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PRACTICAL APPROACH TO STRESS AND DISPLACEMENT CALCULATION OF A CLAMP FIXTURE FOR QUICK OPENINGS OF PRESSURE VESSELS

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Abstract. *Mechanical design is essential for critical equipment that threatens life if it fails. Quick opening closures are used for easy access to insert and removal of PIGs, Pipeline Inspection Gauges, used for cleaning and inspection of pipelines. The opening is a large part of the station that has a similar behavior as a pressure vessel. This work compares two methods, analytic and numerical to dimensioning the closure according to standards. Despite the complex geometry of the closure, it can be designed using analytical methods without the need of expensive numerical tools.*

Keywords: *quick opening closure, analytical method, FEM,*

Nomenclature

ΔDx1 - Axial displacement of the Clamp Arched Body;
ΔDx2 - Flexion Displacement on the Clamp's Lips;
D - Hub Shoulder Diameter;
d - Hub body external diameter;
r - Hub shoulder transition radius;
ΔHx1 - Hub shoulder flexion displacement;
ΔHx2 - Partial axial displacement of the Hub;
ΔHx3 - Hub shell bending displacement;
and - Dimensional characteristic of the U-shaped concentrator;
r - Concentrator radius in U;
Pp - Design pressure;
B - Inner Hub diameter;
g1 - Thickness of the thick part of the Hub;
Mo - Distributed bending load;
Cw - Clamp Width
H - Total Hydrostatic Force
Ci - Internal Diameter in Clamp
Lc - Effective length of clamp lip
Cg - Opening the Clamp
Ct - Effective Clamp Thickness
C - effective diameter of the Clamp reaction circle

1. INTRODUCTION

The Oil and Gas (O&G) Industries must to achieve high efficiency in order to profit on a low price market. To increase efficiency also means better, safe and reliable maintenance operations. Moreover, to efficiency also search to reduce downtime in maintenance operations. Rausand (2010) also affirms that O&G equipment needs high reliability level, equivalent to 97% availability, to industry success. This equipment must to obey the Brazilian and International Norms and Legislation, which stablishes rules for design, manufacturing, operation, maintenance and test parameters. However, the standards do not cover all details in the equipment requirements.

Beyond the standards, designers also face difficulties in having appropriate tools for designing and dimensioning especially when the geometry of the equipment is not covered in the handbooks. In most cases when this happens the designer must use numerical tools to be able to evaluate his design. Nevertheless, in some cases it might not be necessary and expensive software can be avoided. This paper presents an evaluation of analytical and numerical methods to evaluate a design of a quick opening clamps used in pipeline inspections gauge (PIG) launch and receiver stations. During the launch and retrieving procedures of PIGs the station work as a pressure vessel, nevertheless, the design of the opening clamp is not covered by standard design procedures.

2. METODOLOGY

The first step for the design was to evaluate all the standards and references that cover the design of the quick opening and possible similar designs. The main goal for this product that is sold separated from the station was to reduce manufacturing costs and keeping the design safe and reliable. Nevertheless, these design considerations are not the focus of this paper that is concern about the dimensioning of the quick opening closure.

The technological bases to verify stresses and displacement levels on sealing system were based on the product application requirements and acceptance factors for limiting criteria. Based on this, the Boiler and Pressure Vessel Code from ASME (2013), as done by Flanders (1991), was considered to define the main dimensions and the allowable stresses (Flanders, 1991) using the analytical method. As the use of the numerical method, it was used Finite Element Method, FEM (Jones, 1989) as a way of guaranteeing the safety parameters of the product.

Other important factor necessary for the calculus of the equipment was to predict the friction coefficients (Qin, 2011) in order achieve a optimized structure. So, the contact surfaces of the wedges that have higher friction coefficients can guarantee a stable and solidary connection between the moveable parts of the structure. However, the standards (ASME, 2013) refers to "lugs" devices fastened by screws and the design developed doesn’t have this characteristic. Then it is an issue that must be addressed outside ASME guidance. In Figure 1 it is presented the current design and the main components of the quick opening.

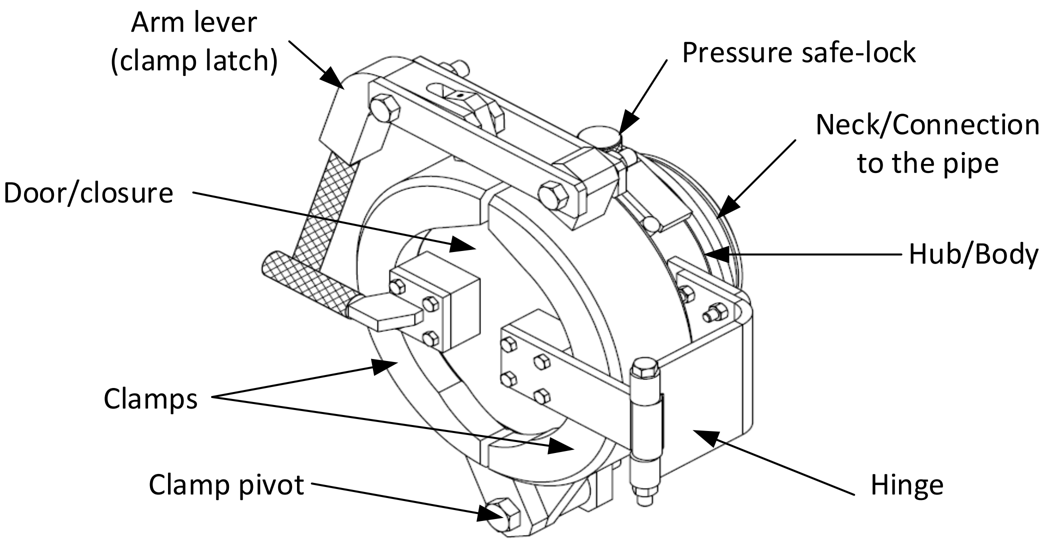

Figure 1. Components of the quick opening closure.

The inner pressure, temperature variation and external forces generates individual components main loads. The main parts that are the hub (body), the closure (door) and the clamps that hold the closure to the hub. For the analytical calculus the equipment was simplified as an axisymmetric solid. In Figure 2, the expected behavior/deformation of the parts is presented. These loads generate a deformed shape on each part, but also generate rigid body displacements among the components due to the mechanical gaps or assembly adjustments and gaps from manufacturing tolerances. This analytical calculus approach objective is to estimate loads on components, predict deformable body, rigid body displacement and applied stress in order to evaluate sealing and integrity of the quick opening.

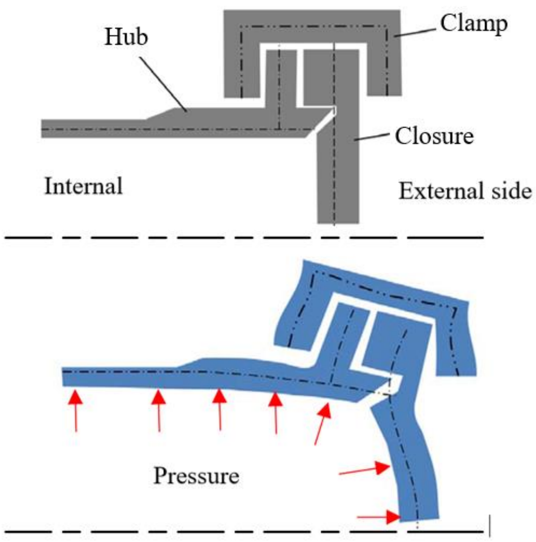

Figure 2. Expected displacement, device axial section view.

2.1 Loads analytical calculation approach model

The clamp lips may have wedges which modifies the stress concentrator at the lip root at the transition between the clamp body (ASME, 2013). However, concentration stress evaluation became an hard engineering task because no previous work was found by analyzing an arcuate body with rectangular notches with rounded corners in the cross section, Figure 3. In this case, the concentrator was considered as an infinite plate with rectangular notch with rounded corner as a "U" shaped element analogy (Pilkey and Pilkey, 2008) and planned to be validated through CAE (Computer Aided Engineering) modeling.

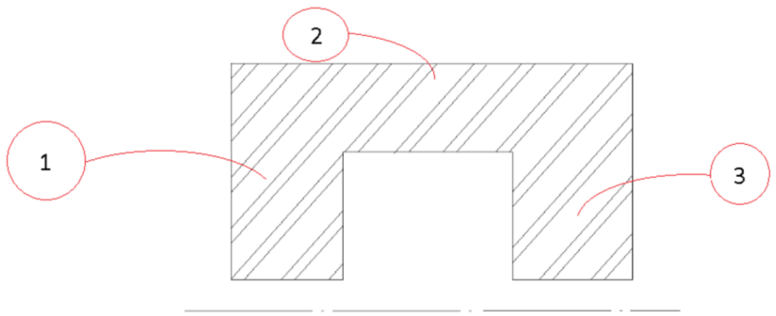

Figure 3. Clamp axial cross section (2- clamp body, 1 and 3: clamp lips).

The approach applied to clamp consider the clamp lips as rectangular plates (Timoshenko and Woinowsky-Krieger, 1959) subjected to simple bend, generating as reaction a uniform linear moment and loads according to Figure 4, respecting the dimensions and thicknesses.

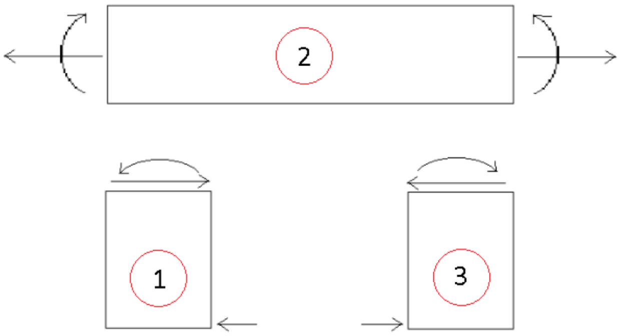

Figure 4. Simplified load distribution for the clamp.

Thus, the clamp body bending load was calculated due to the lips bend moment (eq 1):

$$M_{c13} = \frac{H \cdot l_c}{2 \cdot \pi \cdot (C_i + 2 \cdot l_c)} \tag{1}$$

Bending stress on the clamp's lip (eq. 2) :

$$S_{C13} = 6 \cdot \frac{M_{C13}}{\left(\frac{C_w - C_g}{2}\right)^2} \tag{2}$$

Shear Stress on the clamp's lip (eq. 3):

$$S_{CC13} = \frac{H}{2 \cdot \pi \cdot \left(\frac{C_i}{2} + l_C\right) \cdot \left(\frac{C_w - C_g}{2}\right)} \tag{3}$$

Area of the clamp body subjected to longitudinal stresses (eq. 4):

$$A_{C2} = \pi \cdot \frac{\left(\frac{C_i}{2} + l_C + C_T\right)^2 + \left(\frac{C_i}{2} + l_C\right)^2}{2} \tag{4}$$

Normal longitudinal stress in the clamp (eq 5):

$$S_{C2} = \frac{H}{2 \cdot A_{C2}} \tag{5}$$

Clamp body flexion exertion due to momentum in the lip (eq 6):

$$M_{c13} = \frac{\frac{H}{2} \cdot l_c}{2 \cdot \pi \cdot \frac{(C_i + l_c)}{2}} \tag{6}$$

Bending stress in the clamp body due to the moment in the lip (eq. 7):

$$S_{C13} = 6 \cdot \frac{M_{C13}}{C \cdot t^2} \tag{7}$$

Combined stress on clamp lip root (eq. 8):

$$S_{Ct} = S_{C13} + S_{C2} \tag{8}$$

The acting stress on the clamp body will have the effect of a clamp-shaped tension concentrator, similar to a plate with a rectangular concentrator on a semi-infinite plate subjected to normal stresses (Pilkey and Pilkey, 2008).

2.2 Analytical mathematical approach for for displacements calculation model

For the sealing region, where the function of the system depends on the displacement achieve, displacements calculus was considered on three different components that interact with each other. Displacement in the hub shoulder, at the O-ring, point H; the clamp lips opening effect at point D and cover displacement at o-ring region, point F as indicated in Figure 5.

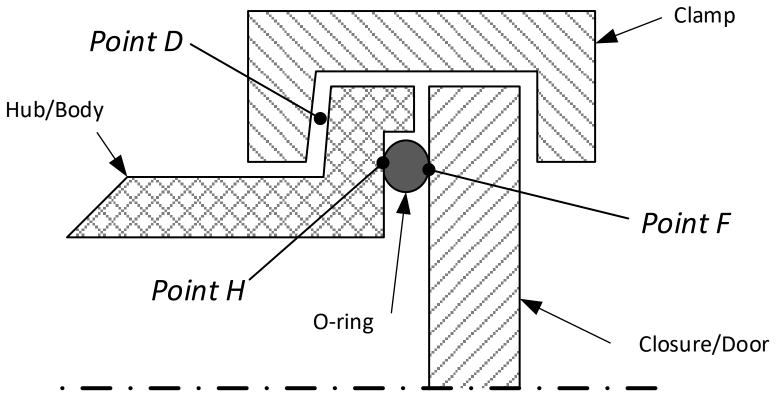

Figure 5. Schematic indication of the sealing region (clearances augmented).

As the body deforms it is expected bending of the clamp lips and also the clamp body elongation. Additionally, the hub shoulder bends and displace. Moreover, the closure plate also bends. All dis deflection/displacements could be summarized as three main critical displacements: ΔD, as total displacement from clamp; ΔH, as hub displacement; and ΔF, as cover plate contribution. The distortion/displacement is indicated in Figure 6.

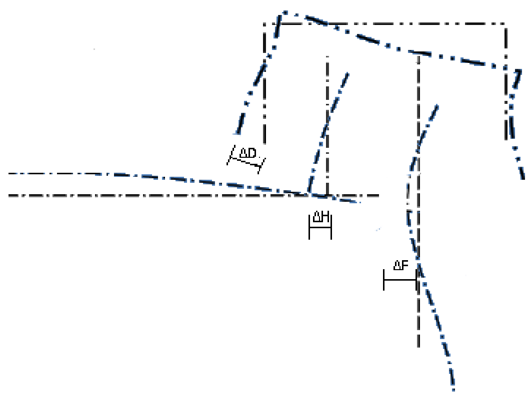

Figure 6. Most important displacements close to the sealing region.

Considering the linear elastic regime for small displacements, it is possible to add the displacements obtained by different loads, using the principle of superposition. Where the displacements due to axial loads will be estimated as $\delta l = FL / EA$ and $y = Ml^2 / 8D$ for the bending displacements.

The displacements of the arched body of the Clamp due to flexion will also be considered, since the body of the Clamp suffers the flexion on the same forces of moment that the lip of the Clamp.

The approach used for the solution was to consider the analogy to a rectangular flat plate, while retaining the original dimensions. This consideration can be made conservatively for the evaluation of the displacement of the body of the Clamp, part 2 of Figure 4, according to Timoshenko and Woinowsky-Krieger (1959).

For the flexion displacement of the Clamp lip, this is considered as a rectangular flat plate, similarly and maintaining the original dimensions, but fixing it on one side, at the junction with the body. In this way, the total displacement, in the X axis, of the clamp only, can be calculated as in Eq.

$$\Delta Dx = \Delta D_{x1} + \Delta D_{x2} \tag{9}$$

For the hub's total displacement, it was made similarly to the clamp. The first part is the component due to the flexion displacement of the hub's shoulder, calculated according to Budynas and Young (2002), as a circular plate with the center set. In addition to the axial displacement, the displacement due to the flexing shown, allows the engineer to analyze the Hub as a thick-walled tube under the effect of internal pressure. The bend displacement of the cap is calculated directly using Budynas and Young (2002), associated with a linear interpolation with respect to the point of maximum displacement, in order to find the displacement in the sealing region.

The value of the full aperture of the sealing region can be compared to the permissible limit deformation in the o-ring without losing the sealing. This can be compared as the sum vector of the hub, clamp and closure displacements (eq. 10).

$$\Delta Hx = \Delta Hx1 + \Delta Hx2 + \Delta Hx3 \tag{10}$$

That said, what is expected is to validate the simple way of estimating the displacement of the H point in Figure 5 and to be able to compare with the acceptable variations for the sealing ring to be selected. And this simple approach would be coherent as long as the verified voltages were below the design limits, especially the flow (Sy) and the analytically calculated voltages converged to those calculated by the Finite Element Method.

2.3 Definitions and descriptions of the numerical model

These numerical models were made by choosing the appropriate elements for analysis and refining in the transition regions. In this way, the meshes were made with quadratic hexahedral elements, with a greater refining in the tension concentrators, as can be seen in the following figures. The bore regions had more than 36 elements along its circumference, and the fillets had more than 8 elements, in order to ensure better results along the discontinuities.

Material models and boundary stresses were defined in the same way as the analytical calculation. The meshes of this model are shown in figures 7 and 8.

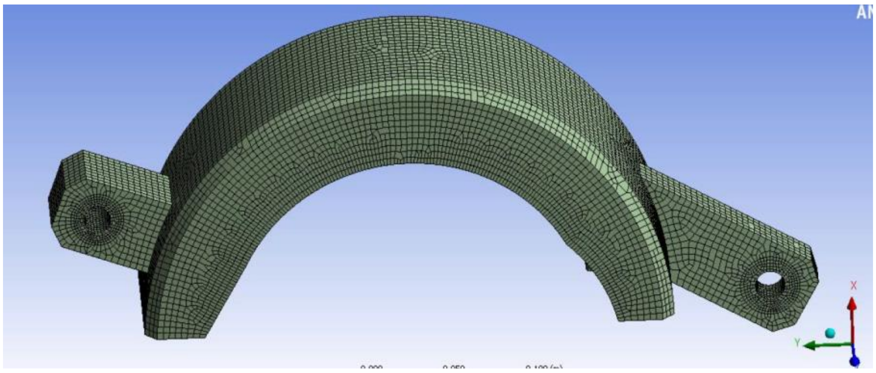

Figure 7. generated finite element mesh.

The loads and boundary conditions applied in these models were exactly the same as those calculated for the design and for the hydrostatic test of the analytical model.

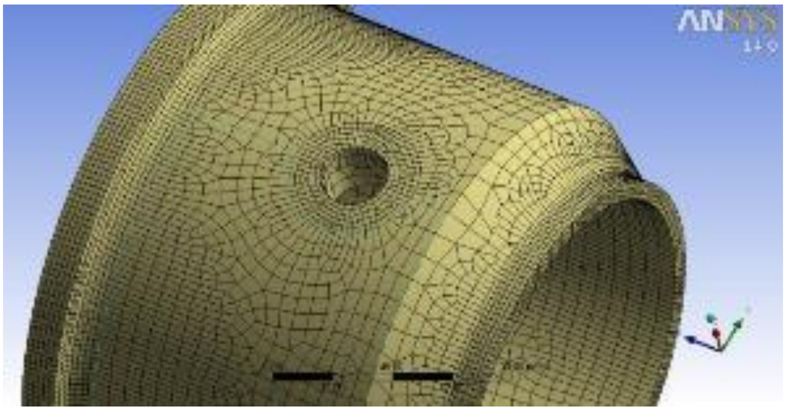

Figure 8. Hub generated finite element mesh.

The hub model was prepared in an analogous way, considering the internal pressure applied on the inner walls, and forces applied due to the action of the clamp lips on the Hub shoulder.

3. COMPARISON BETWEEN ANALYTICAL MODELS AND CAE

For the validation of the models used in the Hub and Clamp for stress calculations, a comparison was made between the most important recurrent voltages between the analytical and finite element models, using the permissible voltages as a comparison.

In the tables below, all values are presented, identifying the regions of interest for the nominal and local stress analyzes, and the percentages of comparison between them and the permissible ones (analytical% and MEF) and between them (% DIV).

Observing the difference between these, it is clear that there is a consistency in the results, since the divergences were less than 8%.

To perform these analyzes, the ANSYS® software was used. Special attention was given to regions of stress concentrations, such as the holes and ears of the clamp. In these regions, the element quality indicator (warping and aspect ratio) was strictly controlled so as not to interfere with the final result of the model.

The tensions concentrated in the Clamp, whose voltage concentration factors were calculated by Eq. 10, according to Pilkey and Pilkey (2008).

$$Kt = 4{,}141-2{,}760 * (e / r) + 0.838 * (e / r) 2-0.082 * (e / r) 3 \tag{10}$$

Figure 9 shows the normal stress distribution in the inner side of the clamp, showing the maximum stress ratios in the concentration regions, considered as "U-Shape".

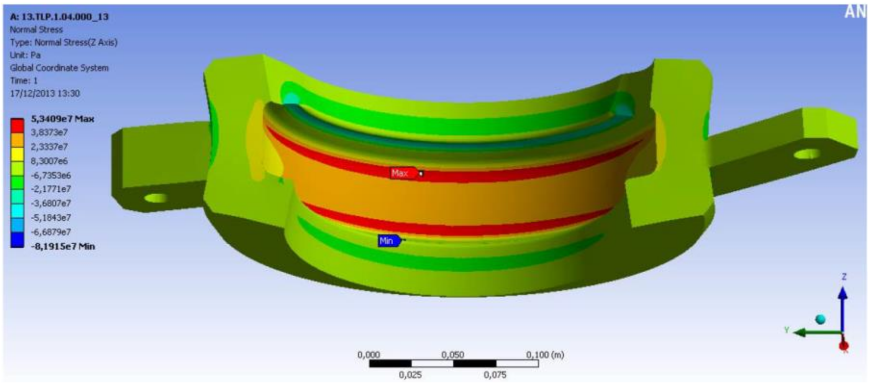

Figure 9. Clamp normal stress distribution.

With the analytical and numerical results, the difference between the stresses calculated by the two methods was very small, due to Sy, with about 1% difference. From the tensions in the clamp one can notice a good correlation between the points of maximum tensions with the suggestions of analytical calculation of Pilkey and Pilkey (2008). The comparison for two points is presents in Table 1.

Table 1. Comparison of longitudinal tensions in the clamp (%Sy)

Place/Type	Numerical	Analytical	Difference
Lip Fillet/ Concentrated (OZ)	46.4%	47.3%	0.9%
Internal Clamp Surf. Body/Nominal (OZ)	27.7%	33.8%	6.1%

Calculated displacements methods were fairly close in the results. The numerical evaluation for the O-Ring region on the Hub revealed an axial displacement of about 0.0124% of Hub thickness, as calculated analytically.

4. CONCLUSIONS

In the absence of numerical analysis tools, analytical approaches can be made, but under more careful and conservative strategies, as keeping stress level under 72% of limiting stress as Contreras (2011), chose adequate material as NACE (2009, 2016) with good surface finish as Campos (2013) recommendations, in order to fatigue problem preventions. These strategies should be adopted at all stages of the project, but more accurately in the evaluation of all the stresses requested, in the dimensional characteristics of the geometry, in the chemical properties, in the mechanical properties of the material, in the surface finish and in the thermochemical treatment.

In view of the convergence of the observed results this approach was considered satisfactory for the conceptual development stage of the piece. But in no case is it recommended that only these actions be done to verify the structural integrity and the functional reliability of the equipment. Physical performance and resistance tests are indispensable and must be done to attest to the reliability of the product as well as determine the standards. Regarding the methodology, this was considered satisfactory, so far, given the convergence of results, from the point of view of stresses, mainly of the calculated voltages and the compatibility of displacements. These were kept within the Linearity regime, which supports the methodology proposed for the calculation of displacements.

5. ACKNOWLEDGEMENTS

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