Development of calculation methodologies for the design of piping systems

João Pedro Amaral Vidigal da Silva
Mechanical Engineering Department, Instituto Superior Técnico, Lisboa, Portugal
e-mail: joaovidigal@ist.utl.pt

Abstract - Piping systems are constantly present in industrial facilities, being in some cases associated with the transport of fuels and steam at very high temperatures. Due to the nature of those fluids, the design of the piping system that transports them is a task of great responsibility, which must follow codes and standards to guarantee the system's structural integrity.

Many times the piping systems operate at a temperature higher than the temperature at which they are assembled, leading to the thermal expansion of the system's pipes and since no piping system is free to expand, the thermal expansion will lead to stresses. Besides the stresses caused by thermal expansion, the studied systems will also be subjected to constant loads, caused by their weight.

In this perspective, will be developed calculation methodologies in order to do quick analysis of the most common configurations, according to the codes ASME B31.1 and B31.3 (2004), allowing that way improvements on the flexibility of the projected systems.

Although the methodology developed may only be used in simple systems and gives very conservative results, in practical cases it can be used to analyze complex systems, by dividing them in simpler cases. Besides that, the method developed, may also be used to analyze simple cases that are frequently present, without having to use commercial software, such as CAESAR II, being helpful to define the system's layout.

Keywords: Piping Systems, Flexibility, Stress Analysis, Thermal Expansion, ASME B31.1, ASME B31.3

I. INTRODUCTION

The origin of the design and construction of piping systems make us go back to the ancient times of the first civilizations. Their evolution and development through the centuries, was the result of the constant need to carry fluids between different points.

The first piping systems were constructed between 3000 b.C. and 2000 b.C. in the ancient Mesopotamia to be used on the irrigation of large areas of cultivated land. Initially used in agriculture, due to the growing need to cultivate larger areas, piping systems also had a crucial role in the development of big cities and during the industrial revolution with the discovery of steam power. Piping systems also turned out to be essential in the exploration of oil (Antaki, 2005).

In the present civilization, piping systems are constantly present, either in residential and commercial buildings, either in industrial facilities. In oil refineries and others industrial process plants, pipelines represent between 25% and 50% of the total cost of the facilities (Nayyar, 2000).

Since piping systems are associated with facilities of high degree of responsibility, stress analysis represent a fundamental stage of the piping design, in order to prevent accidents. Taking into account that piping systems are subjected to multiple loads, stress analysis represents a complex task. Besides the stresses caused by the piping weight, fluids and isolation, piping systems are also subjected to temperature changes, internal and external pressure, and occasional events such as water hammer, wind and earthquakes.

Due to the temperature variations that occur in piping systems, between the installation and operation temperatures, they will be subject to expansion and contraction. In the general terms, both contraction and expansion are called thermal expansion. Since every piping system has restrictions that prevent the free expansion, thermal expansions will always create stresses, but, if the system is flexible enough, the expansion may be absorbed without creating undue stresses that may damage the system, the supports and the equipment to which the pipes are connected (Ellenberger, 2010).

One of the greatest challenges in the pipe stress analysis is to provide the system enough flexibility to absorb the thermal expansions. Even nowadays, that pipe stress analysis covers much more than flexibility analysis, it still is one of the main tasks of the engineers that work in this area (Peng, L.C. and Peng, T.L., 2009). Many times due to the inexistence of a quick method that allows a verification of the flexibility of projected systems, they turn out to be too stiff or too flexible.

Engineers constantly face the need to minimize the costs and at the same time obtain a system with enough flexibility, without sacrificing the security requirements. The shortest the system, the lowest the price, since it will use less material, but this configuration will have flexibility problems in the majority of the cases, due to the incapacity to absorb thermal expansion. On the other hand, systems that are too long may have problems due to pressure drop. The increase of a system's flexibility may be obtain

due to the changes in direction, although, in the cases where the flexibility obtained that way isn't enough, additional flexibility may be obtain using pipe loops and expansion joints (Peng, n.d.).

The attention given to pipe stress analysis has increased in the last decades, due to the high security requirements of the modern process plants. For that reason, the access to an efficient computer program, such as CAESAR II, to perform the stress calculations, reduces the design costs, since it decreases the time necessary to perform the analysis.

In order to prove the structural integrity of a piping system is necessary to follow the procedures and specifications of the piping codes. There are several codes that involve the design of piping systems, but the ones more often used are the ASME B31 Codes.

It is extremely important to make a correct design of piping systems, avoiding their failure, which may cause huge material damage and even loss of human lives. The objective of this paper is to present calculation methodologies for the design of piping systems and to study pipe loops as a way of increase piping systems flexibility.

II. CODES AND STANDARDS

In order to satisfy the security requirements of piping systems, they have to be projected and built according to determinate codes and standards.

In the United States, the American Society of Mechanical Engineers (ASME) has assumed the leadership in the formation of committees that have elaborated The Piping Code. The Piping Code is constituted by a set of requirements that assure a correct and safe operation of de piping systems (Peng, L.C. and Peng, T.L., 2009). The code ASME B31 establishes the allowable stresses, the design, the fabrication, the erection, the tests, the fatigue resistance and the operation for non-nuclear piping systems (Ellenberger, 2010; Wingate, 2007). For this paper we are particularly interested in the facilities covered by the codes ASME B31.1 and B31.3:

- ASME B31.1 Power Piping: For facilities that use steam generation, like power plants or oil refineries, where the steam is used in the process. The safety factor used is about 3.5 against the ultimate strength of the pipe (Woods and Baguley, 1997; Telles, 2005).
- ASME B31.3 Process Piping: For piping systems used in process plants, such as petrochemical plants, is the code that covers more varieties of piping systems (Ellenberger, 2010). In this case, the safety factor used is about 3.0 against the ultimate strength of the pipe (Peng, L.C., Peng, T.L., 2009).

III. THERMAL EXPANSION AND FLEXIBILITY

Most of the piping systems work at temperatures higher than the installation temperature. This temperature raise, will lead to the thermal expansion of the pipes, which for the cases of interest of this paper will always be metal pipes. The thermal expansion of a material is evaluated by the thermal expansion coefficient α. The thermal expansion of a pipe may be calculated by the following expression (Nayyar, 2000):

$$\Delta = \alpha L \tag{1}$$

where L is the pipe length at the reference temperature (usually the installation temperature).

If a piping system does not have enough flexibility, in order to compensate the thermal expansions, the stresses originated may damage the system, as well as the equipment to which it is connected. There are several methods to increase systems' flexibility, being often used the installation of pipe loop as depicted in Fig. 1. In some cases, due to spatial constrains, expansion joints are the alternative to the installation of pipe loops. There are several types of expansion joints, but all of them are elements much more sophisticated than the pipe loops (Nayyar, 2000), which are nothing more than additional sections of pipe. In addition, expansion joints are subjected to breakdowns and require maintenance. For these reasons, the design of pipe loops to increase the systems flexibility is preferable to the use of expansion joints.

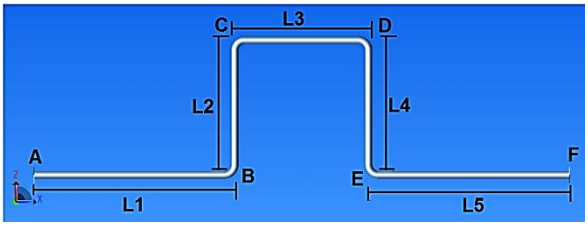

Fig.1 – Pipe loop

In the case of straight sections of pipe with both ends anchored, the thermal expansion will cause strengths and stresses, due to the fact that thermal expansion is absorbed by the compression of the pipe. In this case, flexibility may be increased adding additional sections of pipe perpendicularly to the original section, to absorb the expansion that is the principle of pipe loops. In other words, by using pipe loops, the thermal expansion is absorbed by the bending of the perpendicular sections of pipe (Peng, L.C., and Peng, T.L., 2009).

Besides the use of pipe loops being much simpler than the use of expansion joints, it still matters to know the best pipe loop configurations, in order to maximize its potential to increase flexibility.

In first place, in order to keep forces and moments balanced, the loop as depicted in Fig. 1 must be symmetric. Concerning the relation between the loop dimensions, there is some divergence between companies, while some established design guidelines defining $L_3 = L_2$ (Frankel, 2002), other defined the best configuration as the one that follows the relation $L_3 = 1/2L_2$ (Thermacor, n.d.).

To calculate de length L_2 necessary to absorb the thermal expansion without damaging de pipe, the following expression, which derives from the guided cantilever method, may be used (Peng, n.d.; Kannappan, 1985):

$$L_2 = \sqrt{\frac{3E_h D\Delta}{S_A}} \tag{2}$$

where E_h is the modulus of elasticity of the material at the operation temperature, D is the outer diameter of the pipe and S_A is the allowable expansion stress.

Relatively to loops locations, they must be centered between anchorages, $L_1 = L_5$. In cases that is not possible to center the loop, it should be tried that the pipe sections, at each side of the loop has their dimensions as close as possible (Fluor, 2002).

Besides the anchorages at the ends of the pipe loops, in many cases there are also intermediate guides and vertical supports. The function of the vertical supports is to support the pipes' weight, assuring the allowable pipe span (Kannappan, 1985). The guides are used to control the thermal expansion, assuring that the loops play their role correctly, since they direct the expansion to the sections defined by the tubes of length L_2 and L_4.

IV. CRITICAL LENGTH

During the piping systems process design, it is known that the use of long segments of straight pipe with both ends anchored will cause high stresses and problems of instability due to the axial compression due to thermal expansion. So it is important to know what is the critical length L_{cr}, for which changes in the layout should be made, in order to absorb the thermal expansion.

To find the critical length the Euler's Instability Formula for a pipe with both ends fixed is used, being substituted the variables with the Hooke law and the thermal expansion given by equation (1). The critical length is given by:

$$L_{cr} = \sqrt{\frac{\pi^2 I}{0.25A\alpha}} \tag{3}$$

where A is cross-sectional area of the tube, I is the moment of inertia of the cross section and α is the thermal expansion coefficient of the pipe material.

V. CODE STRESS REQUIREMENTS

According to the Codes ASME B31.1 and B31.3 (2004), only the maximum stresses are calculated, which is implicit in the stress intensification factors (SIF) that derive fundamentally from fatigue tests (Woods and Baguley, 1997).

The expressions to calculate the stresses presented in the ASME B31.1 and B31.3 (2004), are only influenced by the moments, ignoring the forces. This is due to the fact that the stresses originated by forces are usually too low when compared with the stresses originated by moments (Peng, L.C. and Peng, T. L., 2009).

Before calculate the stresses, the moments have to be reoriented accordingly to the planes of the component that is under analysis, due to the different SIF of each direction.

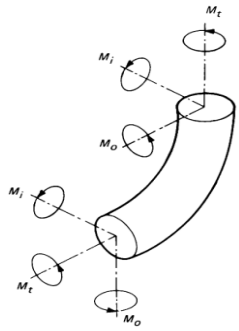

Fig. 2 – Moments in bends

Stresses are calculated in the nodes located at the ends of each element. For the shear stresses, the maximum tension in the outer surface of the pipe, is given by (ASME B31.3, 2004):

$$\tau = S_t = \frac{M_T}{2Z} \tag{4}$$

where M_T is the shear moment and Z is the section modulus.

The bending stress acting on the two different planes can be combined, consequently the combined bending stress S_b acting on the longitudinal direction is given by (ASME, B31.3, 2004):

$$S_b = \frac{\sqrt{(i_i M_i)^2 + (i_o M_o)^2}}{Z} \tag{5}$$

where M_i and M_o, are respectively the inner and outer plane bending moments, and i_i and i_o are respectively the inner and outer plane stress intensification factors.

The flexibility analysis is done by the comparison between the combined effect of multidimensional tensions and the allowable stress. The ASME B31.1 and B31.3 codes use the Tresca criterion to obtain the combined tension effect S_E (Woods and Baguley, 1997), also called expansion stress:

$$S_E = \sqrt{S_b^2 + 4S_t^2} \tag{6}$$

According to the ASME B31.1 and B31.3 (2004) Codes, the stresses to which a piping system is subjected may be separate in three main classes, for which the codes establish limits: the stresses caused by sustained loads (pressure and weight), the stresses caused by occasional loads and the stresses caused by thermal expansion.

A. Sustained stresses

Sustained stresses in piping systems are caused by weight, pressure and any other constant load (ASME B31.1 and B31.3, 2004).

The ASME B31.1 (2004) establishes a limit for the stresses caused by sustained load:

$$S_L = \frac{PD}{4t_n} + \frac{1000\,(0.75i)M_A}{Z} \leq 1.0S_h \qquad (7)$$

where M_A is the resultant moment in the transverse section due to weight [N.mm], i is the SIF, *P* is the internal pressure [kPa], *D* is the outer diameter of the pipe [mm], t_n is the thickness of the main pipe [mm] and S_h is the allowable stress at the operation temperature [kPa].

The ASME B31.3 (2004), also establishes a limit to the sustained stress: $S_L \leq S_h W$, where W is the weld joint strength reduction factor and S_L is sum of the longitudinal stresses, due to sustained loads such as pressure and weight.

B. Occasional stresses

Occasional stresses are caused by occasional events, such as water hammer, earthquakes and wind. According with ASME B31.1 (2004) the following condition must be satisfied:

$$\frac{PD_o}{4t_n} + \frac{1000(0.75iM_A)}{Z} + \frac{1000(0.75iM_B)}{Z} \leq kS_h \qquad (8)$$

where M_B is the resultant moment in the transverse section due to occasional loads.

According to ASME B31.3 (2004) $S_L + S_{oc}$ should be lower than $1.33S_h$, where S_{oc} are the stresses produced by occasional loads (Peng, L.C. and Peng, T.L., 2009).

C. Thermal expansion stresses

The thermal expansion usually leads to fatigue failure, so the system's integrity depends on the stress range and on the number of operation cycles (Peng, L.C., Peng, T.L., 2009). The ASME B31.1 (2004), gives an expression to evaluate the thermal expansion stresses:

$$S_E \leq S_A + f(S_h - S_L) \qquad (9)$$

where $S_A = f(1.25S_c + 0.25S_h)$ and S_c is the allowable stress at the installation temperature.

In the case of ASME B31.3 (2004) the stresses caused by thermal expansion, must satisfy the following condition:

$$S_E \leq S_A \qquad (10)$$

where $S_A = f(1.25S_C + 0.25S_h)$, or $S_A = f[1.25(S_C + S_h) - S_L]$ if $S_h > S_L$. In both cases f is the stress reduction factor, defined by each code.

VI. THERMAL EXPANSION CALCULATIONS

There are several methods to calculate the stresses caused by thermal expansion. The method used in this paper is the Spielvogel Method (Spielvogel, 1961), which is based on the Theory of the Elastic Center and on the Castigliano Theorem.

The work done to deform a pipe of length L, subjected to an axial force F and a moment M, is given by (Hetnarski and Eslami, 2009):

$$W = \int_0^L \frac{P^2 ds}{2AE} + \int_0^L \frac{M^2 ds}{2EI} \qquad (11)$$

where, E is the modulus of elasticity of the material, A is cross-sectional area of the tube and I is the moment of inertia of the cross section.

Considering a piping system in the x-z plane with both ends anchored and with no intermediate restrictions, it is known that due to thermal expansion each end will be subjected to a pair of forces, F_x and F_z, and one moment M_y. Considering the equations of static equilibrium, the forces will have the same modulus in both ends.

From equation (11), the Theory of the Elastic Center and the principal described above, the following system of equations may be deduced (Spielvogel, 1961):

$$\begin{cases} F_x \dfrac{I_{xx}}{EI} + F_z \dfrac{I_{xz}}{EI} = \Delta_x & (12) \\ F_x \dfrac{I_{xz}}{EI} + F_z \dfrac{I_{zz}}{EI} = \Delta_z & (13) \end{cases}$$

where I_{mn} are the line moments of inertia of the system in the plane *m*-*n*, and Δ_x and Δ_z, are respectively the thermal expansion in the direction x and in the direction z.

Solving the system of equations (12) and (13), the forces on the centroid of the system F_x and F_z are obtained. The value of these forces is equal at any point of the system. In order to obtain the bending moment M_y at any point of the system, the following expression can be used:

$$M_y = F_x \bar{z} - F_z \bar{x} \qquad (14)$$

where $\bar{x}$ and $\bar{z}$ are the coordinates of the point in question, in the referential with origin on the centroid.

A. Reactions

According to ASME B31.1 e B31.3 (2004), the calculation of the stresses caused by thermal expansion shall be done using the cold modulus of elasticity E_c. However, accordingly to the same codes, the maximum value of the reactions must be considered at the installation and at the maximum expansion conditions.

For systems with both end anchored and no intermediate restrains, the maximum values of the reactions, depend on the level of cold-spring and are given by (ASME B31.1 and B31.3, 2004):

- For the condition of maximum thermal expansion:

$$R_h = R\left(1 - \frac{2C}{3}\right)\frac{E_h}{E_c} \tag{15}$$

- For the installation condition:

$$R_c = \max\{CR\,, C_1 R\} \tag{16}$$

where C is the cold-spring factor (varying from 0 for systems in no cold-spring, to 1 for system with 100% cold-spring), R is the value of the reaction base on E_c, R_h is the maximum reaction and R_c is the reaction at the installation condition.

B. Stresses in loops with intermediate restrain

The Spielvogel method (1961) was developed for loops without intermediate restrains, being the maximum expansion stress in loops with intermediate guides, as the one depicted in Fig. 3, obtained by the expression established by the Grinnell Corporation (1981):

$$S_B = \frac{L'}{L} S_E \tag{17}$$

where S_B is the maximum expansion stress for the loop with intermediate guides and S_E is the maximum expansion stress for a loop of the same size with anchors in the points where the guides are.

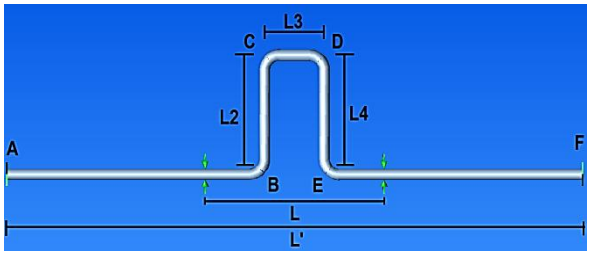

Fig. 3 – Loop with intermediate guides

VII. WEIGHT CALCULATIONS IN PIPE LOOPS

To calculate the forces and the moments caused by the weight on symmetric pipe loops without intermediate restrains, the equations of static equilibrium, the beam deflection formulas and the torsion formulas must be used.

The reactions caused by the weight on the anchored points, considering y as the gravity axis, will be: the force F_{yA}, and two moments M_{xA} and M_{zA}.

Considering w the combined weight of the pipe, fluid and isolation, the static equilibrium is given by:

$$F_{yA} = w\left(L_1 + L_2 + \frac{L_3}{2}\right) \tag{18}$$

Knowing that the moment, in the x direction, in the symmetry point will be null, by the equilibrium of moments:

$$M_{xA} = \left(w\frac{L_3}{2}\right)L_2 + (wL_2)\frac{L_2}{2} \tag{19}$$

Since symmetry only exists in one direction, the moment in the point of symmetry in the direction z $M_{zC'}$ is not null. Therefore the moment M_{zA} is given by:

$$M_{zA} = (wL_1)\frac{L_1}{2} + \left(w\frac{L_3}{2}\right)\left(\frac{L_3}{4} + L_1\right) + (wL_2)L_1 - M_{zC'} \tag{20}$$

The equation for $M_{zC'}$ can be found by dividing half of the pipe loop in three different segments, as illustrated in figures 4 to 6, and applying the boundary condition in the point of symmetry C': $\theta_{zC'}$=0.

Fig.4 – Loads do to weigh in segment AB

Fig.5 – Loads do to weigh in segment BC

Fig.6 – Loads do to weigh in segment CC'

The moment $M_{zC'}$ can be calculated by solving equation (21),

$$\theta_{zC'} = \frac{1}{EI}\left(-\frac{P_B L_1^2}{2} - \frac{wL_1^3}{6} + M_{zB} L_1\right) + \frac{M_{zB} L_2}{JG} + \frac{1}{EI}\left(-\frac{wL_3^3}{6} + M_{zC'} L_3\right) \quad (21)$$

where J is the polar moment of inertia of the cross section, G is the transverse modulus of elasticity of the material, $P_B = F_{yA} - wL_1$ and $M_{zB} = M_{zC'} - \frac{wL_3^2}{2}$.

VIII. SENSITIVITY ANALYSIS

A. Pipe loops

In this section are reported the results of the sensitivity analysis of symmetrical pipe loops with both ends anchored. The objective of the analysis is to verify how the reactions and the stresses caused by the thermal expansion vary with the modification of the operation conditions, namely the temperature, and with variations of the loop and pipe dimensions.

In addition it is intended to compare the difference between the results for the forces, moments and stresses, obtained with different methods: the Spielvogel method (Spielvogel, 1961), the Grinnell method (Grinnell Corporation, 1981), and using the commercial software CAESAR II.

1) Loops without guides: in this case, the reactions caused by thermal expansion, will only be a force in the x direction F_x and a bending moment M_y, as illustrated in figure 7.

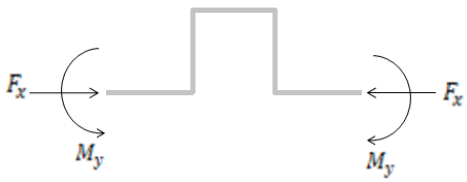

Fig.7 – Free body diagram of a pipe loop, considering only the thermal expansion

In this case, the maximum stress due to thermal expansion will occur in the corner C (Fig. 3). For the stress intensification factor it was considered turns of radius $r = 3/2D$.

First it is studied the variation of the segment L_2 (consequently the variation of L_4, since the loop is symmetric), fixing the remaining dimensions, being obtained the charts for the reactions variation of figures 8 and 9.

Looking at the charts it can be concluded that with the increasing of L_2, the amplitude of the reaction decreases and the results of the three methods converge. The results of the Spielvogel Method and of the Grinnell Method are very similar with a relative difference of about 1%. Besides that, the results of this two methods are much more conservative than the results of the software CAESAR II, with a relative difference around 38% for the forces results and 23% for the moments results.

Regarding the evolution of the maximum stresses $S_{E,max}$ illustrated in Fig. 10, the reduction of stresses with the increase of L_2 can be observed. Once again these methods give very similar results being the relative difference equal to 2.2%. The relative difference between the CAESAR results and the other methods is of about 44%.

From this analysis, it can be conclude that from the configurations suggested by different authors the one more advantageous in terms of stresses and reactions is the one that follows the relation $L_3 = 1/2L_2$.

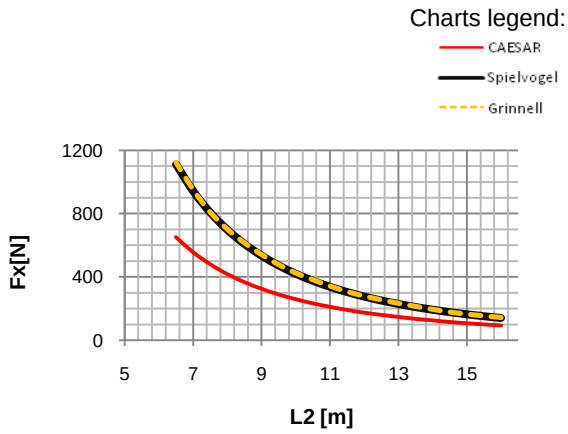

Fig.8 - Chart of F_x vs. L_2

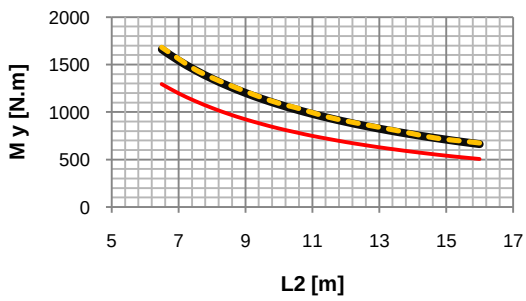

Fig.9 - Chart of M_y vs. L_2

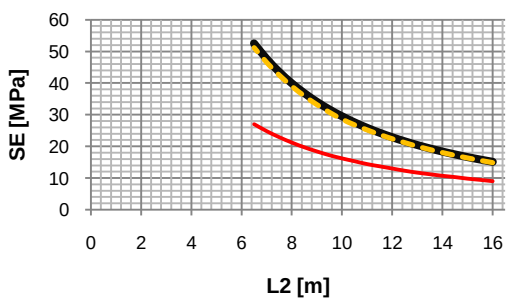

Fig.10 - Chart of $S_{E,max}$ vs. L_2

To continue the sensitivity analysis, the influence of temperature is analyzed, being the loop's dimensions fixed. The charts for reactions vs. temperature are represented in figures 11 and 12. With increasing temperature, the reactions magnitude increase, that can be explained by the increasing thermal expansion in absence of increased flexibility. Also with increasing temperature, the results of the three methods diverge. For temperatures lower than 50 ºC the relative difference between the results of the Spielvogel Method and the Grinnell Method is of 1%, but increases until a maximum of 14.6% for 400 ºC. For the CAESAR results, once again it is verified that are less conservative, with difference relatively to the Spielvogel Method of about 54% for forces and 34% for the moments.

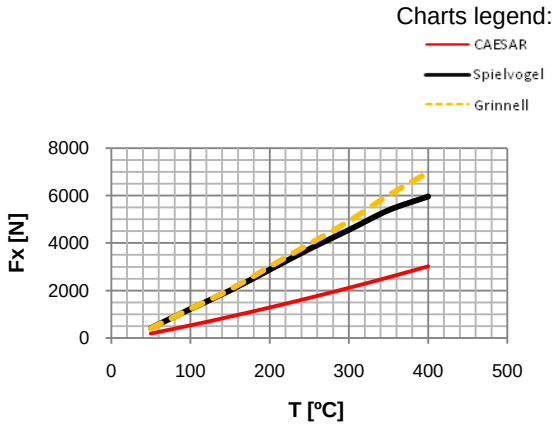

Fig.11 - Chart of F_x vs. T

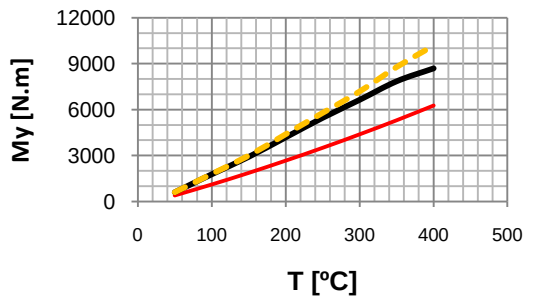

Fig.12 - Chart of M_y vs. T

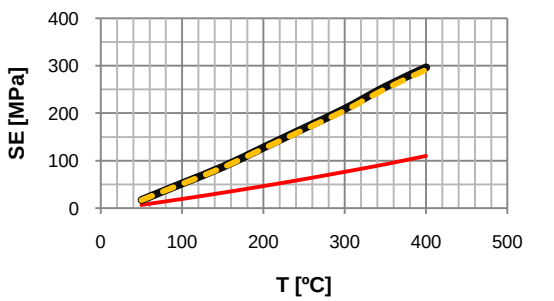

Fig.13 - Chart of $S_{E,max}$ vs. T

The maximum stresses represented in Fig. 13, again reflect the results obtained for the reactions, increasing with the temperature. In this case the Spielvogel Method results are slightly superior relatively to the Grinnell's results, with a relative difference of about 1.7%. The relative difference between the Spielvogel results and the CAESAR results is around 63%.

To conclude the sensitivity analysis for loops without guides, the influence of the pipe thickness was studied, with the loop's dimensions fixed.

For the reactions, the results are illustrated in the charts of figures 14 and 15. As can be confirmed, by increasing the pipe thickness, the reactions will increase, due to the increase of the moment of inertia of the cross section.

The stresses decrease with the increasing thickness, as illustrated in the chart of Fig. 16. The stresses diminution can be explained by the stress intensification factor diminution. Therefore beside the increasing of the reactions magnitude, with the increase of the pipe wall thickness, they are easily absorbed by the system.

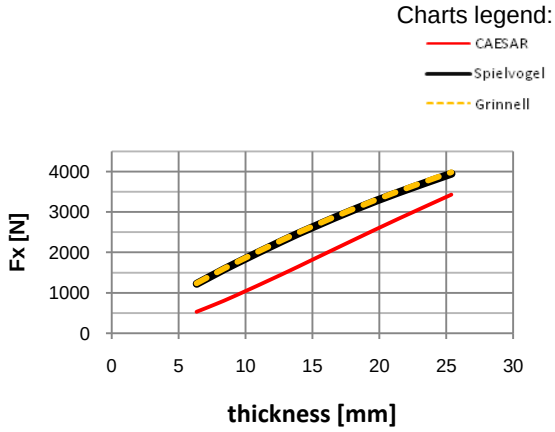

Fig.14 - Chart of F_x vs. thickness

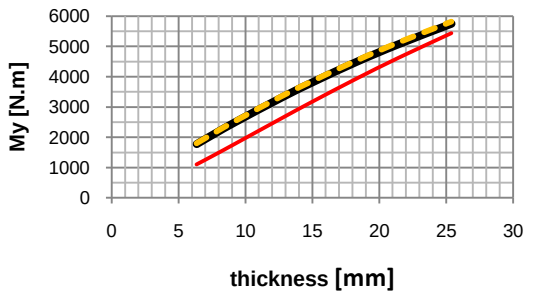

Fig.15 - Chart of M_y vs. thickness

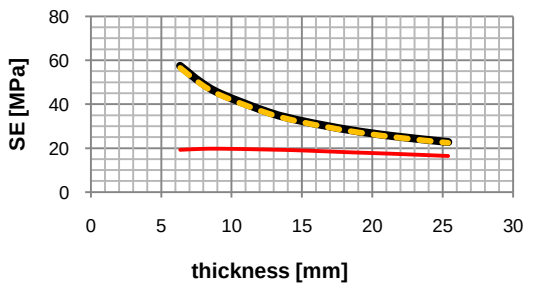

Fig.16 - Chart of $S_{E,max}$ vs. thickness

With the increasing thickness, can be verified a convergence of the results of the three methods. The relative difference between the results of the two methods and of CAESAR decreases with the increasing thickness. In the case of the forces it decreases from 56% to 13%, in the case of the moments from 38% to 5.6%, and in the case of the stresses from 66% to 27%.

B. Critical length

In this section is reported the sensitivity analysis of the critical length L_{cr}, versus the temperature, for different pipe diameters. The result obtained is represented on the chart of Fig.17.

Analyzing the chart, it can be conclude that the critical length, decreases with increasing temperature of operation. Besides that, it can be also observed that with larger diameter, the critical length decreases.

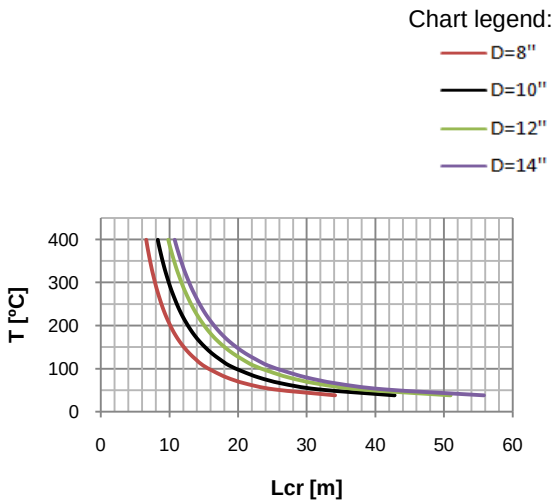

Fig.17 - Chart of temperature vs. L_{cr}

IX. CONCLUSIONS

The objective of this work is to develop calculation methodologies for the design of piping systems of fuels and steam. There are several codes and standards that can be used so assure the integrity of the systems, being the ASME B31.1 and B31.3 (2004) the most used.

According to the ASME B31.1 and B31.3 (2004) Codes, the stresses to which a piping system is subjected may be separate in three main classes, for which the codes establish limits: the stresses caused by sustained loads (pressure and weight), the stresses caused by occasional loads and the stresses caused by thermal expansion. Since the stresses due to occasional loads are only verified in very specific cases, the methodologies developed are only for the sustained loads and thermal expansion. To calculate the stresses according to the ASME B31.1 and B31.3 Codes, it is necessary to know the forces and the moments to which the system is subjected.

Besides the Codes' stress requirements, it is also important to analyze the systems in the operation conditions, namely the loads on the supports and the displacements.

In order to find the forces and the moments caused by the system's weight, the equations of static equilibrium, the beam deflection formulas and the torsion formulas can be used. These methodologies give results very similar to the CAESAR's results, but more conservative, due to the fact of neglecting the curvature of the directions changes. During the development of the methodologies for the calculation of the weight loads, was concluded the pipes torsion cannot be neglected, since it has a great influence in the calculation of the forces, moments and deformations.

The determination of the loads caused by the thermal expansion is a much more complex task, than the determination of the loads due to weight. There are several methods that can be used for that purpose, however, these methods only allow the analysis of systems with both ends anchored and without intermediate restrains, besides that the results obtained are very conservative when compared to the results obtained with the software CAESAR II. The method used to calculate the forces and the moments due to thermal expansion is the Spielvogel Method, which gives results very similar to the Grinnell Company Method, but is more versatile than other methods which are dependent of tables and charts.

Although the methods used to calculate the loads due thermal expansion have several limitations and give very conservative results, they allow the analysis of complex systems by the division of those systems in simple cases, to which the methods can be applied. The method can also be applied in the analysis of simple cases that are constantly present (like pipe loops), without using the software CAESAR II, that can be useful in the definition of the systems' layout. Furthermore the analysis of this method allows understanding the loads caused by the thermal expansion, allowing that way to have a critical opinion about the results obtained with the CAESAR II.

Pipe loops are a very effective way to increase system's flexibility. It have been concluded that from the different configurations, suggested by different authors, for pipe loops, the best is the one that follows the relation $L_2 = 2L_3$. Furthermore it has been verified that the stresses increase with the increasing temperature and with the distance between anchors, but decrease with the increase of the pipe's thickness, even if it means an increase of the system's reactions.

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