STEEL PIPE CLAMPS - STRESS AND FRICTION CAPACITY ANALYSIS

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ABSTRACT

Pipe clamps are often seen as simple elements used for the connection and fixation of pipes and support elements. In case of existing piping, especially in nuclear facilities welding to existing piping is often not acceptable or extremely costly. Friction based supports could offer an alternative approach if their design is proven and the occurring loads can be transferred without sliding or clamp failure.

This paper describes the use of friction clamps for the connection of viscous fluid damper to piping systems that are subject to severe seismic loading. A main advantage of these elements is the ease of installation and the omission of any welding.

The pipe is connected to the damper only by friction basically using a welded clamp consisting of top and bottom parts. A number of accurately pre-stressed bolts provide the necessary compressive force and friction.

For choosing optimal clamp dimensions (thicknesses, widths, bolts number, diameters, etc.) and the proper preload of the bolts a set of design rules has been developed in order to achieve acceptable stress levels in the clamp at maximum design load and sufficient friction capacity in case of torsional loads.

The paper also presents the results of structural finite element analysis of several steel pipe clamps used for connection of viscous dampers to pipes with different dimensions. The main purpose of this analysis is to prove that the clamp sustains the maximal seismic force transferred from the damper without sliding between tube and clamp surfaces and without clamp failure. The paper provides the description of the main steps of analysis and the comparison with the proposed analytical method of assessment.

INTRODUCTION

High Viscous Dampers (HVD) are intensively implemented in Nuclear and Conventional Industry for the protection of piping systems and equipment against a wide range of dynamic loads (Reinsch et al, 2007; Kostarev et al, 1993).

The general design of friction steel clamps used for the connection of viscous dampers with the existing pipe line is shown in Figure 1.
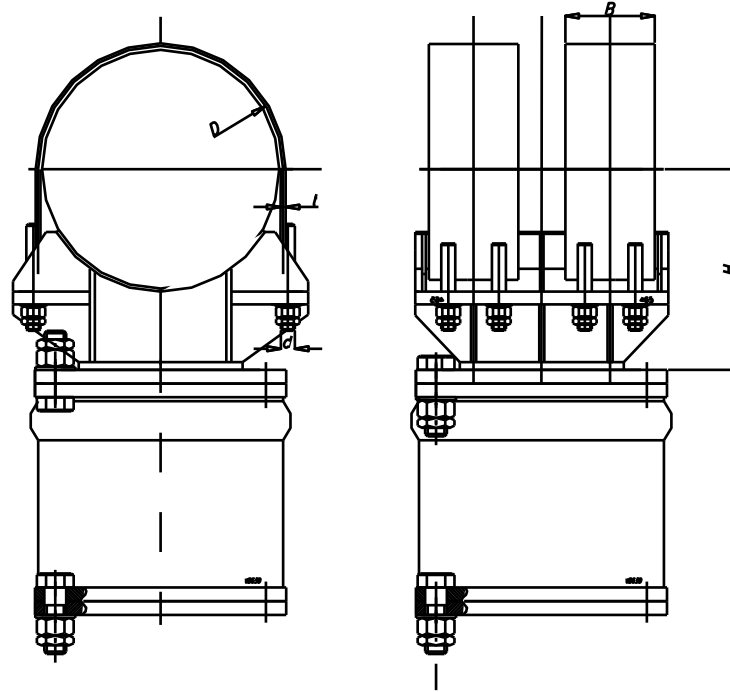
Figure 1. General steel pipe clamp design

The clamp consists of top and bottom parts. Both parts enclose the pipe and are tightened by several bolts. The top part usually consists of thin straps and some kind of stiff connection with bolts. For the general design shown in Figure 1 the bolts are welded to the straps. The bottom part of the clamp is connected to the damper, either directly or by an appropriate adapter. This part is much stiffer than the upper straps and usually has a narrower contact surface with the tube than the upper part.
The particular clamp design can be different but in all cases the pipe is connected to the damper only through the frictional contact. Therefore it is important to choose proper dimensions for the preliminary clamp design to have sufficient frictional capacity under maximal occurring torque loading. In addition all clamp parts must sustain loads and stresses from preloaded bolts and maximum external loads without failure.

The analysis shows that the simplest mathematical schemes with the assumption of uniform contact pressure cannot give reasonable prediction of the clamp friction capacity. Only taking into account the contact pressure gradient and the influence of the internal pressure can give acceptable results (the thermal gradient for hot tubes is also of a considerable influence but it is not studied within this paper).

MATHEMATICAL MODEL

In order to get simple approximate results we can neglect all stresses variations through the clamp width. In real clamps there can be a considerable stress gradient through the strap width but the analysis shows that neglecting this effect helps to get simple but rather accurate equations.

In order to consider the non-uniform pressure distribution between the strap and the tube due to friction the equilibrium of forces for the whole strap and a strap element can be evaluated (see the Figure 2). For the shown conditions of the maximal external torque T (exhaustion of total friction capacity) we can assume that sliding occurs all along the full contact surface between strap and tube and that all friction forces have the same tangential direction.
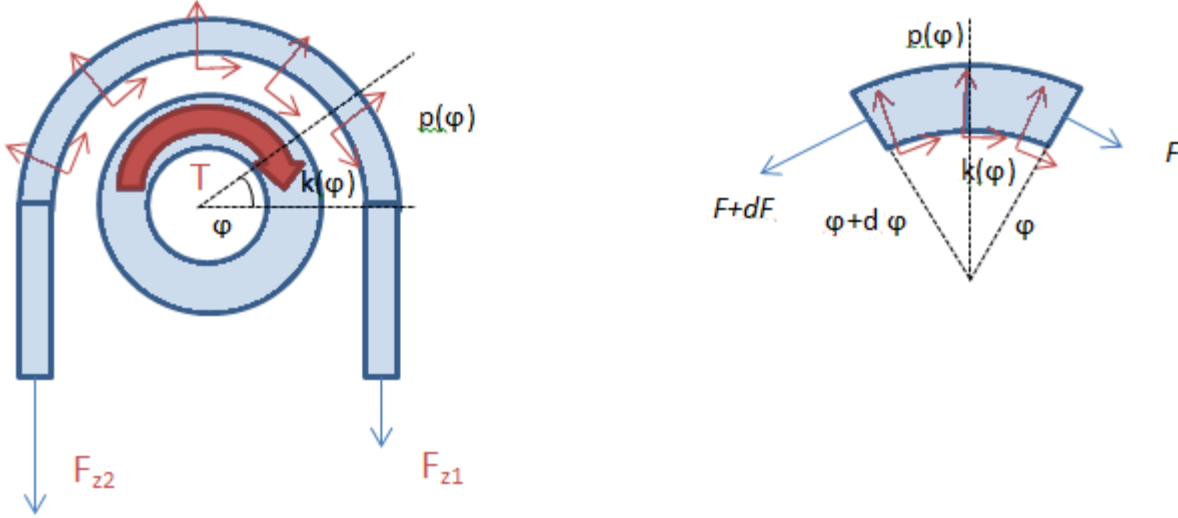
Figure 2. Strap equilibrium

Assuming that on both cross sections of the strap element the forces F and $F+dF$, normal pressure $p(\varphi)$ and tangential load $k(\varphi) = f \cdot p(\varphi)$ (where f is the friction coefficient) are applied we will get the system of Equations 1:

$$\begin{cases} F + dF - F = f \cdot p \cdot B \cdot d\varphi \cdot \dfrac{D}{2} \\ 2 \cdot F \cdot \dfrac{d\varphi}{2} = p \cdot d\varphi \cdot \dfrac{D}{2} \cdot B \end{cases} \tag{1}$$

This gives us the system of Equations 2:

$$\begin{cases} F = \dfrac{p \cdot D \cdot B}{2} \\ \dfrac{dp}{d\varphi} = f \cdot p \end{cases} \tag{2}$$

So the pressure distribution under the strap has the form of Equation (3):

$$p(\varphi) = p_0 e^{f \cdot \varphi} \tag{3}$$

Here the $p_0 = p(0)$ – the minimal pressure (left contact point in Figure 2) the $p(\pi)$ – maximal pressure (right contact point in Figure 2). These exponential relations between the forces in the strap and angle were proposed by Shoghi et al. (2003) for the flat band clamps. The total torque T_s taken by the considered contact is equal to the expression in the Equation (4):

$$T_s = (F_{z2} - F_{z1}) \cdot \frac{D}{2} \tag{4}$$

In the Equation (4) Fz1, F_{z2} are the tensile forces in the strap acting on it's both sides (see Figure 2). From (2), (3) the Equation (5) can be found.

$$\begin{cases} F_{z1} = \dfrac{p_0 \cdot D \cdot B}{2} \\ F_{z2} = \dfrac{p_0 \cdot D \cdot B}{2} \cdot e^{f\pi} \end{cases} \tag{5}$$

To determine the unknown quantity p_0 additional equation is need. The condition of the total tensile force in all bolts remaining can be taken as this additional equation. This condition means that the bolts

pretension decreasing on one side of the clamp is equal to the bolts pretension increasing on another side of the clamp. For a nonlinear contact problem this condition must not be absolutely exact but the analysis shows that this approach gives a reasonable result. If initially z bolts on each side are loaded by the force F_s the condition of total tensile force remaining is the following (Equation (6)):

$$F_{z1} + F_{z2} = 2F_s \cdot z \tag{6}$$

Equations (5) and (6) give the following values of F_{z1} and F_{z2} (Equation (7)):

$$\begin{cases} F_{z1} = \frac{2 \cdot z \cdot F_s}{1 + e^{f\pi}} \\ F_{z2} = \frac{2 \cdot z \cdot F_s}{1 + e^{f\pi}} \cdot e^{f\pi} \end{cases} \tag{7}$$

The second line of the Equation (7) can be used for bolts stress analysis under maximal load conditions. Finally for the total torque taken by the tube/strap contact from Equations (4) and (7) we can get the Equation (8):

$$T_u = \frac{z \cdot F_s \cdot D \cdot (e^{f\pi} - 1)}{e^{f\pi} + 1} \tag{8}$$

For the small friction the Equation (8) turns to simple Equation (9):

$$T_u = f \cdot z \cdot F_s \cdot D \cdot \frac{\pi}{2} \tag{9}$$

In order to calculate the friction capacity of the lower contact between the tube and the bottom part of the clamp we can assume that the contact pressure distribution for this contact is almost uniform. Since the bottom part of the clamp has a complex shape and in any case is relatively stiff this assumption is supposed to be reasonable. The scheme of the contact pressure distribution is presented in Figure 3.
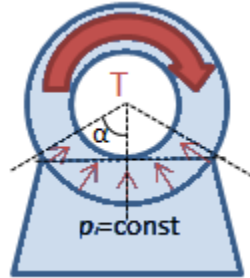

Figure 3. Lower contact scheme

Presuming the angle coordinates of the contact to be from $-\alpha$ to $+\alpha$ (measuring from vertical axis) we can get the expression for the total vertical reaction Q (Equation 10):

$$Q = \frac{D}{2} \cdot \sin(\alpha) \cdot 2 \cdot B \cdot p_l = 2 \cdot F_s \cdot z \tag{10}$$

From the Equation (10) the value of the contact pressure can be obtained (Equation (11)):

$$p_l = \frac{2 \cdot F_{зат} \cdot z}{D \cdot B \cdot \sin(\alpha)} \tag{11}$$

Finally for the total torque taken by the tube/bottom part contact we can get the Equation (12):

$$T_l = \frac{D}{2} \cdot B \cdot f \cdot 2\alpha \cdot \frac{D}{2} \cdot p_l = f \cdot D \cdot z \cdot F_s \cdot \frac{\alpha}{\sin(\alpha)} \tag{12}$$

For $\alpha = \frac{\pi}{2}$ the Equation (12) turns to simple Equation (13):

$$T_l = f \cdot z \cdot F_s \cdot D \cdot \frac{\pi}{2} \tag{13}$$

Combining Equation (8) and Equation (12) the value of total maximal torque taken by support can be obtained (Equation (14)):

$$T = T_u + T_l = z \cdot F_s \cdot D \cdot \left(\frac{e^{f\pi} - 1}{e^{f\pi} + 1} + f \cdot \frac{\alpha}{\sin(\alpha)} \right) \tag{14}$$

The Combination of simplified Equations (9) and (13) gives us the Equation (15).

$$T = f \cdot z \cdot F_s \cdot D \cdot \pi \tag{15}$$

This formula is used in some documents for clamps joints friction capacity calculations (for example Reshetov,1974).
It is found that for the clamps under consideration the Equation (15) is not exact and does not give a conservative result for the total friction capacity estimation and Equation (14) gives results much closer to the numerical results.

The equation (14) was obtained for the bolts pretension + external torque conditions. For the real operating conditions the internal pressure has a considerable influence on the stresses in the clamp and its total friction capacity. The analysis indicates that in case of considerable internal pressure the contact pressure increases and it leads on the one hand to the stresses increasing and on the other hand to the friction capacity raising. It is rather difficult to get the simple analytical formula for this effect but it is possible to get some reasonable approximation using the results of the FEM analysis.

In order to take into account the effect of internal pressure influence the following approach looks preferable: under the internal pressure the straps and bolts take additional load and this change can be expressed by correcting the value of the bolts pretension F_s in Equations (6) - (15). In all equations the F_s must be calculated using Equation (16).

$$F_s = F_{s0} + F_{sP} \tag{16}$$

In the Equation (15) F_{s0} is the assembly bolt pretension and F_{sP} is the additional force due to internal pressure in the tube.
The simplest way to define the F_{sP} value is to consider the straps tension under internal pressure load separately from other loads (see Figure 4)
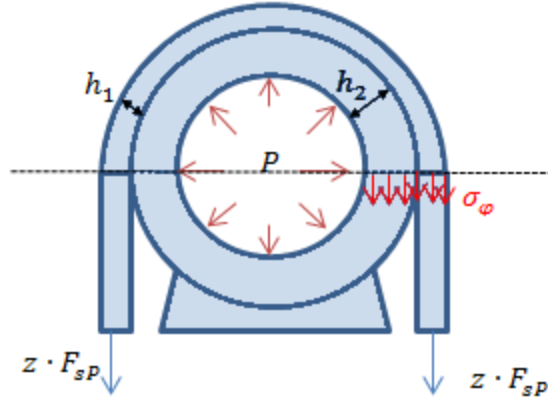
Figure 4. The influence of the internal pressure

Let us suppose that h_1 is the strap thickness, h_2 – tube thickness.

Calculating the balance of the vertical load for the top half of the structure, cut by horizontal plane passing through tube axis (see Figure 4) and neglecting the influence of the parts of the tube outside of the contact area, we will get Equation (17) for the average stress in this section:

$$\sigma_\varphi = \frac{P \cdot D \cdot B}{2 \cdot (h_1 + h_2) \cdot B} = \frac{P \cdot D}{2 \cdot (h_1 + h_2)} \tag{17}$$

Assuming the stress to be uniformly distributed through the considered section we can get the expression for the F_{sP} value (Equation (18)):

$$F_{SP} = \frac{\sigma_\varphi \cdot B \cdot h_1}{z} = \frac{P \cdot D \cdot B \cdot h_1}{2 \cdot (h_1 + h_2) \cdot z} \tag{18}$$

Numerical tests show that Equation (18) gives overestimated F_{sP} value.

To achieve a good agreement with numerical tests it is necessary to introduce an additional correction factor γ which takes into account the nonuniform stress distribution through the considered section (see Equation (19)).

$$F_{SP} = \gamma \frac{P \cdot D \cdot B \cdot h_1}{2 \cdot (h_1 + h_2) \cdot z} \tag{19}$$

The correcting factor γ depends on the flexibilities of the bolts and the lower part of the support. The reasonable agreement for real structures with numerical tests gives the value $\gamma = 0.5$.

Usually the maximum required external force F_D and force arm R_D is specified. In this case the relation between the external force F_D and the torque T is the following (Equation (20)):

$$T = F_D \cdot R_D \tag{20}$$

The formulas for the bolt loads and stresses in the straps estimations can be used for the preliminary hand-calculations and the main dimensions of the bolts and clamp choosing. The sequence of hand calculation can be the following:

1. Based on maximal external load F_D and specified arm of a force R_D the maximum T is calculated (Equation (20)).
2. Initial reasonable values of the strap width B and the number of bolts on each side z are specified.

3. Based on pipe diameter D, specified seat angle coverage α, specified number of bolts z the required bolt pretension F_s is calculated (based on Equation (14)).
4. The maximum force on the bolts F_{z2} is calculated using Equation (7), the maximum force on the one bolt under the torque is obviously equal to $F_{zb}=F_{z2}/z$.
5. Based on allowable stresses for bolt material and maximum bolts forces F_s, F_{zb} the bolt diameter is chosen.
6. Based on the maximal forces in the strap $z \cdot F_s$, F_{z2}, the specified strap width B and the allowable stresses for strap material the strap thickness h_1 is chosen.
7. If the internal pressure is taken into account the necessary bolts assembly pretension F_{s0} is defined based on bolt pretension under operating conditions F_s, internal pressure P, straps and tube thicknesses h_1, h_2, specified straps width B and number of bolts z using Equations (16),(19).
8. The values of B and z can be varied if necessary and then p. 3-7 repeated.
9. Other dimensions (necessary ribs width and thicknesses, flanges thickness, etc.) can be preliminary estimated using obtained values of forces in the bolts. The formulas depend on the specific clamp design.

COMPARISON WITH THE RESULTS OF NUMERICAL ANALYSIS

The Finite element analysis was performed for several different clamps. The main results for the clamp shown in Figure 5 are presented in this paper. All calculations were performed in ANSYS 15.0.
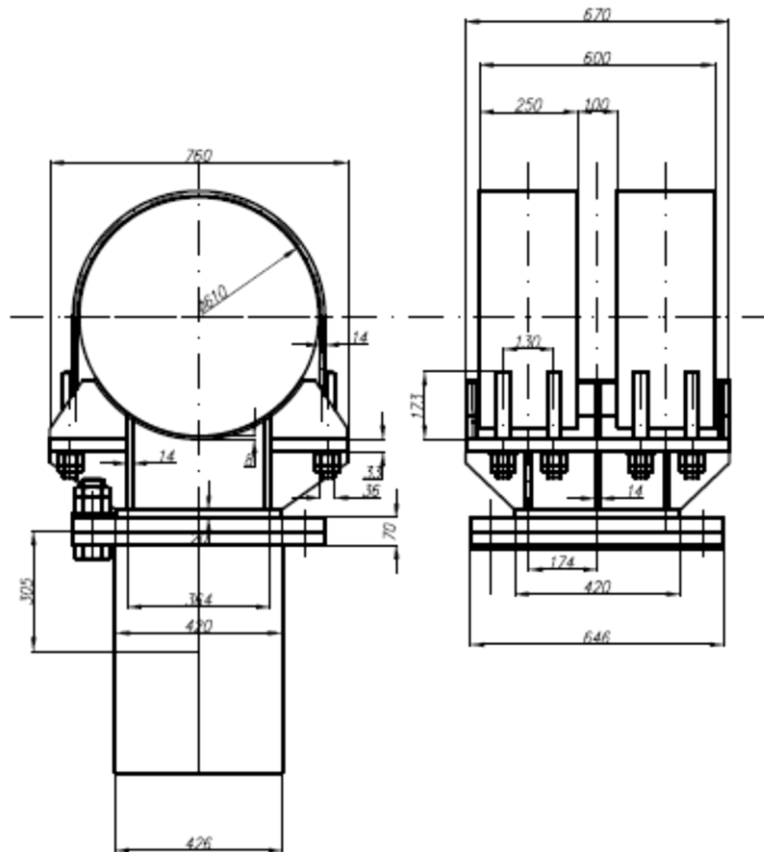

Figure 5. Design of the Clamp for the tube D=Ø610

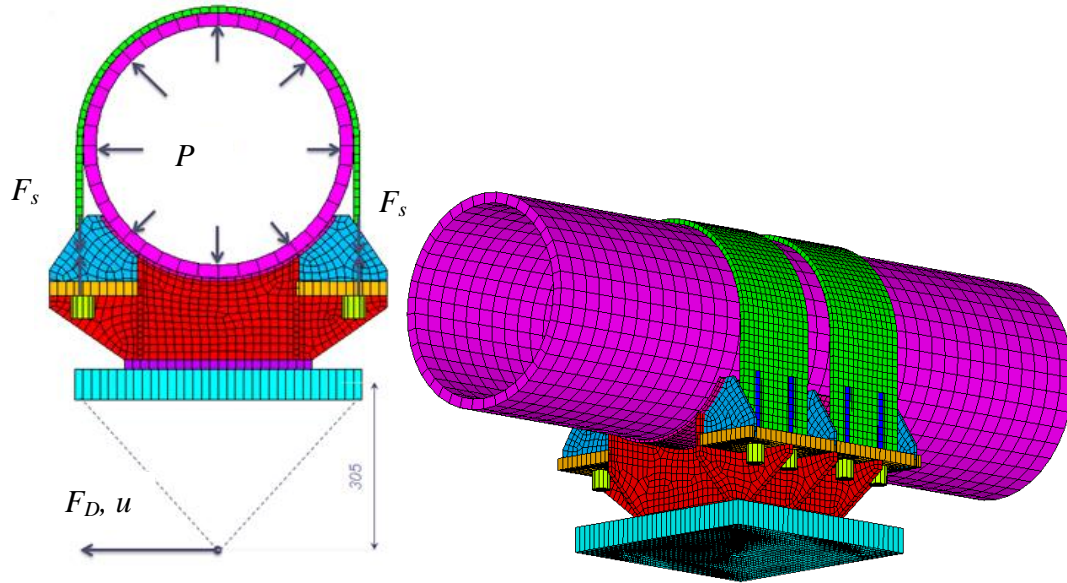The Finite Element model for this part is presented in Figure 6.

Figure 6. FEM model

The shell elements were used for modelling all parts except bolts. The bolts were modelled using beam elements. On the first load step all the bolts were loaded by assembly pretension. On the second load step internal pressure was applied. On the third load step the external force was applied. When friction capacity was defined in order to improve the solution convergence the horizontal displacement instead of external force was applied. The external force F_D was applied on the arm R_D=833 mm.

The results of total friction capacity calculations based Equation (14) and their comparison with numerical results are presented in Figure 7.
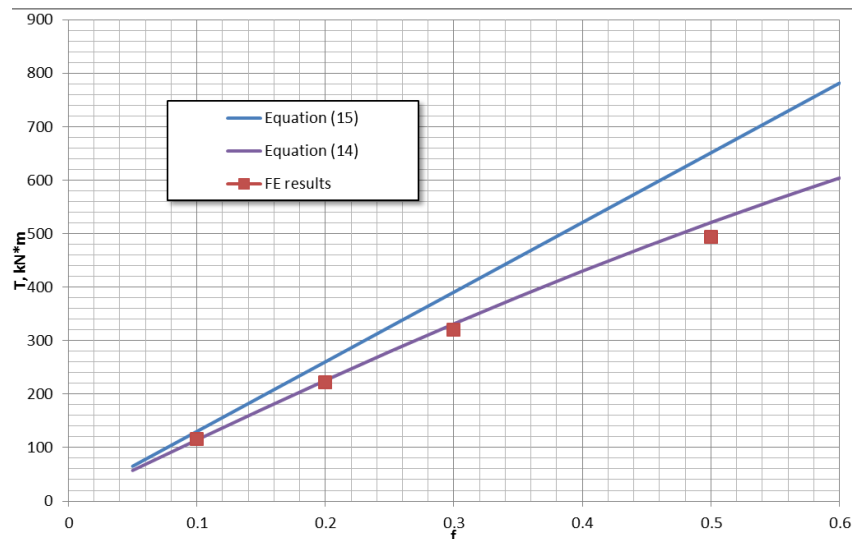

Figure 7. Total friction capacity calculations (Clamp for the tube D=Ø610)

The presented results are obtained for the following dimensions:

α=59°; z=4 (for two straps); F_s=170 kN; D=610 mm; Friction factor: f=0.3; Internal pressure: P=8.63 MPa; h_1=14 mm; h_2=30 mm; B=500 mm (sum of widths for two straps); γ=0.5 – correction factor.

Figure 7 shows that Equation (14) gives reasonable and conservative results, while a simplified equation (15) does not match numerical results and gives a nonconservative prediction. A typical view of relation between the external force F_D and the support displacement u (see Figure 6) under this force in the point of load applying is shown in Figure 8.
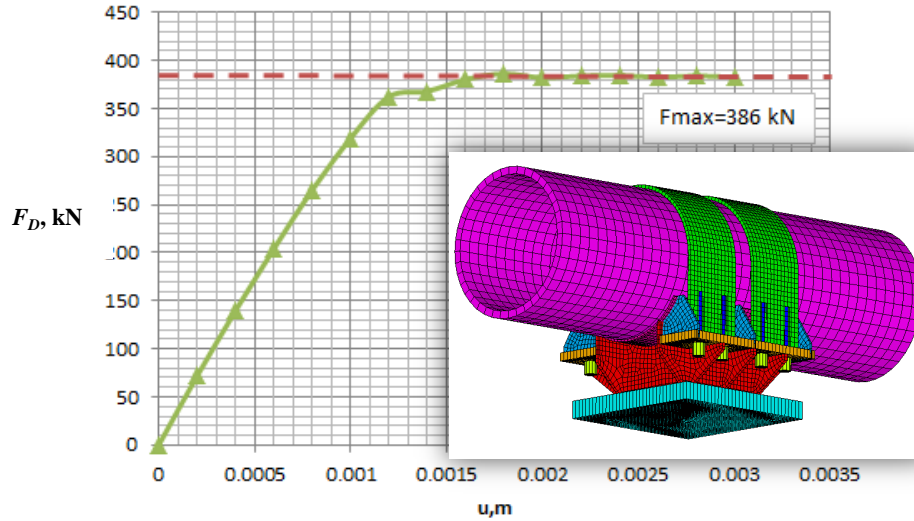

Figure 8. Friction capacity numerical measuring (f=0.3, Clamp for the tube D=Ø610)

Table 1 contains the comparison of the friction capacity values calculated for tube with internal pressure and various straps and tube thicknesses. These numerical tests allow checking the influence of internal pressure accounting.

Table 1: Friction capacity for the tube under internal pressure load

h_1,mm	h_2,mm	f	T, kN*m, Equations (14),(19)	T, kN*m, FEM results	Error, %
14	30	0.3	434	437	0.7
10	30	0.3	412	421	2.1
30	30	0.3	492	479	2.7
14	20	0.3	464	471	1.6
14	50	0.3	402	402	0.1

As it is shown in Table 1 the results based on Equations (14), (19) for considered clamp design match numerical results with relative error less than 3%.

The proposed method of the total friction capacity definition was verified for several different sized clamps. In Figure 9 the force-displacement curve for the clamp with tube diameter D=Ø140 mm is presented.
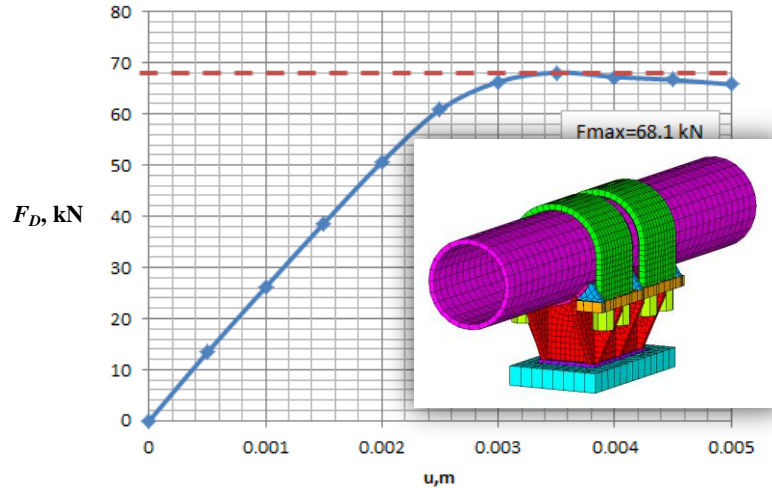
Figure 9. Friction capacity numerical measuring (f=0.3, Clamp for the tube D=Ø140 mm)

The main dimensions of this clamp are: D=140mm, α=60°; z=4 (for two straps); F_s=41.8 кН; Friction factor: f=0.3; Internal pressure: P=8.63 MPa; h_1=10 mm; h_2=8 mm; B=200 mm

The equations (14), (19) give the value of the maximal external force F=68.0 kN; The finite element analysis gives almost the same value F=68.1 kN.

The finite element analysis shows that the real stress distribution in the clamp is very complex and it is difficult to predict analytically the peak stresses in all the parts under operation conditions, but nevertheless the proposed method of the clamps preliminary analysis is fit for some average stresses calculation, preliminary dimensions and bolts pretension choosing and provides good estimation of the total friction capacity. Additional numerical tests show that the proposed method gives reasonable results even for clamps with more complex than "classic" design (shown in Figure 1). For different clamp designs the total friction capacity values obtained from Equation (14) are also close to the values obtained from numerical tests (relative error is less than 10% in all performed numerical tests).

CONCLUSION

A friction clamp is a reliable support for viscous dampers installation and eliminates piping attachement welding. The simplified design by rule approach proposed in the paper is quite appropriate based on comparison with sophisticated FEM analysis.

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