A review of critical issues in the design of lightweight flywheel rotors with composite materials

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Abstract Composite materials are widely used to build high-performance flywheels due to their high material strength and low mass density. The high degrees of freedom in material selection, design, and manufacturing techniques lead to a variety of rotor structures. This paper presents the characteristics of different composite rotors and the critical considerations in terms of designing, manufacturing, and testing them. The introduction starts with the limitations of a single filament-wound composite rim. Then, various rotor structures are presented as well as the critical issues regarding the composite rim design, rim-shaft connection, and rotor failure in order to make safe design recommendations. The aim is to summarize the current techniques and provide references for further developments.

Keywords Composite rotor · Flywheel · High-speed rotor · Lightweight

1 Introduction

Flywheels store kinetic energy in a rotating inertia body. Driven by an electric machine, they can convert kinetic energy into electricity and vice versa. They are used as storage devices in many applications, such as in the utility to stabilize the voltage and frequency [1], in vehicles to accumulate energy from regenerative braking [2, 3], or in aerospace as attitude control actuators for satellites [4], etc. The inertia body of a flywheel can be constructed either by conventional isotropic materials, such as steel, or by lightweight composite materials. In general, for stationary applications, where the weight is not crucial, steel is a good option due to its low costs and mature manufacturing. Examples are the product Powerbridge from company Piller (energy up to 5.8 kWh, max. speed 3600 min⁻¹ [5]) and the product PowerStore from com-
Table 1  Ultimate strength $R_m$, yield strength $R_p$, density $\rho$ and specific strength $R_m/\rho$ of selected materials

| Material                  | $R_p$ [MPa] | $R_m$ [MPa] | $\rho$ [kg/m$^3$] | $R_m/\rho$ [Wh/kg] |
|---------------------------|-------------|-------------|-------------------|--------------------|
| Alloy steel (30CrNiMo8)   | 440         | 500         | 2780              | 50                 |
| Aluminum (AlZnMgCu1.5)    | 700         | 1000        | 7760              | 36                 |
| Titanium (TiAl6Zr)        | –           | 1200        | 4500              | 74                 |
| GFRC [7]                  | 1300        | 2000        | 180               | 180                |
| CFRC (7700+Epox)          | 2450        | 1550        | 439               | 439                |

 GFRC glass fiber reinforced composites, CFRC carbon fiber reinforced composites

Table 2  Shape factor of various rotor geometries. Reprinted from [7], copyright (1985), with permission from Elsevier

| Rotor shape     | Geometry         | Shape factor K |
|-----------------|------------------|----------------|
| Constant stress disk (theoretical) | ![Image](image1) | 1              |
| Constant stress disk (real)    | ![Image](image2) | 0.7 … 0.98    |
| Conical disk       | ![Image](image3) | 0.7 … 0.95    |
| Constant height disk | ![Image](image4) | 0.606         |
| Thin rim          | ![Image](image5) | 0.5           |
| Rim with disk     | ![Image](image6) | 0.4 … 0.5    |
| Pierced constant height disk | ![Image](image7) | 0.303         |

Fig. 1  Geometry of a single rim with circumferentially oriented fibers. ($r_i$ inner radius, $r_o$ outer radius, $h$ axial height, $\Omega$ angular speed)

Company ABB (energy 5 kWh, max. power 1.5 MW, max. speed 3600 min$^{-1}$, rotor mass 3000 kg [6]). However, for higher energy content, lightweight rotors are required, as steel rotors will lead to an extraordinary high weight. Another requirement for lightweight rotors is in mobility or in aerospace, where the weight is strictly constrained. The energy content is usually small, thus the objective is to obtain high specific energy (i.e. energy per mass).

High specific energy can be obtained by high rotor velocity, which is, however, limited by the material strength due to centrifugal forces. As described in (1) by [7], there are two possibilities to increase the specific energy or velocity. One is to use high strength but low density materials, e.g. fiber reinforced composite materials. A comparison of metallic and composite materials is shown in Table 1. The other option is to increase the so-called shape factor $K$, as described in (1). This factor is dimensionless and it corresponds to the stress distribution in the rotor, which depends on the rotor shape (Table 2). If $K=1$, it means the rotor has an optimum shape, that each single volume undergoes the maximum allowable stress. Thus the material is fully utilized, and the rotor can obtain the aimed energy by using the least material.

$$\frac{E}{m} = K \cdot \frac{[\sigma]}{\rho} \propto r_p^2$$

In (1), $E$ stands for the energy content of the flywheel, $m$ is the mass of the inertia body. $[\sigma]$ is the allowable maximum stress, determined by the material strength and a safety factor. $\rho$ is the material mass density. $K$ is the shape factor. For anisotropic material, $K$ also depends on the anisotropy of elasticity modulus and Poisson’s ratio of the material. $v_p$ is the rotor tip velocity.

Steel rotors are typically constructed into the first 4 geometries in Table 2, with a high shape factor from 0.606 to 1. Composite rotors are mostly constructed into a rim shape by orienting the filaments in the circumferential direction (Fig. 1). The radial strength depends mainly on the matrix material, which is much lower. The shape factor is referenced to the strength in the fiber direction. The maximum value is 0.5, which can be only reached by a thin-walled rim. However, a thin-walled rim is normally non-realistic for a storage device, as a quantity of mass is always required to obtain a certain inertia. And a strengthened radial stiffness is also required. However, increasing the wall thickness will increase the radial stress and becomes critical rather than the circumferential direction. The circumferential strength cannot be fully utilized, leading to a low shape factor (e.g. close to 0.1, explained in Section 2). Therefore, regarding (1), one should keep in mind that, using composite materials cannot guarantee high $E/m$, maybe even comparable to metal rotors in case of a low shape factor $K$. The structural design and optimization of composite rotors are important.

Composite materials provide the designers with large degrees of freedom in the material selection, ply design and manufacturing techniques. Thus flywheels applied in the industry are built into a variety of different structures. There are seldom universal de-
design or optimization criteria. However, some design principles and critical issues are in common.

There are many review papers about flywheel technologies [8–11], however very few focus on the critical issues regarding the design and construction of composite rotors. Publication [9] presents general analytical stress calculations for composite rings, showing the problems of low radial strength and radial growth compatibility under centrifugal load. However, the solutions for these problems are not presented in detail. Publication [10] presents some solutions to increase the radial load capability. However, each solution has its drawbacks and limitations, either concerning manufacture or failure modes, which are not discussed in [10] and are also seldom discussed in previous review papers.

This paper provides an overview of the typically used composite rotor structures, focusing on their characteristics and critical issues concerning the design, manufacturing and testing. The purpose is to build a fundamental understanding of the composite rotor technologies and provide an index for relevant literature references. Critical issues introduced in this paper include the composite rim design, rim-shaft connection, rotor failure and safety issues.

The paper is organized in the following way. The topic starts in Section 2 with a single rotating rim, introducing the material orthotropic properties and pointing out the limitations of this structure due to low radial strength. Section 3 presents different solutions to increase the radial load capability. Section 4 introduces the critical issues regarding the rim-shaft connection. Section 5 is concerned with rotor failure and safety issues. A summary and discussions are given in Section 6.

2 Basics of a single rotating rim

2.1 Basic quantities

To start the discussions, some basic variables and formulas are firstly introduced. A single rim with circumferentially oriented fibers has a geometry in Fig. 1, defined by an inner radius \( r_i \), an outer radius \( r_o \), and a constant axial height \( h \). The rotor rotates with an angular speed \( \Omega \). Two key parameters are defined: the aspect ratio \( \alpha \) in (2) and tip velocity \( v_p \) in (3).

\[
\alpha = \frac{r_i}{r_o} \tag{2}
\]

\[
v_p = \Omega \cdot r_o \tag{3}
\]

The inertia \( J \) can be calculated by

\[
J = \frac{1}{2} \cdot m \cdot \left( r_o^2 + r_i^2 \right) = \frac{1}{2} \cdot m \cdot \left( 1 + \alpha^2 \right) \cdot r_o^2 \tag{4}
\]

where \( m \) is the mass. Thus the energy stored in this rim is

\[
E = \frac{1}{2} \cdot J \cdot \Omega^2 = \frac{1}{4} \cdot m \cdot \left( 1 + \alpha^2 \right) \cdot v_p^2 \tag{5}
\]

One can obtain the specific energy

\[
E = \frac{1}{m} = \frac{1}{4} \cdot \left( 1 + \alpha^2 \right) \cdot v_p^2 \tag{6}
\]

The specific energy \( E/\rho \) is one of the most important evaluation indicators for a lightweight flywheel rotor. Another indicator named energy density, i.e. energy per space volume, is defined by

\[
E = \frac{E}{V_o} = \frac{\rho}{4} \cdot \left( 1 - \alpha^4 \right) \cdot v_p^2 \tag{7}
\]

where \( \rho \) is the mass density. \( V_o \) is the space volume in terms of rotor outer layout, with

\[
V_o = \pi \cdot r_o^2 \cdot h \tag{8}
\]

As can be seen, the ratio \( E/\rho \) and \( E/V_o \) are both depending only on the defined aspect ratio \( \alpha \) and tip velocity \( v_p \). The formulas (2) ... (8) are also valid for \( \alpha = 0 \), corresponding to a solid disk or cylinder without central bore.

2.2 Stress in a single rotating rim

Based on the plane stress assumption, which is valid for a small axial height \( h \) with respect to the radial dimension, the stress in the rim under the centrifugal load can be calculated by [7]

\[
\sigma_r = \rho \cdot v_p^2 \cdot H^r_o (\alpha, r/r_o) \tag{9}
\]

\[
\sigma_r = \rho \cdot v_p^2 \cdot H^r_o (\alpha, r/r_o) \tag{10}
\]

where subscripts \( r \) and \( \theta \) denote tangential and radial direction, respectively. The specific stress \( H^r_o \) depends on the aspect ratio \( \alpha \) and material properties (Appendix A).

Taking the composite material: carbon fiber Toray T700 (volume fraction \( \varphi = 0.6 \)) with epoxy resin as an example (properties shown in Appendix B), the distributions of the specific stress \( H^r_o \) for various \( \alpha \) are shown in Fig. 2. The tangential stress is much higher than radial stress. The maximum radial stress occurs in the middle area of the rim. The maximum tangential stress occurs in the middle for a thick rim (low \( \alpha \)) and occurs on the inner surface for a thin rim (high \( \alpha \)). The maximum specific stress for each \( \alpha \) is written as \( H^r_{o, \text{max}} (\alpha) \) and \( H^t_{o, \text{max}} (\alpha) \). Thus, the maximum stress can be calculated by (11) and (12), depending only on \( \alpha \) and \( v_p \), regardless of the geometrical parameters.

\[
\sigma_{r, \text{max}} = \rho \cdot v_p^2 \cdot H^r_{o, \text{max}} (\alpha) \tag{11}
\]

\[
\sigma_{r, \text{max}} = \rho \cdot v_p^2 \cdot H^t_{o, \text{max}} (\alpha) \tag{12}
\]
2.3 Maximum velocity

The maximum velocity is limited by material strength and associated failure criteria. Due to the axisymmetric geometry, theoretically no shear stress occurs for a constant speed rotation [7]. Therefore, the maximum stress failure criterion can be used. A safe design should guarantee that the maximum stress remains lower than the material strength, taking a safety factor into account.

\[
\sigma_{0,\text{max}} \leq \sigma_0 = R_0^r / k_{sf}
\]

\[
\sigma_{r,\text{max}} \leq \sigma_r = R_0^r / k_{sf}
\]

where \( k_{sf} \) is the safety factor. \( R_0^r \) and \( R_0^t \) are tensile strengths in the fiber direction and transverse to the fiber direction (Table B.1). \( [\sigma_0] \) and \( [\sigma_r] \) are stresses allowed in tangential and radial direction.

By substituting (11) and (12) into (13) and (14), one can obtain the maximum velocity \( v_{p,0}(\alpha) \) in (15) and \( v_{p,r}(\alpha) \) in (16). They are limited by the material strength in tangential and radial direction, respectively. Therefore, the allowable velocity should be lower than both limitations [12], as described in (17).

\[
v_{p,0}(\alpha) \leq \sqrt{\frac{R_0^r / k_{sf}}{\rho \cdot H_{r,\text{max}}(\alpha)}}
\]

\[
v_{p,r}(\alpha) \leq \sqrt{\frac{R_0^t / k_{sf}}{\rho \cdot H_{r,\text{max}}(\alpha)}}
\]

\[
v_p(\alpha) \leq \min\{v_{p,0}(\alpha), v_{p,r}(\alpha)\}
\]

For the chosen material and a safety factor \( k_{sf} = 2 \), the velocity limitations are shown in Fig. 3. For a thin rim (large \( \alpha \)), it is limited by the tangential strength. Higher velocity may lead to a failure such as fiber fracture. For a thick rim (small \( \alpha \)), the radial strength is critical, which mainly depends on the “weak” matrix material. The potential failure is radial delamination. The maximum available velocity is the intersection of these two boundary lines: \( v_{p,\text{max}} = 947 \text{ m/s at } \alpha = 0.738 \).

At this velocity, the stress reaches the maximum value in both directions, indicating a maximum material utilization.

It is noted that the maximum tip velocity depends only on material properties, and can be only achieved for a unique aspect ratio \( \alpha \). For example, for the material M46 in Table B.1, the maximum tip velocity is \( v_{p,\text{max}} = 881 \text{ m/s}, \) achieved at \( \alpha = 0.767 \).

2.4 Specific energy, energy density and shape factor

According to (6) and (7), the specific energy \( E/m \) and energy density \( E/V_0 \) are calculated for various \( \alpha \) and \( v_p \), and shown in Fig. 4 and 5, as well as the velocity boundaries.

High \( E/m \) can be achieved by high \( v_p \) for a thin rim (marked by a red circle). The maximum \( E/m \), which is 110 Wh/kg, can be obtained for \( \alpha = 1 \) and \( v_p = 889 \text{ m/s} \). However, this combination results in a zero energy density \( E/V_0 \) in Fig. 5. (The reason is, for a limited mass \( m \), when \( \alpha \rightarrow 1 \), the axial height \( h \rightarrow \infty \), thus \( V_0 \rightarrow \infty \).) The optimum combination is determined to be the maximum velocity \( v_p = 947 \text{ m/s with } \alpha = 0.738 \).

Fig. 2 Calculated specific stress \( H^* \) in a single rim for composite material T700 with epoxy resin. \( r_o \) outer radius, \( r_i \) inner radius, \( \alpha = r/r_o \) aspect ratio, \( \rho \) mass density, \( v_p \) tip velocity, \( \theta \) tangential direction, \( r \) radial direction)

Fig. 3 Calculated velocity limitations by (15) and (16) for material T700 with epoxy resin. \( \alpha = r/r_o \) aspect ratio, \( v_p \) tip velocity, \( \theta \) tangential direction, \( r \) radial direction

Fig. 4 Calculated specific energy \( E/m \) (in Wh/kg) by (6) for the composite material with fiber T700. \( \alpha = r/r_o \) aspect ratio, \( v_p \) tip velocity, \( \theta \) tangential direction, \( r \) radial direction

Fig. 5 Calculated energy density \( E/V_0 \) by (7) for the composite material with fiber T700. \( \alpha = r/r_o \) aspect ratio, \( v_p \) tip velocity, \( \theta \) tangential direction, \( r \) radial direction
where the energy density $E/V_o$ reaches the maximum value and high specific energy $E/m = 96.2$ Wh/kg (slightly lower than the maximum $E/m$) can be also achieved.

The shape factor $K$ can be also calculated by (18) according to (1). The quantity $[\sigma]$ is referred to the circumferential direction, which is calculated by (13).

$$K = \frac{E/m}{[\sigma]/\rho}$$

(18)

The calculated $K$ is plotted in Fig. 6, which has the same distribution as $E/m$ in Fig. 4. A high shape factor ($K \approx 0.5$) can be only achieved at a high velocity by a thin rim. A thick rim leads to a low shape factor ($K \approx 0.1$), due to the low radial strength. One can take the steel 30CrNiMo8 in Table 1 for comparison. The material specific strength $R_m/\rho$ of the composite material (T700 + Epoxy) is 12 times higher than the steel. However, with $K = 0.1$, which is 1/6 of a constant height disk-shape steel rotor, the specific energy $E/m$ of a composite rim is only 2 times of the steel rotor. The benefit of using high-strength composite material is not significant.

2.5 Limitations of a single rim design

Even though using a thin rim can achieve a high shape factor $K$ and high $E/m$, in practice it is only applied in the flywheels with very small energy content (e.g. approx. 0.06 kWh in [12]). For high energy content, a big axial height is required. A thin rim may lead to low eigen-frequencies and buckling problems. The rotor usually requires sufficient radial stiffness considering the dynamic behavior. Meanwhile, a thin rim has larger radial deformation than a thick rim, which brings difficulties for the fixation onto the shaft. Therefore, in most flywheels thick rims are typically preferred. However, they have two main problems: one is the low shape factor $K$ as stated above. The other is the low manufacturing quality. Due to different thermal behaviors of fibers and matrix, residual stress arises inside the body during the curing process, which may lead to delaminations or cracks. Even though this can be improved by a stepwise layer-by-layer winding and curing process, more manufacturing time and costs have to be invested.

To solve these problems, a variety of designs and methods are proposed, which are summarized and presented in Section 3. The relevant critical issues are also introduced.

3 Increase the radial load capability

As stated above, the main limitation of a single filament-wound composite rim is the low radial load capability. To improve this, one can try to reduce the radial stress. One idea is to pre-load the rim with a compressive stress at standstill, so that it will counteract the tensile stress due to centrifugal load. A pre-loading can be realized in different ways, e.g. by using shrink fitted multi-rims or by pre-tightening the fibers during winding process. The second possibility to vary the stress distribution is by varying the rotor geometry profile [13], or by varying the material prop-

Table 3 Characteristics and critical issues in different composite rotors

| Composite rotor | Characteristics and critical issues |
|-----------------|------------------------------------|
| Single rim      | Low radial load capability         |
| Shrink fitted multi-rims | Rim detachment, overstress, assembly difficulties, long-term failure: stress relaxation & creep |
| Pre-tightened wound rim | Quality control (thick rim) |
| Shape-optimized rotor | Special mould design |
| Varied density and stiffness | Manufacture complexity |
| Multi-directional reinforced composites | |
| Stacked-ply rotor (2D) | Complicated fatigue behavior |
| Woven fabric rotor (2D) | Low uniformity, imperfections at fabric tail |
| 3D composite rotor | Local micro-cracking caused by $z$-bundles |
erties during winding process, e.g. changing the fiber content, winding angle, etc [7]. The third possibility is, instead of manipulating the stress distribution, to increase the radial strength by orienting fibers in radial direction. Each approach has its characteristics and the corresponding critical issues regarding the design and manufacture, which are summarized in Table 3.

3.1 Shrink fitted multi-rim design

Instead of one single thick rim, one can pre-fabricate multiple thin rims and assemble them afterwards by interference fittings, thus producing an internal radial compressive stress in each rim layer at standstill, which counteracts the centrifugal forces.

The effect is briefly explained here by an example of a rim with \( \alpha = 0.6 \) in Fig. 4, designed for a rotational speed of 24,000 min\(^{-1}\). Instead of a single rim, it is composed of two rims with identical thicknesses and assembled with an interference fitting of –0.64 mm. This value is the minimum fitting in order to maintain their attachment at rotation. It produces a compressive contact at standstill. At rotational state of 24,000 min\(^{-1}\), Fig. 7 shows the specific stress in this two-layer rim compared to the single rim. The specific radial stress is reduced from 0.047 to 0.015 (by approx. 67.4%). This means that the velocity can be further increased from 689 m/s to 1220 m/s (by 77%) according to (16). However, the maximum tangential specific stress in the outer rim is increased from 0.75 to 0.92 due to the interference fitting. This causes the maximum allowable velocity decreases from 1026 m/s to 925 m/s according to (15). Thus the tangential stress becomes more critical.

For a better understanding, Fig. 8 and 9 show the comparison of the single rim and two-layer rim in terms of specific energy \( E/m \), energy density \( E/V_0 \) and shape factor \( K \). By using a two-layer rim, the velocity boundary limited by the radial stress is increased, while the limitation in tangential direction is decreased. Thus a new feasible region is created. Compared to the single rim design, the tip velocity at low \( \alpha \) significantly increased, thus higher \( E/m, E/V_0 \) and \( K \) can be also achieved for thick rims. Especially for \( E/V_0 \), a maximum value of 92.5 Wh/dm\(^3\) can be obtained for \( \alpha = 0.475 \) and \( v_p = 952 \) m/s, which can be never reached by a single rim (maximum 67.9 Wh/dm\(^3\)).

However, one should also note that, even though multi-rim design improves the performances at low \( \alpha \), the specific energy and shape factor are always lower than a thin rim, due to the additional increased tan-
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K. During the test, it was found that below 36,000 min⁻¹, the rotor showed a slight balance change as the rims started to separate from each other. Further increasing the speed to 47,500 min⁻¹ (1120 m/s) resulted in a rapid imbalance due to further rim-to-rim and rim-to-shaft separation. Higher speed was not available as this may damage the quill shaft due to the increased vibrations.

In order to avoid rim detachment, the interference fittings between each layer should be sufficiently reserved and must be reliable even for long-term operation.

3.1.2 Overstress
As shown in Fig. 7, interference fittings cause extra tangential tensile stress in the outer rim. If too many rims are assembled or if the fittings are too high, a severe tensile stress occurs in the outermost rim. The tangential stress is more critical than the radial one, as it is more likely to cause a burst failure than radial delamination. A burst failure is the most destructive failure mode and the most challenging issue for the containment. Therefore, stress calculations must be carefully performed, considering various conditions, e.g. elevated temperatures, overspeed, even for losing vacuum event. A validation of the failure modes in spin testing is highly recommended.

For inner rims, high compression occurs after assembly, especially for elevated temperatures. The high compression may cause fiber instabilities and permanent strain of the matrix [22]. Therefore, before the structural design, a coupon test of the material is recommended in order to identify the elastic range under compression.

3.1.3 Assembly difficulties
Large interference fittings or too many rims also lead to assembly problems, as large axial forces will act on the rims due to friction during press fitting. Temperature fitting, which is usually used for metal assembly, has little benefit for composite materials due to the low thermal expansion coefficient. Shrink fitting or press fitting are possible. A hint for press fitting is to design a tapered contact surface of the inner and outer rims [23] or use a tapered guiding tool during assembly [19, 24]. The fibers can be also oriented in axial direction to increase the axial strength. One can also use additional glass fiber reinforcement in the axial direction. This will increase not only the shear strength against the axial force, but also increase the fracture toughness which slows down the crack propagation, increasing the fatigue life [25].

3.1.4 Long-term effects
The pre-loaded compression in the rims decreases slowly with time due to the viscoelastic properties of the material. This behavior is called stress relaxation. Stress relaxation and creep are typical behaviors of polymeric materials under load for a long period of time, which should be carefully concerned, as they...
may lead to the loss of the required pre-loads and cause rim separation within the service lifetime.

The viscoelastic behavior was investigated by researchers in [22, 26–30], both theoretically and experimentally. In [26] the stress relaxation in press fitted composite cylinders is calculated for material IM7-graphite/8552-epoxy, which has a creep compliance of $1.1 \times 10^{-10}$ Pa$^{-1}$ in transversal direction at 75°C. The initial interference pressure was $-34.5$ MPa between the two hoop-wound (90°) cylinders after assembly. However, this interference pressure decreases by approx. 5% after one year and 10% for a long period of time ($10^8$ h) according to the calculation. If the fibers are wound with a shift angle, the effect is even more severe due to the hoop viscoelasticity. The effect of creep occurs when a cylinder is subjected to a rotational load. Creep under tensile stress will cause enlargement of the rotor dimension [27] and shorten the fatigue life.

The viscoelasticity of a material is described by creep compliance or stress relaxation modulus. They can be measured in a coupon test, as introduced in [28]. A coupon specimen, composed by material IM7-graphite/8552-epoxy, is loaded by a constant strain of 0.5% at 135°C. After 72 h, it shows 16.4% relaxation under compression and 7% relaxation under tensile load.

3.2 Pre-tightening winding technique

Another approach to realize the compressive pre-loading is to apply a high tensile force (e.g. 100 ... 200 N [31]) when winding the fibers on the mandrel. An internal compressive stress can be obtained in a single rim due to fiber contraction instead of due to assembly fittings. Therefore, theoretically a multi-rim is no more required for the purpose of pre-loading. Nevertheless, practically, the designers also have the freedom to change the fiber materials during winding for the purpose of optimizing the stress distribution or reducing the costs. This so-called “multi-layer” rim is usually thick and is obtained by winding instead of assembly, therefore, solving the rim-detachment and assembly problems. One crucial issue is the quality control of a thick rim during the curing process. To improve this, a pre-curing was performed in [31] each time after the rim is wound by a certain thickness (e.g. 6 ... 10 mm). The temperature for pre-curing is lower than the full-curing temperature, so that the resin is not solidified but maintains a certain viscosity for further winding of the next layer. In this way, the stress distribution in the rim is improved layer by layer. A relatively good quality can be obtained in the end.

3.3 Multi-directional composites

Instead of unidirectional fiber reinforcement in circumferential direction, fibers can also be oriented in radial direction and axial direction. The manufacturing techniques are different than for filament winding. They are usually built into a disk shape instead of a typical filament-wound cylinder shape.

The first method is so-called stacked-ply design, i.e. unidirectional reinforced laminates are stacked layer-by-layer alternatively (Fig. 11). Examples are shown in [32–35]. In [33] the influence of failure criteria on the optimization of stacked-ply composite flywheels is investigated. [35] presents the manufacturing and NDE (non-destructive evaluation) of multi-directional composite flywheel rims.

The second method is weaving techniques. Examples are [36–38]. Fibers in two orthogonal directions, namely warp yarn and fill yarn are woven and form a spiral annular fabric, which is stacked and forms a disk-shape rotor after composition with resin. A scheme is shown in Fig. 12. Different weave style, such as plain, twill and satin can be adopted. The volume fraction of the warp yarn and fill yarn can be varied along the radius [36]. More warp yarn can be arranged more outside in the radial direction in order to increase the specific elastic modulus, as suggested by [37]. In [38], 5 woven rotors are manufactured and tested. A max. velocity of 479 m/s was achieved by one of these rotors. This velocity was lower than expected due to progressive damage caused by material imperfections and low strength at the tail of the stacked configuration.

Generally speaking, the uniformity of a woven rotor is not as good as for a filament-wound rotor. Especially at the tail of the stacked fabric, peeling may occur due to the low strength. In [39] a strengthen method was proposed, sewing the tail to the layer beneath by high strength nylon fibers. The circumferen-
tial length of this strengthening area should be at least 10 ... 30 times the thickness of one single fabric layer.

The publication [40] deals with another issue in a woven rotor. Due to the weaving pattern, the warp yarns at the outermost radius are not reliably constrained and may slip out from the fill yarns. Typically protruding ends of fill yarns are reserved. However, these protruding ends will be squeezed when placing the stacked fabric into the mold for resin impregnation, thus the concentricity cannot be guaranteed. To prevent the slipping of the warp yarn, [40] proposed to use a heat-melt yarn (fibers of thermalplastic resin, such as polyamides, polyetherimide, etc) outside of the outermost warp yarn, so after heating they can fuse the warp yarn to the weft yarns. In this way, the protruding ends of the weft yarns can be shortened, thus easier to position the stacked fabric concentrically into the mold.

Stacked-ply and woven fabric are both two-dimensional reinforcements. If the third dimension, i.e. the through-thickness direction or \( z \) direction, is also reinforced, one speaks of the so-called 3D composites. The three most common manufacturing techniques are 3D weaving, stitching and \( z \)-pining [41]. Hi-

4 Rim-shaft connection

A difficulty in the composite flywheel rotor is the fixation of the large-dimensional rim onto a small-dimensional shaft. Due to the low elastic modulus, the composite rim deforms significantly under high velocity and temperature rise, which is much larger than the metallic shaft. A hub should provide sufficient compliance for a feasible connection. Sometimes a regular metal hub is difficult to fulfill this requirement. A special design, i.e. either a flexible hub which is radially elastic or other special solutions are required.

4.1 Flexible hub

A flexible hub can be constructed into various structures, using either metallic or composite materials [7,
Fig. 16 Rotor structure of a 10kWh flywheel with the power of 2MW (levitated by a high temperature superconducting magnetic bearing), entirely made of carbon fiber reinforced plastic except the machine and bearing components, velocity of the reinforcing ring 360m/s, velocity of the inertia rim 800m/s [49]. Reprinted from [49], with permission from DOE Energy Storage Program website 43–54]. The critical issue is the rotor dynamic stability, as the stiffness of a flexible hub is usually low. The controller of the magnetic bearings is important to stabilize the rotor at resonance. Sometimes massive power has to be invested to pass the critical speed. Another critical issue of the hub is the fatigue life, especially regarding the long-term degradations for composite materials.

Some flexible hub designs are presented here as examples. 

In [43] an aluminum hub ring was split into 24 segments and welded to the shaft at the top and bottom joints, as shown in Fig. 14. During spinning, these split segments are easier to expand and maintain contact with the rim.

[45] describes an aluminum hub, consisting of 4 spokes and a connecting ring between each spoke. The geometries of the spokes are optimized to eliminate stress concentrations. At high speed, the connecting ring maintains contact with the outer rim, while the spokes lose contact. The maximum velocity at the hub outer radius (161 mm) is as high as 369 m/s.

In [48] two shell-like aluminum hubs are discussed, which are designed for a rated speed of 42,000 min⁻¹. Due to the low stiffness of hub, the rotor performs a deflection mode at a frequency lower than the rated frequency. To stabilize the rotor when passing this frequency, the controller of magnetic bearings has to be adapted with large power input. The friction and losses of magnetic bearings caused a high rotor temperature rise. The test had to be stopped at 28,500 min⁻¹.

Apart from metals, composite materials can be also used to construct hubs for the purpose of lightweight or for connection with large dimensional rims. They are typically constructed into conical or dome shape, such as in [44, 47, 49, 52–54]. A scheme is shown in Fig. 15.

If using only a single conical hub or a dome-shape hub, the drawback is that, the self-expansion produces a bending effect on the hub, yielding an additional axial force on the shaft, which may influence the rotor motion mode and cause additional vibrations. Concerning this effect, a symmetric dual-con hub may be a good solution, as shown in Fig. 16. Similar designs are also presented in [48–50]. However, regardless of dual or single cone, the expansion depends on the hub thickness. A thin cone or dome is more flexible but loses stiffness. Additionally, extra axial space requirement is inevitable to accommodate the cone hub, thus it is not suitable to fix a disk-shape rim with large diameter but short axial height. Concerning the above issues, [46] proposed a self-expanding hub as shown in Fig. 17. The hub consists of two symmetric “dual cones”, which are made of steel and are specially curved (marked by a red circle). The expansion depends on the intrinsic flattening effect at the curved area, instead of pure bending of the shell. Therefore, increasing the shell thickness will not lose the self-expansion ability, so that both high radial flexibility and stiffness can be obtained.

4.2 Hub-less design

If the inner diameter of rim is small, it can be mounted directly on the shaft without a hub or with a so-called “solid hub”. One example is shown in Fig. 18 [21]. Due to the small inner diameter, the rim has to be divided into 7-layers. The critical issues come from the multi-rim assembly, which is discussed in Section 3.1. The advantage of such rotors is their compactness and robustness.

For the non-filament-wound flywheels, which are made from 2D or 3D composite materials, they also have small inner diameter and can be also directly mounted on the shaft or via a solid hub. In [22], a ring hub made of polyoxymethylene was used to combine a 3D composite disk-shape rotor with a shaft. The viscoelastic behavior of the material is the critical is-
Flywheel rotor composed of 7-layer composite rims and titanium shaft, developed by Center for Electromechanics, University of Texas at Austin, USA, 2015, for onboard storage on a hybrid electric transit bus, $E = 1.93 \text{ kWh}$, $v_p = 902 \text{ m/s}$, $n_{\text{max}} = 40,000 \text{ min}^{-1}$, $m = 58.2 \text{ kg}$, $r_o = 431 \text{ mm}$, $h = 152 \text{ mm}$ [21]. Republished with permission of SAE International, from [21], permission conveyed through Copyright Clearance Center, Inc.

In [39] a 2D woven composite disk is used as a hub to connect a cylindrical filament-wound rim.

### 4.3 Shaft-less design

An outer rotor topology can eliminate the shaft, as presented in [55–60]. The rotor components of the electric machine and magnetic bearings are assembled on the inner surface a cylindrical rim. The stator and cooling parts are concentrically located inside the rotor parts, as shown in Fig. 19. This is a highly integrated topology. The geometric optimizations of the rim have more freedoms. One problem comes from the stress on the metallic rotor components, which should fit with the rim inner surface. For example in [59], in order to have a better flux concentration, a rotor back iron is designed on the inner surface of the rim. The magnets of the electric machine are segmented and glued on the inner surface of the back iron. This iron is supposed to undergo a maximum velocity of approx. 270 m/s, which is not endurable for steel. Therefore the rotor iron is circumferentially segmented into pieces to reduce the tangential stress, which increases the radial load on the rim. The second critical issue of the outer rotor topology is the rim radial expansion. In [59], the rim has an inner/outer diameter of 290/430 mm (aspect ratio $\alpha = 0.674$). A maximum speed of 11,000 min$^{-1}$ was reached in the experiment. 20% of the air gap increase was observed, which led to a low inductance of the machine and thus high current ripples and losses.

In a similar topology in [60], in order to fit with the radially growing rim, the magnet is made flexible by composites of NdFeB powder and rubber with fiber reinforcement. It has a cylinder shape and is magnetized in a Halbach array, so no back iron is required. This rotor claims a tip velocity of 300 …3000 m/s by using the carbon nanotube to construct the rim. At a speed of 48,500 min$^{-1}$, the 5 mm air gap increases by about 4.2 mm. However, it is stated in [60], the benefit of this radial growth is the automatic field-weakening effect for the machine. The induced voltage approaches nearly a constant value at high speed due to the reduced air gap. Therefore, with the same inverter rating, higher power can be provided compared to a fixed air gap machine. There are also other critical issues in an outer rotor design, such as the outer rotor levitation solutions, the back-up bearing design, etc. Further details can be found in [61–64].

### 5 Rotor failure and safety containment

Perhaps the most essential and critical issue regarding a flywheel is the safety and containment, especially for mobile applications. Underestimation of this safety issue may lead to severe and fatal consequences due to rotor failures. Several examples are reported in [65, 66].

The failure characteristics of composite flywheels are different from metallic rotors [67]. They will not penetrate the containment like the metal fragments. It was believed earlier that they were more likely becoming fuzzy or fluffy in the beginning of the failure instead of destructive crash. Later people found out this may be true for Kevlar fiber or glass fiber composites, which spin at velocities typically lower than 800 …900 m/s [67]. For carbon fiber composite rotors, which contain higher specific energy, the failure is more catastrophic. The mixture of debris and dust behaves like a fluid, flowing along the containment surface, producing axial loads on the top and bottom lids, which has been observed in many tests [68].

If a containment is supposed to contain the full burst energy, the containment mass will be many times higher than the rotor itself, which is impractical, especially for mobile applications. Therefore, according to [65], a typical safety criterion is: 1) The flywheel rotor has to be qualified to assure that it will
be free from structural failure under all conditions over the life of the system. That means that all the potential failure modes of the rotor must be well understood and all the hazards must be avoided in the design stage. 2) The implementation of containment is for the safety purpose in case of unexpected conditions or faults, e.g. assuring containment of the loose but largely intact rotor due to bearing, hub or shaft failure. Thus the containment mass can be reduced.

In the earlier development stage of flywheel technologies, there were no standards established. In order to understand the failure modes, an intensive investigations were carried out in the late 1990s [69–71], both theoretically and experimentally. The results and testing experiences are documented as guidelines for the containment design and safety recommendations [65–67, 72–74]. Some general standards for relevant issues in turbines and systems containing high energy are used for these recommendations. A summary of these standards can be found in [74]. Nowadays, standards regarding flywheels are also established, such as the international standard “ISO 21648:2008 Space system—Flywheel module design and testing” and the American standard “AIAA S-096-2004 Space system—Flywheel rotor assemblies”.

Some safe design principles and testing methods of composite rotors are summarized here.

5.1 Design margin

According to the design recommendation in [65], the rotor design margin has to be verified in a qualification test. In the qualification test, the rotor is accelerated to a maximum speed, either remaining intact or fail. A safe margin is defined that the operating speed of the rotor should be no higher than 70% of the maximum speed achieved in the qualification test, no matter if the rotor failed or not. Also, it is recommended in [65], the qualification tests should be performed 3 times with substantially identical rotors. If a mechanical failure occurs, the failure speed should be validated by repeating 3 times the tests and the failure speed deviations should be within ±3%.

5.2 Testing and NDEs

For a valid design, a series of tests and NDEs are required to understand the material properties, control the manufacturing quality, investigate the failure modes and estimate the lifetime.

Typical tests are introduced in [75]. Before the design, static tests are usually performed on coupons to well understand the material properties. Hydroburst tests can provide the strength, stiffness and fatigue properties for a ring specimen. To control the manufacturing quality, NDE methods, such as computed tomography (CT) scan, ultrasonic scans, scanning electron microscope (SEM), can be used to detect the initial flaws in the pieces. After manufacturing, the rotor is usually tested in a spin test to identify the maximum speed, dynamic behavior, as well as the potential failure modes. The spin test is usually dangerous and expensive. Recently, [76] proposed a static burst test method for a composite rotor with nearly the same stress distribution in the dynamic case. This method was demonstrated to be a controllable and observable possibility to test a rotor in a static way. In [77], a low-cost spin rig was proposed and successfully applied for a steel rotor testing up to 220 m/s at 30,000 min⁻¹. The system setup costs are below 6000€.

Fatigue tests and safe-life tests are essential to estimate the lifetime of rotors. Considering the costs and potential hazards, fatigue tests are typically performed on specimens instead of actual rotors [75, 78, 79]. Load conditions may include transverse cyclic loading, combined axial compression and torsion loads [75], cyclic four-point bending load [80]. According to [74], S-N curves in combination with Palmgren & Miner accumulation methods are used to determine the life cycles. It is recommended to use only 1/3 of the calculated lifetime as a design life. Reasonable inspection intervals should be arranged, especially if no safe containment exists.

One issue in the fatigue tests is that, the state and loading of the specimens should be as close as the actual rotor. This is sometimes difficult because the specimens are typically flat and thin pieces, while the actual rotor is a thicker rim. The manufacturing qualities are thus not comparable. Taking this into account, [80] used a downsized composite cylinder as a specimen and built a cost-effective spin test rig for the fatigue life evaluation. The cylindrical rim is connected to a spindle via a cone-shaped hub. The spindle is magnetically levitated. The test rig is placed in a vacuum chamber and protected by 2 steel cylindrical walls. Additional circumferentially segmented steel ring is placed inside the rim to provide extra load, so that the stress state of the rim is close to the actual rotor. The testing speed was cycling between 15,000 min⁻¹ and 30,000 min⁻¹ and continuously operated up to 200,000 cycles, taking about 83 days.

5.3 Containment validity

In order to investigate the failure scenarios, a number of tests are performed in [69], where the rotors are spun to failure. The measured results are used for a reasonable containment design. Some design concepts are given in [69]. The first direct approach is to use heavy rigid walls. The design method is simple, however, the torque loads will be transmitted to the mounting housing as the debris swirling on the containment inner surface. To absorb this rotational energy, a free-rotating liner can be placed inside the containment, thus the torque transmission can be slowed down, reducing the torque loads on the containment and thus the weight. The third approach is to use energy absorbing liner, which is typically built with “soft”
and ductile materials or structures [81, 82]. The energy of the debris can be quickly absorbed by a shape change of the liners, thus reducing the peak forces. In this way, the size and weight of the containments can be reduced.

5.4 Other safety issues

There are more safety issues regarding the flywheel operations, e.g. the validation of back-up bearings in case of rotor dropdown, protection from overspeed and protections from losing vacuum or control failures, which are not discussed here. For details one can refer to [65].

6 Summary and discussion

This paper presents the characteristics of different composite rotors and the critical considerations in their design, manufacturing and testing.

The discussion starts with a design example of a single filament-wound composite rim. The velocity $v_p$, specific energy $E/m$, energy density $E/V_o$ and shape factor $K$ are used to evaluate the design. It is found out that these parameters depend strongly on the aspect ratio $\alpha$ of the rim. A thin rim (high $\alpha$) usually has excellent performances. A thick rim (low $\alpha$) can dramatically decrease the performances, due to the low radial strength. If improperly designed, the specific energy of a thick rim may be only $2$ times higher than the steel rotors due to the low shape factor. The benefits of using composite materials are wasted. For a single rim without any other boundary conditions, there exists a specific $\alpha$, at which the rim can achieve high $E/m$ and $K$, and the maximum $E/V_o$. But if considering other boundary conditions, such as connection to shaft, rotor dynamics, etc., a thick rim is always preferred in practice.

To improve the performances for low $\alpha$ designs, various solutions are introduced, e.g. using shrink fitted multi-rim design, using pre-tightening winding techniques, or using multi-directional fiber reinforcement. The related critical issues are presented. For multi-rim design, the most dangerous failure is the burst of the outermost rim due to overstress. A qualification test is mandatory for safety concerns. The danger of rim detachment at overspeed or due to long-term stress relaxation should also be carefully considered. For multi-directional reinforced structures, the critical issue is the uniformity and quality control in order to reach higher speed.

The second issue in a composite rotor is the rim-shaft connection. One can either directly mount it or use a hub. A flexible hub is usually preferred for the compliance of the large deformation of the rim. However, a flexible hub usually leads to a low natural frequency, which may worsen the rotor dynamic behavior. The compromise of flexure and stiffness is the key issue. For outer rotor topology, where the shaft and hub are eliminated, the critical issue is the air gap growth due to the rim deformation. The performances of energy converters and magnetic bearings may be influenced.

The safety issues regarding rotor failures are the most essential issues in flywheel rotors. Design principles and testing methods are summarized in this paper according to the given design guidelines and recommendations based on testing experiences and standards.

The flywheel technologies have been intensively studied since the 1990s and nowadays still keep attracting attention from research groups and industries. The trend is higher specific energy, energy density, efficiency and lower costs [83]. Some promising technologies are under development, such as super strong carbon nanotubes (claiming a tensile strength of $11 \ldots 100$ GPa and an expected rotor specific energy from $264$ Wh/kg to thousands Wh/kg [60, 84]), superconducting magnetic levitation, magnetic composite materials and high frequency low-loss power electronics (SiC, GaN semiconductors). These technologies provide a positive future for advanced flywheels, especially with the rapidly increasing integration of renewable energy in the coming decades. Currently, flywheels are competing with supercaps and high power batteries. However, in some applications, they are still irreplaceable concerning their high power density, fast response and long life time without degradations. The future storage market share depends on the developments and costs of all these storage technologies.

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7 Appendix

7.1 Appendix A Formulas for stress calculation

Stresses in a rotating composite rim [7] based on plane stress assumption,

\[ \sigma_0 = \rho \cdot v_p^2 \cdot H_0^2 (\alpha, r/r_0) \] (A.1)

\[ \sigma_1 = \rho \cdot v_p^2 \cdot H_1^2 (\alpha, r/r_0) \] (A.2)
with,

\[
H_0^* (\alpha, r/r_o) = \frac{3 + v_{\perp \perp}}{9 - E_\perp^2} \left( E_{\parallel} \cdot L \left( \frac{r}{r_o} \right)^{E_{\parallel} - 1} + E_{\perp} \cdot (L - 1) \cdot \left( \frac{r}{r_o} \right)^{-E_{\parallel} - 1} \right)
\]

\[
- \frac{E_\parallel^2 + 3 \cdot v_{\parallel \perp}}{3 + v_{\parallel \perp}} \left( \frac{r}{r_o} \right)^2 \right)
\]

\[
H_1^* (\alpha, r/r_o) = \frac{3 + v_{\perp \perp}}{9 - E_\perp^2} \left[ L \cdot \left( \frac{r}{r_o} \right)^{E_{\parallel} - 1} - (L - 1) \cdot \left( \frac{r}{r_o} \right)^{-E_{\parallel} - 1} - \left( \frac{r}{r_o} \right)^2 \right]
\]

where

- \( r \): radial position on the rim, \( r_i \leq r \leq r_o \).
- \( \rho \): material mass density in kg/m\(^3\).
- \( v_{\parallel \perp} \): Poisson's ratio.
- \( E_{\parallel} = \sqrt{E_\parallel/E_\perp}, E_\parallel, E_\perp \): Young's modulus in the fiber direction and transverse direction.
- \( L = \frac{\alpha - E_{\parallel} - 1}{\alpha - E_{\parallel} - 1} \); \( \alpha \): the aspect ratio \( \alpha = r_i/r_o \).

### 7.2 Appendix B Properties of the used materials

| Table B.1 | Properties of the chosen composite materials with a fiber volume fraction \( \phi = 0.6 \) |
|-----------|---------------------------------|
|           | T700 + Epoxy resin | M46 + Epoxy resin |
| **Mass density** | \( \rho \) [kg/m\(^3\)] | 1550 | 1600 |
| **Young’s modulus** | \( E_\parallel \) [MPa] | 125,000 | 245,000 |
| | \( E_\perp \) [MPa] | 7800 | 6900 |
| **Shear modulus** | \( G_{\parallel \parallel} \) [MPa] | 4400 | 3900 |
| | \( G_{\parallel \perp} \) [MPa] | 5330 | – |
| **Poisson’s ratio** | \( v_{\parallel \parallel} \) | 0.34 | 0.34 |
| | \( v_{\parallel \perp} \) | 0.021 | 0.021 |
| | \( v_{\perp \perp} \) | 0.35 | 0.35 |
| **Tensile strength** | \( R_{\parallel} \) [MPa] | 2450 | 2160 |
| | \( R_{\perp} \) [MPa] | 70 | 45 |
| **Compressive strength** | \( R_{\parallel} \) [MPa] | 1570 | 980 |
| | \( R_{\perp} \) [MPa] | 170 | 135 |
| **Shear strength** | \( R_{\parallel \parallel} \) [MPa] | 98 | 59 |
| **Thermal expansion coefficient** | \( a_{\parallel} \) [10\(^{-6}\)/K] | 0.4 | –0.7 |
| | \( a_{\perp} \) [10\(^{-6}\)/K] | 36.1 | 35.5 |

\( \parallel \): fiber direction, \( \perp \): transverse direction
### 7.3 Appendix C List of different flywheel rotor topologies

**Table C.1** List of different flywheel rotor topologies with the performance parameters

| Developer                  | Energy [kWh] | Power [kW] | Speed [min⁻¹] | Max. velocity [m/s] | Rotor topology                          | Ref       |
|----------------------------|--------------|------------|---------------|---------------------|-----------------------------------------|-----------|
| Hanyang University         | 51           | –          | 15,000        | 707                 | Multi-rim + dome-shape composite hub    | [54]      |
| Beacon Power               | 25           | 100        | 16,000        | –                   | Multi-rim                              | [85]      |
| Boeing                     | 5            | 3          | 22,500        | 800                 | 3-layer rim + aluminum hub              | [86]      |
| Stornetic                  | 3.6          | 3          | 45,000        | –                   | Carbon fiber rotor                      | [87]      |
| TU Darmstadt               | 2.4          | 100        | 17,500        | 394                 | Single rim, shaft-less                  | [59]      |
| Magnet-Motor               | 2            | 150        | 12,000        | 390                 | Carbon fiber rim                        | [87]      |
| The University of Texas at Austin | 1.93     | 250        | 40,000        | 902                 | 7-layer rim + titanium shaft, hub-less  | [21]      |
| Kinetic Traction Systems, Inc | 1.5       | 333        | 36,000        | –                   | Composite rim, shaft-less               | [88]      |
| Powertrhu                  | 0.528        | 190        | 53,000        | 1000                | Carbon fiber rim + titanium hub         | [67]      |
| GKN Hybrid Power           | 0.5          | 120        | 36,000        | 660                 | 2-layer rim                             | [67]      |
| Hanyang University         | 0.5          | –          | 40,000        | 712                 | 2-layer rim + split aluminum hub        | [49]      |
| Tsinghua University        | 0.34         | –          | 42,000        | 660                 | 2-layer rim + shell-like aluminum hub   | [48]      |
| PUNCH Flybrid              | 0.111        | 60         | 64,500        | 675                 | Carbon fiber rim + steel hub            | [2, 67]   |
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