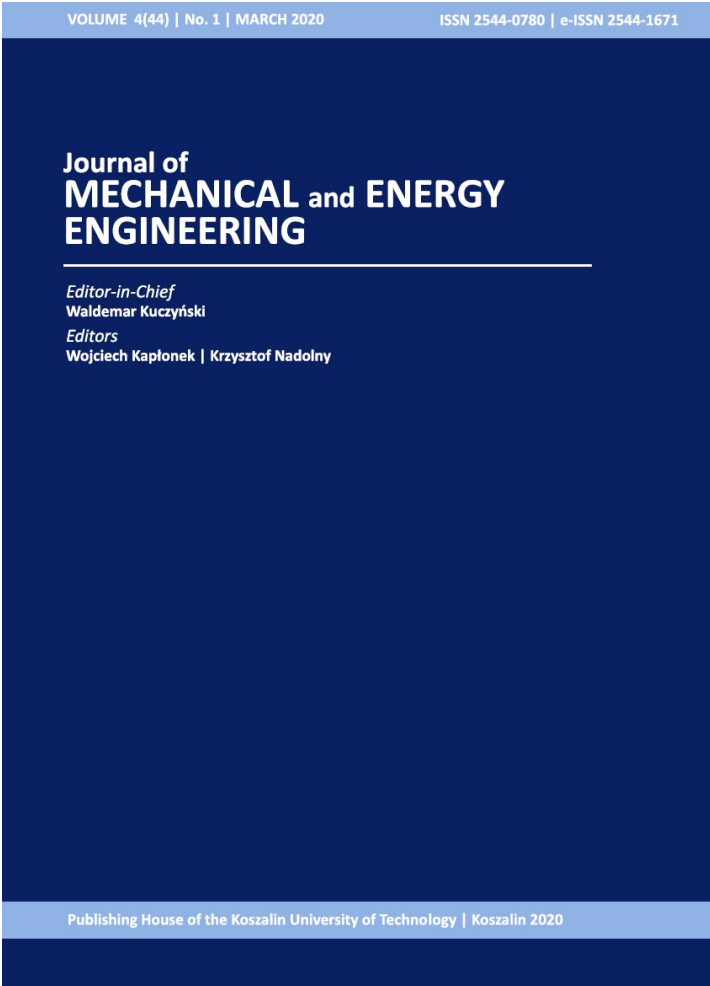# Method for efficient feasibility study of air cooling systems for modern PMSM electric motors in all-electric aviation

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METHOD FOR EFFICIENT FEASIBILITY STUDY OF AIR COOLING SYSTEMS FOR MODERN PMSM ELECTRIC MOTORS IN ALL-ELECTRIC AVIATION

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Abstract: In this paper, the authors present a computational model of a fin-based air cooling system for Permanent Magnet Synchronous Machine (PMSM) electric motors. The model can be used as a method for fast and efficient feasibility studies of air cooling for PMSM motors in hybrid-electric or all-electric aviation applications, supplementing further research (thermal resistance networks, and FEA/CFD-CHT models). In the paper, authors provide temperature distributions along the fin height which are calculated and presented for a straight fin, followed by heat transfer rate from fin surface and fin efficiency. A parameter to compare different fin materials for aviation applications is introduced – heat transfer rate from the fin to fin mass ratio. Aluminum and copper fins are compared. Different shapes of straight fin are considered and compared. The above parameters and comparison are then calculated and given for circular fins. Parameters of the whole fin-based air cooling system for specific 140 kW PMSM motor are calculated and presented.

Keywords: electric motor, air cooling, finned heat sink

1. INTRODUCTION – ELECTRIC MOTORS IN AVIATION

In the field of modern electric machines, the ongoing increase of achievable power density (obtained through progress in the manufacturing methods, improved material properties, or innovative electromagnetic topologies) has led to the situation where thermal constraints of the design process became one of the strong limiting factors of further progress. As losses in an electric motor are directly tied to the output power, increasing the power density of such machines leads to increased heat loss [1]. This in turn has caused the modern, high-power density electric motors to turn from traditional cooling solutions like air cooling or water jackets [2], to-wards innovative, possibly more efficient designs such as direct cooling of the windings [3, 4], oil-spraying [5, 6], heat pipes [7, 8] or using phase-change materials [9, 10].

At the same time, the increase of power density in electric motors made it possible to consider using them in certain high-power applications, where needs regarding weight or volume of the system disqualified electric motors before. One such field is main propulsion in aviation (sometimes described as more-electric and all-electric aircraft) [1,11-13]. With NASA goals to achieve all-electric and hybrid-electric aviation working plane prototypes as soon as early 2020s with small nine seat planes [13], and early 2030s with large 150-300 seat planes [13] it has become imperative to design an electric motor that can work in aviation conditions, and a cooling solution optimal for those conditions, which is lightweight, reliable, and can pass FAA safety tests and regulations.

Taking the above into consideration, it is important to note, that using electric motors as main propulsion system in large aircraft to replace turbo-fan engines opens up new possibilities for air-cooling of the motor [1,11]. In aviation applications, air is an abundant resource. Air velocity in an air-craft engine is far higher than what is typically available for standard air-cooling solutions [14-16], and air at such speeds can result in a very high heat transfer coefficient on the surface of air cooling fins, that might make an air based cooling system viable even for modern high power density electric motors [17].

Considering all of the above, the authors of this paper have created a computational model of fin based air-cooling system for high power electric motors in Mathcad, that can serve as a tool for fast and efficient feasibility studies of using air-cooling. Such a tool can be used as a supplementary assistance for further studies using more complicated models of electric motor cooling (like thermal resistance networks, or FEA and CFD-CHT simulations), allowing for preliminary elimination of certain cases, saving time and effort.

2. PERMANENT MAGNET SYNCHRONOUS MACHINE

Out of different electric motors concepts that can be used as an aircraft drives, the most notable one for this application would be a permanent magnet synchronous machine (PMSM) [1, 12, 13]. Due to PMSM motor construction, a vast majority of heat losses occur in the stator area (mainly in the winding) [4, 17, 18], where the cooling fins are located (placed around the stator casing). Taking this into consideration, a PMSM electric motor was chosen to serve as a basis of calculations presented in this paper. The data presented in subsequent parts is based on a high-performance PMSM motor with power density of 14 kW/kg [19]. The parameters of the motor that served for basis of calculations in this paper are listed in Table 1.

Tab. 1. Parameters of the PMSM motor

Rated power	140 kW
Mass	10 kg
Outer diameter	120 mm
Stator casing length	350 mm
Electromagnetic efficiency	98%

3. STRAIGHT FIN

Straight fins placed on the stator casing along the axis of the motor are the first type of air cooling to be considered. For each parameter, a comparison between aluminum fin and copper fin is presented, to see which material will be optimal to use in this application. The first parameter is the temperature distribution along the height of a fin, which can be obtained from the following equation [20]:

$$T_{fin}(x) := \left[\frac{\cosh\left[z \cdot (H - x)\right]}{\cosh(z \cdot H)}\right] \cdot \left(T_w - T_f\right) + T_f \quad , \quad (1)$$

where: H – height of the fin [m], T_f – temperature of air [°C], T_w – temperature at the base of the fin [°C], x – distance from fin base along the fin height [m], z – given in equation (2).

Parameter z in equation (1) can be obtained from the following equation [20]:

$$z := \sqrt{2 \cdot \frac{\alpha}{\lambda \cdot \delta}}, \quad (2)$$

where: α – convection heat transfer coefficient [W/m²K], δ – thickness of the fin [m], λ – thermal conductivity [W/m K].

Comparison of temperature distribution along the height of a fin be-tween aluminum and copper fins can be found in Figure 1. Comparison is presented for different values of average heat transfer coefficient on the surface of the fin across the whole paper, to illustrate fin performance at different air velocities flowing through the aircraft engine. The limit value of 100 W/m² K is easily achievable across the whole fin surface for air flowing at extreme velocity for aircraft cruise speed.
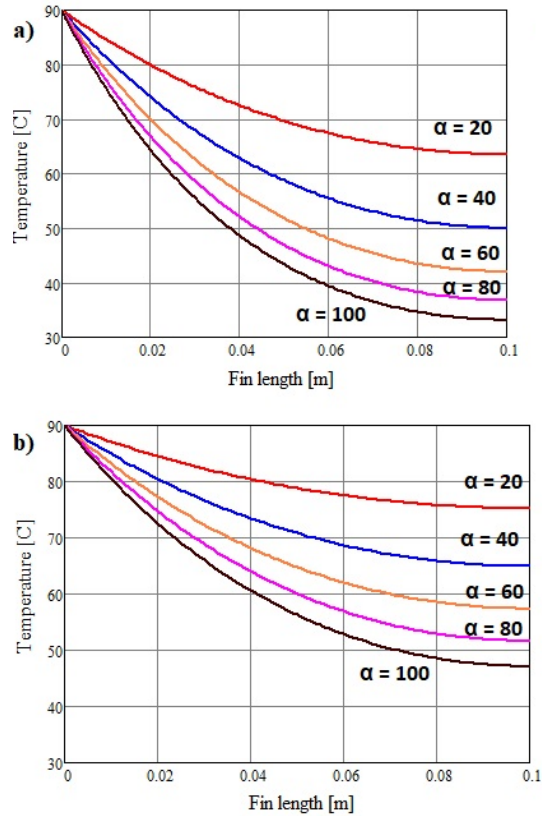
Fig. 1. Temperature distribution along fin height with: a) aluminum fin, b) copper fin

The next considered parameter, heat transfer rate from a single fin depending on fin height, can be obtained from following equation [20]:

$$Q_{sf}(x) := \sqrt{2 \cdot \alpha \cdot \delta \cdot \lambda} \cdot 1 \cdot \left(T_w - T_f\right) \cdot \frac{\left[\left(\frac{\alpha}{z \cdot \lambda}\right) + \tanh(z \cdot x)\right]}{1 + \left(\frac{\alpha}{z \cdot \lambda}\right) \cdot \tanh(z \cdot x)}, \quad (3)$$

where: α – convection heat transfer coefficient [W/m² K], δ – thickness of the fin [m], λ – thermal conductivity [W/m K], l – width of the fin [m], T_f – temperature of air [°C], T_w – temperature at the base of the fin [°C], x – height of the fin [m], z – given in equation (2).

Comparison of heat transfer rate from a fin depending on fin height between aluminum and copper fins can be found in Figure 2. It is clearly visible that copper fin can dissipate more heat than aluminum fin.
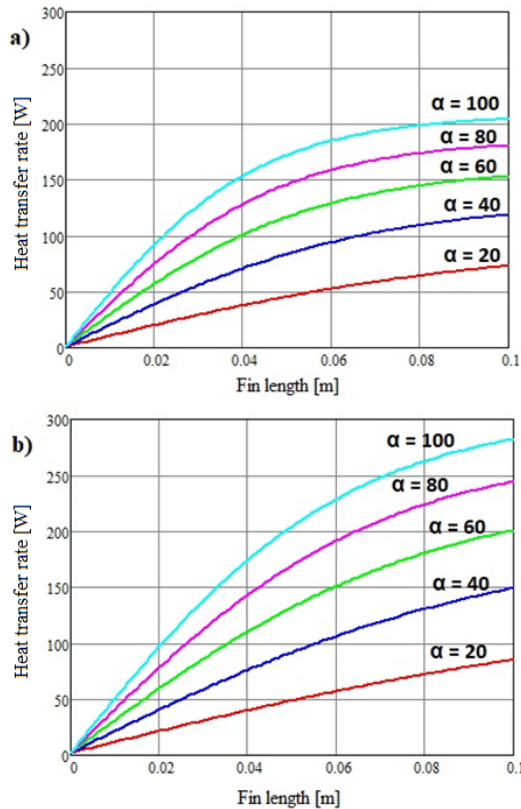

Fig. 2.   Heat transfer rate from a single fin depending on fin height with: a) aluminum fin, b) copper fin

Another important parameter to consider is fin efficiency. While increasing fin height will result in increase of heat transfer area, and thus heat transfer rate from the fin, but also, the longer the fin, the bigger the mass and cost of the fin is. Increasing the fin above the certain height is not practical, and calculation of fin efficiency can help to determine the height of the fin that should not be exceeded. Fin efficiency can be obtained from following equation [21]:

$$\eta_f := \frac{Q_f}{Q_{max}} \quad , \tag{4}$$

where: Q_f – actual heat transfer rate from the fin surface [W], Q_{max} – ideal heat transfer rate from the fin surface (equation 5) [W].

Ideal heat transfer rate from the fin surface can be found from the following equation [21]:

$$Q_{max} := \alpha \cdot A \cdot \left(T_w - T_f \right) \quad , \tag{5}$$

where: α – convection heat transfer coefficient [W/m² K], A – area of the fin surface [m²], T_f – temperature of air [°C], T_w – temperature at the base of the fin [°C].
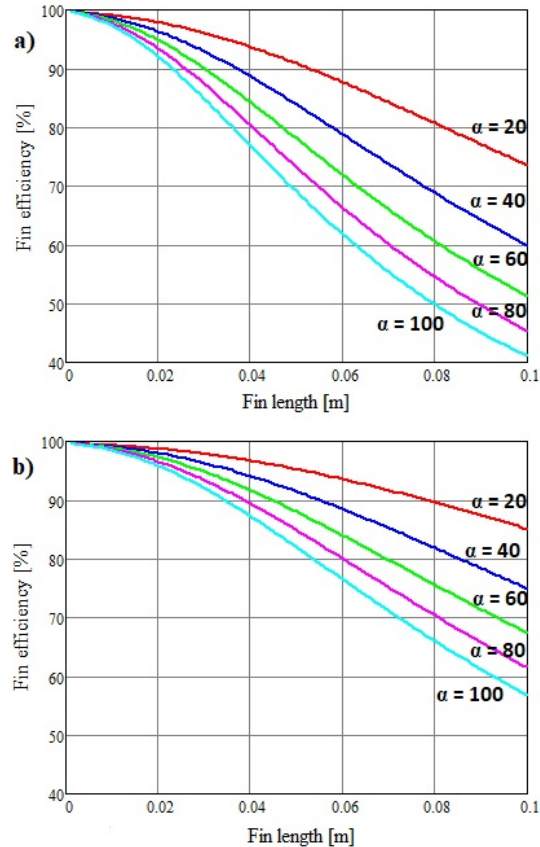

Fig. 3.   Fin efficiency depending on fin height with: a) aluminum fin, b) copper fin

While copper fins seem to perform better than aluminum fins for every above parameter (which is to be expected, based on higher thermal conductivity of copper), a very important factor to consider in aviation centered applications is mass of the whole system. Since copper fin of similar height and thickness would be over three times heavier than aluminum fin (density of copper is 8960 kg/m³ and density of aluminum is 2700 kg/m³), to compare usefulness of both materials in creating heat sinks for aviation electric motors, an additional parameter must be introduced – a ratio of heat transfer rate from a single fin to a mass of a single fin, expressed in Watts per kilogram. The comparison of the heat transfer rate to mass ratio between aluminum fins and copper fins can be found in Fig. 4.
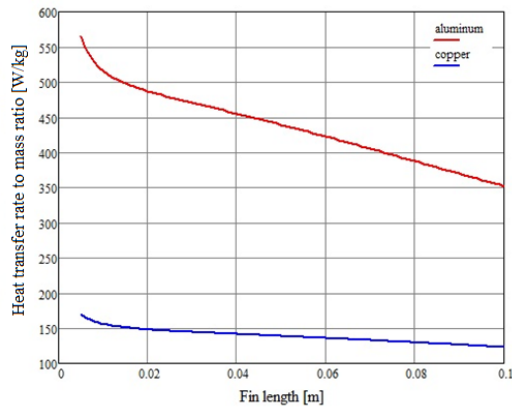
Fig. 4.   Heat transfer rate to mass ratio comparison for aluminum and copper fins

The results presented in Figure 4 show, that even though copper fins boast better performance overall, when introducing the mass dependent parameter, their performance is much worse than that of aluminum fins. Having all these parameters we can finally make a decision, to pick aluminum fin as the basis of the system. Going back to Figure 3a (aluminum fin efficiency), and picking 80% as the acceptable fin efficiency, we can chose 3 cm as the fin height. Based on those factors, we can now calculate the viability of the whole system. The power that has to be dissipated from the stator is equal to 2.8 kW, and the heat transfer rate from a single fin of 3 cm height can be estimated from eq. (5). Having this data allows for calculation of the number of fins needed in the system, the spacing of the fins on the stator casing and the mass of the whole air cooling system. Those parameters can be found in Table 2. The results show that the air cooling system is viable in this application, and it is possible to cool the motor using straight fins. The weight of the cooling system is acceptable, and it brings down the power density of the whole system only slightly, to an acceptable level (to 12.3 kW/kg from 14 kW/kg).

Tab. 2.   Parameters of air cooling system with straight fins

Heat loss in the stator	2.8 kW
Heat transfer rate from a single fin	129.6 W
Number of fins needed	22
Spacing of the fins	17 mm
Mass of the cooling system	1.34 kg

After all of the above is taken into consideration, the next factor we can consider is the shape of the straight fin. Above calculations were made for a standard, rectangular fin shape. We can check how the shape of the fin affects heat transfer from the motor. To do that, we will introduce three functions describing the shape of the fin. Function presented in equation (6)

describes a straight fin, while equation (7) describes a convex trapezoidal fin and equation (8) describes a concave trapezoidal fin. The functions were chosen for a fin parameters similar to the one calculated above, so with fin base thickness of two millimeters, and fin height up to 10 centimeters. Fin shapes resulting from the functions are presented in Figure 5. Functions are as follows:

$$y(x) := 0.001 , \tag{6}$$

$$y(x) := -0.07\,(x - 0.002)^2 + 0.001 , \tag{7}$$

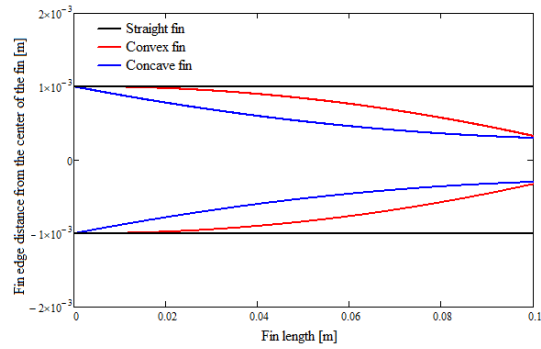$$y(x) := 0.05\,(x - 0.12)^2 + (0.00028) , \tag{8}$$

Fig. 5.   Fin shapes as a function of fin height

After determining shape functions for different fin geometries, we have to introduce equations for cross-sectional area and perimeter of the fin at given location x. Formula for the cross-sectional area will be given in equation (9) and the formula for perimeter in equation (10):

$$A(x) := 2 \cdot l \cdot y(x) , \tag{9}$$

$$P(x) := 4 \cdot y(x) + 2 \cdot l , \tag{10}$$

where: l – width of the fin [m].

Next, we have to introduce a modified z function from equation (2), to include the change of fin thickness along the height of the fin. The new z function is presented in equation (11) below:

$$z(x) := \sqrt{\alpha \cdot \frac{P(x)}{\lambda \cdot A(x)}} , \tag{11}$$

where, α – convection heat transfer coefficient [W/m^2 K], λ – thermal conductivity [W/m K].

Finally, we can introduce a differential equation describing the temperature distribution along the fin height, in the form of temperature difference between the fin and the surroundings [20]. It is given in equation (12):

$$\frac{d^2}{dx^2}\upsilon(x) - z(x)^2 \cdot \upsilon(x) = 0 . \tag{12}$$

A set of boundary conditions needs to be introduced for equation (12), which are difference between fin and surrounding temperature at the base of the fin as $v(0)=70$, and extremum of the function $v(x)$ at the fin tip, as $v'(0.1)=0$. We can solve equation (12) for those boundary conditions, and receive the temperature difference between fin and surroundings along fin height, presented in Figure 6.
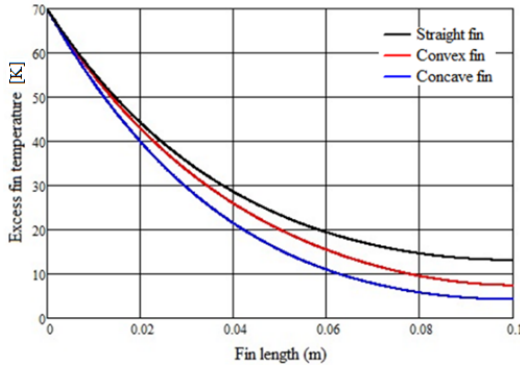

Fig. 6.   Temperature difference between fin and surroundings along fin height.

A set of two additional equations is needed to determine heat transfer between the fin and the surroundings. $Q0$, the heat transfer rate from the fin [20] is presented in equation (13), while Q, heat transfer rate between the fin and the surroundings as a function of fin height [20] is presented in equation (14) below:

$$Q0(x) := -\lambda \cdot A(x) \cdot \frac{d}{dx} v(x) , \qquad (13)$$

where: λ – thermal conductivity [W/m K].

$$Q_1(x) := Q0\left(10^{-6}\right) - \int_0^x \alpha \cdot v(x) \cdot P(x) \, dx , \qquad (14)$$

where: α – convection heat transfer coefficient [W/m² K].

The resulting heat transfer rate for all three chosen shapes of the fin is presented in Figure 7 below.
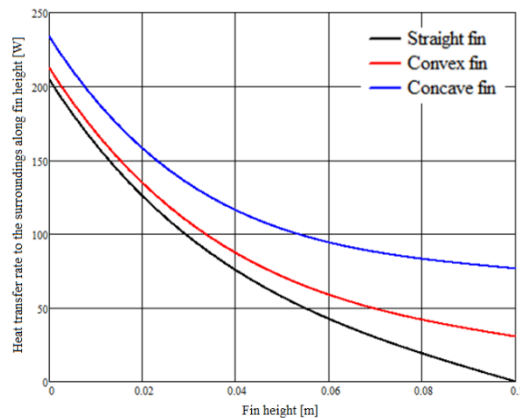

Fig. 7.   Heat transfer rate from the fin along the fin height.

As can be seen in Figure 7, a concave trapezoidal fin has the highest heat transfer rate at any point along the fin height, from the shapes chosen for this comparison. Additionally, it also is the fin with the smallest mass, because of the concave shape. This makes the concave fin the best choice for aviation applications, as the heat transfer rate to mass ratio of the fin is very important parameter, as explained before. After selection of the best fin geometry, it is possible to do further optimization of the concave shape function to maximize heat transfer rate and heat transfer rate to mass ratio from the fin of the same base width and height. This optimization will be the subject of authors further work.

4. CIRCULAR FIN

Second type of analyzed fins are circular fins placed around the stator casing. Similarly to the straight fin analysis, a comparison between aluminum fin and copper fin is presented for each parameter, to see which material will be optimal to use in this application. The first parameter is the temperature distribution along the height of a fin, which can be obtained from the following equation [20]:

$$T(r) := \frac{K1(r_z \cdot z) \cdot I0(r \cdot z) + I1(r_z \cdot z) \cdot K0(r \cdot z)}{K1(r_z \cdot z) \cdot I0(r_w \cdot z) + I1(r_z \cdot z) \cdot K0(r_w \cdot z)} \cdot (T_w - T_f) + T_f , \quad (15)$$

where: Kx, Ix – modified Bessel functions of the appropriate order, z – given in equation (2), r – distance along the radius of the fin [m], rw – motor outer radius [m], r_z – fin radius [m].

Comparison of temperature distribution along the radius of a circular fin between aluminum and copper fins can be found in Figure 8.

The second compared parameter is the heat transfer rate from a single circular fin. It can be obtained from the following equation [20]:

$$Q_o(r) := 2 \cdot \pi \cdot r_w \cdot \sqrt{2 \cdot \alpha \cdot \delta \cdot \lambda} \cdot (T_w - T_f) \cdot \frac{(I1(r \cdot z) \cdot K1(r_w \cdot z) - K1(r \cdot z) \cdot I1(r_w \cdot z))}{I1(r \cdot z) \cdot K0(r_w \cdot z) + K1(r \cdot z) \cdot I0(r_w \cdot z)} , (16)$$

where: α – convection heat transfer coefficient [W/m² K], δ – thickness of the fin [m], λ – thermal conductivity [W/m K], Kx, Ix – modified Bessel functions of the appropriate order, z – given in equation (2), r – fin radius [m], r_w – motor outer radius [m], T_f – temperature of air [°C], T_w – temperature at the base of the fin [°C].

Comparison of heat transfer rate from a fin depending on fin radius between aluminum and copper fins can be found in Figure 9.
Similarly to straight fins, to not increase the fin radius above a point where it would be inefficient to increase it any further from the point of mass and cost of the system, an important parameter to consider is fin efficiency. Fin efficiency for a circular fin can be obtained from equation (4). Comparison of fin efficiency between aluminum and copper circular fins is presented in Figure 10.
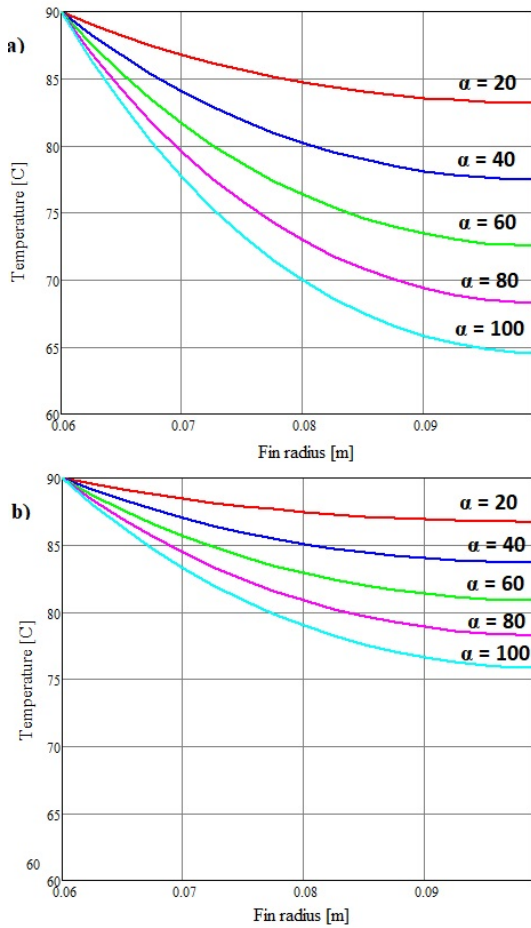
Fig. 8.   Temperature distribution along fin radius with:
           a) aluminum fin, b) copper fin

Fig. 9.   Heat transfer rate from a single fin depending on fin
           radius with: a) aluminum fin, b) copper fin

As we can see in Figure 11, in the case of circular fin, the aluminum fins also have much better mass dependent performance than copper fins. When designing a circular fin based cooling system, aluminum fins should be picked. Going back to f=Figure 10a (aluminum circular fin efficiency), and picking 80% as the acceptable fin efficiency, we can chose 9 cm as the fin radius (so the fin height equals 3 cm). Having the fin radius, the viability of the whole system can now be calculated. The power that has to be dissipated from the stator is equal to 2.8 kW. The heat transfer rate from a single circular fin of 9 cm radius (3 cm height) can be estimated from equation (16). With this data, calculation of the number of circular fins needed for the system, the spacing of the fins on the stator casing and the mass of the whole air cooling system is possible. Those parameters are listed in Table 3.
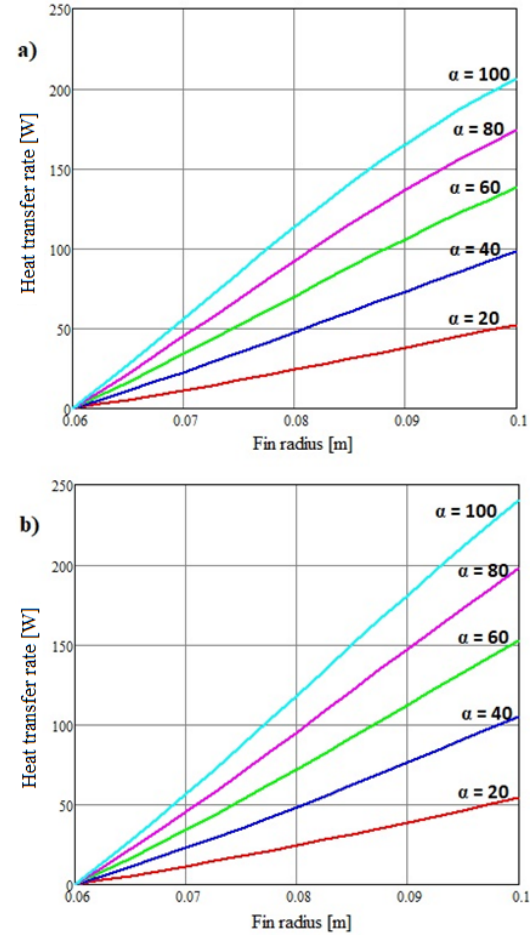
The results shown in Table 3 indicate that the air cooling system based on circular fins is also viable in this case, and it is possible to cool the motor using circular fins. The weight of the cooling system is slightly lower than in case of straight fins, but at a negligible value. The power density of the whole system decreases to a level that is acceptable - 12.4 kW/kg from 14 kW/kg.

Tab. 3.   Parameters of air cooling system with circular fins

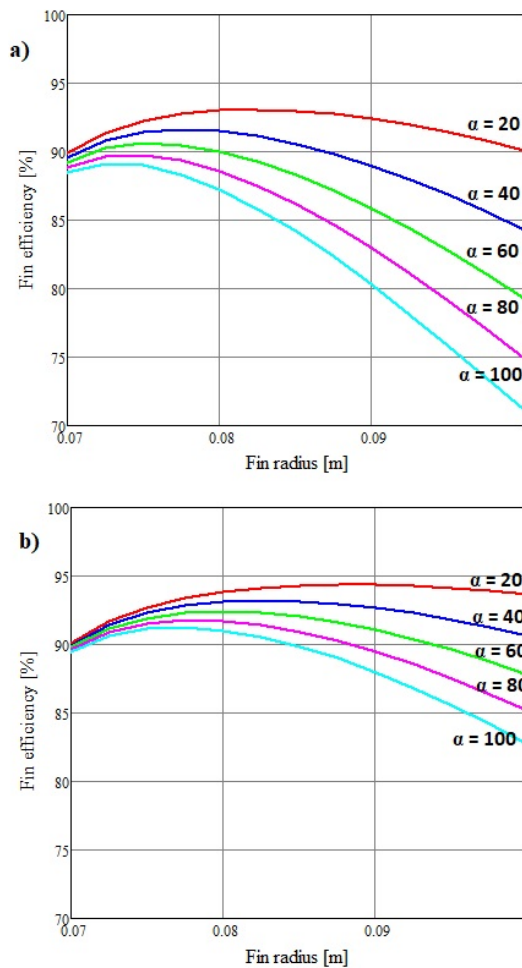|	
Heat loss in the stator	2.8 kW
Heat transfer rate from a single fin	165.2 W
Number of fins needed	17
Spacing of the fins	21 mm
Mass of the cooling system	1.33 kg

Fig. 10. Fin efficiency depending on fin height with:
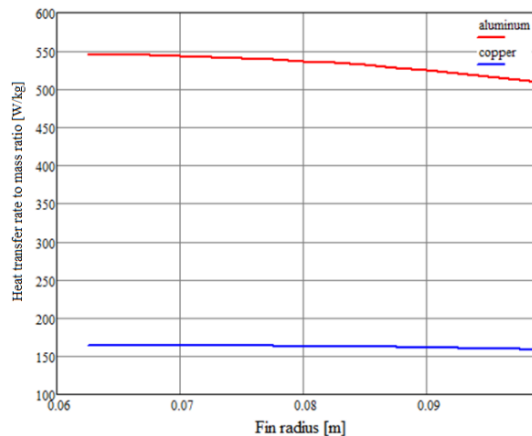a) aluminum fin, b) copper fin

Fig. 11. Heat transfer rate to mass ratio comparison for
aluminum and copper circular fins

5. CONCLUSIONS

This paper presents a computational model of a fin-based air cooling system for electric motors created in Mathcad. The computational model serves as a preliminary step towards design of electric motor

cooling systems, and as a tool for fast and efficient calculation of viability and parameters of air-cooling systems for electric motors that can be used to supplement further work (such as resistance-network thermal models, FEM models and CFD-CHT models), by allowing to narrow down available scenarios and cut down computing time in advanced modelling, by allowing to find viability of the system, optimal fin numbers and fin dimensions beforehand. Calculations presented in this paper show that fin-based air cooling systems can be considered for aviation propulsion based applications, especially for PMSM motors, however it is not the most efficient method and can only work under favorable circumstances (right geometry of the motor, low thermal resistance between the stator winding to the fins located on the stator casing). Further studies show better methods of cooling the high specific power electric motors for aviation, such as direct winding cooling, usage of phase changing materials or oil cooling jackets with special geometry solutions for maximizing the heat transfer area inside the channels..

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Jakub Kapuściński received his M.Sc. degree in Power Engineering from Warsaw University of Technology in 2016. Since 2015 he has been a researcher in the Institute of Aviation in Warsaw, where he currently works as a research associate. He is currently pursuing his Ph.D. degree in cooperation with Warsaw University of Technology, Institute of Aviation and Ohio State University. His scientific interests include issues related to heat transfer, and cooling of electronics and electric equipment.

Roman Domański received his M.Sc. degree (1967), Ph.D. (1976, with honors), and D.Sc. (1988) degree from Warsaw University of Technology. He received Professor degree in 1995. Since 1992 he has been cooperating with Institute of Aviation, where he is currently working as a tenured professor and vice-chairman of the Scientific Council. His scientific interests focus on heat transfer, energy conversion, energy storage, phase-change materials, multi-phase flows, renewable energy sources, cooling of turbine blades, cooling of electronics and electric equipment and thermal barrier coating. He is a member of European Council of Thermal Energy Storage, Thermodynamics and Combustion Committee of the Polish Academy of Sciences, and member of the board of Polish Solar Energy Society, He was a chairman and an organizer of multiple national and international conferences, including FutureStock, Newac and International Nuclear Energy Congress. He is also the author and co-author of over 250 scientific papers and 8 monographs.