FE Analysis of thick test specimens

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Confidential

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## CHANGE RECORD

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<td>1</td>
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1. INTRODUCTION

1.1. Rationale behind the study

In this report, the results of the FE analyses of ECN on the thick OPTIMAT specimen are reported. The thick specimens are aimed at relating the properties of thick and thick repaired laminates to thin and thin repaired laminates, thus extending the validity of the OPTIMAT programme as outlined in the updated detailed plan of action of TG4 [1]. The production and testing of the 2m long specimens is costly and time-consuming, hence it is worthwhile to assess the specimens in closer detail beforehand. As is explained in detail in Chapter 2.1, the thick specimen is based upon the standard OPTIMAT specimen, but five times as large in each dimension, thus nominally 150 mm wide and 32.85 mm thick.

Since LM has a table of 2m long on which the specimens are produced, and some space is needed for draping the vacuum bag etc., the total length of the specimen can only be about 1.9 m. As the specimen length between the tabs is already 1380 mm (see 2.2). This leaves an area of just 140 mm wide and 300 mm long for the tabs, a rather small area for introducing a force of 2500 kN.

Determining the optimal bolt pattern for the test and assuring that the tabs do not fail prematurely is the aim of the FE analysis.

It must be noted here that this study does not actually accurately predict the failure mode as such, since this would require non-linear material behaviour, crack modelling, slippage of the test specimen between the plates all of which are hard to assess. For instance determining the non-linear properties of the material is a project in its own right. Also the FE model would require much more time and CPU time than used for the current assessment.

Instead the FE model should be seen as a first assessment of the stress state and a relative assessment of the various load introduction solutions.

1.2. Choice of repair technique

At the start of the project, the behaviour of the repaired specimens was unknown, but later it turned out that the repaired specimens had an about 20% lower strength than the repaired specimens. The results of the thin repaired specimens produced by LM and tested at WMC is summarised in Figure 1.
The following observations can be made:

- The reference strength of the unrepaired specimens is about 100 kN, or 15% higher than the strength of MD specimens with the standard geometry, which is probably due to the longer length which allows stress raising effects of one tab to be mitigated before reaching the other tab [5].
- The repaired specimens with a slope of 1:50, 1:75, and 1:100 all result in an ultimate strength of about 80 kN, so a loss of 20% in strength compared to the reference specimens, irrespective of slope and repair thickness.
- A steeper repair slope results in a significant decrease in strength, below 50% of the original strength.

As a conclusion, a repair slope of 1:50 seems most efficient. A repair thickness of $\frac{1}{3}$ seems less attractive, since the repaired specimens are already 20% weaker than the reference specimens, fairly close to the expected 33% loss in strength if the repair hadn’t been carried out in the first place. Therefore a $\frac{2}{3}$ repair thickness is preferred.
2. TEST SPECIMEN

2.1. Basis: original MD test specimen

The basis for the thick specimen is the standard Optimat Blade specimen for MD material, which is a rectangular specimen, called R04 in the OPTIMAT project, see below.

The lay-up of the MD laminate \((\pm 45, 0)_4 \pm 45\) consists of 5 layers \(\pm 45^\circ\) (each layer 0.61 mm thick) interspaced with four layers UD (each layer 0.88 mm thick). Total thickness of the MD laminate is 6.57 mm. Tabs are glued on the specimen with a standard epoxy resin, thickness of the glue can be taken close to zero. The tabs of the standard OPTIMAT MD specimen are made of 1 layer of 'tab material' (not used for the thick specimens).

The \(\pm 45^\circ\) layers in the MD laminate are all \(+45^\circ\) on top and \(-45^\circ\) at the bottom, therefore slightly asymmetric, see also Figure 3.
2.2. Thick MD specimen geometry

The original thick specimen geometry was taken to be the same as the original MD specimen R04 but five times as large, thus 32.85 mm thick, 150 mm wide and 750 mm long.

The thickness

The thick specimens are five times as thick. The material properties lay-up remain the same: since the lay-up of the MD is \((\pm 45, 0)_4 \pm 45\) and of the thick MD: \((\pm 45_5, 0_5)\_4 \pm 45_5\), the material properties remain the same, and the lay-up is created by taking each layer 5 times as thick. The reason for this lay-up, rather than interspacing the layers like \((\pm 45, 0)_{20} \pm 45\) which would probably have slightly better properties, is that it more closely resembles the thick laminates as used in turbine blades.

The tabs are cut from the same plate, so that the tabbed area is three times as thick as the free length between the tabs, or 98.55 mm.

For the FE analysis, this means that the material properties remain the same, just the thickness changes, by changing the elements of 6.57 mm to 32.85 mm (MD to thick MD).

The width

The original thick specimen geometry would be 150 mm thick. However, since the thin long specimens exhibited a 15% higher strength than standard MD specimens, reaching about 100kN, the 5·5 larger cross section of the MD specimens would reach 2500 kN, close to the capacity of the test machine at WMC (3000 kN static, 2500 kN fatigue).

In order to create slightly more margin, the width is limited to 140 mm.

The length

The original thick specimen geometry would be 750 mm long. However, it was decided to test thick repaired specimens as well.

Since the thick specimens would also be used for testing repaired specimens with a repair depth of \(2/3\) of the thickness, and a slope of 1:50 (see 1.2), the free length needed between the tabs would be at least \(50 \cdot 2/3 \cdot 32.85 = 1095\) mm. Limiting the width to 140 mm and requiring at least that same length between the tabs and the start or end of the repair, brings the total length of the specimen between the tabs to 1380 mm.

Since LM has a table of slightly over 2m long on which the specimens are produced, and some space is needed for draping the vacuum bag etc., the total length of the specimen can only be about 1.9 m. This leaves just 300 mm for the tabs, for a total specimen length of 1980 mm.
2.3. **FE model of the test specimen**

Because of symmetry, only 1/8 of the plate needs to be modelled. Note that, in contrast to previous FE analyses [5], solid laminate elements have become available in MARC and were employed in this analysis.

**Figure 4** Thick specimen geometry (shown for repaired specimen)

**Figure 5** Part of the test specimen modelled for FE analysis

The thickness of the solid elements is 16.42 mm, 1 row of elements is used for the area between the tabs, and 3 rows were used for the tabbed area.

**Boundary Conditions**

Bolts modelled as rigid cylindrical contact surfaces, fully fixed.

Nodes at x=0 pulled (see below)

Nodes at z=0 fixed in Z-direction at half the thickness of the test specimen.

Nodes at y=0 fixed in Y-direction
Load cases

The three bolts in the first model are modelled as three rigid cylinders which are fixed in space, while the plate is being pulled at by 2500 kN in the middle (modelled as 42 point loads of 14.88 kN for the ¼ of the cross section that is modelled).

The load is calculated for 10 steps, where the load is gradually increased to the full 2500 kN as shown in Figure 6.

Modelled with solid laminate elements, 45 mm thick and thus only 1/8th of the total test specimen.

![Figure 6 Load on test specimen versus time](image)

2.4. Material

The properties of a layer of the reference material are taken from [3]

<table>
<thead>
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<th>fibre and matrix</th>
<th>Silenka E-glass, MY750</th>
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<td>[2] estimated</td>
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<td>In-plane shear modulus ( G_{12} ) [GPa]</td>
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<td>In thickness shear modulus ( G_{13} ) [GPa]</td>
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<td>Transverse tensile strength ( Y_T ) [MPa]</td>
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<td>Thru thickness tensile strength ( Z_T ) [MPa]</td>
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### Non-linear material properties

The non-linear relationships given in [3] do not state whether they are presented for the true stress-strain relation (stresses determined from the cross section under loading) or the engineering stress-strain relation (stresses calculated from the unloaded cross section). It is believed that in the case of FRP with fairly modest strains the difference is negligible. Since MARC requires the true stress-strain relationship the numbers given in [3] should be used as the true stress-strain relation without any attempt at conversion.

Unfortunately, MARC only seems to allow for non-linear material properties in $11(x)$- direction, and this only assuming a symmetrical diagram for tension and compression, so the non-linear shear properties were not used.

### Resulting FE model

The resulting model, consisting of 6900 solid laminate elements is shown below.

![FE mesh](image-url)
3. OVERVIEW OF THE RESULTS OF VARIOUS MODELS

3.1. Introduction

Only the elements in the centre area (green area in Figure 4) are shown, because of symmetry conditions.

In principle only the strains in x-direction is presented for the central 0° layer. This layer is higher loaded than the outer 0° layer of the tab, as can also be observed in Figure 8, which shows the displacements to be larger towards the central plane of the thickness. Where necessary, extra figures are shown.

![Image of displacement whole model](image)

**Figure 8** Displacement whole model (inset) and detail

As can be seen, the centre cross section moves about 12 mm from its unloaded position and even in the tab, there is still 1 mm difference between the central part and the end.

In the next chapters four models will be analysed:

1. The model with 4 M30 bolts as shown above and in Figure 4.
2. Same as the first model, but bolts shifted slightly relative to the boltholes, in order to optimise the force distribution over the bolts.
3. Same as first model, but instead of 3 M30 bolts, an M20, M30 and M40 bolt.
4. Same as the third model, but bolts shifted slightly relative to the boltholes, in order to optimise the force distribution over the bolts.
3.2. Results of model 1
This model is based on 6 M30 bolts, spaced 90 mm apart with holes of 33 mm, as shown in Figure 4.

**Figure 9** Strains in X-direction

**Figure 10** Strains in Y-direction

**Figure 11** In-plane Shear Strains
Figure 12  Stresses in X-direction

Figure 13  Stresses in Y-direction

Figure 14  In-plane Shear Stresses
Observations on model 1:

- The maximum stress is vastly exceeding 1000 MPa (in fact close to 2000 MPa) at the side of the first bolt, whereas the stresses around the other bolts are lower than in the thin part (about 800-1000MPa) and therefore OK. From the low compressive stresses, it can be seen that the other bolts do hardly carry any load.
- The stress in the first bolt vary significantly over the length of the bolt, at the outer surface, the stresses are no higher than in the thin part between the tabs.
- It is obvious that although the stress in the tabs in general are quite acceptable, the stresses in a small area around the first bolt are too high. It is unsure whether these stresses would cause premature failure in the tabs, but it certainly seems possible if not enough stresses are redistributed before failure of the highest loaded part.
- Looking at Figure 15 and Figure 16, the failure criteria are much higher than 100%, so failure of the net section around the first bolt seems quite likely.
3.3. Results of model 2

Since the first bolt was carrying all the load in the first model, it might be beneficial to move the bolts a bit, in order to get a better distribution of the forces across the three bolts. From Figure 8, it can be observed that the difference in displacement in X-direction between the first and third bolt is only about 0.4 mm at the surface. Therefore, two bolts are moved in negative X-direction: bolt 1 is moved 0.4 mm, while bolt 2 is moved 0.2 mm.

![Figure 17 Strains in X-direction](image)

![Figure 18 Strains in Y-direction](image)

![Figure 19 In-plane Shear Strains](image)
Figure 20  Stresses in X-direction

Figure 21  Stresses in Y-direction

Figure 22  In-plane Shear Stresses
Observations on model 2:

- Compressive stresses now occur at bolt 1 and bolt 3, suggesting a better distribution across the three bolts, but still the first bolt carries most of the load.
- If a displacement of 0.4 mm already causes such redistribution in loads, it pays to position the bolts very carefully. It does suggest than in practise the loads will be distributed better due to slight crushing of the material just under the bolts, but it is unclear how much strength will be left at that point: are the loads distributed enough before failure occurs?
- The major problem, the high stresses next to the first bolt are not significantly lower than in model 1 and might still cause preliminary failure in the tabs.
3.4. Results of model 3

In this model, the major problem of high stresses next to the first bolt is tackled by replacing the three 3M30 bolts of the previous models by an M20, M30 and M40 bolt respectively. Furthermore, the bolts are injected with a layer of epoxy of 1.5 mm thick, filling the bolthole. For the epoxy, a Young’s modulus of 16.2 GPa is chosen, a value that corresponds reasonably to the properties in 90° of UD material. Although it could be argued that the value is high for unreinforced epoxy, the fact that the epoxy is locked in could lead to similar stiffness. A lower stiffness would be beneficial, so the assumption seems conservative.

![Figure 25 Strains in X-direction](image)

![Figure 26 Strains in Y-direction](image)

![Figure 27 In-plane Shear Strains](image)
Figure 28  Stresses in X-direction

Figure 29  Stresses in Y-direction

Figure 30  In-plane Shear Stresses
Observations on model 3:

- The maximum stress is now close to 500 MPa at the side of the first bolt, versus over 800 MPa in the section between the tabs, thus the main problem of earlier models seems solved.
- From the low compressive stresses, it can be seen that the other bolts hardly carry any load. Transferring 2500 kN by an M20 bolt is not going to work…..
3.5. Results of model 4

Just as was done in model 2 relative to model 1, this model is based on model 3, this time with the M20 bolt moved -0.4 mm and the M40 bolt moved +0.40 mm in X-direction. The epoxy is modelled with an uneven thickness to reflect the move of the bolts.

![Figure 33 Strains in X-direction](image)

![Figure 34 Strains in Y-direction](image)

![Figure 35 In-plane Shear Strains](image)
Figure 36  Stresses in X-direction

Figure 37  Stresses in Y-direction

Figure 38  In-plane Shear Stresses
Observations on model 4:

- Not much has changed: moving the bolts doesn’t help in case the boltholes are injected with epoxy
4. RESULTING TEST SPECIMEN

4.1. Shear force in the bolts

Can the M40 bolts be replaced by an M30 bolt, making manufacturing a bit easier?

A single 10.9 M30 bolt can take 339 kN per cross section. Since the bolts transfer the load to two steel plates on top of and below the tabs, there are 2 load carrying cross sections, each bolt can carry 678 kN, therefore 4 M30 bolts can just transfer 2500 kN in ideal circumstances.

4 M30 and 2 M20 bolts transferring the loads at both sides of the bolts, can take a maximum force of 4·678 + 2·151=3014 kN, a slim margin if the strength is close to the maximum static force of the machine of about 3300 kN.

Leaving the configuration of 2 M20, 2 M30 and 2 M40 bolts per tab, results in a maximum force of 2·1200+2·678+2·151= 4058 kN, a much more comfortable margin.

However, the net cross section next to the M40 bolt should then be able to carry the load. In order to test this, model 4 was run once again as model 5, with bolt 1 and 2 removed and loaded by 1200 kN in order to check to stresses next to bolt 3 in case the M40 bolts are loaded to their maximum load.

As can be seen in Figure 41, the stresses are quite acceptable, so that the configuration with 2 M20, 2 M30 and 2 M40 bolts per tab seems optimal.

![Figure 41 Stresses in X-direction](image-url)
4.2. Influence of stiffness of epoxy

What changes when the stiffness of the epoxy is taken to be 6.2 GPa, rather than 16.2 GPa?

No influence can be seen, the stresses are the same as for models 3 and 4.
5. LITERATURE


6. NOTES ON USE OF MARC

- Tables in MARC on non-linear material: input as type: plastic strain. The Y-values in the table (stress; X=strain) are multiplied with the value “initial plastic strain”, so leave this value at 1 at all times to avoid confusion.
- Only non-linear in 11-direction and symmetrical for tension and compression. For large strains use linear strain: $\varepsilon_{\text{log}} = \ln(1+\varepsilon)$. For $\varepsilon=10\%$: $\varepsilon_{\text{log}}>9.5\%$, or 5 % difference, for smaller strains difference is small (for $\varepsilon=1\%$, error <0.5%).
- Failure criteria in absolute values (compression failure strength>0).
- Ignoring the difference between engineering stress and true stress and considering just the plastic strain/stress allows for input of the total stress-strain curve as given in [3].
- Only complete set (elastic/plastic/total strain) for MARC is for the local axis system.
- Stresses are output in the direction of the fibres, so be careful which layer to select.