Mechanical Tests for Sub-module

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February 18, 1997

1 Introduction.

According to “A Test Plan for Investigating the Structural Integrity of Tile-Cal modules” by Norman Hill a series of mechanical tests have been carried out. Stiffnesses and loads which could cause failure of the sub-module were measured. Measurements of displacements were carried out using dial indicators. These data have been compared with results of numerical calculations.

2 Shear failure of the front-plate

To simulate front part of three submodules connected with one front-plate, a set-up shown in Fig.1a has been realised. The final load equal to $85\,tons$ caused failure of two cross-sections of the front-plate and some parts of welds. The average value of shear stresses in the front-plate are equal to

$$\tau_{av} = 85\,tons\,\cdot\,9810\,\frac{N}{tons}\,/(2\,\cdot\,1240\,mm) = 336\,MPa.$$

Using the Mises-Hencky theory the equivalent one-dimensional stress can be calculated from the following formula

$$\sigma_{eq} = \sqrt{3}\,\tau_{av} = 582\,MPa.$$

Strength of steel used to produce the front-plate is defined by

$$Yield\,Strength = 250\,MPa,$$

and

$$Tensile\,Strength = from\;\; 565\,MPa\;\; to\;\; 630\,MPa.$$

Values of the equivalent one-dimensional stress versus deformations for this type of material are shown in Fig.1b. From the plot we can notice that deformations can be decoupled to the elastic portion limited by the Yield Strength and the plastic hardening portion limited by the Tensile Strength. In the test the equivalent stress reached nearly the value of the Tensile Strength.

3 The inner radius straps subjected to bearing load

Dimensions of the sub-module used in the test are shown in Fig.2. The sub-module is composed of 27 master-plates and 26 layers of spacers. At the inner radius the width of the straps is equal to $10\,mm$. Moreover, at the inner radius all the master plates and spacers have small holes shown in Fig.2. For the test the $10\,mm$ thick front-plate was welded in the inner radius key way.

A set-up for the test is shown in Fig.3a. In order to equally distribute the load on the contact surfaces we used aluminium shims. The force P was applied in three steps up to $78\,tons$ and than unloaded. The active area of the top strap was

$$10\,mm \cdot 220\,mm = 2200\,mm^2$$

so, the average value of stresses is

$$78\,tons \cdot 9810\,\frac{N}{tons}\,/2200\,mm^2 = 348\,MPa$$

At each step of loading and after unloading displacements of the top and bottom surface of the front-plate were measured (see Fig.3a, point 1 and point 2).

Displacement values of the geometric center of the front plate are shown in Fig.3a. We can see from this plot that plastic deformations of the bottom shim and the sub-module started when the applied load was about $38\,tons$.

4 Master-plates subjected to bearing load

Using the same sub-module as before and a set-up shown in Fig.4a, the test where all the master-plates were subjected to a compression load was performed. At the beginning lead shims were used. Plastic deformations of these shims started at a rather low value of stresses because for lead we have:

$$Yield\,Strength = 5.5\,MPa,$$

$$Tensile\,Strength = 12\,MPa.$$

This is why we changed the top shim to an aluminium one and at the bottom to a bronze shim.

The final value of the applied load (about $225\,tons$) was reached in 6 steps. The contact area ($8100\,mm^2$) was calculated as an area marked on the shims. So, at the highest value of loading the average stress on the top surfaces of the master-plates was $273\,MPa$. Displacements of the top surface of the front-plate are shown in Fig.4b. From this plot we can see that all the master-plates deform elastically. Plastic deformation mostly of the aluminium shim is equal to $0.1\,mm$. Uneven positions of the top and bottom surfaces of the master-plates caused that the contact area was slightly changing when the load was increasing and because of that the stiffness of the sub-module was also changing (see Fig.4b).

5 Sub-module subjected to twisting moment

To apply a twisting moment to the sub-module a set-up shown in Fig.5a was used. Deformations of the sub-module were measured at two points which are shown in Fig.5a as a point 1 and point 2. The total force P was applied in three steps up to $15\,tons$. Sums of vertical displacements at the point 1 and point 2 versus the force P are shown in Fig.5c.

From these experimental results the torsional stiffness of the sub-module can be defined using the following general equation for bars

$$\theta = \frac{Tl}{KG},$$

where
θ - angle of twist (radians),
T - twisting moment,
l - length of the bar,
G - shear modulus
K - factor dependent on the form and dimensions of the cross-section.
The formula above leads to the relation for the K factor of the form

$$K = \frac{Tl}{\theta G} = 8.36 \cdot 10^6\, mm^4,$$

where (see Fig.5b)
$T = 7.5\,tons \cdot 9810\,N/tons \cdot 178\,mm = 13.1 \cdot 10^6\,Nmm,$
$l = 1600\,mm,$
$G = 79600\,N/mm^2,$
$\theta = 5.6\,mm/178\,mm = 0.0315.$

Assuming that the sub-module is a solid piece of steel, a value of the K factor defined analytically would be in such a case 88 times bigger.

In order to make a comparison between the experimental results and numerical calculations, the FEM (ANSYS code [1]), was used to analized the sub-module subjected to the torsional moment. A model of the sub-module is composed of plate elements (the master-plates) connected by link elements which simulate glue layers together with the spacers (see Fig.6). The link elements are in the direction normal and nearly parallel to the master-plates (see Fig.7). Areas of the link element cross-sections were defined in such a way that stiffnesses normal and parallel to the master-plates for the both systems shown in Fig.8a and Fig.8b are the same. So, to define the areas additional calculations for the left system (Fig.8a) were performed where different values of the spacer width ($a = from\ 100\,mm\ to\ 250\,mm$) and different values of the glue layers thickness ($d = from\ 0.02\,mm\ to\ 0.20\,mm$) were assumed. It was check that influence of the glue layer thickness on the shear stiffness is nearly linear (see Fig.8c). Finally, the areas of the link elements were established assuming that the glue layer thickness was $0.1\,mm$ as an average value of all the glue layer thicknesses for the sub-module.

After introducing these data stresses and deformations of the submodule subjected to the torsional moment were calculated. For the final load ($15\,tons$) a measured value of displacements was equal to $5.6\,mm$ and the corresponding displacement obtained from the numerical analysis was $5.2\,mm$. Distributions of stresses in the master-plates are shown in Fig.9. Because of a rather good agreement between the experimental and numerical results of displacements, the finite element model can be used to define the torsional stiffness of the sub-module with different number of the master-plates.

6 Shear load between master-plates

As it it shown in Fig.10, nine of twenty seven master-plates were loaded on their top surfaces and nine master-plates on each side were supported on their bottom surfaces. The maximal value of the applied load was $80\,tons$. This force caused the first failure of glue which could be heard but not seen.

7 Bending of the submodule

The sub-module was loaded and supported in the way as it is shown in Fig.11a. When the load was applied in five steps, vertical displacements at four points 1, 2, 3 and 4 (see Fig.11a) were measured. The average values of displacements at the point 1 and point 2 for all the load steps were calculated and than the average values of displacements at the point 3 and point 4. Differences between these averages for each load step are shown in Fig.11b. The maximal value of the load ($50\,tons$) corresponds to about $61\,MPa$ of stresses in the master-plates.

Using the finite element model of the sub-module as it was described before, numerical calculations for this case of loading were carried out. Results of stresses in the master-plates and deformations of the sub-module are shown in Fig.12.

For loading ($50\,tons$) the maximal value of vertical displacements is about $0.25\,mm$. It should be noticed that the stiffness of the sub-module subjected to the bending load is about 3.5 times smaller than the one obtained from the numerical calculations with the assumption that the master-plates are perfectly aligned.

8 The inner radius straps of two sub-modules subjected to bearing load

A set-up shown in Fig.13a for testing two sub-modules, one on top of the other was prepared. Both of them are the new type sub-modules that is the thicknesses of them are $293\,mm$ and the width of the straps are $7\,mm$.

These two sub-modules were loaded on their inner radius bearing surfaces. Between them was inserted a $1\,mm$ steel shim. Both of the sub-modules were without front-plates. Loading and unloading was perform two times. Displacements were measured at point 1 and point 2 (see Fig.13a). For the first and second loadings results of vertical displacements (average values at the point 1 and point 2) are shown in Fig.13b and Fig.13c, respectively. The final load was $60\,tons$ in the first case and $50\,tons$ in the second one. For the both cases, it was checked that the thickness of the steel shims after unloading were reduced by $0.05\,mm$.

From Fig.13b and Fig.13c we can notice that the stiffness of the system (one sub-module with the support) is much smaller for the load below $10\,tons$ than that for the final load.

When the sub-modules were loaded the second time displacements were measured as well on the bottom surface that is at the point 3 (see Fig.13a). Results of these measurements are shown in Fig.13d. From this plot we can notice quite big change of the stiffness which probably was caused by lack of contact between some parts of the strap and the support. Moreover, it was found out that for the load equal to $50\,tons$ difference between the top surface displacement (the average value of displacements at the point 1 and point 2) and the bottom surface (displacement at the point 3) was $0.23\,mm$. The finite element model shown in Fig.14 was built to check numerically the stiffness of the front part of the sub-module. The calculated value was $0.22\,mm$. Moreover, calculations carried out for the sub-modules with the front-plates show that this stiffness is about 30% bigger.

9 Master-plates of two sub-modules subjected to bearing load

The same sub-modules with $1\,mm$ thick aluminium shims between them were compressed (see Fig.15a). The load was applied on the master-plates through a aluminium shim as well. Vertical displacements were measured at four points (see Fig.15a). Average values of displacements (u_t) of the master-plate top surfaces (the point 1 and point 2) are shown in Fig.15b, the average value of displacements (u_b) of the bottom surfaces in Fig.15c and the difference between the top and bottom surfaces ($u_t - u_b$) in Fig.15d.

From Fig.15d we can notice that for $80\,tons$ the difference ($u_t - u_b$) was about $0.42\,mm$. After unloading this difference was equal to $0.05\,mm$. At the beginning of loading the stiffness of the system is rather small in comparison with the final one. Calculated numerically stiffness only for the master-plates is the same as those which was reached in the test when the load was about $30\,tons$ (see Fig.15d). The total stiffness of two master-plates is about 3.5 times smaller than that for two master-plates alone.

10 Conclusion

From results of the tests the following conclusions can be drawn:

- for the old type sub-module plastic deformations begin when the load was applied on the front bearing surface and reached a value of $38\,tons$,
- the applied load on the master-plates gives the average value of stresses equal to $273\,MPa$ so, plastic deformations appear only in the aluminium shims,

- the torsional stiffness of the sub-module is 88 times smaller than a solid piece of steel with the same dimensions,
- the finite element model of the sub-module gives very good results for the torsional stiffness in comparison with the experimental measurements and can be used to calculate the torsional stiffness for sub-modules with different numbers of the master-plates,
- the stiffness of the sub-module subjected to the bending load is about 3.5 times smaller than the value obtained from the FEM with the assumption that the master-plates are perfectly aligned,
- the stiffness of the front part of the sub-module measured as a difference between displacements of the top surface and the bottom surface for $50\,tons$ is the same as those obtained by th FEM with the error of 5%,
- the calculated value of the front part stiffness of the sub-module is about 30% bigger,
- the total stiffness of two master-plates is about 3.5 times smaller than that for two master-plates alone,
- additional tests checking the front part of the sub-module with the front-plate welded in the key should be carried out.

References

[1] ANSYS - Engineering Analysis System, User's Manual, Swanson Analysis System, Inc., Houston, May 1992.

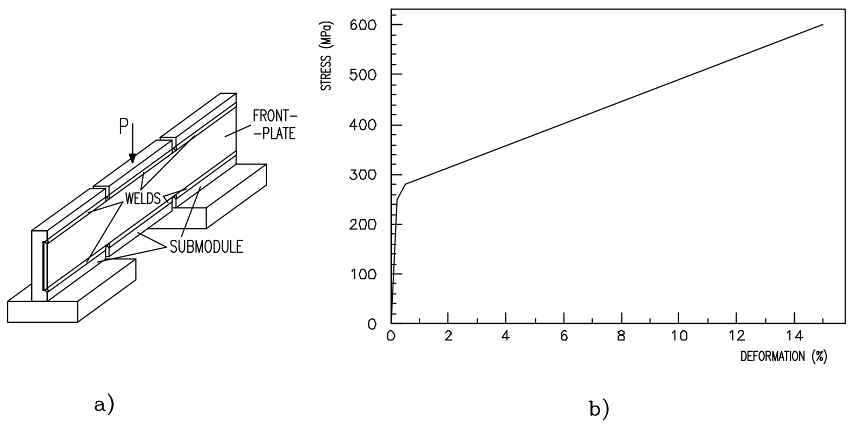

Figure 1: a) Set-up for front-plate shear test; b) Stress-deformation curve.

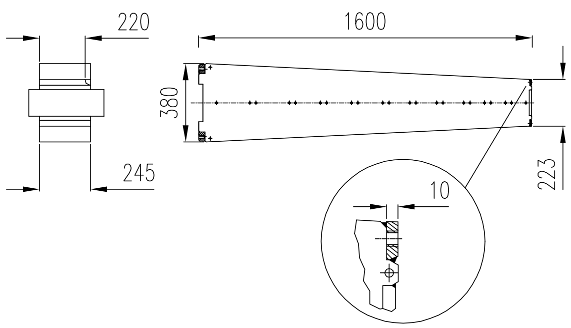

Figure 2: Dimensions of the sub-module.

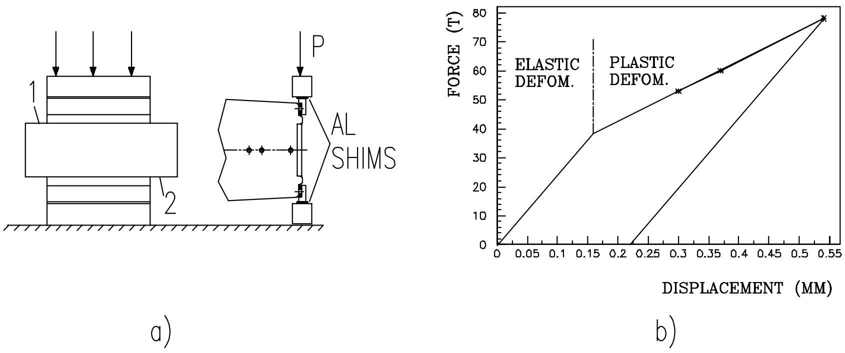

Figure 3: a) Set-up for testing the inner radius straps subjected to bearing load; b) Force-displacement curve.

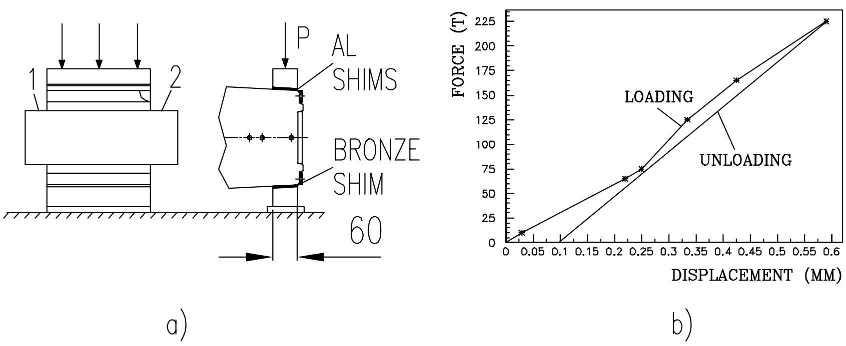

Figure 4: a) Set-up for testing master-plates subjected to bearing load; b) Force-displacement curve.

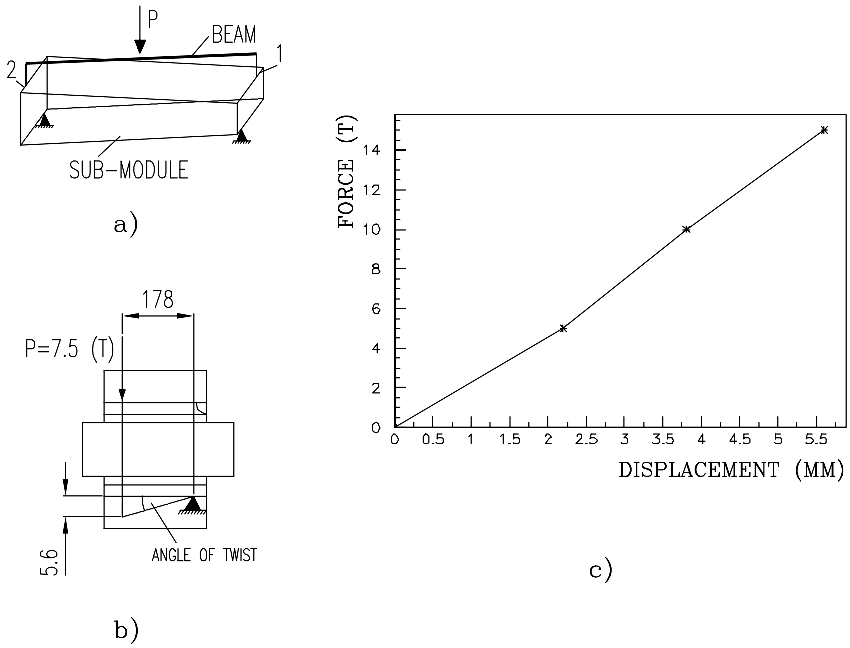

Figure 5: a) Set-up for testing the sub-module subjected to twisting moment; b) View from the end of the sub-module; c) Force-displacement curve.

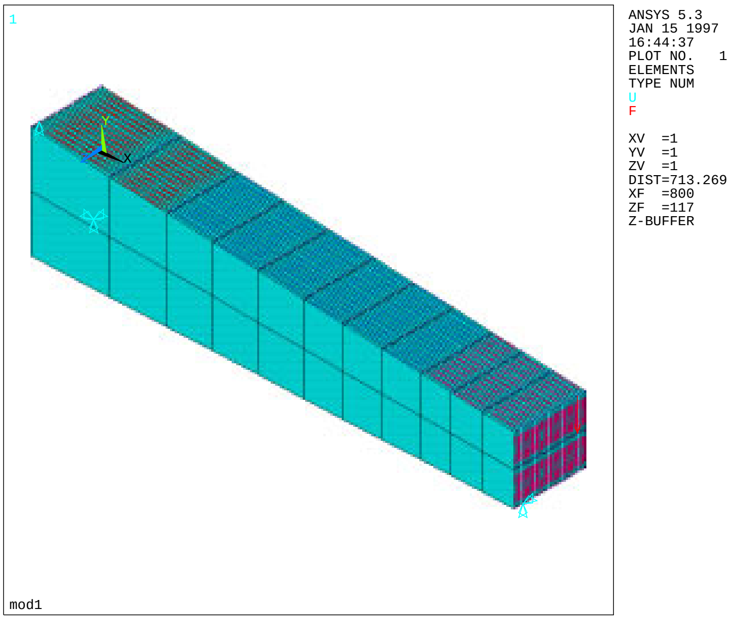

Figure 6: Finite element model of the sub-module.

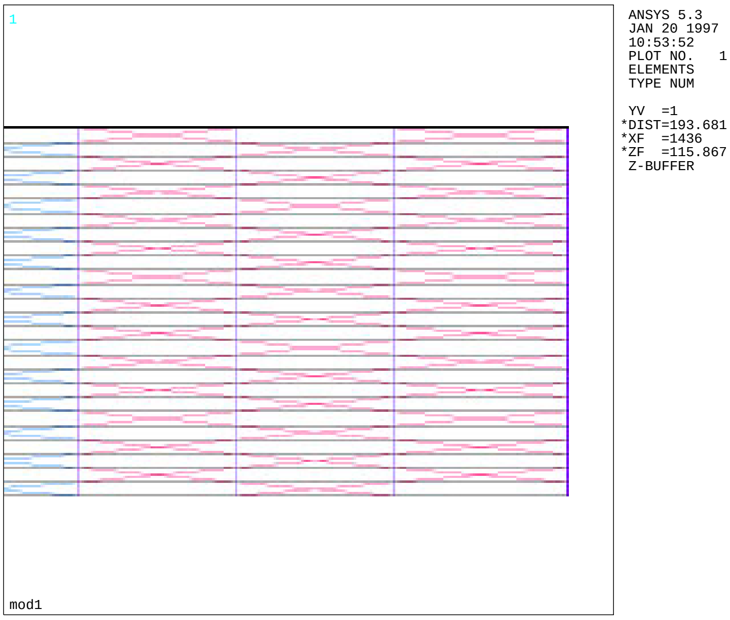

Figure 7: Plate and link elements (the top view of the sub-module).

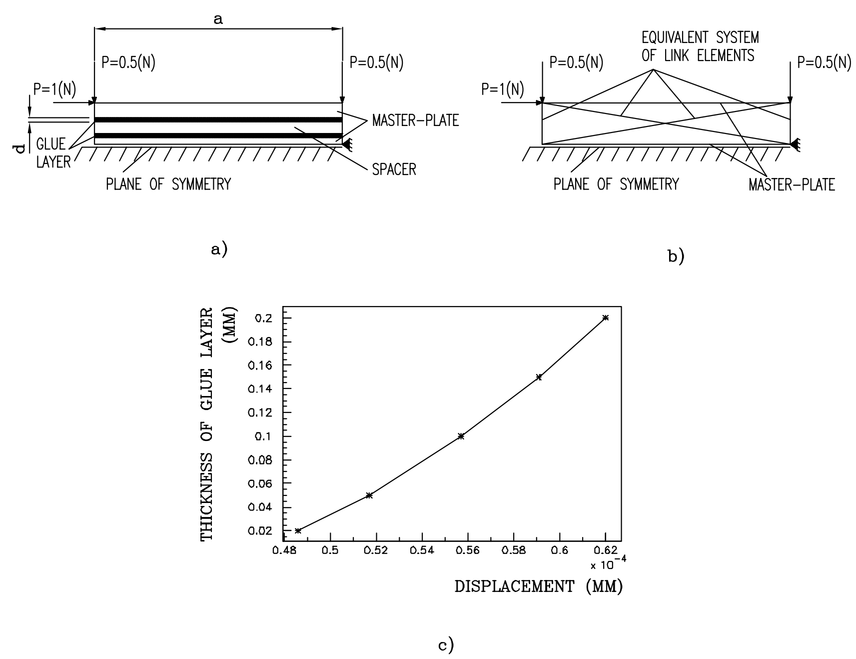

Figure 8: a) Masters and spacers connected by glue; b) Equivalent system of finite elements; c) Thickness of glue layer-displacement curve.

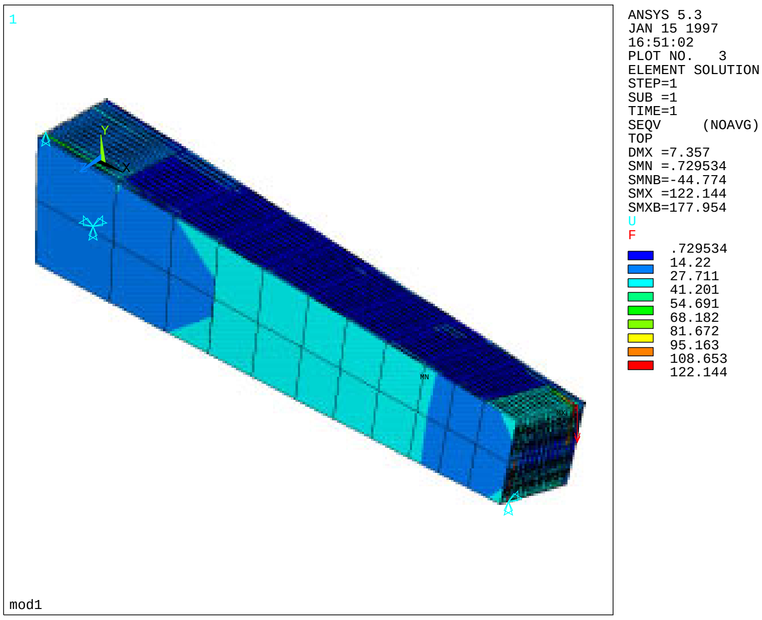

Figure 9: Stress distribution in the master-plates.

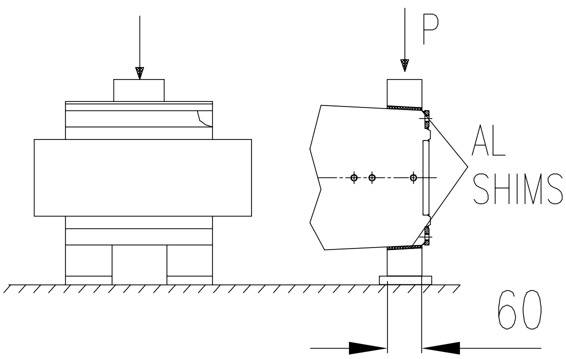

Figure 10: Set-up for testing shear load between master-plates.

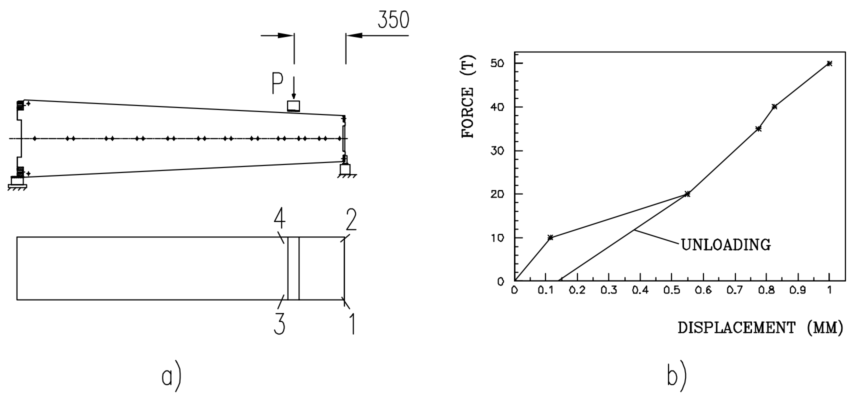

Figure 11: a) Set-up for testing bending load; b) Force-displacement curve.

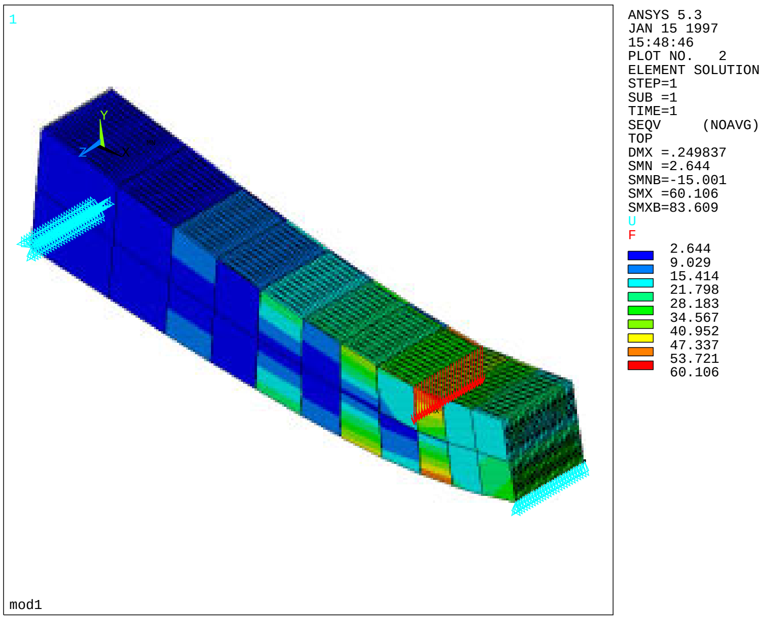

Figure 12: Stresses and deformation of the sub-module in the bending load test.

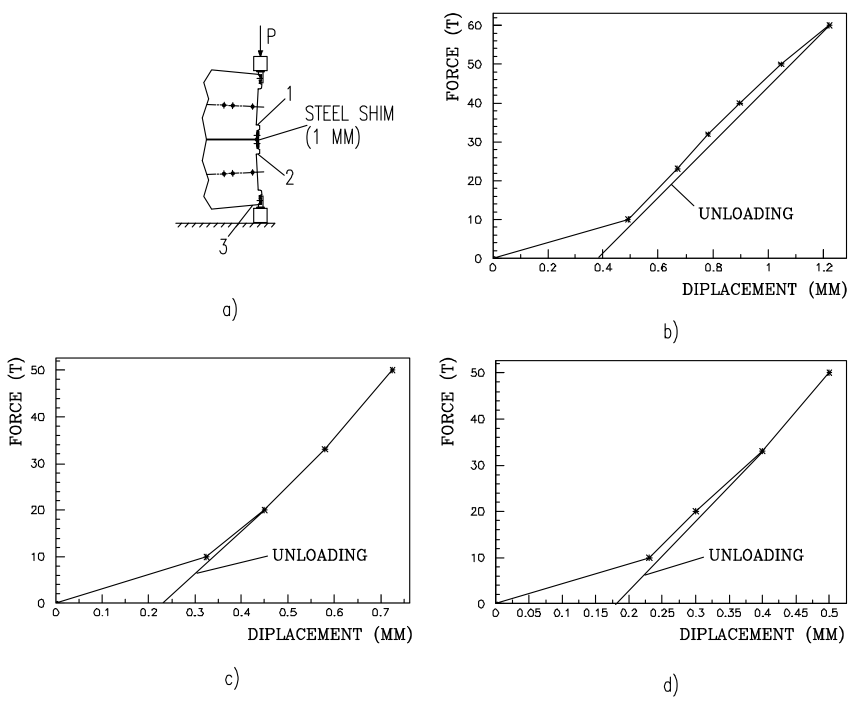

Figure 13: a)Set-up for testing two sub-modules; b) Force-displacement (the point 1 and point 2) curve for the first loading ; c) Force-displacement (the point 1 and point 2) curve for the second loading; d) Force-displacement (the point 3) curve for the second loading.

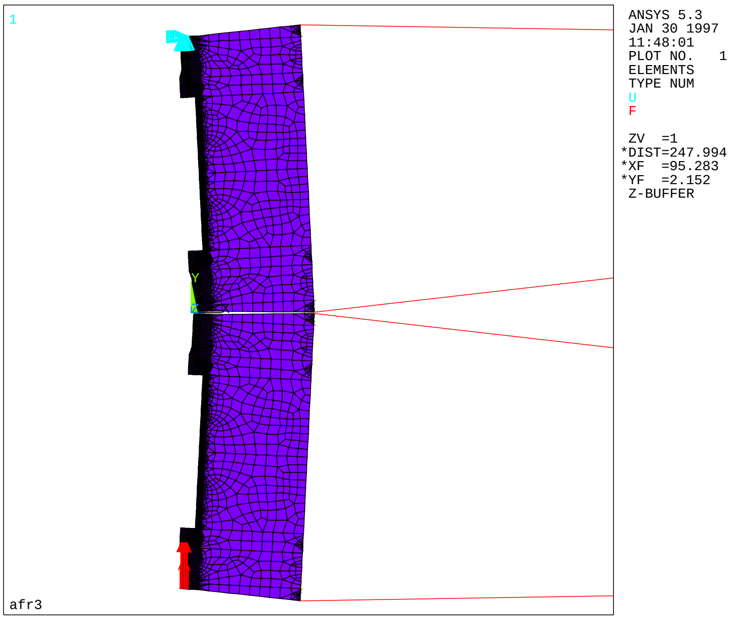

Figure 14: The finite element model of two sub-modules.

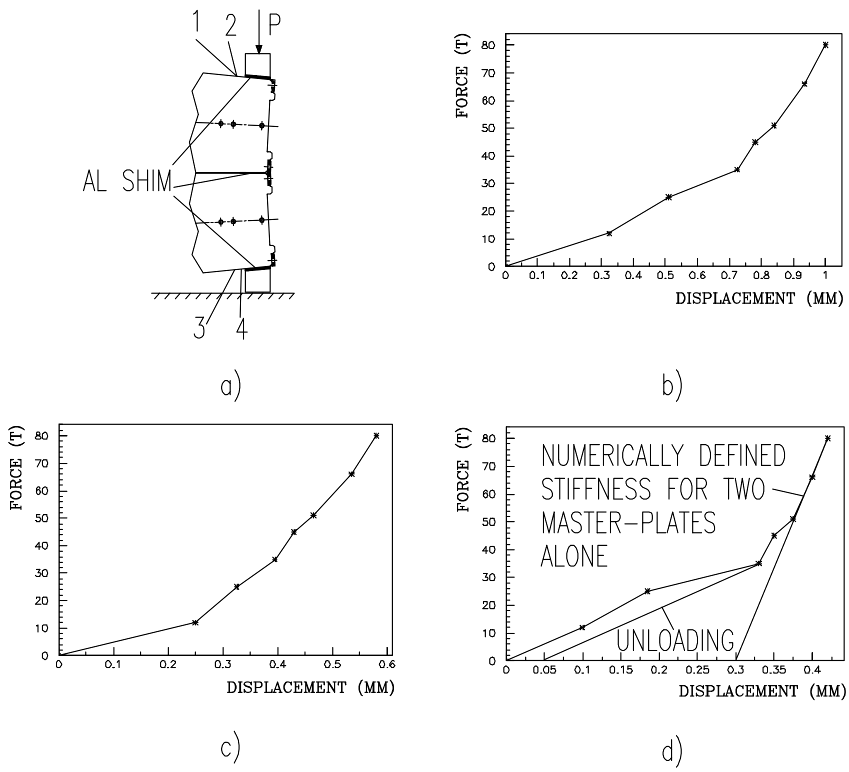

Figure 15: a) Set-up for testing master-plates of two modules subjected to bearing load; b) Force-displacement curve for the top surface; c) Force-displacement curve for the bottom surface; d) Force-displacement curve for the difference between the top surface displacements and the bottom surface displacements.