Electromagnetic performance with and without considering the impact of rotation on convective cooling

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Abstract: Accurate thermal modelling of electrical machines is important for electromagnetic performance determination of a design. This study presents the impact of rotation on convective cooling of a through-ventilated motor – a design equivalent to the Tesla S 60 traction motor design. Machine temperatures after steady state with and without considering rotation effects are compared, and the influence on the machine continuous performance is investigated.

1 Introduction
The electromagnetic performance of an electrical machine is ultimately limited by how the machine temperatures are cooled within their temperature limits. Therefore, there is a requirement of accurate thermal modelling of electrical machines; otherwise, the machine design will not be correctly sized. Without accurate thermal model, a machine designer cannot calculate the machine power/torque density and estimate the costs.

The thermal performance of electrical machines relies mainly on the active convective cooling. Commonly, a coolant is forced into the machines to remove the undesired heat. The coolant is subjected to the effect of rotation due to Coriolis force and centrifugal force. For a given flow rate, these rotational forces give higher convective heat transfer when compared to the stationary condition. Consequently, convective formulae that take into account the rotation effect must be considered during thermal modelling. This appears in lumped parameter thermal network method as the convective heat transfer coefficients are computed from correlations applied.

The objective of the paper is to present the impact of rotation on the thermal performance of a through-ventilated machine. Also, the continuous torque comparison of the machine with and without considering the rotation will be presented in the paper.

2 Through-ventilated machine
The machine used for this analysis is an inductor motor – an equivalent to Tesla S 60 traction motor design (see Table 1 [1]). The original cooling configuration of Tesla motor has a liquid cooled stator and a rotor [1], but in the paper, it is adapted with cooling ducts in rotor and cooling ducts in between housing and the stator core, as shown in Fig. 1. There are 12 circular ducts in the rotor of diameter 12 mm, and another 12 rectangular ducts (41 mm × 7 mm) are formed between the stator and housing. The electromagnetic design is based on the strained winding model in [2]. Due to cooling ducts in the rotor, the electromagnetic performance is re-calculated, as depicted in Fig. 2.

The machine is mainly cooled by air flow through the machine of inlet temperature of 40°C. The total air flow rate is assumed to be 0.09 m\(^3\)/s, which can be supplied by an off-the-shelf fan. After entering into the machine, the air splits between stator ducts, rotor ducts and the annular airgap. The air distribution between these cooling channels depends on their pressure drop (flow resistances), which is calculated using a flow network method. Based on the cooling system, the continuous performance is computed by setting the temperature limits of stator winding and rotor bar to be 180 and 220°C, respectively. Fig. 3 shows the continuous performance.

3 Thermal calculation
The software tool used to perform the analysis is Motor-CAD. It comprises of electromagnetic, thermal and efficiency map modelling modules that allow co-simulation between the electromagnetic model and thermal model for more accurate electrical machine performance. The Motor-CAD thermal module is based on a lumped parameter thermal network method. Hence, the machine is represented by an equivalent thermal circuit with thermal resistances calculated from input geometry, material, losses, cooling methods etc. As mentioned earlier, the machine is mainly cooled by the axial through flow, though the combined effects of axial flow and rotation need to be considered. The aim of the study is to compare the case with and without considering rotation. For convective cooling, the surfaces that are subjected to the effect of rotation are rotor ducts, airgap and end space.

Table 1: Industrial motor details

| Parameter     | Unit | Value |
|---------------|------|-------|
| stator OD     | mm   | 254   |
| stator ID     | mm   | 157   |
| axial active length | mm | 150   |
| airgap        | mm   | 0.5   |
| stator slots  | —    | 60    |
| poles         | —    | 4     |
| rotor bars    | —    | 74    |
| electric steel| —    | M250-35A |
| rotor cage    | —    | copper |

Fig. 1 Geometry design of the through-ventilated inductor motor
(a) Radial view, (b) Axial view
3.1 Rotor ducts

The flow through circular rotor ducts that are parallel to the axis of rotation is influenced by rotational forces: Coriolis force and centrifugal force in a rotation reference frame [3]. The Coriolis force creates a swirl at the duct entrance, as shown in Fig. 4. The Coriolis-induced secondary flow is predominant in the duct entrance region and become less inside the duct when the flow becomes fully developed downstream. Due to heating, the fluid adjacent to the duct wall is less dense than that in the duct centre. The density variation in the centrifugal field results in centrifugal buoyancy, which further enhances the temperature exchange between the hot and cold fluid particles. Hence, inside the duct, the fluid motion is relatively dominated by centrifugal force in a rotating reference frame. Therefore, a more accurate heat transfer correlation is needed to model that. In the thermal model, the correlation proposed by Morris [4], which is suitable for air, is used for the analysis, and the Nusselt number is given as follows:

\[ \text{Nu} = 0.012 \text{Re}^{0.78} \text{Fr}^{0.18} \]  

where Nu is the Nusselt number and Re is the axial flow Reynolds number. Fr is the rotational Reynolds number, which is defined as

\[ J = \frac{\omega m d^2}{8 \nu} \]  

where \( d \) is the duct diameter, \( \omega m \) is the angular velocity and \( \nu \) is the kinematic viscosity. It is also important to note that rotation also increases the resistance of flow passing through the rotor duct and thus higher pressure drop. Detailed experimental investigation can be found in [3, 5].

3.2 Rotor–stator gap

In rotor–stator gap, the flow is highly sheared and unstable due to rotation. This shear flow between concentric cylinders is commonly known as Taylor vortex flow. The formation of Taylor vortices is measured by Taylor number (Ta), and the Taylor vortices are formed in the gap if Ta is greater than a critical value, i.e. \( \approx 41 \) for a narrow annular gap. It is important to note that the Taylor vortices are important for convective heat transfer. The convective heat transfer in the rotor–stator gap can be estimated as follows [6]: When \( Ta < 41 \), the flow is laminar

\[ \text{Nu} = 2 \]  

When \( 41 \leq Ta \leq 100 \), the Taylor vortices are formed in the shear flow that enhances the heat transfer

\[ \text{Nu} = 0.212 \text{Ta}^{0.63} \text{Pr}^{0.27} \]  

When \( Ta > 100 \), the rotor–stator gap is further enhanced as the shear flow becomes fully turbulent with Taylor vortices

\[ \text{Nu} = 0.386 \text{Ta}^{0.5} \text{Pr}^{0.27} \]  

The Taylor number (Ta) is defined as follows:

\[ Ta = \frac{\omega m r_m^3 s^{1.5}}{\nu} \]  

where \( s \) is the airgap size, \( r_m \) is the mean rotor–stator radius, i.e. \( r_m = (r_i + r_o)/2 \), where \( r_i \) and \( r_o \) are inner and outer radii of the airgap, respectively.

For the through-ventilated machine, the superimposed axial flow could provide additional convective heat transfer in the rotor–stator gap. However, at the same time, the superimposed axial flow can suppress and reduce the formation of Taylor vortices when compared to the case without the axial flow. Consequently, the airgap heat transfer with axial flow is very complicated because the existence of the superimposed axial flow can has positive or negative impact. To accurately predict the heat transfer due to the effects of mixed flow, a set of convective correlations is employed that depends on the type of flow in the rotor–stator gap. Basically,
the flow in a rotor–stator gap can be categorised into four different modes: laminar flow, turbulent flow, laminar flow with Taylor vortices and turbulent flow with Taylor vortices, as shown in Fig. 5.

For laminar flow, the convective heat transfer can be approximated using the correlation for the flow between parallel plates as follows [8]:

$$\text{Nu} = 7.54 + \frac{0.03(D_h/L)\text{Pr}}{1 + 0.016[(D_h/L)\text{Pr}]^{0.37}}$$ (7)

where $D_h$ is the hydraulic diameter of the airgap and $L$ is the axial length of airgap.

For turbulent flow, Gnielinski's formula can be used, which is given as follows:

$$\text{Nu} = \frac{(f/8)(\text{Re} - 1000)\text{Pr}}{1 + 12.7(f/8)^{0.3}(\text{Pr}^{0.8} - 1)}$$ (8)

where $f$ is the Darcy friction factor.

For laminar flow with Taylor vortices, the empirical correlation that proposed by Tachibana and Fukui [9] can be applied. The total Nusselt number is basically the sum of Nusselt number due to axial flow ($\text{Nu}_{\text{axial}}$) and Nusselt number due to rotational flow ($\text{Nu}_{\text{rotational}}$):

$$\text{Nu} = \text{Nu}_{\text{axial}} + \text{Nu}_{\text{rotational}}$$ (9)

$$\text{Nu}_{\text{axial}} = 0.015 \left(1 + 4.6 \left( \frac{V}{L/\tau_0} \right)^{1/4} \text{Re}^{0.3} \text{Pr}^{1/3} \right)$$ (10)

$$\text{Nu}_{\text{rotational}} = 0.092 (\text{Ta}^{1/2} \text{Pr})^{1/3}$$ (11)

where $s$ is the airgap size and $r_i$ and $r_o$ are inner and outer radii of the airgap, respectively.

For turbulent flow with Taylor vortices, the empirical correlation that proposed by Gazley [10] can be used. The Nusselt number is influenced by the effective velocity in the rotor–stator gap, which is defined as follows:

$$V_e = \sqrt{U^2 + \left( \frac{V_T}{2} \right)^2}$$ (12)

$$\text{Nu} = 0.03 \left( \frac{D_h V_e}{	ext{Pr}} \right)^{0.8}$$ (13)

where $U$ is the axial flow velocity, $V_T$ is the outer rotor velocity and $\nu$ is the kinematic viscosity of the fluid. Alternatively, the correlations proposed by Kuzay and Scott [11] and Childs and Turner [12] can be used for this flow regime.

### 3.3 End region

Besides the axial through flow driven by the fan, the circulating air flow dragged by the rotor ends also provides extra cooling to the convective surfaces in the end spaces, such as end windings, end caps, housing etc. The convective heat transfer coefficient ($h$) due to the circulating local air velocity ($V$) can be approximated using the following empirical formula:

$$h = k_1 \times (1 + k_2 \times V^{k_3})$$ (14)

where $k_1$, $k_2$ and $k_3$ are empirical coefficients. The first term $k_1$ is used to account for natural convection, while the second term $k_1 k_2 V^{k_3}$ is used to account for forced convection due to rotation. Different values of empirical coefficients have been proposed by different authors [13–17], but all show that the higher the air velocity, the higher the convective heat transfer coefficient, as illustrated in Fig. 6.

It is important to note that (14) is more suitable for distributed winding where the space between conductors in end winding is filled by impregnation resin and the end winding can be treated as a bulk volume. The circulating air has contact to its outer surface only. This is applicable to the studied machine. For form-wound or hairpin winding machines, an alternative formula is required.

The challenge of using (14) is what the representative values of local air velocities are for different surfaces as in fact the local air velocities reduce from the rotational speed of the rotor surfaces to lower values depending on their distance from the rotor. In [6], computational fluid dynamics simulations have been performed to analyse the air flow and convective heat transfer in the end space region in order to calibrate the cooling model based on (14). The analysis shows that the convective heats transfer has strong dependency on the local circulating air velocity. This is in line with their relation in (14).

### 4 Analysis results

As shown in Fig. 2, peak torque of 430 N.m can be achieved up to the base speed of 6500 rpm, considering that the entire motor elements are at a constant temperature of 100°C and peak current of 900 Arms [2]. However, for continuous performance, which is subjected to the stator winding temperature limit of 180°C (i.e. insulation class H), the maximum torque achieved at the base speed is 141 N.m, which corresponds to shaft output power of 96 kW @ 6500 rpm, as shown in Fig. 3. This operating point is an important point for a machine design and therefore is chosen in the present study to investigate the impact of rotation on the thermal performance of the through-ventilated machine (see Fig. 7 and 8). The machine losses used for the thermal analysis are tabulated in Table 2, which are calculated using Motor-CAD software. DC stator copper loss is calculated from the DC-phase resistance at a constant temperature of 100°C, but during thermal calculation, it is scaled according to the calculated temperature. Stray loss is
calculated based on the IEC 60034 standard. The other losses are computed by using a finite element (FE) method.

Based on the losses in Table 2, the machine temperatures after the steady state are calculated by an equivalent thermal network. The effects of rotation on cooling are modelled using the convective correlations described in Section 3. Then, the thermal performances with and without considering effect of rotation are compared. For the case without considering the effect of rotation, the convective heat transfer coefficients of rotor ducts and end space surfaces are only based on the axial flow, not including the influence of swirl flow and circulating air flow induced by the rotor. For the airgap, the convective heat transfer coefficient due to the shear flow (rotational flow) is only considered but not modelling the complexity of superimposed axial flow on the shear flow as described in Section 3.

As shown in Table 3, the maximum stator winding temperature is about 72°C higher if the effects of rotation on convective cooling are not taken into account. The temperature distributions for the cases with and without rotation are illustrated in Fig. 9 and 10, respectively. By comparing the convective heat transfer coefficient used in thermal calculations (see Table 4), the heat transfer coefficients that include rotation effects give much higher values, more than two times of those without including rotation effects for rotor ducts and airgap. The end winding surfaces vary depending on how close the surfaces from the rotor. With rotation effects, the heat transfer coefficients are three/four times higher. The inner end cap and housing also exhibit increment in heat transfer coefficient due to rotation but less significant for the inner housing surface as it is hidden behind the end winding.

Then, the machine continuous performance is re-calculated without considering the effect of rotation on convective cooling in order to analyse the impact on the electromagnetic performance. As mentioned earlier, the equivalent Tesla S 60 traction motor design is adapted from original liquid cooling to air cooling. For reference, with the original liquid cooled stator and rotor configuration [2], the continuous torque is 163 N m at 6500 rpm (base speed), while the continuous power is 110 kW, as shown in Fig. 11. For air cooling, the continuous torque and power is 141 N m and 96 kW, respectively, at 6500 rpm, equivalent to 13% reduction in performance. However, if the effects of ration are not considered, the continuous torque and power is 107 N m and 73 kW, respectively, at 6500 rpm, i.e. 34% less output shaft torque when compared to that of the original liquid cooling configuration. Fig. 11 shows the benefits of rotation on convective cooling and thus higher continuous performance.

**Table 2** Loss components of thermal analysis, 141 N m @ 6500 rpm

| Loss components      | Loss amount, W |
|----------------------|----------------|
| DC stator copper     | 1488           |
| AC stator copper     | 87             |
| rotor cage           | 331            |
| stator iron          | 510            |
| rotor iron           | 9              |
| stray load           | 1499           |

**Table 3** Comparison of machine temperatures after steady state

| Machine temperature      | With rotation effect, °C | With no rotation effect, °C |
|--------------------------|--------------------------|-----------------------------|
| stator winding (max)     | 180                      | 252                         |
| stator winding (ave)     | 162                      | 227                         |
| rotor bar                | 123                      | 133                         |

**Fig. 7** Air velocity streamlines showing the variation of local velocity in the end space

**Fig. 8** Walls temperature at rotational speed equal to 10,000 rpm [18]

**Fig. 9** Machine temperatures at radial cross section for the case with rotation effect being considered

**Fig. 10** Machine temperatures at radial cross section for the case without rotation effect being considered
5 Conclusion

As the electromagnetic performance of an electrical machine is strongly limited by its thermal performance, accurate thermal modelling is important to determine the continuous torque and output power of the design. In the present paper, a through-ventilated machine – an equivalent design to Tesla S 60 traction motor is used to investigate the effects of rotation on convective heat transfer. Suitable formulae for the convective boundaries that are subjected to the effects of rotation are presented. The analysis shows that the values of convective heat transfer that include rotation effects are much higher than the stationary case. Therefore, if the effects of rotation are not included in the thermal calculation, one might likely underestimate the continuous performance of the machine design.

6 References

[1] Kim, K.-C.: ‘Driving characteristic analysis of traction motors for electric vehicle by using FEM’. Proc. ANSYS Users Conf., Seoul, Korea, October 2014
[2] Goss, J., Popescu, M., Staton, D., et al.: ‘Electrical vehicles – practical solutions for power traction drive systems’. Proc. IEEE Workshop Electrical Machines Design, Control and Diagnosis, Nottingham, 2017, pp. 80–88. doi:10.1109/WEMDC.D.2017.7947728
[3] Chong, Y.C.: ‘Thermal analysis and air flow modelling of electrical machines’. Ph.D. Dissertation, University of Edinburgh, Edinburgh, UK, 2015
[4] David, M.W.: ‘Heat transfer and fluid flow in rotating coolant channels’ (Research Studies Press, Letchworth, 1981)
[5] Chong, Y.C., Staton, D.A., Mueller, M.A., et al.: ‘An experimental study of rotational pressure loss in rotor ducts’. Proc. 14th UK Heat Transfer Conf., Edinburgh, UK, 2015
[6] Staton, D., Boglietti, A., Cavagnino, A.: ‘Solving the more difficult aspects of electric motor thermal analysis in small and medium size industrial induction motors’, IEEE Trans. Energy Convers., 2005, 20, (3), pp. 620–628
[7] Kaye, J., Elgar, E.: ‘Modes of adiabatic and diabatic fluid flow in an annulus with an inner rotating cylinder’, Trans. ASME, 1958, 80, pp. 753–765
[8] Mills, A.F.: ‘Heat transfer’ (Prentice-Hall, NJ, USA, 1999, 2nd edn.)
[9] Tachibana, F., Fukui, S.: ‘Convective heat transfer of the rotational and axial flow between two concentric cylinders’, Bull. Jpn. Soc. Mech. Eng., 1964, 7, (26), pp. 385–391
[10] Gazley, C.: ‘Heat transfer characteristics of the rotational and axial flow between concentric cylinders’, Trans. ASME, 1958, 80, pp. 79–90
[11] Kuzay, T.M., Scott, C.J.: ‘Turbulent heat transfer studies in annulus with inner cylinder rotation. Transactions of the ASME American society of mechanical engineers’, Int. J. Heat Mass Transf., 1977, 99, pp. 12–19
[12] Childs, P., Turner, A.B.: ‘Heat transfer on the surface of a cylinder rotating in an annulus at high axial and rotational Reynolds numbers’. Proc. 10th Int. Heat Transfer Conf., Brighton, 1994, pp. 13–18
[13] Mellor, P.H., Roberts, D., Turner, D.R.: ‘Lumped parameter thermal model for electrical machines of TEFC design’, IEE Proc. B – Electr. Power Appl., 1991, 138, (5), pp. 205–218
[14] D’Geraldo Vistoli, A.J.: ‘Thermal networks of induction motors for steady state and transient operation analysis’. Proc. ICEM, Paris, 1994
[15] Schubert, E.: ‘Heat transfer coefficients at end winding and bearing covers of enclosed asynchronous machines’. Elektrotechn., 1969, 62, (6), pp. 219–232
[16] Stokum, G.: ‘Use of the results of the four-heat run method of induction motors for determining thermal resistance’, Elektrotechnika, 1969, 62, (6), pp. 219–232
[17] Hamdi, E.S.: ‘Design of small electrical machines’ (Wiley, Chichester, 1994)
[18] Basso, G.L., Goss, J.Y., Chong, C., et al: ‘Improved thermal model for predicting end-windings heat transfer’. Proc. IEEE Energy Conversion Congress and Exposition, Cincinnati, OH, 2017, pp. 4650–4657. doi:10.1109/ECCE.2017.8096794