DISCUSSION OF INDUSTRIAL PIPING VIBRATION FAILURE CRITERIA

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Abstract. *Piping vibrations consitute a common problem in diverse industrial segments. Several approaches may be considered to control it, such as changes on the support configuration or, when possible, on the piping itself or even through the elimination of the vibration source, for example. However, before initiating the process of controlling or elimination of these vibrations, it is necessary to assess if such vibrations are effectively excessive and if they really need to be reduced. For such, diverse methods are found on literature and on codes. Nevertheless, after an evaluation of the methodologies used on each procedure, one perceives that the extent of these are limited, since some parameters that affect the vibration response of the piping are not considered in these procedures. Additionally, experimental evaluation of vibration levels on industrial piping show that these procedures are excessively conservative. In this paper some of these different methods of verification of vibration acceptability are presented and the applicability of them is discussed. It also presents an initial attempt, through numerical simulations, to correlate vibration parameters. A detailed study of the results is presented, in order to raise the characteristics of each analyzed case.*

Keywords: *Piping Vibration, Dynamical Analysis, Failure Criteria.*

1. Introduction

Problems of piping vibrations are usually found in several industrial segments. They have various excitation mechanisms, such as the two-phase flow. Similarly, the alternative motion of pumps and compressors is an important source of excitation.

The motivation of this work comes from the occurrence of mechanical failure on oil refineries production units and also on offshore platforms. The accumulated experience of the operators has shown that some of these failures are related to excessive vibration of different piping systems, generating, amongst others, nucleation and fatigue propagation of cracks and/or leakage on flanged connections.

Typically, repair cost may be small, but it always causes reasonable losses due to operation interruption or reduction of the affected unit, besides the potential risk of product leakage. It is estimated that the number of failures due to piping vibration in PETROBRAS refineries are more than two per year and per refinery. Depending on the system, in average, four days of downtime are needed to repair a failure, causing a considerable annual loss. Moreover, it is important to point out that this problem is not specific to the oil industry, as literature supplies a survey of similar problems that have occurred on American nuclear plants.

For this reason, many efforts have been made in order to establish procedures to evaluate this type of problem. Several methods are found in literature, including those of the OM-3 code (ASME, 2002a) which prescribes procedures to determine if the piping vibration level is acceptable in the startup of nuclear plants. However, after an evaluation of the methodologies used in each procedure, one perceives that its coverage are limited, since some aspects are not considered, specially regarding piping configuration. These limitations have already been discussed in technical articles. Silva et al. (2004) apply these methodologies to some real cases, showing the main characteristics and limitations of each criterion.

In the present work, partial results of an ongoing numerical analysis on piping vibration limits are presented. This analysis, conducted through a series of finite element results, automatically managed by macros, is performed to correlate piping vibration parameters. A detailed study of the results was made, in order to raise the excellent characteristics of each evaluated case. With these results, one searches the attainment of a new piping vibration failure criterion that is more comprehensive and more suitable to actual industrial piping than the existing criteria.

2. Piping vibration level criteria

Technical literature contains some criteria for the evaluation of the vibration problems in piping, machines and structures, that specify allowable displacement or velocity amplitudes in function of the vibration frequency. The curves presented in Fig. (1) are well known as they were obtained from previous experiences on vibrations and failures in diverse piping configurations of existing installations (Nimitz, 1974). The procedure consists on crossing the measured amplitude of vibration (in thousandth of millimeters, peak to peak) with the vibration frequency (hertz) to determine the vibration danger level. It is a criterion which has been widely used mainly due to its ease of application in the field.

In general, the statistical data used to generate the criterion shows that it applies acceptably on common piping configurations, but the same may not occur in critical applications or on unreinforced connections. Hence, it is advisable to add an additional limit for these cases. No warning on this type of problem is found and a clearer definition of the limits of what constitutes a typical configuration becomes necessary (Wachel, 1982). On the other hand, Lin (1996) states that the limits work quite well with petrochemical installations, where pipes are generally supported in a very flexible way and that the same ones must be modified when used in nuclear plant piping, which are generally stiffer than their petrochemical counterparts. Anyway, the criterion is considered a good starting point for piping evaluation and is widely used.

For a more detailed analysis, there are the methods presented on the OM-3 code (ASME, 2002a). They are intended for vibration assessments, during preoperation and startup of nuclear plant piping that require tests, as specified on the code, but it can also be used for evaluation of vibration levels during operation. The OM-3 code proposes two evaluation procedures, one based on vibration displacement measurements and the other using velocity data.

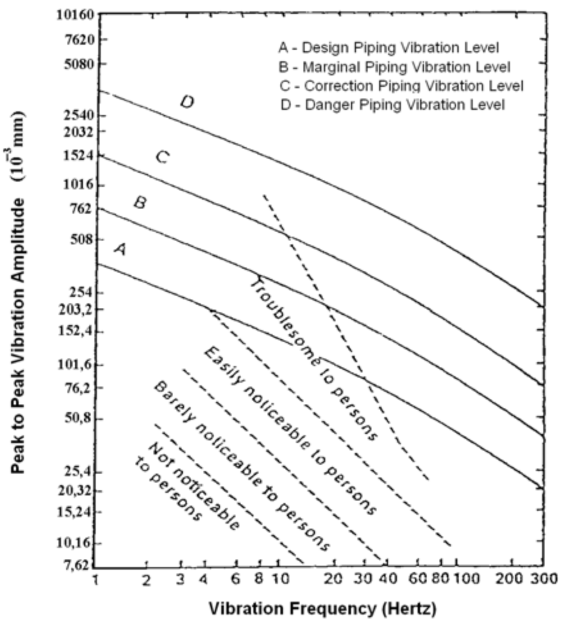

Figure 1. Vibration levels (Nimitz, 1974) and perception limits (Richart, 1962)

Another approach to the problem of determining the acceptability of vibration levels in piping can be based on the comparison of induced stress with the endurance limit to fatigue. However, it is necessary to establish the comparison bases.

The oil refineries piping code, the B31.3 (ASME, 1999) do not supply any details regarding fatigue caused by vibration, because its scope of application is on the design and erection of new plants and, typically, pipings are not designed to support any type of vibration at this level. Yet, there are some cases, such as piping connected to alternative compressors, where vibration evaluations should executed. However, these are exceptions, not the general cases. The vibration evaluation tends to be a post assembly task (Becht, 2002). Although the authors have not consulted the other sections of the B31 code, they believe that they are also omissive on it, since Nayyar (1992) nothing informs on this matter.

The API standard for refinery reciprocating compressors (1995) calls, at its appendix M for the pulsation design of pipes, the fatigue curves of the ASME B&PV code (2002b). For this norm, the endurance limit of low carbon steel was 25000 lbf/pol^2, 172 MPa (peak to peak) at 10^6 cycles and currently is 14000 lbf/pol^2, 96 MPa (peak the peak) at 10^{11} cycles.

On the other hand, as stated in (EDI, 2002) experiments have shown that failures seldom occur on low carbon steel piping when the amplitude of the measured deformation are less than 100 $\mu\varepsilon$, peak to peak. A deformation of 100 $\mu\varepsilon$

represents a stress of 21 MPa peak to peak. Comparing 21 with the fatigue endurance limit of 96 MPa, a ratio of 4.57 is found. This ratio represents, basically, the stress intensification factor for welds on piping.

The stress value that is to be compared with the fatigue endurance limit may be determined through distinct procedures. Wachel (1982), for example, obtains this value through a simplified analytical procedure based on beam deflection analysis combined with appropriate correction factors.

In this work, the adopted approach is the determination of the state of stress by numerical simulation. For such, a model is developed using a finite element program. A modal analysis is performed, extracting the natural frequencies and vibration modes of the model, following a harmonic analysis at the natural frequency of the relevant mode shape. The stresses may be obtained through a macro that calculates the so-called *flexibility stress*, in accordance with the B31.3 code (ASME, 1999). The flexibility stress is Tresca's equivalent stress at the transverse section of the pipe, due to the superposition of the bending and torsional moments on the section. The obtained value is multiplied by the 4.57 factor, which already includes the stress intensification factor, and then compared with the fatigue endurance of low carbon steel at 10^{11} cycles, 48 MPa (zero to peak) (ASME, 2002b).

3. Vibration stress analysis on an L branch

The simplified methods, such as of Fig. (1), are widely used due to its easiness, particularly for on field screening purposes, so only the piping systems that fail to pass are subjected to a thorough numerical analysis. Due to this, a numerical analysis on piping vibration limits is being conducted at UFF, in order to check and enhance the vibration levels chart presented at Fig. (1). The chosen configuration to begin the studies with was a simple but widely found on field one: the L branch.

3.1. Methodology

In order to get the most of information on the induced vibration stresses on an L piping branch (Fig. (2)), it is necessary to perform analyses of several different configurations. In other words, it is required to prepare a parameterized model and an iterative macro in order to analyze and retrieve the results for different parameter values. The chosen parameters were the nominal diameter, nominal wall thickness (schedule), total length of the branch, length ratio between the branch legs, fluid density and internal pressure.

The nominal diameter and wall thickness were based on the ANSI B16.9 standard. The following nominal diameters were used: 2, 4, 6, 8, 10, 12, 16 ,18, 20 and 24 inches, while only the STD and XS schedules were considered. The total length of the branch (A+B) was of 10, 15, 20, 30, 40 and 50 times the outer pipe diameter. The size of each leg was calculated in function of the ratio of the leg A length relative to the total length (A/(A+B)). The range of this length ratio was chosen in order to cover a great amount of cases, varying from 0.1 to 0.5, with an increment of 0.02.

The fluid density affects the mass of the model while the internal pressure, due to a pressure stiffening effect, modifies the stiffness of the bend. This effect is represented by the flexibility factor, which is given in pipe codes and implemented on curved pipe elements of finite element programs. The fluid density parameter was expressed in terms of relative density, being either zero (negligible added mass) or one (the fluid is water) while the internal pressure parameter was expressed as a fraction of zero, half or full allowable pressure. The allowable pressure was calculated from the hoop stress formula considering the basic allowable stress for the API 5L Gr B carbon steel (138 MPa), a common material used in process piping.

For each iteration, that is, for each set of values of the above parameters, the procedure was:

1. Assembly the model in accordance with the given parameters.
2. Perform a modal analysis to extract the first natural frequency, the associated mode shape and the node and direction where the largest displacement occurs.
3. Perform a harmonic analysis at the first natural frequency of the model, applying a 10 mm displacement amplitude at the nodal direction where the largest deflection occurs.
4. Determine the largest flexibility stress caused by the harmonic displacement loading on the L branch.

The ANSYS finite element program was employed to undertake this analysis. The L branch model was mounted using its piping module commands and elements. A macro was written in order to calculate the stresses employing the proper in-plan and out-of-plan stress intensification factors at the bend. Another set of macros was prepared in order to perform the modal and harmonic analyses for each set of parameters.

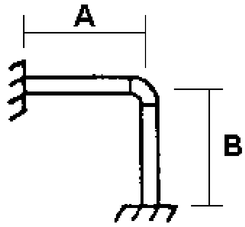

Figure 2 – L branch

3.2. Results of the dynamical stress analysis

After the results of the analysis have been collected, this data may be plotted in different ways in order to determinate the correlation of the stress results with the different parameters. Among these plots, the stress versus frequency of the whole data set, presented in Fig. (3), furnishes a picture of the overall behavior of the results. As explained on the previous item, the plot abscissa represents the first natural frequency of the branch. Each curve of the graph represents the variation of the A/(A+B) length ratio with the other parameters kept fixed. As shown on the plot, the pressure stiffening of the branch bend has small influence on the results. The added mass of the fluid, given by the fluid density parameter, modifies the results, mainly due to the reduction of the natural frequency.

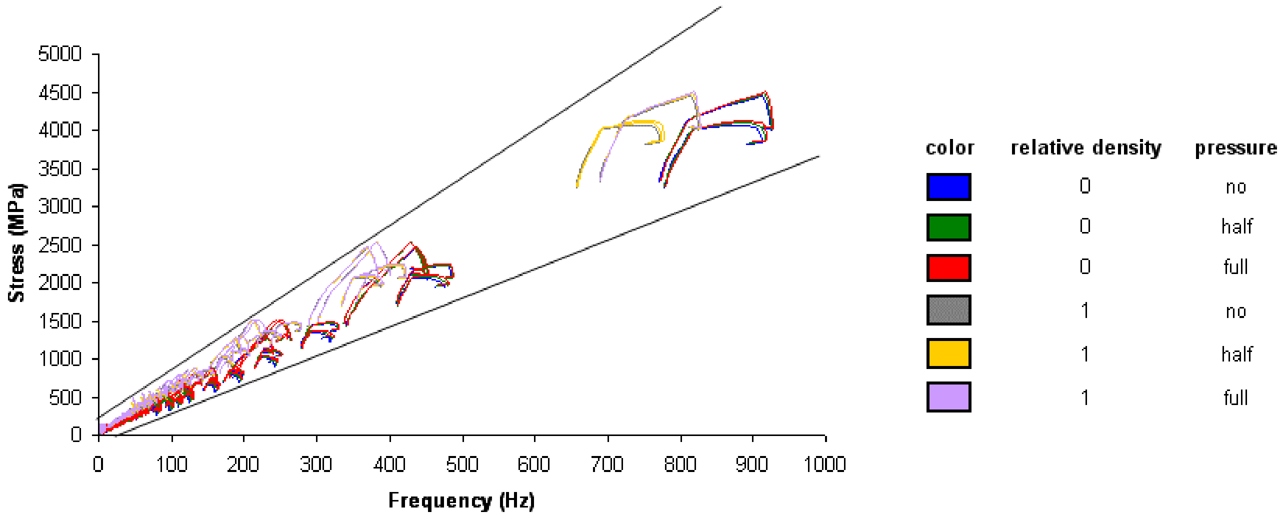

Figure 3 – Stress vs. Frequency: Stress relative to an applied displacement of 10mm with piping vibrating at the first natural frequency. The point of application and the direction of the displacement are supplied by the vibration mode shape.

Some curves presented some points where great stress variations occurred, generating inflection points on the curves. To better explain the behavior of these, the regions shown in Fig. (4) were analyzed in detail. The 24 inches nominal diameter (609.6 mm outer diameter) was chosen because it presented the most expressive results in respect to inflection points. However, the same comments apply to other diameters since they also presented the same behavior, only on a softer form. The results depicted on fig (4) and the following discussion were based on a preliminary analysis, where the length ratio increment was of 0.01.

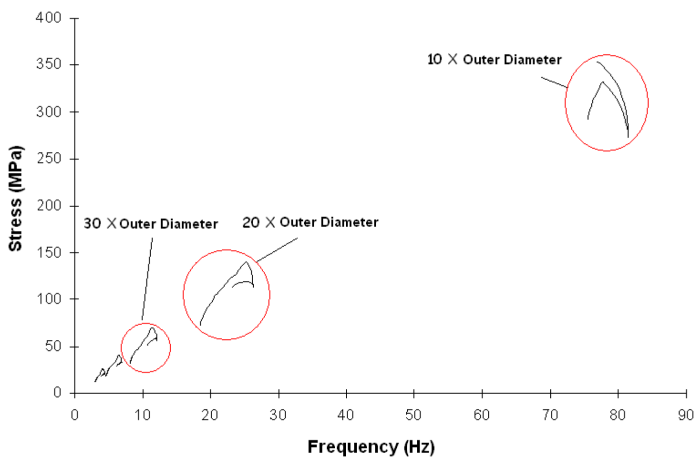

Figure 4 - Stress vs. Frequency for 609.6 mm outer diameter, schedule STD, no fluid and no internal pressure.

Fig. (5) shows detailed plots of the Stress (MPa) versus Frequency (Hz) for the case where the total length corresponds the 10 X outer diameter. In prominence, the regions where the inflections occur. For the first region, the results for cases A, B and C are listed in Tab. (1) while Fig. (6) presents the element numbering of the model. As the A/(A+B) length ratio for the 3 cases are approximately equal, Fig. (6) represents well all these cases. In all three cases, the maximum stress occurs at the first node of element 6, that is, the left support. This is expected since the vibration mode is out of plan.

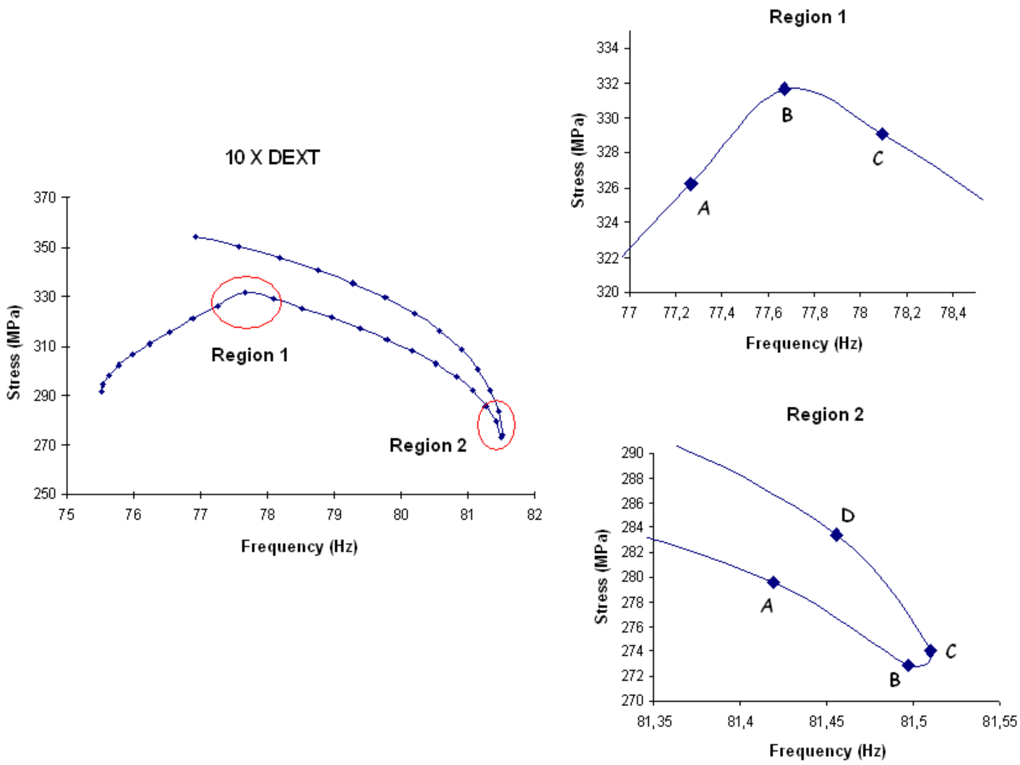

Figure 5 - Stress vs. Frequency –Total Length equal to 10 X Outer Diam., detailed analysis of the inflection points.

Table 1 – Results for the cases shown in region 1 of Fig. (5)

Case	A/(A+B) ratio	Leg A (mm)	Leg B (mm)	Natural Frequency (Hz)	Stress (MPa)	Element	Node
A	0.42	2560.32	3535.68	77.27	326.231	6	first
B	0.41	2499.36	3596.64	77.67	331.648	6	first
C	0.40	2438.40	3657.60	78.09	329.108	6	first

Although the point where the maximum stress occurs is the same for the 3 cases, Fig. (7) shows that the point where the piping presents the largest displacement is different. While for cases A and B the maximum displacement is on the bend, on case C the same point is after the curve. Hence, although for case C the A leg is smaller, the left support is farther from the maximum deflection point, and, therefore, less loaded than the other two cases. The point of inflection of region 1 then represents a change on the vibration mode, modifying the point where the largest displacement occurs.

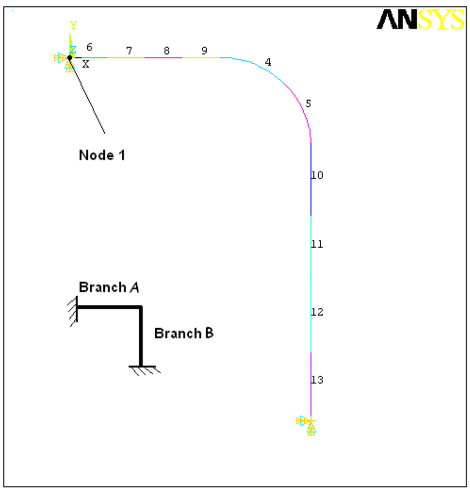

Figure 6 – Element numbering; region 1

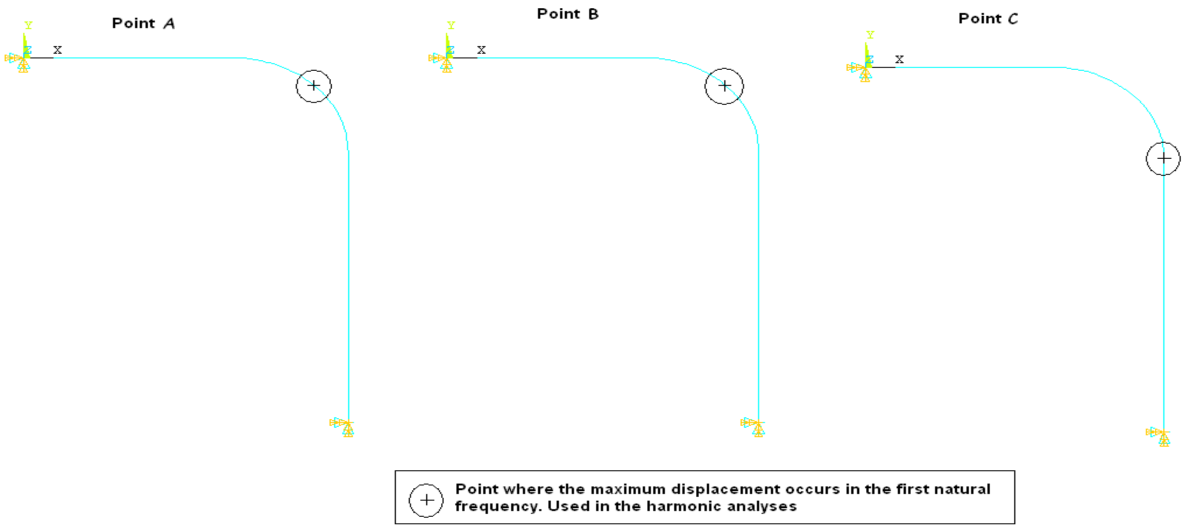

Figure 7 - Points where the maximum deflection occurs for cases A, B and C of Fig. (5), region 1.

For the second region of Fig. (5), the results for the cases A, B, C and D are listed in Tab. (2). The first node of element 6, as shown on Fig. (8), is located at the left support, and as expected, it is the most requested point for the A and B cases of region 2.

Table 2 – Results for the cases shown in region 2 of Fig. (5)

Case	A/(A+B) ratio	Leg A (mm)	Leg B (mm)	Natural Frequency (Hz)	Stress (MPa)	Element	Node
A	0.30	1828.80	4267.20	81.42	279.532	6	first
B	0.29	1767.84	4328.16	81.49	272.877	6	first
C	0.28	1706.88	4389.12	81.51	274.084	13	sec.
D	0.27	1645.92	4450.08	81.46	283.391	4	first

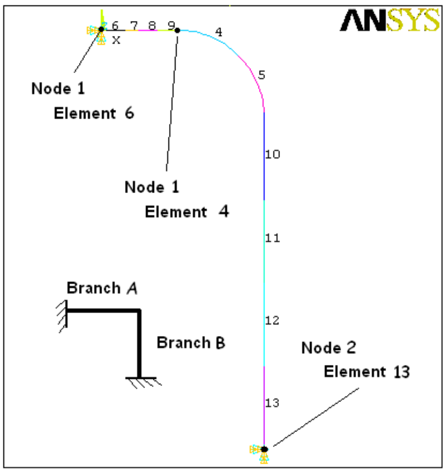

Figure 8 – Element numbering; region 2

Although, for these four configurations, the vibration mode is the same, a change in the localization of the largest stress occurs. In the configuration referring to the points A and B this is located at the left support, while in points C and D, changes occur. In configuration C, the maximum stress is located at the other support, however it is important to mention that the stress at the first node of element 4 (curved element) is very close to this (273.981 MPa). From point D onward the largest stress is located at the first node of element 4.

The cause of this new stress localization, next to the curve, happens mainly because of the use of the stress intensification factors prescribed by the B31.3 code.

For the configurations whose total length corresponds to 20 X outer diameter and 30 X outer diameter, the results are similar to the previously presented ones, i.e., modifications on the vibration mode occur, dislocating the point of maximum displacement, or there are changes on the point where the largest stress ocurrs. If the analyses are performed without any the stress intensification factor in curved elements, the changes only occur for the 24 inches nominal diameter piping with total length equal to 10 X outer diameter. This leads to the conclusion that the inflections for the other cases are only due to changes of the maximum stress position from one support to the other.

In addition to the results displayed in fig. (3), it is also possible to establish an initial curve of allowable amplitude with the information generated from the analyses carried out. As discussed in section 2, an allowable stress value of 21 MPa (peak the peak) is employed. This curve is plotted at fig. (9) together with the vibration level graph discussed previously.

4. Conclusion

The analysis of the L branch shows that for a group of curves, similarities exist among the curves, which suggests the existence of a standard.

The results potted at fig. (9) shows clearly the possibility of establishing a simplified criteria of evaluation of levels of vibration through numerical simulations. This has not been found in current literature.

It is important to stand out that these are only initials results and greater studies are necessary. The same analyses need to be performed for other configurations, such as Z branches (in and out of plan), for example.

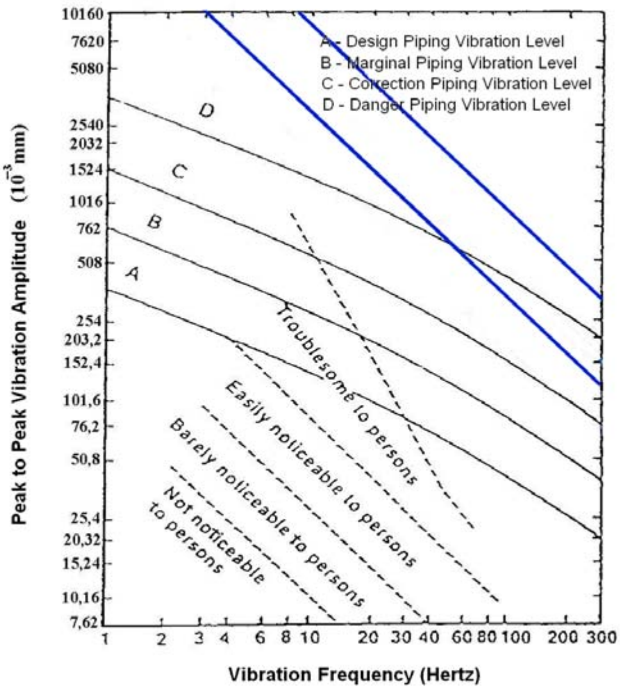

Figure 9 - Comparison of the numerical results with the displacement amplitude criterion

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