Mechanical Design and Analysis of a Permanent Magnet Rotors used in High-Speed Synchronous Motor

Bogdan Dumitru VĂRĂTICEANU, Paul MINCIUNESCU, Daniel FODOREAN *

Abstract

In high-speed machines, the rotor mechanical design often become more critical that the electromagnetic or thermal design. The main focus is to find out and reduce the mechanical stress distribution in the rotor's component parts. For preliminary mechanical design, the analytical approach is used. Further design and optimisation implies 2D finite element analysis. In this paper two types, of permanent magnet rotors (with surface-mounted magnets and with buried magnets) are modelled and analysed.

Keywords: high-speed motors, mechanical design, tensile stress, permanent magnet rotors

1. Introduction

Using compact high-speed motors in different applications, such as hybrid propulsion system for vehicles is an attractive choice due to reduction of number of drive components, increased system reliability and reduce the entire system cost [3].

The opportunity of using those motors in such applications leads to high power density, reducing the transmission losses, the system maintenance, the system noises and increasing the efficiency.

In high-speed motors, beside the electromagnetic and thermal design, the mechanical aspects need to be considered. Different types of machines are qualified for high-speed application, such induction machines with squirrel-cage rotor, permanent magnet synchronous machines and switched reluctance machines [16], [17].

Due to permanent magnets as source of rotor magnetic field, some advantages as: small rotor losses, minor thermal rotor expansion and increased efficiency, leads to widespread usage of the permanent-magnet synchronous machines PMSM in high-speed applications.

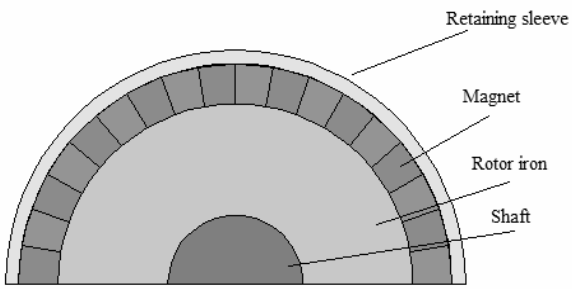
As operating speed increase, the mechanical design of the rotor became critical in the machine design process and primary factor in determining the rotor dimensions and configuration.

Electromagnetic design is often the secondary factor and concerns, with generating, as much torque as possible from the given rotor volume. Electromagnetic design is also deal with minimizing the losses and avoiding the permanent magnets demagnetisation.

This paper is concerned with rotor mechanical design of the permanent-magnet synchronous machine.

In case of PMSM, there are two different rotor configurations (Figure 1):

- surface-mounted magnets;
- buried magnets.



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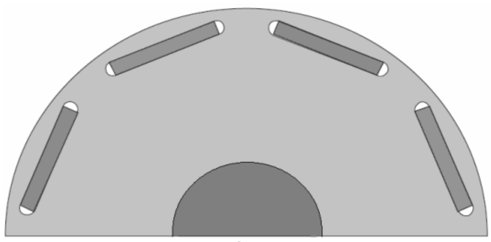

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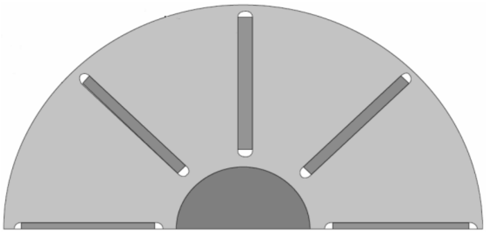
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Figure 1. Rotor configuration for high-speed PMSM: a) Surface-mounted magnets; b) Buried magnets; c) Buried magnets with flux concentration.

The rotor with surface-mounted magnets uses a retaining ring or sleeve to withhold the permanent magnets to the lamination stack or to the shaft. The rotors with buried magnets means that the magnets are placed in a slot below the surface of the rotor or the magnets are placed in a carrier, which is held by high strength retaining components [8]-[10]. While buried permanent magnet rotor have attractive magnetic properties, such as increased resistance to demagnetisation, they have been shown to be mechanically weaker compared to rotors with surface mounted magnets, this results in limitations of volume for the same mechanical speed [2], [12], [14], [15]. The sleeved rotors present a very simple design that maximises the magnetic field strength and rotor volume by distributing the forces evenly around the retaining ring and avoiding stress concentration [11]. The operating principle of the retainer sleeve is to generate a compressive force to be applied to the permanent magnets at all operational speed, allowing the torque transmission among the magnets and the other rotor parts (mainly to the shaft), and to prevent significant tensile stresses occurring in the fragile permanent magnets.

The retaining sleeves used in surface-mounted magnets should be able to sustain large operating stresses and these can be manufactured from different type of materials, such as high strength steel, fibre composite, titanium, etc [2], [18], [19], [20]. To generate the compressive force among the sleeve and magnets an interference fit is used. Depending on the materials used for the retaining sleeves the interference fit can be either done: by thermal shrinkage in case of metal sleeves or by wrapped under tension in case of composite materials.

In the study presented in this paper, only the centrifugal forces are considered, as this is likely to be the dominant source of mechanical stress in high-speed design [13].

2. Mechanical design of permanent-magnet rotors

The mechanical design and optimisation of the PM rotors has been made by two different approaches:

- an analytical method, which is an extremely fast way for design, but some simplifications and assumptions are made that can influence the precision of the method;
- a numerical method, usually the finite element method, that can take in consideration complex rotor structures having a high precision, but is not time efficient during all design procedures, since demands excessive computation time and resources.

At high rotational speed, high centrifugal forces are exerted on the permanent magnets. Thus, when rotors with surface-mounted magnets are used, retaining sleeves must be employed to reduce the stress in the magnets and to firmly retain the magnets in their position.

Because the mechanical stress in the rotor parts are mainly in the tangential direction, only the tangential tensile stress σ_t will be computed using the analytical method.

The mechanical design parameters of the PM rotor were calculated in the first stage using an analytical method. After that, in the second stage, a FE analysis is used for final optimisation and for accurate mechanical stress distribution in the rotor.

2.1. Analytical mechanical design for rotors with surface-mounted magnet

For this type of configuration, the geometry is simplified by assuming that the permanent magnet is continually annular and the contact force due to the glue among the magnets and rotor back iron is neglected. The rotor design can be done using the following formulas [3] and [4]:

$$p_c = p_{c,fit} - p_{\omega,m} - p_{\omega,s} > 0 \quad (1)$$

$$\sigma_t = \sigma_{t,fit} + \sigma_{t,\omega} < \sigma_{t,\max} \quad (2)$$

In (1) the contact pressure p_c (at rotational speed n) among the sleeve and the permanent magnets should be positive at any operational speed. This contact pressure is due to the interference fit contact pressure $p_{c,fit}$ reduced by the centrifugal forces on magnets $p_{\omega,m}$ and sleeves $p_{\omega,s}$. The tangential tensile stress (2) in the rotor magnets and sleeves is due to the tangential stress caused by interference fit

$\sigma_{t,fit}$ and by rotational speed $\sigma_{t,\omega}$ and should be kept below the maximum permissible tangential stress $\sigma_{t,max}$. Figure 2 shows the rotor structure for rotors with surface-mounted magnets.

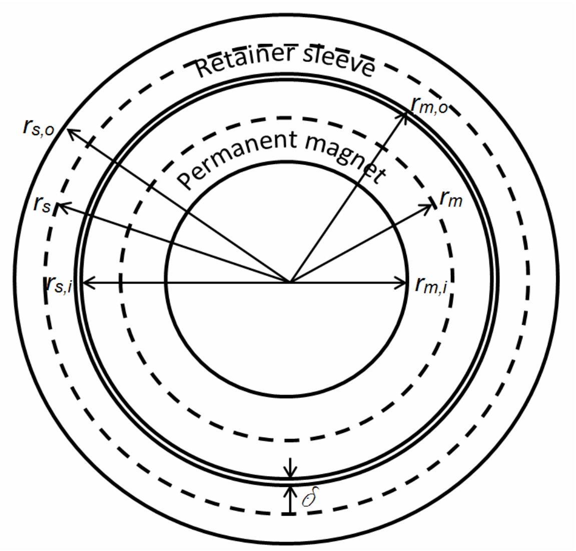

Figure 2. Geometry assumption of permanent magnet and retainer sleeve in a rotor with surface-mounted magnets

The centrifugal forces on magnets and sleeves reduce the contact pressure among the shaft and permanent magnets, and are:

$$p_{\omega,m} = r_m \cdot \rho_m \cdot \omega^2 \cdot h_m \tag{3}$$

$$p_{\omega,s} = r_s \cdot \rho_s \cdot \omega^2 \cdot h_s \tag{4}$$

where r_m, r_s, ρ_m, ρ_s, h_m and h_s are the average radius, the mass density and the height of the magnets and sleeves.

The tangential stress due to interference fit and contact pressure due to shrinking or wrapping of the sleeve onto the rotor are:

$$\sigma_{t,fit} = \frac{\delta}{d_o} \cdot E \tag{5}$$

$$p_{c,fit} = \frac{\sigma_{t,fit} \cdot h_b}{r_s} \tag{6}$$

$$p_{c,fit}(r) = \sigma_{t,fit} \cdot \left(\frac{r_{s,i}^2}{r_{s,o}^2 - r_{s,i}^2} \cdot \left(1 + \frac{r_{s,o}^2}{r^2}\right)\right)^{-1} \tag{7}$$

where δ is the interference fit required to develop the contact pressure from (6), E is the Young's modulus of the sleeve, d_o is the rotor outer diameter and $r_{s,I}$, $r_{s,o}$ are the inner and outer sleeve radius. The contact pressure due to interference fit for a sleeve considered as a "thin shell" is presented in (6) and for a sleeve considered as a "thick shell" is presented in (7). A thin shell is defined as a shell with a thickness, which is small compared to the rotor diameter.

The additional tangential stress due to rotation of the magnets or sleeve is:

$$\sigma_{t,\omega} = \rho_s \cdot \omega^2 \cdot r_s^2 \tag{8}$$

$$\sigma_{t,\omega}(r_i) = 0.4125 \cdot \rho \cdot \omega^2 \cdot \left(0.424 \cdot r_i^2 + 2 \cdot r_o^2\right) \tag{9}$$

where ρ, r_i and r_o are the mass density, inner and outer radius of the magnets or sleeve. The equation (8) is valid for a thin shell sleeve and equation (9) is valid for a thick shell sleeve. In the retainer sleeve, the maximum tangential stress is found at the inner surface of the sleeve. In the permanent magnets the total tangential stress can be computed using equation (2) considering $\sigma_{t,fit}$ being negative.

Depending on the materials used for the retainer sleeve, the thermal expansion can be neglected in case of fibre composite materials or taken into consideration for metal retainer sleeve. Thermal expansion of other metal rotor parts such as shaft, lamination stack or magnets will develop a significant strain on the retainer sleeve. The temperature difference to allow the metal retainer ring to slip over the magnets is:

$$\Delta T = \frac{\delta}{r_{m,o} \cdot \alpha_T} \tag{10}$$

where $r_{m,o}$ is the magnet outer radius and a_T is the coefficient of thermal expansion of the retainer sleeve.

An adequate safe margin between 5 %-10 % of the rotational speed should be chosen in the mechanical stress calculation.

2.2. *Analytical mechanical design for rotors with buried magnets*

The forces that act in a high-speed rotor are dominated by the centrifugal force. Analytical calculation of the peak mechanical stress acting in a tangential direction is a challenging task and a FE analysis is recommended. In this type of rotor, the magnets are inserted into the rotor slots and no interference fit is needed. The lamination stack bridge must withstand the centrifugal force produced by both bridge and magnets.

For a simple configuration, as the rotor structure presented in Fig. 1 (b), the geometry is simplified by assuming that the permanent magnet and the back iron bridge are transformed in an equivalent ring with an artificially increased mass ρ_{equiv} Figure 3 [3].

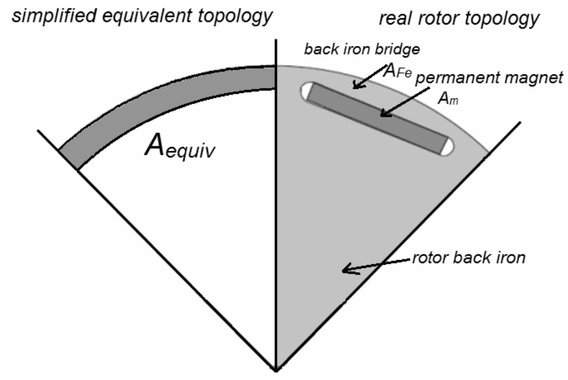

Figure 3. Simplified assumption of a rotor with buried magnets

The height of the equivalent ring h_{equiv} is chosen to be equal to the narrowest high of the back iron bridge that covers the permanent magnets.

The artificial ring equivalent mass density is:

$$\rho_{equiv} = \rho_{Fe} \cdot \frac{A_m + A_{Fe}}{A_{equiv}} \qquad (11)$$

where ρ_{Fe} is the mass density of the laminated back iron, A_m, A_{Fe} and A_{equiv} represent the surface of the permanent magnets, back iron and equivalent ring. The equivalent mass substitute both the permanent magnets and the back iron masses, and the equivalent ring suffers from the same centrifugal forces as the original rotor structure.

The tangential stress inside the equivalent ring, due to rotational speed can be computed as:

$$\sigma_{t,equiv} = \left(\frac{r_{equiv,o} + r_{equiv,i}}{2}\right)^2 \cdot \omega^2 \cdot \rho_{equiv} \qquad (12)$$

The maximum tangential stress $\sigma_{t,max}$ inside the rotor is the local peak stress caused by the uneven distribution of the permanent magnets and by the magnets edges and is computed as:

$$\sigma_{t,max} = 2 \cdot \sigma_{t,equiv} < R_{p0.2} \qquad (13)$$

where $R_{p0.2}$ is the laminated back iron yield strength. The yield strength is defined as the amount of stress that will lead to a permanent (irreversible) deformation of 0.2 %. This defines the maximum accepted limit of the stress that can be applied to the material.

3. Finite element analysis

For accurate distribution and prediction of the tensile stress and for optimization of the mechanical design of high-speed rotors, finite element method is used [1].

The numerical analysis was carried out using the JMAG Designer v12.1 [7].

As a starting point, for verifying the correct formulation of the mechanical problem, the mechanical stress distribution was computed in a rotor configuration from article [5]. Good arrangement was obtained when comparing the results with the ones given in this article.

Both rotors configuration (with surface-mounted magnets and buried magnets) was mechanical analyzed and optimized using finite element method.

3.1. Numerical analysis for rotors with surface-mounted magnet

In high-speed machines, the rotors normally have two or four magnetic poles in order to minimise the electric frequency and improve the machines efficiency. In this study, the rotor with surface-mounted permanent magnets from a high-speed machine with two magnetic poles and a rated power of 40 kW at 40000 rpm was numerically analysed. A retainer ring of titanium alloy was chosen to firmly hold and protect the fragile permanent magnets due to its high maximum permissive mechanical tension ($\sigma_{t,max}$>850 N/mm^2).

To simplify the numerical problem formulation, all the materials are considered anisotropic.

For rotors with surface-mounted magnets, two rotor types were analysed. The key parameters of the first rotor type are given in Table 1.

Table 1. Rotor dimensions

R_s	shaft radius	11.9 mm
$R_{m,i}$	magnet inner radius	21.9 mm
$R_{m,o}$	magnet outer radius	27.9 mm
$R_{g,i}$	inter-pole gap radius	23.9 mm
$R_{b,o}$	bandage outer radius	29.9 mm
δ	interference fit	0.1 mm
α_m	magnet open angle	153^0
l	rotor active length	114 mm

The mechanical properties of the materials used in construction of the rotor with surface-mounted magnets are presented in Table 2.

Table 2. Mechanical proprieties of the materials

Material	Young's modulus [GPa]	Density [kg/m^3]	Poisson's ratio
titanium alloy	110	4500	0.33
smco	150	8300	0.27
steel	193	8000	0.3
epoxy resin	5	1300	0.4
lamination steel	215	7300	0.3

The rotor was tested at 48000 rpm. An interference fit of 0.1 mm was chosen, so that the contact pressure among the retainer ring and permanent magnets is positive at any operational speed. The inter-pole gap between the two magnetic poles was filled with a non-conducting epoxy resin with a low mass density.

The mechanical tensile stress distribution in form of von Mises stress is shown in Figure 4.

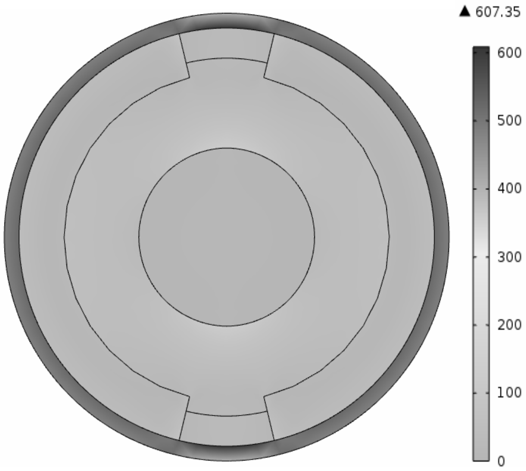

Figure 4. Von Mises stress distribution (MPa) in the first rotor

The maximum mechanical stress has developed as expected in the retainer ring. An unexpected issue is that the tensile stress has concentrated on the retainer ring in the region that covers the inter-pole gap Figure 5.

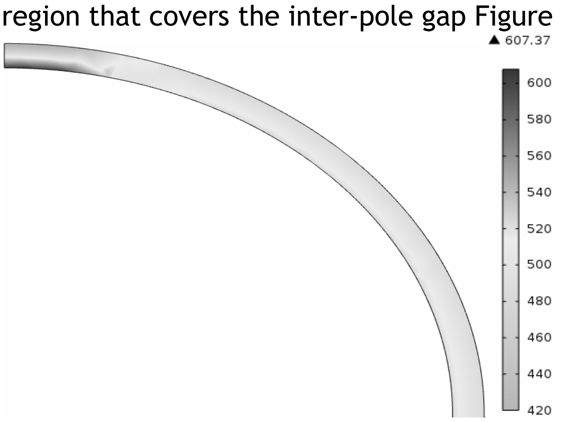

Figure 5. Mechanical stress concentration (MPa) on the retainer ring

The analytical result for tangential stress on the retainer ring, treating it as a thin shell sleeve, is 335 MPa.

The maximum tensile stress (607 MPa) developed in the sleeve is below the maximum allowable limit. However, the stress concentration over a short surface leads to the risk that the retainer ring can break along this surface [3]. The thermal shrinkage of the retainer ring in order to provide the interference fit can be harmful to the titanium alloy along the inter-pole gap. This causes a shear stress in the inner layer of titanium alloy in the region that covers the inter-pole gap and leads to a weakening of the entire retainer ring. In addition, due to the epoxy resin's low mass density (compared with the permanent magnets mass density), at high rotational speed the retainer ring will suffer a deformation on the region with mechanical stress concentration. This deformation will lead to an additional edge effect of the permanent magnets that cut into the titanium alloy ring.

The key parameters of the second rotor type are given in Table 3.

Table 3. Rotor dimensions

R_s	shaft radius	11.9 mm
$R_{m,i}$	magnet inner radius	21.9 mm
$R_{m,co}$	magnet outer radius in magnet centre	27.9 mm
$R_{m,eo}$	magnet outer radius in magnet extremities	25.9 mm
$R_{g,i}$	inter-pole gap radius	23.9 mm
$R_{b,o}$	bandage outer radius	29.9 mm
δ	interference fit	0.1 mm
α_m	magnet open angle	154.8^0
l	rotor active length	114 mm

The permanent magnets have a concentric shape with a bigger height in the magnet centre and lower height outwards. This shape was chosen in order to obtain a sinusoidal distribution of the magnetic field in the air-gap.

The materials used in this rotor construction are described in Table 2 (see above).

As in the previous rotor type, the inter-pole gap was filled with the same epoxy resin. The test conditions are the same as those used for the first rotor.

The mechanical tensile stress distribution in the second rotor type in form of von Mises stress is shown in Figure 6.

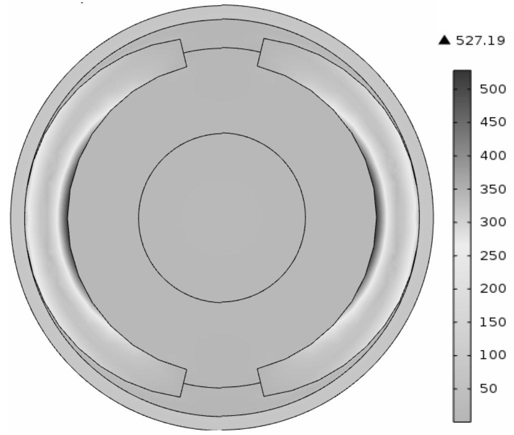

Figure 6. Von Mises stress distribution (MPa) in the second rotor

The maximum mechanical stress has developed in the fragile permanent magnets. The maximum tensile stress (527 MPa) is far beyond the SmCo permanent magnets allowable limit.

The analytical result for tangential stress on retainer ring, treating it as a thick shell sleeve, is 88 MPa.

The permanent magnets shape leads us to conclude that the contact pressure between the retainer ring and permanent magnets (due to the interference fit) is made on a relatively reduced surface, only where the magnets have the maximum height. Due to this reduced contact surface, the tensile stress distribution in permanent magnets exceeds the upper allowable limit, leading to magnets cracking.

3.2. Numerical analysis for rotors with buried magnets

Rotors with buried magnets proved being mechanically weaker compared to rotors with surface mounted magnets [2]. Due to this mechanical reduction in maximum allowable tensile stress, it results a limitation of rotational speed or a limitation of volume for the same rotational speed. The initial design of the rotor and key parameters are presented in Figure 7 in connection with Table 4.

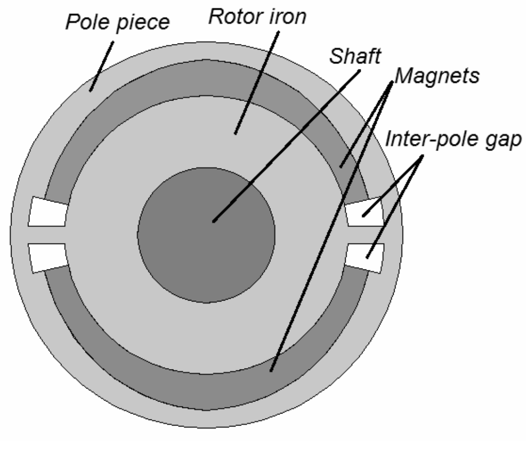

Figure 7. Initial structure of rotor with buried magnets

Table 4. Rotor dimensions

R_s	shaft radius	11.3 mm
$R_{m,i}$	magnet inner radius	23.3 mm
$R_{m,co}$	magnet outer radius in magnet centre	29.3 mm
$R_{m,eo}$	magnet outer radius in magnet extremities	27.3 mm
$R_{g,i}$	inter-pole gap radius	29.3 mm
$R_{b,o}$	rotor outer radius	32.3 mm
δ	interference fit	0.1 mm
α_m	magnet open angle	154.8^0
l	rotor active length	350 mm

The advantages of this type of rotors come from their simple manufacturing process and lower production costs. Assembly difficulties with associated impact on performances (retainer sleeve made of carbon fibre) and relatively higher cost, make the rotors with surface-mounted magnets to be used only in application that involve very high rotational speed.

Assuming that the main factors, that affect the peak mechanical stress, are the outer diameter of the rotor and the rotational speed, the von Mises stress distribution is directly proportional to each of this factor.

Hereinafter, we present the numerically analysis of the rotor with buried magnets from a high-speed machine with two magnetic poles and a rated power of 26 kW at 26000 rpm.

As in the second rotor with surface-mounted magnets, in this type of rotor, the permanent magnets have a special shape with bigger height in the magnet centre and lower height outwards.

The laminated steel pole piece from Figure 7 is only attached to the rest of the laminations by the thin steel bridges at each end. The centrifugal load on the pole piece is not evenly distributed across the entire surface, causing a substantial concentration of mechanical stress on the two retaining bridges.

The challenge in the mechanical design of the rotors with buried magnets is to model the shape and dimensions of those bridges to sustain the large operational stress [6].

The mechanical properties of the materials used in the construction of the rotor with buried magnets are presented in Table 5.

Table 5. Mechanical proprieties of the materials

Material	Young's modulus [GPa]	Density [kg/m^3]	Poisson's ratio
Vacodur 50	250	8120	0.3
SmCo	150	8300	0.27
Steel	193	8000	0.3

Another important detail in the mechanical design of the retaining bridges is the limitation of the edges notch effect. Designing round-shaped edges can influence the maximum stress concentration in the bridges. Various types of retaining bridge were numerically analysed in order to reduce the mechanical stress concentration.

Figure 8 shows the mechanical stress distribution in MPa for different shapes of pole piece bridges, where:

a) zero slot edge radius;

b) 0.25 mm slot edge radius;
c) 3 mm slot edge radius;
d) 3 mm slot edge radius and bridge height increased by 0.5mm;
e) different shape with 1 mm slot edge radius;
f) different shape wit 0.5 mm slot edge radius;
g),(h) different bridges shape;
i) 1 mm slot edge radius and retainer sleeve with 0.05 mm interference fit;
j) 1 mm slot edge radius and retainer sleeve with 0.1 mm interference fit.

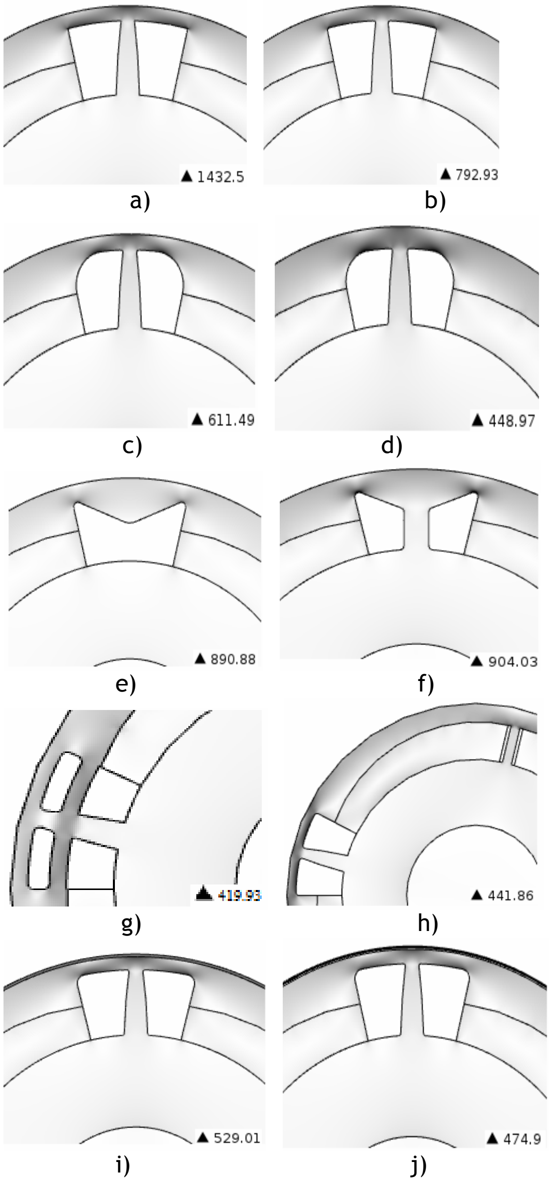

Figure 8. Various shape and dimensions of the pole piece bridges

The final version of the rotor and key parameters, that result from finite element analysis are presented in Figure 9 in connection with Table 6.

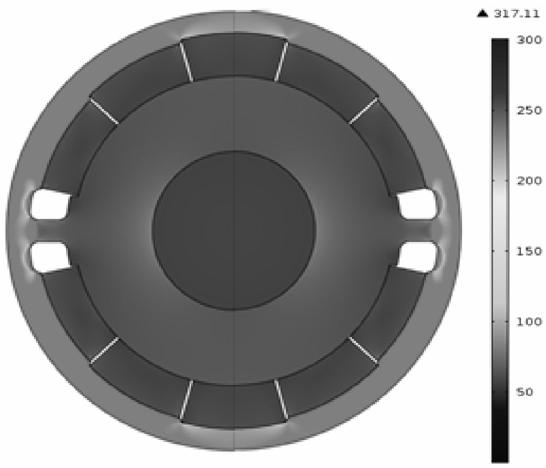

Figure 9. Von Mises stress distribution (MPa) in final version of rotor with buried magnets

Table 6. Rotor dimensions

R_s	shaft radius	12 mm
$R_{m,i}$	magnet inner radius	21.5 mm
$R_{m1,o}$	highest magnet outer radius	26.5 mm
$R_{m2,o}$	middle magnet outer radius	27 mm
$R_{m3,o}$	smallest magnet outer radius	27.5 mm
$R_{g,i}$	inter-pole inner gap radius	22.5 mm
$R_{g,o}$	inter-pole outer gap radius	27.5 mm
$R_{b,o}$	rotor outer radius	31.5 mm
α_m	magnet open angle	154.8^0
l	rotor active length	350 mm

The circumferential speed of the rotor is 83 m/s.

The permanent magnets were divided in five distinct pieces, in order to obtain a constant outer radius for each piece, and to maintain the sinusoidal distribution of the magnetic field in the air-gap. The maximum tensile stress developed in the retaining bridges (317 MPa) is below the maximum allowable limit of Vacodur 50 (390 MPa). The analytical result for maximum tangential stress in retainer bridges is 293 MPa.

4. Conclusions

In this paper, the mechanical design and analysis of high-speed permanent magnet rotors are presented. A simple analytical process has been described to estimate the stress distribution in rotor parts. Accurate stress prediction implies finite element analyses.

In the rotor with surface-mounted magnets, presented in Figure 4 and Figure 5, the 2D numerical predicted mechanical stress differs from the analytical computed one, due to unevenly stress distribution. To avoid the mechanical stress concentration on the retainer ring or on the permanent magnets, the inter-pole gap must be filled with a material with similar mechanical properties to those of permanent magnets.

In the rotor with buried magnets, the 2D numerical predicted mechanical stress is in good arrangement with analytical computed one. The maximum stress concentration in the retaining bridges can be reduced by designing bigger radius at the slot edges.

5. Acknowledgment

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6. References

[1] Robert D. Cook, *Finite Element Modeling for Stress Analysis*, John Wiley & Sons, USA, 1995.

[2] Smith D.J.B., Mecrow B.C., Atkinson G.J., Jack A.G., Mehna A.A.A., *Shear Stress Concentrations in Permanent Magnet Rotor Sleeves*, ICEM, Rome, 2010.

[3] Binder A., Schneider T., Klohr M., “Fixation of Buried and Surface-Mounted Magnets in High-Speed Permanent-Magnet Synchronous Machines”, in *IEEE Transactions of Industry Applications*, Vol. 42, No. 4 August 2006.

[4] Fei W., Luk P.C.K., El-Hasan T.S., “Rotor Integrity Design for a High-Speed Modular Air-Cored Axial-Flux Permanent-Magnet Generator”, in *IEEE Transactions of Industry Electronics*, Vol. 58, Issue 9, 2011.

[5] Fei W., Luk P.C.K., Jinupun K., “Design and analysis of high-speed coreless axial flux permanent magnet generator with circular magnets and coils”, in *IEEE Transactions of Electric Power Applications*, Vol. 4, Issue 9, November, 2010.

[6] Lovelace E.C., Jahns T.M., Keim T.A., Lang J.H., “Mechanical Design Considerations for Conventionally Laminated, High-speed, Interior PM Synchronous Machine Rotors”, I *IEEE Transactions of Industry Applications*, Vol. 40, No. 3 June 2004.

[7] JMAG Designer v12.1, User’s manual, http://www.jmag-international.com/

[8] Bremner R.D., “Bridge Stresses and Design in *IPM Machines”, Eurocon*, May 2009.

[9] Kawase Y., Yamaguchi T., Oni Y., Koyanagi W., Nakamura M., “Magnetic Field and Stress Analysis of Interior Permanent Magnet Motor with Off-Centered Rotor Using 3-D Finite Element Method”, in *International Conference on Electric Machines* (ICEM), Sept. 2012.

[10] Eyin H.Z, Haodong Y., Yangsheng C., “Investigation of the Rotor Mechanical Stresses of Various Interior Permanent Magnet Motors”, in *International Conference on Electric Machines and Systems* (ICEMS), Nov. 2009.

[11] Smith D.J.B., Mecrow B.C., Atkinson G.J., Jack A.G., Mehna A.A.A., “Shear Stress Concentrations in Permanent Magnet Rotor Sleeves”, in *International Conference on Electric Machines* (ICEM), Sept. 2010.

[12] Lee K.-J., Ki-Chan K., Ju L., “Rotor Optimization of Interior Permanent Magnet Synchronous Motor Considering Mechanical Stress”, in *Digests of the IEEE International Magnetics Conference*, April 2005.

[13] Calverley S.D., Jewell G.W., Saunders R.J., “Calculation of Centrifugal Stress in Four-Pole Switched-Reluctance Rotors”, in *IEEE Proceedings Electric Power Applications*, Jun 2003.

[14] Jae-Woo J., Byeong-Hwa L., Do-Jin K., Jung-Pyo H., Jae-Young K., Seong-Min J., “Mechanical Stress Reduction of Rotor Core of Interior Permanent Magnet Synchronous Motor”, in *IEEE Transactions of Magnetics*, Vol. 48, No. 2, February 20012.

[15] Jason D.E., Zhu Z.Q., Howe D., “Rotor Resonances of High-Speed Permanent-Magnet Brushless Machines”, in *IEEE Transactions of Industry Applications*, Vol. 39, No. 6 November 2002.

[16] PARK C.H., CHOI S.K., HAM S.Y., “Design and Experiment of 400,000 RPM High Speed Rotor and Bearings for 500W Class Micro Gas Turbine Generator”, in *International Conference on Micro and Nanotechnology for Power Generation and Energy Conversion Applications* (PowerMEMS), 2011.

[17] Krack M., SECANELL M., MERTINY P., “Rotor Design for High-Speed Flywheel Energy Storage Systems”, http://www.intechopen.com/books/energy-storage-in-the-emerging-era-of-smart-grids /rotor-design-for-high-speed-flywheel-energy-storage-systems

[18] http://www.specialmetals.com/documents/Inconel%20alloy%20X-750.pdf

[19] http://www.vacuumschmelze.com/fileadmin/docroot/medialib/documents/broschueren/htbrosch/Pht-004_e.pdf

[20] http://www.carbonfiberglass.com /Composites-Manufacturing/carbon-fiber-glass-composites-services/high-speed-composite-rotor-sleeves.html

7. Biography

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