# Design of Tension Circular Flange Joints in Tubular Structures

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## INTRODUCTION

As the use of circular steel tubes in different structures has increased dramatically in recent years the connections between tubes have become very important. One of the most important and common methods of making connections, especially field connections, in tubular structures is with bolted circular flanges. The design of such a joint under an overall axial tension force requires the determination of flange dimensions and the number and arrangement of bolts.

A typical circular flange joint is shown in Figure 1. Two flanges are welded to the tubes and fastened by a number of bolts and these bolts may be preloaded. $T$ is the tensile force uniformly acting around the tubes. Under the action of $T$, the flanges will bend and parts of the flanges may separate while other parts remain in contact. Due to the existence of contact force (prying force) between the two flanges, the total bolt force is higher than the applied tension force. These factors make the prediction of total bolt force in a joint and the behavior of the joint more complex.

Because the joint in Figure 1 is symmetrical about the middle surface between two flanges, only half of the joint actually needs to be modelled in an analysis. The half joint is very similar to the case when a circular column is connected to a circular base plate under uplift loading, which may be the governing load case for the base plate in single-storey buildings in some countries such as Australia.[1] Hence the design method for circular bolted flange joints in tubular structures is also applicable to circular base plates connected to circular columns under uplift loading.

Research work on bolted circular flange joints in tubular structures has been carried out for many years and different methods for the design of the joints have been developed. From an investigation by Cao[2] (1995) it has been found that the methods currently used are either empirical rules or based on some assumptions which are not applicable to all joints. Further research work on circular flange joints, which includes finite element analyses, mathematical analyses and tests, has been carried out and a better understanding of the behavior of these joints has been obtained. In this paper different methods currently used are compared and the deficiencies and shortcomings of the methods are analyzed. A new design method based on this research work is proposed and design charts, which are in accordance with the AISC LRFD specification for structural steel buildings[3] (1993), are created to simplify the design procedure. It will be seen that the method is accurate, reliable and easy to use.

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## 1. NOTATION

| | |
|---|---|
| $A$ | Cross-sectional area of tube. |
| $A_b$ | Total nominal body area of bolts in a joint. |
| $d$ | Diameter of bolt. |
| $D$ | Diameter of tube. |
| $D_p$ | Diameter of bolt pitch circle. |
| $D_f$ | Diameter of flange outside circle. |
| $F_t$ | Nominal tensile strength of bolt. |
| $F_{yf}$ | Yield stress of flange. |
| $F_{yt}$ | Yield stress of tube. |
| $M$ | Interactive bending moment between tube and flange per unit length. |
| $M_{max}$ | Maximum moment in a flange per unit length. |
| $M_p$ | Plastic moment of flange plate per unit length. |
| $M_{rj}$ | Bending moment at tube/flange junction in a flange per unit length. |
| $M_{rp}$ | Bending moment at bolt pitch circle in a flange per unit length. |

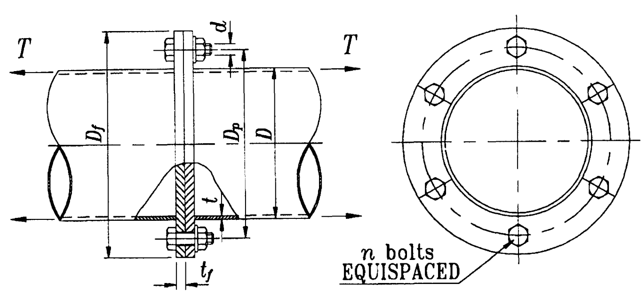



*Fig. 1. A typical circular flange joint.*

| | |
|---|---|
| $M_y$ | Yield moment of flange plate per unit length. |
| $n$ | Number of bolts in a joint. |
| $P$ | Total prying force in a joint. |
| $r$ | Radius of flange. |
| $R$ | Function of dimension and material property ratios used for calculating $M_{max}$. |
| $S$ | Function of dimension and material property ratios used for calculating $t_f / t$. |
| $t$ | Thickness of tube. |
| $t_f$ | Thickness of flange. |
| $T$ | Calculated value of total tensile force in a joint. |
| $T_b$ | Total bolt force in a joint. |
| $T_{b0}$ | Total bolt preload in a joint. |
| $T_r$ | Total bolt tension capacity in a joint. |
| $V$ | Interactive radial force between tube and flange. |
| $\phi$ | Resistance factor of tube material. |
| $\mu$ | Slope of bolt force vs. tension force line. |
| $\gamma$ | Coefficient taking bolt preload into account in calculation of $T_b$. |
| $\nu$ | Poisson's ratio. |
| $\omega_0$, $\omega_1$, $\omega_2$, $\omega_3$, $\omega_4$ | Coefficients for calculation of $\mu$. |

## 2. COMPARISON OF THE METHODS CURRENTLY USED

Based on the test results for a large number of circular flange joints some empirical rules for the design of these joints were proposed by Rockey and Griffiths[4] (1970) and other researchers[5,6] (1974, 1975) in the UK and these rules have been incorporated in British Standard BS 8100[7,8] (1988, 1986) for the design of towers and masts. In the Stelco design manual[9] (1981) for hollow structural section connections, a method for the determination of minimum flange thickness for a joint was provided to ensure that the flange designed had sufficient stiffness to resist prying action. This method was included in an early design guide for hollow steel sections by CIDECT[10] (Comité International pour le Développement et l'Etude de la Construction Tubulaire) (1984). Tests and theoretical analysis on circular flange joints have been carried out by Igarashi *et al.*[11,12] (1985, 1987) in Japan and a method for the determination of flange thickness and number of bolts in a joint has been proposed. This method has been adopted in a late design guide for circular hollow section (CHS) joints by CIDECT[13] (1991) and was also incorporated into a recent book by Packer and Henderson[14] (1992).

From research work by Cao[2] and others, it has been found that the bolt pitch circle in a joint should be as small as possible and the flange outside circle is usually decided by the arrangement of bolts. Hence the main task for the design of a circular flange joint is to determine the thickness of flange and the total bolt tension capacity required, which is used to decide the diameter, number and grade of bolts, for the joint. A summary of the methods used by BS 8100, Stelco and Igarashi *et al.* is presented in Table 1.

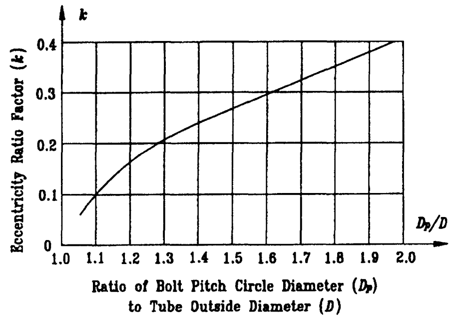
In order to compare the three different methods presented in Table 1, they are used to determine the flange thicknesses and the minimum numbers of bolts for the four joints listed in Table 2. The flange bolt pitch circle diameters, flange outside diameters and bolt diameters have been decided. The yield stress for all the flanges is 36 ksi (A36), the yield stress for all the tubes is 50 ksi (A50) and A325 bolts are used for the joints. Three different levels of tension forces ($T_1$, $T_2$ and $T_3$ in Table 2) are applied to each of the joints, which correspond to 90 percent, 75 percent and 60 percent of the squash load (yield stress multiplied by cross-sectional area) of each of the tubes. The design results, in terms of flange thickness and number of bolts required, are presented in Table 3.

From Table 3 it can be seen that the results from the three methods are quite different. The flange thicknesses from BS 8100 are much lower than those from Stelco. From Cao & Bell's analysis[15] (1996) it has been found that the total bolt force in a joint may be much higher than 120 percent of the tension force applied to the joint. Hence the flange thickness and bolt capacity determined by BS 8100 for a joint are small and a joint designed based on BS 8100 may not be safe, especially for the cases with high tension forces. Because Stelco's method attempts to reduce prying force in a joint to zero, the flange designed by this method seems too thick. Although thicker flanges for a joint can reduce prying force, the prying force may not be eliminated totally so that the bolt capacity may not be enough if no prying force is considered. The method proposed by Igarashi *et al.* is based on the analysis of plastic failures of flanges obtained using yield line theory, but the yield line models were not proven by the tests conducted by Igarashi *et al.* In some cases flanges may need to be designed using elastic analysis, as recommended in EC 3[16] (clause 6.1.3). In addition, the bending moment between tube and flange as well as the bolt stiffness are neglected in Igarashi's analysis. Hence this method still cannot be used confidently in all situations.

## 3. FURTHER RESEARCH WORK

Cao and Bell[17] (1994) have carried out finite element analyses using the package ABAQUS[18] (1994) for circular flange joints. Axisymmetric and 3-D models were used to analyze a number of joints with different dimensions and bolt arrangements. Bolt forces, contact forces between flanges, displacements and stresses in the tubes and the flanges in these joints were obtained from the analyses. The influences of bolt preload, tension force and joint geometry on joint behavior were investigated.

From the analyses it has been found that the discrete bolt arrangement in a joint can be represented by a uniformly annular arrangement without significant error. For most joints analyzed with relatively high tension forces, only the edges of the two flanges in a joint were in contact and this contact

| Table 1. Summary of Design Methods | | |
|---|---|---|
| **Method** | **Details** | |
| BS 8100<br>Refs. 4–8 | $t_f \ge 2t \cdot \frac{F_{yt}}{F_{yf}}$ | $T_r \ge 1.2T$ |
| Stelco | $t_f \ge \sqrt{k\frac{T}{F_{yf}}}$ |   |
| Refs. 9–10 | $T_r \ge T$ | |
| Igarashi<br>*et al.* | $t_f \ge \sqrt{\frac{2T}{\pi F_{yf} f_3}}$ | $T_r \ge \left[1 - \frac{1}{f_3} + \frac{1}{f_3 \ln(D_f / D_p)}\right] T$ |
| Refs. 11–14 | $f_3 = \frac{1}{2k_1}\left[k_3 + \sqrt{k_3^2 - 4k_1}\right]$ | $k_1 = \ln\frac{D_p}{D-t}$ $k_3 = 2 + k_1$ |

(prying) force was about 30 percent to 70 percent of the applied tension force. For each joint analyzed the maximum bending stress in the tubes was much higher than the maximum stress in the flanges. As the applied tension force was increased the tubes at the junctions with the flanges were found to yield first. The failure of a practical joint, in which the flange thickness is considerably greater than the tube wall thickness, was usually caused by excessive plastic deformations in the tubes and when a joint failed the flanges in the joint either remained elastic or had just started to yield.

Based on the results from finite element analyses a model for mathematical analysis for this type of joints has been created by Cao and Bell[15] (1996). In this model total bolt force and prying force are modelled as uniformly annular forces around the bolt pitch circle and the edge of the flange. From the analysis of this model, formulas for the calculation of total bolt force in a joint have been developed. Detailed analysis about the prying action in circular flange joints has also been done by Cao and Bell.[19] The determination of total bolt force in a joint is essential for the design of flanges and bolts for the joint.

Further tests have been undertaken by Cao and Bell[20] (1996) to confirm the results from finite element analyses and mathematical analysis. In these tests static tension forces, bolt forces, displacements and stresses have been measured and the influences of bolt preloads and flange dimensions on the behavior of flange joints have been investigated. Good agreement between the test results and the results from finite element analyses and mathematical analysis has been found.

## 4. DESIGN METHOD FOR CIRCULAR FLANGE JOINTS

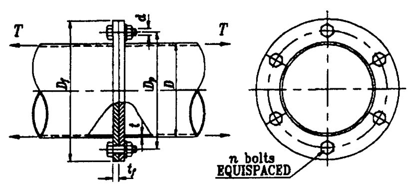
From the further research work[15,17,19,20] described above the behavior of circular flange joints has been investigated thoroughly. Although it is not practical to apply the formulas derived in the work to regular designs directly, the formulas really provide a basis for the formation of a rational design method.

As explained before, the main task for the design of a circular flange joint is to determine the thickness of flange and the diameter, number and grade of bolts. Other flange dimensions can be decided empirically first and then the flange thickness can be determined based on the assessment of maximum flange bending moment and the bolts can be determined based on the assessment of total bolt force.

### 4.1 Determination of $d$, $D_p$ and $D_f$

Bolt diameter $d$ should be decided first. Flange bolt pitch circle diameter $D_p$ should be kept as small as possible but the clearance between the nut and the weld should be not less than 0.25 inches. The distance between flange outside radius and bolt pitch circle radius can be taken as the same as the distance between bolt pitch circle and tube surface, as suggested by Packer and Henderson,[14] i.e. $D_f - D_p = D_p - D$. Cao[2] has shown

| **Table 2. Information for Joints** | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Joint No. | Tube (inch) | | Flange (inch) | | Bolt (inch) | Tension Forces (kips) | | |
| | $D$ | $t$ | $D_f$ | $D_p$ | $D_0$ | $T_1$ | $T_2$ | $T_3$ |
| J-1 | 10 | 0.375 | 18 | 14 | 1.25 | 510 | 425 | 340 |
| J-2 | 7 | 0.312 | 13 | 10 | 1 | 295 | 246 | 197 |
| J-3 | 5 | 0.25 | 10 | 7.5 | ¾ | 168 | 140 | 112 |
| J-4 | 3 | 0.203 | 7 | 5 | ⅝ | 80 | 67 | 54 |




Materials

Tubes: A50
Flanges: A36
Bolts: A325

| **Table 3. Comparison of Different Methods** | | | | | | | |
|---|---|---|---|---|---|---|---|
| Joint No. | Tension Force (kips) | BS 8100 | | Stelco | | Igarashi *et al.* | |
| | | $t_f$ (inch) | $n_{min}$ | $t_f$ (inch) | $n_{min}$ | $t_f$ (inch) | $n_{min}$ |
| J-1 | 510 | 1.04 | 7.4 | 1.84 | 6.2 | 1.26 | 9.3 |
| | 425 | 1.04 | 6.2 | 1.68 | 5.1 | 1.15 | 7.7 |
| | 340 | 1.04 | 4.9 | 1.51 | 4.1 | 1.03 | 6.2 |
| J-2 | 295 | 0.87 | 6.7 | 1.43 | 5.6 | 0.99 | 8.4 |
| | 246 | 0.87 | 5.6 | 1.31 | 4.6 | 0.90 | 7.0 |
| | 197 | 0.87 | 4.5 | 1.17 | 3.7 | 0.81 | 5.6 |
| J-3 | 168 | 0.69 | 6.8 | 1.11 | 5.6 | 0.79 | 8.5 |
| | 140 | 0.69 | 5.6 | 1.02 | 4.7 | 0.72 | 7.0 |
| | 112 | 0.69 | 4.5 | 0.91 | 3.8 | 0.64 | 5.6 |
| J-4 | 80 | 0.56 | 4.7 | 0.85 | 3.9 | 0.60 | 5.8 |
| | 67 | 0.56 | 3.9 | 0.77 | 3.2 | 0.55 | 4.8 |
| | 54 | 0.56 | 3.1 | 0.69 | 2.6 | 0.49 | 3.9 |

that the increase of flange outer diameter in a joint has very little influence on flange strength and can reduce total bolt force in the joint. Hence the method developed based on the basis of $D_f - D_p = D_p - D$ can actually be used for designs with the case of $D_f - D_p > D_p - D$.

### 4.2 Calculation of Total Bolt Force

Cao and Bell have carried out detailed analyses[15,17,19,20] for the calculation of total bolt force in a circular flange joint. In Figure 2 the "real curve" was obtained from a finite element analysis for a typical joint with total bolt preload $T_{b0}$. It can be seen that the curve can be represented by two straight lines without significant error and the two lines can be expressed as:

$$T_b = \begin{cases} 1.1T_{b0} & (T_b \le 1.1T_{b0}) \\ \mu \cdot T + \gamma \cdot T_{b0} & (T_b > 1.1T_{b0}) \end{cases} \tag{1}$$

where:

$T_b$ = Total bolt force in the joint
$T_{bo}$ = Total bolt preload in the joint
$T$ = Tension force applied to the joint

In Equation 1, $\mu$ is the slope of the bolt force vs. tension force line and $\gamma$ is used to take account of bolt preload. Formulas for the calculation of $\mu$ and $\gamma$ have been derived by Cao and Bell[15] but they are tedious for practical designs. It has been found[15,17,19,20] that the values of $\mu$ for some joints can be up to 1.7 but the values of $\gamma$ are usually small and can be taken as 0.1. Hence the evaluation of $\mu$ is much more important in the calculation of bolt force and for a joint under a relatively high tension force, as shown in Figure 2, the effect of bolt preload can be neglected. From Cao and Bell[15] the formula for $\mu$ can be expressed as:

$$\mu = \frac{1 - \omega_0\omega_1}{\omega_4 - \omega_2 - \omega_0\omega_3} \tag{2}$$

$\omega_0$, $\omega_1$, $\omega_2$ and $\omega_3$ are non-dimensional coefficients related to four dimensional ratios of $D / D_f$, $D / D_p$, $t / D$, and $t_f / t$. $\omega_4$ is a coefficient, which represents the ratio of bolt axial stiffness to flange bending stiffness at the bolt pitch circle. $\omega_4$ relates to the above four dimensional ratios and the ratio of the total cross sectional area of the bolts, $A_b$, to the square of the flange thickness $t_f$. Hence $\mu$ can be expressed as the following general equation:

$$\mu = f\left(\frac{D}{D_f}, \frac{D}{D_p}, \frac{t}{D}, \frac{t_f}{t}, \frac{A_b}{t_f^2}\right) \tag{3}$$

As a rational design for a joint, bolts should be decided according to the total bolt force in the joint. In the AISC LRFD design specification,[3] the design tensile strength is based on the lower value obtained from the limit states of yielding in the gross section (gross area × yield strength × resistance factor of 0.9) and fracture in the net section (net area × ultimate strength × resistance factor of 0.75). For bolted flange joints the tube net area equals the gross area. Furthermore, yielding of the gross section will typically be the design criterion since, even for grade A50 hollow sections, the yield to ultimate strength ratio = 50/62 = 0.81 < 0.75/0.90 = 0.83.

The design tension strength for all the bolts in a joint is $0.75A_bF_t$ (AISC LRFD section J3.6) where $F_t$, the bolt "nominal tensile strength", is specified in the corresponding Table

J3.2. Hence the required bolt area, $A_b$, to develop the maximum tension design capacity of the connected tube is:

$$A_b = \frac{\mu T}{0.75F_t} = \frac{0.9AF_{yt}}{0.75F_t}\mu = 1.2\pi t(D-t)\frac{F_{yt}}{F_t}\mu \tag{4}$$

So Equation 3 can be expressed as:

$$\mu = f\left(\frac{D}{D_f}, \frac{D}{D_p}, \frac{t}{D}, \frac{t_f}{t}, \frac{F_{yt}}{F_t}\right) \tag{5}$$

If $D_f - D_p = D_p - D$, $D/D_f$ in Equation 5 can be deleted and $\mu$ is a function of the other four ratios.

### 4.3 Calculation of Maximum Moment in a Flange

From the theoretical analysis[21] (1993), finite element analyses[17] (1994) and tests[20] (1995) carried out by Cao and Bell, it has been concluded that the maximum moment in a blank flange is either the bending moment at the tube/flange junction or the bending moment at the bolt pitch circle.

As shown in Figure 3(a), the flange in a joint is acted on by a uniformly annular tension force $T$, bolt force $T_b$, prying force $P(P = T_b - T)$, interactive moment $M$ and interactive radial force $V$ between the flange and tube. Because the tube is relatively thin, the interactive moment $M$ is low and the stress caused by this moment is usually much lower than the stresses caused by $T_b$ and $P$. The effect of radial force $V$ is also small so that the moment $M$ and force $V$ in Figure 3(a) can be neglected to make the calculation of maximum moment in a flange simpler (but the moment $M$ and force $V$ have been considered in the calculation of $P$ and $T_b$). Then the flange can be analyzed using the model shown in Figure 3(b), which is a symmetrically loaded circular plate. Solutions for the calculation of bending and hoop moments at any position of the flange plate can be obtained from an analysis carried out by Cao and Bell[22] (1992) (radius of central hole is zero here). If bolt preload is neglected, bolt force $T_b$ and prying force $P$ can be expressed as $T_b = \mu T$ and $P = (\mu - 1)T$. Hence the critical moments in a flange can be found to be:[22]

Bending moment at the tube/flange junction $M_{rj}$:

$$M_{rj} = \frac{-T}{8\pi}\left[(1-\nu)\frac{D_f^2 - (D-t)^2 - (D_f^2 - D_p^2)\mu}{D_f^2} + 2(1+\nu)\left(\ln\frac{D_f}{D-t} - \mu \cdot \ln\frac{D_f}{D_p}\right)\right] \tag{6}$$

Bending moment at the bolt pitch circle $M_{rp}$:

$$M_{rp} = \frac{T}{8\pi}\left[(1-\nu)\frac{D_f^2 - D_p^2}{D_f^2}\left(\mu - \frac{(D-t)^2}{D_p^2}\right) + 2(1+\nu)\ln\frac{D_f}{D_p}(\mu - 1)\right] \tag{7}$$

It is noted that the existence of prying force in a joint will reduce the moments at the tube/flange junction of the flange. Thicker flanges are usually needed for a joint if prying force is neglected in the assessment of flange strength.

If $\nu = 0.3$ is used and Equation 5 is considered, the maximum moment, $M_{max}$, for a flange can be obtained from Equations 6–7 and expressed as:

$$M_{max} = T \cdot R\left(\frac{D}{D_f}, \frac{D}{D_p}, \frac{t}{D}, \mu\right) = T \cdot R\left(\frac{D}{D_f}, \frac{D}{D_p}, \frac{t}{D}, \frac{t_f}{t}, \frac{F_{yt}}{F_t}\right) \tag{8}$$

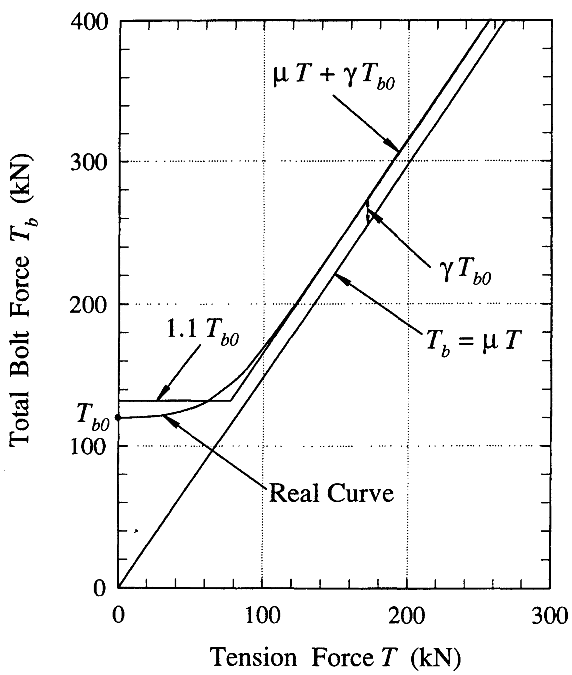



*Fig. 2. Relationship between bolt force and tension force.*

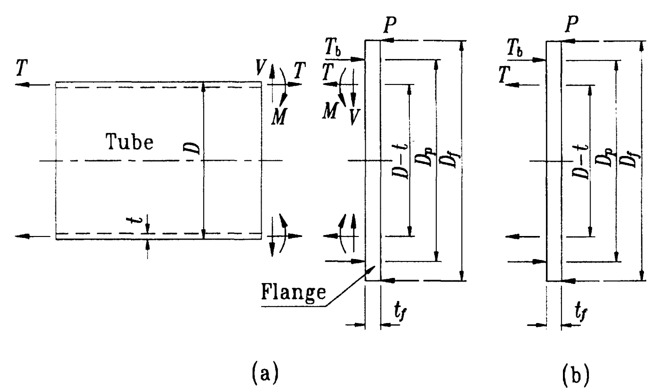



*Fig. 3. Calculation of bending moment in flange.*

Similar to μ in Equation 5, if $D_f - D_p = D_p - D$ is used, the maximum bending moment in the flange in Equation 8, $M_{max}$, is the product of tension force $T$ and a function of four ratios (without $D/D_f$ in function $R$).

### 4.4 Design Charts for Flange Thickness and Bolt Capacity

*If a flange is designed elastically* the maximum bending moment, $M_{max}$, in the flange should be limited not to exceed the yield moment, $M_y$, of the flange:

$$M_{max} \le M_y = \frac{t_f^2}{6} F_{yf} \tag{9}$$

Inserting Equation 8 gives:

$$\frac{6 \cdot T}{t_f^2} R \le F_{yf} \tag{10}$$

If the joint is designed to develop the full tension capacity of the tube, $T = 0.9AF_{yt}$, then Equation 10 can be expressed as:

$$\frac{t_f^2}{t^2} \ge 6(0.9)\pi \frac{D-t}{t} \frac{F_{yt}}{F_{yf}} R \tag{11}$$

Considering Equation 8 and $D_f - D_p = D_p - D$ and taking the critical state for the flange gives:

$$\frac{t_f}{t} = \sqrt{6(0.9)\pi \frac{D-t}{t} \frac{F_{yt}}{F_{yf}} R} = S\left(\frac{D}{D_p}, \frac{t}{D}, \frac{t_f}{t}, \frac{F_{yt}}{F_t}, \frac{F_{yt}}{F_{yf}}\right) \tag{12}$$

Hence the ratio of flange thickness to tube thickness, $t_f/t$, is a function of two dimensional ratios and two material property ratios. If materials for tube, flange and bolt are determined the ratio $t_f/t$ is a function of two dimensional ratios $D/D_p$ and $t/D$. Although it is not easy to find an explicit solution for Equation 12 the relationship between $t_f/t$, $D/D_p$ and $t/D$, for a material combination, can be determined by using an iteration process.

The preferred yield strength for round Hollow Structural Sections in the U.S. is 50 ksi (A50), the yield stress for structural plate is generally 36 ksi (A36) and the nominal strengths for bolts are 90 ksi (A325) and 113 ksi (A490). The results from Equation 12 for these two types of bolts are presented in Figures 4 and 5 (solid lines).

Equation 5 shows that after the material properties $F_{yt}$ and $F_t$ are decided the value of μ is a function of three dimensional ratios ($D/D_p$, $t/D$ and $t_f/t$). If the critical values of $t_f/t$, determined by Equation 12 are used for Equation 5, μ is also a function of $D/D_p$ and $t/D$. The results for μ (the prying ratio) are also presented in Figures 4 and 5 (dashed lines) for the two types of bolts. As explained before, the effect of bolt preload is negligible if tension force is relatively high, so the value of μ represents the ratio of total bolt force to applied tension force.

Similarly, *if a flange is designed plastically* using $M_p (M_p = 1.5M_y)$, of the flange in Equation 9, the charts for flange plastic design can be also obtained, as shown in Figures 6 and 7.

It can be seen that after the tube and flange pitch circle diameter for a joint are specified, the ratio $t/D$ (vertical axis in Figures 4 to 7) and the ratio $D/D_p$ (horizontal axis in Figures 4 to 7) can be determined. Then the critical value for ratio $t_f/t$ and the corresponding ratio of $T_b/T$ can be determined from the figures easily for the joint.

Although the charts are designed to develop the full tension capacity of the tube ($0.9AF_{yt}$) at a joint, the charts can also be

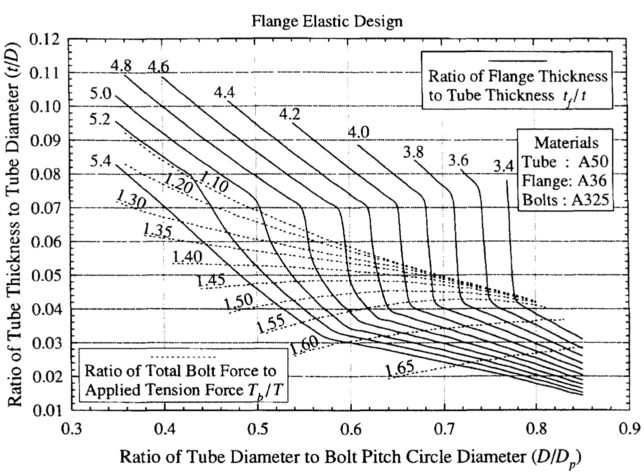



*Fig. 4. Design chart for flange elastic design using A325 bolts for circular flange joints.*

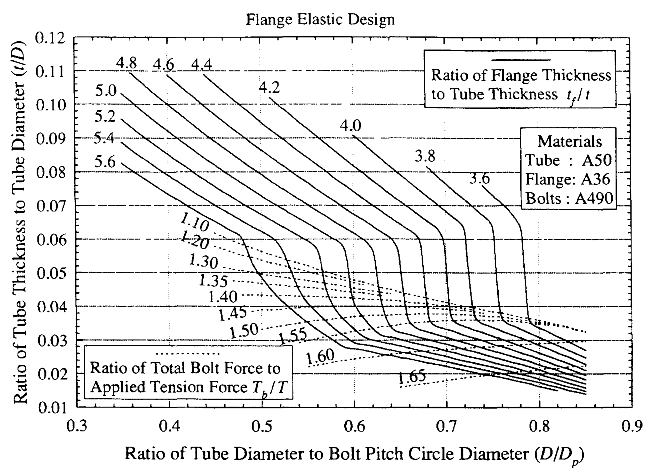



*Fig. 5. Design chart for flange elastic design using A490 bolts for circular flange joints.*

used for joints with lower tension loads. If the tension load for a joint is $x \cdot A F_{yt}$, the ratio of required flange thickness to tube thickness for the joint can be determined as following:

$$\left(\frac{t_f}{t}\right)_x = \left(\frac{t_f}{t}\right)_{0.9} \times \sqrt{\frac{x}{0.9}}$$

where:

$\left(\frac{t_f}{t}\right)_x$ = ratio of flange thickness to tube thickness for the joint

$\left(\frac{t_f}{t}\right)_{0.9}$ = ratio of flange thickness to tube thickness obtained from Figures 4 to 7

If a material other than A50 is used for the tube or a material other than A36 is used for the flange for a joint, the ratio of $t_f / t$ can be obtained by multiplying the ratio from Figures 4 to 7 by

$$\sqrt{\frac{36}{F_{yf}} \times \frac{F_{yt}}{50}}$$

where

$F_{yt}$ and $F_{yf}$ = yield stresses of the tube and the flange in ksi

The ratio of total bolt force to applied tension force can be taken to be the same.

## 5. EXAMPLES

The design charts, Figures 4 to 7, can be applied to the designs of circular flange joints easily. Table 4 shows the results for elastic and plastic designs for the joints listed in Table 2. The bolt pitch circles of the flanges have been determined based on the bolt diameters selected, as presented in Table 2, so the ratios of $t / D$ and $D / D_p$ can be obtained. Then the ratios of $t_f / t$ and $T_b / T$ can be found from Figure 4 and Figure 6 (using A325 bolts), for both flange elastic design and flange plastic design respectively. Three different levels of tension force, representing 90 percent, 75 percent and 60 percent of the squash load, are considered for each of the joints. The flange thickness and the minimum number of bolts for each of the joints under tension force $T_1$ can be found directly from the ratios of $t_f / t$ and $T_b / T$. The flange thicknesses for each of the joints under tension forces $T_2$ and $T_3$ can be obtained by multiplying the flange thickness under $T_1$ by $\sqrt{0.75 / 0.9}$ and $\sqrt{0.6 / 0.9}$, respectively. The minimum number of bolts, for each of the joints, under tension forces $T_2$ and $T_3$ can be obtained by multiplying the minimum number under $T_1$ by $0.75 / 0.9$ and $0.6 / 0.9$, respectively.

The number of bolts for a joint should be not less than four and the circumferential distance between two bolts should be greater than $3d$, as required in AISC LRFD design specification.[3] If the number of bolts for a joint is too big or too small the diameter of bolts should be changed and the joint should be redesigned.

It can be seen, from Table 4, that although the flange thicknesses are reduced by using plastic design, the numbers of bolts are increased. From Cao and Bell's tests[20] it has been found that the deformation of flanges can cause bending of bolts so that flanges should be rigid enough to resist excessive deformations. Hence the charts for flange elastic design may be preferable for some situations.

Comparison of Table 3 and Table 4 shows that the flange

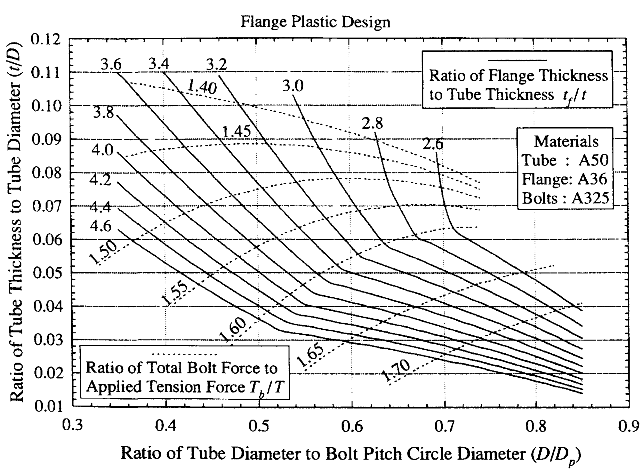



*Fig. 6. Design chart for flange plastic design using A325 bolts for circular flange joints.*

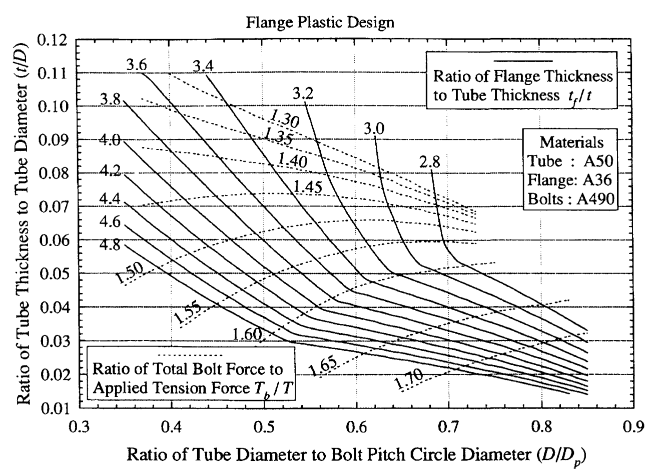



*Fig. 7. Design chart for flange plastic design using A490 bolts for circular flange joints.*

| Table 4. Examples | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Joint No. | Bolt Dia. | Parameters | | | | Tension Forces | | | | | |
| | | | | | | $T_1$ (90%) | | $T_2$ (75%) | | $T_3$ (60%) | |
| | $D_0$ inch | $t/D$ | $D/D_p$ | $t_f/t$ | $T_b/T$ | $t_f$ inch | $n_{min}$ | $t_f$ inch | $n_{min}$ | $t_f$ inch | $n_{min}$ |
| **Flange Elastic Design Using Figure 4** | | | | | | | | | | | |
| J-1 | 1.25 | 0.0375 | 0.714 | 4.09 | 1.59 | 1.53 | 9.8 | 1.40 | 8.1 | 1.25 | 6.5 |
| J-2 | 1 | 0.0446 | 0.700 | 3.96 | 1.51 | 1.24 | 8.4 | 1.13 | 7.0 | 1.01 | 5.6 |
| J-3 | ¾ | 0.0500 | 0.667 | 4.09 | 1.38 | 1.02 | 7.8 | 0.93 | 6.5 | 0.84 | 5.2 |
| J-4 | ⅝ | 0.0677 | 0.600 | 4.54 | 1.10 | 0.92 | 4.2 | 0.84 | 3.6 | 0.75 | 2.9 |
| **Flange Plastic Design Using Figure 6** | | | | | | | | | | | |
| J-1 | 1.25 | 0.0375 | 0.714 | 3.51 | 1.67 | 1.32 | 10.3 | 1.20 | 8.6 | 1.08 | 6.9 |
| J-2 | 1 | 0.0446 | 0.700 | 3.24 | 1.65 | 1.01 | 9.2 | 0.92 | 7.6 | 0.83 | 6.1 |
| J-3 | ¾ | 0.0500 | 0.667 | 3.16 | 1.63 | 0.79 | 9.2 | 0.72 | 7.6 | 0.65 | 6.1 |
| J-4 | ⅝ | 0.0677 | 0.600 | 3.05 | 1.55 | 0.62 | 6.0 | 0.57 | 5.0 | 0.51 | 4.0 |

thicknesses from elastic design in Table 4 are between the results from Stelco and the results from Igarashi *et al.* The flange thickness results from Igarashi et al. are very similar to the results from plastic design in Table 4, so the flanges designed in accordance with Igarashi *et al.* are rather flexible. Also, the plastic design method presented herein is a little more conservative than Igarashi *et al.* with regard to the requirements for the number of bolts.

## 5. CONCLUSIONS

Based on previous theoretical research work and tests on circular flange joints in tubular structures, a new design method for these joints is proposed. Design charts, which are in accordance with the AISC LRFD design specification, have been produced and can be used to decide flange thickness and number of bolts for a joint very easily. These design charts are produced for one tube grade and one flange grade but can be modified for various grades of tube and flange plate materials, both A325 and A490 bolts, and various levels of tension load applied to the connection. It has been shown that the method is accurate, reliable and easy to use.

The design examples given in the paper use the nominal geometric properties of several ASTM A500 grade round Hollow Structural Sections. However, the ASTM A500 standard[23] permits a hollow section wall thickness as much as 10 percent below the nominal wall thickness, without specifying any mass (or weight or cross-sectional area) tolerance. This can have a major negative effect on the assumed (nominal) structural properties[24] as most U.S. manufactures now tend to produce thin sections. Both the Steel Tube Institute of North America and the American Institute of Steel Construction will shortly be publishing new section properties for hollow sections produced in accordance with ASTM A500. These will be based on a tube wall thickness, for design purposes, of 0.93 times the nominal tube wall thickness. The design procedures and charts presented herein will still be applicable for these new properties as the tube wall thickness is an explicit variable in Figures 4 to 7.

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