High load introduction into thick-walled CFRP structures

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Abstract

The subject of an ongoing research project is to substitute a steel gear component by an equivalent carbon fibre reinforced polymer (CFRP) structure. This component belongs to an eccentric drive of a knitting machine which transfers dynamic loads up to 30,000 N to the powering con-rods. The high loads are introduced into the structure by spherical roller bearings. The geometry of the CFRP component is a plane solid structure with a thickness varying between 43 and 48 mm slightly modified in comparison to the original steel component shape. It is planned to apply the same bearings which are used in the existing steel structure. The design of the CFRP structure was accomplished by considering three different load cases which include the mechanical service loads of 30,000 N, the mechanical effort due to the assembly of the roller bearings as well as a bearing temperature difference of +80 K due to in-service conditions. A safety concept was developed to ensure a sufficient reliability of the CFRP component to include manufacturing tolerances, fluctuations of the material properties and service life effects of the structure. The part geometry and the associated load cases were implemented into a finite element model. The results of the numerical analysis were compared to an analytical calculation from which a reasonable agreement was found.

Keywords: CFRP, load introduction, high wall thickness, roller bearing, thermal fitting, force fitting, safety concept.

1 Introduction

Ball and roller bearings are often used to introduce high loads into gear components. In general, these components are made of steel, and the bearings are assembled by force fitting in most cases.
Within an ongoing research project a steel gear component is to be substituted by an equivalent CFRP structure. The integration of conventional roller bearings and the reliable load introduction into the surrounding structure is to be realised.

Two different fitting methods, the thermal and the forced fitting, were examined and compared. A safety concept was developed in order to assure reliability, and a safety factor was defined to determine the allowable stresses. Furthermore, analytical and numerical analyses were carried out to obtain results about the required press fit and the resulting stresses in the individual layers of the CFRP laminate.

2 Part geometry and loading

The geometrical shape of the CFRP gear component and the position of the roller bearings are shown in fig. 1. The CFRP part is a plane solid structure with slight modifications compared to the original steel component shape. The thickness of the component varies between 43 mm at the outer edges and 48 mm in the centre. It is planned to apply the same bearings which are used in the existing steel structure which are spherical roller bearings with an outer diameter of 80 mm in the left and the right positions and a 100 mm bearing in the centre of the part.

The design of the CFRP structure was accomplished by considering three different load cases which include the mechanical loads of 30,000 N, the mechanical effort due to the assembly of the roller bearing as well as a bearing temperature difference of +80 K due to in-service conditions.

![Figure 1: Principle geometrical shape and mechanical in-service loads (geometrical dimensions in mm, forces in N).](image)

The geometry and the load cases were implemented into a finite element model and a finite element analysis (FEA) was performed. The results of the FEA were compared with an analytical calculation which is described in section 4.

A plate of 500 mm x 500 mm x 50 mm was manufactured by vacuum infiltration of an Epoxy resin into 128 dry carbon fibre fabric, twill 2/2, layers.
The stacking sequence was defined as \{[(±45°)/(0°/90°)/(±45°)]_{11S}/(0°/90°)_{18}\} where in plies designated as 0°/90° layers the warp direction of the fabric is oriented parallel to the x-axis of the part (fig. 1). After curing, fibre and void volume fractions of 54 ± 3 % and 1.35 ± 0.5 % were observed.

3 Safety concept

A safety concept was developed to ensure a sufficient reliability of the component which includes manufacturing tolerances, fluctuations of the material properties and the cyclic fatigue behaviour of the CFRP material as the component must be able to sustain 10⁻⁹ load cycles. Due to the lack of experimental data for the applied CFRP material S-N curves of comparable quasi-isotropic CFRP laminates from a database were used [1]. The safety concept is based on two different factors of safety. A general safety factor of 2.0 which considers all manufacturing tolerances and material fluctuation was fixed. The estimation of a factor taking the cyclic fatigue behaviour into consideration is shown in fig. 2. By means of S-N curves from three representative quasi-isotropic CFRP laminates the arithmetic mean of their static strengths were related to their averaged cyclic strengths after 10⁸ and 10⁹ load cycles from which safety factors for fatigue of 2.0 and 2.5 were determined. The global safety factor is equivalent to the product of the individual safety factors (4.0 and 5.0) from which the allowable tensile and compressive stress parallel to the fibre direction and the allowable intralaminar shear stress were defined as 150 and 120 MPa as well as 21 and 17 MPa, respectively.

![Figure 2: Estimation of safety factor (γ) to account for cyclic fatigue.](image)

4 Analytical and numerical analysis of the CFRP component

The press fitting of the roller bearings was designed by defining the diameter of the bearing seat with the ratio of the Young’s moduli of steel and CFRP based on
the experience available for the press fit qualities from the existing steel component. To qualify and compare the assembly of roller bearings into steel and CFRP an analytical model was developed to describe the mechanical behaviour of a press fit. A numerical simulation was carried out to verify the calculated results of the analytical model.

![Figure 3: Press fit “ring-in-ring” model.](image)

The analytical model corresponds to a “ring-in-ring” model as illustrated in fig. 3. This model is based upon the use of the common relation which describes the circumferential tensile stress of pressure vessels [2]. This equation was modified for the combination of an internal steel ring and an external CFRP ring, eqns (1)-(4). It is assumed that the internal ring has a taller outer diameter compared to the inner diameter of the external ring. Equilibrium appears if the external pressure of the inner ring is equal to the internal pressure of the outer ring, eqns (5)-(7). Solving eqn (6) for the contact pressure $p$ and implementing eqn (6) into eqn (7) three major relations can be developed to describe the resulting fitting diameter $D_\infty$ and the contact pressure $p$, eqns (8)-(10).

$$
\sigma_{u,CFRP} = \frac{D_{s,CFRP} \cdot p}{2 \cdot t_{CFRP}}, \quad \varepsilon_{u,CFRP} = \frac{D_{s,CFRP} \cdot p}{2 \cdot E_{CFRP} \cdot t_{CFRP}}
$$

(1), (2)
\[ \sigma_{u,St} = -\frac{D_{S,St} \cdot p}{2 \cdot t_{St}}, \quad \varepsilon_{u,St} = -\frac{D_{S,St} \cdot p}{2 \cdot E_{St} \cdot t_{St}} \] (3, 4)

\[ D_{A,St,p} = D_{1,CFRP,p} = D_{\infty} \] (5)

\[ D_{1,CFRP,p} = D_{1,CFRP} \cdot (1 + \varepsilon_{u,CFRP}) \] (6)

\[ D_{A,St,p} = D_{A,St} \cdot (1 + \varepsilon_{u,St}) \] (7)

\[ D_{\infty} = \frac{D_{A,St} \cdot D_{1,CFRP} \cdot (1 + K)}{D_{A,St} \cdot K + D_{1,CFRP}} \] (8)

\[ K = \frac{D_{S,St}}{D_{S,CFRP}} \cdot \frac{E_{CFRP} \cdot t_{CFRP}}{E_{St} \cdot t_{St}} \] (9)

\[ p = \left( \frac{D_{\infty}}{D_{1,CFRP}} - 1 \right) \cdot \frac{2 \cdot E_{CFRP} \cdot t_{CFRP}}{D_{S,CFRP}} \] (10)

The material properties of the CFRP structure were determined applying the classical laminated plate theory (CLT) based on the stacking sequence as defined in chapter 2. The properties of a single layer were estimated according to the rules of mixture [3] taking a fibre volume fraction of 45% into account. For the single ply a Young’s modulus of 53,000 MPa was determined in the warp and weft directions from which a tensile modulus of 32,600 MPa of the total laminate was calculated.

Figure 4: FE half-model.
Figure 5: Distribution of stresses in (a) $0^\circ/90^\circ$ and (b) $+45^\circ/-45^\circ$ layers along bearing seat perimeter due to mechanical loads, bearing assembly and temperature increase.
The geometry, material properties and the loads were implemented into a finite element model (fig. 4) to carry out a numerical analysis in order to verify the analytical results. The component was modelled using 8-node layered shell elements for the CFRP laminate and 8-node structural shell elements for the outer rings of the roller bearings. In the symmetry axis of the structure constraints with symmetry boundary conditions were applied. Link elements which were arranged in the radial direction and which are designed to transfer compression loads only were used to simulate the load transfer characteristic of roller bearings into the CFRP structure. Furthermore, contact elements between the outer ring of the bearing and the CFRP structure were applied to simulate the press fit and the temperature increase.

The results of the analytical and numerical calculations and the comparison to the steel structure are shown in table 1. It can be seen that nearly the same diameter of the deformed bearing seat was determined for the chosen fitting conditions while a slightly higher contact pressure was estimated compared to the steel configuration.

It could be shown that the diameter of the deformed bearing seat agrees reasonably with the analytical solution. The distribution of the first principle strain is approximately constant and oriented circumferentially around the bearing seat which confirms the assumption of the “ring-in-ring” model.

The evaluation of the stress distribution around the bearing seat was carried out for each layer. In fig. 5 the laminar stresses of the $0^\circ/90^\circ$ and $+45^\circ/-45^\circ$ layers are illustrated for the superposition of all load cases (mechanical loads, bearing assembly and temperature increase) where $\sigma_1$ and $\sigma_2$ correspond to stresses in the warp and weft direction. It can be concluded that the stresses parallel to fibre direction do not exceed 85 MPa and that the shear stresses are not higher than 7 MPa. Furthermore, a sinusoidal stress distribution around the bearing seat is observed for each layer which is primarily influenced by the bearing assembly and the temperature increase.

### Table 1: Results and comparison of the analytical calculations.

<table>
<thead>
<tr>
<th></th>
<th>steel component</th>
<th>CFRP component</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_{LSI}$, $D_{LCFRP}$ mm</td>
<td>79.985</td>
<td>79.960</td>
</tr>
<tr>
<td>$D_\infty$ mm</td>
<td>79.990</td>
<td>79.989</td>
</tr>
<tr>
<td>$p$ MPa</td>
<td>3.87</td>
<td>4.54</td>
</tr>
</tbody>
</table>

### 5 Thermal and mechanical bearing assembly

To ensure a reliable load introduction into the CFRP structure press fitted bearings need to be realised. As a result of the analytical and numerical calculations, the seat diameter was determined to be 79.96 mm with a roller bearing diameter of 80.00 mm which can be classified as 80 R6 h5 according to ISO notation. In order to avoid pre-damaging of the CFRP during assembly due
to micro-cracking and delamination it was first decided to thermally fit the bearings by a sufficient cooling with liquid nitrogen. From the coefficient of thermal expansion of $1.1 \cdot 10^{-5}$ K$^{-1}$ of steel a temperature difference of –70 K is required to shrink the outer ring of the roller bearing into the milled hole within the CFRP structure which was maintained at room temperature. This procedure enabled to adjust and join the bearings without any mechanical effort. On the other hand it turned out to be impossible to demount the bearing without mechanical effort. The reason for that behaviour is the reduced surface area of the assembled roller bearing for liquid nitrogen coverage as well as the interaction with the surrounding CFRP which prevented to reach a sufficient cooling.

As this procedure could not be maintained for later application it was decided to assemble the bearings by force fitting. In fig. 6 the fitting of a roller bearing by using a positioning tool is shown. The positioning tool consists of two sockets, one which adjusts the roller bearing while the other one pushes the bearing into its seat by turning a nut along a threaded rod. After demounting the roller bearing from the CFRP structure a smoothened surface was observed in the contact area in comparison to the remaining hole surface.

![Figure 6: Bearing assembly by force fitting with positioning tool (photograph by courtesy of Oskar Dilo Maschinenfabrik KG).](image)

6 Light microscopy investigation

Light microscopy tests on specimens cut out of the bearing seat area as well as the remaining portion of the milled hole were carried out to look for potential pre-damaging caused by the force fitted bearing assembly as described before. In fig. 7 the sampling points of specimen around the bearing seat are shown. Specimens of the bearing seat area affected by the bearing assembly were
labelled “mL” while unaffected specimens of the remaining hole portion were designated as “oL”.

The microscopy tests of the “oL” specimens showed a typical surface roughness caused by the milling process (fig. 8) with a depth up to \(20 \times 10^{-3}\) mm. Equivalent results were observed by evaluating the “mL” specimens, and, in particular, no indications with regard to matrix cracking, local delaminations or fibre kinking were found (fig. 8).

Figure 7: Specimen position for microscopy investigation (dimensions in mm).

Figure 8: Left: “oL” specimen; right: “mL” specimen.
7 Conclusion

It was shown that higher contact pressure acts between the external ring of a steel roller bearing and the surrounding CFRP structure compared with the former press fit into a steel component. The resulting stresses in the CFRP laminate according to assembly and in-service loads are lower than the allowable stresses determined by applying a conservative safety concept. Finally, it can be concluded that force fitting is a possible assembly alternative for the joining problem at hand without pre-damaging the laminate in the vicinity of the bearing seat.

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