

**THE FENET PLATE/SHELL FABRICATION PROCEDURAL
BENCHMARKS AND 'ROUND-ROBIN' EXERCISE**

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SUMMARY

Plate and shell construction is common across many industrial sectors and covers components and structures that range from the relatively unimportant to safety-critical. The details used in plate/shell structures, in any industry sector, are no doubt a reflection of tradition, as well as market forces and regulation. As a result, for example, full penetration butt-welds will be more common in the nuclear industry, while fillet weld details will be more common in many 'every-day' fabricated structures, ranging from lamp posts to 'bin' lorries.

As finite element technology has moved from the so-called 'right-first-time' sectors into general industry, today's powerful analysis and simulation technology is being adopted by more and more organisations, including SMEs, which generally do not have an 'analysis tradition'. In addition, coverage of the assumptions inherent in shell theory generally falls into the postgraduate educational domain. The staffing challenges facing SMEs in particular in this area are therefore significant. Furthermore, it is also argued that many of the details commonly found in fabricated plate/shell structures are often not subjected to widely recognised and commonly accepted cross-industry analysis procedures. The procedural benchmarks and 'round-robin' exercise, detailed herein, were seen as an excellent opportunity to examine such practice and to observe resulting educational and quality assurance related issues.

Observations from the 'round robin' provide some surprising results. In the first two benchmarks about half of the respondents supplied results which suggested that they had made modelling errors. In the third example, only two out of ten respondents realised that this was a nonlinear geometric problem. Whilst some contributions were no-doubt completed under time pressure, it can be argued that this is a reflection of the everyday industrial environment for many engineers. The resulting levels of human error and lack of results checking, for what some might regard as simple case studies, must be of wider interest and concern. The general spread of results arising from the different modelling and assessment strategies should also be of interest. The outcomes certainly confirm the ongoing role that organisations such as NAFEMS have, in ensuring quality and promoting the education and development of analysts and engineers.

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1 Introduction

The procedural benchmarks defined in section 2, formed the basis of a voluntary 'round-robin' exercise, in which participants were encouraged to apply their normal industry modelling practices. The aim of the exercise was to identify, if possible, elements of best practice and to disseminate this across the various industry sectors. It was also recognised that the results from such exercises generally provide useful fruit for debate and discussion in matters related to education and quality assurance.

The aim of the benchmarks and the 'round-robin' exercise reported in the following sections was not to try to promote higher quality (and invariably higher cost) detail, but to recognise the details in common use and to see whether a round-robin exercise involving particular geometry, could identify 'best practice' or perhaps highlight modelling and assessment deficiencies. Neither was the aim to present the most comprehensive analysis possible of each detail, but rather to use 'industry typical' procedures, sufficient for the determination of preliminary scantlings and to highlight the issues involved in such approaches. However, more sophisticated/detailed non plate/shell models invariably proved useful as a reference, in highlighting the limitations of simpler approaches. The main fabrication details selected for examination were:

- The modelling and assessment of intersections;
- The modelling of reinforcement/cover/wrapper plates;
- The modelling of offset shell midsurfaces.

The rationale behind this selection is discussed in the following section. All of the details included welds and the modelling and fatigue assessment of these was an integral part of the whole exercise, although not the *raison d'être*. The results were assessed according to various industry standards, where appropriate.

The three 'procedural benchmarks' were developed to reflect the above modelling issues. These benchmarks and selected reference solutions (detailed in reference[1]), should also provide a worthwhile educational resource. While these have many of the characteristics of traditional benchmarks, they differ in that they were designed to focus on the modelling issues that analysts are faced with and the various *procedures* adopted in the analysis and assessment process.

2 Procedural Benchmarks Outline

At the outset, the following 'characteristics' were identified as being desirable for the definition of procedural benchmarks:

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- The problem definition should be as simple as possible, whilst still capturing the essence of the modelling and assessment challenges.
- The problem, as posed, should be as relevant to as many industry sectors as possible, without compromising the essence of the modelling and assessment challenges.
- The benchmarks developed should embrace the most common and relevant element types. The assessment should address both static strength and fatigue.
- The problems posed should avoid the need for explicit modelling of material non-linearity, although reference could be made to more rigorous modelling and analysis scenarios for comparative purposes.
- Given that the modelling approach adopted is invariably linked to the purpose of the analysis, which in many cases is inherently linked to Codes of Practice and allowables therein, it was recognized that the benchmarks should therefore make reference to the form of structural assessment being used.
- The benchmarks should specify solution quantities that could in turn be used in various assessment strategies. Simple “check” values, that may not be part of any engineering assessment, would also be useful for comparing solutions.

Models with plate/shell elements will obviously reflect the approximations and assumptions associated with the various plate/shell theories, including mid-surface representation of geometry, linear through-thickness stress and others, dependent upon the plate/shell theory inherent in the element formulation. **Many of the challenges presented by plate/shell structural detail, will fall into one or more of the following categories:**

1. Intersections with slope continuous mid-surfaces (with or without a discontinuity in the plate/shell thickness).
2. Intersections with discontinuous mid-surfaces (with or without a discontinuity in the plate/shell thickness).
3. Over-laid plate/shell construction, which presents an indeterminate degree of through-thickness connectivity and resulting bending stiffness. This detail also invariably results in an off-set midsurface. Such construction detail is commonly used in an attempt to reduce local stresses or spread load, although a lap-joint, which also falls into this category, is simply used as a basic means of connection.
4. Connections using bolts or welds have long been recognised as presenting particular challenges to routine finite element analysis. In addition to the need to represent the joint stiffness and perhaps local stress distribution accurately, clearly the correct sizing of the bolts and welds themselves is also an important part of the overall process.
5. The transition of geometry from an area that may be considered from a ‘mechanics of materials’ viewpoint as being ‘thin’ to one that may be

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regarded as 'thick' or even 'three-dimensional' also presents modelling challenges. There are, of-course, various ways of connecting shell elements to brick elements and many of these are now automated in today's analysis systems. What is sometimes not so apparent, is how these element connections affect local stresses and stiffness.

6. Stiffened plate/shells using structural sections are common in industry. Predicting the correct structural behaviour, particularly when using combinations of beams and plate/shell elements, can be a significant challenge to the analyst.

It was also noted that many of the above challenges may also have more significance to some analysis types (e.g. buckling) than others.

The above issues provide a backdrop to the selection and development of the following procedural benchmarks. The formal specification sheets and results templates, are presented in reference[1]. It should be noted that the modelling issues addressed by these benchmarks are applicable to a wider range of industries than those simply involved in fabricating 10mm thick steel plate!

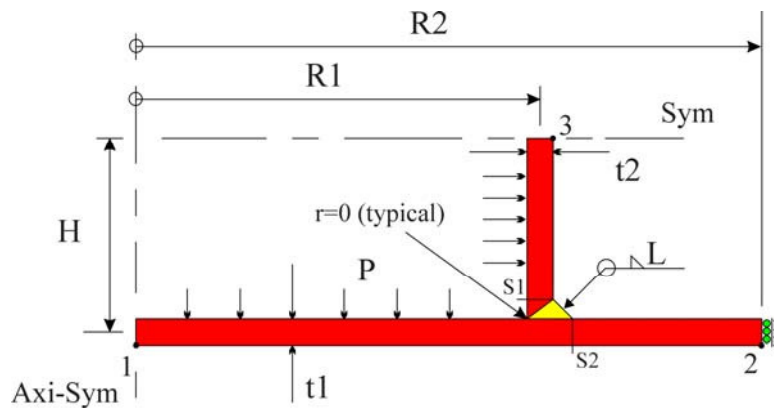
2.1 FENET_E&D1 Shell Intersection

The main purpose of this procedural benchmark was to identify the limitations of modelling practices currently in use, using plate/shell elements, for adequate representation of the stiffness and stresses in large fabrications containing intersections **that exhibit a slope discontinuity in shell/plate midsurfaces.**

The stresses and deflections in the fabricated detail shown were to be determined using common industrial modelling practices. Participants were asked to use any elastic failure criteria appropriate to their industry sector, to establish margins of safety against static and fatigue failure.

The target solution quantities required for comparison were the deflections and principal stresses at points 1, 2 and 3 and the principal stress distributions through the thickness at sections s1 and s2. Participants were also asked to indicate the elastic stress(es) to be used for assessment of static failure margin(s) and the "hot-spot" stress(es) for fatigue assessment.

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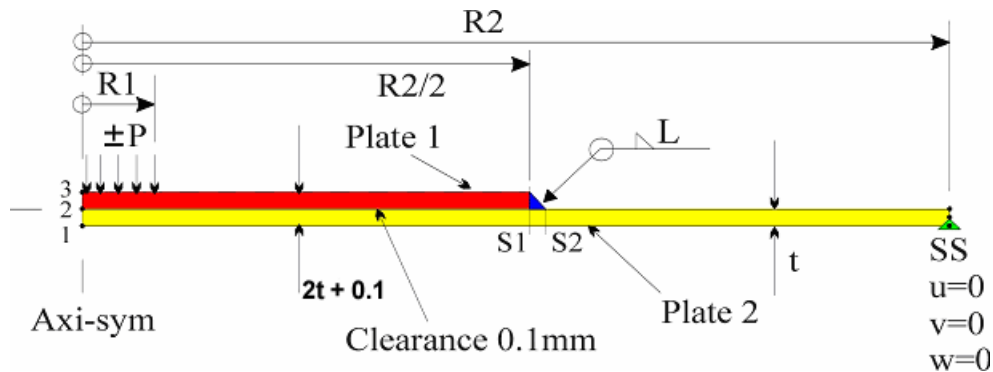


Data: $R1=650$ mm; $R2=1000$ mm; $H=300$ mm; $t1=20$ mm; $t2=15$ mm; $L=15$ mm (leg length). Neglect self-weight; 45 degree full penetration fillet. Internal pressure $P=0.2$ N/mm² @ 2×10^6 cycles (0...P...0). EN10025 S355 JR steel in the as-rolled, as-welded condition. Young's Modulus=200000 N/mm²; Poisson's Ratio=0.3; Minimum Yield Strength=355 N/mm² for $t < 16$ mm; Fatigue strength (stress range) for plain plate = 280 N/mm² with a 2.3% (2SD) probability of failure. Tensile Strength 560 N/mm².

Figure 1: Shell Intersection Benchmark

2.2 FENET_E&D2 Reinforcing Plate

The main purpose of this procedural benchmark was to identify the limitations of modelling practices currently in use, using plate/shell elements, for adequate representation of the stiffness and stresses in large fabrications containing reinforcing (wrapper, compensation, spreader) plate detail.



Data: $R1=50$ mm; $R2=1000$ mm; $t=15$ mm; $L=15.1$ mm (leg length); 45 degree fillet; Neglect self-weight. Load case 1: pressure $P=+2.5$ N/mm² @ 2×10^6 cycles (i.e. upwards 0...+P...0); Load case 2: pressure $P=-2.5$ N/mm² @ 2×10^6 cycles (i.e. downwards 0...-P...0). EN10025 S355 JR steel in the as-rolled, as-welded condition. Young's Modulus=200000 N/mm²; Poisson's Ratio=0.3; Minimum Yield Strength=355 N/mm² for $t < 16$ mm; Fatigue strength (stress range) for plain plate=280 N/mm² with a 2.3% (2SD) probability of failure. Tensile Strength 560 N/mm².

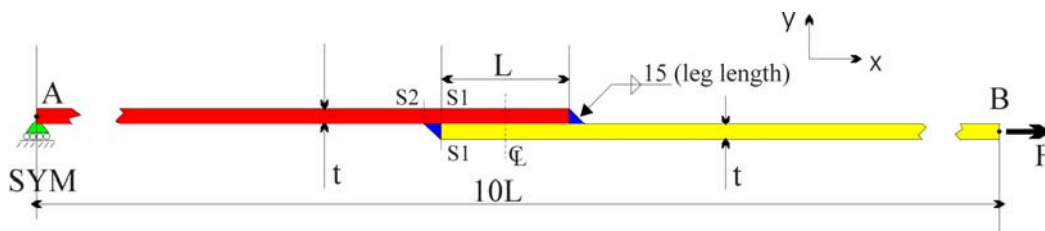
Figure 2: Reinforcing Plate Benchmark

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The target solution quantities required for comparison were the deflections at points 1,2 and 3; hoop and radial stresses at points 1, 2 and 3; hoop and radial stress distributions through the thickness at sections s1 and s2; contact radius. Participants were also asked to indicate the elastic stress(es) to be used for assessment of static failure margin(s) and the “hot-spot” stress(es) for fatigue assessment.

2.3 FENET_E&D3 Single Lap Joint

The main purpose of this procedural benchmark was to identify the limitations of modelling practices currently in use, using plate/shell elements, for adequate representation of the stiffness and stresses in large fabrications **containing an offset in the shell/plate midsurface.**



Data: $L=100$ mm; Plate width= 100 mm; $t=15$ mm; Neglect self-weight; 45 degree fillets. Force $F=15$ kN @ 2×10^6 cycles (0...F...0). EN10025 S355 JR steel (old BS 4360 Grade 50B) in the as-rolled, as-welded condition. Young's Modulus= 200000 N/mm²; Poisson's Ratio= 0.3 ; Minimum Yield Strength= 355 N/mm² for $t < 16$ mm (345 for $16 < t < 40$); Fatigue strength (stress range) for plain plate= 280 N/mm² with a 2.3% (2SD) probability of failure. Tensile Strength 560 N/mm².

Figure 3: Single Lap Joint Benchmark

The target solution quantities required for comparison were the X and Y components of deflection at B; Longitudinal stress distribution through the thickness at A; Longitudinal stress distributions through the thickness at sections s1 and s2. Participants were also asked to indicate the elastic stress(es) to be used for assessment of static failure margin(s) and the “hot-spot” stress(es) for fatigue assessment.

3 Observations from Results

Unfortunately it is not practical to provide the detailed results from all of the anonymous participants in the ‘round-robin’ exercise. Interested readers should consult reference[1] for more detailed information.

Despite the lack of complexity, response to the call for participation in the ‘round-robin’ must be regarded as poor, with only 10 contributors over the 3 procedural benchmarks (8 minimum on any benchmark). The reasons for this are not apparent, although lack of financial incentive to participate and the

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resulting possible perception of a 'no-win/only lose' outcome, no doubt had some influence. However, there were enough responses to ensure that this was a worthwhile exercise, with submissions from the civil, structural, shipbuilding and general industry sectors as well as academia, research and a software vendor.

The Excel spreadsheets containing the results from all contributors to the 'round-robin' exercise, along with some explanatory comments, are presented in reference[1]. It was agreed from the outset of this exercise that the identity of all contributors would remain anonymous.

3.1 Generic observations:

1. Significant variation in the modelling, results, assessment and conclusions relating to fitness for purpose of such detail is apparent across analysts and industry sectors, for both static and fatigue situations. The spread of results, for what on the face of it are relatively simple details, should certainly provide fruitful avenues of discussion in the education, validation and QA areas. It may come as a surprise to many, that these sort of observations are not unusual in "round-robin" exercises in this general area[2-5].
2. Human error is apparent, including:
 - a. Mis-interpretation of boundary conditions;
 - b. Incorrect use of modelling functionality, which altered physical response;
 - c. Reporting wrong results;
 - d. Using stress output directly at singularities;
 - e. Using averaged stresses at shell intersections.
3. Not all contributors checked that the field stresses compared well with hand calculations, where appropriate. Such a simple check would have flagged errors prior to submission.
4. From this limited linear elastic exercise:
 - a. The need for finite element knowledge is confirmed;
 - b. The need for general engineering education is confirmed;
 - c. The need for industry specific knowledge is confirmed;
 - d. The need for validation is confirmed;
 - e. The need for adequate QA procedures is also confirmed.
5. Established 'common best practice' across the various industry sectors is not apparent with, for example, only 1 contributor making specific reference to IIW Guidelines. The use of experimentally derived results on real weld geometries is perhaps widely recognised as a necessary part of the assessment process for fatigue. However, it is how FEA results are obtained (often at locations where singularities exist) that provides scope for variations in approach. The IIW, in particular, has done much work in this area and definitive guidelines are eagerly awaited. **There would however appear to be a need for a wider**

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dissemination of any such guidelines, given the wide use of welded fabricated construction.

6. As with all analyses, the adequacy of any idealisation must be judged in terms of the purpose of the analysis being conducted. The idealisation of such fabrication detail will affect static and fatigue (as well as dynamic, buckling, limit and fracture) assessments to different degrees.
7. The influence of fabrication detail can be local or global in nature and this fact should be considered when judging adequacy. The details selected for consideration as "procedural benchmarks" have both local and global measures selected as targets and all showed variation.
8. It is possible to use shell models to obtain necessary stress data for fatigue assessment of such details, but care and understanding is necessary. It should also be recognised that shell models will, in general, produce finite converged results at intersections.
9. There are two distinct approaches to obtaining "hot-spot" stresses from finite element models:
 - (a) A stress linearization procedure (not required with shell representations), designed to remove the peak stress component and leave the membrane and bending stress components. Such an approach will include gross geometric stress concentration effects. Some finite element systems provide post-processing tools for defining the "assessment or classification section" in both 2D and 3D representations and for linearising the results. This approach is common in the Pressure Vessel industry.
 - (b) A number of "extrapolation" approaches, with variations in the extrapolation procedures, are in use. These approaches typically involve element sizes of the order of $0.4t$ and differ in whether linear or quadratic extrapolation is used to the hot-spot location (which is often a singularity in non-shell models). It is however recognised that specific Codes of Practice may not give the analyst any choice in which procedure to adopt.
10. Some fatigue assessment procedures require the use of the nominal stress range on the weld throat area. Stresses plotted across the throat will show a highly non-linear variation. Although not always clear from submissions, it is likely that a simple 'membrane+bending' value of stress, with peak component removed, has been used, to provide consistency with hand calculations.
11. Some analysts used thick shell elements. The fact that most did not, would perhaps indicate that participants from different disciplines do not have a common understanding of when plates and shells become thick.
12. Given the nature of the Displacement Finite Element Method and the details examined, it is perhaps not surprising that greatest agreement is apparent for global stiffness (as measured by overall displacements), closely followed by field stresses (by definition away from local stress

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concentrations). Greatest variation is apparent for local stresses, which include finite values derived from distributions in the vicinity of singularities. In addition, as would be expected, greatest variation is apparent for very small target values, with best agreement generally for large values.

3.2 Observations for E&D1:

1. For this detail, the practice of displacing shell stress distributions by half a shell wall thickness before interpolating values at the 'notional' position of weld toes (hot-spot), would seem unnecessary.
2. The various results provided by 'Analyst Identifier 1' show remarkably little variation amongst 2D-Axi and shell models (with and without weld representation). Whilst inclusion of the weld stiffness (by whatever means) improves the comparison with the highly refined 2D-Axi results, it is not apparent that this additional complexity is merited over a simple shell representation and use of stresses at the location corresponding to the weld toe.
3. Although not considered in these benchmarks, it is noted that a simple shell intersection representation already has too much mass, without the addition of any measures designed to include the effect of the weld. This will have some bearing on dynamic analyses and weld models will result in a greater need to reduce the density of local elements for accurate representation.

3.3 Observations for E&D2:

1. The reason for the large variation in contact radius results for load case 2 is not apparent. This is clearly a function of global stiffness representation as well as the effectiveness of the contact methods used. Given the relatively good agreement on overall deflections in most cases, it must be assumed that the differences are largely due to the contact methods. This fact emphasizes the need for adequate contact benchmarks.
2. The various results provided by 'Analyst Identifier 1' show remarkably little variation amongst 2D-Axi and shell models for deflections, field stresses and weld-toe stresses, for load case 1. The poor comparisons for load case 2 are due to the lack of contact simulation in the shell model for load case 2.
3. Assuming the reinforcing plate to be integral did not provide good agreement for field stresses at the plate centre. The comparison of local stresses in the region of the weld are reasonable, particularly for those of larger magnitude. It is clear therefore, that if such an assumption is to be made, then care must be taken to ensure that both plates effectively act as one through use of a suitable number of spot or puddle welds. The results for the 'central spot-weld' idealisation, would indicate that

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an 'integral' behaviour may be possible with relatively few plate connections.

4. Neglecting the offset due to the reinforcing plate and assuming a double thickness integral representation over the reinforced area produced similar results to the 'integral' idealisation with offset.

3.4 Observations for E&D3:

1. Only two participants showed that the problem was in fact 'large displacement', in spite of the tip deflection being less than the thickness of the plate. Reductions in deflections and stresses are significant. The rules of thumb commonly used as a guide to when large displacement effects become significant for beams, plates and shells are clearly not applicable for this problem. The reason for this is apparent when the source of the non-linearity is given due consideration.
2. 3D models (shells and bricks) show variations in results across the width, which are obviously absent from 2D results. Not all contributors commented on this effect.
3. Neglecting the offset, even with correct plate thicknesses, fails to predict adequate values for overall stiffness, field stresses and local stresses. Analysts should therefore think carefully before neglecting offsets in plate/shell mid-surfaces, as their effects can have a global nature as well as local. For thinner plates/shells, large displacement effects may act to reduce the global effect of the offset, through local bending of the joint and effective realignment of the mid-surfaces.
4. The modelling of contact between the lapped plates appears irrelevant for the relative joint sizes considered. The results for separate plates, with and without contact, and models where the plates were assumed integral appear similar. The latter model however, fails to pick up the stress singularity that exists at both ends of the lap running between fillet weld roots. Almost all contributors failed to highlight this singularity, in any model.

4 Concluding Remarks

Observations from the 'round robin' detail some surprising results. In the first two benchmarks about half of the respondents provided results which suggested that they had made modelling errors. In the third example, only two out of ten respondents realised that this was a nonlinear geometric problem. Whilst some contributions were no-doubt completed under time pressure, it can be argued that this is a reflection of the everyday industrial environment for many engineers. The resulting levels of human error and lack of results checking, for what some might regard as simple case studies, must be of wider interest and concern. The general spread of results arising from the different modelling and assessment strategies should also be of interest. The outcomes certainly confirm the ongoing role that organisations such as NAFEMS have,

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in ensuring quality and promoting the education and development of analysts and engineers. It is interesting to reflect on the fact that the same exercise and same general conclusions, could probably have been made 30 years ago. The main difference today being that the **same mistakes can now be made more quickly and conveniently.**

Round-robin exercises often show up surprising variations in results amongst participants, even when such participants may be judged as 'expert' or 'competent' beforehand. Of some relevance to the present FENET work, is the round-robin exercise reported by Katajamaki et al[2]. The aim of the round-robin was to validate the different fatigue analysis approaches, involving the determination of hot-spot stresses and results post-processing. The structure analysed was an I-beam with a gusset welded to the flange. There were 3 loading cases. A consortium of 12 Nordic industrial companies, universities and research institutes took part in the exercise, producing 28 different solutions. It was concluded that shell models were less successful (underpredicting) than brick models in predicting the stress concentration factor. However, it is not clear whether the shell models had been completed in accordance with IIW guidelines. From the results, it was clear that it was possible to obtain adequate results from shell idealisations, with care. In addition, it was noted that the hot-spot approach was better than the nominal stress method when comparing fatigue lives. Quite a number of participants had used highly refined meshes, untypical of routine analysis tasks. Three participants failed to establish the nominal stresses adequately, for the simplest loading case. The third loading case produced even greater variation. The authors report that a previous Nordic round-robin[3] involving the static loading of a welded lug attachment had produced higher than expected scatter, partly due to the analysts inability to define the location of maximum principal stress. The authors also report that a round-robin on a similar component, reported by Niemi[4][5] to the IIW working group on hot-spot stresses, showed significant differences in the choice of position and direction of the extrapolation line.

It is recommended that NAFEMS organise, through its working groups, an on-going series of such 'round-robin' exercises across all analysis areas. The results from such an effort and investment, could have valuable quality and educational benefits.

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