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Fin plate joint using component method of EN 1993-1-8

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Summary. Fin plate joint is widely used in steel structures. In many cases the joint can be classified as a hinge following the standard EN 1993-1-8. In the standard a very generic component method is given which can be used to analyze the behavior of the joints both in ambient and fire conditions. The behavior means in this case the resistance and the stiffness of the joint. In this paper an application of the component method is proposed for fin plate joints. The results of the component method are compared to the test results and more comprehensive finite element simulations available in the literature. In fire cases large rotation takes place in the joint and then the rotational stiffness and the moment resistance of the joint increase considerably. This is due to fact that the gap between the beam flange and the column flange is closed.

Key words: fin plate joint, component method, fire and ambient conditions.

Introduction

This paper is based on the seminar presentation of the first author in a PhD seminar in Tampere University of Technology during spring 2010 [1]. Fin plate joints are characterized as joints, where the beam web is bolted to the vertical plate which is welded to the column face or to the beam web and flanges as shown in Figure 1.

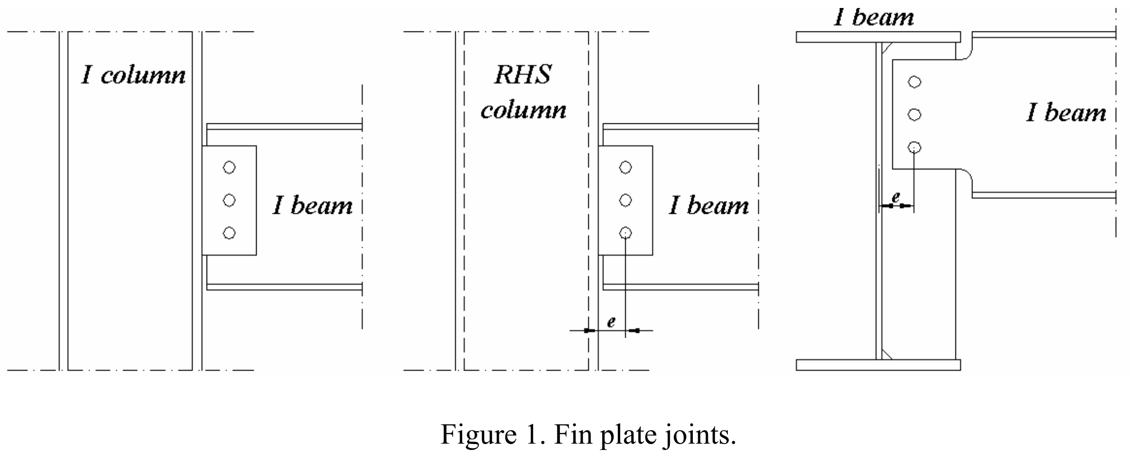

Figure 1. Fin plate joints.

Fin plate joints are considered as simple joints [2] transferring mainly shear forces from the beam web to the supporting member. Fin plate joints are classified often as hinges, meaning that no bending moment will be transferred from the beam end to the supporting structures. Following the standard [3] the classification is done using the equation

$$S_{ini}L_b/(E_bI_b) \leq 0.5 \tag{1}$$

where S_{ini} is the initial rotational stiffness of the joint, L_b the span of the beam to be jointed, E_b the elastic modulus of the beam to be jointed, I_b the inertia moment of the beam to be jointed.

So, in order to verify the behaviour of the joint the initial rotational stiffness S_{ini} of the joint should be calculated. Moreover, the span of the beam L_b should be verified. The definition of the span of the supported beam is not quite clear when looking at the joints in Figure 1.

When considering the joint in fire, then the initial rotational stiffness of the joint should be reduced following the rules of [4]. If it is supposed that the entire joint, meaning all parts of the joint, have the same elevated temperature in fire, then the reduction of the rotational stiffness is done by reducing the rotational stiffness using the factor $k_{E,\theta}$ of the standard [4], i.e. use the reduction factor for the elastic modulus of steel at the elevated temperature θ. If it is supposed that the entire beam is at the same elevated temperature, then the same reduction is used for the quantity E_b and the equation (1) is the same for the fire case.

To calculate the stiffness and the resistance of the joint under different actions (loads) appears to be rather complicated task, although the joint layout is considered simple. To systematize the calculations the component method proposed in the standard [3] has proven to be a generic powerful tool to analyse many kinds of structural joints [5] - [8]. When constructing the local analysis model of the joint the basic components' behaviours, the location of the model, composition of the components are the main steps.

In this paper the component model is proposed for the fin plate joints used in beam to I-column flange joints (Figure 1, left hand side joint). The local deformations due to the bearing near the bolts of the beam web and of the fin plate and, the bolt shear deformations are taken into account. The deformations of the fin plate itself and the column are neglected in this study, as well the deformations of the welds. The equations for stiffness and resistances of the deformable components are given following the standard [3] and the composition of those is given in this paper. Bi-linear elastic-ideal-plastic force-deformation relationships are supposed to be valid for the components. Tri-linear relationships are available [9] for most of the components presented in the standard [3], but they are not used in this study. The results are compared to the tests and more comprehensive finite element method results available in the literature both in ambient and fire conditions.

Bolt in shear

The basic component of the fin plate joint is two plates bolted together with one bolt and loaded by the shear force. If the plates are large, meaning in this case that the edges of the plates are far away from the bolt, the shear resistance failures can be:

- Bolt in shear,
- Bearing resistance, contact failure between the bolt side and the plate.

The bearing resistance should be checked for both plates connected in the joint. The deformation of the joint is composed of the same deformations as is done for the resistance of the joint:

- Bolt in shear deformation,
- Bearing deformation at both plates.

If the connected plates are not large, then the new failure mode may become critical:

- Tension tearing at the edge of the plate.

This failure mode is not listed into the design rules of the standard [3]. It is included in the equations of the standard.

The design of the joint should be done so, that the failure mode is ductile. This means that the critical failure mode is the bearing resistance. To ensure this the following safe rules of thumb can be derived based on experiments and analysis [10]:

- Edge distance of the bolt should be at least two times the bolt diameter,
- Bolt diameter should be at least two times the thickness of the thinnest plate in the joint.

When the load is acting in any angle against the edge, then the resistance can be checked following [3] by checking the component perpendicular to the edge and by checking the component parallel to the edge. It is supposed in this paper that the same rule holds for the stiffness, too. This means that no reduction in the interaction is needed.

If there is more than one bolt in the joint, this question arises first: how will the total load of the joint distribute to the bolts? Two ways can be used to define the bolt loads: elastic and plastic analysis. Also new failure modes can be critical in this case. Those are such as block tearing and effect of neighbouring bolts to bearing and tension tearing. The block tearing is not considered in this paper. The effects of other failure modes are included in the equations of the standard [3].

Failure modes for the shear, bending and buckling of the fin plate, the shear failure of the supported beam, failures of welds and the failure of supporting column are not considered in this paper. These are considered in [2]. Also, the ductility and rotation requirements are considered in [2].

Bolt in shear component

Figure 2 illustrates two plates connected with one bolt and the joint is loaded by the shear force F.

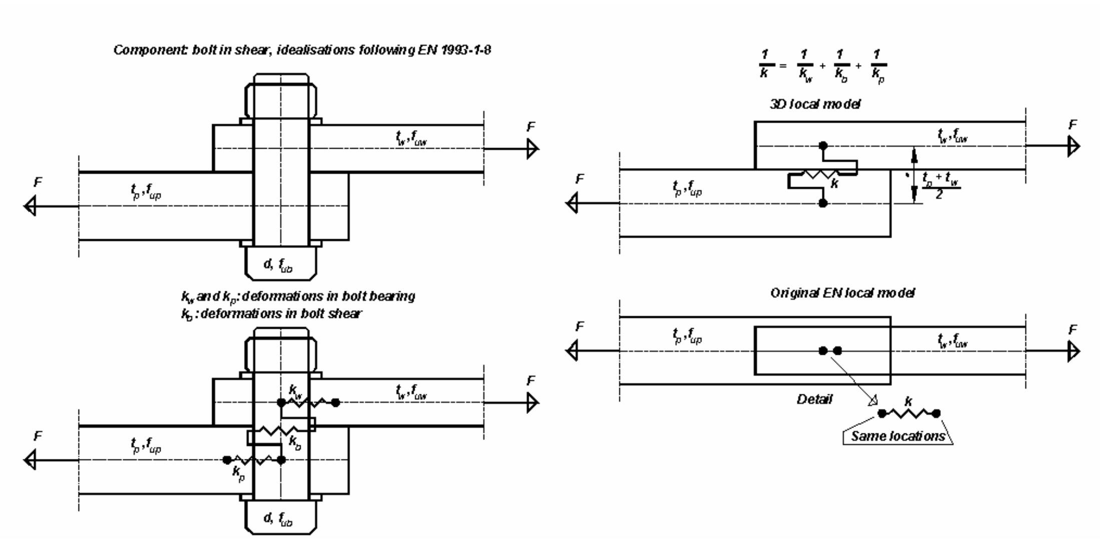

Figure 2. Bolt in shear component construction.

Thicknesses and ultimate tensile strengths of the plates are t_p, f_{up} and t_w, f_{uw}, respectively, where the index p refers to the fin plate and the index w refers to the beam web. The nominal bolt diameter is d and the ultimate tensile strength of the bolt is fub. In Figure 1 the first idealization with the spring stiffnesses k_p, k_b and k_w is given. These are the basic components of the joint. For all these spring stiffnesses are given the equations in the standard [3], as well for their resistances. The stiffness is given in the standard both for non-preloaded and preloaded bolts. If the bolt is preloaded then the stiffness of all the basic components is infinite in this joint.

The three springs are combined to one spring with the spring stiffness k using the basic systems of springs in the line. This stiffness is calculated using the equation:

$$\frac{1}{k} = \frac{1}{k_p} + \frac{1}{k_b} + \frac{1}{k_w} \tag{2}$$

The resistance of this final one spring is the minimum of the resistances of the three springs because the same force acts in all springs. It can be seen in Figure 2 that this idealisation means, that the fin plate location is moved to the mid line of the beam web in the final component model used in 2D. In the final model the spring with the stiffness k is modelled using two points with the same locations and the spring is between these points.

Consider next an example. Consider the joint where one bolt connects two plates as shown in Figure 3. The joint is loaded with a tensile force causing shear to the joint. The results of experiments (Richard Ralph Experiments) for this joint are given in [10]. The results based on a comprehensive finite element method (FEM) model (ABAQUS model) are given in [11]. These results and the results of the component method are given in Figure 3. The dimensions of the joint are rounded from US units to SI units.

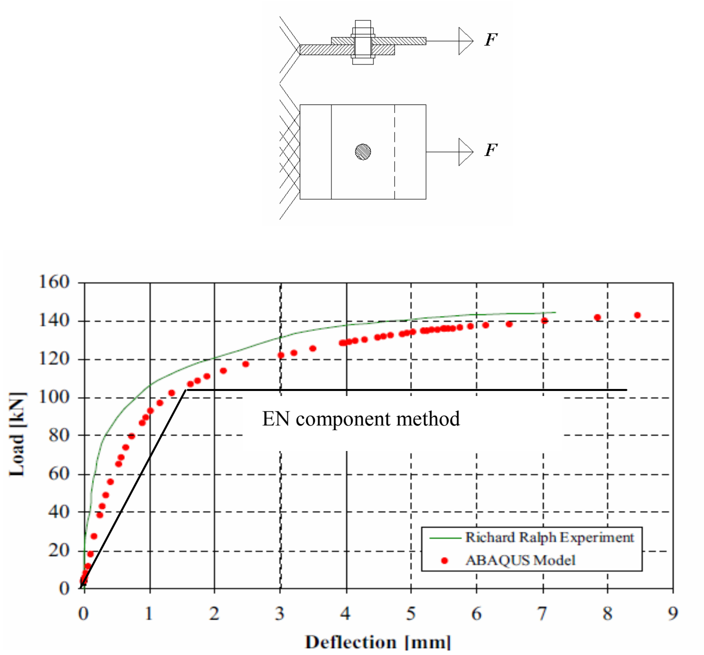

Figure 3. One bolt joint.

The analysis using the component method is as follows. The shear resistance $F_{v.Rd}$ and the shear stiffness k_b of the bolt is [3]:

$$F_{v.Rd} = 0.6 A_s f_{ub} / \gamma_{M2} \qquad (3)$$
$$k_b = d^2 f_{ub} \qquad (4)$$

where in this case (we compare the results to the tests) the material factor is $\gamma_{M2} = 1$, A_s is the stress area of the bolt, f_{ub} is the ultimate tensile strength of the bolt and d is the nominal diameter of the bolt. The factor 0.6 in Eq. (3) is valid for the bolt grade 8.8 and it is supposed that the threads of the bolt are locating at the contact plane of the plates in the joint.

Note 1: In the standard [3] are given the stiffness factors divided by the elastic modulus of the bolt or the plate. Here the spring factors with the units N/mm are used, which are got from the values of the standard by multiplying them with the elastic modulus, that of bolt (E_b = 210000 MPa) in this case.

Note 2: The unit of the stiffness k_b of Eq. (4) is N/mm in this case although Eq. (4) does not imply this. It is supposed that the diameter d is in the unit mm and the tensile strength of the bolt f_{ub} is in the unit MPa.

Note 3: When calculating the stiffness in fire, then the entire value of k_b should be multiplied by the reduction factor of the elastic modulus of the bolt. The ultimate strength f_{ub} is not reduced in fire case in Eq. (4), but is reduced in Eq. (3).

Note 4: Here the stiffness value of the bolt without preload of the bolt is used. If the bolt is preloaded such that slip is prevented at the joint then the stiffness k_b is infinite.

The bearing resistance and the corresponding stiffness is [3]:

$$F_{i.Rd} = k_1\, a_b f_{ui}\, d\, t_i / \gamma_{M2},\ (i = p, w) \qquad (5)$$
$$k_i = 24\, k_x\, k_y\, d\, f_{ui},\ (i = p, w) \qquad (6)$$

where in this case

- $k_1 = \min [2.8\ e_2/d_0 - 1.7\ ;\ 2.5]$ (7)
- $a_b = \min [e_1 / (3\ d_0)\ ;\ f_{ub} / f_{ui}\ ;\ 1]$, $(i = p, w)$ (8)
- but in this case $k_1\ a_b \leq 1.5$ (9)
- $k_x = \min [\min [0.25\ e_1 / d + 0.5\ ;\ 1.25]\ ;\ 1.25]$ (10)
- $k_y = \min [1.5\ t_i / 16\ ;\ 2.5]$, $(i=p, w)$ (11)

and e_2 is the edge distance to the edge parallel with the load, e_1 is the edge distance to the edge perpendicular to the load, f_{ui} is the ultimate tensile strength of the plate (index p) or the web (index w), t_i is the thickness of the plate or the web and d_0 is the hole diameter for the bolt. Notes 1-4 are valid for this case, too.

The following notations can be done when studying Eqs (5) - (6). If the load is opposite to that presented in Figure 3 then the quantity e_1 can be considered large and the resistance and the stiffness of the bolt may be different from that presented in Figure 3. When studying the equations of the standard [3] it can be seen, that the resistance and the stiffness of the bolt is different to that presented in Eqs (5) – (6) if the load is in different angle compared to the edges of the plates than that presented in Figure 3, or if there are more bolts in the joint.

This means, that the resistance and the stiffness of the joint is highly dependant on the direction of the loads acting to the bolts and the layout (number and locations of the bolts) of the joint. This means, further, that the analysis of the joint is extremely difficult, if all the features are taken into account which are possible when applying the standard [3]. However, in mass production and using some computer application all the features can be taken into account and the standard [3] enables this. In daily engineering practice this is not possible and approximations can and should be done. One approximation which is done often is that for the joints including many bolts the minimum resistance and stiffness of all bolts is calculated and it is used for all bolts.

For the example case of Figure 3 the following input values are used:

- A_s = 220 mm2
- d = 19 mm
- f_{ub} = 800 MPa
- f_{up} = 400 MPa
- f_{uw} = 400 MPa
- t_p = 12.7 mm
- t_w = 9.525 mm
- e_1 = 40 mm (estimated)
- e_2 = 60 mm (estimated)
- d_0 = 20.6 mm

These mean the following results

- $F_{v.Rd}$ = 106 kN
- $F_{p.Rd}$ = 145 kN

- $F_{w.Rd}$ = 109 kN
- => $F_{.Rd}$ = 106 kN
- k_b = 288800 N/mm
- k_p = 222699 N/mm
- k_w = 167118 N/mm
- => k = 71780 N/mm

This result is shown in Figure 3 as a bi-linear curve (EN component method). The deflection in the corner is calculated as 106000/71780 = 1.48 mm. The following conclusions can be done based on Figure 3:

- The initial stiffness of the joint is larger than computed. This is due to preload at the bolt in tests, which has been reported in [10] and it was not considered in the analysis.
- The resistance of the joint is larger in tests than computed value. This is due to used nominal values for the strengths in calculations. Measured strengths for the parts of the joint were not reported in [10].
- The resistance calculated using the component method is about 0.75 times the tested resistance.
- The test verified very large deflection of the joint (7 mm) meaning very ductile behaviour, although the bolt shear resistance was critical in the component method.

Consider next the component model for the fin plate joint with many bolts.

Component model for many bolts

Consider next the fin plate joint presented in Figure 4. This joint has been tested in Sheffield University both in ambient and fire conditions [12]. The short column was fixed to the reaction frame and the joint was loaded with the force *P* acting at the end of the beam as shown in Figure 4. It can be seen, that the joint was loaded by the axial and shear force and by the bending moment. The whole structure shown in Figure 4 was located in a furnace.

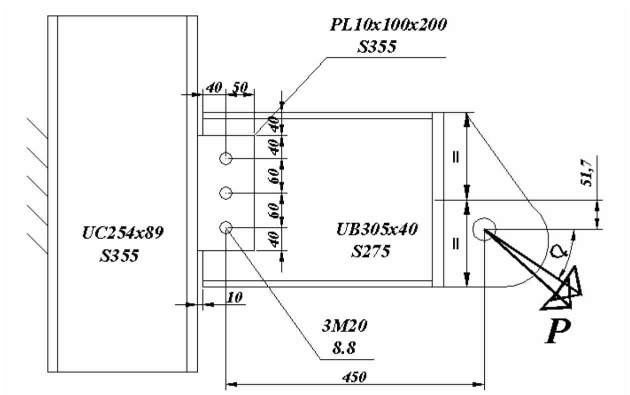

Figure 4. Fin plate with three bolts.

Consider firstly the location of the component model. In the standard [3] the location of the component model is given to the intersection of the mid-lines of the column and the beam. Using this rule the designer should take care that the eccentricity moment of the vertical beam support force is taken into account separately after the analysis, which does not mean rational way to perform the calculations. Moreover, if the column cross-section is large and the beam is short, then the beam span is overestimated in the calculations.

In [9] the location of the component model is proposed at the mid-line of the beam end plate for the end plate joint between the beam and the column. This reduces the problems of previous location for the eccentricity moment and for the beam span. In [13] the function for the location of the component model is given for the end plate joint at the beam end. The component model should locate at the point where the beam analysis model is connected to the local joint analysis model, component model in this case. In the component model all the deformations taking place at the join are capsulated. In [13] was considered a three dimensional component model and the beam analysis model was based on the finite elements including Vlasov's torsion. This means that at beam end node there are seven degrees of freedom and the component model was fixed to that. The location of the component model was in [13] at the point where the beam analysis axis, located at the shear centre of the beam cross-section, meets the surface of the end plate.

In this case the two dimensional case is considered. The beam analysis axis is located at the shear centre of the beam, because the cross-section (UB305x40) is double- symmetric. There are three degrees of freedom at the end node of the beam analysis model. As shown before for one bolt, all the deformations near the bolt were capsulated to one component having the spring stiffness *k*. Here we use the similar techniques for the three bolt joint of Figure 4 and end up to the component model shown in Figure 5.

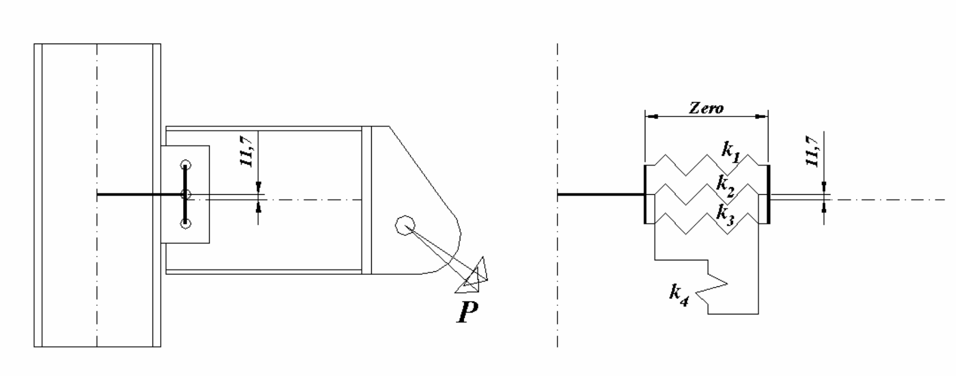

Figure 5. Component model for the fin plate joint.

The component model consists of three rigid links. The horizontal rigid link is connected rigidly to the column analysis node located at the column analysis line. There are two vertical rigid links which have the same locations and they are connected with four springs. The springs with stiffness k_1 - k_3 capsulate the deformations in horizontal direction near the bolt rows 1-3. The spring stiffness k_4 is supposed to infinite in this study. This spring transfers the vertical loads from the beam to the column. In [14] the finite stiffness is taken into account for the vertical force transfer. The spring 4 can be compensated with three springs located at all bolts if more accurate results are required. The horizontal rigid link is connected rigidly to the left hand side of the vertical rigid link. The right hand side of the vertical rigid link is connected rigidly to the beam analysis end node. If needed the local deformations taking place at the column web can be capsulated to this component model, too, as was done in [9]. The same techniques can be used for the fin plate deformations. The component model for the fin plate joint is now ready.

In [2] it is proposed that for the fin plate joint the component model should be located in two locations depending on which quantity is considered. If the welds of the fin plate are considered then the component model should be locates at the bolt line, as above. If the bolts are considered then the component model should be located at the column flange surface. This means that when the shear force is the only joint load, then the bolts get not only vertical but horizontal loads, also, due to the eccentricity moment. In this study it is supposed, that the component model is locating as described above. The column is supposed to be infinitely stiff in this study.

Consider next the behaviour of the component model when it is loaded at the beam end node with the axial and shear force and with the bending moment. The shear forces are transferred with the spring of stiffness k_4 to the column. It was supposed, that there are no interactions between vertical and horizontal loads in the bolt stiffness and resistances. So the shear force of the joint can be divided to three bolts using either elastic or plastic theory. Because the vertical links are rigid, then using elastic theory all bolts get the same shear force. In plastic theory the shear forces are distributed to the bolts using the proportional shear resistances of the bolts. After distributing the forces to the bolts the resistance check of the entire joint against the shear force is done.

If the load of the joint is only shear then only vertical loads are coming to the bolts using this model i.e. no horizontal loads for bolts in this case. When considering e.g. beam to beam joint shown in Figure 1, then typically the rotation of the supporting beam is prevented by the joint. Then the hinge is supposed to be located at the supporting beam and there exist horizontal loads at the bolts due to the eccentricity moment caused by the eccentricity e shown in Figure 1. If the column is tubular (as shown in Figure 1) and very thin walled then it may be wise to prevent the rotation of the fin plate. This means also horizontal loads to the bolts. If the supporting member may deform then it is recommended in [2], that the eccentricity is taken into account both when checking the resistances of bolts and the welds of the fin plate. In our case there are no horizontal loads at the bolts, if the load is shear only. What is the limit for the stiffness of the supporting member so that this assumption is correct is outside the scope of this paper.

In the example case the shear resistances of the bolts for the vertical load are using the equations of [3]:

- Bolt row 1: 118 kN (bolt shear critical)
- Bolt row 2: 85 kN (bearing, web critical)
- Bolt row 3: 118 kN (bolt shear critical)
- Plastic shear resistance of the joint: 118 + 85 + 118 = 321 kN.

Now the bolt shear failure is critical for two bolts, so the use of plastic theory is not suitable. The shear resistance of the joint is 3x85 = 255 kN.

Consider next the stiffness $k_1 - k_3$ and corresponding resistances. There should be some stiffness in the horizontal springs so that the component model is not a mechanism. If there are not springs 1 and 3 then the joint acts as a hinge and all possible axial loads of the beam are transferred through the spring 2. In this case the beam end rotates freely around the spring 2. Note, that there may be vertical eccentricity between the spring 2 and the beam axis, as is the in Figure 5 (11.7 mm). If there are not springs 1 and 2 then the beam end rotates freely around the spring 3 etc. In the case of one bolt in the joint the model works logically. If there is axial load at the beam, and if the joint is not symmetric with respect to the beam axis then there exists the eccentricity moment at the beam end. In one bolt case the joint is a hinge.

If there are two bolts in the joint, then the axial force at the beam end can be distributed to the bolts using the equilibrium equations. The same holds for the bending moment at the beam end. After calculation of the spring forces, the displacements at the springs can be calculated. Using these displacements the vertical displacement and the rotation of the beam end can be calculated at the right hand side vertical rigid link of the component model. By these means the axial and the rotational stiffness at the beam end can be calculated supposing no displacements and rotation at the horizontal rigid link.

If there are three or more bolts in the joint, as is in Figure 5, then the spring system is statically indeterminate and the spring forces cannot be calculated using the equilibrium equations, only. The compatibility condition is needed, too. The unit force method is suitable to solve the spring system mechanics in these cases. In Appendix A the solution for this case is given.

One problem arises, when using the developed component model. Typically steel structures are statically indeterminate as a whole. We do not know in advance which are the loads e.g. for the joints because they depend on the stiffness of the joints. Moreover, the stiffness of the spring may depend on the direction of the load, as given before. The same problem is in the component model presented in [3] as shown in [13] for the base bolt joints. This may mean iterations when completing the global analysis of the structures.

In our case the following spring stiffness and corresponding resistances are got using the equations of [3]:

- Row 1: Tension: $F_{1.Rd}$ = 77.4 kN, k_1 = 64000 N/mm
- Row 1: Compression: $F_{1.Rd}$ = 77.4 kN, k_1 = 74000 N/mm
- Row 2: Tension: $F_{2.Rd}$ = 66.0 kN, k_2 = 64000 N/mm
- Row 2: Compression: $F_{2.Rd}$ = 77.4 kN, k_2 = 74000 N/mm
- Row 3: Tension: $F_{3.Rd}$ = 77.4 kN, k_3 = 64000 N/mm
- Row 3: Compression: $F_{3.Rd}$ = 77.4 kN, k_3 = 74000 N/mm

In this case the equation (9) is active in all cases but for the bolt in the row 2 in tension. The displacement of the component model under axial load and the bending moment is illustrated in Figure 6 supposing large axial force and then all springs are under tension.

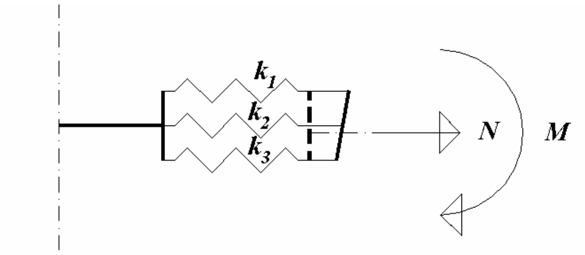

Figure 6. Displacement of the component model if axial force is large.

Consider next the behaviour of the joint in Figure 4 when the load *P* is increasing from zero. It should be noted, that the loads of the joint are not increasing proportionally because the angle α of the load *P* varied between 33.7 – 42.2 degrees in the test. The maximum load *P* reached in the test in ambient temperature was about 185 kN with the angle 42.2 degrees.

Three bolt joint solution using the component method

We start the analysis with the small load P = 10 kN and the angle $\alpha = 36^o$. The loads of the joint are:

$$N = \cos(\alpha) \times P = 8.09 \text{ kN} \quad (12)$$
$$Q = \sin(\alpha) \times P = 5.88 \text{ kN} \quad (13)$$
$$M = 0.450 \times Q - 0{,}0517 \times N = 2.23 \text{ kNm} \quad (14)$$

Firstly, it can be seen that the shear resistance of the joint is not critical in any case ($P \leq 185$ kN). Using the equations of Appendix A the following spring forces and displacements can be got supposing tension for the springs at the rows 1 and 2 ($k_1 = k_2 =$ 64000 N/mm) and compression at the row 3 (k_3 = 74000 N/mm):

- Row 1: Force = 20.14 kN, displacement = 0.315 mm
- Row 2: Force = 3.39 kN, displacement = 0.053 mm
- Row 3: Force = -15.45 kN, displacement = -0.209 mm

and minus means compression at the spring. Now the rotation at the beam end and the corresponding initial rotational stiffness can be calculated:

$$Rotation = (0.315+0.209)/(2\text{x}60)\text{x}1000 = 4.36 \text{ mrad} \quad (15)$$
$$S_{ini} = 2.23/0.00436 = 511 \text{ kNm/rad} \quad (16)$$

We can linearize the problem using the mean stiffness (64000 + 74000)/2 = 69000 N/mm for each spring. Then we get:

- Row 1: Force = 20.49 kN, displacement = 0.297 mm
- Row 2: Force = 2.70 kN, displacement = 0.039 mm
- Row 3: Force = -15.10 kN, displacement = -0.219 mm

and the rotation and the initial stiffness

$$Rotation = (0.297+0.219)/(2x60)x1000 = 4.30 \text{ mrad} \quad (17)$$
$$S_{ini} = 2.23/0.00430 = 519 \text{ kNm/rad} \quad (18)$$

We can see, that the differences in the spring forces and in the initial stiffness are not large when using the linearization in this case.

Using Eq (1) we can estimate the beam span when the joint should be considered as semi-rigid in this case. The span of the beam starts in this case from the point where the component model is connected to the beam model. Using the inertia I_b = 8.505x10^{-5} mm^4 we get using the result of Eq (16) that the beam span should be larger than 17.4 m and then the joint should be considered as semi-rigid. For this beam size this means that practically all the cases can be considered as hinges. However, in some limiting cases the small rotational stiffness may reduce the deflections of the beam.

It can be seen that in this case to row 1 bolt reaches its resistance firstly. The load 10 kN means 20.14 kN load of the bolt which resistance in tension is 77.4 kN. The load can be 77.4x10/20.14 = 38.4 kN and then the row 1 bolt starts to yield and the force at the row 1 bolt remains to the value 77.4 kN. The spring forces and displacements are at this limit load:

- Row 1: Force = 77.4 kN, displacement = 1.210 mm
- Row 2: Force = 13.0 kN, displacement = 0.204 mm
- Row 3: Force = -59.3 kN, displacement = -0.802 mm

The loads of the joint are at the limit case supposing the same angle 36^o for the load:

$$N = \cos(36^o) \text{ x } 38.4 = 31.31 \text{ kN} \quad (19)$$
$$Q = \sin(36^o) \text{ x } 38.4 = 22.75 \text{ kN} \quad (20)$$
$$M = 0.450 \text{ x } Q - 0.0517 \text{ x } N = 8.62 \text{ kNm} \quad (21)$$

The rotation with this moment is 8.62/511 = 16.9 mrad.

When the load P increases from the value 38.7 kN then the component model is shown in Figure 7.

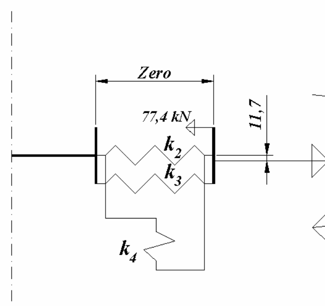

Figure 7. Behaviour of the joint after yield in spring 1.

This system is statically determined and the spring forces can be solved using the equilibrium equations only. The system in Figure 7 is drawn without displacements originating from the previous phase. In this linear analysis it is supposed that these initial displacements have no effect to the equilibrium equations of this new system.

Solving the system of Figure 7 the following results are got:

- Row 2: Force = $M/p + (1-a/p)N - 2\text{x}77.4$ kN (22)
- Row 3: Force = $-(-77.4 \text{ kN} + M/p - (a/p)N)$ (23)

where a is the distance between the spring 2 and the beam axis (Figure 7, a = 11.7 mm) and p is the distance between the bolts (p = 60 mm in this case). The loads N and M of the joint can be calculated as before, but now using the value $\alpha = 39^o$. The load P can increase up to 39.7 kN and the spring forces are then

- Row 2: Force = 30.8 kN
- Row 3: Force = -77.4 kN

and the row 3 spring starts to yield. The moment of the joint at this limit is 9.65 kNm. The increases of the bolt forces and the corresponding increases in the spring displacements are

- Row 2: Force: from 13.0 to 30.8 kN. Displacement: from 0.204 to 0.482 mm
- Row 3: Force: from -59.3 to -77.4 kN. Displacement: from -0.802 to – 1.044 mm

The displacement at the row 1 is increased from 1.210 mm up to 2.016 mm. The rotational stiffness for the system of Figure 7 is 379 kNm/mrad.

After yielding of the row 3 bolt the system is acting as a hinge and starts to rotate around the spring at the row 2. The moment resistance of the joint is at this limit 9.65 kNm, meaning not much increase from the previous limit 8.62 kNm.

The beam end can rotate around the spring 2 until the gap between the beam bottom flange and the column flange is closed. The rotation at this limit is about 67 mrad. After the gap is closed, then the joint behaviour changes totally from the previous. The component model for this phase is shown in Figure 8. It should be noted, that the effect of the closed gap can be taken into account when analysing composite structures in fire following the standard [15].

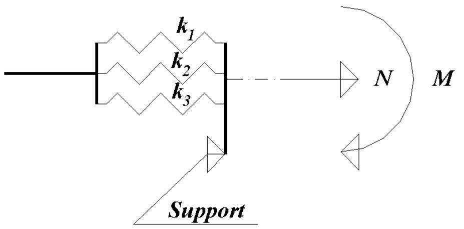

Figure 8. Component model when the gap is closed.

There will be a new force distribution for the bolts at this phase. At the bolt rows 1 and 2 the load level is now lower and the yielding stops there. It is supposed, that when the forces at the springs 1 and 2 decrease after yield no displacements happen in springs and springs continue to behave as elastic up to the next yield. At the bolt row 3 the compression changes to tension. It is supposed, that no stiffness is at the spring 3 until the bolt is in contact at the tension side. The behavior of the joint is illustrated in Figure 9 at different phases when the load is increasing. It can be seen in the last situation, when all the springs yield very large displacements are especially at the spring 1. In the test the structure was enough ductile to reach these displacements.

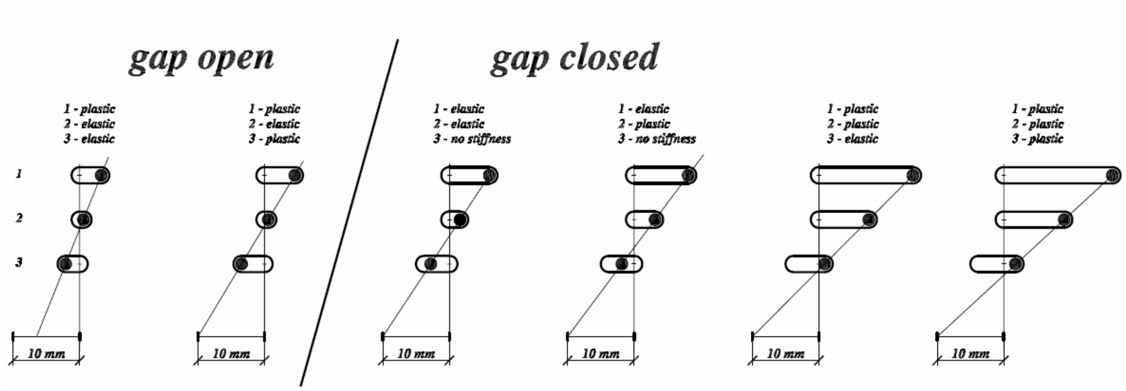

Figure 9. Joint behaviour in different phases.

The calculations are done in [1] for the joint in Figure 4. The moment-rotation curve for all the phases when the load P increases from zero up to the value 185 kN is shown in Figure 10. The maximum moment in the test in ambient conditions was 48.6 kNm which is much larger than using the component method 24.2 kNm. In Figure 10 the test results and the calculated results in ambient conditions (20 deg) and at elevated temperatures are presented.

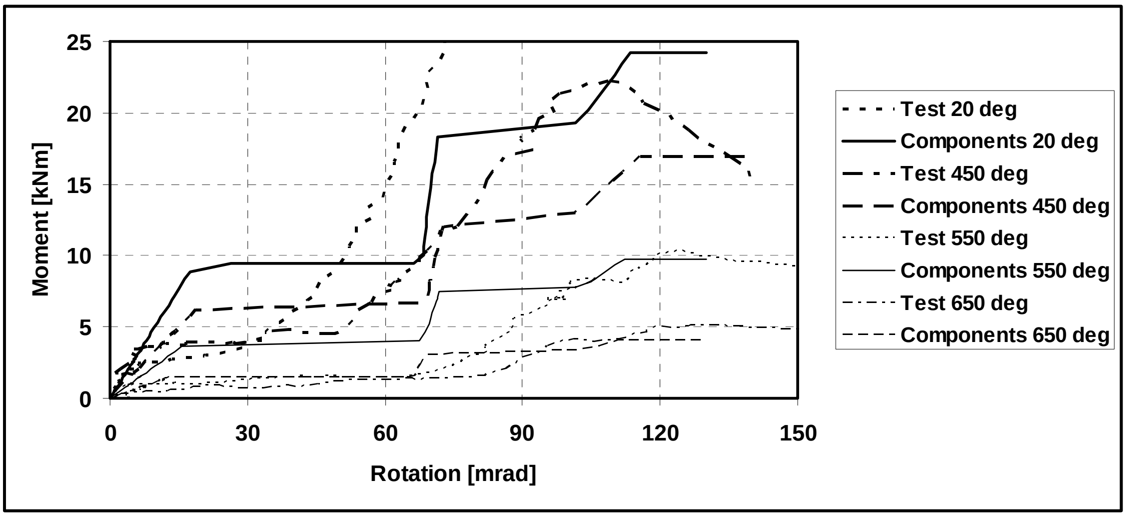

Figure 10. Moment-rotation diagram for the fin plate joint in all phases.

It can be seen that there are big differences between the calculated and the measured results. Sources for the differences are such as:

- Preloads of bolts were not taken into account in the calculations. The possible preloads are not reported in the test report.
- The gaps in the bolt holes were not taken into account in the calculations.

– Nominal strength values were used in the calculations. Measured values were not reported in the referred test report.

The moment resistance in fire can be calculated using the reduction factors of bolt strengths given in [4]. In Table 1 the calculated plastic moment resistances are given using the model of Figure 8 and the corresponding test results.

Table 1. Moment resistances at elevated temperatures.

Temperature [^{o}C]	Moment resistance [kNm] Component method	Maximum measured moment [kNm]
20	24.2	48.6
450	17.0	22.2
550	9.7	10.3
650	4.0	5.2

It can be seen, that the results of the component method predict very accurately the moment resistance of the joint in fire. In ambient conditions the result of the component method is very conservative.

Fin plate joint in ambient conditions

The last example deals with the fin plate joint presented in Figure 11. The dimensions of the joint are rounded from US units to SI units also in this case.

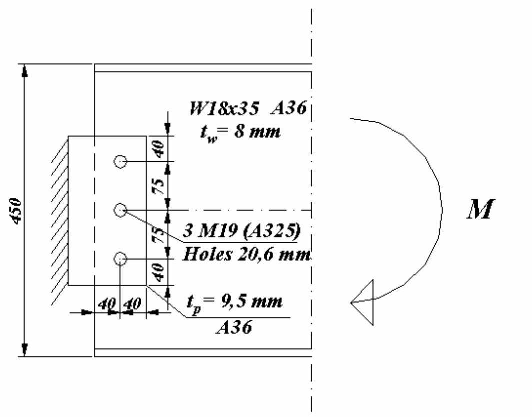

Figure 11. Three bolt fin plate joint [11].

The results for this joint are given in [11] using the comprehensive finite element model (ABAQUS Model) and based on tests (Richard Experiments) originating from [10]. This case is solved using the linearization proposed above. This means that we use the mean of spring stiffness for horizontal springs. Only moment load is considered. The joint is symmetric with respect to the beam axis. Then the analysis of the joint is rather easy.

The material properties for the beam, plate and bolts are the same as in one bolt case before. The resistance against horizontal loads of the top and bottom row bolt is

1.5x400x8x19/1000 = 91.2 kN. The moment resistance of the joint is 0.15x91.2 = 13.7 kNm. The mean stiffness of the springs against the horizontal loads is k = (60272+70231)/2 = 65251 N/mm. The initial rotational stiffness is (h is the distance between the top row and the bottom row bolt centers)

$$S_{ini} = h^2 k / 2 = 0.15^2 \text{ x } 65251 / 2 = 734 \text{ kNm/rad} \tag{24}$$

The rotation when the moment is 13.7 kNm is 13.7/734 = 0.0186 rad = (180/π) x 0.0186 = 1.07 Degrees. This result (Components without preload) and the result with preloads at the bolt (Component with preload) are given in Figure 12 together with the tests results (Richard Experiment) and FEM results (ABAQUS Model).

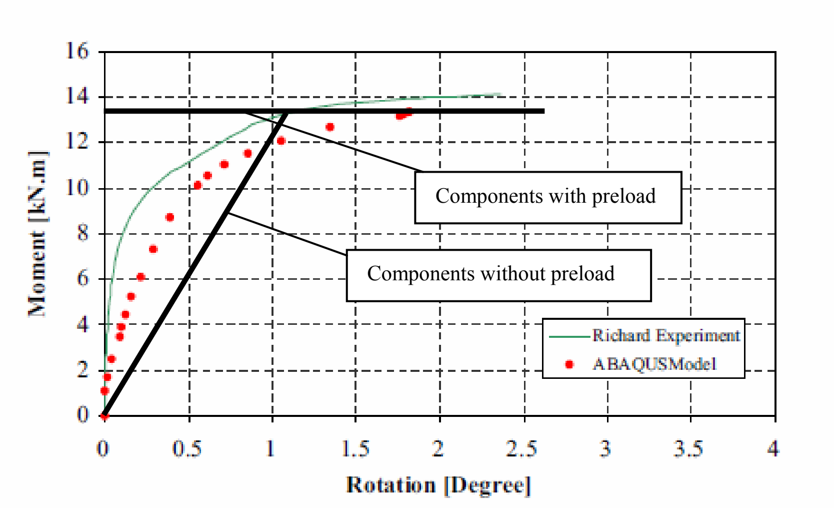

Figure 12. Moment-rotation curves for three bolt fin plate joint.

It can be seen in Figure 12 that the initial stiffness using the component method and non-preloaded bolts is smaller than got from tests and FEM. The test results and the FEM results are between the two lines of the component method. The moment resistance using the component method is slightly on the safe side in this case. This means that the component to describe the stiffness of the preloads in bolts is needed to predict the stiffness of the joint better than the using the two limit cases: with or without preloads appearing now in the standard. However, using the assumption of non-preloads the stiffness is on the safe side.

Conclusions

The component method for the fin plate joint was proposed in the paper. The results using the proposed component method give safe results for the moment resistance of the joint both in ambient and fire conditions in the cases considered in the paper. The comparisons were done against the tests and the comprehensive FEM results available in the literature. The stiffness of the joint did not match very well to the tests and FEM results being much lower than in the tests if non-preload in the bolts was assumed. Also the effects of clearances in bolt holes were not taken into account in the component method.

The appearance of the horizontal loads at the bolts from the shear force of the joint remains open in this study. Their appearances depend on the stiffness of the supporting member as explained in [2]. Until this has been solved it is recommended that the resistance checks of the joint are done following [2]. The proposed component model can be used when analyzing the joint, and the resistance checks are done following [2], meaning that the horizontal loads from the shear force are calculated as in [2].

Numerical results show, that the fin plate joint can be considered as a hinge in practical cases. The computed rotational stiffness was so small, that using the criteria of the standard [3] the joint could be considered as a hinge in practical cases. It should be noted that the computed rotational stiffness was too low compared to the tests, so the situation in real structures may be different. The small rotational stiffness of the joint may be important when considering the deflections of the beams in practice.

The tests showed large moment potential in the joint with large rotations when the gap between the beam bottom flange and the column face was closed. The resistance using the component method was on the safe side in this case. This feature of the joint is essential and it can be taken into account in the analysis, especially in fire cases as proposed in [15].

The component method is very generic and can be used to the analysis of many kinds of joints. It suits well to the automatized computer aided design because it can be constructed based on the geometrical entities of the joint. The component method leads to the non-linear global analysis, but it can be linearized as proposed in the paper. This linearization reduces the computing time of the global system, if needed. The linearization may be done in the preliminary analysis. The final analysis should be done using the non-linear analysis to get proper, safe result, because the linearization leads in some design quantities to safe and in some quantities to unsafe results. In the joints considered the differences in resistances and stiffness were not large whether using the non-linear or the linear component model.

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Appendix

Consider the statically indeterminate three spring system shown in Figure A1. Use the spring force F_2 (tension is positive for spring forces) as a parameter X. Then the statically determined structure is as shown in Figure A1. This structure is loaded by the loads N and M of the original structure and by the unit load $X = 1$ as shown in Figure A1. The corresponding displacements are δ_{10} and δ_{11}.

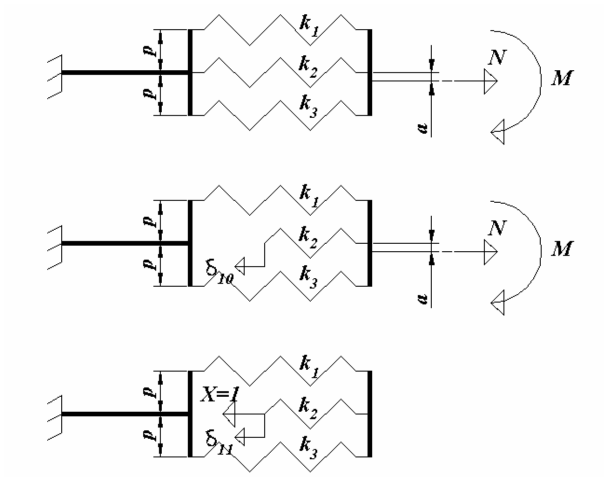

Figure A1. The original structure and statically determined structures.

The spring forces F_{10} and F_{30} are from the original loads and the spring forces F_{11} and F_{31} are from the unit load. The compability condition is $\delta_{10} + X\,\delta_{11} = 0$, which yields:

$$F_2 = X = -\frac{\delta_{10}}{\delta_{11}} \Rightarrow F_1 = F_{10} + X \cdot F_{11}; \quad F_3 = F_{30} + X \cdot F_{31} \qquad \text{(A1)}$$

The equilibrium equations yield:

$$F_{10} = \frac{p-a}{2 \cdot p} \cdot N + \frac{M}{2 \cdot p} \qquad \text{(A2)}$$

$$F_{30} = \frac{p+a}{2 \cdot p} \cdot N - \frac{M}{2 \cdot p} \qquad \text{(A3)}$$

$$F_{11} = F_{31} = -\frac{1}{2} \qquad \text{(A4)}$$

The displacements are:

$$\delta_{10} = -\frac{\frac{F_{10}}{k_1} + \frac{F_{30}}{k_3}}{2} = -\left(\frac{F_{10}}{2 \cdot k_1} + \frac{F_{30}}{2 \cdot k_3}\right) \qquad \text{(A5)}$$

$$\delta_{11} = \frac{\frac{1}{2 \cdot k_1} + \frac{1}{2 \cdot k_3}}{2} + \frac{1}{k_2} = \frac{1}{4 \cdot k_1} + \frac{1}{4 \cdot k_3} + \frac{1}{k_2} \qquad \text{(A6)}$$

After the spring forces F_1, F_2 and F_3 are calculated then the corresponding displacements at the springs can be calculated. From these the rotation and the axial displacement at the loading point can be calculated.

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