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Development of an all composite drive shaft

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Summary

The use of composites in primary aerospace structures is increasing gradually. Until recently one of the most important reasons for using composites instead of metals for these structures was the reduction of weight. However, the last few years a change from 'Design for Minimal Weight' to 'Design to Cost' can be observed. The main goal of this 'Design to Cost' approach is to achieve a reduction in number of parts of a structure. A way to realize this cost reduction is to fuse several parts to one by developing new composite materials and fabrication concepts for these materials. One of these fabrication techniques is creating a preform by overbraiding in conjunction with 'resin transfer moulding' (RTM).

An example of such a part is a composite drive shaft with integrated flanges as a replacement for a shaft made by filament winding with assembled preserved aluminium flange parts. The potential advantages of these whole composite drive shafts are (besides a reduction in weight) a reduction in fabrication and maintenance costs. The purpose of the programme was to demonstrate, in a relatively short period, the feasibility of a composite drive shaft with integrated flanges as a replacement for an assembled composite/metal version.

The FEM analysis has been performed with Patran - Nastran 2005 r2. The geometry of the FEM-model is derived from a 3D Catia model and translated to a mid plane surface model. To simplify the FEM model without losing accuracy, only one half of the shaft was modelled. After establishing the design a buckling and an Eigen frequency analyses was performed to check the stability of the shaft.

To minimise development costs, without losing examination of the production method, it was decided to produce only one half of the shaft. An RTM mould and a modular overbraiding concept have been developed. Three composite half shafts were fabricated. One half shaft was subjected to a test programme in which the shaft was loaded to Design Ultimate Load in both torsion directions. The shaft did not fail at this load level.

The weight of the whole composite shaft was 1633 g whereas the shaft with the metal flanges weighted 1996 g, which means that a weight reduction of 18 % has been realized. Because no data was available on the costs of the shaft with the metal flanges no cost comparison between the whole composite shaft and the shaft with the metal flanges could be made.

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1 Introduction

The use of composites in primary aerospace structures is increasing gradually. Until recently one of the most important reasons for using composites instead of metals for these structures was the reduction of weight. However, the last few years a change from 'Design for Minimal Weight' to 'Design to Cost' can be observed. The main goal of this 'Design to Cost' approach is to achieve a reduction in number of parts of a structure. A way to realize this cost reduction is to fuse several parts to one by developing new composite materials and fabrication concepts for these materials. One of these fabrication techniques is creating a preform by overbraiding in conjunction with 'resin transfer moulding' (RTM).

The overbraiding concept is based on a core that contains the inner shape of a composite product. The core is positioned in an overbraiding machine were bobbins, wound with dry carbon fibres, are moving around the core creating a well draped tight fibre reinforcement. It is a computer controlled process where the fibre angles can be adjusted while braiding. This creates the possibility to give the material adjustable mechanical properties. Braiding several layers creates a preform with a certain thickness.

The RTM fabrication concept is based on the injection of resin into a mould cavity containing this preform. During the injection process, air in the mould cavity is replaced by resin and the fibres are impregnated. The RTM process has been in use within the automotive industry for many years for limited-production run parts where the cost of tooling for pressed steel construction would be prohibitive, e.g. for sports cars and special purpose vehicles. RTM has also been in use in the aerospace industry for many years for secondary parts such as radomes and flap track fairings. However, until recently, RTM has not been used routinely in the aerospace industry for primary structures because of the lack of high-quality RTM resins and the lack of available material databases are adequate for structural substantiation and certification.

Now that high-quality RTM resins are becoming commercially available, RTM is becoming increasingly popular in the aerospace industry. The main improvement of these new RTM resins (besides their improved mechanical properties) is that they have low viscosity for a reasonable time, enabling large products with high fibre volume fractions to be produced without the use of excessively high injection pressures. Although RTM moulds often are very complex and expensive, RTM has several advantages compared to autoclave prepreg fabrication concepts which, at this moment, are the standard used in aerospace industry. Some of these advantages are as follows:

- Net shaped products can be made, reducing the amount of trimming required for the cured product.
- Two-sided tooling concepts can be used assuring tight outer-dimensional tolerances, reducing the amount of shimming during assembly.
- No high capital investments (for instance an autoclave) are required.
- Three-dimensional double-curved products can be fabricated which cannot be made using standard autoclave fabrication techniques.

An example of a three dimensional double curved composite component is a composite drive shaft with integrated flanges as a replacement for a shaft made by filament winding with assembled preserved aluminium flange parts. The potential advantages of these whole composite drive shafts are (besides a reduction in weight) a reduction in fabrication and maintenance costs. A major reduction in fabrication costs can be achieved by creating the whole shaft out of one preform using the overbraiding technology. The RTM mould for such a part can be relatively simple and is refunding for small series in comparison to assembled parts. The reduction in maintenance costs can be attributed to the excellent fatigue properties of composites and the absence of risk for corrosion occurring between metal and composite parts.

In the investigation described in this paper a composite drive shaft with integrated flanges for rotorcraft application has been developed. Helicopters and tilt rotor aircrafts have similar drive shafts, to connect rotors to engines, or rotors to rotors. High revolution numbers, up to 9000 rpm, can be observed. The shaft was fabricated by overbraiding technology in conjunction with RTM. To evaluate the composite design it was compared with an existing composite shaft made by filament winding with mounted metal flanges as its counterpart. The main goal of the investigation was to demonstrate the feasibility of integrated composite flanges as a replacement for mounted metal flanges. The work is carried out in co-operation between NLR, Onera and Eurocarbon.

2 The composite shaft made by filament winding with mounted metal flanges.

The composite shaft with mounted metal flanges, which was used as reference for this study, is presented in Fig 1. The composite tube was made with filament winding technology and was made of carbon fibres and epoxy resin. The flanges were made of anodised machined aluminium and were mounted on the tube

Figure 1: Composite shaft made by filament winding.

with blind rivets. The weight of this assembled shaft was 1996 g. In service the shaft was connected on both ends to a flexible coupling connected by three M10 titanium precision bolts. The design ultimate load for this shaft was 2.1 kNm in torsion. This torsion load was introduced through these three precision bolts.

3 Composite materials used.

The laminate of the composite shaft was composed of angle plies and longitudinal fibres (see Fig. 2). The following materials were used for the composite shaft with integrated flanges:

1. Torayca T800H 12K (445 g/1000m) carbon fibres - these fibres have a fair stiffness and an excellent tensile strength which is the perfect match for overbraiding where the fibres are pulled through the machine making several sharp bends. Fibres with a higher stiffness would give unacceptable fibre breakage during overbraiding. The T800H fibres are used for the angle fibres of the braid.
2. Due to the lack of T800H 24K fibres it was decided to use Tenax IMS 5131 24K (820 g/1000m) for the longitudinal fibres. The mechanical properties of these fibres are equal to the T800H fibres and therefore perfect for overbraiding.

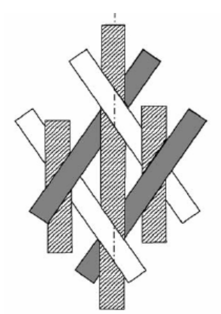

Figure 2: Braided ply.

Table 1. Test results (Onera).

E_{11}	65,5 GPa
E_{22}	46,3 GPa
ν_{12}	0,399
G_{12}	25,5 Gpa

High-temperature curing epoxy resin RTM 6 from Hexcel was used to impregnate the fibres. The material properties of the materials used (needed as input for the finite-element analysis) were determined by testing tension and compression specimens (Onera). The specimens where fabricated by RTM and had a fibre volume fraction of 58,5%. All tests were performed at ambient conditions. The results of these tests are presented in Table 1.

4 Analyses of the composite drive shaft with integrated flanges

The purpose of the programme described in this paper was to demonstrate the feasibility of a composite drive shaft with integrated flanges as a replacement for an assembled composite/metal version. It was decided to use the same number of bolts in the composite flanges as were used in the metal flanges. Pre-tensioned bushes were shrunken into the composite flange holes for two reasons. The first is to prevent hole damage during cyclic loading and the second is lowering the bearing stress by increasing the size of the inside faces of the holes. This solution increases the life cycle of this component.

The composite drive shaft with integrated flanges had to be a functional replacement for the assembled shaft. However, it was allowed that the global geometry of the whole composite shaft differed from the assembled shaft. The overbraiding technology distributes a certain amount of fibres on the circumference of the core. The thickness of the braid is directly proportional to the size of the circumference. Due to this phenomenon one braided layer of the shafts preform will have a variable thickness and therefore the flange will get a tapered shape.

The FEM analysis has been performed with Patran - Nastran 2005 r2. The geometry of the FEM-model is derived from a 3D Catia model (Fig. 3) and translated to a mid plane surface model. To simplify the FEM model without losing accuracy, only one half of the shaft was modelled. The model is meshed with quadrangle shell elements (see Fig. 4).

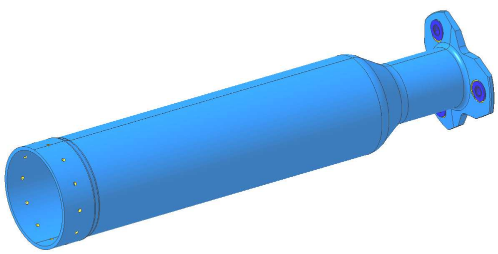

Fig. 3: 3D Catia model

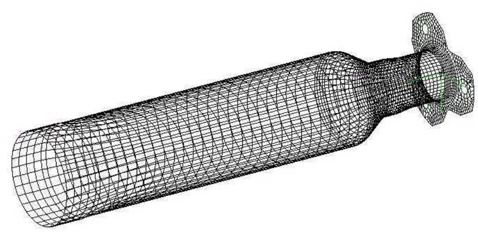

Fig. 4: Meshed FEM model.

The bolt-loaded holes were modelled by applying a displacement load on the nodes (see Fig. 5). The three loads are representing perfect torsion on the shaft. Because only one half of the shaft was modelled, the other side of the model is locked in all six degrees of freedom.

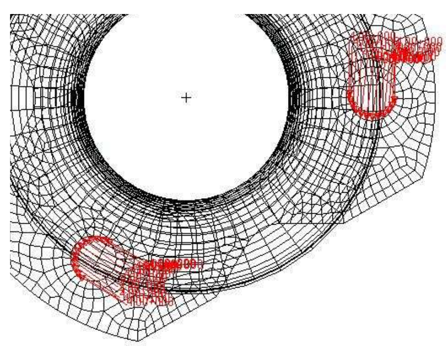

Figure 5: Displacement loads.

The bolt-loaded holes were not modelled in detail. To design these holes in the shaft, conside-ring the use of the shrunken pre-tensioned bushes, the following design stress levels were used [1]: $\sigma_{bearing}$ = 400 MPa, $\tau_{shear-out}$ = 90 MPa and $\sigma_{tension}$ = 388 MPa. These stress levels, in combination with the diameter of the holes, determine the dimensions required for the shaft near the holes (see Fig. 6). These minimum dimensions were used in the determination of the outside diameter of the flange and were set to values which ensured a bearing failure mode as this failure mode has a fail-safe character [1].

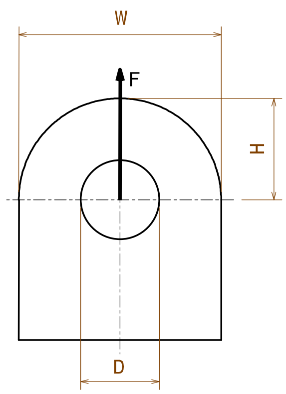

Figure 6: Sizing of a bolt-loaded hole

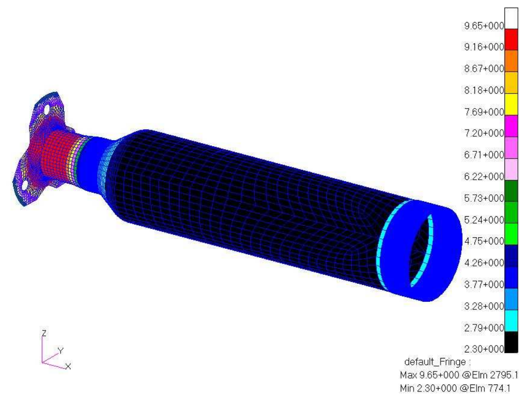

Figure 7: Thickness variations of the model.

To prevent high bearing stresses it was needed to increase the thickness of the flange. Due to this addition, together with the regular thickness variations, the shaft was divided into nine sections (see Fig. 7).

Fig. 8 en 9 shows the stress distribution of the model. Note the stress concentrations in the neck of the shaft. Due to the thickness variations in the neck, several stress levels are obtained. The tubular part of the shaft has a lower stress level and is designed for stiffness to prevent damage, caused by vibrations at higher revolution numbers (Eigen frequency). Fig. 10 presents the dimensions of the shaft.

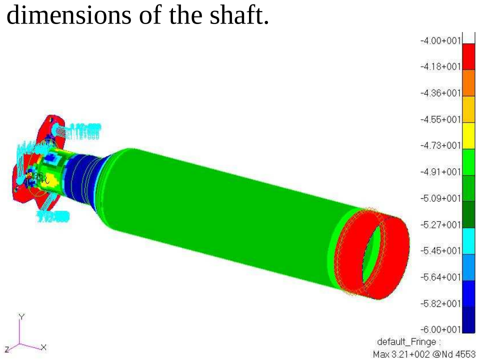

Figure 8: Stress distribution.

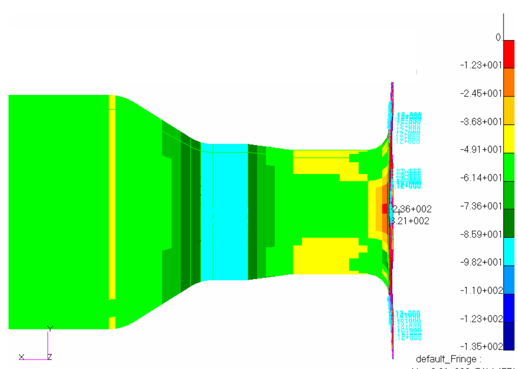

Figure 9: Strain distribution

Finally a buckling and an Eigen frequency analyses was performed to check the stability of the shaft. Fig. 11 presents the first buckling mode which occurred at 4x Design Ultimate Load. Fig. 12 presents the first Eigen frequency mode which occurred at 2,3x the maximum revolution number.

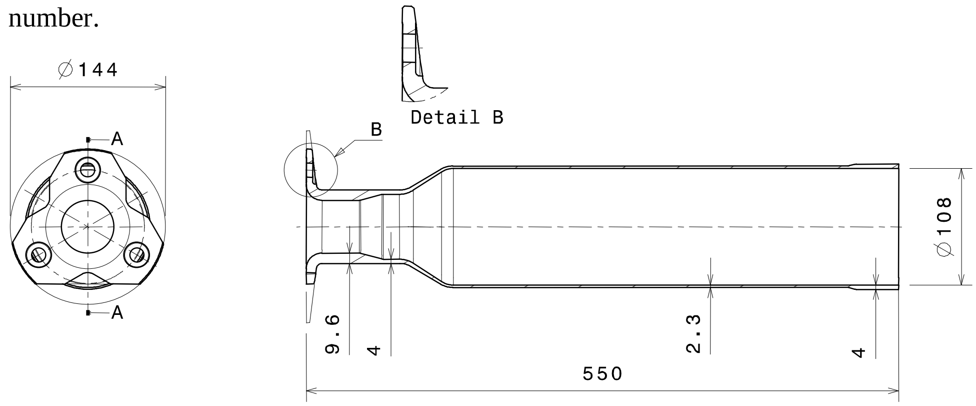

Figure 10: Dimensions of the shaft.

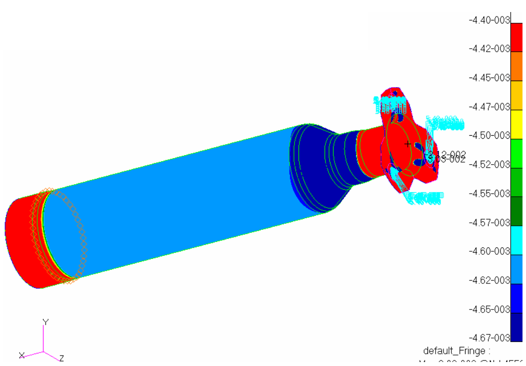

Figure 11: Buckling.

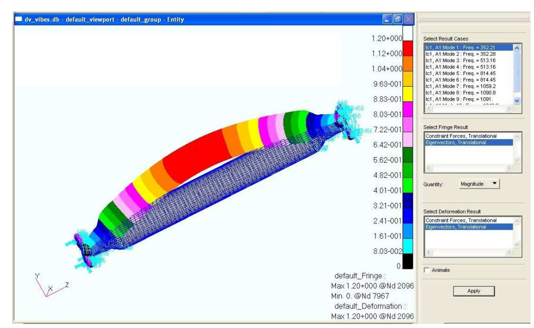

Figure 12: Eigen frequency.

5 Fabrication of the shaft

The purpose of the programme described in this paper was to demonstrate the feasibility of a composite drive shaft with integrated flanges as a replacement for an assembled composite/metal version. To minimise development costs it was decided to produce only one half of the shaft.

A cost analysis showed that it was wise to overbraid two test shafts in one braiding cycle. Figure 13 represents the different elements of the core that contains two times the inner shape of a test shaft, with the flange parts in the middle. It is not possible to overbraid a core that contains an angle of inclination steeper than 45° without losing control about the thickness of

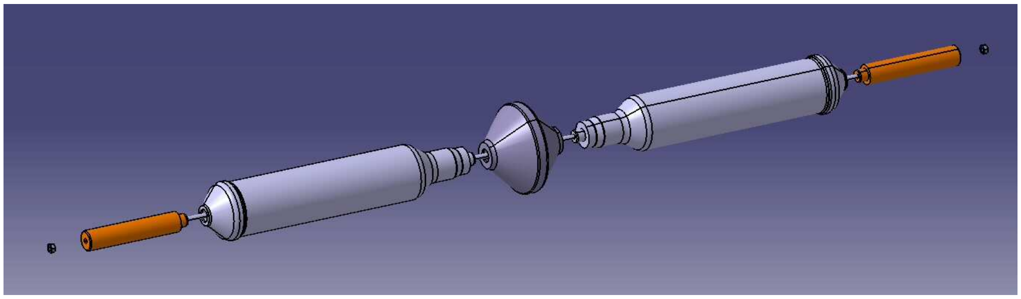

Figure 13: Wooden core

the layer. Therefore the flange parts are braided over this 45º shape and must be formed into the 90º shape afterwards. To save production costs the contour parts (grey) were made of wood (MDF) finished with a coat of varnish. Because of the modular character of the overbraiding core, the braid could easily be taken of by cutting through the braid between both flange parts before disassembling the overbraiding core.

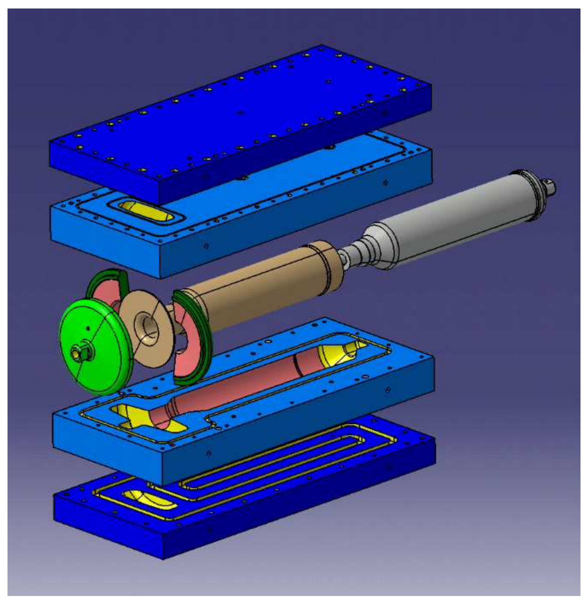

Figure 14: RTM mould

Figure 14 represents the different elements of the RTM mould. All parts are made of steel with the exception of the core (grey) which was made of aluminium. Aluminium was selected because of its high coefficient of thermal expansion which eases demoulding of this mould element after curing the shaft. Because of the modular character of both overbraiding core and RTM mould, the braid could be easily taken from the wooden core and positioned on the metal version. After positioning the braid, the flange could be formed with a form tool. This is a laborious part of the fabrication and should be examined for automation. For the feasibility study it was a sufficient method for getting the flange into its shape.

After cutting the preform (beige) the steel shells (green) were positioned around the flange of the preform. Together with the aluminium core and the shells the preform was positioned in the cavity of the lower mould part. Then the mould was closed by bolting the upper mould part on the lower mould part. Resin was injected through a hole in the lower mould part and is guided through the shells to create a circular injection front. Four vents, located in the tubular part of the shaft, were used to evacuate the air during resin injection.

Based to cost estimation, it was decided not to fabricate the shaft net shaped but to machine the cured shaft to the required dimensions, as cutting the preform to the net shaped dimensions without fibre distortion at the edges would become very difficult, time consuming and expensive owing to the small dimensions of the flange of the shaft.

Three shafts were fabricated. During resin injection the mould had a temperature of 120°C. Resin was injected without vacuum assistance. The RTM injection pressure, needed to inject the resin, varied from 0,3 bar at the beginning to 9 bar at the end of the RTM process. After 45 minutes the preform was wetted. However, in order to ensure a complete impregnation of the fibres in the preform, the resin injection was continued for 30 minutes. A fibre volume fraction of 58,5% was achieved with a good laminate quality. The C-scans made indicated that there was no entrapped air or dry spots. Figure 14 shows the shaft before machining. Figure 15 shows the shaft after machining with installed bushings and prepared for testing.

Figure 14: Test shaft before machining

Figure 15: Test shaft prepared for testing

The weight of the half shaft after machining including the shrunken pre-tensioned bushes was 924 g. Translating this value to the fully shaft, considering the extended length of the half shaft for introduction of the torsion load, gives a weight of the fully shaft of 1633 g whereas the assembled shaft with mounted metal flanges weighted 1996 g. This means that a weight reduction of 18 % had been realized. Unfortunately, because no data were available on the cost of the assembled shaft no cost comparison between the whole and the assembled shaft could be made.

6 Testing the shaft and test results

One of the three shafts was tested statically in torsion. Four rosettes were used to measure strains during the tests (see figure 15, 16 and 17). The clockwise and counter clockwise torsion loads were introduced to the shaft by a metal lever driven by a hydraulic cylinder. To introduce perfect torsion the shaft is mounted on two flexible couplings on both ends of the shaft. The tests performed were torsion controlled with a velocity of 0,3 kNm/min. The shaft was subjected to the following test programme.

- Test 1 : 0.143 x Torsion design ultimate load clockwise
- Test 2 : 0.143 x Torsion design ultimate load counter clockwise
- Test 3 : 0.286 x Torsion design ultimate load clockwise
- Test 4 : 0.286 x Torsion design ultimate load counter clockwise
- Test 5 : 0.429 x Torsion design ultimate load clockwise
- Test 6 : 0.429 x Torsion design ultimate load counter clockwise
- Test 7 : 0.571 x Torsion design ultimate load clockwise
- Test 8 : 0.571 x Torsion design ultimate load counter clockwise
- Test 9 : 0.714 x Torsion design ultimate load clockwise
- Test 10 : 0.714 x Torsion design ultimate load counter clockwise
- Test 11 : 0.857 x Torsion design ultimate load clockwise
- Test 12 : 0.857 x Torsion design ultimate load counter clockwise
- Test 13 : 1.000 x Torsion design ultimate load clockwise
- Test 14 : 1.000 x Torsion design ultimate load counter clockwise

The shaft was designed for Design Ultimate Load. The shaft did not fail at this load level and was not loaded to failure at this stage because it will be subjected to a fatigue programme in the near future. Fig. 18 represents the principle strains of rosettes 1 and 2 (see Fig. 16) measured during test number 14 (Design Ultimate Load counter clockwise). Fig. 19 represents the principle strains of rosettes 3 and 4, located in the neck of the shaft. Fig. 19 indicates that the calculated strains are lower than the measured strains. This is caused by overbraiding a high amount of fibres on a small circumference. Due to this fact the 55º fibre angle couldn’t be achieved and minor fibre wrinkling was necessary to get the first layers into shape. A lower fibre angle causes less torsion stiffness and a lower fibre volume fraction. Those facts caused a higher strain level compared to the calculated strain level, were the material of the specimens were used as input for the FE analyses.

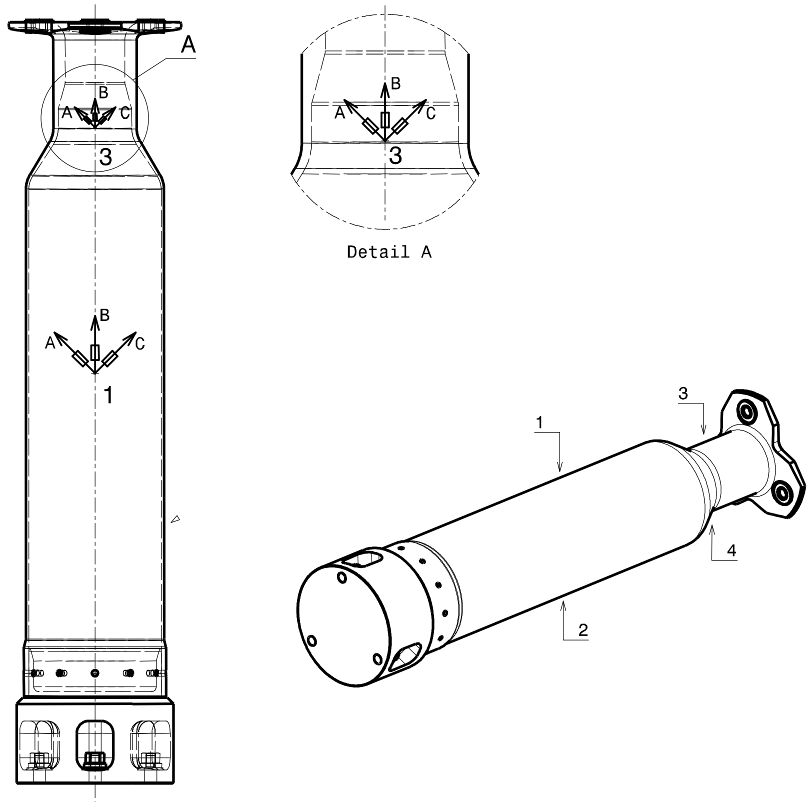

Figure 16: Instrumentation of the shaft.

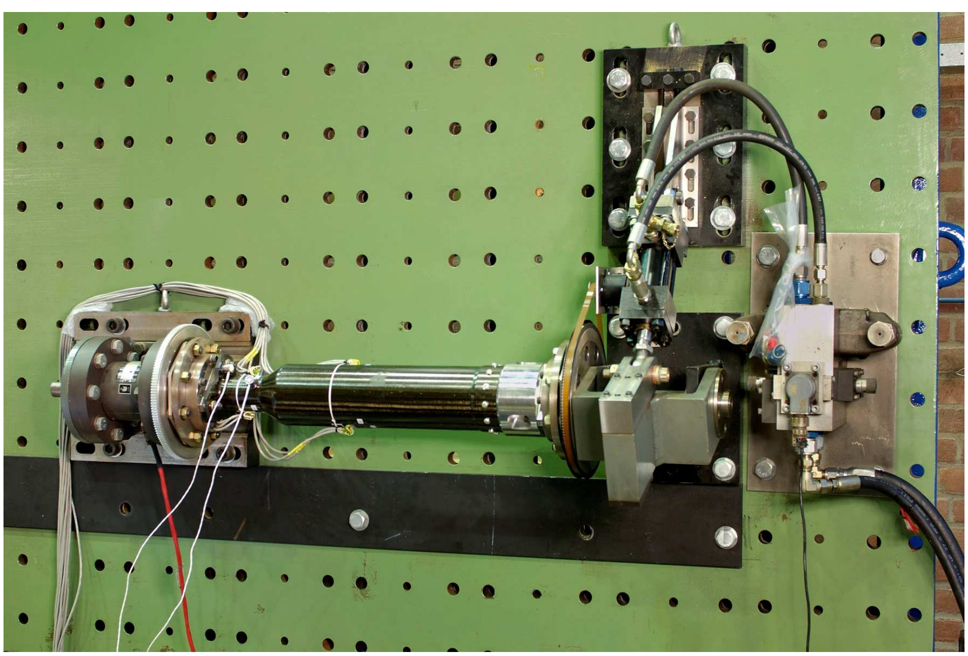

Figure 17: Test set-up

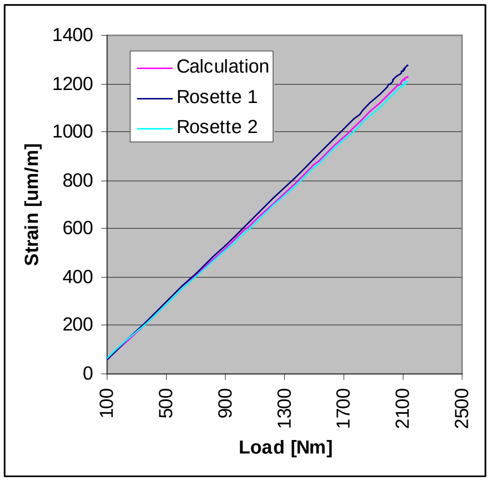

Figure 18: Calculated en measured load-strain levels of rosettes 1 and 2.

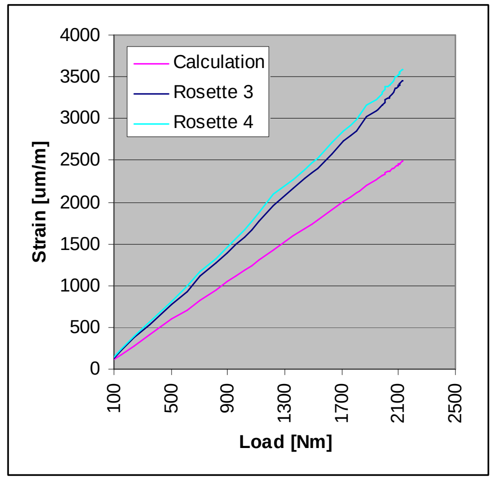

Figure 19: Calculated en measured load strain levels of rosettes 3 and 4.

7 Conclusions

A braided composite drive shaft with integrated flanges for rotorcraft applications as a replacement for a shaft made by filament winding with mounted metal flanges has been developed. The finite-element method was used to design the shaft.
The weight of the whole composite shaft was 1633 g whereas the shaft with the metal flanges weighted 1996 g, which means that a weight reduction of 18 % has been realized. Because no data was available on the costs of the shaft with the metal flanges no cost comparison between the whole composite shaft and the shaft with the metal flanges could be made.
An RTM mould and an overbraiding concept have been developed. Three composite half shafts were fabricated. One half shaft was subjected to a test programme in which the shaft was loaded to Design Ultimate Load in both torsion directions. The shaft did not fail at this load level.

8 References

1. Thuis. H.G.S.J., Design stress levels and rough design rules for CFRP structures with concentrated load introductions – an experimental study. NLR technical report TR 97045 L, 1997.