

CFD MODELLING OF AN ENTIRE SYNCHRONOUS GENERATOR FOR IMPROVED THERMAL MANAGEMENT

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Abstract

This paper is the first in a series dedicated to investigating the airflow and thermal management of electrical machines. Due to the temperature dependent resistive losses in the machine's windings any improvement in cooling provides a direct reduction in losses and an increase in efficiency. This paper focuses on the airflow which is intrinsically linked to the thermal behaviour of the machine as well as the windage power consumed to drive the air through the machine. A full CFD model has been used to analyse the airflow around all major components of the machine. Results have been experimentally validated and investigated. At synchronous speed the experimentally tested mass flow rate and torque were under predicted by 4% and 30% respectively by the CFD. A break-down of torque by component shows that the fan consumes approximately 87% of the windage torque.

1 Introduction

This paper discusses airflow management for a mid-sized four pole synchronous generator utilising Computational Fluid Dynamics (CFD) modelling and experimental validation. Electrical machines require cooling due to resistive losses created within the core rotor and stator windings. The windings are electrically insulated but the insulation will severely degrade above a peak temperature for the given insulation class. Adequate cooling must be targeted at key regions to avoid exceeding the maximum permissible temperature. By cooling the windings, temperature dependent resistive losses are reduced which directly improves the machines' efficiency [1-3]. Airflow and thermal management is key to improving the machine's overall efficiency. Thermal aspects in the generator are directly linked to the airflow as the air velocity over a surface directly impacts the local heat transfer. Understanding where cooling is required, improving cooling by tailoring the airflow and reducing airflow in regions where it is not required is key.

Airflow modelling of electrical machines is traditionally carried out by designers using empirical correlations [4, 5]. Empirical relationships are limited to the specific geometries of a given machine [6]. Flow resistance networks in conjunction with correlations [7-9] provide an improved

modelling technique, but are not able to provide the detail required to accurately predict the important surface heat transfer which dictates the conduction paths within the important solid windings and laminations [6, 10]. Experimental testing of surface heat transfer is expensive and discrete so does not allow a full understanding of the heat transfer at the surfaces [11]. CFD allows the user to predict the airflow throughout the machine to a far finer resolution [6]. Local velocities may be calculated allowing accurate heat transfer coefficients to be obtained. Using CFD to predict airflow and surface heat transfer has been used an approach in the past [2, 12-14] for generator analysis. However, these investigations have been with 2D or 3D partial geometries. Conduction in solid regions has also been studied but is segregated from the airflow and relies on the mapping of discrete heat transfer coefficients. The novelty of this work is the attention to mesh detail in the airflow case presented here with a view to incorporating full conjugate heat transfer modelling to fully understand both the thermal and airflow properties of the generator in future work within this project. Within this paper a full 360° CFD model of the generator is presented. The exciter, 4-pole-rotor, stator, centrifugal fan and casing are modelled. Details of an experimental rig, used to provide validation for the CFD model, are provided. Aerodynamic losses around specific geometries, such as the fan, are analysed in detail to identify strategies to reduce windage losses.

Beyond the initial work reported here, a full combined airflow and conjugate heat transfer CFD model will be run to give a full thermal model of the machine.

2 Experimental Methodology

A test rig has been commissioned to validate the CFD airflow modelling carried out in this project. The mass flow rate through the machine, measured shaft windage torque, speed and pressure drops across the machine core allow the CFD model to be validated. The testing described here was run with the generator driven at speed but electrically isolated generating no load. This facilitates the measurement of windage loss.

Mass flow rate is measured through the machine. A sealed plenum is created (*Figure 1*) around the machine inlet. Air enters the plenum through a calibrated conical inlet (*Figure 1*)

manufactured according to B.S.848. A ‘Furness Controls FC012’ digital manometer is used to obtain the pressure drop over the inlet which is converted to a mass flow rate. Air is exhausted from the generator back into the room perpendicular to the inlet to avoid interfering with the intake measurements. The laboratory is maintained at a constant temperature during testing, although heat created during testing is minimal and any reduction in air density due to a temperature increase is considered negligible. The manometer has an accuracy of $\pm 2.5\%$ and the mass flow rate result has an overall uncertainty of $\pm 2.1\%$ found using the Kleine McClintock method for uncertainty analysis [15].

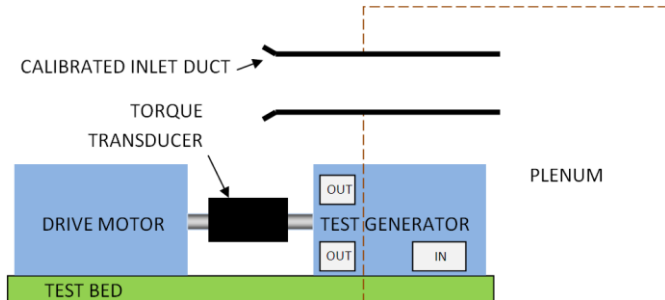


Figure 1: Schematic of experimental rig

Torque and speed measurements are taken from the shaft mounted ‘Datum Electronics M420’ torque transducer which is situated between the test generator and drive motor (*Figure 1*). The results provide the windage power losses. The torque transducer has a manufacturer’s supplied accuracy of $\pm 0.135\%$. Understanding the distribution of viscous and pressure torque losses through the machine allows a greater understanding of the airflow, allowing improvements via geometry alterations. The torque measurement provides a total loss, which includes the bearing losses and windage, whilst the CFD predicts the pure aerodynamic windage.

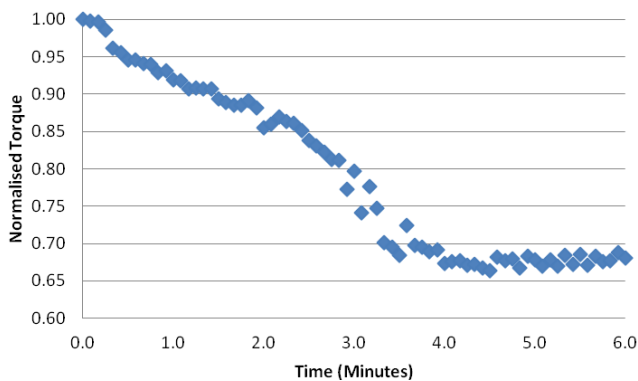


Figure 2: A graph showing the variation of torque due to bearing warm-up over the first 6 minutes of operation at synchronous speed

During an 80minute test it was noted that there is a reduction in torque over the first 5 minutes of operation at full speed due to the grease packed bearings warming and reaching a steady state operating condition (*Figure 2*).

The mean steady state torque is 68% of the initial full speed peak torque with a standard deviation of 3%, as shown in *Figure 2*. The estimated bearing losses are subtracted from the measured torque using the ‘SKF’ simple ‘Estimation of the frictional moment’ model [16].

Future work will investigate more accurate modelling of the bearing losses and an experimental tare down test for bearing losses. All tests are carried out after this warm-up period. Between changes of shaft speed the bearing torque is stabilised for 1 minute before readings are collected. The machine under test has a synchronous speed of 1500rpm. However, to aid understanding of the behaviour of the components within the generator, and to provide more validation of the CFD modelling, test data is taken for a range of shaft speeds.

Experimental testing is carried out for a variety of cases including with and without grilles over the outlet areas. The CFD model presented does not have grilles modelled over outlets so the experimental results without grilles are used for validation.

3 CFD Methodology

The numerical CFD code used in this project is “ANSYS’ Fluent version 13.0”[17]. This is a Reynolds Averaged Navier-Stokes (RANS) solver. The equations are solved within a meshed fluid domain for the airflow simulation.

Firstly a full 360° geometry of all the major components in the generator is created in “ANSYS ICEM CFD version 13.0” (ICEM). The 360° model is required due to the lack of reflective symmetry in some key geometrical features (*Figure 3 and Figure 4*).

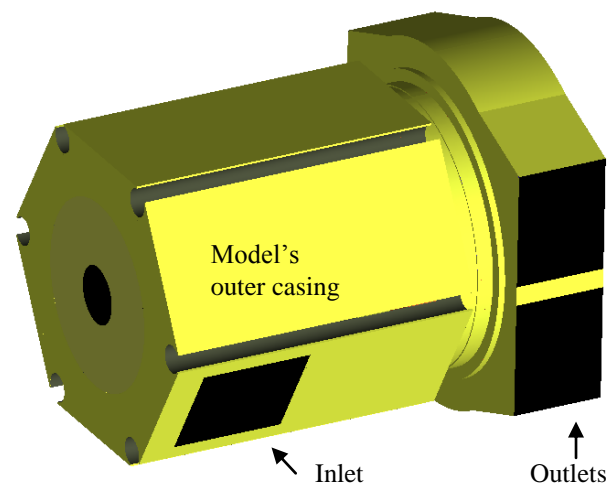


Figure 3: Machine external geometry model in 'ICEM CFD'

A three dimensional hybrid mesh is then constructed within the domain using “ICEM”. Generators prove an interesting challenge to mesh efficiently with geometry scales of $< 1\text{mm}$ in and around the airgap to open regions of $> 100\text{mm}$ and for this reason a hybrid mesh utilising hexahedral and tetrahedral elements is employed. Tetrahedral elements are very efficient within large volumes whilst dense hexahedral meshes provide

the level of density required in areas of high shear. The airgap between the rotor and stator which is an annular region of the order of a few millimetres is meshed independently using ICEM's hexahedral 'blocking' feature. This allows a consistent mesh density of 10 cells across the airgap to be maintained which is essential for predicting the complex flow within the airgap which are a key driver of heat transfer and also a key component of the windage torque created. This distribution of cells is important to the success of the solution. The hybrid mesh used for the simulation described in this paper is 8 million cells. Mesh independence is checked by doubling the mesh size and ensuring that changes in the solution are negligible. In all cases the flow is assumed to be steady state, turbulent, isothermal and incompressible.

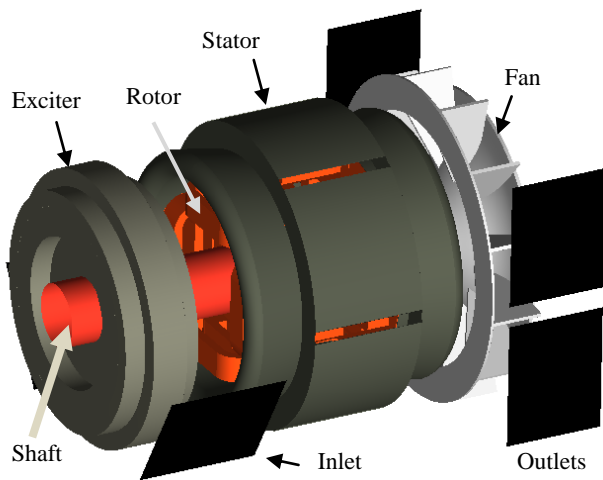


Figure 4: Inner components of the machine's geometry in 'ICEM CFD'

The main model discussed in this report is the machine with the casing, inlets and outlets as boundaries to the model. This assumes at the inlets and outlets are constant pressure over the surface and flow enters the domain normal to the boundary. To validate this assumption, a model including the surrounding room was created. This added just over 10% additional cells to the model as a cell growth ratio of 1.2 was employed from the machine into the room. Mass flow rate through the machine increased by 3% and torque increased by 3.5% when the room was included in the model with respect to the machine-only model. Although slightly more accurate, it is considered that the additional computational resource is not beneficial. Future work will identify the optimum inlet and outlet zone sizes to improve modelling but at the minimal increase in cell count. The assumption at the outlets is reasonable but it is important to understand that the inlet and outlet effects have been simplified.

As the synchronous generator operates at a steady speed a steady-state solution is sought. Shanel [18] found that the significant cost of transient sliding mesh simulations was not worth the minor benefit of monitoring the non-steady pumping effects of the rotor. As the generator is operated without electrical load for the validation tests the assumption of isothermal flow is valid. The rotor speed is at Mach 0.053 and hence the flow is within the incompressible range.

Fluent's Multiple Reference Frame (MRF) or 'Frozen rotor' technique is employed in this model (ref) to simulate the relative motion of the rotor and stator. The fluid region is split into two concentric cylinders by an interior surface. The inner fluid region is set to rotate about the fixed rotor geometry at the synchronous speed. The outer fluid region is set to be stationary (Figure 5). Mixing occurs across the middle interior surface. The benefit of this technique is a more efficient computation time to solution. However the interaction of the rotor poles and stator slots is not resolved as would occur in reality and the pumping effects of the rotor poles are not modelled.

The standard k-epsilon model [17] is used for turbulence modelling within Fluent as it is more stable for rotating simulations [17]. To resolve the boundary layer a wall function is used. The location of the first cell at a wall is determined by the wall function requirement and is indicated by the y^+ value which is a non dimensional distance required for the first cell from the wall based on the shear stresses at the wall [17]. The 'enhanced' wall function [17] is used for all cases described in this paper, this allows a wall function to be enabled in regions of relatively coarse mesh whilst enabling a finer mesh to predict the shear behaviour in the boundary layer of the rotating components. For instance in the airgap the geometry scale and complexity determines that the cell sizing must be far smaller than the standard wall function [17] would require. The enhanced wall function automatically assigns the method for resolving the boundary based on the y^+ value. At low y^+ values of ~ 1 the boundary layer is resolved to the viscous sublayer. At values $y^+ < 11.225$ a wall function is used to compute turbulent region which is particularly valid for $30 < y^+ < 60$. At values $3 < y^+ < 10$ the linear laminar logarithmic turbulent laws are blended [17].

The PC running the simulation is a 3GHz Intel Quad core CPU with 16GB RAM running the Windows 7 64-bit operating system. A converged steady-state solution is run in around 12 hours.

Boundary conditions are essential to the success of a CFD simulation. Wall boundaries are used to represent most surfaces in the model. Walls in contact with rotating fluid zones are set stationary with respect to the adjoining fluid zone. Pressure inlets and outlets are used to represent the inlets and outlets respectively (Figure 3 and Figure 4). There are 2 inlets and 4 outlets from the machine with reflective symmetry about the vertical plane in the axial direction (Figure 3 and Figure 4). Fluid motion through the machine is driven by the rotation of the fan surfaces.

Several geometrical simplifications are employed within the CFD model. These are listed below, each is employed to reduce the number of cells required. As part of the modelling process the significance of each of these assumptions will be investigated to verify the simplifications are valid.

Geometry Assumptions:

- Exciter airgap is neglected – this will reduce the aerodynamic torque on the shaft by a small amount.

However, the relative size of the exciter compared to the main rotor determines that it is negligible

- Stator end-windings are simplified by assuming the spaces between windings leaving the stator are neglected as the majority of this space is filled with excess slot liner
- Windings are modelled as bulk solid regions in all areas as they are randomly wound during manufacture. Therefore any attempt to model these individually would only be appropriate for one machine.
- Grilles over the outlets are not modelled as an experimental case that matches this is utilised. A case investigating the inclusion of the model of the room around the generator found there to be a negligible effect on the flow.
- The airgap is considered to have uniform spacing between rotor poles and stator. In reality there is a small grading in the airgap

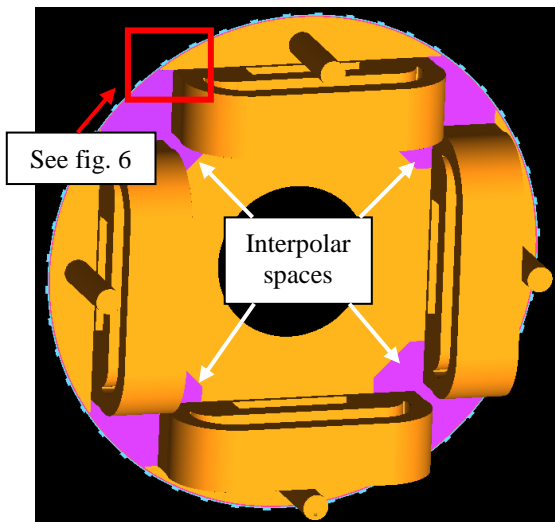


Figure 5: Interior surface labels of the interpolar spaces shown on a midway axial plane through generator

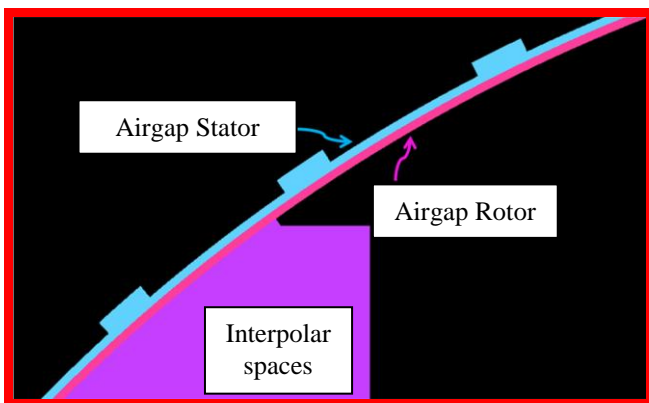


Figure 6: Interior surface labels of the airgap shown on a midway axial plane through generator

4 Experimental Results and CFD validation

The key experimental data for validation of the CFD model includes shaft torque and mass flow of air through the machine at a range of shaft speeds.

A matrix of experimental tests has been carried out, all without electrical load on the generator. The tests have included the rotation of the generator in both directions, an investigation on the effects of grilles over outlets, the effect of closing outlets and a completely throttled case where both inlets and outlets were blocked. This combination of tests has provided insight into the impact of inlet and outlet configurations at a range of shaft speeds on both torque and mass flow rate through the machine. This has been used to validate the CFD model as described below.

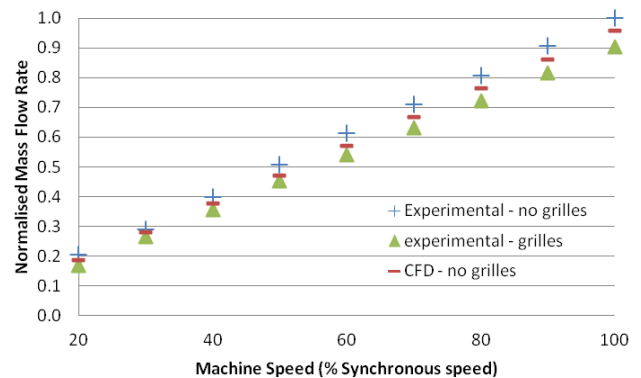


Figure 7: A graph to show the effect of grilles on mass flow rate and the validation of CFD results without grilles

Figure 7 shows two experimental cases, one with grilles present over the outlets and one without, and the CFD results across a range of shaft speeds. It is clear from Figure 7 that the grilles restrict the air flow rate through the machine. By removing the grilles, 10.4% more mass flow rate at full speed may be obtained with an increase of 7.8% torque (Figure 8). The CFD results in Figure 7 clearly predict the trend of mass flow rate and only under predict the experimental mass flow rate at the greatest point by 4% at synchronous speed. This is considered a good match between CFD and experimental data and sufficient to validate the approach to modelling the mass flow rate.

Figure 8 presents the comparison between the raw experimental torque data, the experimental data processed to remove the bearing torque and the CFD prediction which includes torque due to windage alone.

The raw experimental torque data is the total torque to drive the machine which includes bearing losses. During the processing of experimental data the bearing torque is estimated as described in the experimental methodology section (2) above. The bearing torque is currently modelled using a simplistic model and this undoubtedly introduces an error into the processed experimental data and for this reason future plans include capturing measured bearing torque data across a range of shaft speeds.

Figure 8 shows that the CFD under predicts the experimental torque, after allowing for estimated bearing losses, by 30%. This is a much larger difference than with the mass flow rate validation and may be as a result of an incorrect estimation of bearing loss. However the trend with increasing shaft speed is clearly captured. Further investigations into accurate bearing loss predictions are required.

Significantly, accurate trends are being predicted by the CFD for both mass flow rate and torque.

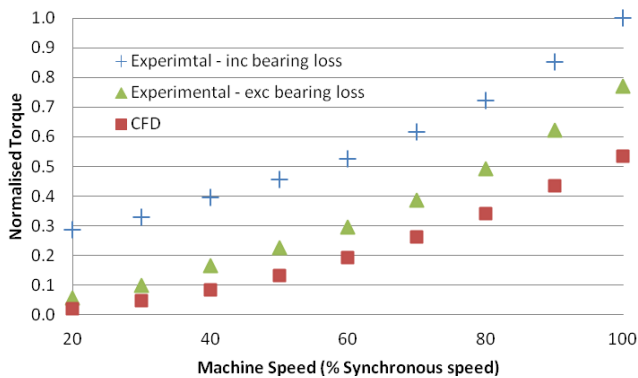


Figure 8: A graph to show the validation of CFD torque results using experimental data

5 CFD Results

Results at synchronous speed and their validation are of the greatest importance. However, by running experimental tests and CFD cases at a range of speeds, trends can be matched which is very important to show the physics is correctly being modelled. Trends in the machine's airflow behaviour also allow a greater understanding of whether components' share of windage alters through the speed range.

The flow paths through the machine are monitored using interior surface patches. Interior surfaces are created over the drive and non-drive end of the airgap and interpolar regions as well as at the non-drive end in the stator channels between the stator and casing (Figure 5 and Figure 6).

Flow in and out of the machine is monitored through the inlet and outlet boundaries. Across the speed range, a nearly constant 32% of flow passes through the airgap and interpolar regions with the remainder passing behind the stator.

Both inlets are located on the lower half of the machine opposite each other (Figure 1, Figure 3 and Figure 4). Around 12% more air flows in through the inlet whose inflow joins the direction of rotation.

Only around 12% of the air which passes in between the rotor and stator enters through the airgap even though it contributes to 17% of the area in this region (Figure 6). The rest enters through the larger interpolar spaces.

Around 35% of the mass flow through the front of the airgap falls into the interpolar spaces by the time it reaches the rear of the airgap increasing their flow by around 5% each (Figure 5 and Figure 6). 90% of the flow leaves through the outlets

whose normal are aligned to the tangential direction of rotation. This suggests that the other outlets are poorly located. This, however, is emphasised due to the pressure outlet normal outflow condition mentioned earlier.

CFD results allow the total airflow related windage torque to be identified. The windage due to the fluid may be broken down to pressure and viscous components and these can be further broken down by each important geometry feature.

Pressure torque contributes the majority of the total torque over the speed range investigated with a mean value of 97.8% of the total torque with a standard deviation of 0.3%.

This suggests that there might be very little in the way of improvement in machine efficiency associated with any improvements made to the viscous related properties of the machine such as surface roughness.

As would be expected, the key region for the viscous dependent torque on the rotating parts is the rotor (Figure 9), due to the high gradients of shear stress experienced in the airgap. However this equates to only around 1.2% of the entire windage torque, suggesting it is not worth focussing too much attention on improving this aspect of the machine. The viscous torque contributions for other components are negligible also (Figure 9). The exciter airgap is not being modelled here to reduce computational expense. This means it's viscous torque contribution is under predicted. However, this is a reasonable assumption since the exciter is shorter than the rotor which is considered negligible so any exciter viscous losses would too be negligible.

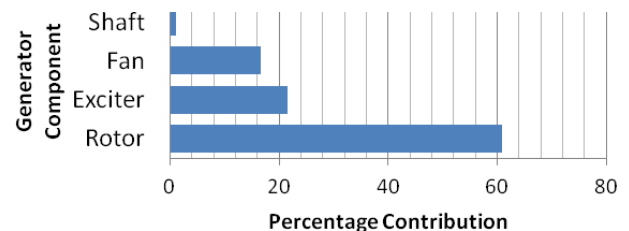


Figure 9: Percentage contribution to the viscous torque by generator component

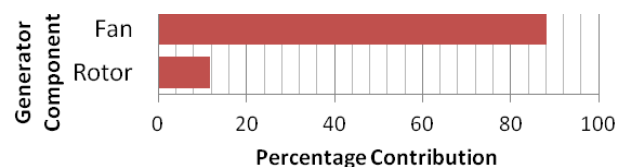


Figure 10: Percentage contribution to the pressure torque by generator component

The contribution of each component to the pressure torque is steady through the investigated speed range (Figure 10). The fan is the most significant contributor to the more important pressure component of the torque on the rotating parts (Figure 9). This is considerably higher than the rotor part.

If a reduction of flow could be achieved with minimal effect on the thermal performance of the machine, these results suggest fan design is the key region for airflow related efficiency gains.

6 Conclusions

A full 360° airflow model of a synchronous machine has been successfully created and experimentally validated. At synchronous speed the experimentally tested mass flow rate was under predicted by 4% by the CFD. Torque was under predicted by 30%. The windage torque is difficult to accurately quantify due to the inclusion of bearing losses in the experimental data. Inclusion of the room around the generator reduces the difference to experimental data to 1.5% and 28% for mass flow rate and torque respectively. Although these results provide a closer match to experimental data, the computational cost of 11% more cells in the model is considered too significant for the improvement in predictions. Analysing the CFD torque results showed that the pressure component of torque was 98%, with the viscous component responsible for the small remainder. These results suggest that there is little point in targeting viscous associated properties of the machine, such as surface roughness for reducing windage, even on the rotor pole surfaces which account for approximately 60% of the viscous torque. The fan is by far the most significant component related to windage. It consumes 87% of the total windage torque from the machine. Aerodynamic only improvements should be targeted at the fan design based on these results. Care must be taken not to segregate these findings from thermal analysis as they are both intrinsically linked.

6.1 Future Work

The airflow modelling shown here is to be used as part of a full conjugate heat transfer analysis of the flow. This will allow detailed linked airflow and thermal management to take place ensuring thermal issues are targeted by the cooling air.

Further specific airflow modelling will look at the effect of transient flow using 'Fluent's sliding mesh model' [17]. Zones around inlets/outlets will be created to ensure accurate modelling at the lowest computational expense. Bearing losses from the experimental results will be investigated further to improve their modelling with experimental testing of bearing losses investigated. Local fluid velocities will be experimentally tested between the stator barrel and casing to further validate the airflow CFD modelling. Detailed modelling of the grilles will be evaluated in future models.

Acknowledgements

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