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DESIGNING OF STRESS RELIEVE GROOVE PARAMETERS FOR FLAT ENDPLATES OF PRESSURE BOILERS

Bogdan Szybiński, Marek Barski, Mateusz Wygoda

Cracow University of Technology
Department of Mechanical Engineering
Jana Pawła II Avenue 37, 31-864 Krakow, Poland
tel.: + 48 12 3743388, + 48 12 3743389, fax: + 48 12 3743360
e-mail: boszyb@mech.pk.edu.pl, e-mail: mbar@mech.pk.edu.pl
comfort1991@interia.eu

Abstract

The application of flat ends as closures in pressure boilers is a certain alternative for commonly used dished ends. However, it is inevitably joined with the stress concentration presence in the vicinity of the junction between the cylindrical pipe and the ending flat plate. The notch effect, which is observed in the connection area, is usually caused by the abrupt change of the inner wall shape and the lack of the curvature continuity in this region. The problem of the stress reduction in the transition zone between the cylindrical pipe and the endplate has been studied by researchers from many years and several proposals how to modify the shape of the respective elements have been established. On the base of these results, several designs for flat end closures are proposed in the codes applied in pressure boilers calculations. This paper shows the effectiveness of one of the design proposals, admitted by code EN 13445-3. In the proposed analysis the special attention are paid to flat ending plates with the circular stress relieve grooves. The parameters of the groove radii and location of its centre are set by a system of inequalities, which usually defines the region of admissible values for radii and position of the circular groove centre. The choice of these values strongly influences the stress concentration and in the considered code, no clear suggestions or justification is given how to choose the best groove parameters, providing the minimum stress concentration. In this report, the Authors show how to choose the best combination of the design parameters providing the minimum stress concentration.

Keywords: *pressure boilers, flat ends, stress concentration, FEM analysis*

1. Introduction

Welded connections of a pressure boiler cylinder and a head of the boiler are studied since many years due to the observed presence of stress concentration in the respective connection [4, 8, 10, 12, 14]. The proper assessment of the notch effect is possible only by numerical methods, as the analytical solutions do not exist. The minimization of the stress peak mainly relies on introduction of such heads, which shape provides the continuity of the curvature on inner skin of the vessel. However, this approach is still not commonly applied in the construction of boiler heads and still the ellipsoidal heads are in major use. The sizes of these heads are limited to certain range of diameters and pressures, and in nonstandard uses, particularly for higher pressures and bigger thickness flat welded ends can be a rational alternative. These heads are less complex in shape and much cheaper in fabrication. The main disadvantage of such designs is the presence of the severe stress concentration in the junction of the flat plate edge and the shell wall appearing inside the boiler. In order to minimize the notch effect the stress relieve groove in the form of a cut-out in the flat end plate is applied in the junction area. This provides more smooth shape transition in the shell-endplate connection. Also in this case the lack of the curvature continuity observed in the area when the inner edge of the cylindrical wall develops to the curved one still results in the peak of the stress.

The commonly used design codes (EN 12952:3 [1], EN 13445-3 [2]) suggests the application of the circular shape for the stress relief groove edge. However, the design of the junction vicinity is slightly different in both addressed codes. The main differences in both designs show the Fig. 1. As it can be seen, the change of the topology relies on introduction of the certain shell wall reinforcement of the cylindrical part in the vicinity of the shell-the endplate connection. These results in different admissible groove parameters calculated according to the respective formulas given in both codes. Additionally, the formulas for the calculation of the endplate thickness are different too. These are as follow:

$$e_h = C_1 \cdot C_2 \cdot C_3 \cdot d_i \cdot \sqrt{\frac{p_c}{f}}, \tag{1}$$

for the code EN 12952:3. Here, constants C_2 and C_3 take the value of 1.0 when the boiler with circular cross-section and endplate without opening is under construction, constant C1 depends on the ratio between applied pressure pc and the admissible stress f, and C_1 value varies between 0.41 and 0.82, and is taken form the plot included in the code. In the code EN 13445-3 the endplate thickness is calculated as follow:

$$e_h = \max\left\{ C_1 d_i \cdot \sqrt{\frac{p_c}{f}}, \quad C_2 d_i \cdot \sqrt{\frac{p_c}{f_{\min}}} \right\}. \tag{2}$$

Here the d_i is the inner boiler diameter measured in the zone where the wall thickness is equal to e_s (see Fig. 1b). The admissible stress f equals to $f_{\min}$ if the material chosen for the cylindrical and flat parts of the boiler are the same and the p_c is, the internal pressure. The constants C_1 and C_2 are taken from the respective plots presented in the code EN 13445-3.

It is also worth mentioning that the formulas for calculation of the cylindrical wall thickness for a given inner pressure are different for both codes. The first one –EN 12952:3 – uses the Tresca-Guest formulae for the equivalent stress calculation. On the base of such approach, the cylindrical wall thickness is derived, while the second norm – EN 13445-3 applies the Huber – Mises – Hencky criterion for the equivalent stress calculations, which is used further in the wall thickness calculations for cylindrical part. This difference leads to a certain, small differences in the wall thickness in these two methods, depending on the combination of the applied internal pressure p_c and the inner diameter d_i of the studied boiler. The differences of these two designs are shown in Fig. 1.

In case of using EN 12952:3 the set of following expressions (3) is used for calculations of radius r_{ik} and the minimum end-plate thickness in the groove e_{h1}.

$$\begin{cases} e_{h1} \geq e_s, \\ e_{h1} + r_{ik} \leq e_h, \\ r_{ik} \geq \max\{0.2 \cdot e_s, \quad 5\,\text{mm}\}, \\ e_{h1} \geq 1.3\left(\dfrac{d_i}{2} - r_{ik}\right) \cdot \dfrac{p_c}{f}. \end{cases} \tag{3}$$

The reported set of inequalities gives certain ranges for the groove radius r_{ik} and the minimum end-plate thickness e_{h1}. Here e_s is the shell or pipe wall thickness and d_i stands for the inner boiler diameter, p_c is the internal pressure and f stands for the admissible stress. Here the searched values usually cover certain polygonal area. While the code EN 13445-3 is used than the set of the respective conditions is different.

$$\begin{cases} e_{h1} \geq e_s, \\ e_{h1} + r_d \leq e_h, \\ r_d \geq \max\{0.25 \cdot e_s, \quad 5\,\text{mm}\}. \end{cases} \tag{4}$$

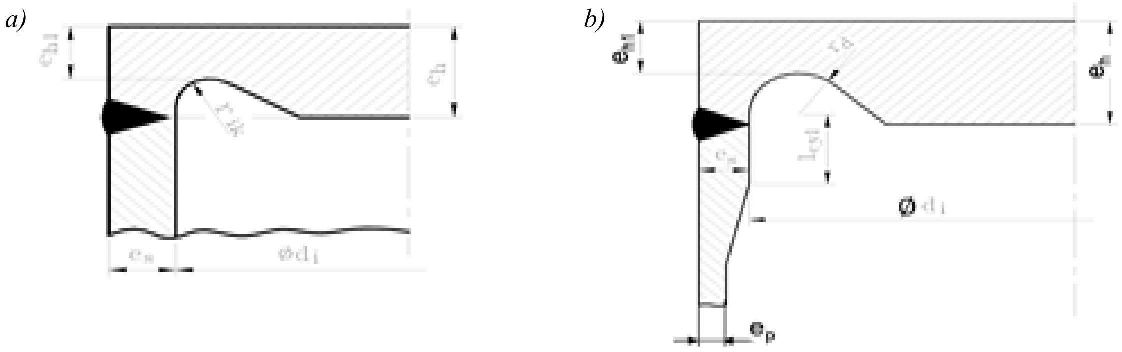

Fig. 1. Two recommended designs for boilers with flat ends with stress relief grooves: a) EN 12952:3 code, b) EN 13445:3 code

The main difference with former approach relies on the introduction of locally increased the wall thickness e_s instead of the calculated wall thickness e_p. Additionally the fourth condition presented in the set (3) is not included now and no other equivalent expression is included. Besides of this, the additional formulae for the l_{cyl} – the length of the reinforced cylindrical part – is introduced in the form as follow:

$$l_{cyl} = \sqrt{(d_i + e_s) \cdot e_s} \, . \tag{5}$$

The code EN 13445-3 also recommends that the transition zone in the cylindrical part, in which the wall thickness increases from e_p to e_s should be smooth with the inclination angle not bigger than 30°. Unfortunately, no clear suggestion is given in the code how to assume the thickness e_s in the thickened area. This gives certain freedom in the designing process and opens the possibility of the search of the optimal combination of the investigated structure parameters (dimensions) in the junction providing the minimum stress concentration. The set of inequalities presented in (4) results in a triangular area in r_{ik} and e_{h1} coordinates. This area limits the values for the lowest thickness in the groove and the radius of the groove. The up to date experiences shows that the inclination angle of the groove shown on the vertical axle side has no real influence on the stress concentration when its value exceeds 60°, so that all numerical calculations presented below assume the inclination angle value of 90°. In such a case, the problem of the search of optimal parameters reduces to investigations of only two design variables when using the EN 12952:3 code. These are the minimum thickness of the endplate measured at the bottom of the groove – e_{h1} and the radius of the groove – r_{ik} (see Fig 1a). In case of using EN 13445-3 code the additional, third design variable appears, which the thickness es (see Fig. 1b) is. The stated problem of the search of optimal design variables, which provide the minimum stress concentration, can be regarded as a parametric optimisation, in which the optimal vector of design variables is obtained on the base of analysis of the sequence of solutions obtained for different combinations of design variables. Different optimization criteria and optimization tools can be applied in search of the optimal groove parameters. The most common approach in the structural problems is the definition of the objective function as the minimization of the maximum value of the stress concentration factor, which is expressed as follow:

$$F_s = \min\left\{\max \frac{\sigma_{eqv}}{R_e}\right\}, \tag{6}$$

here R_e is the yield limit, while σ_{eqv} defines the equivalent stress following the von Mises or Tresca-Guest hypothesis. This formula is fully justified for the structures, which exhibit only elastic response to the applied load, but usually fails in case of structures where elastic-plastic deformation may appear. In such situation, the minimization of the maximum value of the equivalent plastic strain seems to be more convenient and efficient:

$$F_e = \min\{\max \varepsilon_{pl\_eqv}\}, \tag{7}$$

here, ε_{pl_eqv} is the equivalent plastic strain.

In case of the optimization, process including only two design variables the simple search method in the admissible design space is one of the solution methods. This approach enables to study the distribution of the objective function over the whole domain and prevents to get stuck the solution in the local minimum. The above approach can also be used in case of three design variables, as in the case of EN 13445-3 code. Here, the assumed value of the shell thickness e_s determines the area of search in the sense of the inequalities given in (4). So that the bounding values for e_{h1} and r_d become different for different values of e_s. This means that the simple search method can be done for the established value of e_s. In the numerical tests it was observed that the increase of the e_s reduces the value of the inner diameter of the cylindrical part d_i, which inevitably causes the reduction of the endplate thickness e_h, and reduces the limits of admissible changes for the groove radius and the minimum thickness at the groove.

2. Numerical model and results of FEM calculations

The analytical solution of the cylindrical shell – flat end-plate junction subjected to internal pressure does not exist at all, so that only certain assessments with various simplifying assumptions can be analytically done [6, 10]. This inconvenience is omitted by application of the finite element software. In the investigated problem the finite element code ANSYS [1] is applied. In this code various structural problems, including material and geometrical nonlinearities can be solved. In case of axisymmetric structures, certain reduction of the calculation time can be achieved by using the axisymmetric model, which helps to reduce the analysed problem of the boiler with two flat end plates to the symmetric quadrant of the axial cross-section of the whole structure. Additionally to keep the solution error as low as possible in the area where the severe stress concentration appears, the transition zone of small regular elements around the curved edge was introduced. The generation of the finite elements in the area neighbouring the curve edge of the groove was organized as automatically adjusted, which provides the proper mesh refinement and the control of the solution quality. Such approach for finite element mesh generation helped to keep the solution error in stress energy below 2.5% in each of the analysed examples, which was fully acceptable from the engineering point of view. The one of the used meshes shows the Fig. 2 with the groove, curved area magnified in the left bottom quadrant.

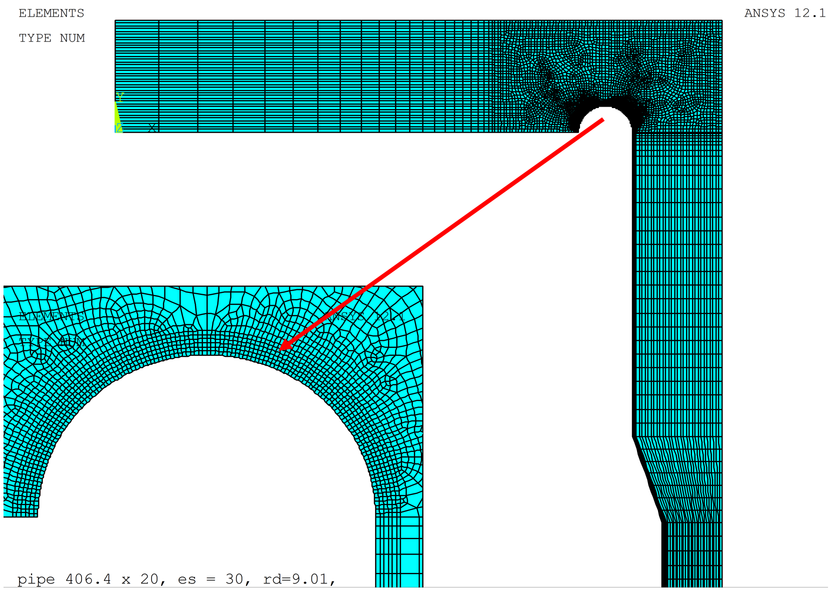

Fig. 2. Exemplary mesh used in numerical calculations

Two structures were under numerical investigations. The first one was made by means of welding form the steel 16Mo3, which is the low alloy steel widely used in construction of pressure appliances due to its good mechanical properties. The second boiler was fabricated from the S235JR low carbon, weldeable steel, which is also used for vessels manufacturing. Both materials exhibit the elastic-plastic properties and their stress-strain curves were approximated by models introduced on the base of several results of the mechanical tension tests. Here, the numerical results for boiler made from S235JR are presented. The region of plastic deformation can be modelled as nonlinear one, but to stay on the safe side the linear hardening in the region of plastic deformations was used. In the proposed approximation the Young modulus was assumed to be equal: $E = 2.0\times10^5$ MPa, while hardening module: $E_t = 570$ MPa, and elastic limit $R_e = 225$ MPa and tensile strength $R_m = 360$ MPa, and the maximum strain $\varepsilon_{max} = 0.26$.

The presented detailed results were obtained for cylindrical shell following the shape presented in EN 13445-3 with the outside diameter Ø406.4 and the basic wall thickness $e_p = 20$ mm. For these values, the maximum operating pressure was calculated: $p_{int} = 15.597$ MPa. In the study the thickness e_s for the reinforced part of the cylinder, was changing from 20.0 mm to 30 mm with a step of 1 mm. It appeared that, for $e_s = 31$ mm the admissible area for the minimum thickness at the groove bottom and the radius of the groove became empty – there were no admissible values of the design variables fulfilling the set of inequalities (4). The Tab. 1 shows the sets of variables used in calculations depending on the increased thickness e_s. Fig. 3 shows the reduction of the admissible areas of search for e_{h1}, r_d due to the increase of the cylindrical shell thickness.

Tab. 1 Design parameters: e_s, d_i, l_{cyl}, e_h and design variables (e_{h1},r_d) range

e_s	d_i	l_{cyl}	e_h	r_d		e_{h1}	
				min	max	min	max
20.00	366.40	0.00	63.22	5.00	42.14	20.00	58.22
21.00	364.40	89.96	56.28	5.25	35.28	21.00	43.99
22.00	362.40	91.96	46.64	5.50	24.64	22.00	39.49
23.00	360.40	93.91	44.06	5.75	21.06	23.00	38.20
24.00	358.40	95.80	43.24	6.00	19.24	24.00	37.36
25.00	356.40	97.65	42.43	6.25	17.43	25.00	36.31
26.00	354.40	99.45	41.62	6.50	15.62	26.00	35.58
27.00	352.40	101.21	40.82	6.75	13.82	27.00	34.41
28.00	350.40	102.93	40.25	7.00	12.25	28.00	33.36
29.00	348.40	104.62	39.79	7.25	10.79	29.00	32.43
30.00	346.40	106.26	39.01	7.50	9.01	30.00	31.51
31.00	344.40	107.87	38.23	7.75	–	31.00	–

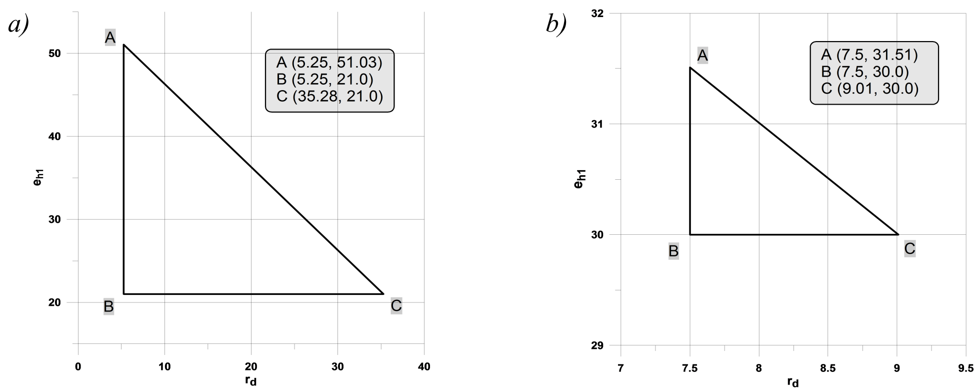

Fig. 3. Admissible area of search for optimal parameters for radius of groove and minimum endplate thickness in groove: a) e_s = 21 mm, b) e_s = 30 mm

The optimization process and details of numerical calculations were presented in papers [12, 14] As a results for all analysed cases, using both the codes [1, 2] the optimal values for the groove radii and the minimum thickness of the end-plate appeared on the border line AC of the admissible area, where: $e_{h1} + r_d = e_h$. Depending on the shell thickness, the optimum points may move toward the maximum admissible value of the groove radius (point C). In case of using the code EN 13445-3 if only e_s is greater than e_p, than the optimal point locates in the corner of the admitted area, which corresponds to the maximum admissible value of the groove radius (point C). The exemplary results for optimal configurations of the groove radii and the minimum thickness of the endplate, obtained for two different values of e_s, namely 21.0 mm and 30.0 mm, are presented in Fig. 4 and 5. These are the equivalent stress and equivalent plastic strains distributions in the vicinity of endplate-shell junction. It can be seen that the maximum equivalent stress value is much less sensitive than the maximum equivalent strain value with respect to the changes of e_s. This justifies the application of the objective function in the form given in (7) rather than this shown in (6). The dependency between the maximum equivalent plastic strain obtained for the optimal parameters of the groove radii and the minimum endplate thickness presents the Fig. 6.

a) b)

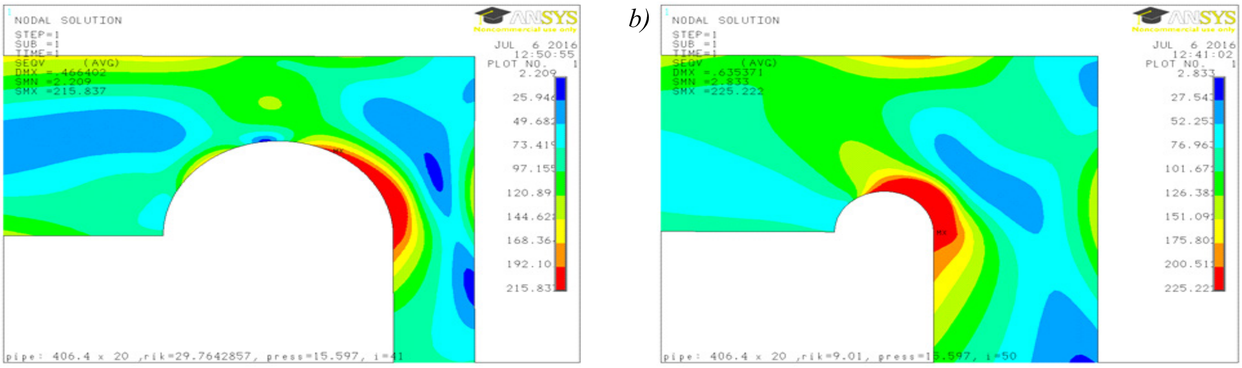

Fig. 4. Distribution of equivalent stresses for optimal parameters for radius of groove and minimum endplate thickness in groove: a) e_s = 21 mm, b) e_s = 30 mm

a) b)

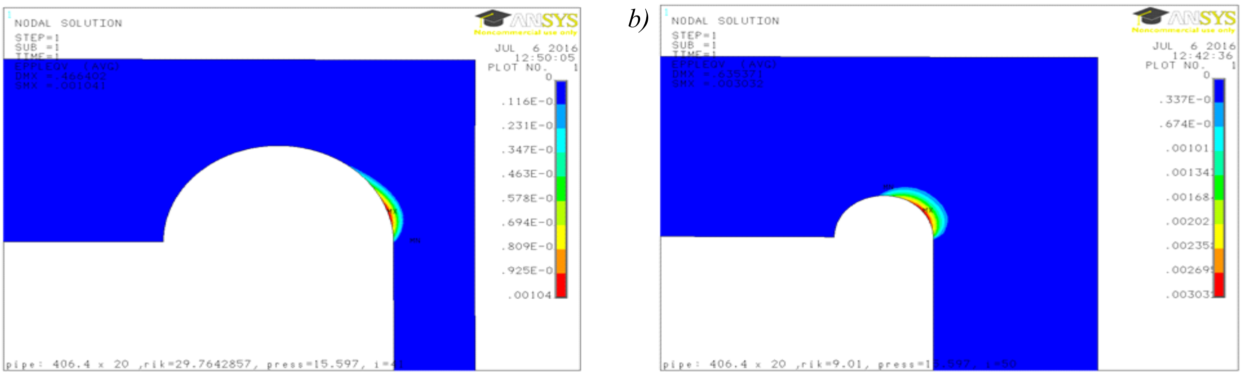

Fig. 5. Distribution of equivalent plastic strains for optimal parameters for radius of groove and minimum endplate thickness in groove: a) e_s = 21 mm, b) e_s = 30 mm

3. Conclusions

The performed study has shown that the minimum value of the plastic strains (the minimum stress concentration) is obtained for the shell with no local reinforcement applied, namely $e_p = e_s$. This means that the proposed in the code EN 13445-3 local reinforcement of the pipe wall does not bring any profits. When using both calculation codes the plastic deformations are still present in the groove area, which are usually unacceptable in problems where the cyclic loadings appear. In such a case, the maximum applied inner pressure should be properly lowered in order to avoid

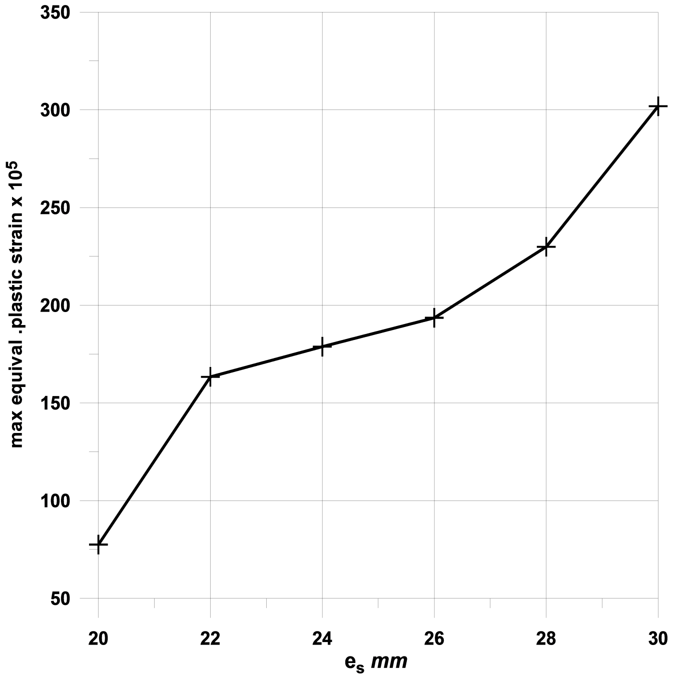

Fig. 6. Maximum equivalent strain distribution with respect to thickness of reinforced part of cylindrical shell e_s

the presence of plastic deformations and their possible development in the following load cycles. The additional study with different shapes for stress relieve grooves should be performed, this can provide further reduction of stress concentrations as it was observed in structures analysed in papers [5, 9].

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