Thermal Management of Electromechanical Actuators using Natural Convection

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***Abstract*—Electromechanical actuator's performance gets limited due to the temperature rise in the motor windings. This work deals with estimating the temperature rise and designing the actuator's housing to optimize Thermal performance by employing Natural Convection.**

Keywords—BLDC Motor, Heat loss, Motor model, Natural Convection, Thermal Management, Annular Fins, Housing.

I. INTRODUCTION

This Electromechanical actuators (EMA) have found wide applications in the field of legged robotics. EMA majorly comprises of a motor driver or electronic controller, a servo motor which is a Brushless DC (BLDC) motor for our case, a gearbox, and the output rotor.

Fig. 1. Block diagram of an EMA.

One of the challenges with these actuators is the management of heat generated in the motor, which limits the operation time of the motor, which in turn effects the operation time of the entire actuator. Two mechanisms, namely Active cooling and Passive cooling can be used for the thermal management of these actuators which will help us to increase the amount of time for which we can operate our actuator. Active cooling techniques are those which require an external energy source to transfer heat from the source to the sink, for example: fans, liquid coolant being pumped in and out, etc. On the other hand, passive cooling mechanisms do not require any external source of energy for heat transfer. Few examples of passive cooling mechanisms are: free convection using air, heat pipes, etc.

In this project, we will be exploring how to optimally use free convection over the actuator's housing so that we can increase its operation time. First step towards this would be to build a heat transfer model of the motor which replicates the physics of heat transfer inside the motor. This model, then can be used with any of the cooling mechanisms to make appropriate design decisions. Secondly, we will be focusing on optimally designing the actuator's housing using extended surfaces or fins, the part of the actuator that interacts with the atmosphere, for efficient heat transfer through natural convection. The housing of our actuator in particular and the housing of most EMAs in general are cylindrical in shape. Also, their orientation during operation in a legged robot in most of the cases is horizontal. For a horizontal cylinder, addition of annular fins to increase the heat dissipation from the surface exposed to the surroundings has been explored well in the literature, and we will also study a similar design to deal with our problem.

II. HEAT TRANSFER MODEL OF THE MOTOR

In this section we will discuss the heat transfer model of the motor developed, and further try to validate it by using the experimental data available in the motor's datasheet. The actuator being considered comprises of an in-runner type of BLDC motor (shown in Fig.2) i.e., the stator is in the outer periphery and the rotor rotates within the stator. Since, majority of the power loss in the form of heat is generated in the motor windings, which is contained within the stator of a BLDC motor, the idea of exploiting passive cooling mechanism for thermal management- free convection for our case becomes more lucrative.

Fig. 2. In-runner BLDC motor.

To build the model of our motor we will make a finite-element model in Abaqus [1]. As major amount of heat comes from the winding losses also known as copper losses, we will approximate that the entire heat is generated in the motor windings and its rate can be given as:

$$\dot{Q}_{gen} = I^2 R_{phase}$$

where,

$\dot{Q}_{gen}$: rate of heat generation within the winding

I: current flowing in the motor winding

R_{phase}: phase resistance of the motor winding

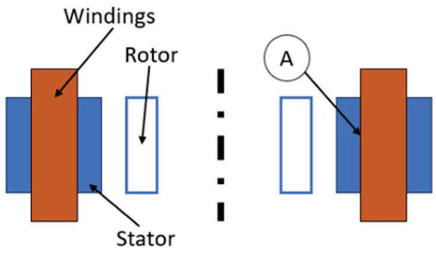

Fig. 3. Schematic of In-runner BLDC motor.

The results of experimental and numerical study based on CFD simulations of an In-runner BLDC motor given in [2] suggests that it would be a good approximation to consider the surface A in Fig.3 to be insulated, that is no heat transfer takes place in the inward direction from the motor stator. We can analyze these results using Fig.4 and Fig.5 in the following way:

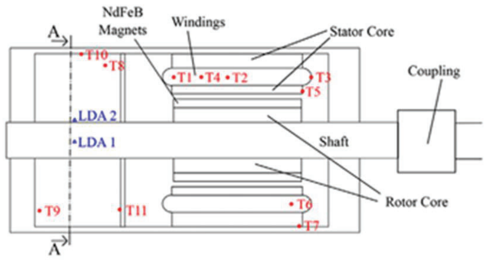

Fig. 4. Location of thermocouple positions for the experimental study [2].

- T_i in Fig.4 shows the location of thermocouples mounted in the in-runner BLDC motor for experimental study.
- Plot in Fig.5 compares the temperature values generated using CFD simulations (red squares) and that recorded using experiments (blue dots) at different thermocouple locations.
- T_1, T_4, T_2, and T_3 values are temperatures at different winding locations, and are nearly same for both experiment and CFD analysis. From this we can conclude that the temperature of the entire winding volume remains roughly constant over the winding region.
- T_5 is the temperature at inner wall of the stator core, and is nearly same as T_1, T_4, T_2, and T_3 for experimental analysis. From this we can conclude that since there is no temperature difference between the winding volume and inner wall of the stator core, nearly no heat flows occur in this direction and hence the surface A in Fig.3 can be assumed to be insulated.

Now in addition to the assumptions made, considering the motor windings as a lumped system as in [3], we can write the energy balance equation as:

$$\dot{Q}_{gen} - \frac{T_w - T_s}{R_{th}} = m_w C_w \frac{\partial T_w}{\partial t}$$

T_w: temperature of the motor windings

T_s: temperature of the outer surface of the stator

R_{th}: thermal resistance between motor winding and outer surface of the stator

m_w: mass of the copper windings contained in the motor

C_w: specific heat capacity of the motor windings

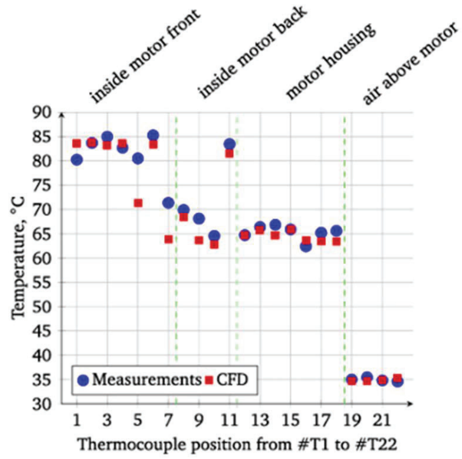

Fig. 5. Temperature comparison between CFD simulation and experimental results [2].

Since our developed model should replicate the governing heat transfer equation discussed in finite element simulations (performed in Abaqus), we do the following:

- Since stator is cylindrical and we will be designing a cylindrical housing, cylindrical shapes will be used to make a complete assembly of the motor model.
- Motor windings is assumed as a very thin cylindrical layer with very high thermal conductivity such that the temperature T_w remains constant over the winding volume V_w, also the entire heat generated will be applied as body heat flux to this layer.
- After this layer we have a cylindrical layer whose thermal resistance is equal to R_{th} and density is zero so that no temperature rise occurs within itself.
- These two layers connected in series comprise the motor model and after this layer we can have a cylindrical housing of our choice.

Few of the geometric dimensions of these layers will be taken equal to dimensions of the motor as per model assembly requirements, and rest will be calculated such that the thermal resistance of the R_{th} layer is equal to the thermal resistance of the motor. Fig.6 shows the schematic of the motor model with few important dimensions marked. We will be using the following notations for specifying and calculating the required dimensions:

R_o: outer radius of the motor stator

R_i: inner radius of the R_{th} layer

L: axial length of the motor stator

k: thermal conductivity of the R_{th} layer

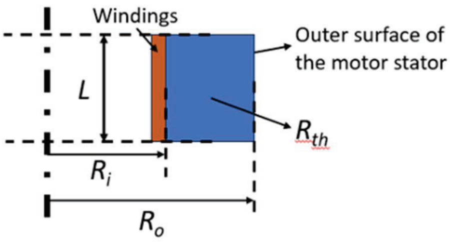

Fig. 6. Schematic of the motor model.

For a cylinder, using R_{th}, R_o, and L from the motor datasheet, we know:

$$R_{th} = \frac{ln\frac{R_o}{R_i}}{2\pi kL} = 1.67$$

$$R_o = 42.5\ mm$$

$$L = 26.3\ mm$$

For ease of calculation, we will assign $k = 1Wm^{-1}°C^{-1}$ to the R_{th} layer which gives us,

$$R_i = 32.25\ mm$$

Now, for the winding layer to be in series with the R_{th} layer, its outer radius will be taken equal to R_i. Since, we also want the winding layer to offer negligible thermal resistance such that the T_w remains constant over the winding volume V_w, we will model the winding layer to be very thin. For simplicity we can take the inner radius of the winding layer to br 32 mm, which will ensure a thin winding layer of thickness 0.25 mm. To further ensure T_w does not vary over V_w, we will assign a very high thermal conductivity (say, $10^6 Wm^{-1}°C^{-1}$) to the winding layer. Since we have fixed the winding volume V_w by fixing the winding layer dimensions, for transient model we have to assign the density of this layer such that its product with V_w equals the mass of the copper windings contained in the motor i.e., m_w. But, for our case we will be performing only steady state analysis and thus any arbitrary density values can be assigned to the winding layer.

Now, assembling the two layers discussed with proper constraints and assigning them the properties and dimensions as discussed completes our motor model for performing simulations in Abaqus. We just need to assign proper loads (heat generated in the winding volume as body heat flux), appropriate boundary conditions (surface film condition for the case free convection), and adding a housing in series with the R_{th} layer, we can get an approximate idea of heat flow and temperature distribution across the entire motor system.

A. *Verification of the Motor Model*

The motor datasheet provides the steady state winding temperature for two operating conditions given as rated conditions, with a specific housing and heat sink design. To verify that the developed model works in accordance with the real system, we add a cylindrical housing and heat sink (provided as a rectangular plate but assumed to be a disc of diameter same as the plate's breadth) with material specifications of Aluminum to our model, as given in the motor datasheet. The surface film conditions for convection through the housing and heat sink surface are applied appropriately for an ambient temperature of 25°C. Heat dissipation due to radiation is assumed to be negligible due to low emissivity of Aluminum. The steady state analysis for a body heat flux load in the winding layer is performed for two operating conditions as given in Table.1.

TABLE I. MOTOR OPERATING CONDITIONS WITH PEAK WINDING TEMPERATURE VALUES

Rated Torque	***Rated Speed***	***Peak winding Temperature***
2.39 Nm	2600 rpm	155°C
1.58 Nm	2700 rpm	85°C

The body heat flux load to be applied for the first operating condition can be calculated as follows:

$$\dot{Q}_{gen} = I^2 R_{phase} = I^2(1.5XR_{line})$$

$$I = \frac{\tau_{rated}}{k_t} = \frac{2.39}{0.166} = 14.397\ A$$

$$\therefore \dot{Q}_{gen} = 41.0435W$$

$$\frac{\dot{Q}_{gen}}{V_w} = 30.926X10^6 Wm^{-3}$$

where, apart from the notations discussed,

R_{line}: line resistance of the motor winding

τ_{rated}: rated torque of the motor

k_t: torque constant of the motor (N/A_{rms})

$\frac{\dot{Q}_{gen}}{V_w}$: body heat flux applied as load

For the second operating condition we calculate the load similarly.

From Fig.7, and Fig.8, we can see that the steady state T_w (which happens to be the maximum temperature region as expected for the motor model) comes out to be very close to the values specified in the motor datasheet (Table.1).

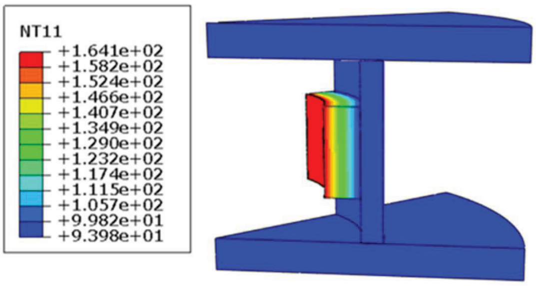

Fig. 7. Temperature distribution for first operating condition.

To further verify, similar models of motor from different series (different dimensions and torque capacity) were built and simulated to get the results close to the experimental values provided. Hence, we can conclude that the developed motor model is a good approximate of the real system.

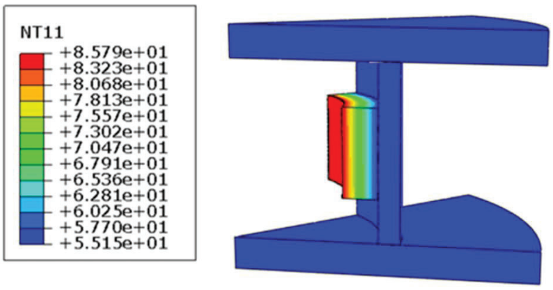

Fig. 8. Temperature distribution for second operating condition.

III. Problem Formulation for Thermal Management

Having discussed the heat transfer model of the motor, we are ready to focus on optimizing the design of our housing for effective heat dissipation. In this small section we will completely state the problem formulated for thermal management of our actuator.

Assuming ambient temperature to be 25 ◦C and neglecting the heat rejected due to radiation, the problem of thermal management for the specified motor model, with constant body heat flux as load, can be formulated as follows:

- How annular fins can be designed optimally over the cylindrical housing to minimize the rise in winding temperature of the motor at steady state?
- Can we design the annular finned housing such that the maximum rise in winding temperature over the ambient temperature is within the safe limits of operating the motor, that is 130°C (Tw = 155°C)?

IV. Natural Convection over Annular Finned Horizontal Cylinder

To address the formulated problem we need to first discuss how to model or calculate the surface film condition over annular finned horizontal cylinder for natural or free convection. The effects of different design parameters related to annular fins on heat transfer due to natural convection have been studied in the literature. For the case of cylindrical surface of the annular finned horizontal hollow cylinder maintained at constant temperature i.e., isothermal wall condition, [4] shows experimentally that the Nusselt number or Nu correlation depends on the fin diameter D. For a similar system, [6] experimentally suggests the existence of an optimal fin spacing S_{opt} such that the total heat transfer due to convection from the finned cylinder is maximized. [5], using results from CFD simulations gives a Nu correlation which can be used to find out the constant average convective heat transfer coefficient i.e., h_{avg} for an annular finned horizontal cylinder under isothermal wall condition. The simulation results in [5] also validate the idea of optimal fin spacing that maximizes the heat transfer rate.

A constant surface film coefficient (h_{avg}) can give us a good idea from design point of view about the system behavior during free convection at steady state, keeping the analysis simple. For our design purpose, we will also use the Nu correlation given in [5] to calculate the surface film coefficient to be assigned as interaction in the Abaqus simulation. Since for our case we do not have isothermal wall boundary condition, rather we are applying a constant body heat flux load, we will have to modify the calculation procedure for h_{avg}.

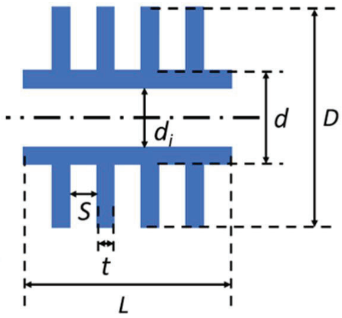

Fig. 9. Schematic of Annular Finned Horizontal Cylinder.

Before going ahead, let us list down the notations we are going to use in this section:

d_i: inner diameter of the cylinder

d: outer diameter of the cylinder

D: diameter of the annular fins

L: length of the cylinder

S: fin spacing

t: thickness of the fin

ζ: fin to cylinder diameter ratio, D/d

T_s: cylinder surface temperature

T_{inf}: ambient temperature

h: average heat transfer coefficient

k_a: thermal conductivity of air

Ra_S: Rayleigh number based on S

Nu_S: Nusselt number based on S

g: acceleration due to gravity

α: thermal diffusivity of air

β: thermal expansion coefficient of air

ν: kinematic viscosity of air

The Nu correlation as given in [5] which we will be using is given as:

$$Nu_S = -3.827 + 0.047 Ra_S^{0.348} \zeta^{0.173} + 1.039 Ra_S^{0.175} + 0.009 \zeta^{2.548} \quad (a)$$

$$(5 \leq Ra_S \leq 10^8)$$

$$Ra_S = \frac{g\beta(T_s - T_{inf})S^3}{\nu\alpha}\left(\frac{S}{D}\right)$$

For a given set of other design parameters, we have to find optimal fin spacing (S_{opt}) such that we can minimize the rise in winding temperature of the motor at steady state, as T_s has a dependence on fin spacing S. As discussed, unlike isothermal wall condition we do not have a constant T_s, rather it is the parameter which needs to be minimized by designing the annular fins optimally, the calculation of h does not come directly from:

$$Nu_S = \frac{hS}{k_a} \quad (b)$$

In paper [5], the goal was to maximize heat transfer rate for a given Ts, whereas we want to minimize Ts for a given heat transfer rate (constant for steady state analysis) i.e., $\dot{Q}_{gen}$. We will be calculating h as follows:

- We will have to first calculate the approximate mean temperature of the cylinder surface exposed to free convection at steady state, that is T_s.
- T_s is calculated as follows:
  - We start with initial guess value of T_s
  - Using $h = \frac{\dot{Q}_{gen}}{A_{conv}(T_s - T_{inf})}$ in (b),

    where A_{conv} is the surface area of the cylinder exposed to free convection, we get,

    $$Nu_1 = \frac{\dot{Q}_{gen} S}{A_{conv}(T_s - T_{inf})k_a}$$
  - Then we iteratively calculate Nu_1 and Nu_S, with T_s being updated at each iteration, until $|Nu_1 - Nu_S|$ converges to a very small tolerance value ($< 10^{-2}$)
  - Using the final converged Nu_S, we can calculate h or h_{avg} using (b)
  - Also, the final value of T_s will be very close to the mean surface temperature of the cylinder, which can be confirmed using simulations where h is applied as the surface film coefficient over A_{conv}. This further gives a mathematical validation to our model.

Iterating over the entire range of possible values of fin spacing S (which is restricted by the axial length of the housing L) and using the above procedure to calculate T_s followed by h, an optimal value of S i.e., S_{opt} can be found for which T_s is minimum. Using this approach we develop an optimization code in python which gives us S_{opt} and h to be applied, with other design parameters related to housing {d_i, d, L, t, D} being fixed. The values of the fixed design parameters are:

$$d_i = 85 \text{ mm}$$

$$d = 90 \text{ mm}$$

$$L = 26.3 \text{ mm}$$

$$t = 1 \text{ mm}$$

$$D = 95 \text{ mm}$$

This answers the first part of our problem formulation. For, the second part, besides considering S as the only design variable, we have to also consider fin diameter D as a design variable. It can be easily shown using (a) and (b) that increasing D will decrease Ts. Therefore, we increase D until we get a Ts value which when used in the energy balance equation discussed in section 2, gives us a $T_w \leq 155°C$ in both hand calculation and simulation. This gives us a D = 194 mm, with rest of the parameters being same and S_{opt} can be found out using the steps discussed earlier.

V. Results

Temperature distribution for a housing without any extended surface is shown in Fig.10. Now if we add small annular fins (D = 95 mm) to the cylindrical housing with optimal fin spacing (S_{opt}), we can see a large drop in the steady state motor winding temperature Tw as shown in Fig.11. For this case S_{opt} is found to be 4.2 mm, and hence a total of 5 annular fins can be accommodated for the given L = 26.3 mm. The variation of mean surface temperature of the housing (T_s) with all the possible fin spacing is represented by the plot shown in Fig.12, where the red dot marks the point of minimum T_s at S_{opt}.

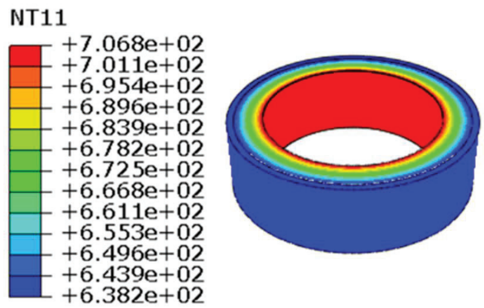

Fig. 10. Temperature distribution for motor with Un-finned Housing.

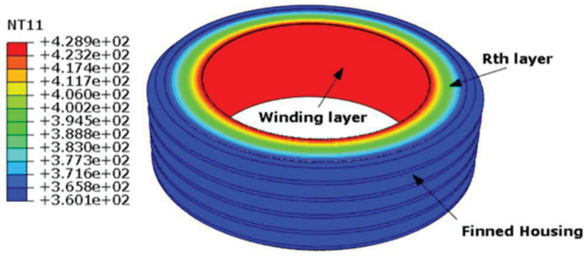

Fig. 11. Temperature distribution for motor with Finned Housing (D = 95 mm).

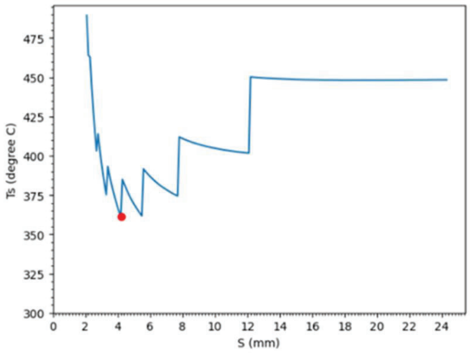

Fig. 12. T_s vs S curve for D = 95 mm

The design of the housing for which $T_s \leq 155°C$, has a fin diameter D = 194 mm. For this case S_{opt} is found to be 5.5 mm which can accommodate 4 annular fins for the given L. For this case, the temperature distribution for this is shown in Fig.13 and variation of T_s with possible values of S is represented in Fig.14, where the red dot marks the point of minimum T_s at S_{opt}.

For each simulation, convergence was checked by varying the mesh sizes and also changing the element type from linear to quadratic, and the results converged with negligible variation.

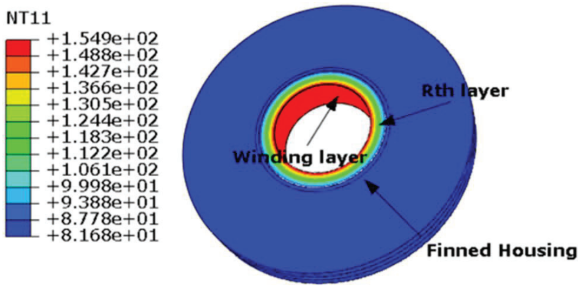

Fig. 13. Temperature distribution for motor with Finned Housing (D = 194 mm).

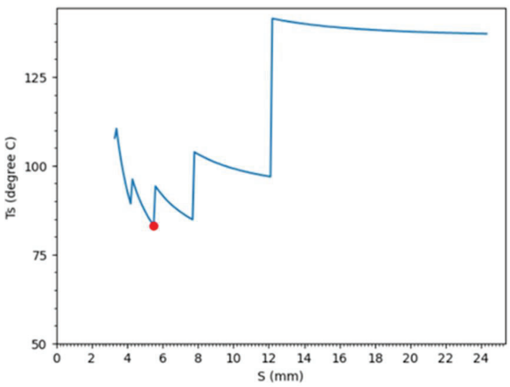

Fig. 14. T_s vs S curve for D = 194 mm

VI. CONCLUSION

Fig.12 and Fig.14 describes how the choice of S, with other fin design parameters kept fixed, can effect our housing surface temperature T_s which in turn effects the winding temperature T_w. We also found out that the design of cylindrical housing with annular fins under natural convection involves the optimization of the fin spacing S and a calculated choice of D. The addition of small annular fins (D = 95mm) helped us to minimize T_s significantly when compared to the un-finned housing design, but was not able to bring it down to the allowable limits at steady state. The obtained design of the annular finned housing which keeps the motor operating in its safe limits at steady state is not a feasible design due to very large fin diameter (D = 194 mm). So, if we want to operate our motor and hence the actuator at rated condition with our own housing design for longer periods of time (steady state), leveraging free convection alone cannot solve our problem. But, definitely the housing design with annular fins can increase the operation time of the actuator up to a certain extent.

The developed heat transfer model of the motor can be extended to transient or time dependent motor model by incorporating the mass of the copper windings contained within the motor stator in the model. Apart from finite element model, CFD based motor model can be built and compared as they are able to capture the flow of air over complex geometries during convection accurately. The scope of this project deals with effective thermal management solutions for an electromechanical actuator. Besides free convection, effectiveness of other passive cooling mechanisms like heat pipes and vapor chambers can be explored on the developed heat transfer model of the motor in the future. Heat dissipation through active cooling mechanisms can also be compared and a design decision can be made in the direction of combining different cooling mechanisms in a calculated manner without increasing the complexity of actuator design.

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