considered as a structural element in which axial forces do not cause strains and the friction coefficient is constant. For a 1D layout, as in the illustrative examples, Fig. 3 shows how the loads are computed.

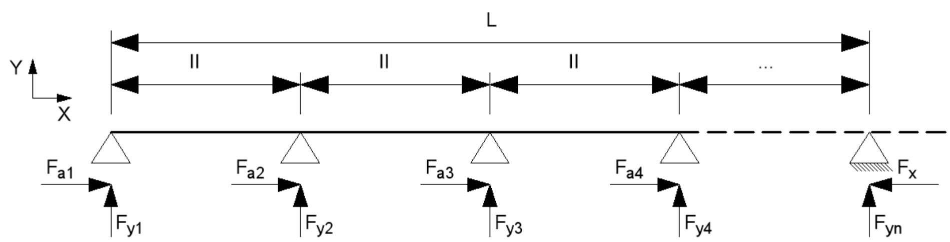


Figure 3. Computed loads representation for a 1D layout.

The vertical forces are calculated as:

$$F_{y1} = 0.5 \left[\frac{P}{(n-1)} \right] \tag{1}$$

$$F_{y2} = F_{y3} = F_{y4} = F_{yn} = \frac{P}{(n-1)} \tag{2}$$

where

$$P = \frac{\pi}{4} g L \left[\rho_f {D_i}^2 + \rho_p \left({D_e}^2 - {D_i}^2 \right) \right] \tag{3}$$

The horizontal forces (friction) on simple supports are calculated as:

$$F_a = \mu F_y \tag{4}$$

Finally the total horizontal load due to the friction acting at the pipe stop or anchor, computed only for the section L presented in the Fig. 3, is calculated as:

$$F_x = \sum_{n=1}^{n-1} F_{an} \tag{5}$$

In the first simulation all the pipe is uniformly heated at design temperature, as per ASME, and the stop has no gap. Table 1 shows the Caesar II output, where the node 110 is where the stop is located.

Table 1. Pipe uniformly heated.

Node	Load Case	$F_x\ (N)$	$F_y\ (N)$	$F_z\ (N)$	$M_x\ (Nm)$	$M_y\ (Nm)$	$M_z\ (Nm)$
110		Rigid +Y; Rigid X					
	(OPE)	0	-6573	0	0	0	0

It is found that the software calculated a zero axial load F_x as a consequence of the model symmetry and the steady state, once the upstream and the downstream linear thermal expansion from the anchor is maximum and therefore the friction loads, which are a consequence of the pipe displacement on the support beam, cancel out.

In the second simulation, half of the pipe among the nodes 10 to 110 experiences an abrupt heating, while the other half is at ambient temperature. The stop has no gap. Results are shown in Tab. 2.

Table 2. Abrupt heating with no gap stop.

Node	Load Case	$F_x\ (N)$	$F_y\ (N)$	$F_z\ (N)$	$M_x\ (Nm)$	$M_y\ (Nm)$	$M_z\ (Nm)$
110		Rigid +Y; Rigid X					
	(OPE)	18732	-6573	0	0	0	0

In this simulation the pipe applies $18732\,N$ to the structure by the intermediate stop. This load is due to the friction forces calculated from the node 10 to 100 due to the thermal expansion. This condition is never evaluated once the calculation is done for the steady state with the design temperature defined by the standard. Such abrupt heating is not real, however it is feasible that there is a time instant, as will be demonstrated by this paper, in which the pipe upstream side from the stop will experience a differential of temperature while the downstream side remains at ambient temperature.

Caesar II simulations illustrate the importance of the thermal transient when a pipe system is analyzed, especially regarding the loads transmitted by the pipe to structures and equipment, being these loads not part of the ASME B31.3 scope.

3. MATHEMATICAL EQUATIONS

The mathematical model considers a internal steady state flow through a pipe. Initially the fluid and the pipe temperature are both ambient. Suddenly the temperature of the fluid entering the pipe is increased (Zargary and Brock, 1973). The external boundary conditions are convective and the ambient temperature remains constant.

Fundamentals assumptions used to define the mathematical model are:

- Physical properties are constant.
- Flow is turbulent, fully developed and incompressible.
- Viscous dissipation and axial heat conduction in both pipe and fluid are negligible comparing with the heat transfer to the pipe wall.
- Inlet temperature is uniform.
- External convection is natural and uniform.
- Potential energy variations are negligible.

The mathematical model equations are presented below. In these equations temperature T, pressure p and velocity u are time averaged for turbulent flow, while they are instantaneous for laminar flow.

3.1 Continuity equation

$$u = u(r) \tag{6}$$

$$u_\theta = u_r = 0 \tag{7}$$

3.2 Momentum conservation equation

$$\frac{1}{r}\frac{\partial}{\partial r}\left[(\nu + \varepsilon_M)\, r \frac{\partial u}{\partial r}\right] = \frac{1}{\rho_f}\frac{\partial p}{\partial x} \qquad (0 \le r \le r_0) \tag{8}$$

where, for laminar flows, the momentum eddy diffusivity ε_M is zero. Boundary conditions are:

$$\left(\frac{\partial u}{\partial r}\right)_{r=0} = 0 \tag{9}$$

$$u(r_0) = 0 \tag{10}$$

The momentum eddy diffusivity ε_M used is the Reichardt's correlation (Kays and Crawford, 1993):

$$\frac{\varepsilon_M}{\nu} = \frac{0.4 y^+}{6}\left(1 + \frac{r}{r_0}\right)\left[1 + 2\left(\frac{r}{r_0}\right)^2\right] \tag{11}$$

$$y^+ = (r_0 - r)\,\frac{U\sqrt{f/8}}{\nu} \tag{12}$$

The term $\frac{\partial p}{\partial x}$ is the head loss per unit length and for turbulent flow it is written in terms of the Darcy friction factor f, which in turn is calculated by using Swamee and Jain correlation (Potter *et al.*, 1997):

$$\frac{\partial p}{\partial x} = -f\,\frac{\rho_f\, U^2}{2\, D_i} \tag{13}$$

$$f = 1.325\left\{\ln\left[0.27\left(\frac{e}{D_i}\right) + 5.74\left(\frac{1}{Re}\right)^{0.9}\right]\right\}^{-2} \qquad 5000 < Re < 10^8 \tag{14}$$

3.3 Energy equation

3.3.1 For the fluid

By applying Eq. (7) and assumptions, it results:

$$\frac{\partial T_f}{\partial t} + u\frac{\partial T_f}{\partial x} = \frac{1}{r}\frac{\partial}{\partial r}\left[r\,(\alpha_f + \varepsilon_M)\,\frac{\partial T_f}{\partial r}\right] \qquad (0 \le r \le r_0) \tag{15}$$

where, for laminar flows, the thermal eddy diffusivity ε_H is zero. The initial and boundary conditions are:

$$T_f\,(x, r, 0) = T_\infty \tag{16}$$

$$T_f\,(0, r, t) = T_{me} \tag{17}$$

$$k_f\left(\frac{\partial T_f}{\partial r}\right)_{r=r_0} = h_i\left[T_p\,(x, r_0, t) - \bar{T}_f\right] \tag{18}$$

$$\left(\frac{\partial T_f}{\partial r}\right)_{r=0} = 0 \tag{19}$$

Internal convection heat transfer coefficient is calculated using Gnielinski's correlation (Kays and Crawford, 1993):

$$h_i = \frac{k_f}{D_i}\frac{(Re - 1000)\; Pr\; (f/8)}{1 + 12.7\,(f/8)^{1/2}\,\left(Pr^{2/3} - 1\right)} \qquad 0.5 \le Pr \le 2000 \qquad 2300 \le Re \le 5 \times 10^6 \tag{20}$$

The average fluid temperature $\bar{T}_f$ is calculated as:

$$\bar{T}_f(x, t) = \frac{2}{U\, {r_0}^2}\int_0^{r_0} u\,(r)\; T_f\,(x, r, t)\; r\, dr \tag{21}$$

The friction coefficient, f, is calculated by using Eq. (14). Reynolds analogy ($\varepsilon_H = \varepsilon_M$) is used in this study.

3.3.2 For the pipe

$$\frac{\partial T_p}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}\left(r\,\alpha_p \frac{\partial T_p}{\partial r}\right) \qquad (r_0 \le r \le r_e) \tag{22}$$

The initial and boundary conditions are:

$$T_p\,(x, r, 0) = T_\infty \tag{23}$$

$$k_p \left(\frac{\partial T_p}{\partial r}\right)_{r=r_0} = h_i \left[T_p\,(x, r_0, t) - \bar{T}_f\right] \tag{24}$$

$$k_p \left(\frac{\partial T_p}{\partial r}\right)_{r=r_e} = h_e \left[T_\infty - T_p\,(x, r_e, t)\right] \tag{25}$$

External convection heat transfer coefficient is calculated using Churchill and Chu (Incropera and De Witt, 2003) correlation:

$$h_e = \frac{k_{air}}{D_e}\left\{0.6 + \frac{0.387 Ra^{1/6}}{\left[1 + (0.559/Pr)^{9/16}\right]^{8/27}}\right\}^2 \qquad Ra \le 10^{12} \tag{26}$$

$$Ra = \frac{g\beta\,(T_p - T_\infty)\,D_e^3}{\nu\alpha} \tag{27}$$

$$\beta = \frac{1}{T} \tag{28}$$

3.4 Solution methodology

Equation (8) was first solved and this solution was introduced in Eqs. (15) and (22), which were solved simultaneously by a finite-difference procedure. As pointed by Kawamura (1976), all properties were evaluated at T_{me} as the calculated Reynolds numbers are not small and the considered domain is far from the entrance region.

4. TRANSIENT THERMAL EXPANSION

After knowing the temperature distribution as a function of time, the transient thermal expansion is calculated as:

$$\delta(x, t) = \int_0^x \varphi\left[\bar{T}_p(\xi, t) - \bar{T}_p(\xi, 0)\right] d\xi \tag{29}$$

where the pipe average temperature, $\bar{T}_p$, is calculated as:

$$\bar{T}_p(x, t) = \frac{2}{{r_e}^2 - {r_0}^2}\int_{r_0}^{r_e} T_p(x, r, t)\, r\, dr \tag{30}$$

Steel and copper are considered as pipe materials. Below are presented the correlations used for calculate the pipe linear thermal expansion coefficient, being these equations adapted from ASME B31.3 (2010).

The steel total linear thermal expansion is correlated as:

$$\varphi = 0.0077\,\bar{T}_p + 10.721 \tag{31}$$

The copper total linear thermal expansion is correlated as:

$$\varphi = 0.0058\,\bar{T}_p + 16.651 \tag{32}$$

In Eqs. (31) and (32) the linear expansion is in $\mu m/(m\,^oC)$ and the average pipe temperature $\bar{T}_p$ is in ^oC.

5. CASE STUDY

Figure 4 shows the geometry used to make the thermal study. The pipe is anchored at the points A and B and the flexible element allows the pipe to expand freely. Domain of study is between $x = 50\,m$, the pipe rack entrance, and $x = 200\,m$, the middle of the pipe rack where the anchor is located. Equation (29) can be rewritten for this case as:

$$\delta(x, t) = \int_{200}^x \varphi\left[\bar{T}_p(\xi, 0) - \bar{T}_p(\xi, t)\right]\, d\xi \qquad (50 \le x \le 200\,m) \tag{33}$$

The pipe rack is symmetric and has a length of $300\,m$ with a span of $5\,m$ between beams. As the total thermal expansion is a very important design criterion, it is fixed to $130\,mm$, being the reason why the pipe that will expand has a length of $150\,m$.

Water is considered as the fluid and steel and copper are considered as the pipe materials. The flow velocity range is $2\!-\!5\,m/s$, the recommended velocity for process pipe (Silva Telles, 1999). The average inlet temperature is $T_{me} = 373\,K$ $(100°C)$ and the ambient temperature is $T_\infty = 298\,K$ $(25°C)$. Pipes are standard, as per ASME B.36.10.

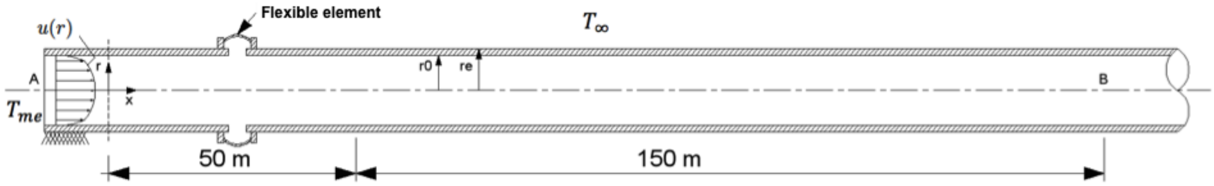


Figure 4. Geometry for case study.

6. TRANSIENT THERMAL EXPANSION ANALYSIS

After demonstrating the loads related to the thermal transient by Caesar II simulations, it must be pointed out that the considered abrupt heating is not real, as will be shown. In terms of the external forces to the pipe, friction and normal reaction are the only present in the model of Fig. 4 and, therefore, are the only ones considered in this study.

Figure 5(a) shows the pipe thermal expansion profile for $D_n = 200\,mm$ steel pipe with average velocity $U = 3\,m/s$. It is shown how the pipe expands from the position $x = 200\,m$, where the pipe is anchored (see Fig. 4). It is found that in $20\,s$ the length between $50\,m \leq x \leq 75m$ expanded. At $t = 40s$ the length between $50\,m \leq x \leq 155\,m$ expanded while at about $t = 48\,s$ all the upstream pipe from the anchor tends to displace whereas the downstream pipe is at ambient temperature. At about $t = 108\,s$ the upstream pipe reaches the steady state. It should be noted that the reference for $t = 0$ is $x = 0$.

Figure 5(b) was obtained considering the same properties and pipe diameter used in the Fig. 5(a) simulation, except that $U = 5\,m/s$. This simulation illustrates the effect of velocity on the studied thermal transient.

By increasing the average flow velocity, the time the pipe takes to achieve the maximum thermal expansion δ is about $65\,s$. At about $t = 26\,s$ the upstream pipe tends to expand and from this instant to the maximum thermal expansion it takes $39\,s$ ($\sim 60s$ with $U = 3\,m/s$). Such a result shows that by increasing the average velocity the thermal expansion rate will be higher, firstly due to the shorter time to fill the pipe with heated fluid and also due to the higher Reynolds number and, therefore, the higher Nusselt number.

Figure 5(c) was obtained considering the same parameters used in the Fig. 5(a) simulation, except that the pipe diameter is $D_n = 100\,mm$. This simulation illustrates the effect of the diameter with respect to studied thermal transient.

By comparing Fig. 5(a) and 5(c), it is found that in both two cases it takes about $48\,s$ for the upstream pipe to begin to displace, what is not a surprise as the average velocities are equal. However, even though the $D_n = 200\,mm$ pipe has a thicker wall and a higher Grashof number, it is found that the $D_n = 200\,mm$ pipe heat rate is slightly higher as a consequence of the higher Reynolds and Nusselt number, what is confirmed by the Lin and Kuo (1988) and Yan *et al.* (1989) conclusions. The maximum thermal expansion rates are similar (about $3\,mm/s$).

Figure 5(d) was obtained considering the same parameters used in the Fig. 5(a) simulation, except that now the pipe material is copper. This simulation illustrates the effect of the pipe material thermal diffusivity α_p and the thermal expansion coefficient φ on the thermal expansion rate.

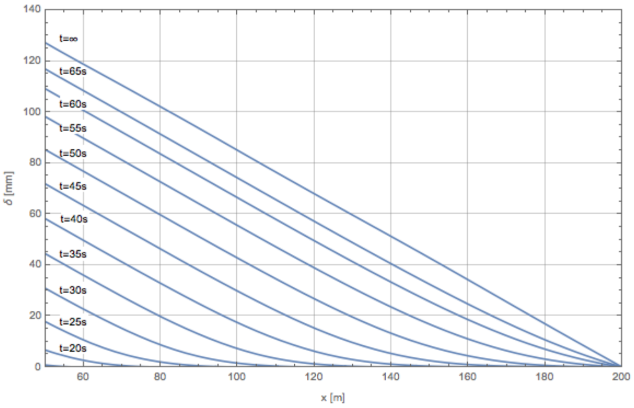
As the Grashof number and the average velocity are equal in both simulations and thermal diffusivity is greater for copper than for steel, the copper pipe heating rate is higher. In order to respect the design criteria by fixing the maximum thermal expansion in $126\,mm$, which is the maximum calculated expansion for the steel pipe, the pipe length now is $98\,m$ (the anchor is located at $x = 148\,m$). It is found that the copper pipe takes $75\,s$ to reach the maximum thermal expansion while the steel pipe takes about $108\,s$ to reach the same expansion. This observation shows that the greater the thermal expansion coefficient the greater the thermal expansion rate. However, as the pipe length is shorter and therefore the volume is smaller, not necessarily the load on the anchor will be higher.

6.1 The inlet temperature effect

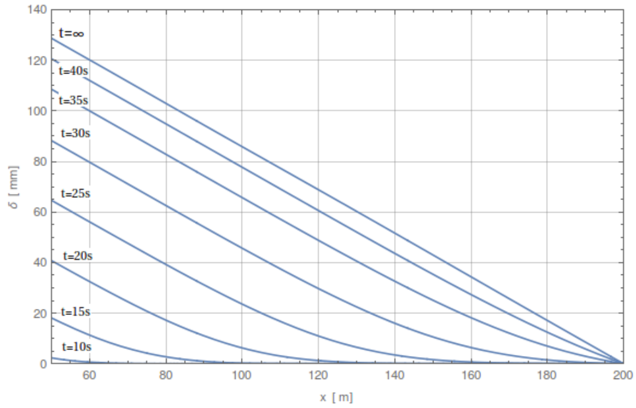
In order to illustrate the inlet temperature effect, a simulation was performed considering the same properties and pipe diameter used in the Fig. 5(a) simulation, except that now the inlet temperature is $T_{me} = 570\,K$. As per Eq. (31), the thermal expansion coefficient linearly increases with temperature, which means that the pipe length shall be shorter in order to respect the maximum thermal expansion allowed. Figure 5(e) shows the pipe thermal expansion profile obtained from this simulation.

The simulation shows that by increasing the inlet temperature, even thought the pipe material thermal diffusivity and the Grashof number are smaller at $570\,K$, the heat rate is higher as a consequence a larger Reynolds and Nusselt numbers.

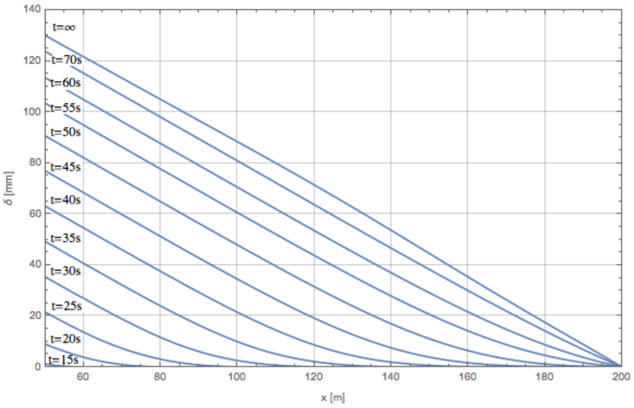
By comparing Fig. 5(a) and 5(e), it was found that the thermal expansion rate is higher at $T_{me} = 570\,K$ as a consequence of higher heat rate and mainly smaller pipe length.



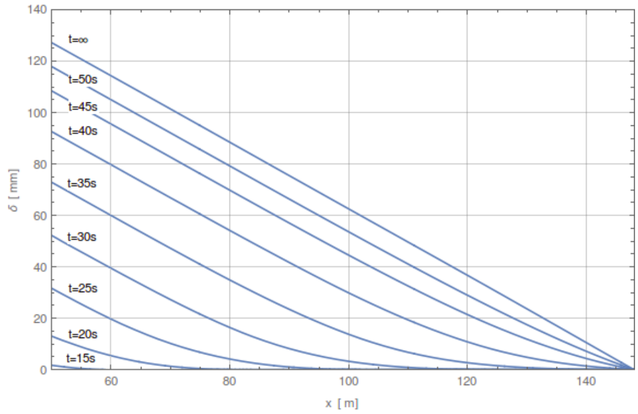
(a) Pipe thermal expansion profile. Steel pipe, $D_n = 200\,mm$, $U = 3\,m/s$, $Re = 2.09 \times 10^6$, $Nu = 6654.4$, $h_i = 22290.5\,W/(m^2\,K)$, $Gr = 41.2$, $h_e = 5.44\,W/(m^2\,K)$.



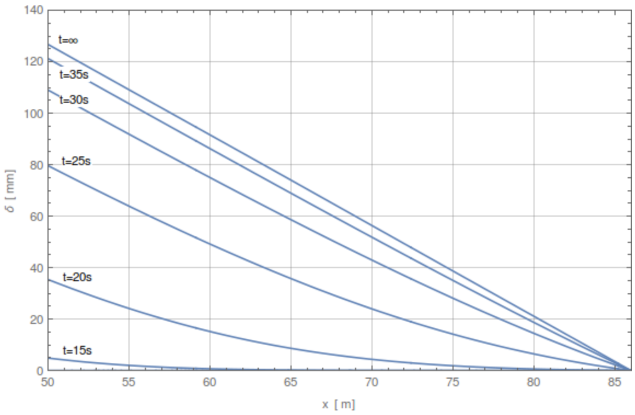
(b) Pipe thermal expansion profile. Steel pipe, $D_n = 200\,mm$, $U = 5\,m/s$, $Re = 3.48 \times 10^6$, $Nu = 11047.9$, $h_i = 37007.8\,W/(m^2\,K)$, $Gr = 41.2$, $h_e = 5.44\,W/(m^2\,K)$.



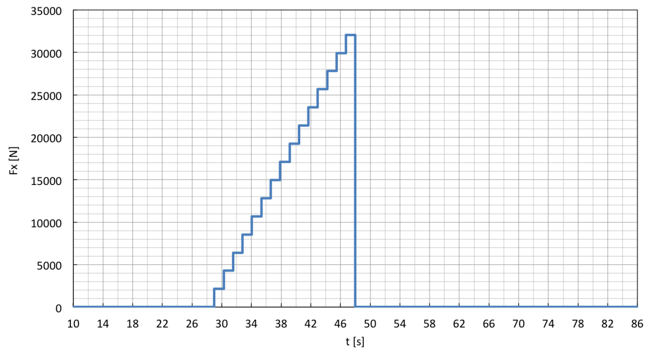
(c) Pipe thermal expansion profile. Steel pipe, $D_n = 100\,mm$, $U = 3\,m/s$, $Re = 1.05 \times 10^6$, $Nu = 3889.2$, $h_i = 25862\,W/(m^2\,K)$, $Gr = 23.1$, $h_e = 5.84\,W/(m^2\,K)$.



(d) Pipe thermal expansion profile. Copper pipe, $D_n = 200\,mm$, $U = 3\,m/s$, $Re = 2.09 \times 10^6$, $Nu = 6654.4$, $h_i = 22290.5\,W/(m^2\,K)$, $Gr = 41.2$, $h_e = 5.44\,W/(m^2\,K)$.



(e) Pipe thermal expansion profile. $T_{me} = 570\,K$. Steel tube, $D_n = 200\,mm$, $U = 3\,m/s$, $Re = 4.81 \times 10^6$, $Nu = 10644$, $h_i = 28734.8\,W/(m^2\,K)$, $Gr = 44.9$, $h_e = 7.42\,W/(m^2\,K)$.



(f) Vertical load behavior at the anchor B of Fig. 4 as a consequence of the transient thermal expansion of Fig. 5(a).

Figure 5. Simulation results.

6.2 Loads due to the transient thermal expansion

For the methodology classically used in Pipeline Engineering to compute force, the thermal expansion rate does not affect the calculated loads. Despite this, by knowning the gradual and progressive nature of the pipe heating, it is possible to do a horizontal sum of forces by using Eq. (2) and (5).

In order to illustrate the load behavior at the pipe anchor point as a function of time, it is considered the pipe thermal expansion profile of Fig. 5(a), where it is considered water, steel pipe with $D_n = 200\,mm$ and $U = 3\,m/s$. It is important to note that it is considered a symmetric pipe rack with length of $300\,m$ and $5\,m$ span between beams.

Figure 5(f) shows the vertical load behavior at the anchor B indicated in Fig. 4. The used friction coefficient is $\mu = 0.3$.

It is found that the force associated with the linear thermal expansion starts to be transmitted to the anchor in a abrupt way and increases in steps until its maximum value. After this, the force falls abruptly to its minimum value once the downstream thermal expansion from the anchor tends to displace all this pipe section. This minimum value is the computed load for the stead state.

An interesting result related to the thermal transient loads is the time instant in which the load where the pipe is anchored is maximum. In the model used in this study the load in the anchor B is maximum when the thermal profile is between the anchor and the support immediately before this anchor. Moreover, the time is short, about $1.5\,s$.

7. CONCLUSIONS

A methodology for calculating expansions in pipes during thermal transients was developed. Simulations were made, taking into account the influence of velocity, pipe material, diameter and inlet fluid temperature on pipe displacements and horizontal forces.

A case study was presented. Simulations showed that during the transient, forces can be higher than the ones predicted at steady state.

It was found that the higher the flow velocity, the Nusselt number and the thermal diffusivity of the pipe material, the higher the pipe heating rate. Therefore, the higher the heating rate and the thermal expansion coefficient, the higher the linear thermal expansion rate of the pipe. Moreover, the force associated with the linear thermal expansion starts to be transmitted to the pipe fixed point (anchor or stop) in steps until its maximum value, then falls abruptly to its minimum value, being this the load obtained in the usual pipe load analysis. The instant where the load on this fixed point is maximum is very short and depends on the average flow velocity and pipe span.

This study shows the necessity of taking into account the transient thermal behavior in the design of pipe systems.

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