On the ductility of bolted steel joints at ambient and elevated temperatures

Fernando C.T. Gomes,
João Paulo C. Rodrigues

Department of Civil Engineering, Coimbra University, Portugal

Abstract

Beam-to-column joints must be ductile enough to sustain plastic deformation without premature failure when the moment resistance of the joint is less than the moment resistance of the connected beam. In order to prevent premature failure, the Eurocode 3 imposes that the rotation capacity of a joint must be checked if the design moment resistance of the joint is less than 1.2 times the design plastic moment resistance of the connected beam. The Eurocode 3 gives a simple rule for the rotation capacity of bolted joints, based on the assumption that joints designed with that rule are ductile enough to prevent premature failure of the bolts. This paper analyses the Eurocode 3 ductility rule and shows that this rule is in certain cases unsafe and can lead to brittle failure of joints. An alternative rule is proposed for beam-to-column bolted joints at ambient and elevated temperatures.

Introduction

The ductility of steel structures is a research topic that has deserved the attention of the scientific community. A comprehensive overview on this topic was made by Gioncu and Mazzolani,[1] mainly for seismic resistant steel structures, but also important in the case of other structural actions.

In what concerns the ductility of joints, the research is much less abundant. In the case of beam-to-column joints, the trend has been to avoid the formation of a plastic hinge at the joint, by using expensive full strength joints or, for instance, by using the technique of the *dog bone*, developed by Plumier,[2] that imposes the formation of a plastic hinge in the beam.

Modern design codes like the Eurocode 3 (EN 1993-1-1:2005, EN 1993-1-8:2005)[3,4] permit the use of partial strength beam-to-column joints where a plastic hinge may form. These joints have to accommodate the required rotations, mainly due to plastic deformation of ductile components. Failure of brittle components has to be avoided which implies an accurate evaluation of the ductile component strength. However, the main difficulty is to evaluate the behaviour of plate components like end-plate or column flange.[5]

Numerical simulations by finite elements have been used to analyse the behaviour of bolted joints. Complex discretization is necessary to obtain accurate enough results but the conclusions are limited.[6] Haremza *et al.*[7] presented the results of an experimental investigation on a two dimensional composite steel-concrete beam-to-column sub-frame. The main objective of the research was to provide a detailed analysis of the heated joint behavior subject to variable bending moments and axial loads when the column fails. The loss of a column under localised fire induces large vertical displacements in the above floors. To reach equilibrium in the deformed configuration and avoid progressive collapse of the building, membrane forces in the slabs, and catenary forces in beam elements should develop.[8,9] The connections are required to have sufficient ductility in order to sustain large rotations without brittle failure. It was concluded that due to the low slenderness of the composite beam this topology of composite joint does not possess sufficient capacity of rotation to reach the equilibrium deformed configuration.

Analysis of the Eurocode 3 ductility rule

According to the Eurocode 3,[4] the rotation capacity of a joint must be checked if the design moment resistance of the joint is less than 1.2 times the design plastic moment resistance of the cross section of the connected member. One possibility is to determine the rotation capacity of the joint by testing in accordance with EN 1990, Annex D. As an alternative to testing, the following rule is given in EN 1993-1-8:2005, 6.4.2:[4]

A joint with either a bolted end-plate or angle flange cleat connection may be assumed to have sufficient rotation capacity for plastic analysis, provided that both of the following conditions are satisfied: (a) the design moment resistance of the joint is governed by the design resistance of either the column flange in bending or the beam end-plate or tension flange cleat in bending; (b) the thickness t of either the column flange or the beam end-plate or tension flange cleat [not necessarily the same basic component as in (a)] satisfies:

$$t \le 0.36d\sqrt{\frac{f_{ub}}{f_y}} \tag{1}$$

where d is the nominal diameter of the bolt, is the yield strength of the relevant basic component and f_{ub} is the ultimate strength of the bolt.

At elevated temperatures, the ductility condition (1) will be necessarily different because the reduction of f_{ub} is different from the reduction of f_y. The EN 1993-1-2:2005[10] defines strength reduction factors for plates ($k_{y,\theta}$) and bolts ($k_{b,\theta}$) at elevated temperatures (Figure 1). Following the procedure indicated in the EN 1993-1-2:2005[10] to evaluate the design resistance of plates in bending, the term f_y/γ_{M0} (at ambient temperature) should be replaced by $k_{y,\theta}f_y/\gamma_{fi}$ (at elevated temperature). When the design resistance at ambient temperature depends on the ultimate strength (for instance, shear failure), the term f_u/γ_{M2} should be replaced by $k_{y,\theta}f_y/\gamma_{fi}$. For the design resistance of bolts at elevated temperatures, the term f_{ub}/γ_M should be replaced by $k_{b,\theta}f_{ub}/\gamma_{fi}$. In all cases, γ_{fi} is a national determined parameter with a recommended value γ_{fi}=1.0 (EN 1993-1-2:2005).[10]

The Eurocode 3 rule (1) is based on the assumption that bolted joints have sufficient rotation capacity if the resistance of each individual bolt is greater than the resistance of one of the connected plates (end-plate, column flange or cleats) in order to prevent premature failure of the bolts. The design resistance of a bolt in tension is given in the EN 1993-1-8:2005[4] as:

$$F_{t,Rd} = \frac{0.9 A_s f_{ub}}{\gamma_{M2}} \tag{2}$$

where A_s is the tensile stress area of the bolt and γ_{M2} is a partial safety factor.

According to the EN 1993-1-8:2005,[4]

Correspondence: João Paulo C. Rodrigues, Department of Civil Engineering, Coimbra University, Rua Luis Reis Santos, 3030-786 Coimbra, Portugal.
Tel: +351.239797237.
E-mail: jpaulocr@dec.uc.pt

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the design resistance of a plate reaches the maximum value in the case of a circular mechanism, which is

$$F_{p,Rd} = \frac{\pi t^2 f_y}{\gamma_{M0}} \tag{3}$$

where t is the plate thickness and γ_{M2} is a partial safety factor.

The base assumption of ductility condition (1) may be written in the form

$$\gamma_{ov} F_{p,Rd} \le F_{t,Rd} \tag{4}$$

where γ_{ov} is an overstrength factor, $F_{t,Rd}$ is given by Equation (2) and $F_{p,Rd}$ is given by Equation (3). By introducing the equivalent diameter given by the approximate relation $d_s \approx 0.88d$ (Table 1) and the partial safety factors given by the recommended values $\gamma_{M0}=1.0$ and $\gamma_{M2}=1.25$, the inequality (4) is rewritten in the form

$$t \le \frac{0.37d}{\sqrt{\gamma_{ov}}}\sqrt{\frac{f_{ub}}{f_y}} \tag{5}$$

The inequality (5) becomes identical to the Eurocode 3 rule (1) if the overstrength factor takes the value

$$\gamma_{ov}=1.08 \tag{6}$$

Gioncu and Mazzolani[1] and ECCS[11] discuss the overstrength factors, proposed in the Eurocode 8 (EN 1998-1:2004),[12] to prevent brittle failure of steel joints in the case of seismic design. They indicate that the overstrength factors should be increased.

Results and Discussion

Bolt head diameter

The EN 1993-1-8:2005[4] assume in Equation (3) that the force applied by the bolt head or nut is concentrated in a point, *i.e.*, the actual dimension of the bolt head or nut is not taken into account. In the following, the resistance of a circular plate is evaluated as a function of the mean diameter d_m of a bolt head (or nut), Figure 2, which is defined by

$$d_m = \frac{d_1 + d_2}{2} \tag{7}$$

The mean diameter d_m is frequently greater than three times the thickness t of the plate. In fact, taking for instance f_{ub}=800 MPa and f_y=355 MPa, the ductility condition (1) gives for the plate thickness $t \le 0.54d$. And, using the approximate value of the mean diameter of the bolt head $d_m \approx 1.7d$, the condition (1) becomes

$$d_m \ge 3.2t \tag{8}$$

Equation (3) under-evaluate the maximum resistance of a plate loaded by a single bolt mainly because it considers that the bolt act as a point load. In the following, a detailed plastic analysis of a circular plate is performed in order to determine the influence of the bolt head diameter d_m on the resistance of the plate, considering bending failure as well as shear failure mechanisms.

Bending failure mechanism

If a yield line is formed by bending only, *i.e.*, if normal and shear forces are negligible, the plastic moment per unit length of a plate is given by

$$m_{pl} = \frac{1}{4} t^2 f_y \tag{9}$$

where t is the plate thickness and f_y is the yield strength. The yield line method of Johansen (1962)[13] uses a square yield criterion given by

$$\max(|m_1|,|m_2|) = m_{pl} \tag{10}$$

where m_1 and m_2 are the bending moments per unit length in the principal directions 1 and 2. The square yield criterion of Johansen is compared in Figure 3 with the criterion of Tresca, represented by the hexagon, and with the criterion of Von Mises, represented by the ellipse.

The failure load of a clamped circular plate with a rigid central core representing a bolt head and nut (Figure 4) is given by the criterion of Johansen as:

$$F_{JOHANSEN} = \frac{\pi t^2 f_y}{1-x} \tag{11}$$

where

$$x=d_m/D \tag{12}$$

If we neglect the dimension of the bolt head, *i.e.*, if we consider that the bolt act as a point load (x=0), equation (11) becomes identical to the Eurocode 3 formula (3) with $\gamma_{M0}=1.0$, depicted in Figure 5 by the horizontal line.

Using the criterion of Tresca, the failure load of a circular plate with a rigid central core of diameter d_m, Figure 4, calculated by Ilyushin,[14] is equal to the failure load of a circular plate with a linear load on a circumference of diameter d_m calculated by Sawczzuk and Jaegar.[15] This failure load, depicted in Figure 5 by the curve TRESCA, is given by

$$F_{TRESCA} = \frac{1}{2}\pi t^2 f_y\left(1 - \frac{1}{\ln \rho}\right) \tag{13}$$

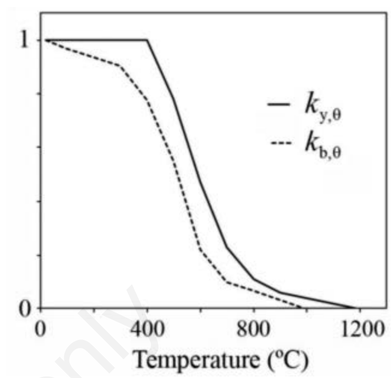

Figure 1. Strength reduction factors for plates and bolts at elevated temperatures.

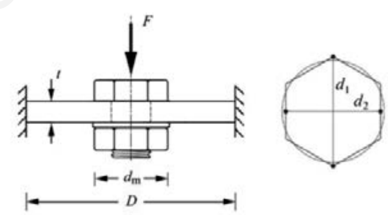

Figure 2. Mean diameter d_m of a bolt head or nut.

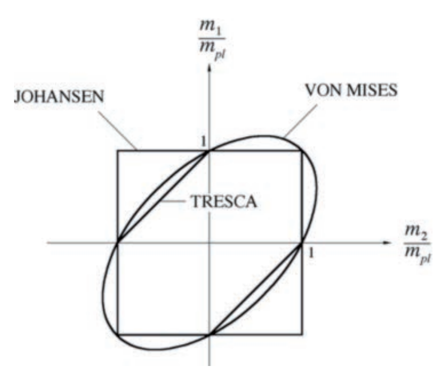

Figure 3. Yield criteria of Johansen, Tresca and Von Mises.

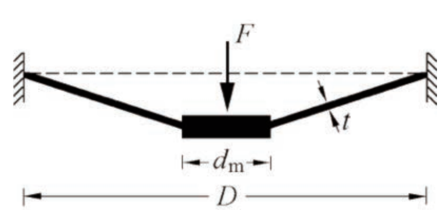

Figure 4. Bending failure mechanism of Johansen.

where the values of r are obtained from the equation

$$\rho - x(1 - \ln\rho) = 0 \tag{14}$$

and x is given by Equation (12).

There is no analytical solution by the criterion of Von Mises for the circular plate with a rigid central core, Save *et al*. (1997). However, if we multiply by $2/\sqrt{3}$ the values of F_{TRESCA}, we get

$$F_{\text{VON MISES (envelope)}} = \frac{2}{\sqrt{3}} F_{\text{TRESCA}} \tag{15}$$

which is an upper bound solution since the ellipse of Von Mises is circumscribed by the hexagon depicted in Figure 6 as Von Mises (envelope). The failure load obtained by this upper bound criterion is shown in Figure 5.

Figure 5 shows that the yield line method of Johansen provides a solution almost always between the Tresca and the Von Mises solutions, except for $x<0.3$. In particular, when x tends to zero (point load), the Tresca and Von Mises solutions are 50 and 57.7% of the Johansen solution, respectively. However, very small values of x are uncommon since minimum values of x are imposed by the ductility condition as, for instance, condition (8). Therefore, because the values of x are in general greater than 0.3, the method of Johansen gives a good solution for the bending failure load of circular plates used in ductile joints. Figure 5 also shows that, in general, the Eurocode 3 failure load given by Equation (3) underestimates the failures load.

Shear failure mechanism

In the case of a punching shear failure mechanism, the maximum shear force occurs in the yield line around the bolt head or nut (Figure 7). According to the Von Mises criterion, the maximum shear stress is $f_y/\sqrt{3}$. Thus, the plastic shear force per unit length of yield line reaches

$$v_{\text{pl}} = \frac{tf_{\text{y}}}{\sqrt{3}} \tag{16}$$

Considering the bolt and nut as a central core with diameter d_{m}, the force per unit length of yield line $F/(\pi d_{\text{m}})$ reaches the plastic value v_{pl}. In this case, the shear failure load is

$$F_{\text{shear}} = \pi\, d_{\text{m}} v_{\text{pl}} = \frac{\pi\, d_{\text{m}} t f_{\text{y}}}{\sqrt{3}} \tag{17}$$

Combined bending and shear mechanism

The combined bending and shear mechanism of a clamped circular plate with a central rigid core is depicted in Figure 8. The central core is representative of a bolt head or nut with mean diameter d_{m}. The failure mechanism is defined by the virtual displacements δ_1, δ_2 and Δ. The yield line 1 around the central core is subject to the rotation θ and the shear displacement δ_1, simultaneously, while the yield line 2 is subject to the rotation θ and the shear displacement δ_2. Supposing small rotations ($\theta \approx \tan\,\theta$), the displacement Δ and the rotation θ are related by the geometric relation:

$$\Delta = \frac{D - d_{\text{m}}}{2}\theta \tag{18}$$

Interaction m-v: normality law

A combined bending and shear yield line is represented by a section in a plane perpendicular to the yield line. The yield line is characterized by the rotation θ and the shear displacement δ (Figure 9), related to the bending moment m and the shear force v per unit length of yield line, respectively.

There is no exact solution for the interaction m-v.[16] However, the approximate solution of Drucker[17] may be used in the form (Figure 10).

$$\frac{m}{m_{pl}} = 1 - \left(\frac{v}{v_{pl}}\right)^4 \tag{19}$$

where m_{pl} and v_{pl} are given by Equations (9) and (16), respectively.

In a combined bending and shear yield line, the rotation θ and the shear displacement δ are related by the normality law, according to which the vector defined by the couple (δ,θ) is perpendicular to the interaction curve m-v (Figure 10). Consequently, the normality law is given by

$$\frac{\theta}{\delta} = \frac{-1}{\dfrac{\text{d}m}{\text{d}v}} \tag{20}$$

Using the interaction curve (19), the normality law (20) imposes

$$\frac{\delta}{\theta} = -\frac{\text{d}m}{\text{d}v} = \frac{4m_{pl}}{v_{pl}}\left(\frac{v}{v_{pl}}\right)^3 \tag{21}$$

Taking into account Equations (19) and (21), the internal virtual work by unit length of yield line is given by

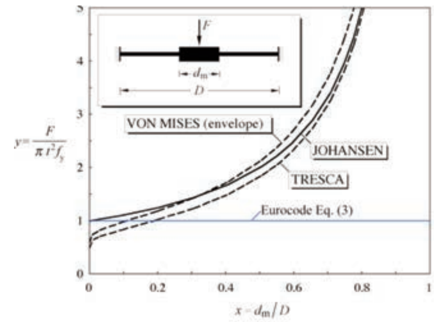

Figure 5. Failure load by bending mechanisms. Influence of the yield criterion.

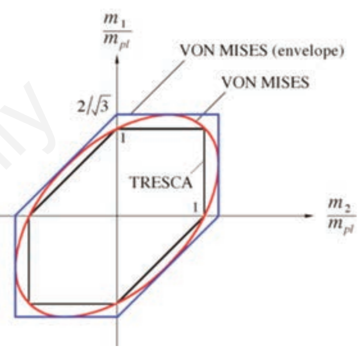

Figure 6. Comparison of different yield criteria.

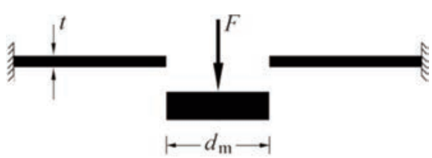

Figure 7. Punching shear failure mechanism.

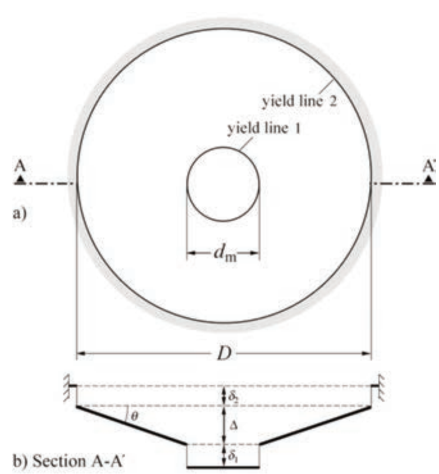

Figure 8. Combined bending and shear failure mechanism.

$$m\theta + v\delta = m_{pl}\theta\left[1 + 3\left(\frac{v}{v_{pl}}\right)^4\right] \tag{22}$$

Failure load

The plastic failure load may be obtained by the principle of virtual work,

$$W_{\text{int}} = W_{\text{ext}} \tag{23}$$

where the external virtual work is

$$W_{\text{ext}} = F(\delta_1 + \delta_2 + \Delta) \tag{24}$$

and the internal virtual work is given by the sum of the three parts

$$W_{\text{int}} = W_1 + W_2 + W_3 \tag{25}$$

where W_1 is the virtual work in the yield line 1, subject to m_1 and v_1

$$W_1 = \pi d_{\text{m}}(m_1\theta + v_1\delta_1) \tag{26}$$

W_2 is the virtual work in the yield line 2, subject to m_2 and v_2

$$W_2 = \pi D(m_2\theta + v_2\delta_2) \tag{27}$$

and W_3 is the virtual work in the zone between the two yield lines, subject to m_{pl}

$$W_3 = \pi(D - d_{\text{m}})m_{\text{pl}}\theta \tag{28}$$

The shear forces in the yield lines 1 and 2 are, respectively

$$v_1 = \frac{F}{\pi d_{\text{m}}} \text{ and } v_2 = \frac{F}{\pi D} \tag{29}$$

From Equation (21) we get

$$\delta_1 = \theta\frac{4m_{pl}}{v_{pl}}\left(\frac{v_1}{v_{pl}}\right)^3 \tag{30}$$

$$\delta_2 = \theta\frac{4m_{pl}}{v_{pl}}\left(\frac{v_2}{v_{pl}}\right)^3 \tag{31}$$

Furthermore, introducing Equation (19), the failure load for the combined bending and shear mechanism may be obtained from Equation (23), which becomes the following fourth degree equation

$$\frac{y^4}{2\alpha^4}\left(\frac{1}{x^3} + 1\right) + y(1 - x) - 1 = 0 \tag{32}$$

where

$$y = \frac{F}{\pi t^2 f_y} \tag{33}$$

$$\alpha = \frac{D}{t\sqrt{3}} \tag{34}$$

and x is given by Equation (12). The value of α is equal to the value of y for x=1.

Figures 11 and 12 depict the resistance of circular plates with slenderness D/t equal to 10 and 7, respectively. In these two figures the curve *Bending*, the line *Shear* and the curve *Bending & shear* represent Equations (11), (17) and (32), respectively. The Eurocode 3 Equation (3) is also represented in these figures, showing that this equation clearly underestimates the failure load. The curve *Bending & shear* is always under the curve *Bending* meaning that the failure load is always less than that given by the bending mechanism. The failure load may be less or equal to that given by the shear mechanism, which means that, in the case of Figure 11, for x<0.2 the failure is determined by the shear mechanism while the combined bending and shear mechanism governs for x>0.2.

In the case of thick plates, for instance $D/t \leq 7$, the failure load becomes slightly lower than or equal to the *Shear* load (Figure 12).

Design formulae at ambient and elevated temperatures

At ambient temperature, the design resistance $F_{\text{p,Rd}}$ of the circular plate with a central core, representative of a bolt head with mean diameter d_{m}, may be determined from Equation (32) with a slight modification of the parameter α that takes into account the safety factors (γ_{M0} and γ_{M2}) as well as strain hardening (f_{u} instead of f_y). The non-dimensional parameter α becomes then

$$\alpha = \frac{F_1}{F_0} \tag{35}$$

$$F_1 = \frac{\pi D t f_{\text{u}}}{\gamma_{\text{M2}}\sqrt{3}} \tag{36}$$

where

$$F_0 = \frac{\pi t^2 f_y}{\gamma_{\text{M0}}} \tag{37}$$

Thus, the design resistance $F_{\text{p,Rd}}$ at ambient temperature may be obtained from the equation

$$\frac{y^4}{2\alpha^4}\left(\frac{1}{x^3} + 1\right) + y(1 - x) - 1 = 0 \tag{38}$$

where x and α are given from equations (12) and (35), respectively, and

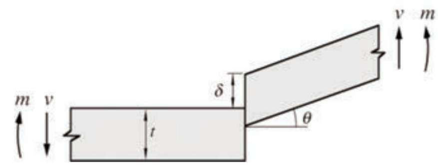

Figure 9. Combined bending and shear yield line.

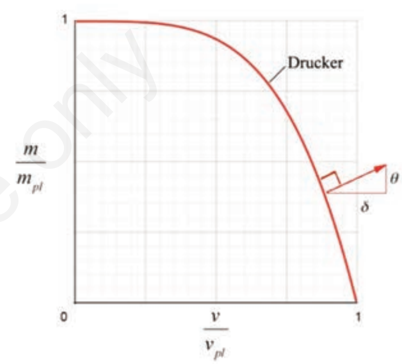

Figure 10. Interaction between bending moment m and shear force *vs* normality law.

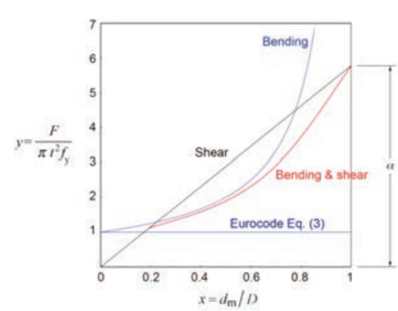

Figure 11. Failure load for *D/t*=10.

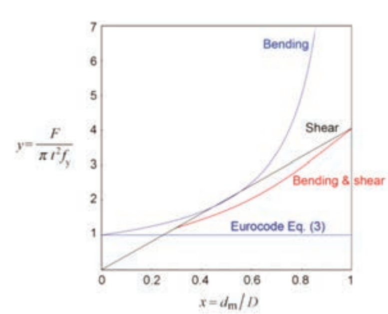

Figure 12. Failure load for *D/t*=7.

$$y = \frac{F_{\text{p.Rd}}}{F_0} \quad (39)$$

At elevated temperatures, following the procedure indicated in the EN 1993-1-2: 2005, the term f_y/γ_{M0} in equation (37) should be replaced by $k_{y,\theta}f_y/\gamma_{fi}$ and the term f_u/γ_{M2} in Equation (36) should be replaced by $k_{y,\theta}f_y/\gamma_{fi}$, where γ_{fi} is a national determined parameter with a recommended value γ_{fi}=1.0. Therefore, the design resistance $F_{p,Rd,fi}$ at elevated temperatures may be obtained from Equation (38) where y and α are modified as follows:

$$y = \frac{F_{\text{p.Rd.fi}}}{F_{0.\text{fi}}} \quad (40)$$

$$\alpha = \frac{F_{1.\text{fi}}}{F_{0.\text{fi}}} \quad (41)$$

where

$$F_{0.\text{fi}} = k_{y,\theta}\frac{\pi t^2 f_y}{\gamma_{\text{fi}}} \quad (42)$$

$$F_{1.\text{fi}} = k_{y,\theta}\frac{\pi D t f_y}{\gamma_{\text{fi}}\sqrt{3}} \quad (43)$$

Equations (41) and (34) are identical if we take into account Equations (42) and (43).

Proposal of a ductility rule

The plate ductility is influenced by membrane effect, as illustrated in Figure 13b. In fact, due to membrane effect, the plate resists beyond the plastic limit F_{pl} until punching shear failure occurs around the bolt head, B_p (Figure 13a). The punching shear resistance B_p is an upper bound of the plate resistance. Thus, in order to prevent brittle failure, the resistance of the bolt in tension F_t should be greater than the resistance of plate in punching shear B_p.

The greater is the slope of the membrane effect curve the smaller is the ultimate displacement δ_u (Figure 13a). The topic of the plate's membrane effect is well documented.[16]

In this sense, instead of inequality (4), the ductility rule should be written in the form

$$\gamma_{ov} B_{\text{p.Rd}} \le F_{\text{t,Rd}} \quad (44)$$

where γ_{ov} is the overstrength factor, $B_{p,Rd}$ is the design resistance of the plate in punching shear, given in EN 1993-1-8:2005[4] as

$$B_{\text{p.Rd}} = \frac{0.6\pi d_m t f_u}{\gamma_{M2}} \quad (45)$$

and $F_{t,Rd}$ is the design resistance of a bolt in tension, given in EN 1993-1-8:2005[4] by Equation (2). If we take d_s≈0.88d (Table 1) and d_m≈1.7d, the inequality (44) becomes

$$t \le \frac{0.17 d f_{ub}}{\gamma_{ov} f_u} \quad (46)$$

or, considering γ_{ov}=1.25, we get the following ductility rule in the case of ambient temperature

$$t \le \frac{0.14 d f_{ub}}{f_u} \quad (47)$$

The ductility rule (47) and the Eurocode 3 rule (1) are compared in Table 2 for one bolt steel (grade 8.8) and two plate steels (S235 and S355). The last column of Table 2 shows that the Eurocode 3 rule (1) gives a plate thickness that may be more than twice the thickness calculated by the proposed rule (47), which means that the Eurocode 3 rule (1) may lead to brittle failure of bolts.

At elevated temperatures the ductile rule becomes

$$t \le \frac{0.14 d k_{b,\theta} f_{ub}}{f_{u,\theta}} \quad (48)$$

where $f_{u,\theta}$ is given by

$$f_{u,\theta} = \begin{cases} 1.25 k_{y,\theta} f_y & \text{for } \theta < 300^\circ\text{C} \\ k_{y,\theta} f_y (2 - 0.0025\theta) & \text{for } 300^\circ\text{C} \le \theta < 400^\circ\text{C} \\ k_{y,\theta} f_y & \text{for } \theta \ge 400^\circ\text{C} \end{cases} \quad (49)$$

where θ is the steel temperature and the reduction factors $k_{y,\theta}$ and $k_{b,\theta}$ are defined in EN 1993-1-2:2005 (Figure 1).

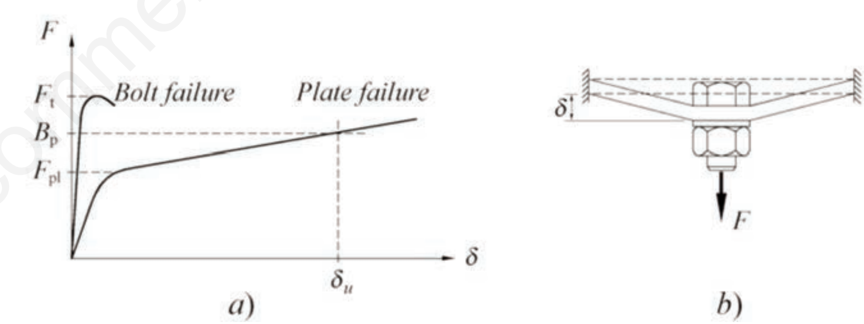

Figure 13. Membrane effect *vs* ductility.

Table 1. Equivalent diameter of a bolt.

Bolt	M16	M20	M24	M30
Nominal diameter d (mm)	16	20	24	30
Tensile stress area A_s (mm²)	157	245	353	565
Equivalent diameter $d_s=2\sqrt{A_s/\pi}$ (mm)	14.1	17.7	21.2	26.8
d_s/d	0.88	0.88	0.88	0.89

Table 2. Comparison of the ductility rule (47) with the Eurocode 3 rule (1).

Bolt Steel	Bolt f_{ub} (N/mm²)	Plate Steel	Plate f_u (N/mm²)	$t_{max(47)}/d$ Ductility rule (47)	t_{maxEC3}/d Eurocode rule (1)	$t_{maxEC3}/t_{max(47)}$
8.8	800	S235	360	0.31	0.66	2.1
8.8	800	S355	510	0.22	0.54	2.5

Conclusions

In bolted steel joints, the resistance of bolts in tension should be greater than the resistance of the steel end-plate, cleat or column flange in tension. In this paper, the detailed analysis of a circular plate demonstrates that it is necessary to evaluate correctly the resistance of the steel plate taking into account the actual dimension of the bolt head or nut, in order to avoid brittle failure of bolts in tension.

The Eurocode 3 ductility rule may lead to a brittle failure of the joint. In fact, the plate components resistance is underestimated by the Eurocode 3, which is safe for the resistance evaluation but is unsafe for ductility control.

As a conclusion, in order to avoid brittle failure of bolted beam-to-column joints, the currently possibilities are: (a) to prevent the formation of a plastic hinge in the joint by using full-strength joints, or (b) to use the proposed ductility rule for ambient and elevated temperatures.

Finally, it must be noted that the proposed ductility rule is rather restrictive. Therefore, it is necessary more research on failure mechanism of plates (end-plate, column flange, angles, *etc.*) that will lead to a less restrictive ductility rule. Experimental tests could be useful showing that EN 1993-1.8 formulation may lead to brittle failure of bolts and thus to insufficient rotation capacity of some joint configurations.

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