Abstract – The present paper investigates the effects of rotation on flow resistance of the flow passing through the rotor-stator gap of radial flux electrical machine. Rotational pressure loss was measured and compared with the results predicted using computational fluid dynamic (CFD) methods. By using dimensional analysis, correlation for the shock loss coefficient is proposed. It provides a significant contribution to the field of thermal modelling of electrical machines, such as thermofluid modelling using Motor-CAD.

Index Terms—Correlation coefficient, cooling, electric machines, fans, fluid dynamics, friction, pressure measurement, rotating machines, rotors, thermal analysis.

I. NOMENCLATURE

\( a \)  
Rotor radius (m).

\( D_h \)  
Hydraulic diameter (m).

\( e \)  
Surface roughness (m).

\( p, \Delta p \)  
Pressure, pressure loss (Pa).

\( f_s \)  
Friction factor for stationary condition (dimensionless).

\( f_r \)  
Friction factor for rotating condition (dimensionless).

\( k \)  
Turbulent kinetic energy (m²/s²).

\( K \)  
Loss coefficient (dimensionless).

\( L \)  
Length of flow path (m).

\( \mu \)  
Dynamic viscosity (Pa·s).

\( \omega \)  
Angular velocity (rad/s).

\( \omega_0 \)  
Specific dissipation rate (s⁻¹).

\( \Omega \)  
Volumetric flow rate (m³/s).

\( Re \)  
Axial Reynolds number (dimensionless).

\( Re_0 \)  
Rotational Reynolds number (dimensionless).

\( \rho \)  
Fluid density (kg/m³).

\( s \)  
Gap size (m).

\( V_f \)  
Axial flow velocity (m/s).

\( V_T \)  
Periphery velocity of rotor outer surface (m/s).

\( y' \)  
Dimensionless wall distance (dimensionless).

\( \frac{1}{2} \rho U^2 \)  
Dynamic pressure (Pa).

II. INTRODUCTION

Due to the high power density of electrical machines nowadays, cooling medium is often forced to pass over the heated surface to remove the undesired heat. In order to predict the electrical machine thermal performance accurately thermal analysis of electrical machines must include fluid flow modelling. The coupled model between equivalent thermal network and flow network has been demonstrated by [1] for dual mechanical port machine, [2] for axial flux permanent magnet (AFPM) machine and [3] for radial flux synchronous machine.

The radial flux machine is the most common machine topology used in industrial application. The rotor-stator gap is an inevitable flow path of a throughput ventilated machine and is one of the main areas of where the convective heat transfer takes place. However, the cooling medium passing through the rotor-stator gap is subjected to the effects of rotation. A better understanding of the flow phenomena will significantly benefit the thermal management of radial flux machines.

The topic of Taylor vortex flow between two concentric cylinders was first introduced in [4]. Since then, much work has been done on the convective heat transfer in annular gap (with rotating inner cylinder and stationary outer cylinder) with superimposed axial flow, e.g. [5]-[10]. These published heat transfer correlations can be applied to throughflow ventilated machines. Besides the Taylor number or rotational Reynolds number which is normally used to characterize the rotating effects, the convective heat transfer in an annular gap strongly depends on the axial Reynolds number. In fact, the flow rate of cooling medium is decided and limited by the fan performance and system flow resistance. The impact of ventilation on the temperature rise of a throughflow ventilated AFPM machine has been presented in [11].

A number of studies in [12]-[15] have demonstrated that the rotating effects increase the flow resistance, causes an additional pressure loss to the ventilation system and reduces the obtainable flow rate through the cooling system. Consequently, this must be included in the flow network analysis. However, the review of existing literature reveals that the correlation of describing how the rotating effects increase the pressure loss is very limited. More research needs to be carried out on this topic, and is the aim of the present study. In the perspective of fluid mechanics, it is important to determine how the effects of rotation affect the flow distribution in a rotor-stator system before the heat transfer correlations can be applied.

III. PRESSURE LOSS CALCULATION

For real fluids, the flow in a bounded system experiences energy losses due to frictional and flow separation effects as defined by [16]. The flow separates from the pipe walls when it passes through a disturbance resulting in the formation of turbulence eddies and consequent pressure loss. As separation losses associate with the effect of turbulence, the derivation of expressions defines the effect of turbulence is difficult owing to the nature of turbulent flow. Therefore, a separation loss is usually quantified with empirical loss coefficient, \( K \), based upon the flow kinetic energy as:

\[
\Delta p = K \times \frac{1}{2} \rho U^2
\]  

(1)

Therefore, the pumping pressure provided by a fan is mainly used to maintain the motion of bounded flow.
A. Pressure Losses in an Rotor-Stator Gap

As the flow passing through the rotor-stator annular gap suffers additional flow resistance due to the rotating effects caused by the rotor, the pressure loss of a rotor-stator gap can be expressed as the sum of the stationary and rotational pressure losses. Hence, the total rotational pressure loss can be determined from the difference of rotating and stationary conditions by assuming the stationary loss preserves in the same amount in the rotating condition.

\[ \Delta p = \Delta p_r + \Delta p_s \]  \hspace{1cm} (2)

where \( \Delta p_r \) and \( \Delta p_s \) are rotational and stationary pressure losses respectively. The rotational loss that arises from this flow problem is assumed to be in the form of additional friction loss due to rotation and shock loss.

\[ \Delta p_r = \left[ (K_{fr} - K_{fo}) + K_{shock} \right] \times \frac{1}{2} \rho U^2 \]  \hspace{1cm} (3)

where \( K_{fr} \) is the friction loss coefficient for stationary case, \( K_{fr} \) is the friction loss coefficient with rotation.

The stationary loss can be determined using the relationship between the stationary pressure drop and inlet flow rate that obtained over a range of flow rate. Therefore, the rotational loss is the excess pressure drop for a given flow rate.

1) Friction Loss

With rotation, the characteristics of fluid in the rotor-stator gap are different from stationary case and the flow is in helical form as described by Kuzay and Scott [8]. The flow helix is strongly affected by the rotor speed and flow rate passing the rotor-stator gap. As the fluid friction is associated with shear stresses developed between layers within the fluid that are moving relative to each other, the formulas of friction factor for stationary case cannot be applied for rotating case. However, the friction factor of air flow passing through an annular gap with an inner cylinder rotating for turbulent flow can be estimated using the following correction factor, \( C \), proposed by Yamada [15]:

\[ C = \frac{f_r}{f_0} = \left[ 1 + \left( \frac{2}{7} \right)^2 \left( \frac{Re_{aw}}{2Re} \right)^2 \right]^{0.38} \]  \hspace{1cm} (4)

The correction factor was used to multiply with stationary friction factor for friction factor that accounts for the influence of rotation. According to [15], the fluid friction of a laminar flow is independent of the rotating effects.

2) Shock Loss

The shock loss was first introduced by Webb [17] for rotor ducts that rotating about a parallel axis. When compared to stationary condition, the fluid suffers additional pressure drop because of the abrupt change of flow direction of the fluid entering into the passages in rotor due to rotation. Therefore, a rotor-stator gap also has this form of pressure loss.

As the additional friction loss can be computed using (4), the shock loss can be quantified and the shock loss coefficient, \( K_{shock} \), can be obtained by non-dimensionalizing the corresponding rotational pressure loss with the kinetic energy of fluid passing through the rotor-stator gap as:

\[ K_{shock} = \frac{2K_{fr}}{\rho U^2} - (K_{fr} - K_{fo}) \]  \hspace{1cm} (5)

B. Dimensional Analysis

Due to the complexity of the flow passing through a rotor-stator gap, the problems of heat transfer and fluid mechanics are normally studied using dimensional analysis to identify the factors involved in the problems. Dimensional analysis is used to establish the functional relationship between geometric dimensions of annulus, operating condition (e.g. rotor speed) and flow condition on the shock loss coefficient. The functional equation of general coefficient of rotational pressure loss, \( K_r \), can be written as:

\[ K_r = f \left( \frac{L}{D_h}, \frac{e}{D_h}, \alpha, \frac{V_T}{U}, Re_{aw}, Ta, Re, Pr \right) \]  \hspace{1cm} (6)

To satisfy the principle of dimensional homogeneity, all the terms of (6) are defined in dimensionless form. The parameters in Table I are used to compute for the dimensionless parameters. This allows the investigations in the present study to be shared through the rule of similarity – geometric similarity and dynamic similarity.

The hydraulic diameter \( D_h \) of a rotor-stator gap is \( 2s \). For fluid mechanics, the length-to-diameter ratio, \( L/D_h \), is a measure of equivalent length of a flow path in terms of its hydraulic diameter. It is normally used to account for the developing flow. The surface roughness of the flow path is taken into account by the relative roughness, \( e/D_h \). Both \( L/D_h \) and \( e/D_h \) are assumed to be only important for friction loss. Therefore, they can be discarded from (6).

The geometric effect of rotor-stator gap is specified by the gap ratio, \( a/s \), which is used to measure the annular gap size relative to the inner cylinder radius.

Besides the dimensionless geometric parameters, other dimensionless groups of fluid mechanics used by the published literature such as rotation ratio, \( \frac{V_T}{U} \), rotational Reynolds number, \( Re_{aw} \), Taylor number, \( Ta \), axial Reynolds number, \( Re \) and Prandtl number, \( Pr \), are also included in (6). In order to reduce the number of terms in (6), it is necessary to discard those dimensionless groups which are irrelevant and unimportant for the present study. As the review of cooling methods for electrical machines reveals that the cooling medium used for the rotor-stator gap and passages in rotor is mostly air, hence the Prandtl number can be removed from (6). Both rotational Reynolds number and axial Reynolds number can be combined as follows:

\[ \frac{Re_{aw}}{Re} = \frac{\rho_o m a D_h}{\mu} / \frac{\rho U D_h}{\mu} = \frac{V_T}{U} \]  \hspace{1cm} (7)

Equation (7) shows that the ratio of the rotational Reynolds number to the axial Reynolds number results in a dimensionless parameter, which is exactly the same as the rotation ratio. In the review of existing literature, the rotation ratio was already suggested by [8], [9] and [17] to account for the effects of rotation on the heat transfer and pressure drop for mixed axial and rotational flow. As the Taylor number is mainly used to measure the stability of rotational flow between concentric cylinders and does not
account for the influence of axial flow on the rotational flow. Due to this reason, the rotation ratio is the most suitable parameter to characterize the rotating effect with superimposed axial flow. Hence, the Taylor number can also be discarded. Thus, the functional equation of the shock loss coefficient of an annular gap becomes:

\[ K_{\text{Shock}} = f \left( \frac{S}{a} \frac{V_f}{U} \right) \]  

(8)

After discarding those irrelevant dimensionless parameters for the present study, the number of dimensionless parameter is reduced from eight to two. This considerably simplifies the process of investigation of the shock loss.

IV. COMPUTATIONAL FLUID DYNAMICS

It has been well-known that CFD method is a very useful tool to investigate complex fluid flow problems, especially those problem with flow paths that are difficult to get access for accurate measurement. The CFD software programme STAR-CCM+ was used in the present study to investigate the effects of rotation on the flow passing through the rotor-stator gap and to characterize the rotational pressure loss.

A. Air Flow Models

For representatives of throughflow ventilated machine, machines of [17] and [18] were selected for the investigation of the shock loss because the machine geometries are known from the literature. In addition, the machine geometry of the experimental test rig in Section V was also used for the CFD modelling. Therefore, the CFD models can be validated by experimental results.

<table>
<thead>
<tr>
<th>TABLE I</th>
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<tr>
<td>GEOMETRIES OF ROTOR-STATOR GAP OF AIR FLOW MODELS</td>
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<table>
<thead>
<tr>
<th>Model</th>
<th>Dimension</th>
<th>Dimensionless Parameters</th>
<th>Machine</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>( \alpha ) (mm)</td>
<td>( L ) (mm)</td>
<td>( s ) (mm)</td>
</tr>
<tr>
<td>A</td>
<td>127</td>
<td>228.6</td>
<td>3</td>
</tr>
<tr>
<td>B</td>
<td>254.8</td>
<td>614</td>
<td>10</td>
</tr>
<tr>
<td>C</td>
<td>75</td>
<td>150</td>
<td>4</td>
</tr>
<tr>
<td>D</td>
<td>75</td>
<td>150</td>
<td>2</td>
</tr>
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</table>

It is important to note that the rotor-stator gap was treated as plain annular gap between concentric cylinders. The slots were not modeled. As the gap size of [17] is not provided in the literature, it was assumed to be 3 mm. Due to limitations of computational resource, an annular gap of 1.6 mm of [18] was not possible to model because the gap is relatively small compared with the rotor radius. Thus, it was increased to 10 mm. Although the exact gap size of that machine was not modelled, the air flow models in Table I with a range of gap ratio would able to provide insight into the impact of annular gap size on the shock loss.

B. Model Definition and Mesh

Since the rotor-stator gap is formed between the rotor and stator, it is cyclic symmetry so only one-sixteenth (22.5°) segment was modelled in three dimensions for Model A. As shown in Fig. 1, periodic planes of this segment are represented by \( p_1 \)-\( p_1 \) and \( p_2 \)-\( p_2 \) respectively. The CFD domain is separated into stationary domain (blue) and rotational domain (grey). Conformal mesh interfaces were created between stationary and rotational domains.

Polyhedral and prism layer meshers were employed. Within the rotor-stator gap of Model A, there are 33 cells across the gap. The minimum mesh size of the Model A was set to 0.006 mm close to the walls, which corresponds to 1/250 of the fluid boundary layer, i.e. half of the gap size. The maximum mesh size of the Model A was set to 5 mm to save the computational cost. The total number of computational cells of the Model A is about 3.5 million. The axial domain lengths up and downstream of the rotor are 2.5 times and 4 times the rotor length respectively for flow to develop. The meshes of the Model B, C and D have similar density.

C. Turbulent Flow Modelling

As steady state operations of the machines were investigated, steady state models were chosen for CFD simulation. The present study had made use of a rotating reference frame to simulate the relative motion of the rotor and stator for time-averaged steady-state solutions. The shear stress transport (SST) \( k-\omega \) (Menter) turbulence model was used to provide closure of the Reynolds-Averaged Navier-Stokes (RANS) equations in the present study. The effect of gravity is ignored. As the Mach number of the cooling medium used in most conventional electrical machines is less than 0.3, the air was assumed to be incompressible with constant density, viscosity, thermal conductivity and specific heat.

D. Boundary Condition

The governing equations are subjected to the following boundary conditions: At the inlet, a range of flow velocity was set. As the motion of the cooling medium of a fan-cooled system is induced by the fan pressure, therefore it is important to note that the pressure drop of the simulated flow rate was within the pressure range of commercially available fans. At the outlet, the static pressure, \( p = 0 \) Pa. At both the inlet and outlet, the turbulent kinetic energy and specific dissipation rate were set at default to be 0.001 Jkg\(^{-1}\) and \( 1 \times 10^{-4} \) s\(^{-1}\) respectively.

Both rotor and stator surfaces are smooth and no-slip. The local speed of the rotor surface is equal to \( \omega D_b \). The
maximum peripheral speed of the rotor was set below \( Ma < 0.3 \). It is determined by the rotor outer radius of the CFD models.

### E. Wall Treatment

As the turbulence model is valid only outside the viscous affected region of the boundary layer, a wall treatment model was adopted to specify profiles of the mean flow quantities in the wall boundary layers. With the intention of resolving the viscous sublayer, a high boundary-layer mesh resolution was created with the wall cell \( y^+ \leq 1 \). All \( y^+ \) Wall Treatment approach was used to compute the wall shear stress, turbulent production and turbulent dissipation.

### F. Solution Strategy

The governing equations were numerically discretized using a finite volume method to a system of linear algebraic equations. These were solved simultaneously based on the conservation of mass, momentum and turbulence parameters using the second-order upwind discretization scheme.

### V. EXPERIMENTAL APPARATUS

Fig. 2 shows the experimental test apparatus to investigate the effects of rotation on the flow resistance of the flow paths of throughflow ventilated machine. The experimental objectives are to validate the findings obtained using the CFD methods in Section IV. The air flow test rig mainly consists of a fan inlet tube that connecting between an inlet orifice meter and an adjustable speed fan, a transparent fan outlet tube that connected to the fan through an expansion adaptor and a test section that accommodates a stator and a rotor driven by an inverter drive induction motor. The rotor can rotate up to about 3000 rpm. The speed was calibrated using a RS TM-2011 Tachometer.

#### A. Test Section

In the test section, the stator and rotor are made of steel laminations to simulate the surface condition of electrical machine. The axial length of stator and rotor laminations is approximately equal to 150 mm. The stator laminations have an inner diameter of 158 mm and outer diameter of approximately 190 mm to suit the inner diameter of the fan outlet tube. The stator laminations are positioned by the stator sleeves (of the same inner and outer diameters of stator laminations) to ensure the stator laminations are placed right over the rotor laminations. The rotor laminations have the outer diameter of 150 mm. This forms a rotor-stator annular gap of 4 mm providing a gap ratio of 0.0533. The rotor laminations are mounted on a shaft and are held in place by 2 bearing mounting plates. The distances between the bearing mounting plate and rotor laminations are approximately 80 mm for the front end and 74 mm for the rear end.

#### B. Surface Roughness Measurement

As friction loss is one of the main system pressure losses, the surface roughness of steel laminations was examined using a standardized method, namely RepliSet. Replicas of steel laminations are shown in Fig. 3 (see Appendix for the replica surface condition under microscopic examination). By taking the mean values of these 9 replicas, the surface roughness of steel laminations is approximately equal to 17.6 \( \mu \text{m} \). (17.6 ± 4 mm gives a 95% confidence interval for the proportion in the measurements). It is used to calculate the friction factor.

#### C. Flow Measurement and Conditioning

An inlet orifice meter with corner tap was selected for the device of measuring the volumetric flow rate. The geometric design, installation and conditions of use of the inlet orifice meter strictly follow the standard specified in BS 848-1 [19]. An expansion adaptor was fitted after the fan as a diffuser to eliminate undesirable effects of adverse pressure gradient and flow separation. A NEL (Spearman) flow conditioner was installed at a distance of one tube inner diameter behind the expansion adaptor to remove the swirl that might be introduced by the fan and to redistribute the air flow velocity profile to a condition close to fully developed flow. Consequently, flow rate determination from traverse measurements using Pitot-static tube conducted in accordance with the BS 848-1 [19] can be performed to calibrate the flow rate measurement using inlet orifice meter.

#### D. Pressure Measurement

In the fan outlet tube, a total of 12 pressure tapping holes were drilled at cross section planes of distance of 1\( D \), 2\( D \)
and 3D from the flow conditioner. $D$ is the fan outlet tube inner diameter. On each cross section plane, four pressure tapping holes are arranged at 90° intervals as shown in Fig. 4. The static pressure at a cross section plane was measured from four pressure tappings that were connected together in a “triple-T” arrangement. As the static pressure measurement is required to make under developed flow condition, it was conducted at a distance of three tube inner diameters after the flow conditioner, namely Plane 3. The measured pressure at Plane 3 reflects the total pressure drop of the test section because the test rig outlet was opened to the atmosphere. The pressure measurements were conducted using an Omega PX277-05D5V differential pressure transducer. The pressure tapping holes were also used for the Pitot-static tube traverse.

VI. COMPARISON OF CFD AND EXPERIMENTAL RESULTS

A. Flow Field in Rotor-Stator Gap

With rotation, the flow passing through the annular gap becomes helical as illustrated in Fig. 6. Model C was modelled using a periodic sector and also solid rotor, but they have no significant difference. Hence, the length of pathway travelled by the rotational flow is longer. This increases the friction loss. As proposed by Kuzay and Scott [8], the flow helix can be identified by the rotation ratio because the rotation ratio accounts for the effect of mixed axial and rotational flow. As mentioned in Section III, the rotation ratio is the simplified form of the ratio of the rotational Reynolds number to the axial Reynolds number as shown in (7). As (4) is comprised of the ratio of the rotational Reynolds number to the axial Reynolds number, it is suitable to be employed in the present study to calculate the friction factor of flow passing through the rotor-stator annular gap.

B. Shock Loss Coefficient of Rotor-Stator Gap

The relationship between the stationary pressure drop and inlet flow rate for Model A, B, C and D was obtained from the CFD results. Then, the shock loss coefficient of annular gap of different gap ratio was computed by using (5). Fig. 5 shows that the shock loss coefficient increases in a parabolic relationship with the rotation ratio. However, the CFD results indicate that the values of shock loss coefficient of different gap ratio (i.e. $s/a$) follow the same trend. Thus, it can be concluded that the gap size has no influence on the shock loss for gap ratio from 0.0236 to 0.0533. The shock loss coefficient of the annular gaps can be correlated as:

$$ K_{shock} = 0.043(V_f/U)^2 $$

In order to ensure the axial Reynolds number has no significant impact on the shock loss coefficient of rotor-stator gap, a range of flow rate passing through the rotor-stator system was simulated for each CFD model. The correlation is valid for the axial Reynolds number in the annular gap ranges from 2300 to 12000.
C. Experimental results

It is important to note that no significant temperature difference between the ambient and the air at the outlet was found (less than 2 °C). Therefore, the assumptions of incompressibility and constant fluid properties are accepted for the present study. Fig. 7 shows the relationship of the air flow rate passing through the test rig and the pressure drop from the Plane 3 to the outlet of the test rig over a range of inlet flow rate. The experimental measurements of pressure drop were repeatable. Due to the consistency of the experimental results, the relationship between the air flow rate and the stationary pressure loss from the Plane 3 to the outlet can be expressed as:

$$\Delta p_s = 276392Q^2 + 2229.7Q$$  \hspace{1cm} (10)$$

where $Q$ is the inlet flow rate. As the rotor-stator gap is the only flow path in the rotor-stator system, the gap flow rate is equal to the inlet flow rate.

As shown in Fig. 8, rotation apparently affects the system curves and increases the resistance of flow passing through the airgap with rotor rotating. The pressure requirement for a given flow rate is higher for rotating condition. Hence, the experimental results confirm the presence of additional pressure loss due to the effects of rotation. The influence of rotation is less marked at higher flow rate when compared to the lower flow rate. Equation (10) was used to compute the stationary loss from the measured inlet flow rate.

By applying (2)-(5), the shock loss coefficient can be obtained and it was plotted against the rotation ratio, i.e. $V_T/U$. The experiments were repeated and the values obtained for the shock loss coefficient were very consistent. As shown in Fig. 9, the shock loss can be neglected when the rotation ratio is less than unity and an empirical correlation for the shock loss coefficient can be proposed for the testing conditions. For gap ratio of 0.0533, the shock loss coefficient can be correlated with the rotation ratio as:

$$V_T/U > 1, \hspace{1cm} K_{\text{shock}} = 0.1(V_T/U)^2 - 0.06(V_T/U)$$  \hspace{1cm} (11)$$

$$V_T/U \leq 1, \hspace{1cm} K_{\text{shock}} = 0$$

The correlation is valid for the axial Reynolds number in the rotor-stator gap ranges from 2400 to 11000, the rotational Reynolds number up to 12700, the Taylor number ($Ta = \rho_0 \omega^2 R^4 \mu / \nu$) up to 1470 and the rotation ratio, $V_T/U$ up to 5.3.

![Fig. 8. Flow resistance curves of the test section for speed up to 3000 rpm.](image1)

![Fig. 9. The variation of shock loss coefficient with rotation ratio for gap ratio of 0.0533.](image2)

The comparison between the CFD and experimental results indicated that the CFD results agree with the experimental results, following the trends experienced in measurements in response to the rotation ratio. Discrepancy between the CFD and experimental results are due to the limitations of two-equation turbulence models based on Boussinesq’s isotropic eddy viscosity assumption in modelling the flow in rotor-stator gap with large rapid changes in strain rate. This includes SST $k-\omega$ model used in the present study. Direct Numerical Simulation (DNS) or Large Eddy Simulation (LES) of the higher order should provide better accuracy. However, due to the expensive computational cost, the large number of CFD simulations required in the present study is difficult to be finished using DNS or LES approaches. In addition to that, as the shock loss is obtained by subtracting the additional friction loss from the total rotational loss, hence the discrepancy also consists of the uncertainties from (4).

VII. DISCUSSION AND CONCLUSION

In this paper, the investigation of the pressure loss that suffered by the air passing through the rotor-stator annular gap is presented. It demonstrates that the flow resistance caused by rotation can be significant, e.g. the rotational pressure loss can go over half of the total system pressure loss. Ignoring rotational pressure loss may lead to
overestimation of the cooling performance achievable by an electrical machine. Machine design engineers should avoid a machine to be operated at a condition with high rotation ratio to high rotational loss.

Based upon dimensional analysis, the correlation of shock loss coefficient is proposed, which is suitable to be applied for the radial flux machines. Such correlation provides a significant contribution to the field of thermal modelling of electrical machines. This will be implemented in the flow network analysis of Motor-CAD for improved thermofluid modelling.

VIII. APPENDIX
Below is the surface condition of a replica viewed from the side under microscopic examination

IX. ACKNOWLEDGMENT
The authors gratefully acknowledge the contributions of Fountain Design Ltd. for their work on building the test rig for the research.

X. REFERENCES

XI. BIOGRAPHIES
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