Thermal Design of a Magnet-free Axial-Flux Switch Reluctance Motor for Automotive Applications

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Abstract—With the requirement to develop magnet-free motors for automotive application, more design effort is needed for thermal design as the machine temperature limits the motor performance. Thermal design procedure of automotive motor has to fully exploit the modern approaches in electrical machine thermal analysis in terms of accuracy and solving time. This paper proposes a computational-cost-effective way to obtain the optimum thermal design of an axial-flux switched reluctance motor by means of lumped circuit methods with some inputs from Computational Fluid Dynamic simulations.

Keywords—Automotive; axial-flux; computational fluid dynamics; switched reluctance motor; thermal analysis; thermal design and thermal network.

I. INTRODUCTION

The heavy dependence on rare-earth materials used in electrical machines for the industrial applications has caused the magnet price to increase in the recent decade. Due to the limited supply and availability of rare-earth materials, the price of magnet could increase to a level that could make the electrical motors based on permanent magnet to be too expensive for hybrid and electric vehicles. Therefore, there is a demand to improve the performance and power density of magnet-free motors that suitable for automotive applications.

The switched reluctance motor (SRM) has been chosen over the other motor types in this paper due to the following reasons:

- Simple construction and ease to manufacture,
- Low cost for series production,
- Good torque-speed characteristics,
- Offer the possibility for dual coil and further improvements.

As machine operating temperature limits the electric loading, the power output of an electrical motor is strongly affected by its thermal performance. An electrical motor thermal performance is predominantly affected by two factors. One is amount of heat sources within the motor generated from losses. The other is how well the generated heat can be dissipated out from the motor. Furthermore, the machine temperature can affect the electrical efficiency because some electromagnetic losses are temperature dependent. For example, the electrical resistance of copper windings increases with temperature and hence too high operating temperature may lead to unnecessary power losses. Temperature rise will change material properties which can cause severe failure. From a structural point of view, elevated temperatures can induce mechanical stresses due to thermal expansion, and even thermal fatigue if the machine is operated at variable speed. Thus, motor designers must put more effort in thermal design to ensure machine reliability, life time and efficiency.

This paper provides a method to define the thermal design of magnet-free axial-flux switch reluctance motor (AFSRM) that is more suitable for automotive applications by using analytical lumped-parameter thermal network (LPTN) method with some inputs from Computational Fluid Dynamic (CFD) simulations.

II. AXIAL-FLUX SWITCHED RELUCTANT MOTOR

Fig. 1 shows the CAD geometry of the AFSRM by using axial-flux configuration. The motor comprises a rotor disc carrying 8 rotor poles and 12 stator coils assembly placed on the periphery of the rotor disc. Each C-core is fixed axially by the endcaps, radially by an inner support ring, and circumferentially by trapezoidal bulges on both sides. The winding of the AFSRM are wound about the C-cores as shown in Fig. 1 (b). Each coil consists of 22 turns of own made Litz wires. There are 60 strands of conductor diameter of 0.6 mm for each turn of the winding. The prototype of the AFSRM is estimated to have copper fill factor of 0.41.

The motor is totally enclosed and is mainly liquidcooled, i.e. 50/50 Ethylene glycol and water solution. The design flow rate is 16 litre/min and the inlet temperature is assumed to be 50 °C. The cooling channels are doublespiral shaped in the endcaps of drive end (DE) and nondrive end (NDE). They are connected in parallel through external connecting pipes. The circulating air induced by the rotor disc also aids active parts in dissipating heat.





Fig. 1: (a) CAD geometry of prototype of AFSRM under construction. (b) Rotor and stator arrangement by using axial-flux configuration.

A. Electromagnetic Performance

The modularity of this motor topology makes phases to be magnetically decoupled, thus motor performance can be extrapolated from the analysis of one pole pair and taking into account the number of active phases and poles. As the motor has three-phases, only one of them should be simultaneously active to generate torque in one direction. With 12 stator poles in the motor, there are 4 poles per phase. Therefore, motor parameters (torque, losses, etc.) are 4 times higher than in one pole pair.

The motor is designed with the help of an analytical electromagnetic design tool [1] which estimates overall performance parameters (average torque, base speed, electrical losses), which are after validated and completed both with electromagnetic FEA and dynamic simulations.

3D FEA simulations get accurate curves of switched reluctance motor's characteristic figures (flux vs. current, and torque vs. position). In these 3D FEA magneto static simulations a unique C-core together with three rotor poles is simulated as shown in Fig. 2. The coil in the C-core is fed with different current levels and a movement from 0° to 45° is performed to take a complete torque cycle. From these simulations torque (Fig. 3) and flux linkage (Fig. 4) curves are obtained so then dynamic simulations may be performed in Matlab Simulink software. The electromagnetic performance of the machine shown in Fig. 5 is defined from these dynamic simulations.



Fig. 2: Flux arrows in the 3D FEA simulation



Fig. 3: Torque vs Current and position curves (Each curve corresponds to a current level, starting at 0A and reaching 910A)

As joule heating loss is temperature dependent, the current considered for the thermal validation of the model in this paper is $I_{ph} = 135$ A rms. This was chosen as nominal current for the motor as a result of the iteration between the thermal analysis and the resistance of the copper windings. Based upon the class H wire insulation, the coil temperatures are aimed to be cooled below 180 °C.

With m = 3 phases and only one of them active, the duty cycle is $\delta = 1/m = 1/3$. Therefore the amplitude of the square waveform to be applied is $I_{ph,amp} = I_{ph}/\sqrt{\delta} = 235$ A.

Coil resistance at 180 °C is calculated to be 8.3 m Ω considering its length and copper section. Thus the copper loss in each coil is $P_{cu} = R_c \cdot I_{ph}^2 = 152.2$ W and total copper loss is 1826 W as given in Table 1.



Fig. 4: Flux linkage vs current and position curves (Each curve corresponds to a current level, starting at 0A and reaching 910A)



Fig. 5: Dynamic torque curve



Fig. 6: Evolution curve of the losses in the stator C-cores and rotor poles.

Magnetic losses are estimated by means of 3D FEA simulations. In this case transient simulations are performed so the loss calculation tools in Cedrat FLUX can be used. The machines rotates at 3500 rpm and is fed with 235A square waveform current, activating a single phase at each moment assuring that the maximum possible torque is produced. The losses in the stator C-cores and the rotor poles are calculated using LS model in FLUX. The time evolution of the losses is shown in Fig. 6. The average power losses are shown in Table 1.

TABLE 1: AVERAGE POWER LOSSES OF AFSRM

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Power loss	Prediction method	Value		
Coils	Analytical method	1826 W		
C-cores	FEA simulation	4312 W		
Rotor poles	FEA simulation	728 W		

The power losses in Table 1 are only taken into account in the paper. It is important to note that they are assumed to be sufficient for thermal design considerations of the AFSRM.

III. THERMAL ANALYSIS OF ELECTRIC VEHICLE MOTORS

The modern approaches in thermal analysis of electrical machines are analytical lumped-parameter thermal network (LPTN) method, numerical finite element analysis (FEA) and Computational Fluid Dynamic (CFD) methods [2]. All these approaches have both advantages and disadvantages. One of the main differences is the solving time. Once the thermal models are setup, the computational cost of analytical LPTN method is less demanding when compared to the numerical method. This makes LPTN method an attractive solution in machine design for sensitivity analysis. In addition to that, in order to calculate the complex thermal performance due to duty cycles, this paper complex demonstrates а computational-cost-effective way to obtain optimum thermal design of novel AFSRM by means of LPTN method. For complex fluid motion and convective heat transfer phenomena, CFD method is only used to provide inputs into the lumped parameter thermal network model.

A. Lumped Parameter Thermal Network Model

The equivalent thermal network has been developed using PORTUNUS, system simulation software package. The use of PORTUNUS in modelling an axial-flux permanent-magnet (AFPM) machine can be found in [3]. Due to the machine symmetry, only half of one-twelfth (30°) segment is modelled for the stator and half of onesixteenth (22.5°) segment is modelled for the rotor as shown in Fig. 9. It is important to note that the cuboidal element technique proposed in [4] has been employed in the thermal network for three-dimensional thermal modelling, especially useful to model internal heat generation and a material with anisotropic thermal conductivity. Fig. 7 shows the CAD geometry of C-core and coil. The cuboidal elements are superimposed on the same drawing to indicate how they are used to model the stator. It is important to note that only half of a C-core and coil are modelled. The stator lamination and winding are represented by four cuboidal elements respectively (green cubes are for C-core and yellow cubes are for coil) as shown in Fig. 7. The same cuboidal elements are also visible in the actual thermal network of the stator as shown in Fig. 8. The other heat transfer paths, i.e. conduction (R = L/kA) and convection (R = 1/hA) are represented by normal resistive elements as shown in Fig. 8.



Fig. 7: CAD geometry of C-core and coil with superimposed cuboidal elements.

The difficult aspects of thermal analysis of this motor include equivalent thermal conductivity across the coil, convective heat transfer coefficients and thermal contact resistance between motor components.

1) Equivalent coil thermal conductivity

The heat transfer in the coil is the most complicated conduction heat transfer in electrical motors because of a composite material which is a mixture of copper conductor, conductor insulation and impregnation materials. The appropriateness of using equivalent thermal conductivity to model the winding heat transfer has been studied by [5], [6]. The equivalent thermal conductivity across a coil can be approximated using the formulation proposed by Haskin and Shtrickman [7] that used to determine the effective thermal conductivity of a composite material constituted of two dominant isotropic materials. Hence, the equivalent thermal conductivity across a coil (k_e) can be calculated as:

$$k_e = k_{imp} \frac{(1+v_c)k_c + (1-v_c)k_{imp}}{(1-v_c)k_c + (1+v_c)k_{imp}}$$
(1)

where k_c and k_{imp} are the conductor and impregnation material thermal conductivities respectively, v_c is the volume ratio of conductor while the impregnation material occupies the volume ratio, v_p with $v_c + v_p = 1$. It is important to note that this homogenization relation assumes the conductor insulation material has similar thermal conductivity to that of the impregnation material.



Fig. 8: Equivalent thermal network for stator and housing assembly.



Fig. 9: Equivalent thermal network for rotor

Since the thermal conductivities of copper conductor and impregnation material (i.e. Varnish) are 401 W/m.K and 0.2 W/m.K respectively, based upon copper fill factor of 0.41, the equivalent thermal conductivity across and along the coil are 0.48 W/m.K and 165 W/m.K respectively. The use of cuboidal element and equivalent coil thermal conductivity allows the complex winding heat transfer to be modelled with reasonable accuracy and less computational time.

2) Convective heat transfer coefficients

The AFSRM is mainly cooled by water jacket in the endcaps and also the internal circulating air, the convective heat transfer is modelled as convective thermal resistance in the thermal network as:

$$R_c = \frac{1}{hA} \tag{2}$$

where h and A are the convective heat transfer coefficient and surface area respectively. The h value of water jacket can be estimated preliminarily by using Dittus-Boelter equation as:

$$Nu = \frac{hD_h}{k_{fluid}} = 0.023Re^{0.8}Pr^{0.4}$$
(3)

where Nu, Re and Pr are the dimensionless Nusselt, Reynolds and Prandtl numbers, D_h is the hydraulic diameter of the cooling channel and k_{fluid} is the fluid thermal conductivity.

Due to the novelty of this motor topology, the heat transfer coefficients at various convective surfaces inside the motor are obtained using CFD method as shown in Section III.B.

3) Thermal contact resistance

As the heat generated from losses is primarily dissipated to the water jacket, thus it is important to minimize the interface gap between stator lamination (C-core) and endcap for lower thermal contact resistance. For conventional radial flux machines, the contact resistance between stator lamination and housing frame is affected by shrink fit pressure, materials used, surface roughness, thermal interface material, assembly procedures and even the thermal expansion effects [8], [9]. The combination of these factors makes the exact calculation of contact resistance to be not possible. The usual procedure is to perform sensitivity analysis by varying the interface gap between 0.01 mm and 0.08 mm. This will indicate the upper and lower temperature range of the machine components.

However, the stator assembly concept of the AFSRM is very different from the radial flux motors. Due to the concentricity issues between the motor shaft and bearing, the interface gap between the C-core and endcap is strongly affected by the assembly procedure. The potential gap size cannot be identified until testing carried out on the actual prototype. If the contact resistance is found to be large, the interface gap will be filled with thermal interface material with high thermal conductivity. Despite of that, in the design state, the similar procedure that used for radial flux machines described above is also used for this AFSRM.

В.	Computationa	l Fluid Dy	namic Models	
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Fig. 10: (a) Internal view at axial cross section and (b) internal view of radial cross section of the CFD model of AFSRM.

Two CFD models have been developed for the thermal design of the AFSRM. The first CFD model is used to investigate the performance of housing cooling jacket, e.g. ensuring its pressure drop meet the pump characteristics of powertrain cooling loop, flow distribution between DE and NDE is similar and estimating temperature rise of housing and endcaps due to the electrical losses. The second model is used to investigate the convective cooling inside the motor due to the circulating air, e.g. computing convective heat transfer coefficient at every surface. By splitting the complete machine into two separated sub models, this can save the computational recourses considerably. The coupled simulation is then performed by making use of the CFD results in the LPTN model in Section III.A.

The second CFD model is the only focus in this paper. As shown in Fig. 10, only one-sixth (60°) segment was modelled in three dimensions for simulating flow and heat transfer. Periodic planes *p1-p1* and *p2-p2* are used for this segment. The fluid domain is split into stationary and rotating regions. Alongside the fluid domain, the solid domain representing the rotor and stator was also included to calculate power dissipation with conduction. All material properties specified in the CFD model are the same as those of the prototype (i.e. C-core and rotor pole -M270-35A, coil - coil, rotor plate and pole retainer -Aluminium Alloy). Conformal mesh interfaces between solid and fluid domains were created to maintain the continuity of heat flux and temperature. It is important to note that in the CFD model the coil was simplified by a bulk region with anisotropic thermal conductivity. The effective thermal conductivity across and along the coil are specified as that of in LPTN model. As the material of liner was not physically modelled, the thermal resistance per unit area is computed as:

$$R_{liner}' = \frac{l_{liner}}{k_{liner}} \tag{4}$$

 l_{liner} is the thickness of liner and k_{liner} is the thermal conductivity of liner. It was applied as the thermal contact _ resistance of the interfaces between coil and C-core.

Polyhedral and prism layer meshers were employed. With the intention of resolving the viscous sublayer, a high boundary layer mesh resolution was created with the wall cell y+ is close to 1.

1) Turbulent flow Modelling

In the present study a rotating reference frame was used to simulate the relative motion of the rotor and stator for time-averaged steady-state solutions. As the nominal speed of the motor is 3500rpm, a turbulence model is = necessary. Due to rotating flow, Reynolds Stress Transport (RST) turbulence model was used to provide closure of the Reynolds-Averaged Navier-Stokes (RANS). The effect of gravity was ignored. The air was assumed to be an ideal gas. The viscosity and thermal conductivity of air vary with temperature based upon the Sutherlands' law.

2) Bondary Conditions

The governing equations of the CFD model are subjected to the following boundary conditions:

a) As the heat dissipation with conduction in outer housing and endcaps has already been estimated in the first CFD model. Consequently, the housing and endcaps' temperatures are imposed in the second CFD model.

b) The power losses described in Section II.A are injected into the CFD model as internal heat generation and they are assumed to be evenly distributed within the relevant regions.

c) Both rotor and stator surfaces are smooth and there is no-slip. The local speed of the rotor surface is equal to $r_{rotor}\omega_m$. In the rotating air region, additional momentum source, S_m is added in the momentum equation which is the sum of the Coriolis force and centrifugal force due to the effects of rotation as:

$$S_m = -[2\rho(\omega_m \times \bar{V}) + \rho\omega_m \times (\omega_m \times r)]$$
(5)

where *r* is the position vector of a general point in the flow relative to the rotating frame and ω_m is angular velocity.

3) Convective Heat Transfer Coefficient

Based on the CFD solution, the mean heat transfer coefficients, \bar{h} at various surfaces can be obtained as:

$$\bar{h} = \frac{q}{T_{wall} - T_{ref}} \tag{6}$$

where q is heat flux, T_{wall} and T_{ref} are wall temperature and reference temperature respectively. It is important to note that T_{ref} should refer to the local air temperature normal to the convective surface based on heat transfer and fluid dynamic practice. However, the average temperature of air domain is used as T_{ref} which is sufficient for the LPTN model.

TABLE 2: AVERAGE HEAT TRANSFER CORRELATIONS

Convective surface	Α	В
S1 - Coil (top)	0.0140	0.678
S2 - Coil (side)	0.0300	0.658
S3 - Coil (endcap)	0.3660	0.454
S4 - Coil (slot)	0.0381	0.693
S5 - C-core (leading edge)	0.2930	0.471
S6 - C-core (trailing edge)	0.7007	0.440
S7 - Endcap (inner)	0.0377	0.611
S8 - Housing (inner)	0.0236	0.658
S9 - Rotor ends (radial)	0.6090	0.497
S10 - Rotor (circumference)	0.0015	0.936

The rotor outer radius, r_o is 0.16 m and the axial air gap size, s is 0.5 mm. The heat transfer results are correlated with the rotational Reynolds number Re_{θ} for disc type machine according to a power law [10], [11] and [12] as:

$$Re_{\theta} = \frac{\omega_m r_o^2}{v} \tag{7}$$

$$\overline{Nu} = \frac{\overline{h}r_o}{k} = ARe_{\theta}{}^B \tag{8}$$

The air properties, i.e. kinematic viscosity, ν and thermal conductivity, k are also evaluated at the average air domain temperature. The constants A and B of the average Nusselt number \overline{Nu} are given in Table 2:



Fig. 11: Average Nusselt number for AFSRM internal surfaces

Based on the heat transfer practice, the average \overline{Nu} is plotted against the Re_{θ} using logarithmic scale (with base of 10) as illustrated in Fig. 11. The correlations are only for valid for air, range of Re_{θ} from 2.13×10^5 to 9.08×10^5 (i.e. 2000 rpm to 8300rpm). They were then applied into the analytical thermal model. They can also be applied to the machine that of geometric and dynamic similarities.

IV. RESULTS AND COMPARISON

As conjugate heat transfer analysis has been performed using CFD simulations as described in Section III.B. The CFD results can provide verification to the LPTN model. The comparison between the CFD and LPTN temperature predictions after steady state are given in Table 3 for nominal speed.

TABLE 3: PREDICTED TEMPERATURES FROM CFD AND LPTN THERMAL MODELS (IN $^{\circ}$ C)

Temperature	CFD Model	LPTN Model
Coil (Maximum)	217.7	223.3
Coil (Average)	200.1	189.2
C-core (Average)	115.6	102.5
Tooth tip	124.1	133.8
Housing inner	61.4	62.7
Interface between C-core and endcap	65.5	64.7
Rotor pole (Maximum)	145.2	154.0
Inside air	116.9	120.2

The temperature rise of coil and C-core is illustrated in Fig. 12. As shown in Table 3, the predicted temperatures from CFD and LPTN models have good agreement. Hence, the validated analytical thermal model is suitable for thermal design of the AFSRM. This provides computational-cost-effective approach to obtain the optimum thermal design and perform transient analysis.

It is important to note that the AFSRM is in the early stage of the mechanical design process (where some details are missing) and the thermal model development, thus the temperature prediction might not be truly representative of the prototype that is under construction. Nevertheless, the coil temperature is estimated to be higher than expectation of 180 °C as described in section II. A. This however does not affect validity to the thermal model and validation presented here.



Fig. 12: Temperature contour of coil and C-core axial cross section at the mid-plane of C-core (nominal speed).

A. Sensitivity analysis using LPTN model

By using the LPTN model, sensitivity analysis is performed to investigate the impact of interface gap between endcap and stator C-core on the coil temperature rises. This is carried out by varying the interface gap from 0.01 mm to 0.08 mm.



Fig. 13: The variation of coil temperature with interface gap between endcap and C-core.

As demonstrated in the Fig. 13, the sensitivity analysis shows that the interface gap has significant impact on the AFSRM thermal performance. The air temperature inside the AFSRM also increases by having larger interface gap.

B. Transient analysis using LPTN model

A simple transient has been performed using the LPTN model to evaluate the time taken to reach steady state condition. As shown in Fig. 14, the temperature rise of machine components is plotted against time. The AFSRM reaches steady state condition after about an hour. The computational time required is only 45 seconds approximately using Lenovo ThinkPad T440p.



Fig. 14: The variation of machine temperatures with time from ambient temperature of 50 $^{\circ}\mathrm{C}.$

The aim of this section is to demonstrate that the demand of computational cost is very low. Therefore, the LPTN model can be easily used for a more complex duty cycle analysis of automotive application.

V. CONCLUSION

The paper was aimed to develop a computational-costeffective approach for analyzing and predicting the thermal performance of electrical machine of novel topology. The approach is extremely useful for complex duty cycle analysis such as automotive application.

The LPTN model is verified with CFD simulation. The comparison between LPTN and CFD results demonstrate that the analytical thermal model can perform the thermal analysis with reasonable accuracy and minimum computational cost. Sensitivity analysis can be easily performed to obtain optimum thermal design.

Based on dimensional analysis, a set of the convective heat transfer correlations has been developed for the AFSRM due to the internal circulating air. They can also be applied to the machine that of geometric and dynamic similarities.

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