Mass Optimization Method of a Surface-Mounted Permanent Magnet Synchronous Motor Based on a Lightweight Structure

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ABSTRACT To solve the problem that the propulsion motor of the solar-powered aircraft is difficult to meet the high-torque-density requirement, the mass optimization method of a surface-mounted permanent magnet synchronous motor (SPMSM) based on a lightweight structure is proposed in this paper. According to the motor load generated by propeller, the strength and stiffness analytical models of the mechanical structure are established. By correlating the mechanical dimensions with the electromagnetic dimensions, a full-scale parametric model of SPMSM is obtained. A 4kW and a 167W SPMSMs are designed based on the parametric model. Analysis results show that with the increase of the ratio of motor diameter to length, the motor mass firstly decreases and then increases, and there is a minimum value. The validity of the lightweight structure and analytical models are verified by the finite element models (FEM) and prototypes.

INDEX TERMS Analytical model, lightweight structure, mass optimization, solar-powered aircraft, strength, stiffness, surface-mounted permanent magnet synchronous motor (SPMSM).

I. INTRODUCTION
Solar-powered aircraft is an electric-driven aircraft with solar energy source [1]. The propulsion motor is required light weight, high efficiency, high torque density and high reliability. Therefore, permanent magnet synchronous motor (PMSM) and brushless DC motor (BLDC) have become the first choices for most solar-powered aircrafts [2]–[7].

With the continuous improvement of the high-altitude long-endurance capability of solar-powered aircraft, the torque density of the propulsion motor is required larger than 6Nm/kg, which brings a great challenge to the motor mass reduction. Under the constraints of electric, magnetic and thermal loads, the motor electromagnetic mass is usually difficult to be further reduced, while the mechanical structure still has a lightweight design space under the premise of ensuring strength and stiffness. By referring to [2]–[10], the motor shaft can be hollow and the supporting structure can be composed of beams. The low-density and high-strength material, such as aluminum alloy can be adopted. Unlike traditional commercial motors, the mechanical mass of motors for electric-driven aircraft tends to be lower than electromagnetic mass [3], [10].

The propulsion system of Zeyphr 6 adopted BLDC with external rotor structure. The end cap and stator support structure were both designed with six beams which greatly saved the motor mass. The electromagnetic mass was 385g, and the shaft and support structures were 277g, the maximum torque density was 3.78Nm/kg [3], [4]. Based on Zeyphr motor, the traditional copper wire was replaced with the pre-compressed aluminum wire to reduce the winding mass. The motor was 6.22kg and the maximum torque density was up to 9.64Nm/kg [5]. Heliplat selected BLDC with axial flux disc structure. Instead of traditional coreless supporting structure, advanced composite plastic material is used to further reduce mass [6], [7]. In addition to solar-powered aircraft, light aircraft or gliders also adopted permanent magnet motors in propulsion system [8]–[10]. In [8], special duct design was adopted between rotor and shaft to enhance heat dissipation and reduce motor mass. The motor was 22kg and the maximum torque density was 4.3Nm/kg. In [9], a direct-drive solution of traction motor for all-electric airplane was presented. The motor was 35kg and the torque density was larger than 6.5Nm/kg. To meet NASA’s goal of the turbo-electric propulsion system, an air-core
PM motor was designed and the torque density was 7.5Nm/kg [10].

In the case of lightweight structure, the strength and stiffness must be checked at the design stage. An analytical model of structural mass of axial-flux PM machines for a range of typical wind-turbine ratings was presented in [11]. Results showed that the inactive mass was greater than 60% of the machine total mass, and at multimegawatt ratings it was almost 90% of the total mass. A mechanical design of a PM motor with inside-out configuration for future 737 class hybrid-electric aircrafts was carried in [12]. To reduce the breakage of the rotating shaft for submersible PMSM, two improved schemes with optimal mechanical structures were proposed and verified in [13].

The motor parametric model is helpful to give a lightweight design scheme quickly in the initial design stage and provide reference for subsequent optimization. In recent years, the parametric models of SPMSM stator core and winding have been established based on the motor split ratio [14]–[17], and the parameters of the magnetic poles and rotor core can also be deduced by magnetic circuit method and scalar magnetic potential method [18]–[21]. However, little literatures have discussed the parametric model of the motor mechanical structure. In [22], [23], the parametric models of the internal rotor, external rotor and axial flux motor for airborne wind turbines were proposed respectively. The parametric models included the mechanical structure and described the electromagnetic and thermal behavior.

The paper is organized as follows. In Section II, the SPMSM lightweight structure is introduced and the motor load are summarized. In Section III, the strength and stiffness analytical models of the mechanical structure are established. In Section IV, a full-scale parametric model of SPMSM is obtained. In Section V, a 4kW and a 167W SPMSMs are designed based on the full-scale parametric model and the motor mass distribution is discussed. In Section VI, the validity of the lightweight structure and its analytical models are verified by the FEMs and prototype machines.

II. LIGHTWEIGHT STRUCTURE AND MOTOR LOAD

The SPMSM lightweight structure is shown in Fig.1. The electromagnetic structure includes stator core, winding, magnetic poles and rotor yoke. The mechanical structure includes end caps, housing, shaft and bearings.

The front and rear end cap are respectively equipped with a rabbet for positioning the stator core. The shaft is hollow, and the supporting structures of the rotor and end cap are composed of several beams. The housing integrated with the front end cap not only can protect the stator core, but also provides the installation standard to ensure the concentricity of the front and rear end caps. Practical experience shows that if the housing diameter is large and the thickness is thin, the housing is prone to deformation resulting in the concentricity worse. Therefore, the sufficient housing thickness is maintained and set as a constant in the following parametric model.

The motor load at typical flight states of solar-powered aircraft are shown in Fig.2. During the flight, the propeller will encounter the incoming flow. Under the action of the incoming flow, the propeller will have a speed $n_w$. If the motor speed $n$ is larger than $n_w$, the propeller will generate a positive pull, which is the normal state. If $n$ is less than $n_w$, the propeller will generate a negative pull (thrust), which is the windmill state. At normal state, the motor drives the propeller to rotate forward, and the propeller generates reverse torque to balance the motor torque. The propeller pull is transmitted to the front end cap through the shaft. At windmill state, the propeller drives the motor to rotate forward. For the protection of the battery, the motor is required not to generate reverse torque. Therefore, the motor driver will control the output current to 0, ie, no torque is generated. The propeller thrust is transmitted to the rear end cap through the shaft. Regardless of the flight state, the propeller will both generate an alternating bending moment due to the side-wind. There will be a maximum total bending moment on the shaft when the bending moment coincides with the propeller gravity direction. The bending moment and propeller gravity will be transmitted to the end caps through the shaft. The end caps will also provide reactions to the shaft to balance the bending moment and propeller gravity.

The forces on the shaft and end caps at typical flight states are summarized in Table 1. It is worth noting that the
TABLE 1. Forces on mechanical structure.

| State          | Shaft   | Front end cap | Rear end cap |
|----------------|---------|---------------|--------------|
| Normal         | 1. torque | 1. propeller pull | 1. bending moment |
|                | 2. reverse torque | 2. bending moment |               |
|                | 3. propeller gravity |               |               |
|                | 4. bending moment |               |               |
|                | 5. end cap reaction |               |               |
| Windmill       | 1. propeller gravity | 1. bending moment | 1. propeller thrust |
|                | 2. bending moment |               | 2. bending moment |

III. ANALYTICAL MODEL OF MECHANICAL STRUCTURE

A. SHAFT

The views and dimensions of shaft are shown in Fig.3. There are \( n_{rb} \) supporting beams with equal distribution. \( D_{rbi} \) and \( D_{rb2} \) are the diameters of the supporting beam outer ring. \( D_{rb3} \) and \( D_{rb4} \) are the diameters of the supporting beam inner ring. The length, width and height of the rotor supporting beam is \( L, \ h_{rb} \) and \( h_{rb} \). \( D_{sho} \) and \( D_{sh} \) are the outer and inner diameter of the extended shaft. \( r_{new} \) is the outer radius of end-winding, \( l_{bs} \) is the length of bearing seat, \( l_{sh} \) is the length of the extended shaft and \( l_{shp} \) is the length of propeller mounting surface. It is known that the sizes of the front and rear bearing are the same. The thicknesses of the inner and outer rings of the bearing are both \( h_b \), and the bearing width is \( l_b \). The length of the bearing seat is usually taken as \( l_{bs} = l_b \). The height of the bearing seat should be taken as \( h_{bs} < h_b \).

The height of the non-positioning shoulder \( h_{bs} \) for convenient installation of bearing is determined according to the inner diameter of the appropriate bearing \( D_{bi} = D_{sho} + 2h_{bs} \).

It is assumed that the motor torque \( T \) is evenly distributed on the rotor yoke, and the propeller reverse torque is evenly distributed on the mounting surface. Then the maximum shear stress \( \tau \) is on the extended shaft surface and the torsion angle per unit length \( \varphi \) of the extended shaft is the largest.

\[
\begin{align*}
\tau &= \frac{T}{W_p} < \sigma_{rb} \\
\varphi &= \frac{2T}{GD_{sho}W_p} < [\psi] \quad (1)
\end{align*}
\]

where \( G \) is the shear modulus, \( W_p \) is the torsional section coefficient and \( [\psi] \) is the allowable torsion angle per unit length. For general drive shaft, \( [\psi] \) is taken within 0.5-1.0°/m.

The supporting beam of the shaft is equivalent to a cantilever beam. The inner ring of the supporting beam is regarded as the fixed end, then the maximum stress is on the inner ring and the maximum deformation is on the outer ring.

\[
\begin{align*}
\sigma_{rb} &= \frac{h_{rb}b_{rb}T}{n_{rb}D_{rb2}l_{rb}} < [\sigma_{rb}] \\
W_{rb} &= \frac{2h_{rb}^3T}{3n_{rb}D_{rb2}EI_{rb}} < [w_{rb}] \quad (2)
\end{align*}
\]

where \( E \) is Young’s modulus of shaft material and \( I_{rb} \) is the moment of inertia of the supporting beam. \( [\sigma_{rb}] \) is the allowable stress of shaft material and \( [w_{rb}] \) is the allowable deformation which can be taken as 0.002\( l_{rb} \).

If the rear end cap is regarded as a fixed hinge support and the front end cap as a movable hinge support, then the shaft can be equivalent to an overhanging beam including three segments as shown in Fig.4.

FIGURE 3. Views and dimensions of shaft.

FIGURE 4. Equivalent overhanging beam.
Then the bending moment of the three segments can also be derived. The maximum bending moment is at the junction of segment I and II. Because the outer diameter of the extended shaft is the smallest, the cross section \((x = \ell_{mo})\) is considered dangerous and the corresponding bending stress is the largest.

\[
\sigma = \frac{0.5G I_{shp} + G I_{sh} + M}{W_b} \tag{4}
\]

where \(W_b\) is the bending section coefficient.

The composite stress of the dangerous section can be calculated as (5), where the bending stress \(\sigma\) is symmetrical cyclic variable stress, and the torsional shear stress \(\tau\) is static stress. The conversion coefficient \(\alpha = 0.3\) is introduced to consider the different effects of cyclic characteristics.

\[
\sigma_{-1} = \sqrt{\sigma^2 + 4(\alpha \tau)^2} \leq [\sigma_{-1}] \tag{5}
\]

where \([\sigma_{-1}]\) is allowable bending normal stress of the shaft under symmetrical cyclic variable stress.

It is assumed that the shaft including three segments is of equal cross-section, then the deflection curve equations of three segments can be obtained.

\[
\begin{align*}
 w_1(x) &= \frac{1}{EI} \left[ \frac{(I_{shp} + 2I_{sh})G + 2M_{add}}{12\ell_{mo}} x^3 + A_1 x \right] \\
 w_2(x) &= \frac{1}{EI} \left[ -\frac{G}{6}(x - \ell_{mo})^3 + B_1 x + B_2 \right] + \frac{1}{3} \left( 2GI_{sh} + GI_{shp} + 2M_{add} \right) x^2 \\
 w_3(x) &= \frac{1}{EI} \left[ G \left( \frac{1}{12}(x - \ell_{mo} - I_{sh})^3 \right) + C_1 x + C_2 \right]
\end{align*}
\]

(6)

where \(I\) is the moment of inertia, \(A_1\), \(B_1\), \(B_2\), \(C_1\) and \(C_2\) are coefficients.

\[
\begin{align*}
 A_1 &= -\frac{(I_{shp} + 2I_{sh})G + 2M_{add}}{12\ell_{mo}} \ell_{mo} \\
 B_1 &= -\frac{(I_{shp} + 2I_{sh})G + 2M_{add}}{12\ell_{mo}} \ell_{mo} \\
 B_2 &= \frac{(I_{shp} + 2I_{sh})I_{shp} + 2M_{add}}{12\ell_{mo}} I_{mo} \\
 C_1 &= \frac{1}{6} \left( 3I_{sh}^2 - 2I_{sh}I_{mo} + 2I_{sh}I_{mo} \right) + \frac{1}{3} M \ell_{mo} + 3I_{sh} \\
 C_2 &= \frac{1}{12} G \left( 2I_{sh}^2 + 6I_{sh}I_{mo} + 4I_{sh}I_{mo} - \ell_{mo}^2 \ell_{sh} \right) - \frac{1}{6} M \left( 2I_{mo}^2 + 3I_{sh}^2 + 6I_{sh}I_{mo} \right)
\end{align*}
\]

(7)

Then the deflection angles of the front bearing \(\theta_1\) and rear bearing \(\theta_2\), radial runout of the shaft end \(w_1\) and maximum radial deformation of the shaft at air gap \(w_2\) can be obtained and the following conditions should be met.

\[
\begin{align*}
 \theta_1 &= w_1'(l_{mo}) = \frac{(I_{shp} + 2I_{sh})G + 2M_{add}}{6EI} \ell_{mo} \leq [\theta] \\
 \theta_2 &= |w_1'(0)| = \frac{(I_{shp} + 2I_{sh})G + 2M_{add}}{12EI} \ell_{mo} \leq [\theta] \\
 w_1 &= w_1(l_{mo} + I_{sh} + I_{shp}) \leq [w_1] \\
 w_2 &= w_2(\sqrt{3} \ell_{mo}) = \frac{\sqrt{3} (I_{shp} + 2I_{sh})G + 2M_{add}}{54EI} \ell_{mo}^2 \leq [w_2]
\end{align*}
\]

(8)

where \([\theta]\) is the allowable deflection angle of shaft, for the ball bearings, \([\theta] = 0.001\) rad. \([w_1]\) is the allowable radial runout of the shaft end. To avoid the motor rotor being in contact with the stator, \([w_2] = 0.05 l_g\), \(l_g\) is the air-gap length.

**B. END CAP**

The views and dimensions of end cap are shown in Fig.5. There are \(n_{c,b}\) beams with equal distribution. \(D_{cp1}\) and \(D_{cp2}\) are the diameters of the end cap outer ring. The thickness of the outer ring is equal to the housing thickness plus the rabbet height. To avoid the interference of the end cap and the end-winding, the length of the end cap is taken as \(l_p = r_{oe} + h_{bo}\). \(D_{cp3}\) and \(D_{cp4}\) are the diameters of the end cap inner ring. The length, width and height of the end cap beam is \(l_{cb}, b_{cb}\) and \(h_{cb}\). The bearing outer diameter is \(D_{bo}\). The length and height of the bearing chamber are \(l_{bb} = l_b, h_{bb} < h_b\) respectively.

**FIGURE 5. Equivalent overhanging beam.**

It is assumed that the rotation angles of the bearing chamber and outer edge are zero under the propeller pull \(F\), then the end cap can be equivalent to a beam \(AB\). The bearing chamber is equivalent to the midpoint \(C\) of the beam, and the equivalent axial force \(F_a\) acts at the midpoint \(C\). The \(AC\) and \(BC\) segments represent the two opposite beams of the end cap. The horizontal reactions are ignored, then the basic statically determinate system is selected as simply supported beam shown in Fig. 6.

Because the geometric shape, physical properties and load of the beam are symmetrical to the midpoint \(C\), the reaction forces will also be symmetrical. According to the static equilibrium equation and the deformation geometric compatibility equation, the supporting reactions and couple moments
The change of the angle between the rods can be neglected. It is assumed that the rods’ forces are tension, then the tension of each rod can be derived from Hooke’s law and the equilibrium equation of node O.

\[
F_i = \frac{2F_{N'}}{n_{cb}} \cos \alpha_i \quad (12)
\]

Then the maximum tension stress and the maximum deformation are both located at the beam in the \( F_{N'} \) direction.

\[
\begin{align*}
\sigma_r &= \frac{2F_{N'}}{n_{cb}b_i h_{cb}} < [\sigma_{cb}] \\
\Delta l_{cb} &= \frac{2F_{N'} l_{cb}}{n_{cb}E_{cb}b_i h_{cb}} < [\Delta l_{cb}]
\end{align*}
\]

where \([\Delta l_{cb}]\) is the allowable radial deformation which is usually taken as 0.01mm to ensure the concentricity.

To make sure that the beams under compression can be in stable equilibrium state, the maximum tension \( F_{r\max} \) must meet the following condition.

\[
F_{r\max} = \frac{2F_{N'}}{n_{cb}} < F_{cr} = \frac{\pi^2 E_{cb} I_{cb}}{l_{cb}^2} \quad (14)
\]

where \( F_{cr} \) is the critical force of the beam under compression.

**IV. FULL-SCALE PARAMETRIC MODEL**

For sinusoidal current-driven PMSM, the electromagnetic power can be given by

\[
P_{em} = 3E_1 I_1 \quad (15)
\]

where \( I_1 \) is the fundamental effective value of phase current, \( E_1 \) is the fundamental effective value of no-load phase back-electromotive force (EMF) which can be derived as

\[
E_1 = \frac{\pi}{30} N_a K_{dpl} B_{sl} D_{si} L n \quad (16)
\]

where \( N_a \) is the number of turns-in-series per phase, \( K_{dpl} \) is the fundamental winding factor, \( B_{sl} \) is the fundamental effective value of air-gap flux density, \( D_{si} \) is the stator inner diameter, \( L \) is the motor active length, and \( n \) is the speed.

The electric load is defined as

\[
A = \frac{6N_a I_1}{\pi D_{si}} \quad (17)
\]

Then the main dimensions of motor can be obtained according to (15)-(17).

\[
D_{si}^2 L = \frac{2}{\pi} \frac{T_{em}}{K_{dpl} B_{sl} A} = C \quad (18)
\]

where \( T_{em} \) is the electromagnetic torque and \( C \) is a constant. It is clearly shown that if the pole and slot combination is maintained and the magnetic load \( B_{sl} \) and electric load \( A \) remain unchanged, the main dimensions of motor \( (D_{si}^2 L) \) will be fixed for a given electromagnetic torque.

As the motor is powered by batteries, the bus voltage is fixed. The motor phase back-EMF must be designed constant. Then \( N_a \) will be proportional to \( D_{si} \).

With the center line of the magnetic pole as the coordinate origin, the radial component of magnetic field in the slotless
air gap of SPMSM has been derived in [18].

\[ B_{sl}(\theta) = \sum_{i=1,3,5,...}^{\infty} \frac{4 B_r}{\pi} \sin \left( \frac{i \pi \alpha_p}{2} \right) \frac{i p}{(ip)^2 - 1} \]

where \( B_{sl} \) is the \( i \)th harmonic of the air gap flux density \( B_\delta \), \( D_\delta \) is the air gap diameter, \( D_m \) and \( D_r \) is the outer and inner diameter of magnetic pole respectively, \( p \) is the pole pair, \( \alpha_p \) is the pole arc coefficient, \( B_r \) is the magnet remanence, \( \mu_r \) is the relative recoil permeability. According to (19), with the change of \( D_{st} \), the magnetic load \( h_m \) can be guaranteed fixed by adjusting the magnetic pole thickness \( h_m \).

It is assumed that there is no leakage and the magnetic flux enters the core vertically, then the stator tooth width \( b_{st} \), stator yoke thickness \( b_{sy} \) and rotor yoke thickness \( b_{ry} \) can be derived based on the theorem of magnetic flux continuity [14–17].

\[ b_{st} = \frac{D_\delta}{2B_{sm}} \int_{\frac{\pi}{2}}^{\frac{\pi}{2}} B_\delta(\theta) d\theta \]

\[ b_{sy} = \frac{D_\delta}{4B_{sm}} \int_{\frac{\pi}{4}}^{\frac{\pi}{4}} \frac{B_\delta(\theta)}{b_{st}} d\theta \]

\[ b_{ry} = \frac{D_\delta}{4B_{rm}} \int_{\frac{\pi}{2}}^{\frac{\pi}{2}} \frac{B_\delta(\theta)}{b_{st}} d\theta \]

where \( \gamma_s \) is the slot pitch, \( \tau \) is the pole pitch, \( q \) is the slot number per pole, \( B_{sm} \) and \( B_{rm} \) is the maximum flux density in the stator and rotor core respectively.

Neglecting the tooth tip, the area of the simplified stator slot can be obtained as

\[ A_s = \frac{\pi}{4N_s} (D_{so}^2 - D_{si}^2) - \frac{\pi}{4N_s} (D_{so}^2 - 2D_{sy} - D_{si}) \]

Then the current density of the winding can be given as

\[ J = \frac{6N_s I_1}{K_{Cu} N_s A_s} \]

where \( K_{Cu} \) is the slot fill factor, \( N_s \) is the slot number.

The thermal load is defined as

\[ H = AJ \]

To keep the thermal load unchanged, the current density \( J \) also needs to be fixed. With the constant \( K_{Cu} \), the appropriate stator outer diameter \( D_{so} \) can be deduced.

Then the average end-winding length of the double-layer concentrated winding can be obtained by referring to [15].

\[ l_{ew} = \pi r_{ew} \]

where

\[ r_{ew} = 0.25 b_{st} + 0.125 (D_{so} - 2b_{sy} + D_{si}) \sin \left( \frac{\pi}{N_s} \right) \]

In view of the above, the flow chart of the electromagnetic dimension design is shown in Fig.8. The main dimensions can be decided firstly based on the magnetic and electric loads. Then the remaining dimensions can be deduced based on the main dimensions and thermal load.
Finally, according to the propeller interface, the extended shaft length and outer diameter are obtained. Then the bearing type can be selected. Next, combined with the electromagnetic dimensions, some dimensions of the shaft and end cap can be directly determined. Finally, the remaining dimensions can be deduced based on the analytical models in Section III.

Based on the above two flow charts, the full-scale parametric model of SPMSM including the electromagnetic structure and mechanical structure is established.

V. MASS OPTIMIZATION

After getting all the dimensions of the motor, the total mass of the motor can be calculated, which is the objective function of the optimization. The mass is divided into electromagnetic part and mechanical part. The specific calculations are as follows.

\[
m_{\text{total}} = \left( \frac{m_{\text{sc}} + m_{\text{ry}} + m_{\text{mag}} + m_{\text{Cu}} + m_{\text{im}}}{\text{electromagnetic mass}} \right) + \left( \frac{m_{\text{shaft}} + 2m_{\text{cap}} + 2m_{b} + m_{\text{hou}}}{\text{mechanical mass}} \right)
\]

(28)

where \(m_{\text{sc}}, m_{\text{ry}}, m_{\text{mag}}, m_{\text{Cu}}, m_{\text{im}}, m_{\text{shaft}}, m_{\text{cap}}, m_{b}, m_{\text{hou}}\) is the mass of stator core, rotor yoke, magnets, winding, impregnation, shaft, end cap, bearing and housing respectively.

This paper focuses on the details of the design and optimization of the 4kW motor. The main parameters of the 4kW motor are shown in Table 2. The miniature heat pipes are applied to insert into the stator slots to cool the winding. Due to the excellent heat dissipation effect, the current density of the overload torque can be designed as 10 A/mm\(^2\). Therefore, the current density of the rated torque is set to 4.84 A/mm\(^2\).

\[
\begin{array}{|c|c|c|c|c|c|c|c|}
\hline
\text{Rated power (kW)} & 4.09 & \text{Rated speed (rpm)} & 1700 \\
\text{Rated torque (N·m)} & 23 & \text{Overload torque (N·m)} & 50 \\
\text{Magnetic load (T)} & 0.82 & \text{Bus voltage (V)} & 180 \\
\text{Electric load (kA/m)} & 24.65 & \text{Current density (A/mm\(^2\))} & 4.84 \\
\text{Pole number} & 22 & \text{Slot number} & 24 \\
\hline
\end{array}
\]

(34)

TABLE 3. Electromagnetic dimensions (mm).

\[
\begin{array}{|c|c|c|c|c|c|c|}
\hline
D_{s} & L & D_{sh} & h_{b} & b_{a} & b_{n} & r_{ew} \\
\hline
120.0 & 61.0 & 160.3 & 4.8 & 6.4 & 3.2 & 8.9 \\
130.0 & 51.9 & 171.3 & 4.3 & 7.0 & 3.5 & 9.6 \\
140.0 & 44.8 & 182.2 & 4.1 & 7.5 & 3.8 & 10.3 \\
150.0 & 39.0 & 193.2 & 3.9 & 8.1 & 4.0 & 10.9 \\
160.0 & 34.3 & 204.0 & 3.8 & 8.6 & 4.3 & 11.6 \\
170.0 & 30.4 & 214.8 & 3.7 & 9.1 & 4.6 & 12.3 \\
180.0 & 27.1 & 225.8 & 3.6 & 9.7 & 4.8 & 12.9 \\
190.0 & 24.3 & 236.5 & 3.5 & 10.2 & 5.1 & 13.6 \\
200.0 & 21.9 & 247.3 & 3.5 & 10.8 & 5.4 & 14.2 \\
210.0 & 19.9 & 258.0 & 3.4 & 11.3 & 5.6 & 14.9 \\
220.0 & 18.1 & 268.8 & 3.4 & 11.8 & 5.9 & 15.6 \\
\hline
\end{array}
\]

The motor load is shown in Table 4 and the constraints of strength and stiffness are summarized in Table 5, in which the materials of the shaft and end cap are both aluminum alloy, and the safety factor is 5.

\[
\begin{array}{|c|c|c|c|}
\hline
\text{Propeller pull/thrust (N)} & 300 & \text{Propeller gravity (N)} & 32 \\
\text{Reverse torque (N·m)} & 50 & \text{Bending moment (N·m)} & 40 \\
\hline
\end{array}
\]

(37)

The proposed optimization method is not limited by the power level of the motor. A 4kW motor and a 167W motor for two solar-powered aircrafts are chosen as examples to illustrate the effectiveness of the optimization method.

According to the flow chart in Fig.8, combined with Table 2, the electromagnetic dimensions with different ratios of diameter to length (\(D_{s}/L\)) are listed in Table 3.

\[
\begin{array}{|c|c|c|c|c|c|c|}
\hline
D_{s} & L & D_{sh} & h_{b} & b_{a} & b_{n} & r_{ew} \\
\hline
120.0 & 61.0 & 160.3 & 4.8 & 6.4 & 3.2 & 8.9 \\
130.0 & 51.9 & 171.3 & 4.3 & 7.0 & 3.5 & 9.6 \\
140.0 & 44.8 & 182.2 & 4.1 & 7.5 & 3.8 & 10.3 \\
150.0 & 39.0 & 193.2 & 3.9 & 8.1 & 4.0 & 10.9 \\
160.0 & 34.3 & 204.0 & 3.8 & 8.6 & 4.3 & 11.6 \\
170.0 & 30.4 & 214.8 & 3.7 & 9.1 & 4.6 & 12.3 \\
180.0 & 27.1 & 225.8 & 3.6 & 9.7 & 4.8 & 12.9 \\
190.0 & 24.3 & 236.5 & 3.5 & 10.2 & 5.1 & 13.6 \\
200.0 & 21.9 & 247.3 & 3.5 & 10.8 & 5.4 & 14.2 \\
210.0 & 19.9 & 258.0 & 3.4 & 11.3 & 5.6 & 14.9 \\
220.0 & 18.1 & 268.8 & 3.4 & 11.8 & 5.9 & 15.6 \\
\hline
\end{array}
\]

Then the mechanical dimensions can be optimized based on the electromagnetic dimensions, motor load and constraints of strength and stiffness. The constant mechanical dimensions in the analytical model which are not changed during the optimization are listed in Table 6. It should be noted that the outer diameter and length of the extended shaft are determined by the propeller interface.

A. SHAFT

According to the formulas (1) (4) (5), \(\psi\) and \(\sigma_{-1}\) are independent of \(l_{mo}\), and are related to \(D_{sho}, D_{shi}, l_{shp}\), and \(l_{sh}\). As \(D_{sho}, l_{shp}\), and \(l_{sh}\) are fixed by the propeller interface, \(\psi\) and \(\sigma_{-1}\) are only affected by \(D_{shi}\). The variation of \(\psi\) and \(\sigma_{-1}\) with \(D_{shi}\)
TABLE 5. Constraints of strength and stiffness.

|        | Shaft | End cap |
|--------|-------|---------|
| $[\sigma_h]$ | 91Mpa | $[\sigma_s]$ | 91Mpa |
| $[w_{al}]$ | 0.002lsh | $[w_{as}]$ | 0.1mm |
| $[\sigma_s]$ | 45Mpa | $[\Delta l_h]$ | 0.01mm |
| $[\phi]$ | 0.8° | $F_{cr}$ | $\pi^2E_{al}/l^2$ |
| $[\theta]$ | 0.001rad | — | — |
| $[w_1]$ | 0.12mm | — | — |
| $[w_2]$ | 0.05mm | — | — |

**TABLE 6. Constant mechanical dimensions (mm).**

|        | Shaft | End cap |
|--------|-------|---------|
| $D_{A1}$-$D_{A2}$ | 8 | $D_{Ap1}$-$D_{Ap2}$ | 14 |
| $D_{A3}$-$D_{A4}$ | 4 | $D_{Ap3}$-$D_{Ap4}$ | 14 |
| $D_{As}$ | 38 | $D_{As}$ | 62 |
| $h_{bp}$ | 88 | $h_b$ | 12 |
| $l_s$ | 55 | $h_{sh}$ | 2 |
| $l_a$ | 12 | $h_{cb}$ | 7 |
| $h_{a}$ | 2 | — | — |
| $h_{sh}$ | 1 | — | — |

is shown in Fig.10(a). According to the formulas (6)-(9), $w_1$, $w_2$ and $\theta_1$ are affected by both $l_{mo}$ and $D_{shi}$. Since one $D_{sh}$ corresponds to one $l_{mo}$, the variations of $w_1$, $w_2$ and $\theta_1$ with $D_{shi}$ and $D_{si}$ are shown in Fig.10(b)-(d). With the increase of $D_{shi}$, $\phi$, $\sigma_{-1}$, $w_1$, $w_2$ and $\theta_1$ all increase slowly first, then increase rapidly. Under the same $D_{shi}$, $w_1$, $w_2$ and $\theta_1$ all decrease with the increase of $D_{si}$. Within the range of stator inner diameter (120mm-220mm), $[\phi]$ is the constraint for first limiting the further increase of the shaft inner diameter. Therefore, the maximum $D_{shi}$ will not change with $D_{si}$.

Taking six beams as example, the deformation and stress of the shaft supporting beam with different widths are shown in Fig.11. It is shown that the deformation constraint firstly limits the further reduction of the beam width.

**B. END CAP**

It can be seen from (11) that when the propeller pull $F$ and $l_{sh}$ remain unchanged, under the constraint of $[w_{cb}]$, the beam volume is inversely proportional to the square of $h_{cb}$, but independent of $n_{cb}$ and $b_{cb}$, that is, $n_{cb}$ and $b_{cb}$ are inversely proportional to each other. Therefore, to reduce the end cap mass, $h_{cb}$ should be increased. Besides, the beam volume is proportional to the fourth power of $l_{cb}$, so the beam volume will increase with the increase of the motor outer diameter.

It can be seen from (13) that when the force $F_N'$ from shaft and $l_{sh}$ remain unchanged, under the constraint of $[\Delta l_{cb}]$, the beam volume is independent of $n_{cb}$, $b_{cb}$ and $h_{cb}$, that is, the total cross-sectional area of the beams is kept unchanged.

The deformations of the end cap consisted of six beams with different beam widths under propeller pull and bending moment are respectively shown in Fig.12 and Fig.13. It is shown that $[w_{cb}]$ firstly limits the further reduction of the beam width. It should be noted that the stresses of end cap under propeller pull and bending moment are both much smaller than the boundary condition, so they are not shown.

**FIGURE 10. Shaft under bending moment and torque.**

After the dimensions of the electromagnetic and mechanical structures are determined, each part mass can be obtained. The mass distribution of electromagnetic structure is shown in Fig.14 a). With the increase of the ratio of diameter to length, the masses of stator core and magnets decrease obviously. The masses of winding and impregnation firstly decreases and then increases. The rotor yoke mass hardly changes. On the whole, with the increase of the ratio of diameter to length, the electromagnetic mass firstly decreases rapidly and then increases slowly.
The mass distribution of mechanical structure is shown in Fig. 14 b). As the outer diameter and length of the extended shaft are determined by the propeller interface, and the inner diameter of the extended shaft remains unchanged under the boundary conditions, the mass of the extended shaft will not change with the stator inner diameter. For the part of the shaft between the bearings, its mass decreases with the decrease of the motor length. As the shaft outer diameter is fixed, a bearing with a suitable inner diameter to match the shaft will be found. When the motor diameter changes, the height of the supporting beams will change accordingly, while the outer diameters of the shaft and bearing can remain unchanged. Therefore, the bearing size will not change, that is, the bearing mass remains fixed. To guarantee the axial deformation, as the length of the end cap beam increases, the corresponding beam width also increases. Therefore, the mass of end caps increases obviously with the increase of the stator inner diameter. Since the housing thickness is fixed to 5mm in the parametric model, the housing mass will decrease. On the whole, with the increase of the ratio of diameter to length, the mechanical mass firstly decreases slowly and then increases rapidly.

The total mass distribution is shown in Fig. 14 c). As the stator inner diameter increases from 120mm to 220mm, the total mass firstly decreases and then increases. When the stator inner diameter is 160mm, i.e., $\frac{D_s}{L} = 4.7$, the total mass reaches a minimum of 6.79kg. The corresponding mass
proportions of the electromagnetic and mechanical structure are 65% and 35% respectively.

To fully prove the generality of the variety law of motor mass based on the proposed lightweight structure, the mass distribution of 167W motor is shown in Fig.15. It can be seen that the mass variety trend is basically the same as that of 4kW motor. When the stator inner diameter is 80mm, i.e. \( D_{si}/L = 4.0 \), the total mass reaches a minimum of 976g. The corresponding mass proportions of the electromagnetic and mechanical structure are 75% and 25% respectively.

VI. VERIFICATION

According to the dimensions in Table 3, the corresponding 2-D electromagnetic FEMs (1/2 model) are established, part of them are shown in Fig.16. The element type for all meshed models is triangle.

In consideration of the motor loss, the electromagnetic torque is designed to be 24N·m. As shown in Fig.17, the simulated values are close to the target value as a whole and the maximum error is no more than 2%. The main reason of error is that the effect of slotting on the air-gap magnetic field is not considered in the analytical model, that leads to the smaller simulated torque. Another reason is that \( N_a \) in analytical model can be decimal, while it must be set as integer in FEM.

Taking the design of \( D_{si} = 180 \)mm as an example, the corresponding shaft and end cap dimensions are listed in Table 7 and Table 8 respectively. Then the mechanical FEMs are established. The meshed FEM of end cap consists of 37596 nodes and 20623 elements, and the meshed FEM of shaft consists of 441279 nodes and 289906 elements. Element type for both models is tetrahedron.
The simulations of shaft under torque are shown in Fig.18. The maximum deformation of the supporting beam is at the shaft outer ring and the maximum stress is at the shaft inner ring. As the simulation only shows deformation, the torsion angle per unit length needs to be calculated indirectly by the deformation, and the extended shaft radius.

The simulations of shaft under propeller gravity and bending moment are shown in Fig.19. The most serious deformation is at the shaft front end, and the maximum bending stress is at the extended shaft root. In Fig.19, the two bearing seats are set as fixed surfaces, so the simulated $w_2$ and $\theta_1$ are very small, which are much smaller than the analytical results, which is caused by the inconsistency between the simulation setting and the analytical model.

The simulations of end cap under propeller pull are shown in Fig.20. The maximum deformation of the beam is at the bearing chamber and the maximum stress of the beam is at the end cap outer edge. The simulations of end cap under shaft force are shown in Fig.21. The shaft force is equivalent to a radial force which is uniformly distributed on the internal surface of the bearing chamber. The maximum deformation of the beam is also at the bearing chamber. The maximum stress of the beam is uniformly distributed on the beam which is consistent with the direction of the radial force.

The comparisons between the analytical and simulation results are shown in Table 9. The maximum error is less than 15% and all errors are acceptable.

The 4kW and 167W prototypes and test platforms are shown in Fig.22 and Fig.23 respectively, and their main dimensions are listed in Table 10. Both motors adopt cooling method of heat pipe inserting into winding. The stator inner diameters of 4kW and 167W prototypes are 177mm.
and 74.8mm respectively. According to the mass distribution results in Fig.14 and Fig.15, the diameters are in the range of low total mass. There are some differences between the prototype mechanical structures and the analytical models proposed in this paper. For the 4kW prototype, the end cap adopts 8 supporting beams, and each beam is connected by a ring to provide better strength under the action of bending moment. The housing is partially hollowed out, and the strength is guaranteed by the triangular beam structure. The shaft adopts 5 supporting beams, and the beams are provided with stiffeners, which improve the torsional resistance. For the 167W prototype, the end cap also adopts 8 supporting beams including 4 main beams and 4 auxiliary beams. There is no housing design, and the front and rear end caps are fixed on the stator core by 4 bolts. It should be noted that 167W motor is small in volume and low in torque, so the end caps can be fixed in this way. 4kW motor has also used 8 bolts instead of housing to fix the end caps, but the motor vibration was serious. Only when the housing has been added, the vibration can be eliminated.

The mass details of 4kW and 167W prototype are shown in Table 11. The mass details of the Zephyr 6 propulsion motor in [3] is also cited for reference, and the motor power is 88W. Comparing the masses in table 9 with those in Fig.14 and Fig.15, it can been seen that the electromagnetic mass of 4kW prototype is 260g more than the design, and the mechanical mass is 170g lower than the design. The electromagnetic mass of 167W prototype is 63g more than the design, and the mechanical mass is 5g lower than the design. The reason for the error is that the prototype needs to have a more detailed electromagnetic optimization to consider the distortion rate of the back-EMF and the loss of each part. Meanwhile, the mechanical structure also needs to be adjusted to meet the actual assembly requirements and the interface requirements of the propeller and the installation.

| Dimension          | Symbol | 4kW | 167W |
|--------------------|--------|-----|------|
| Stator outer diameter | $D_{s}$ | 226 | 100.5 |
| Stator inner diameter  | $D_{s}$ | 177 | 74.8 |
| Rotor outer diameter  | $D_{r}$ | 175.2 | 73.6 |
| Rotor inner diameter  | $D_{I}$ | 156 | 66.4 |
| Stator length       | $L$   | 28 | 26 |
| Magneter height     | $b_{m}$ | 3.6 | 2.3 |
| Shaft outer diameter | $D_{h}$ | 38 | 18 |
| Shaft inner diameter | $D_{i}$ | 29 | 14 |
| Extended shaft length | $l_{e}$ | 88 | 27 |
| Propeller mounting length | $l_{p}$ | 55 | 25 |
| End cap beam height | $k_{s}$ | 7 | 2.5 |
| End cap beam width  | $b_{s}$ | 7.5 | 6 |
| Shaft supporting beam width | $b_{s}$ | 4 | 1.8 |
| Bearing outer diameter | $D_{o}$ | 62 | 32 |
| Bearing inner diameter | $D_{i}$ | 40 | 20 |
| Bearing width       | $l_{b}$ | 12 | 7 |

The mechanical mass of 4kW and 167W prototypes accounts for 34% and 23% of the total mass respectively, and the mechanical mass of 88W motor accounts for 42%. Although the motor is external rotor structure, which is different from the internal rotor structure in this paper, it can be seen from [3] that 88W motor has only one end cap. If the internal rotor structure with two end caps is adopted,
the proportion of 88W motor mechanical mass should be higher. Therefore, by referring to the Zeyphr 6 motor, it is proved that the proposed lightweight structure can further reduce the proportion of mechanical mass.

Both prototypes have been successfully tested for load and the experiment data are shown in Table 12. In addition to the mass in Table 11, the prototype mass in Table 12 also includes the cooling system, the resolver and the cable joint. At room temperature, under forced air cooling of 4m/s, the temperature rise of two prototypes are 47°C and 53°C respectively at the overload point. The data of the 88W motor are also listed as a comparison standard. It can be seen that the two prototypes have higher torque density. It should be noted that the 88W motor adopts a brushless dc drive, the overload current in Table 12 is the peak value of the square wave current, and it is estimated based on the current waveform at 1.9N-m provided in [3]. The 88W motor has a larger overload current density, but the temperature rise is under 30°C, which should benefit from the good heat dissipation conditions of the open external rotor structure.

VII. CONCLUSION
To meet the high-torque-density requirement, the mass optimization method of a SPMSM for solar-powered aircraft based on a lightweight structure is proposed. The strength and stiffness analytical models of the shaft and end cap are obtained from the overhanging beam, simply supported beam and hinged rod models. An analytical full-scale parametric model of SPMSM including mechanical and electromagnetic structures is established. A 4kW and a 167W SPMSMs are designed based on the analytical model. The results show that with the increase of the ratio of motor diameter to length, the electromagnetic mass firstly decreases rapidly and then increases slowly, the mechanical mass firstly decreases slowly and then increases rapidly, and the total mass firstly decreases and then increases and there is a minimum value. The validity of the lightweight structure and its analytical models are verified by the FEMs and prototype machines. By referring to the mass distribution of Zeyphr 6 motor, the proposed lightweight structure is proved to further reduce the proportion of mechanical mass.

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