

Calculation Methodologies for The Design of Piping Systems

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Abstract- Piping systems are constantly present in industrial facilities, being in some cases associated with the transport of fuels, processing of crude oils and chemical plants. 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, as well as occasional loads like wind, earthquake. In this perspective, calculation methodologies were developed in order to do quick analysis of the most common configurations, according to the codes like ASME B31.3, 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 analyse complex systems, by dividing them in simpler cases.

[6, 8, 10]

Keywords

Piping Systems, Flexibility, Stress Analysis, Thermal Expansion, ASME B31.3, design methodology, expansion loop.

I. Introduction

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.

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.

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 failures and cause of 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.

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. 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 obtained using pipe loops and expansion joints. The attention given to pipe stress analysis has increased in the last decades, due to the high safety requirements of the modern process plants. For that reason, the access to an efficient computer program, such as CAESAR II, ANSYS, AUTOPIPE 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, it 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. However there are design limitations stated in Codes, these are needed to be considered during analysis assumptions and deciding methodology.

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.

[1, 4, 5, 8]

II. CODES AND STANDARDS

In order to satisfy the safety requirements, local regulations, design constraints of Client, piping systems have to be designed 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 the piping systems. 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. For this paper we are particularly interested in the facilities covered by the codes B31.3.

ASME B31.3 Process Piping: For piping systems used in process plants, such as petrochemical plants, this is the code that covers almost all the requirements to design, erection, testing of piping systems. The stress analysis requirements are detailed in this code can be applied to all the plants designed according to this Code. This Code is constantly improved considering latest industrial trend and feedback from various committee members, consultants and industrial operation groups.

There are other Codes and Standards from different countries like UK, EU, Japan which also are used and applied to the stress analysis. There are consultancy practices which provide guidelines and rules to design piping system.

[3, 6, 8]

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 metallic 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:

$$\Delta = \alpha L \quad (1)$$

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.



Fig. 1

Consider piping connecting from Tank T1 to Tank T2. If the piping is straight as shown in the Fig. 1 and fluid with higher temperature flow through the piping, the section of pipe between T1 and T2 will expand. However, there is no space to expand and result in high loads on the nozzles. However if we provide the loop as shown in the Fig. 2, the nozzle loads can be decreased.

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.

There are several methods to increase systems' flexibility, being often used the installation of pipe loop as depicted in Fig. 2. In some cases, due to spatial constraints, 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, 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.

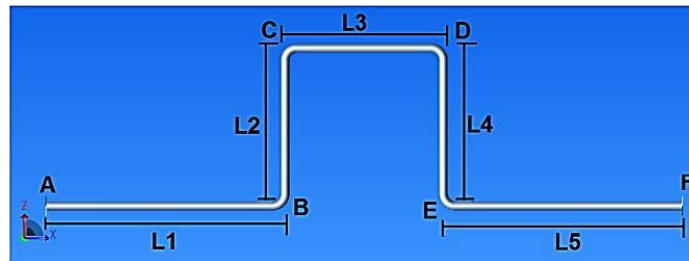


Fig.2 – Pipe loop

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. 2 must be symmetric. Concerning the relation between the loop dimensions, there is some divergence between companies, while some established design guidelines defining $L3=L2$, other defined the best configuration as the one that follows the relation:

$$L3 = 1/2L2 \quad (2)$$

To calculate the length $L2$ necessary to absorb the thermal expansion without damaging the pipe, following expression, which derives from the guided cantilever method, may be used:

$$L2 = \sqrt{\frac{3E_h D \Delta}{S_A}} \quad (3)$$

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, $L1 = L5$. 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.

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. 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 $L2$ and $L4$.

[5, 9, 11, 12]

IV. CODE STRESS REQUIREMENTS

According to the Codes ASME B31.3, only the maximum stresses are calculated, which is implicit in the stress intensification factors (SIF) that derive fundamentally from fatigue tests. Code requirements are specified with the cautions and limitations which need to be carefully considered while designing piping system. Even Code specifies the actions when the condition cannot fully comply with Code requirements. The expressions to calculate the stresses presented in the ASME B31.3, 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 moment. 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. The SIF calculation is the most crucial step and careful considerations are required. The in plane and out of plane moment need to be considered. The stresses developed are more in the cases like direction change, inside diameter change, sudden obstructions. The example of elbow is shown in Fig. 3, shows inner and outer plane moments.

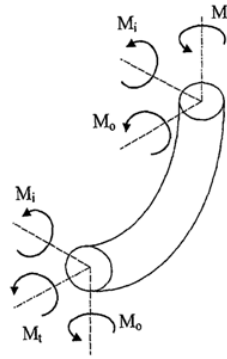


Fig. 3 – Moments in bends

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):

$$\tau = S_t = \frac{M_t}{2Z} \quad (4)$$

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):

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

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 (Fig- 3).

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

$$S_E = \sqrt{S_b^2 + 4S_t^2} \quad (6)$$

According to the ASME B31.3 Code, the stresses to which a piping system is subjected may be separate in three main classes, for which the codes establish limits:

- A) The stresses caused by sustained loads
- B) The stresses caused by occasional loads
- C) 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 B31.3).

The ASME B31.3, establishes a limit to the sustained stress:

$$S \leq S_h W \quad (7)$$

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. The allowable stress at the operation temperature is denoted as S_h .

B. Occasional stresses

Occasional stresses are caused by occasional events, such as water hammer, earthquakes and wind. According to ASME B31.3, $S_L + S_{oc}$ should be lower than $1.33S_h$, where S_{oc} are the stresses produced by occasional loads.

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.

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

$$S \leq S_A \quad (8)$$

Where,

$$S_A = f(1.25S + 0.25S_h)$$

OR

$$S_A = f(1.25 S_c + S_h - S) \quad \text{if } S_h > S_L.$$

In both cases f is the stress reduction factor, defined by each code and S_A is the allowable stress.

[2, 3, 5, 6, 9]

V. 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, 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:

$$W = \int_0^L \frac{P^2 ds}{2AE} - \int_0^L \frac{M^2 ds}{2EI} \quad (9)$$

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 (9), the Theory of the Elastic Center and the principal described above, the following system of equations may be deduced:

$$F_x \frac{I_{xx}}{EI} + F_z \frac{I_{xz}}{EI} = \Delta x \quad (10)$$

$$F_x \frac{I_{xz}}{EI} + F_z \frac{I_{zz}}{EI} = \Delta z \quad (11)$$

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

Solving the system of equations (10) and (11), 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:

$$M_y = F_x z - F_z x \quad (12)$$

where x and z are the coordinates of the point in question, in the referential with origin on the centroid.

A. Reactions

According to ASME B31.3, 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 B31.3):

- For the condition of maximum thermal expansion:

$$R_h = R \left(1 - \frac{2C}{3} \right) \frac{Eh}{E_c} \quad (13)$$

- For the installation condition:

$$R = \max \{CR, C_1, R\} \quad (14)$$

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. 4, obtained by the expression established by the Grinnell Corporation (1981):

$$S = \left(\frac{L'}{L} \right) S_E \quad (15)$$

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.

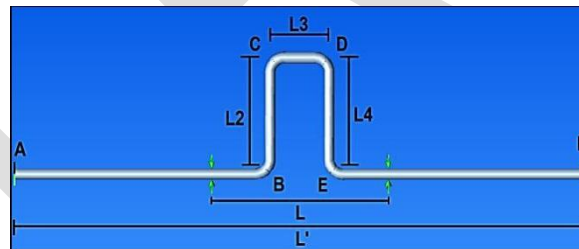


Fig. 4 – Loop with intermediate guides

[1, 6, 11]

VI. Sensitivity Analysis

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 and using the commercial software CAESAR II.

In Loops without guide 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 Fig -5.

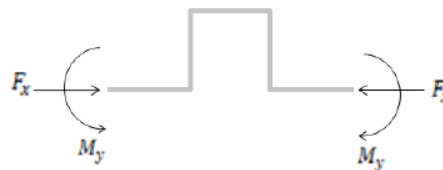


Fig.5 – Free body diagram of a pipe loop, considering only the thermal expansion

In this case, the maximum stress due to thermal expansion will occur in the corner C (Fig. 4). For the stress intensification factor it was considered turns of radius $=3/2D$.

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 6 and 7.

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 these 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 $SE_{E,ax}$ illustrated in Fig. 8, 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 $L_3=1/2L_2$.

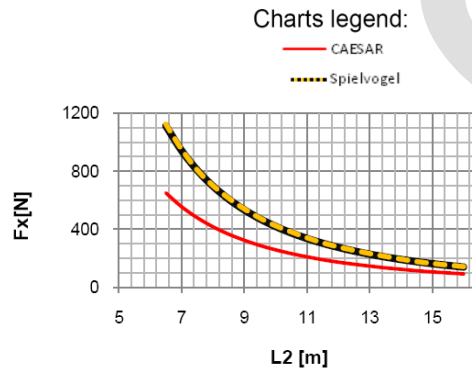


Fig. 6 – Chart of F_x Vs. L_2

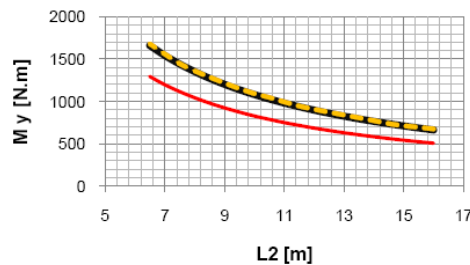


Fig. 7 – Chart of M_y Vs. L_2

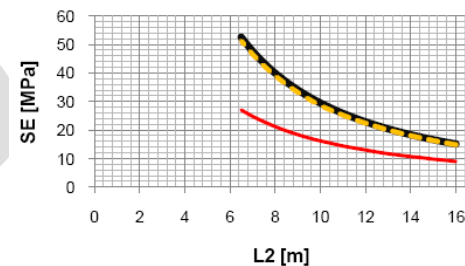


Fig. 8 – Chart of $SE_{E,max}$ Vs. L_2

[4, 6, 9, 11]

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Conclusions

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

According to the ASME B31.3 Code, 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, 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. 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. The determination of the loads caused by the thermal expansion is a much more complex task. The method used to calculate the forces and the moments due to thermal expansion is the Spielvogel Method, it is more versatile than other methods which are dependent of tables and charts. 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 $L_2=2L_3$.

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, more detailed investigation of SIF, more specific parameters based on actual condition, less assumption in analysis software. Even there are limitations to Codes, so rather than over designing, the conservative approach may be applied based on actual conditions and the specific analysis so that material and cost can be saved. Selection of proper methodology is the key to the design optimization.

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