Experimental Investigation of a Novel Direct Rotor Cooling Method for an Interior Permanent Magnet Synchronous Machine

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***Abstract*— A measurement setup is designed and constructed to experimentally investigate the thermohydraulic performance of a novel direct rotor cooling method for an interior permanent magnet synchronous machine, where oil is pumped through weight saving channels of the rotor iron close to the magnets. The setup consist of a stationary rotor assembly where heat is dissipated in an aluminum sleeve surrounding the rotor outer surface. Measurements are performed for different oil inlet temperatures, flow rates and heat losses. It is shown that tangential temperature variations can be neglected and axial variations are as expected. The conductive resistance within the rotor assembly is similar for each measurement point as the differences are within the measurement uncertainty. The convective resistance decreases with an increasing flow rate, inlet temperature and heat dissipation. From correlations in scientific literature, it is expected that the convective resistance will further reduce with increasing rotational speed of the rotor. A comparison of these results to a standard hollow shaft cooling method shows a great potential of the novel direct rotor cooling method to further increase the power density of electric machines.**

Keywords—Electric machine cooling, Direct rotor cooling, Experimental investigation

I. INTRODUCTION

Electric vehicles have a great potential to help in the overall reduction in emissions and pollution in the transport industry. One of the main components in the drivetrain of these electric vehicles is the electric machine. These need to fulfill certain requirements to act as a viable alternative to fossil fuel drive-trains. They have to be light weight while at the same time providing sufficient output power to be able to compete. As a result, electric machines get more power dense, requiring sufficient cooling in order to avoid damage to the stator coils and permanent magnets of e.g. an internal permanent magnet synchronous machines (IPMSM) [1].

In case temperatures of the stator coils exceed the temperature to which the electrical insulation is resistant, these can get damaged and short-circuits can occur. Permanent magnets can have their remnant magnetism decreased or outright removed because of strong magnetic fields. As the temperature increases, the magnitude of the magnetic field needed to (irreversibly) demagnetize the magnet, becomes smaller, resulting in the motor to be damaged or even break down [1]. Cooling of electric drive-trains is therefore paramount to make the electric drivetrain work for at least the lifetime of the vehicle. Cooling strategies for the stator coils have been widely investigated and different solutions exist [2]. For the cooling of the rotor and permanent magnets, recently several possibilities have been investigated [3] [4], such as hollow shaft liquid cooling, rotating heat pipe cooling and rotor spray or jet cooling. The hollow shaft liquid cooling method is the most commonly used method because it is relatively easy to manufacture [5]. The latter cooling method however suffers from relatively high interface resistances from heat source (rotor core and magnets) to the location of cooling within the shaft [3].

Within this study, a variation on the hollow shaft liquid cooling method is proposed where weight saving spaces within the rotor iron close to the permanent magnets are used as cooling channels. The electric machine under investigation has the dimensions and specifications as shown in TABLE I. Since there is direct contact between the coolant and corrosion sensitive components, a lubrication oil is used as coolant which is already available in an electric vehicle.

TABLE I INTERIOR PERMANENT MAGNET SYNCHRONOUS MACHINE GEOMETRY AND SPECIFICATIONS

Stator/ rotor poles	48/8
Active length	140 mm
Shaft diameter	42 mm
Rotor outer diameter	119 mm
Stator inner diameter	120.5 mm
Stator outer diameter	180.5 mm
Housing diameter	216.5 mm
Max power	124 kW
Max speed	15000 rpm

II. EXPERIMENTAL SETUP

The experimental setup consists of an assembly of the stationary shaft, rotor iron, magnets and balancing plates, since these are the only components of interest to study the proposed cooling method. Within the rotor iron, 8 weight saving channels are provided which are used as the cooling channels. In the rotating machine operation, hysteresis and eddy-current losses are induced in the magnets and rotor iron due to an alternating electromagnetic field, mostly close to the airgap [1]. Such a field is not present within the stationary experimental setup, so an aluminum sleeve with cartridge heaters is used as an alternative, surrounding the rotor over the whole active length as shown in Fig. 2.

The sleeve contains 8 drilled holes in which 16 cartridge heaters are inserted resulting in a maximum heat dissipation of 1 kW, supplied by an EA-PS 9072-170 power supply. The heat spreading in the aluminum sleeve was simulated in 2D with the finite element package FEMM [6] and with a thickness of 50mm, the temperature non-uniformity at the inner diameter of the sleeve is lower than 0.2%. Oil is provided from an oil conditioning circuit shown in Fig. 1 at the desired inlet temperatures T_{in} and flow rates $\dot{V}$. The oil is pumped from a reservoir through a plate heat exchanger (HEX) to condition the oil with a water-glycol chiller. Next a three-way-valve settles the flow rate to the test section by recirculating a part of the flow back to the reservoir. The oil temperature at the inlet of the test section is accurately set at the desired temperature by a PID controller on the electrical heater. After the test section the flow is directed to one of the two Krohne Optimass coriolis flow meters to measure the flow rate, one for the low flow range (6000FS08) and one for the high flow range (6000FS15) and flows back to the reservoir.

An overview of the test section and oil flow path is shown in Fig. 2. The oil inlet and outlet are both located in the shaft at the front of the motor. The oil first flows through an axial centered bore hole to the back of the rotor, from where it goes to the back balancing plate collector. The oil then flows through the rotor channels after which it is collected in the front balancing plate. From there the oil flows back through radially drilled channels to the radially spaced axial channels in the shaft. Inlet and outlet temperatures are accurately measured with 1.5mm T-type mineral insulated thermocouples downstream from a twisted tape insert flow mixer providing a uniform temperature profile (see Fig. 1).

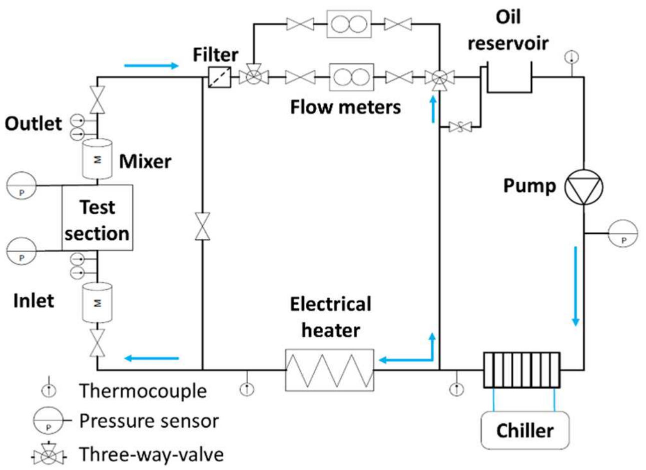

Fig. 1. Oil conditioning circuit and test section

Several temperature sensors are mounted onto different locations of the rotor assembly to determine the thermal performance of the cooling method. Mineral insulated 0.5mm K-type thermocouples are used to measure the temperature of the rotor iron between the rotor channels (T_{RI}) and the oil temperature within these channels (T_C). The rotor outer surface temperatures (T_{RO}) are measured with ultra-fine wire PFA K-type thermocouples. The exact locations are shown on Fig. 2. The inlet and outlet pressure are measured directly before the shaft inlet with a Gems 2200AGB2501A2UA001 and after the outlet with a Gems 2200AGB1001A2UA001 pressure sensor to determine the hydraulic performance. The range and accuracy of all used sensors are shown in TABLE II.

TABLE II RANGE AND UNCERTAINTY OF THE MEASUREMENT EQUIPMENT.

Sensor	Range	Accuracy
K-type TC	$20..150°C$	$\pm 0.1°C$
T-type TC	$20..120°C$	$\pm 0.08°C$
Inlet pressure	$0..25bar$	± 0.0625bar
Outlet pressure	$0..10bar$	± 0.025bar
Current	$0..170A$	$\pm 0.17A$
Voltage	0..72V	$\pm 0.14\ V$
Flow meters	$0..5\ l/min$ $5..30\ l/min$	$\pm 7.5\ ml/min$ and $\pm 0.1\%$ $\pm 35\ ml/min$ and $\pm 0.1\%$

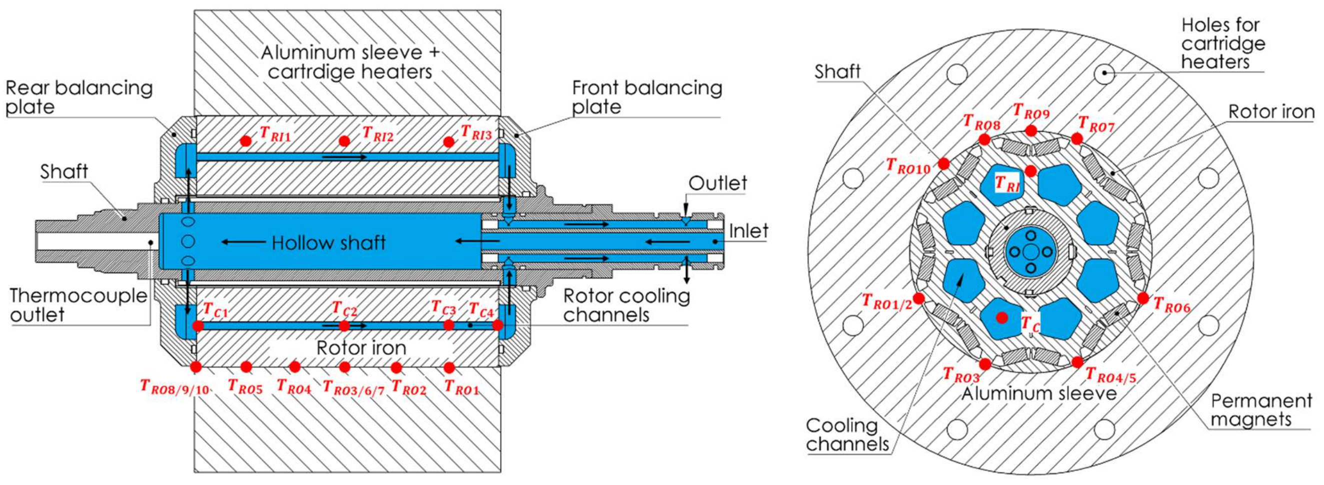

Fig. 2. Axial (left) and perpendicular (right) section view of the test section with indication of the sensor locations.

III. Measurement results

Measurements are performed for different flow rates ($\dot{V} = 2, 5\ and\ 10\ l/min$) at different inlet temperatures ($T_{in} = 40, 60\ and\ 80°C$) and for different heat losses in the aluminum sleeve ($Q_{el} = 200, 500\ and\ 800W$). In a first step, the heat balance is checked, which is the difference between the electric power Q_{el} dissipated in the sleeve and the thermal power Q_{th} measured in the oil. Taking into account the uncertainty on the thermal power, the heat balance is always closed within the measurement error. At the highest power, the relative difference is smaller than 5%.

Next, the axial and tangential temperature gradients within the rotor are verified, which will be done for measurement point $T_{in} = 60°C$ and $Q_{el} = 500$. The tangential temperature variation is measured at the same axial location by thermocouples T_{RO3}, T_{RO6} and T_{RO7} and the measurement results are shown in Fig. 3 (left). It is concluded that the tangential variation can be neglected, since the relative temperature variations $(T_{RO,max} - T_{RO,min})/(T_{RO} - T_{in})$ are smaller than 1.4%. The axial temperature variation is measured with sensors $T_{RO1}, T_{RO2}, T_{RO3}, T_{RO4}$ and T_{RO5} and the results are shown in Fig. 3 (right). The trends in the variation are as expected, since the temperatures near to the balancing plates are expected to be lower than the center temperatures due to the additional heat transfer area to the oil in the balancing plate spaces compared to the rotor channel area.

To compare the effect of the inlet temperature, flow rate and heat dissipation on the thermal performance, a conductive and convective thermal resistance are defined. The conductive resistance is calculated by the temperature difference between the center outer sensor measuring the maximum temperature at the outer surface (T_{RO3}) and the center sensor within the rotor iron (T_{RI2}) which is located close to the heat transfer wall: $R_{cond} = (T_{RO3} - T_{RI2})/Q_{el}$. The convective resistance is defined between T_{RI2} and the temperature sensor within the oil channel (T_{C2}): $R_{conv} = (T_{RI2} - T_{C2})/Q_{el}$. The calculated resistances for each measurement point are shown in Fig. 4.

The conductive thermal resistance is similar for each measurement point and the differences are close to the measurement uncertainty, which is to be expected since it is independent of the flow conditions and heat dissipation. It is shown that the convective part of the thermal resistance is dependent on all measurement variables. With an increasing flow rate, a decrease in thermal resistance is seen due to an increasing Reynolds number and heat transfer coefficient and decreasing outlet and bulk temperature of the oil. With an increasing inlet temperature, the convective resistance decreases because of a decrease in oil viscosity, resulting in a higher Reynolds number and heat transfer coefficient between oil and rotor channel. Finally, with an increasing heat dissipation, the convective heat transfer coefficient also decreases. This is caused by a higher wall temperature, decreasing the viscosity in the boundary layer which improves the heat transfer. The decrease in resistance at higher heat dissipation rates is more pronounced for lower flow rates than for higher flow rates. This might indicate that natural convection has a significant contribution to the total heat transfer between rotor and oil compared to the forced convection contribution.

To show the potential of the proposed cooling method, a comparison will be made with conductive resistance from the rotor surface to the inside of the shaft based on a 2D FE simulation in FEMM [6]. The simulation uses the exact same geometry as the experimental setup while the rotor core is assumed to be solid (without weights saving spaces). Thermal properties are used from Staton et al. [7], including the contact resistance from the rotor core to the shaft. The contact resistances between magnets and core are neglected because of the presence of the epoxy potting. The total calculated conductive resistance for the hollow shaft cooling method is $R_{cond,hslc} = 0.146\ K/W$, of which half is caused by the contact resistance between core and shaft. Even when eliminating the contact resistances, the conductive resistance for the hollow shaft cooling method is of the same order of magnitude as the total resistance of the proposed method. Taking into account that a convective resistance should be added to this conductive part $R_{cond,hslc}$ due to the fluid convection in the hollow shaft, the great potential of proposed cooling method for electric machines is shown.

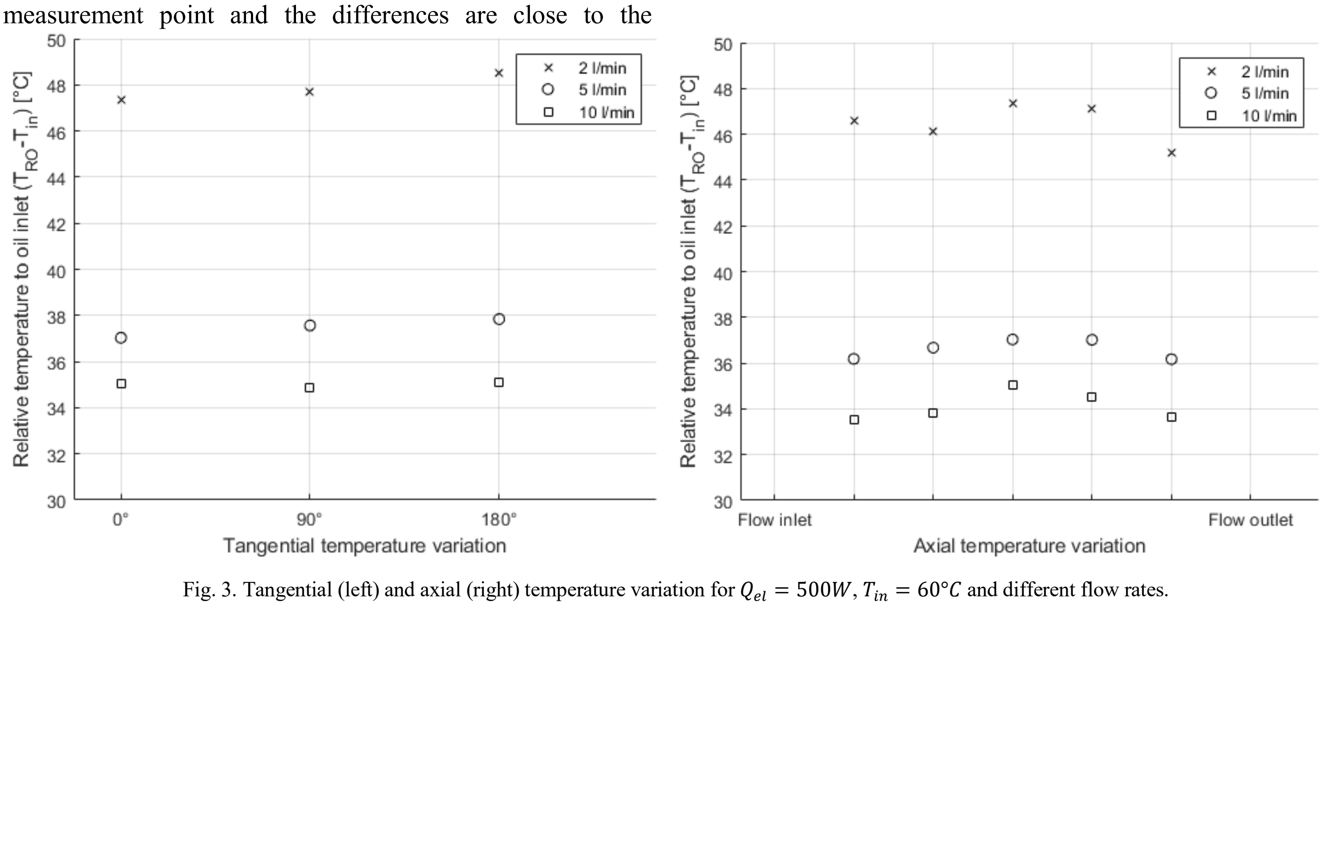

Fig. 3. Tangential (left) and axial (right) temperature variation for $Q_{el} = 500W$, $T_{in} = 60°C$ and different flow rates.

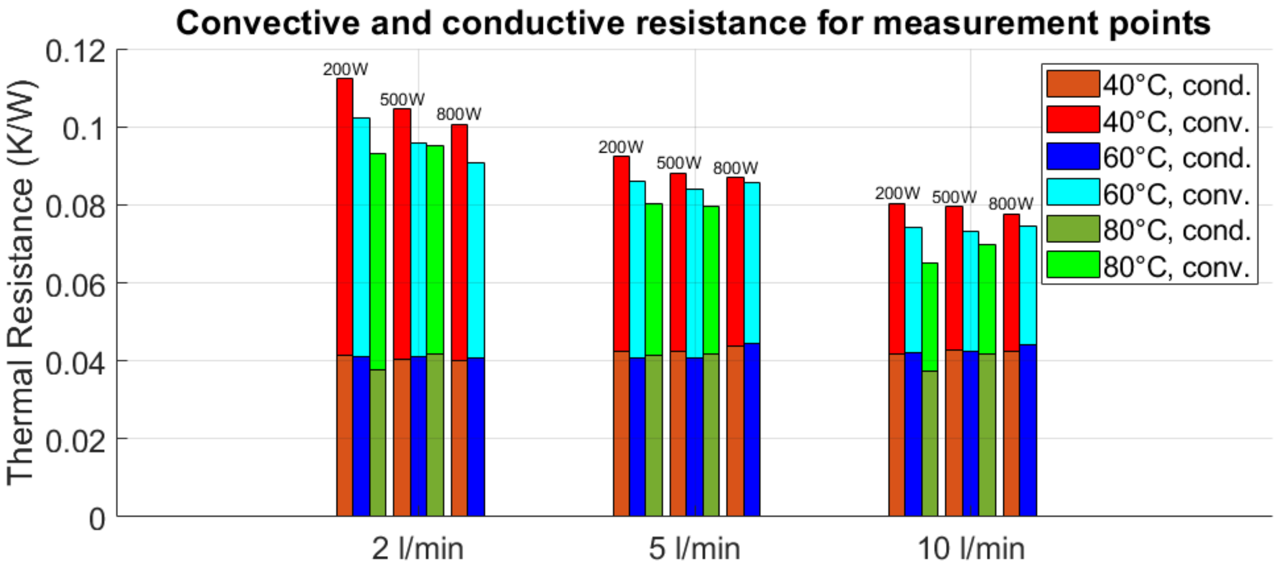

Fig. 4. Conductive and convective thermal resistance for the different measurement points.

Since experiments were done on a stationary rotor setup, the effect of the rotating channels could not be investigated experimentally. However, based on the results of Mori et al. [8] it is expected that the convective heat transfer and pressure drop will increase with rotational speed due the secondary flow created by the rotating motion and the temperature difference between the boundary layer and the core flow. The boundary layer has a higher temperature and therefore has a lower density which through buoyancy forces induces additional flows to form between the boundary layer and remaining fluid. The overall effect is that the heat exchange is increased compared to the stationary flow. Overall it can be concluded that with the proposed method, heat losses up to 800W and more can be dissipated within the rotor iron and magnets without exceeding the maximum temperature of the magnets.

IV. Conclusions

A measurement setup is designed and constructed to experimentally investigate the thermohydraulic performance of a novel direct rotor cooling method for an interior permanent magnet synchronous machine. With this method, the weight saving channels of the rotor iron which are located close to the magnets are used as cooling channels. The setup consist of a stationary rotor assembly where heat is dissipated in an aluminum sleeve surrounding the rotor outer surface at different rates ($200, 500 \; and \; 800W$). Oil is pumped axially through the cooling channels at different inlet temperatures ($40, 60 \; and \; 80°C$) and flow rates ($2, 5 \; and \; 10 \; l/min$). Based on the measurement results, it is shown that the heat balance is closed within the measurement uncertainty and tangential temperature variations within the rotor can be neglected. The axial variations are as expected, where lower temperatures are measured near the balancing plates due to the increased surface area for cooling. The conductive resistance within the rotor assembly is similar for each measurement point as the differences are within the measurement uncertainty. The convective resistance decreases with an increasing flow rate, inlet temperature and heat dissipation. From correlations in scientific literature, it is expected that the convective resistance will further reduce with increasing rotational speed of the rotor. When comparing these results to a standard hollow shaft cooling method, the measurements show a great potential of the novel direct rotor cooling method to further increase the power density of electric machines.

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