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END CONNECTIONS OF TUBULAR MEMBERS BY CIRCULAR PLATE AND CENTRAL FLANGE

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The connection of tubular members, used as bridge hangers or in 3-D coverings, has been widely researched, both for resistance and for fatigue. Such members are also often used in scaffoldings or as lighter bracings. The connection of the tube to a single bolted or welded plate consists of a circular end plate, covering the tube's cross section on which a perpendicular rectangular plate is welded. No analytical approximation is available to calculate stresses in such a circular plate. Hence, FE-models have to be used for each application. As this is time consuming, a systematic survey of normal and shear stress has been conducted, for 6 unity connection forces of the gusset plate. The latter are normal force, shear in 2 directions, torsion and bending moments in 2 directions. The 3 types of stresses are kept separate, to allow for linear superposition, simulating any kind of real situation. The tube diameter and plate thickness have been varied to allow regression and obtain closed formulas. It appears that larger tube diameters have the capacity to distribute force, albeit the stress should be larger due to the increase of diameter. The former effect seems to overrun the latter and stress conditions are lower for wider tubes, which is rather surprising and may prove to be useful in practice. This is also confirmed by the displacement patterns, which show larger effective width of the higher diameter circular plate.

Keywords: Biaxial stress, Circular steel tube, Connection gusset plate, Parametric analysis, Steel scaffoldings.

1 INTRODUCTION

Connecting a circular section hollow core tube to other members requires sudden or gradual modification of the cross-section. This may be to a single plate, a smaller tube, a rectangular or H-cross section. From these, the former is probably the most drastic modification. The transition of geometry also introduces stress concentrations, which are heavily aggressive for fatigue resistance.

In view of this, various types of connections have been developed and thoroughly researched. For bridge hangers Agerskov and Bjornbak-Hansen (1979) made extensive fatigue tests on welded connections in round steel bars. They have developed an elaborate and fatigue resistant method of inserting a single plate into a slot of the tube. An application for hanger tubes can be seen in Figure 1. Hoang *et al*. (2013) have tested a simpler connection of tubes, by a circular flange plate and circumferential bolts. They conclude that EN 1993-1-9 (2004) does not entirely account for the geometrical parameters of the connection. This implies that the hot spot stress method would be more adapted to deal with this type of problems.

Choi and Najm (2020) have investigated the fatigue resistance of fillet-welded connection of tubes for sign support structures. They have provided fatigue life data of the connection details, damage and safety factors for this type of connection. The strength of tube connection by lateral gusset plates has been researched by Kim (2001). The numerical simulation and test program have resulted in proposing formulas to assess strength taking into account the eccentricity of normal force of the tubular bracing bars.

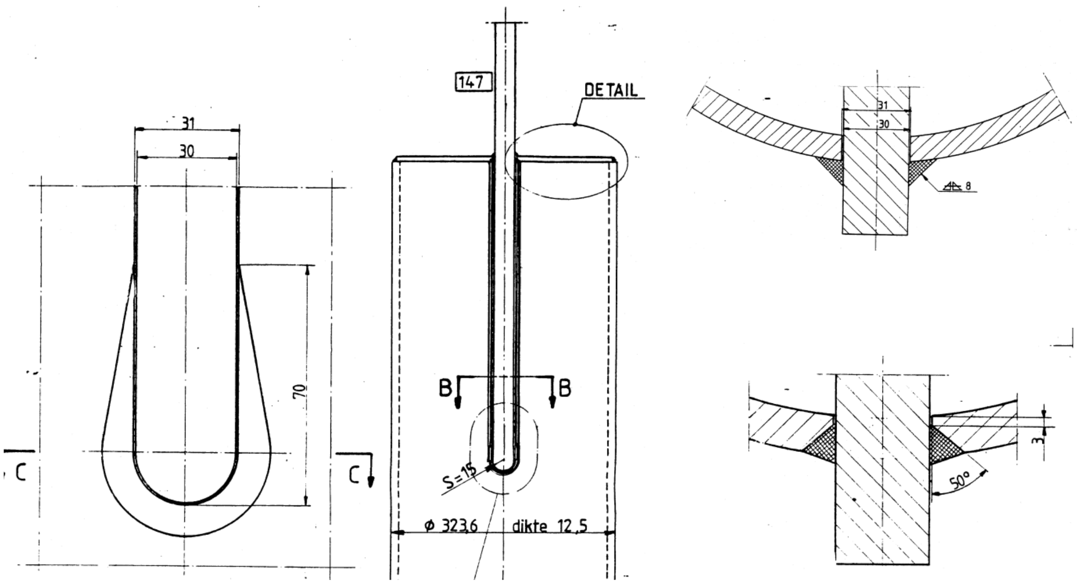


Figure 1. Fatigue-resistant connection of gusset plate welded is slotted opening of tube.

The present research concentrates on tubular structures for temporary use, fatigue not being the main issue for the connection. In addition, the type of connection is intended not to introduce eccentricity and additional bending of the tube wall and aims to include all types of loads. Figure 2 shows a scaffolding, including such connection, as well as the principle. At the end of a tubular member, a closing circular plate is welded. On top of this, a perpendicular gusset plate is welded, allowing connection by bolts to other parts.

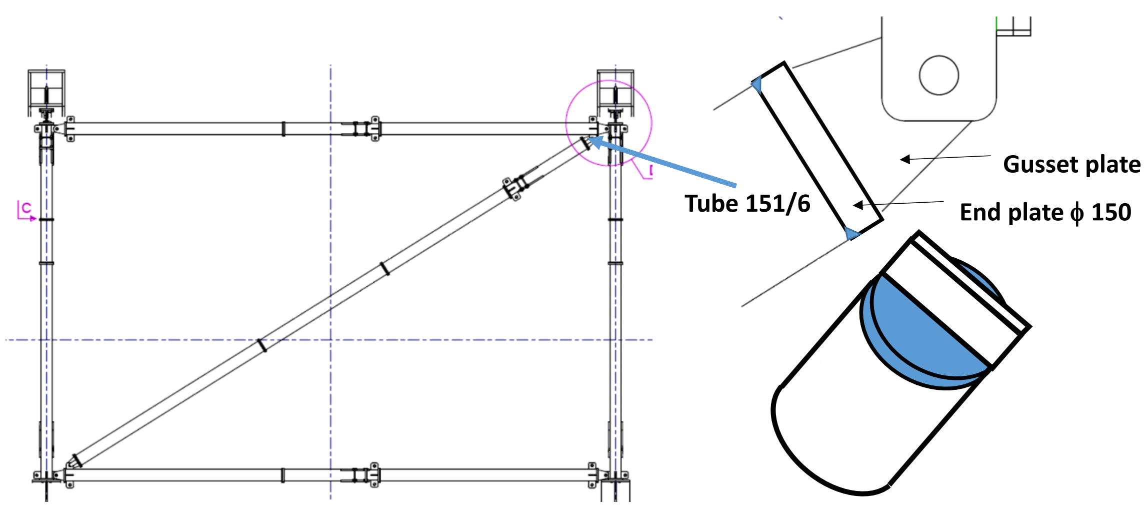


Figure 2. Scaffolding with node detail of circular disk and perpendicular gusset plate.

The stress in the gusset plate and in the tubes can easily be calculated through classical formulas. However, after survey in literature, especially highly developed work as Flügge (1973), no analytical formulas are available to calculate the stresses in the circular plate. In more recent research. Hence, FE-simulations seem to be necessary.

2 BOUNDARY AND LOAD CONDITIONS

As the end plate is the main object of this research, the model needs to consist of the circular plate and the perpendicular gusset plate. Loads will be applied at the gusset plate edge rib. Hence the circular plate needs boundary conditions, which correspond to the connected tube. These tubes are rather stiff in the circumferential direction, but may be flexible in the perpendicular direction. The latter is due to the tube thickness, which is considerably lower than for the end plate. Hence, the assumption is to simulate this connection by hinged supports.

All loads are applied at the edge rib of the gusset plate. The 6 types of loads, normal force, shear in both perpendicular directions, torsion and bending around 2 axes are considered. These are shown in Figure 3. Clearly, all forces and moments are uniformly distributed

Obviously, some of these individual load cases are not due to axial loads in the tubes. For instance, torsion may occur, due to load eccentricity, whereas some parts of the bending moments can be due to wind load.

3 PLATE STRESS DUE TO INDIVIDUAL LOAD CASES

After sufficient mesh refinement, the 3 types of skin stresses, σ_x, σ_y, τ_{xy}, for the various load cases can be found from FE-simulation, through the use of the software Samcef-Siemens. Stress concentrations are located in typical areas, corresponding to the nature of the loads. For the circular plate, Figure 4 shows some of the typical areas, which are not necessarily identical for all cases, although they show large similarity.

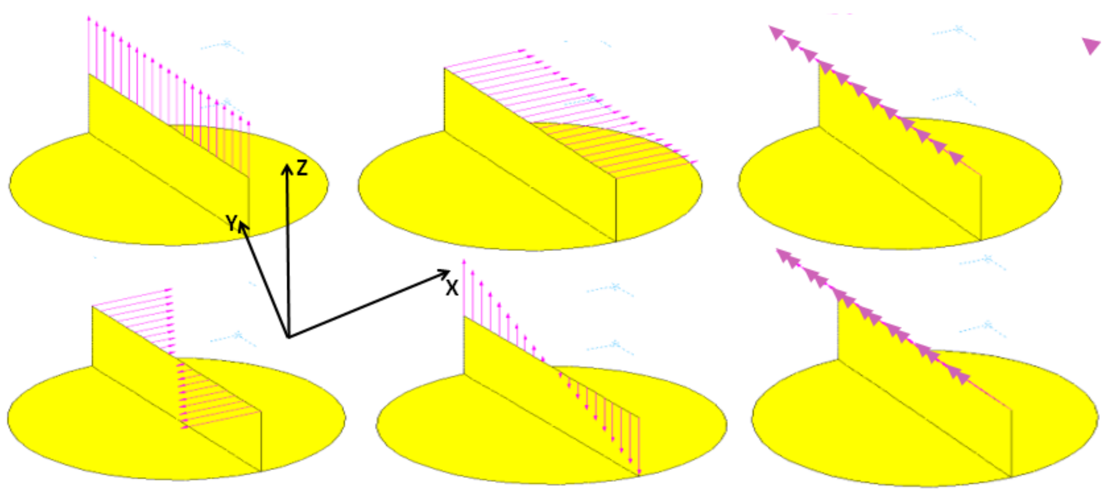
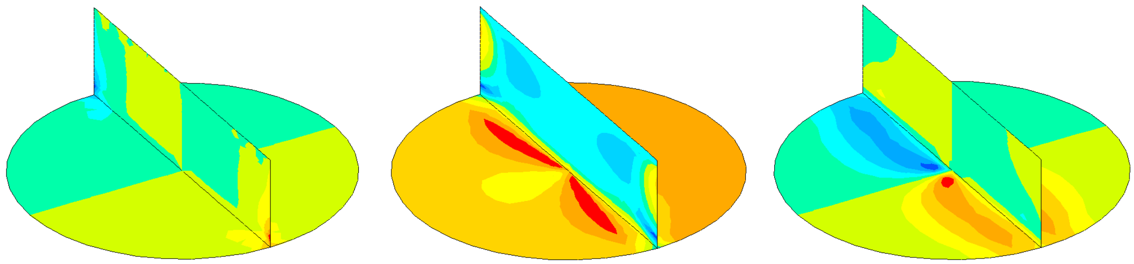


Figure 3. Six considered loading cases.

The loads which act in the plane of the gusset plate, as the axial force F_z, in plane shear F_y and in-plane bending moment M_x generate normal stress at the tips of the connection of tube, gusset and circular plate, as shown in Figure 4a. This is due to the discontinuity between open air and the intersection of various plates and occurs at any similar geometry. However, concentration of shear stress occurs at the same location for normal force F_z and parallel bending M_x only. Parallel in plane shear force F_y renders a rather distributed shear stress pattern, since the latter requires to build up from the free edge towards the center, as shown in Figure 4b. A similar pattern as in Figure 4b is found for the normal stress due to perpendicular shear F_x and due to parallel bending M_x.



(a) (b) (c)

Figure 4. Stress distributions in circular plate.

Finally, concentration of shear stress occurs near the circular plate's center for perpendicular shear F_x, for torsion M_z and for perpendicular bending M_y. The latter is due to building up of shear from both the two free edges towards the center. Thus, the stress distributions can easily be explained, as they correspond to physical insight. Indeed, a direct normal force is diverted towards the stiffer area of the gusset plate, while bending of the latter renders large deformation at the plate's center and shear also concentrates near that center. The deformations of the circular plate are not shown, although they confirm the preceding comments, as the applied forces F_x, F_y, F_z, and M_x, M_x and M_z introduce vertical deformations or curvature of respectively 0.043, 0.040, 0.016, 0.032, 0.064 and 0.177 mm, corresponding to the locations of Figure 4.

The aim of this study is to provide a tool for combining various load cases and assessing safety through the unity-check, based on the equivalent (von mises-) stress. However, by nature linear superposition of the latter type of stress is impossible and thus the individual stresses σ_x, σ_y, τ_{xy}, are to be used. The maximum values of these quantities do not occur at the same location. Hence combination of the maximum values leads to overestimating the actual stress state. In the following, this overestimating is accepted, as it is mostly shear stress which appears at a different location and this type of stress has a considerably lower value.

4 PARAMETRIC ANALYSIS

Attempting to generalize the results of plate stresses, a parametric variation was carried out. Since force and bending moments are reduced to unit quantities, the main influence parameters are the tube diameter D and the circular plate thickness t. Both have been considered separately. Regarding the relation of stress with the diameter D, the diagram of Figure 5 shows the values of σ_x, σ_y, and τ_{xy} divided by the normal force F_x, which is causing these stresses. The diagram is concerned with absolute values of maximum quantities.

Although a second-degree parabola is also close, the best fitted regression line seems to be a power function, which corresponds closely to the individual values. For τ_{xy} this correspondence is weaker, since the shear stress varies only mildly. A striking result is that the stress intensity decreases with the tube diameter. At first sight, it may be expected stress is increasing with the diameter, since the span of the reversed T-beam consisting of the circular and gusset plate becomes larger.

However, the latter effect is counteracted by the fact that a larger diameter also implies a larger effective width of the horizontal flange of a T-beam. Hence, compression or tensile forces in the circular plate can disperse over a larger portion of the plate, thus causing decrease of the normal stress. The latter effect seems to overrun the former one and this applies to each of the various external forces. In addition, for those external forces, causing stress concentrations at the gusset plate tip, do not experience any effect of the larger diameter.

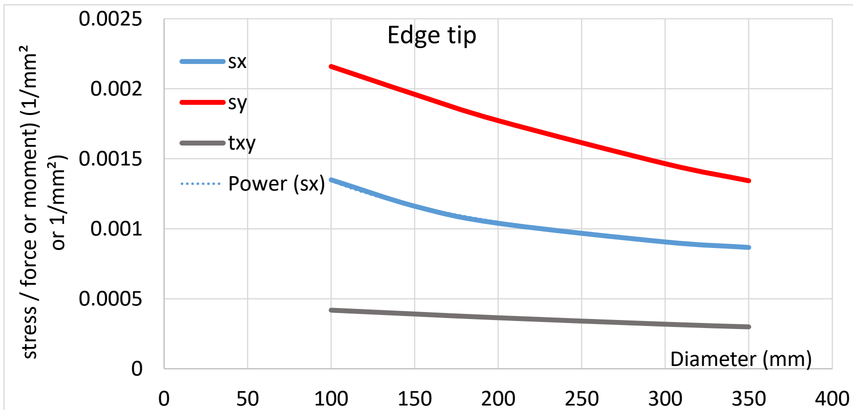


Figure 5. Stress as a function of the tube diameter.

Regression curves for the parameter t are more difficult to grasp in typical functions and are not shown here because of space shortage. The power series is also the most adequate one for this relation. Stress intensities are decreasing with larger thickness, as expected.

In the graph from Figure 5, stress is divided by either unit force or by unit bending or torsion moment. Hence, it is expressed in 1/mm², which is not without dimension. To express these quantities in a dimensionless manner, the following transformation into Eq. (1) and (2) were made.

$$\frac{\sigma_x D t}{F_z}, \frac{\sigma_y D t}{F_z}, \frac{\tau_{xy} D t}{F_z}, \frac{\sigma_x D t}{F_y}, \frac{\sigma_y D t}{F_y}, \frac{\tau_{xy} D t}{F_y}, \frac{\sigma_x D t}{F_z}, \frac{\sigma_y D t}{F_z}, \frac{\tau_{xy} D t}{F_z} \tag{1}$$

$$\frac{\sigma_x D t^2}{M_x}, \frac{\sigma_y D t^2}{M_x}, \frac{\tau_{xy} D t^2}{M_x}, \frac{\sigma_x D t^2}{M_y}, \frac{\sigma_y D t^2}{M_y}, \frac{\tau_{xy} D t^2}{M_y}, \frac{\sigma_x D t^2}{M_z}, \frac{\sigma_y D t^2}{M_z}, \frac{\tau_{xy} D t^2}{M_z} \tag{2}$$

$$\frac{\sigma D t}{F} = C t^{\alpha} D^{\beta}, \quad \frac{\sigma D t^2}{M} = C t^{\alpha} D^{\beta} \tag{3}$$

From the various power regressions, the power values of α and β can be derived, which are defined by the general expression (3). The values of C are the constants that may allow to establish a reliable assessment of the circular plate's stresses. Thus, there are different values of C for each type of force or moment. The values of α and β can be taken from Table 1.

Table 1. Power exponents for t and D.

α	σ_x	σ_y	τ_{xy}		σ_x	σ_y	τ_{xy}
F_z	0.108	-0.069	-1.205	F_x	-0.628	-0.344	-2.069
F_y	0.395	0.468	0.253	M_z	-0.351	1.892	0.583
M_x	1.129	1.159	1.236	M_y	1.107	0.355	0.318
β	σ_x	σ_y	τ_{xy}		σ_x	σ_y	τ_{xy}
F_z	0.746	0.617	0.729	F_x	0.210	-0.078	0.149
F_y	-0.380	-0.314	-0.301	M_z	-0.501	-0.592	-0.275
M_x	-0.358	-0.445	-0.605	M_y	-0.069	0.257	-0.172

Since the stresses have been multiplied either by D* t or by D * t², they no longer decrease all with the diameter D, nor with the plate thickness t. In fact, some values are increasing with D, the order being reversed. Concerning the constants C from equations (3), they have been summarized in Table 2; It should be mentioned this table gives 95% fractional values, since the quantities have been averaged for all considered values of D, ranging from 100 to 350 mm and t, varied from 10 to 25 mm.

Table 2. C-constants.

α	σ_x	σ_y	τ_{xy}		σ_x	σ_y	τ_{xy}
F_z	0.050	0.247	0.606	F_x	18.01	23.40	277.1
F_y	4.315	5.461	1.732	M_z	7554	30.78	51.09
M_x	17.21	54.00	23.47	M_y	12.91	10.50	59.13

The C-values are sufficiently reliable, the standard deviation being 5.8 % on average, except for M_z or torsion and M_y bending in plane of the gusset plate with standard deviation of 46.1 %, where the dispersal is relatively high. The reason for this has to be investigated further.

A similar result is obtained if the values from the numerical simulations are compared to the outcome of the formulas (3), the ratio being on average 0.937 for F_x, F_y, F_z, and M_z and 0.254 for both M_x and M_y. Hence, the formulas are overestimating the stresses from both last moments.

5 CONCLUSIONS

A comprehensive set of formulas, including constants and exponential values for the parameters plate diameter and thickness have been derived from a series of numerical simulations. This set of formulas allows determining the stresses in the circular plate that is used to connect a gusset plate to a circular plate thus transmitting forces and moments between both structural elements.

For most types of external forces, the derived equations are reliable, with small standard deviations of the necessary constant factor relating the dimensionless force or moment to both the parameters. This does not apply to stresses due to torsion and bending in the plane of the gusset plate, which show large deviations. The latter may be due to the need for refinement of the proposed formulas and will be researched in future.

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