Direct Thermal Management of Windings enabled by Additive Manufacturing

N. Simpson, G. Yiannakou, H. Felton, J. Robinson, A. Arjunan and P. H. Mellor

Abstract—The electrification and hybridization of ground- and air-transport, in pursuit of Carbon Net Zero targets, is driving demand for high power-density electrical machines. The power-density and reliability of electrical machines is ultimately limited by their ability to dissipate internally generated losses within the temperature constraints of the electrical insulation system. As the electrical windings are typically the dominant source of loss, their enhanced design is in the critical path to improvements in power-density. Application of metal additive manufacturing has the potential to disrupt conventional winding design by removing restrictions on conductor profiles, topologies and embedded thermal management. In this paper, a modular end-winding heat exchanger concept is presented, which enables effective direct cooling without occupying valuable stator slot cross-section. In addition, this arrangement eliminates the need for a good stator-winding thermal interface, thereby allowing mechanical or other less permanent winding retention methods to be used, facilitating non-destructive disassembly and repair. A prototype winding is fabricated and experimentally tested to demonstrate the feasibility of the concept, yielding promising results.

I. INTRODUCTION

The electrification and hybridization of ground- and air-transport, in pursuit of Carbon Net Zero targets, is driving demand for high power-density electrical machines. To meet requirements across traction, propulsion, generation and actuation, the UK Advanced Propulsion Centre (APC) and Aerospace Technology Institute (ATI) envisage a need for 9-25 kW/kg by 2035 in stark contrast to the 2-5 kW/kg commercially available at present, [1], [2]. Typically, at the high operating fundamental frequencies of modern high specific output electrical machines, AC winding loss is the dominant loss component, followed by iron loss, [3]. The trend toward higher fundamental electrical frequencies, intended to re-ap system level power-density benefits, further exacerbates AC loss components leading to higher overall loss densities which need to be managed through enhanced thermal design.

In general, power-density improvements are brought about if the following winding design goals can be achieved:

1) minimise winding loss (AC and DC)
2) minimise non-active winding volume through use of concentrated coils or alternative machine topologies
3) raise electrical insulation temperature rating
4) enhance heat extraction (to dissipate loss)

Incremental improvement across 1-4 can be achieved, for example, by adopting concentrated Litz wire windings, modern high-temperature insulation coatings with circa 240 °C temperature rating, [4]–[6], and liquid cooling of housings complemented by established end-winding cooling techniques, [7]. However, the step change in power-density needed calls for exploration of potentially disruptive technologies which can simultaneously address design goals 1-4 whilst ideally providing additional integration possibilities and value-add features, such as integrated terminals and routes to non-destructive disassembly and repair.

One such approach is to exploit the unparalleled geometric freedom offered by metal Additive Manufacturing (AM) in which feedstock is selectively bonded in a succession of 2D layers to incrementally form a 3D part. In the case of electrical winding design, metal AM of copper and aluminium windings (Cu, CuCrZr, AlSiMg) has been demonstrated with designs incorporating shaped conductor profiles, [8], [9], targeted loss mitigation, [10], variable conductivity, [11] and through-conductor or flooded stator cooling, [9], [12]–[15]. In each case, there is scope for integrated terminals and voids for temperature and other on-line monitoring sensors. Alongside through-conductor and flooded stator cooling, AM has been used to demonstrate in-slot cooling via conduction to metal, ceramic or polymer heat exchangers, or heat guides, [16], [17]. Further, a technique to cool end-windings by conduction to a liquid cooled annulus was proposed in [14], [15]. These cooling schemes typically occupy valuable stator slot cross-sectional area, [9], [12]–[14], [16]–[19], which could otherwise be used for conductor material to reduce DC loss, or they rely on thermal conduction across material interfaces which tend to be poor, [16], [17], [20].

In this paper, an integral end-winding heat exchanger concept is presented which alleviates the need to occupy stator slot volume or traverse poor material interfaces. An analytical model capable of predicting the turn-by-turn temperature distribution for non-uniform heat generation caused by AC loss is developed and a simple method for rapidly identifying the optimal heat exchanger location is shown. A prototype winding is manufactured and subject to experimental validation, yielding promising results.

II. END-WINDING HEAT EXCHANGER CONCEPT

The use of AM enables a fluid heat exchanger to be directly incorporated into a winding to form a single contiguous component, as illustrated in Fig. 1. This avoids the need for separate entities in contact and the typically poor thermal interfaces between them. However, the turns of such a winding must remain electrically isolated. As such, the contact cross-section between the heat exchanger and the winding is limited to that of the end-winding cross-section of a single turn, as shown in Fig. 1a. Nevertheless, the direct through-conductor
A. Heat Exchanger Infill

A key component of the heat exchanger is an infill structure that enhances heat transfer to the coolant fluid by maximising the contact area without incurring excessive pressure drop, since this would increase the required fluid pumping power at a system level. Conventional Computer Aided Design (CAD) systems use boundary representation (B-rep) in which 3D objects are made up of exact vertices, edges and faces where each entity is typically modelled explicitly by a designer. This approach is highly complementary to subtractive manufacturing methods. However, AM allows extremely complex geometries, such as structured infill and textured surfaces to be realised which are difficult or impossible to model using conventional CAD systems. Instead, functional representation (F-rep), [22], can be used to easily generate complex geometry. Here, the zero-level isosurface of a continuous real valued function is used alongside surface meshing techniques such as Marching Cubes, [23], or Dual Contouring, [24], to form 3D objects. The ability to generate geometry from mathematical functions underpins the idea of field-driven design, highly applicable to AM, in which physics-based simulations are used to influence geometry in design and optimisation processes. For example, this approach is demonstrated in the design of shaped profile windings to minimise AC loss in [8].
Fig. 2 illustrates how the heat exchanger conformal infill is created by firstly generating an F-rep gyroid infill covering the minimum bounding box of the heat exchanger cavity, followed by a union operation. The F-rep conformal infill is then converted to a mesh and inserted into the B-rep winding assembly, ready for AM. Table I shows the heat exchanger cavity and infill properties, where the gyroid infill exhibits almost double the fluid-solid contact area to promote heat transfer, at the expense of increased pressure drop (reduced cavity volume) and consequently fluid pumping power. The heat exchanger cavity volume, inlet and outlet diameter, infill structure, density and orientation should be optimised to maximise heat transfer. However, optimisation of the heat exchanger is beyond the scope of the present paper.

TABLE I
HEAT EXCHANGER PROPERTIES

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
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<tbody>
<tr>
<td>$k_x$ x axis</td>
<td>1.26</td>
<td>N/A</td>
</tr>
<tr>
<td>$k_y$ y axis</td>
<td>1.26</td>
<td>N/A</td>
</tr>
<tr>
<td>$k_z$ z axis</td>
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<td>N/A</td>
</tr>
<tr>
<td>$W_t$</td>
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<td>mm</td>
</tr>
<tr>
<td>Cavity Volume (hollow)</td>
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<td>mm$^3$</td>
</tr>
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<tr>
<td>Surface Area (infilled)</td>
<td>3058</td>
<td>mm$^2$</td>
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</table>

III. THERMAL MODEL

A three-quarter section illustration of a concentrated $N_t = 18$ turn winding originally designed for a 48 V, (7.5 kW continuous, 15 kW peak) Integrated Starter Generator (ISG), [26], is shown in Fig. 3. The conductors are assumed to be of uniform cross-section rather than shaped profile, Fig. 1a, [8], to simplify analysis. Assuming that the heat exchanger connected to the end winding of $N_{h_x}$ is the only cooling mechanism and that DC is applied to the winding, leading to uniform heat generation along each turn, then there exists two primary heat transfer paths. The first is through-conductor, where heat follows along the helical path of the winding to the heat exchanger. The second is through-insulation, where heat flows between turns via the inter-turn electrical insulation layers. The two heat paths can be represented as an equivalent circuit, as illustrated in Fig. 3b, where the voltage source, $T_{h_x}$, represents the temperature of the heat exchanger (neglecting the associated thermal resistance to the coolant fluid), the current sources, $Q_n$, represent the heat generation of each turn, $n$, and the resistances, $R_n$, represent the through-conductor thermal resistance given by (3) or the through-insulation thermal resistance given by (4). This assumes that the thermal resistance of one of the heat transfer paths is much larger than the other and can be neglected in the analysis. If this is not the case, then both heat transfer paths should be modelled in parallel and the equivalent circuit modified accordingly. In (3) and (4) $k$, $r$, $l$, $w$, $h$, $A$ and $t$ refer to the mean thermal path length, width, height, cross-sectional area and thermal conductivity respectively.

The ratio of through-conductor to through-insulation resistance is given by (5) showing that the magnitude of through-insulation resistance is only 3.5% that of the through-conductor resistance for the present geometry. Table II. This indicates that the dominant heat transfer path is through the inter-turn insulation where the turn incorporating the heat exchanger, $N_{h_x}$, behaves like a heat sink. Hence, the through-conductor heat transfer path may be neglected when considering the through-insulation heat transfer path in this case. The nodal temperatures, $T_n$, of the equivalent circuit, Fig. 3b, can be solved analytically using (6). The expression

$$k = \frac{2\pi}{U_c}$$  \hspace{1cm} (2)

Fig. 3. Illustration of a concentrated winding mounted on a stator segment.

$R_c = l_t \frac{k_c A_c}{k_c w_c h_c}$  \hspace{1cm} (3)

$R_{ins} = l_{ins} \frac{k_{ins} A_{ins}}{k_{ins} w_{ins} l_t}$  \hspace{1cm} (4)

$R_{ins} \over R_c = h_t \frac{l_{ins}}{l_t} \frac{k_c}{k_{ins}}$  \hspace{1cm} (5)

TABLE II
WINDING PROPERTIES

<table>
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<th>Parameter</th>
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<th>Unit</th>
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<td>$w_c$</td>
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<tr>
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<td>mm</td>
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<td>W/m.K</td>
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<tr>
<td>$k_{ins}$</td>
<td>0.1</td>
<td>W/m.K</td>
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</table>
is validated against steady state thermal 3D Finite Element Analysis (FEA), Fig. 4, where a fixed temperature boundary condition represents the heat exchanger, \( T_{hx} = 20 \, ^\circ C \), and \( I_{DC} = 35 \, A_{rms} \) giving a uniform heat generation of 3.9 \( W \) over the winding. (8). A fine mesh size of 0.5 \( mm \) is used to minimise mesh dependency. Fig. 4a shows the temperature of each turn for different heat exchanger locations, \( N_{hx} \), predicted using (6) and 3D FEA where the inter-turn insulation is assigned a thermal conductivity \( k_{ins} = 0 \, W/m.K \) to emulate through-conductor heat transfer only. In spite of the high thermal conductivity of Cu \( (k_c = 394 \, W/m.K) \), the resulting absolute temperature values are high due to the high thermal resistance of each turn \((25 \, ^\circ C/W)\) owing to a relatively long mean path length and low cross-sectional area, (3). As can be seen by the equivalent circuit structure, the heat generated in turn 1 traverses turns 1 to \( N_{hx} \) while the heat generated in turn \( N_{hx} - 1 \) only traverses a single turn which leads to compounded temperatures at the extreme turns, 1 and \( N_t \). As expected, for uniform heat generation turn \( N_{hx} = 9 \), representing the middle of the coil, with balanced loss either side, yields equal peak temperatures. The shape of the analytical and 3D FEA temperature profiles show close agreement with the differences in magnitude attributed to the lumped parameter nature of the analytical model. Splitting the winding into a greater number of segments would improve temperature prediction at the expense of complexity. Fig. 4b shows the resulting temperature profile for the same thermal problem with \( k_{ins} = 0.1 \, W/m.K \) to include through-insulation heat transfer. This value accounts for imperfect impregnation and bonding by reducing the nominal insulation thermal conductivity of DOLPHON C(a)-1114 from 0.8 \( W/m.K \), [27], to 0.1 \( W/m.K \). This gives an inter-turn thermal resistance of 0.9 \( ^\circ C/W \). As a result, the peak temperatures reduce substantially, demonstrating the dominance of the through-insulation heat transfer path and reinforcing the feasibility of such a cooling concept.

\[
T_n = \begin{cases} 
T_{hx} + \sum_{i=n}^{N_{hx}} \left( R_{N_{hx}+n-1} \sum_{j=1}^{N_{hx}+n-i} Q_j \right) & n \leq N_{hx}, \\
T_{hx} + \sum_{i=n}^{N_{hx}} \left( R_i \sum_{j=n}^{N_{hx}} Q_j \right) & n > N_{hx} 
\end{cases}
(6)
\]

IV. AC LOSS MODEL

Under DC conditions the heat generation of each turn is uniform, hence, the ideal location for the heat exchanger, assuming no additional cooling from the stator slot, is approximately \( Nt/2 \), Fig. 4. However, under AC conditions the heat generation becomes non-uniform, due in part, to the impact of skin effect and proximity effect. The loss tends to increases toward the air-gap since the magnitude of the impinging slot leakage flux density is sourced from the ampere-turns enclosed below each layer, as illustrated in Fig. 5. The frequency dependent heat generation of each turn is modelled using a technique adapted from [28], neglecting rotor effects, (7). The DC loss component in the active length, \( P_{DC_{actn}} \), of each turn, \( n \), is multiplied by a frequency dependent AC loss factor, \( K_{ACn} \), while the end-winding is assumed to exhibit DC loss only, \( P_{DC_{endn}} \), since the impinging flux density is comparatively low in this region. The active length and endwinding DC resistances are calculated using the appropriate geometry and electrical conductivity \( \sigma \), (8). The AC loss factor \( K_{ACn} \) is given by (9) with coefficients \( C_1 \), (10), and \( C_{II} \), (11), dependent on the characteristic dimension \( \eta \), (12), which is in turn dependent on the lateral slot fill factor, \( e_l \), (13), where \( w_s \) is the slot width, and \( \delta \) is the skin depth at frequency \( f \), (14).

\[
Q_n = 2(K_{ACn}P_{DC_{actn}} + P_{DC_{endn}}) = 2(K_{ACn}I^2R_{DC_{actn}} + I^2R_{DC_{endn}}) 
(7)
\]

\[
R_{DC} = \frac{l}{\sigma A} = \frac{l}{\sigma w_sh_c} 
(8)
\]

\[
K_{ACn}(\eta) = C_1 + C_{II}(2n-1)^2 
(9)
\]
The total AC loss factor of the winding is predicted using the analytical model, (7), and a single slot time-harmonic 3D FEA model up to 1 kHz showing similar trends. The deviation (maximum of 15%) arises since the end-winding length of the present winding geometry is close to that of the active length, meaning that any error in the assumption of DC loss only in the end-winding is proportionally significant. In addition, the leakage flux impinging the end-winding region is not accounted for and acts to raise the AC loss above that attributed to DC loss alone. Nevertheless, the analytical model is computationally efficient and allows the heat generation distribution to be predicted with acceptable accuracy considering that the optimal heat exchanger location is necessarily an integer choice. The spread of the turn level AC loss factor, $K_{AC}$, with increasing frequency is illustrated in Fig. 7.

The impact of non-uniform heat generation is demonstrated by Fig. 8, where at DC, Fig. 8a, the optimum heat exchanger position is $N_{t}/2 = N_{hx} = 9$ yielding balanced temperatures at the first and last turn of 117 °C. At 1 kHz the spread of heat generation is such that the optimal heat exchanger position shifts to $N_{hx} = 12$ yielding temperatures of 179 °C close to the stator slot back-iron and 160 °C close to the air-gap, a difference of 19 °C. In a practical application this temperature difference would likely be reduced due to the supplementary cooling effect of the stator slot on the conductors close to the back-iron. In this example, $I_{DC}$ is increased from 35 $A_{RMS}$ to 85 $A_{RMS}$ to give peak steady state operating temperatures close to the limit of Class H insulation of 180 °C at 1 kHz, with a total loss dissipation of 70 W. The ideal heat exchanger location, $N_{hx}$, can be represented as a simple optimisation problem using the analytical thermal model, (6), and loss model, (7), where the difference between the temperature of the first and last turn is minimised, (15), as shown in Fig. 8.

$$C_T(\eta) = \frac{\eta \sinh(\eta) + \sin(\eta)}{2 \cosh(\eta) - \cos(\eta)}$$

$$C_{II}(\eta) = \frac{\eta \sinh(\eta) - \sin(\eta)}{2 \cosh(\eta) + \cos(\eta)}$$

$$\eta = \sqrt{\frac{h_c}{\delta}}$$

$$\epsilon_t = \frac{w_c}{w_s}$$

$$\delta = \sqrt{\frac{1}{\sigma \mu_0 \pi f}}$$
with associated high electrical and thermal conductivity is challenging due to the reflective and thermally conductive nature of the Cu feedstock, making it difficult to achieve high melt pool temperatures with comparatively low laser power, [29], among other challenges, [30]. Mass densities and International Annealed Copper Standard (IACS) conductivity close to 100% have been achieved with high laser powers (up to 1 kW), [31]. However, in this instance the prototype was shown to be porous via pressure testing of the heat exchanger and through DC electrical testing, [8], giving $R_{DC} = 5.2 \, \text{m}\Omega$ at ambient temperature, above the expected 3.2 m$\Omega$, implying 62 % IACS. The impact of porosity on the internal structure of AM parts is illustrated via Computed Tomography (CT) scanning of CuCrZr material samples, Fig. 9, where high effective energy density (left) leads to higher density material, in contrast to the porous material (right).

![CT cross-section image of CuCrZr material sample produced by LPBF with left) high effective laser power and right) low effective laser power.](image)

It is important to note that porosity can be reduced to very low levels with improvements in processing parameters yielding high thermal and electrical conductivity and water-tight manifolds. In the present study, the reduced electrical conductivity is assumed to result in a commensurate reduction in thermal conductivity according to the Wiedemann–Franz law ($k_c = 244 \, \text{W/m.K}$) and the heat exchanger is sealed using an epoxy overmoulding, as shown in Fig. 11. The temperature rating of the epoxy material limits experimental thermal testing to 80 °C peak.

![Experimental arrangement used to assess the thermal performance of the winding under DC conditions and varying coolant flow rate.](image)

The winding Device Under Test (DUT) is subject to DC thermal characterisation using the experimental arrangement shown in Fig. 10. DUT current is supplied by a high-current DC supply (Farnell AP60/150) and measured using a precision Hall effect transducer (IT 400-S ULTRASTAB) in conjunction with a power analyser (Fluke Norma 4000), to ascertain power dissipation. A chiller unit with an uncontrolled impeller pump (ATC K3) supplies temperature controlled coolant (Coolflow ECO20, [32]) to the cold side of a heat exchanger (Xylem BP410) set to 10 °C. A low temperature is used to ensure that the thermal tests do not exceed the 80 °C limitation imposed by the epoxy overmoulding of the present heat exchanger prototype. The hot side is connected to the DUT via a positive displacement peristaltic pump (Watson Marlow 505U) and flow rate meter (LMX24SS) enabling precise, validated, flow rate control, in this instance between 0 and 420 ml/min. The fluid inlet and outlet temperature is measured using Class A Resistance Temperature Detectors (RTDs) in contact with the fluid (IFM Electronic PT100). Type K thermocouples, subject to a calibration procedure, are placed on turns 1, 9 and 18 as shown in Fig. 11. Two Data Acquisition (DAQ) units (Measurement Computing USB-TEMP and USB-1208FS) collect the relevant temperature and flow rate data as a function of time until thermal steady state is reached. The DUT is first characterised with the turns splayed, as illustrated in Fig. 11, to assess the impact of through-conductor cooling alone.

![Illustration of the splayed and insulated coil configurations used to test the through-conductor and through-insulation heat flow regimes.](image)

However, the splayed conductors act as an effective heatsink structure giving rise to a significant convective cooling component which would not be present in practice since the conductors would normally be insulated and in close contact. To assess the through-insulation thermal path, representative of a practical use-case, the DUT is then insulated with epoxy resin (DOLPHON C(a)-1114) using a polymer based conformal mould, fabricated using AM, and a vacuum chamber to minimise voids. Features in the mould are used to maintain a consistent spacing between turns of 0.36 mm. The insulated winding, Fig. 11, exhibits imperfect coating in terms of uniformity and surface finish, however, DC resistance testing shows that electrical separation of turns is maintained. Improvements in the impregnation process and mould design will enhance the coating quality of future prototypes. Each experiment is repeated three times and the results averaged.

**VI. RESULTS**

Fig. 12 shows thermal images of the splayed and insulated coil configurations with minimum coolant flow rate (0
ml/min) and maximum coolant flow rate (420 ml/min) when subject to a constant DC current of $I_{DC} = 55$ A. The temperature difference between minimum and maximum flow rate of Fig. 12a is small since the primary cooling mechanism is by convection. In contrast, the temperature difference is significant in Fig. 12b at 31 °C, with a minimum temperature of 45 °C along the centre line, turn 9, where the heat exchanger is connected.

![Thermal image of the splayed coil configuration.](image1)

![Thermal image of the insulated coil configuration.](image2)

Fig. 12. Thermal images of the coil configurations with 0 ml/min and 420 ml/min flow rates.

Fig. 13 shows the temperature of turns 1, 9 and 18 as a function of fluid flow rate for the splayed coil configuration. A constant DC current of $I_{DC} = 55$ A is applied resulting in $P_{DC} = 20.8$ W in the steady state with zero flow rate. With no flow, the heat exchanger is ineffective, as shown in Fig. 12a, and the convection cooling from the splayed turns acts to reduce the temperature of extreme turns, 1 and 18, more than the centre turn, 9, with a difference of approximately 11 °C. As the flow rate is increased, the temperature of turn 9 drops to an average of 28 °C while turns 1 and 18 remain at an average of 68 °C. The 40 °C difference in temperature implies that through-conductor heat transfer is poor compared to the convective cooling component. As the flow rate is trebled, turn 9 reduces by 4 °C and turns 1 and 18 by only 2 °C suggesting that the thermal resistance between the heat exchanger and the winding is the limiting factor rather than the solid-fluid heat transfer coefficient of the heat exchanger, which should increase with flow rate. For a constant current, the power dissipation reduces by 0.6 W between zero and maximum flow rate, confirming the small overall reduction in winding temperature linked to the temperature coefficient of conductivity of Cu.

![Temperature vs Flow Rate](image3)

Fig. 13. Measured temperature of turns N1, N9 and N18 for the splayed coil configuration as a function of coolant flow rate.

![Temperature vs Flow Rate](image4)

Fig. 14. Measured temperature of turns N1, N9 and N18 for the insulated coil configuration as a function of coolant flow rate.

For a fixed current of $I_{DC} = 55$ A and zero flow, the average temperature of the insulated coil is 5 °C above that of the splayed coil. This is because the convective heat transfer coefficient is proportional to exposed surface area, which is approximately 20 times larger in the splayed case than the insulated case and is not representative of a practical use-case. At the first flow rate of 140 ml/min the average temperature reduces by 21°C but only reduces by an additional 6 °C and 2 °C for 280 ml/min and 420 ml/min, respectively. This further demonstrates that the limiting factor is the thermal resistance of the winding to heat exchanger connection rather than the solid-fluid heat transfer coefficient. As in the previous case, the reduction in average winding temperature is confirmed by the decrease in power dissipation at fixed DC current, since the DC resistance and therefore DC power loss are temperature sensitive.

Fig. 15 shows the temperature difference between the heat exchanger inlet and outlet for the three flow rates tested in both the splayed and insulated cases. The power absorbed by the coolant fluid, $P_{hx}$, is given by (16) where $c_p$, $\rho$, $V$ and $\Delta T$ are the coolant specific heat capacity (J/kg. °C), mass density (kg/m³), volume flow rate (m³/s) and the difference between inlet and outlet temperature (°C) respectively, [33]. Assuming fixed values for $c_p$ and $\rho$, the heat exchanger in the insulated case extracts more than twice the power than in

![Temperature vs Flow Rate](image5)
for changes in conductor thermal conductivity or convective heat transfer coefficient. Comparing this to the original ISG thermal resistance of the insulated winding is approximately 3.1 °C-1 mΩ, and 

\[ R_{DC}(T_{op}) = R_{DC}(1 + \alpha (T_{op} - T_{amb})) \]  

(17)

If a flow rate of 140 ml/min is assumed, the overall thermal resistance of the insulated winding is approximately 3.1 °C/W, meaning that 58 W could be dissipated for an average winding temperature of \( T_{op} = 180 °C \), commensurate with Class H insulation. Adjusting the measured \( R_{DC} = 5.2 \) mΩ at ambient, \( T_{amb} = 20 °C \), for \( T_{op} = 180 °C \), (17), where \( \alpha = 0.0038 °C^{-1} \) as with pure Cu, gives \( R_{DC}(T_{op}) = 8 \) mΩ, and \( I_{DC} = 85 A \) for DC conditions. Repeating the analysis for \( R_{DC} = 3.2 \) mΩ, representative of 100% IACS Cu, gives \( I_{DC} = 105 A \), not accounting for changes in conductor thermal conductivity or convective heat transfer coefficient. Comparing this to the original ISG windings which were designed for a continuous current of 58 A\(_{RMS}\) with stator housing cooling, [26], gives a potential 81% increase in continuous current capability of the end-winding heat exchanger concept, neglecting AC loss effects. This is under the assumption that the 27% natural convection cooling component, Fig. 16, would be provided by the stator-winding interface in a practical application. Hence, the end-winding heat exchanger concept coupled with through-insulation heat conduction is shown to be an effective thermal management method and could afford enhanced current ratings leading to higher power-density electrical machines.

VII. DISCUSSION

The end-winding heat exchanger concept shows promise in providing enhanced thermal management and hence improved power density in electrical machines. However, there are important practical challenges in realising a robust system. For example, the Coolflow ECO20 must be substituted with a dielectric fluid such as VoltCool, [34], in high voltage applications. Each coil effectively forms a four-terminal module with two electrical and two fluid terminals which conceptually requires a current busbar and a fluid busbar. As with the electrical configuration, the fluid configuration may have coils in parallel to reduce the overall pressure drop and subsequent fluid pumping power required, at the expense of housing a more complex and potentially physically larger fluid busbar. In order to minimise impact on packaging length, the heat exchanger concept coupled with through-insulation heat transfer on an application basis. Given the large number of degrees of freedom, such an optimisation problem will prove challenging and experimental results such as those presented here can be valuable in establishing accurate mathematical models. In this work, the performance improvement is estimated assuming DC loss, however, AC loss effects must be included to establish real-world improvement, which is the subject of further work.

VIII. CONCLUSION

In this paper, an integral end-winding heat exchanger concept is introduced, made possible by metal AM. The topology is modelled analytically and validated via 3D FEA. It is shown that for the present geometry, the through-insulation heat path dominates over the through-conductor route. A lumped parameter thermal model, able to predict the temperatures of each turn from an analytical AC loss model, is presented and used to demonstrate how the optimal heat exchanger location can be identified. A pure Cu prototype is manufactured using LPBF and thermally tested in a splayed state to assess the through-conductor heat path. The heat sinking capability of the insulated case, which is most representative of a practical application, is highlighted by the steady state temperature reduction as flow rate increases. This is further supported by an energy balance analysis. There are important practical challenges to be addressed, for example, the use of dielectric coolant fluids. However, the concept alleviates the need to occupy valuable stator slot volume and the reliance on good stator-winding thermal contact. The approach could provide...
a route to non-destructive disassembly and repair, particularly advantageous for high-value, low-volume manufacture of electrical machines. The cooling efficacy is strongly coupled to the winding geometry where significant temperature gradients may exist for relatively long active length windings. In this case, through-conductor liquid cooling could be extended to strategically selected turns, to further enhance the thermal performance whilst minimising the occupied stator slot volume and pressure drop penalty. The end-winding heat exchanger concept illustrates the new thermal management opportunities arising from AM technologies which will prove valuable in pursuit of high power-density electrical machines, necessary to meet ever pressing electrification targets.

REFERENCES


