

Numerical and Experimental Investigation of Heat Flow in Permanent Magnet Brushless DC Hub Motor

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ABSTRACT

This paper investigates the heat dissipation in the hub motor of an electric two-wheeler using lumped parameter (LP), finite element (FE) and computational fluid dynamic (CFD) models. The motor uses external rotor permanent magnet brushless DC topology and nearly all of its losses are generated in the stator. The hub motor construction restricts the available conductive paths for heat dissipation from the stator to the ambient only through the shaft. In contrast to an internal rotor structure, where the stator winding losses are diffused via conduction, here convection plays a major role in loss dissipation. Therefore, a LP thermal model with improved convection modelling has been proposed to calculate the temperature of the components inside the hub motor. The developed model is validated with the FE thermal model and the test data. In addition, CFD tools has been used to accurately model the internal and the external flow as well as the convective heat transfer of the hub motor. Finally, an optimization study of the hub motor has been carried out using the CFD model to improve heat transfer from the stator.

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INTRODUCTION

Major cities around the world are either implementing or considering the stringent environmental pollution regulations. This, along with the increasing price of fossil fuels and the growth of renewable energy has brought electric propulsion back to the market. The electric propulsion uses an electrical energy storage such as batteries or fuel cells, along with electric drives instead of an IC engine powertrain. Electric motor in an electric drivetrain can be incorporated in number of ways and hub motor, especially in direct drive configuration, offers a number of benefits such as improved efficiency, increased cabin space in four wheelers and weight reduction in two wheelers without affecting the ride and handling [1].

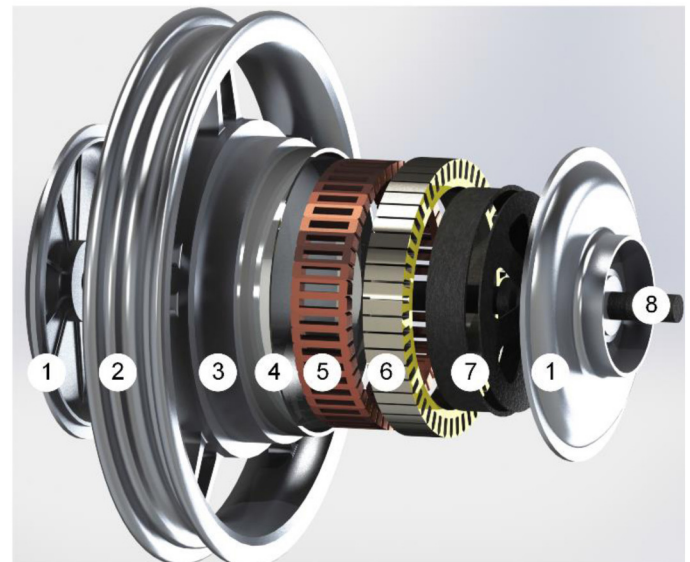


Figure 1. Exploded view of the motor (1. Endplate, 2. Rim, 3. Rotor core, 4. Magnet, 5. Winding, 6. Stator core, 7. Spider and 8. Non rotating shaft).

The hub motor investigated in this paper is taken from an electric two wheeler currently available in the market. The continuous rating of the motor is 750 W at 550 rpm and it has an external rotor radial flux permanent magnet brushless DC (PMBLDC) topology. The exploded view of the motor is shown in Fig. 1. The stator comprising of the stator core and the winding is attached to a non-rotating shaft via spider. The shaft is then connected to the two wheeler chassis. The wheel houses the rotor core and magnets and they are connected to the shaft via a set of endplates with bearing. An air gap of 0.5 mm separates the stator core and magnets.

A good thermal design of an electric motor not only helps to limit the temperature of its components within the thermal limit of their materials, but also allows the overall size reduction of motor [2]. A detailed thermal study is essential in case of the PMBLDC hub motor as nearly all of its losses are generated in the stator and their construction restricts available conductive paths for the heat dissipation to ambient only through the shaft. In contrast to an internal rotor structure, where the stator winding losses are diffused via conduction, here convection plays a major role in dissipating the losses.

In this work, a lumped parameter (LP) thermal model of PMBLDC hub motor is developed based on already available models [3, 4, 5, 6, 7]. The LP thermal model can be solved in minimum time and hence, it could be part of an electrical design program. This will help the designer to evaluate the effects of change in ambient temperature and increased internal temperatures due to loading, on the motor performance. This is especially important in case of electric vehicle powertrain, as the motors are exposed to considerable temperature variation compared to the motors used in other applications. In addition, due to significant simplification, LP thermal model is computationally less demanding and could be used as a part of control algorithms to assess critical temperature rise in a real time and adjust control accordingly.

The developed LP model is validated with the finite element (FE) thermal model and the test data. The FE method can model more complex shapes and therefore predict more accurately than the LP thermal model. A sensitivity analysis of the LP thermal model has revealed that, the temperature of the components are highly influenced by the thermal resistance linked to the internal air and the motor casing. Subsequently, a computational fluid dynamic (CFD) model based study has been carried out to assess the internal and the external flow and the convective heat transfer. Further, an optimization study of the hub motor has been carried out using the CFD model to improve heat transfer from the stator.

LUMPED PARAMETER THERMAL MODEL

Losses in PMBLDC Motor

The resistive losses are generally the major component of power losses in PMBLDC motors and they are calculated as

$$P_r = m I_{ph}^2 R_{ph} \quad (1)$$

where m is the number of phases, I_{ph} is the RMS phase current and R_{ph} is the phase resistance of winding. The resistance varies with temperature and can be calculated from a known resistance at another temperature as [8]

$$R_{T2} = R_{T1} \left(\frac{234.5 + T2}{234.5 + T1} \right) \quad (2)$$

where R_{T1} and R_{T2} are resistances at temperatures $T1$ and $T2$ °C respectively. The hub motor considered here has coils with a turn diameter of 0.5 mm. Therefore, the ac effects such as skin, proximity and circulating current are not considered.

The stator core is made of insulated silicon steel laminations. Therefore, the magnetic flux variation due to the rotation of magnets generates eddy and hysteresis losses in laminations. These losses are estimated individually for stator teeth and yoke as the field variations in them are different [9]. The loss per unit mass due to hysteresis (P_h), eddy current in stator yoke (P_{ey}) and eddy current in stator tooth (P_{et}) are calculated as [9]

$$P_h = K_h f^\alpha B_m^\beta \quad (3)$$

$$P_{ey} = K_e \frac{f^2 B_m^2}{\pi \beta_m} \quad (4)$$

$$P_{et} = K_e \frac{4 f^2 B_m^2}{\pi \alpha_{tt}} \left[2 - \frac{\pi - \beta_m}{\alpha_{tt}} \right] \quad (5)$$

where f is the frequency of flux variation, B_m is the peak flux density, α_{tt} and β_m are the tooth arc and the pole width respectively in electrical radians, K_h , K_e , α and β are the curve fit constants calculated from the loss data of the laminations.

Besides the resistive losses and core losses, a PMBLDC motor will have electromagnetic losses in rotor core and in magnets. These are neglected in this study as they are found to be of very low value. The frictional loss in bearing and air friction loss are calculated using the standard formulas available in literature [4].

Thermal Network of Radial Flux PMBLDC Hub Motor

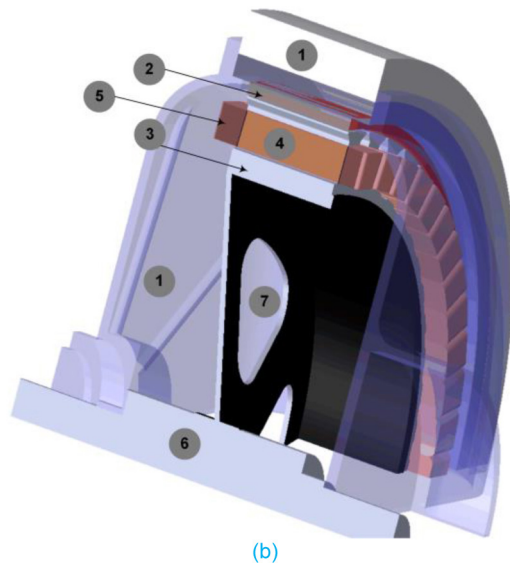
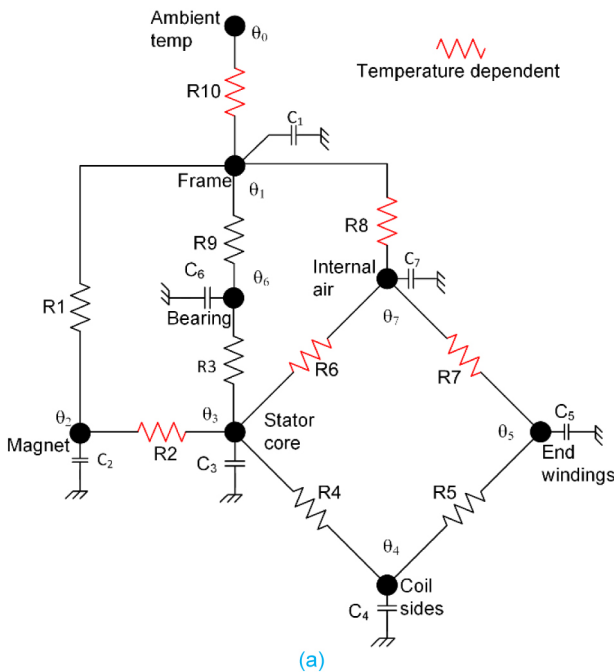


Figure 2. (a) Thermal network of PMBLDC hub motor and (b) position of nodes in motor.

The seven node thermal network of the PMBLDC hub motor is shown in [fig. 2a](#); where θ_0 is the ambient temperature, θ_n is the nodal temperature rise from the ambient temperature, R_n is the thermal resistances between two nodes and C_n is the thermal capacities of the components near node n . Additionally, the thermal resistances that are function of temperature is highlighted in red color. The position of each node inside the motor is shown in [fig. 2b](#). The nodes and the thermal resistance of the motor components are selected based on the specific characteristics such as temperature distribution, mechanical complexity and the material properties.

In general, the winding is the hottest part of the machine. A section of winding rests inside the slot (the node 4 in [fig. 2b](#)) and remaining, known as overhang (the node 5 in [fig. 2b](#)), is surrounded by the internal air. The heat from the winding is dissipated via slot walls to the stator core and also, via overhang to the internal air. This results in different thermal characteristics for the winding inside the slot and in the overhang; therefore modelled with two nodes. The temperature gradient inside the stator core for small machines are not very significant and hence, modelled with a single node. The motor uses a sintered Neodymium-Iron-Boron (NdFeB) magnets. The performance of the magnet is strongly influenced by the temperature and they could even get demagnetized if exposed to high temperature. Therefore, one node is assigned to the magnet. The rotor core, core housing and end plates are made of solid metals and they have good contact area. In order to simplify the modelling; only one node is assigned to these three components. Thermal and mass flow properties of air changes with the temperature. Therefore, accurate estimation of thermal resistance to the heat transfer via internal air requires its temperature and thus a node is assigned. The bearing temperature is a critical parameter that determines the life of electrical machines and hence a node is assigned to monitor the temperature of the bearing.

Thermal capacity of different components of motor are assigned to nearest nodes and represented in thermal network as capacitor connected to the nodes. Thermal capacity of an element consists of n components can be expressed as

$$C = \sum_{i=1}^n m_i c_i \quad (6)$$

where m_i and c are respectively the mass and the specific heat capacity of component i . Thermal resistance of components such as shaft, spider, magnet, rotor core and motor housing can be estimated using the expression for thermal resistance to conductive heat transfer as [\[8\]](#)

$$R = \frac{t}{kA} \quad (7)$$

where t is the effective length of heat flow path, k is the thermal conductivity of the material and A is the cross sectional area of the component. Thermal resistance of stator teeth, thermal resistance between slot and winding and thermal resistance between winding in the slot and in the overhang are calculated based on the approach described in [\[5\]](#). The thermal resistance of bearing is calculated using the equivalent airgap assumption as described in [\[10\]](#).

Air Gap Heat Transfer

The air gap heat transfer in radial flux motors represents heat transfer between concentric cylinders. There are a number of literature discussing the stationary outer cylinder and rotating inner cylinder [2,7,11]. However in hub motors, the outer cylinder is rotating while the inner cylinder is stationary. In [12], authors addressed a smooth outer cylinder rotation with a stationary smooth inner cylinder for an air gap to diameter ratio of 0.05 and concluded that Nusselt number (Nu) is nearly one. Therefore, the flow is laminar and heat transfer via air gap is conductive. However, in this study when using Nusselt number equal to one, resulted in higher temperature of components. Authors in [13], studied the effect of slot on flow between concentric cylinders. They concluded that the flow is influenced by a geometrical parameter, hydraulic diameter (D_h) and can be calculated as

$$D_h = Q_s h_o + 2 \frac{\pi(R_2^2 - R_1^2) + Q_s h_o w_o}{\pi(R_1 + R_2)} \quad (8)$$

where Q_s is the number of slot, h_o is depth of slot opening, w_o is the width of slot opening and R_1, R_2 are radius of inner and outer cylinder respectively. Further, Taylor number (Ta) which characterizes flow in annular region can be calculated using hydraulic diameter as [13]

$$Ta = \frac{\omega^2 R_1 \left(\frac{D_h}{2}\right)^3}{\nu^2} \quad (9)$$

where ω is the angular speed of rotation and ν is the kinematic viscosity of air. Subsequently, the Nusselt number for air gap region can be calculated as [13]

$$\begin{aligned} 6000 < Ta < 1.4 \times 10^6: Nu &= 0.132(Ta)^{0.3} \\ 4 \times 10^6 < Ta < 2 \times 10^7: Nu &= 0.029(Ta)^{0.4} \end{aligned} \quad (10)$$

The Nusselt number calculated for air gap region of hub motor at 350 rpm is found to be 5.75, much larger than the value discussed in [12]. The heat transfer coefficient (h) can be calculated using Nusselt number as

$$h = \frac{Nu k}{2(R_2 - R_1)} \quad (11)$$

Therefore, the equivalent thermal resistance of the air gap is

$$R_2 = \frac{1}{hA} \quad (12)$$

where A is the inner surface area of the outer cylinder.

End Space Heat Transfer

The end space heat transfer includes heat flow from overhang winding to internal air, from stator core and spider to internal air and internal air to end plate. In [14], authors summarised number of studies and concluded that heat transfer coefficient follows a relation

$$h = k_1 + k_2 V^{k_3} \quad (13)$$

where k_1, k_2, k_3 are curve fit coefficients and V is the peripheral velocity.

Heat Transfer from Wheel to Ambient

When wheel with spokes rotate, it could create turbulence. Therefore, this study models the wheel as a disc in a turbulent flow. In [15], author has discussed the heat transfer from a disc under various flow condition and for this study the lowest value of Nusselt number corresponding to a turbulent flow has been used. The average heat transfer coefficient of wheel is given by

$$h = \frac{Nu k_{air}}{r_{wl}} \quad (14)$$

where k_{air} is thermal conductivity of air at ambient temperature and r_{wl} is the wheel radius. Thermal resistance of equivalent disc to ambient can be calculated using the eq.12.

Solving the Thermal Network

The thermal network in fig.2a can be solved to find the steady state temperature rise with respect to ambient as well as the temperature rise versus time. For a thermal network of n nodes, with each of them connected to the other via thermal resistances, the steady state relationship between heat generated at node i and nodal temperature rises can be expressed as [5]

$$P_i = \frac{\theta_i}{R_{i,i}} + \sum_{j=1}^n \frac{\theta_i - \theta_j}{R_{i,j}} \quad (15)$$

where P_i is the heat generated in node i , θ_i is the temperature rise at node i , $R_{i,i}$ is the thermal resistance between the node i and the ambient and $R_{i,j}$ is the thermal resistance between the node i and j . The eq.15 can be expanded to include heat generated in all node as

$$\begin{bmatrix} P_1 \\ P_2 \\ \vdots \\ P_n \end{bmatrix} = \begin{bmatrix} \sum_{i=1}^n \frac{1}{R_{1,i}} & -\frac{1}{R_{1,2}} & \dots & -\frac{1}{R_{1,n}} \\ -\frac{1}{R_{2,1}} & \sum_{i=1}^n \frac{1}{R_{2,i}} & \dots & -\frac{1}{R_{2,n}} \\ \vdots & \vdots & \ddots & \vdots \\ -\frac{1}{R_{n,1}} & -\frac{1}{R_{n,2}} & \dots & \sum_{i=1}^n \frac{1}{R_{n,i}} \end{bmatrix} \begin{bmatrix} \theta_1 \\ \theta_2 \\ \vdots \\ \theta_n \end{bmatrix} \quad (16)$$