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Cooling of 1 MW Electric Motors through Submerged Oil Impinging Jets for Aeronautical Applications

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Abstract: Electrification of aircraft is a very challenging task as the demand for energy and power is high. While the storage and generation of electrical energy are widely studied due to the limited specific energy and specific power of batteries and fuel cells, electric machines (power electronics and motors) which have years of experience in many industrial fields must be improved when applied to aviation: they generally have a high efficiency but the increase in power levels determines significant thermal loads which, unlike internal combustion engines (ICE), cannot be rejected with the exhaust. There is therefore a need for thermal management systems (TMSs) with the main objective of maintaining operating temperatures below the maximum level required by electric machines. Turboprop aircraft, such as the ATR 72 or the Dash 8-Q400, are commonly used for regional transport and are equipped with two gas turbine engines whose combined power is in the order of 4 MW. Electric and hybrid propulsion systems for these aircraft are being studied by several leading commercial aviation industries and start-ups, and the 1MW motor size seems to be the main option as it could be used in different aircraft configurations, particularly those that exploit distributed electric propulsion. With reference to the topics mentioned above, the present work presents the design of a TMS for a high-power motor/generator whose electrical architecture is known. Once integrated with the electrical part, the TMS must allow a weight/power ratio of 14 kW/kg (or 20 kW/kg at peak power) while maintaining the temperature below the limit temperature with reasonable safety margins. Submerged jet oil is the cooling technique here applied with a focus on diathermic oil. Parameters affecting cooling, like rotor speed and filling factor, are analysed with advanced CFD.



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1. Introduction

The electrification of aircraft, as a necessary response to the impact of aviation on global warming, follows two parallel paths, the first of which is based on the replacement of non-propulsive subsystems now driven by mechanical, hydraulic, and pneumatic energy with more efficient electrical subsystems. The second path is intended to replace internal combustion engine (ICE) propulsion systems with hybrid and hopefully all-electric propulsion.

According to [1], carbon dioxide (CO₂) emissions produced by the mobility sector will increase by 80% by the end of 2050. About 20% of the foreseen increase should be due to aviation [2].

The Intergovernmental Panel on Climate Change (IPCC) issued the special report “Summary for Policymakers” on the impacts of global warming of 1.5 °C above pre-industrial levels and related global greenhouse gas emission pathways [3]. Subsequently, the European Commission (EC) issued the report “A Clean Planet for All” [4], underlining the need for decarbonisation by 2050. Subsequently, the European Parliament declared the

climate and environmental emergency [5] and “its commitment to urgently take the concrete action needed to fight and contain this threat before it is too late.”

This commitment translated to the launch of the European Green Deal [6], which fixes the goal to “transform the EU into a fair and prosperous society, with a modern, resource-efficient and competitive economy where there are no net emissions of greenhouse gases in 2050 and where economic growth is decoupled from resource use”, and of the Industrial Strategy for Europe [7] stating that “there should be a special focus on sustainable and smart mobility industries [...] to drive the twin transitions towards climate neutrality and digital leadership, to support Europe’s industrial competitiveness and improve connectivity. This is notably the case for [...] aerospace [...] as well as for alternative fuels and smart and connected mobility”.

Aviation stakeholders committed to reducing global net aviation carbon emissions by 50% by the year 2050 compared to 2005 [8]. This target is narrowed into specific goals in terms of reducing emissions per passenger per kilometre by the long-term roadmap of the Advisory Council for Aviation Research and Innovation in Europe (ACARE) Flightpath 2050 [8], according to which a reduction of 75% for CO₂ and 90% for nitrogen oxide (NO_x) emissions is expected.

These goals can be achieved only by introducing new aircraft configurations and disruptive technologies [9], changing the paradigm of the current aviation market, which is still based on the use of jet engines introduced in the 1950s.

In this scenario, one of the most promising technologies under study is hybrid/electric propulsion for aircraft with powers greater than 250 kW, while for lower-class aircraft, the approach with fuel cells, batteries, and supercapacitors is starting to make technical sense. However, the electrification of flight represents a promising technology that can achieve the goals mentioned above, even if it introduces new challenges, both environmentally and technologically.

Several studies have addressed the problem of flight electrification, focusing in particular on propulsion systems that deal with low values of the specific powers and specific energies of the electrochemical sources for energy storage. In [10], a study is reported in terms of energy budget and weights for two types of missions for a General Aviation (GA) aircraft, a 15-min touch-and-go training flight and a 1-h cruise flight. A hybrid electric propulsion system of an aircraft with approximately 60 kW of maximum power is numerically analysed in [11] with a lumped parameter-based model in which the complete system of the converter, motor, batteries, and propeller is coupled with a traditional internal combustion engine. In [12,13], the management of multiple energy sources (fuel cells, lithium batteries, and super-cap, each with its own charging and discharging rate) allows a 60 kW two-seater aircraft to successfully fly for a 1 h mission, ensuring acceptable weight and centre of gravity displacement.

A key component of the fully electric or hybrid propulsion system is the motor whose power is progressively large depending on the size of the aircraft. As already seen, GA aircraft require moderate motor shaft power but their fleet and their use result in a low environmental impact. Motor power for propeller-driven commuters (<19 passengers) and regional aircraft (about 80 passengers) ranges from 1 MW to 5 MW. Within the overall commercial aviation sector, regional aircraft account for an estimated 3% of CO₂ emissions. In this case, the development of cleaner propulsion is not so important for reducing emissions but for their application on increasingly larger aircraft for which the energy and specific power of electrochemical sources represent the real bottleneck.

High-power electric motors for aviation are currently under development as the requirements are different from those designed for land and maritime applications. A power output of 500 kW for a single motor would imply a number of 6–8 motors in a regional aircraft, which would also have an effect on the overall configuration of the aircraft, including weight, power distribution, and aeroacoustics. The motor size being considered to equip regional aircraft is based on a power output of 1 MW [14]. From a technical point of view, the high thermal loads associated with them make traditional TMSs impractical.

The present article focuses on this technical challenge: designing a new TMS for the thermal control of a MW-class permanent magnet (PM) electric motor. The TMS for these motors requires, in addition to the obvious ability to dissipate thermal loads and maintain a high overall volumetric density, to consider effective, reliable, and low-impact cooling solutions on weight.

From the examination of the most common cooling techniques presented in Section 2, oil jet cooling was considered to have the highest potential for the TMS design, also taking into account the characteristics of the electrical machine to be coupled. Therefore, the topic of oil jets has been covered extensively in Section 3 with an in-depth look at diathermic oil. As regards the electric machine, this is presented in Section 4, where the TMS design is fully developed considering the geometric and weight constraints at the system level. To allow safe operations, it is analysed using a state-of-the-art CFD tool for the identification of areas where the temperature reaches the highest values. In particular, to validate this solution, preliminary CFD analyses were performed using state-of-the-art tools to identify the thermal map of the proposed electrical machine. Based on the results of this preliminary analysis, a TMS has been developed, considering geometric and weight constraints at the system level, to allow safe operations of the proposed PM electrical machine. This study introduces an innovative configuration of submerged impinging oil jets for the direct cooling of critical components in high-power electric motors for aeronautical applications, significantly reducing the size and weight of the cooling system. The article demonstrates how the proposed system can maintain temperatures within material resistance limits even at maximum power, offering thermal optimisation and effective lubrication with minimal increase in friction losses, opening new perspectives for thermal management in aeronautical electric motors.

The activities here summarised have been carried out within the European Union (EU) co-funded ORCHESTRA (Optimised Electric Network Architectures and Systems for More-Electric Aircraft) Project [15], which aims to design new technologies that allow a 10% efficiency increase and 25% weight reduction of Electric Power Systems (EPSs) compared to the state-of-the-art.

2. Review of Cooling Technologies

Over the last decade, electric motors have taken on an increasingly significant and central role in the transport sector, from automotive to the first concepts of power applications for aviation. They can offer many advantages: high efficiency, high specific power, low weight, relatively small dimensions, and high ease of use. In addition to this, they are an excellent resource for reducing polluting emissions. However, despite the positive aspects, electric motors, as seen above, are affected by numerous power losses. This logically does not only concern motors and/or generators in the aeronautical sector but, in general, any electric motor.

It is pretty straightforward, however, that the most disadvantaged are those used for propulsion of aircraft, as these are required to have increasingly higher speeds and specific powers, with the addition of being compact. As a result, this causes motors to have smaller dimensions and even higher temperatures, pushing them further and further to the limits of their capabilities. Consequently, to try to stem these problems, increasingly efficient cooling systems are required in motors [16]. Today, it is possible to find numerous strategies to remove heat and reduce losses, especially in automotive companies, where improvements need to be made [17].

Most cooling systems are based on the physical principles of conduction, particularly convection [18,19]. Convective heat exchange is the first resource used for motor cooling. Consequently, finding strategies based solely on the conductive process is not easy. The latter methodology is based on transferring molecular energy through the molecules' vibrational, rotational, and translational motions. But it does not exploit the macroscopic movement of matter in any way.

This section briefly overviews TMS technologies, starting with systems with higher Technology Readiness Level (TRL [20]) and moving on to lower ones [16].

- Liquid cooling systems: These systems use water or onboard coolants, such as engine oil and fuel, to remove heat from the equipment and cool it down. They are well known and widely used in several applications [17] (e.g., in the automotive field Figure 1a), and their TRL is higher than 7.
- Forced air cooling systems: These systems adopt the same principle as liquid cooling systems [17], using air as a coolant to transfer heat from the source to the external ambient [21]. As for liquid cooling systems, air cooling systems (see Figure 1b) have been applied to existing piston aircraft [22], even if it is less effective during low-speed operations (i.e., ground operations, take-off, holding, and all the other high-power/low-speed operations). This type of cooling is commonly used in electric motors, where air is circulated over the surface of the motor to dissipate heat. Studies explore factors influencing cooling effectiveness, such as fan design, airflow, and heatsink configurations. While ensuring construction simplicity and reduced weights, this approach cannot easily guarantee heat dissipation in the reference volumes. TRL is higher than 7.

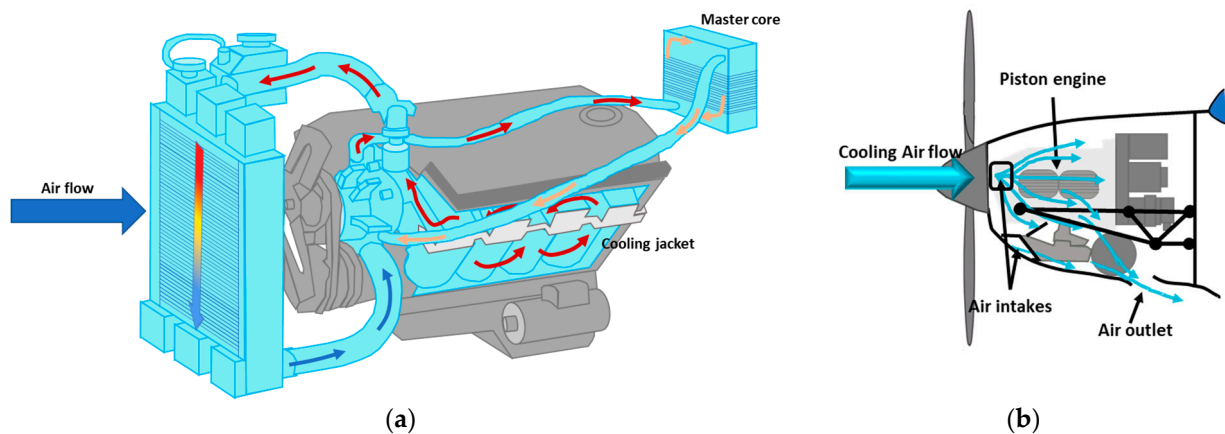


Figure 1. Typical scheme of cooling: (a) liquid cooling system for an automotive engine; (b) air cooling system for a piston engine.

- Skin heat exchangers: This type of system uses ambient air as the cold side of the cooling system [22], while a fluid transporting waste heat from the heat source is the hot side of the system (see Figure 2). Such a system has a TRL higher than 7.
- Passive systems: Such cooling systems use fluid moving in a closed case to cool down the equipment. There are three different typologies of passive systems [21]:
 - Heat pipes: Refrigeration fluid is heated by the heat source, changing phase (from liquid to vapour), thus absorbing heat. The vapour moves from the hot to the cold zone, condensing and releasing heat outside;
 - Thermosyphons are similar to heat pipes but use gravity and natural convection;
 - Vapour chambers are flat heat pipes that transfer heat in 3-D.
- Pump two-phase system: This is a hybrid cooling loop system consisting of an evaporator, a mechanical pump, a reservoir/condenser, and connecting pipes, as can be seen in the schematic of Figure 3a [23]. Such a system has already been implemented on existing aircraft, assuring a TRL higher than 7.
- Phase Change Materials (PCMs) are gaining importance as passive cooling solutions for electric motors. PCMs absorb and release heat during phase transitions, maintaining a stable temperature within the motor [24]. Studies investigate suitable types of PCMs, their encapsulation methods, and their integration within motor systems. Figure 3b shows the crystalline configuration of a generic PCM in the heat absorption and release phase, with the temperature curves as a function of time [25].

PCMs are very effective for lowering the temperature locally, but mainly for transient conditions. Various problems [26,27] have been detected regarding the material to be used, assuring a melting temperature of 450 K, suitable for our applications, and above all, the maximum amount of heat they can dissipate. The TRL of PCM ranges between 4 and 6.

- Absorption refrigerator: Such systems, whose TRL ranges between 4 and 5, are driven using low-quality heat, as seen from [28]. Coolant is inserted in the evaporator, which absorbs heat from the component to cool down and changes its phase into vapour (see Figure 4a). This vapour enters the absorber at low pressure, reacting with another fluid (e.g., water) to form a compound at higher pressure without any compressor. A pump directs this mixture to a generator, which is heated by a low-quality heat source (e.g., a hot system). This heat addition induces the separation of water vapour from the coolant vapour. A proper filter separates these two vapours: water vapour is sent back to the absorber, while the coolant goes to the condenser where it condenses, moving into the liquid phase.
- Vortex tube: This system, shown in Figure 4b, also known as the Ranque–Hilsch tube, is a mechanical device that separates compressed gas of homogeneous temperature in a stream hotter (up to +200 °C) than the incoming flow and another cooler one (up to −50 °C) simultaneously [29]. Thanks to the geometry of the tube, which includes a control valve, an outlet, and a spin chamber, the TRL is lower than 4.
- Thermoelectric effects: Some materials show a coupled thermal/electric behaviour, enabling direct conversion between electrical and thermal energy. There are two types of thermoelectric effects in Figure 5a:
 - Peltier effect: The thermoelectric material heats up or cools down at an electrified junction [30];
 - Seebeck effect: The thermoelectric material converts heat directly to electricity.
 Thermoelectric materials have poor power density and efficiency. Their TRL is between 3 and 4, although some thermoelectric generators are already used in the aerospace sector [31].
- Thermionic energy converter: This system comprises a heated surface and a collector separated by a vacuum (see Figure 5b). The heated surface emits electrons flowing towards the cold surface, producing an electromotive force that can be used to absorb heat [32]. The TRL of this TMS is lower than 3.
- Caloric materials: This type of materials generates cooling effects by the influence of magnetic (magnetocaloric materials), electric (electrocaloric materials), or mechanical (mechanocaloric materials) forces, using a reversible transformation. The TRL is between 2 and 3. In Figure 6a, a schematic of the principle of the magnetocaloric effect is shown.
- Joule–Thomson effect: If a highly compressed gas suddenly expands, the pressure reduction lowers rapidly (almost immediately) the temperature of the gas, which can be used as a heat extractor from a heat source. TMSs based on such an effect are generally utilised on coolers to allow cryogenic performances. The system consists of a fluid isolated in a volume (see Figure 6b), cooled using a proper heat sink, before entering an isolated chamber, where there is a nozzle for sudden gas expansion through a valve into the isolated chamber itself. The expansion rapidly reduces fluid temperature, creating a very cold volume, which cools down the hot source. The TRL ranges between 1 and 3.
- Cryo-cooling systems: In Figure 7a, this type of system uses cryo-refrigerants to remove large amounts of heat, reaching cryogenic temperatures [33]. To remove this heat, these systems can be based on both boiling or sublimation phenomena at low temperatures (depending on the coolant used: liquid or solid). A standard system is the so-called Reverse Bryton Cycle Cryocooler (RBCC). The TRL is between 1 and 3.
- Thermoacoustic heat engines: These systems (see Figure 7b) are composed of a resonator filled with a working fluid and heat exchangers in a tube. Their geometry is

studied to convert heat into small air vibrations (i.e., acoustic power). The TRL is between 1 and 3.

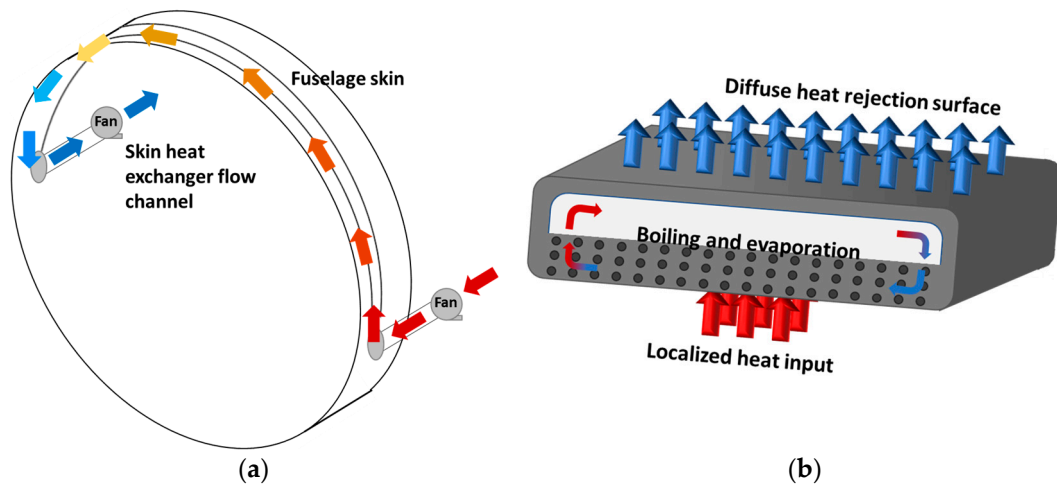


Figure 2. (a) Skin heat exchangers and (b) vapour chamber scheme.

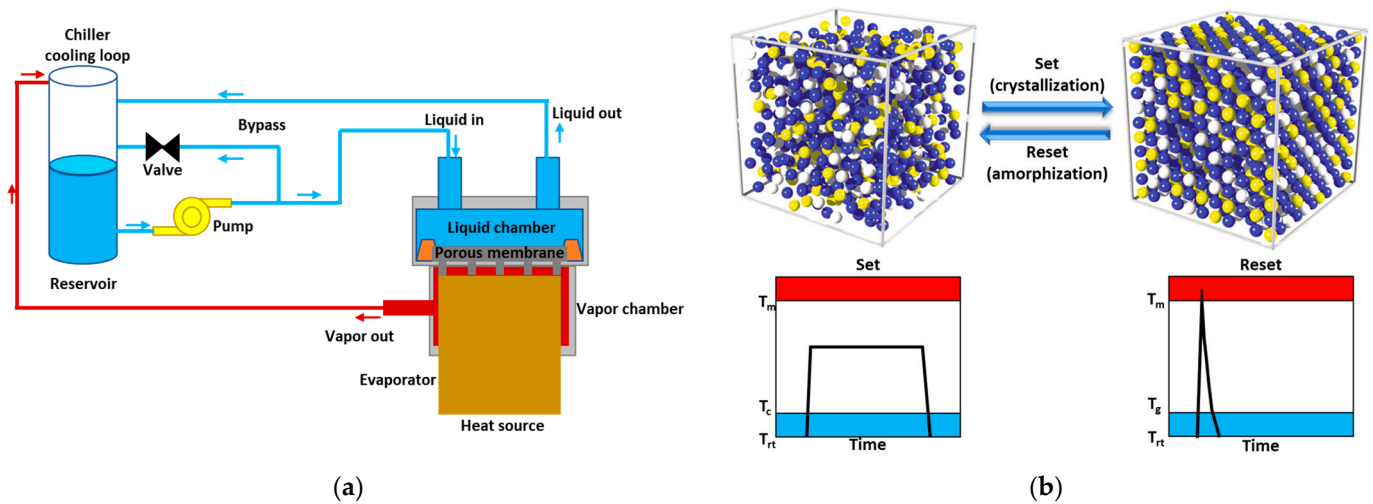


Figure 3. (a) Pump two-phase system scheme; (b) PCM as TMS principle.

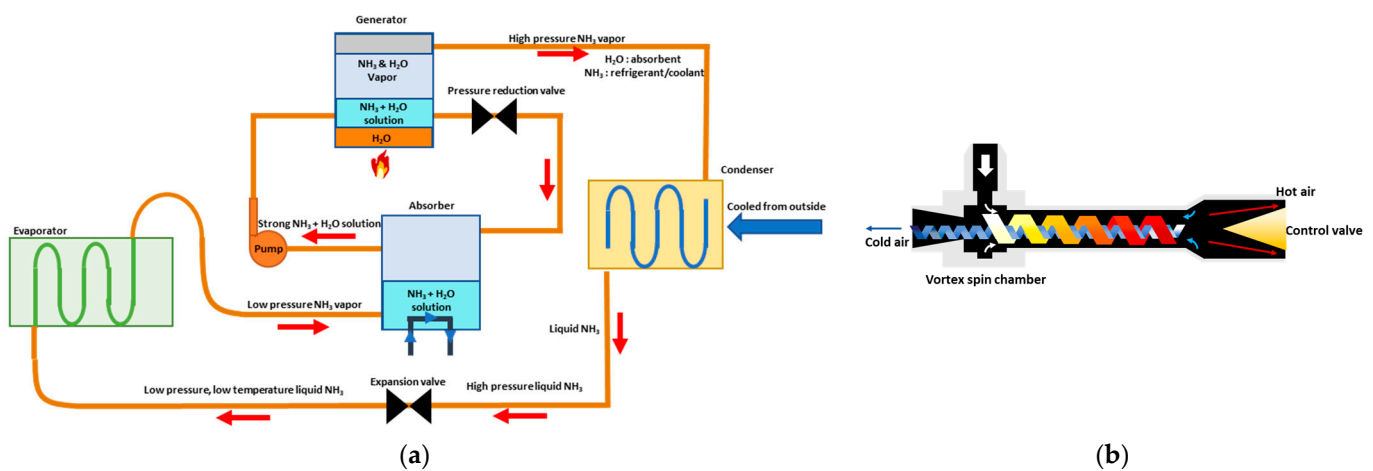


Figure 4. (a) Vapour absorption refrigerator cycle; (b) vortex tube scheme.

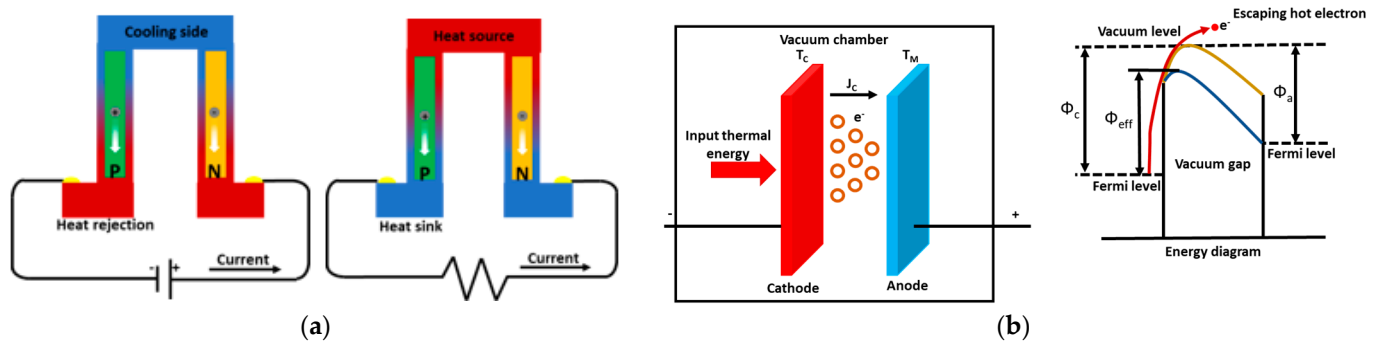


Figure 5. (a) Peltier principle (left) and Seebeck effect (right); (b) thermionic energy converter.

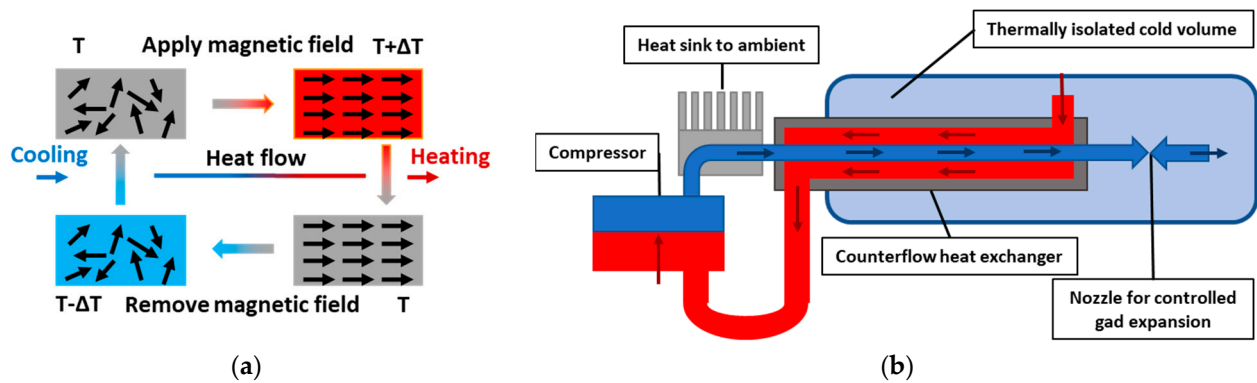


Figure 6. (a) Magnetocaloric principle; (b) Joule–Thomson effect.

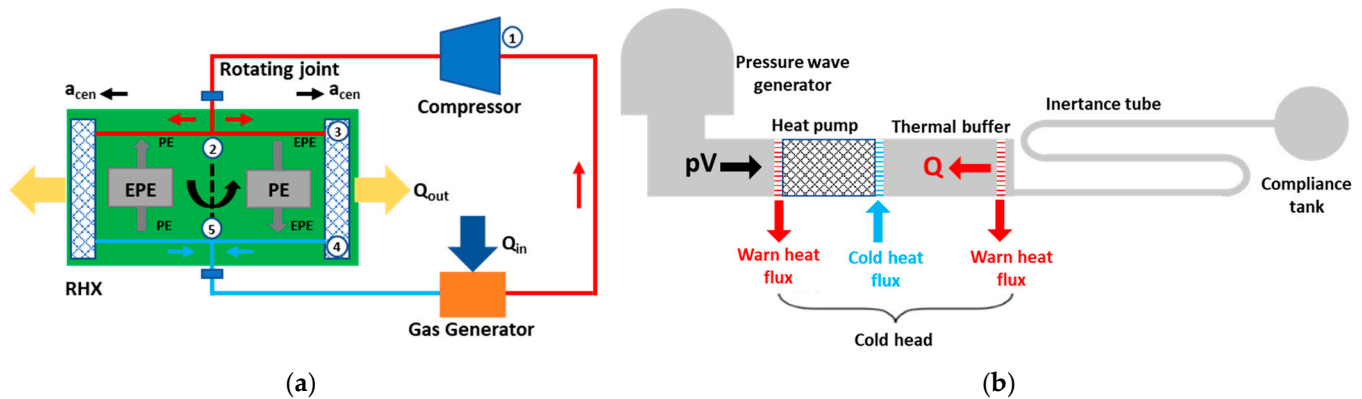


Figure 7. (a) Centrifugal RBCC; (b) thermoacoustic heat engine.

3. Cooling Methodologies for High Power Electric Motor

This paragraph describes the type of PM electric machine (either motor or generator) that must be used for applications on regional aircraft of the 1 MW class and which guarantees high values of power and specific energy. The typical losses of an electric motor (EM) or electric generator (EG) are responsible for the increase in weight of the overall system or the degradation in performance. In the case of conventional propulsion systems, although the total heat loads are higher, this heat is eliminated mainly at the exhaust. This heat must be managed locally and eliminated in electric aircraft with a TMS. The temperature limits of the materials that make up a permanent magnet motor, generally below 480 K, represent a challenge for the design of the TMS, which, in addition to being efficient in terms of overall power dissipated, avoids hot spots in all operating conditions, must guarantee a reduced weight, such that the sum of the EM and the associated TMS gives a power-to-weight ratio greater than 10 kW/kg, the minimum for aeronautical applications.

Subsequently, the primary cooling techniques for machines of this power level will be reported, as well as the typical values of the convective heat transfer coefficients valid for the pre-design of the TMS on each subcomponent of the motor; finally, a focus on the cooling technique of impinging jets submerged with diathermic oil, which was conceived in this research work, will be presented.

3.1. Description of Oil Cooling Techniques for the 1 MW PM Electrical Machine

Electric motors are critical components in various applications, particularly for transport vehicles, of which aircraft impose the most stringent efficiency and reliability requirements. This literature review summarises the primary research and advances in thermal management techniques for electric motors. Many studies focus on developing accurate thermal models to understand the mechanisms of heat generation and dissipation in electric motors. CFD simulations and analytical models are widely used to analyse the thermal behaviour of motors. These models help identify critical points and optimise the design of cooling systems. This study is for designing and optimising a heat management and disposal system for an ultra-compact electric machine of approximately 1 MW. Improving heat transfer within electric motors is another crucial area of research. Some studies explore using heat pipes, radiators, and microchannel heat sinks to improve heat dissipation efficiency. Advanced materials like graphene and carbon nanotubes are being studied for their high thermal conductivity properties. It is therefore necessary to assess those improving techniques, which, despite being simple at a construction level, can guarantee the control of the maximum temperature in all operating conditions. One of the first approaches examined was cooling via an external cooling jacket on the stator [34]. This reasonably simple solution was unsuitable for the thermal powers involved, even when using fluids other than water as the cooling fluid unless too high flow rates were used and couplings with internal phase change materials were considered. The problem is that the hottest points are near the teeth and the end winding, far from the cooling jacket, and different heat passage paths follow, as shown in Figure 8.

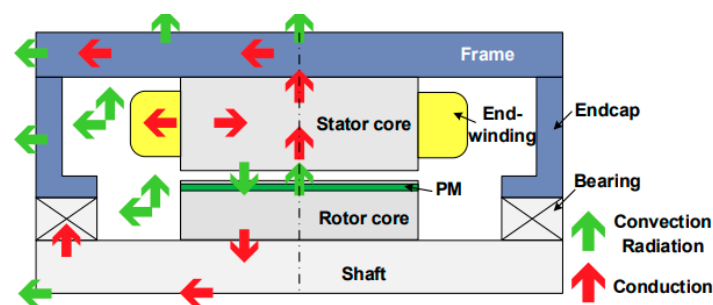


Figure 8. Schematic example of heat paths inside an electric machine.

In this study, different numbers of windings on the cooling jacket from 4 to 10 are examined, from which it is possible to estimate the work of the convective heat exchange coefficient and derive maximum values of the thermal power exchanged. Table 1 shows schematically the correlations of the dimensionless heat transfer coefficient (Nusselt Number—Nu) for the various cooling techniques mentioned.

Table 2 shows the fluid dynamic correlations for calculating the rotor and stator resistance coefficients for various conditions and the different areas of the electric machine. From these values, it is then possible to estimate the thermal resistance of each macro component of the motor and, finally, the overall one.

Table 1. Heat transfer coefficient vs. cooling system.

Natural Cooling		Ref.
Cylinder housing	$Nu = 0.525(Gr \cdot Pr)^{0.25}; Gr \cdot Pr < 10^9$ $Nu = 0.129(Gr \cdot Pr)^{0.33}; Gr \cdot Pr > 10^9$	[35]
Finned housing	$Nu = 5.22 \cdot 10^{-3} (Gr \cdot Pr \cdot NS/L)^{0.57} (S/L)^{0.412} (H/L)^{0.656};$ $\text{for } 10^6 < Gr \cdot Pr \cdot NS/L < 2.5 \cdot 10^7$ $Nu = 2.78 \cdot 10^{-3} (Gr \cdot Pr \cdot NS/L)^{0.57} (S/L)^{0.412} (H/L)^{0.656};$ $\text{for } 2.5 \cdot 10^7 < Gr \cdot Pr \cdot NS/L < 1.5 \cdot 10^8$	[36]
Forced Cooling		
Cylinder housing	$Nu = 0.664 Re^{0.5} \cdot Pr^{0.33}; (Re < 5 \cdot 10^5; 0.6 < Pr < 50)$ $Nu = (0.037 Re^{0.8} - 871) \cdot Pr^{0.33}; (Re > 5 \cdot 10^5)$	[35]
Finned housing	$Nu = 0.03 Re^{0.8} \left\{ 1 - 0.23 \left(L/S \right)^{0.5} (L - H/L)^{1.5}; \text{For Turbulent flow} \right\}$	[37]
Housing jacket	$Nu = 3.66 + \frac{0.668 Re \cdot Pr \cdot \frac{D}{L}}{\{1 + 0.04(Re \cdot Pr \cdot D/L)^{0.667}\}}; Re > 2300$ $Nu = 0.125 \cdot m \cdot (Re - 1000) \cdot \frac{Pr}{\{1 + 4.49 m^{0.5} (Pr^{0.667} - 1)\}}; 3000 < Re < 5 \cdot 10^5$	[38]
Rotational hallow shaft	$Nu = 0.019 Re^{0.93} + 8.51 \cdot 10^{-6} Re_r^{1.45}$ $(0 < Re < 3 \cdot 10^4); (1.6 \cdot 10^3 < Re_r < 2.77 \cdot 10^5)$ $Nu = 0.019 Re^{0.93} + 2.85 \cdot 10^{-4} Re_r^{1.19}; (Re_r > 2.77 \cdot 10^5)$	[39]
Spray cooling	$Nu = Pr^{0.4} \left\{ 0.785 Re^{0.5} \cdot \frac{L}{D} \cdot A_r + 0.0257 Re^{0.83} \cdot L/L^* \cdot (1 - A_r) \right\}$ $A_r = \frac{\pi(1.9d)^2}{L^2}; L^* = \frac{0.5 \cdot (1 + \sqrt{2}) L - 3.8d}{2}$	[40]

Table 2. Flow resistance coefficient by the literature.

Friction Loss Coefficient		Ref.
Stationary pipe $k = f_s \cdot L/D$	$f_s = \begin{cases} \frac{64}{Re}, Re < 2300 \\ \frac{0.316}{Re^{0.25}}, 4000 < Re < 1 \cdot 10^4 \end{cases}$	[41]
Rotating shaft $k = f_r \cdot L/D$	$\frac{f_r}{f_s} = \begin{cases} 1; V_r/V < 0.35 \\ 0.579 \left(V_r/V \right)^{-0.52}; 0.35 \leq V_r/V \leq 0.8 \\ 0.47 \left(V_r/V \right)^{-1.42}; 0.8 < V_r/V < 1.2 \end{cases}$	[42]
Air gap $k = f_r \cdot L/D$	$\frac{f_r}{f_s} = \left\{ 1.765 \cdot \left(\frac{Re_r}{2 Re} \right)^2 \right\}^{0.38}$	[42]
Rotor ducts $k = f_r \cdot L/D$	$\frac{f_r}{f_s} = \begin{cases} 0.5 Re_r^{0.16} Re^{-0.03}; 900 < Re < 9880 \\ 0.842 Re_r^{0.023} Re^{0.002}; Re > 9880 \end{cases}$	[43]
Sudden Expansion, Contraction Loss Coefficient		
Stationary pipe	$k = (1 - A_i/A_0)^2$	[41]
Entrance of air-gap	$k = 0.1 \left(V_r/V \right)^2 - 0.06 \left(\frac{V_r}{V} \right); V_r/V > 1$	[42]
Entrance of rotor ducts	$k = 0.234 \left(V_r/V \right)^2 - 0.043 \left(\frac{V_r}{V} \right); V_r/V > 0.5$	[43]

3.2. Submerged Jets Oil Cooling Techniques

In general, the effectiveness of a cooling system greatly depends on the type of cooling liquid used. Coolants should have high thermal capacities to absorb heat without significant temperature changes. They should also have high thermal conductivity and high thermal stability (low freezing and high boiling points). Water is one of the most commonly used

liquids because it has high thermal capacity. However, ethylene glycol and water (50/50) are often used [44]. The studies [45–47] investigate the cooling enhancement using confined impinging jets, with and without water-based nanofluids. From them it is possible to derive the Nusselt correlations as a function of the geometric parameters of the jets (z and D) and of the Reynolds number of the jets.

Oil, especially engine oil or Automatic Transmission Fluid (ATF), is another coolant commonly used for TMSs due to its many advantages from both thermodynamic and physical points of view. It has a thermal conductance similar to water and is an excellent electrical insulator with low dielectric constant and high electrical resistivity. In addition, it is chemically stable, non-toxic, and non-flammable. These aspects make oil a valid alternative to water or various mixtures, especially if the coolant is to be placed in direct contact with internal motor parts such as stator heads or windings. In fact, because of oil insulating capabilities, it is unnecessary to worry about whether electrical or magnetic effects may be created or about all the insulated circuits the electric motor is composed of.

In addition, a secondary but not insignificant benefit is that by taking advantage of motor oil or ATF, there is no need for a second pumping system to circulate the liquid since the one already present is used [48]. In recent years, by exploiting the properties of oil, more and more new solutions have been patented: oil jet, oil spray, and oil immersion cooling. Table 3 shows the thermophysical properties of the primary cooling fluids.

Table 3. Typical cooling fluids are used in electric motors.

Fluid	Thermal Conductivity λ [W/mK]	Specific Heat c_p [kJ/kgK]	Density ρ [kg/m ³]	Kinematic Viscosity ν [m ² /s]
Air (@ sea level)	0.0264	1.006	1.1174	1.57×10^{-5}
Water	0.56	4.217	1000	1.78×10^{-6}
EGW 50/50	0.37	3.0	1088	7.81×10^{-6}
EGW 60/40	0.34	3.2	1100	1.36×10^{-5}
Engine Oil	0.147	1.796	899	4.28×10^{-3}
Mobil jet oil	0.149	1.926	1014	1.88×10^{-4}
Paratherm LR	0.1532	1.925	778	3.43×10^{-6}
PGW 50/50	0.35	3.5	1050	1.90×10^{-5}
PGW 60/40	0.28	3.25	1057	3.31×10^{-5}
Dynalene HF-LO	0.1126	2.019	778	3.20×10^{-6}
Brayco Micronic	0.1344	1.897	835	1.35×10^{-5}
RF 245 FA	0.014	0.975	10.51	1.03×10^{-5}
Silicone KF 96	0.15	1.5	1000	8.00×10^{-5}
Skydrol 500-4	0.1317	1.75	1000	3.50×10^{-6}

In oil-immersion cooling, the motor is cooled by running oil through its internal parts in a set path. This method successfully cools the motor entirely and more efficiently as there is direct contact with the various internal components. In addition, due to the properties of oil, the magnetic fields on which the operating principles of the electric motor are based are not disturbed or disrupted. An example of a motor that utilises such a cooling methodology is the YASA P400R [44].

Cooling methods using oil jets and oil sprays are still under development, and many experiments on them have been made over the years, as well as fluid dynamic studies. However, to date, such a type of TMS is still not so widely used. This is because there are many physical parameters and phenomena to take into account, and up to now, the majority of the studies have focused on testing the effectiveness of such solutions on simple test surfaces rather than on real stator windings. Both methods base their operation on the direct injection of oil onto the internal parts of the motor, particularly the stator heads. In this way, direct liquid-surface contact occurs, and more effective heat transfer is assured than in the solutions described above. Successively, the oil falls by gravity to the lower part

of the motor, where it is collected using a particular outlet. Finally, it will be cooled and re-circulated for use again.

This type of system is considered one of the few capable of operating variably: when loads are low, or the use is short, the system delivers less oil or no oil at all; vice versa, for high loads or prolonged uses, it is expected to provide a quantity proportional to the workload. As can be seen, depending on the fluid and thus on the various properties (see Tables 1 and 3), the convective heat transfer coefficient values will tend to vary, as well as the heat transfer effectiveness. In addition, we must also consider that convection can be natural or forced. In fact, in the case of natural convection, the cooling fluid is the air, for which typical values of h are $(5 \div 10) \text{ W/m}^2\text{K}$. In the forced case, the air has h value of $(10 \div 300) \text{ W/m}^2\text{K}$. On the other hand, liquids can reach values of $(50 \div 20,000) \text{ W/m}^2\text{K}$ [48]. This study focused on oil jets directly conveyed at stator windings, which appears to be a highly efficient and flexible method of transferring heat: a liquid flow directed against a surface can absorb, very efficiently, large amounts of heat energy.

Compared with conventional systems, where the flow is confined in a circuit and not in direct contact, this solution provides heat transfer coefficients up to three times higher for a given maximum flow velocity.

This is because of turbulence generated by shear stresses between the jet and the circulating air, which is carried into the boundary layer of the surface. In addition, the flow required by an oil injection device can be up to two orders of magnitude less than that required in systems with cooling circuits. Unlike the spray case, the oil injection mechanism relies on nozzles with significantly lower pressure.

Figure 9a shows a dense liquid column impinging on the solid surface at the nozzle's exit.

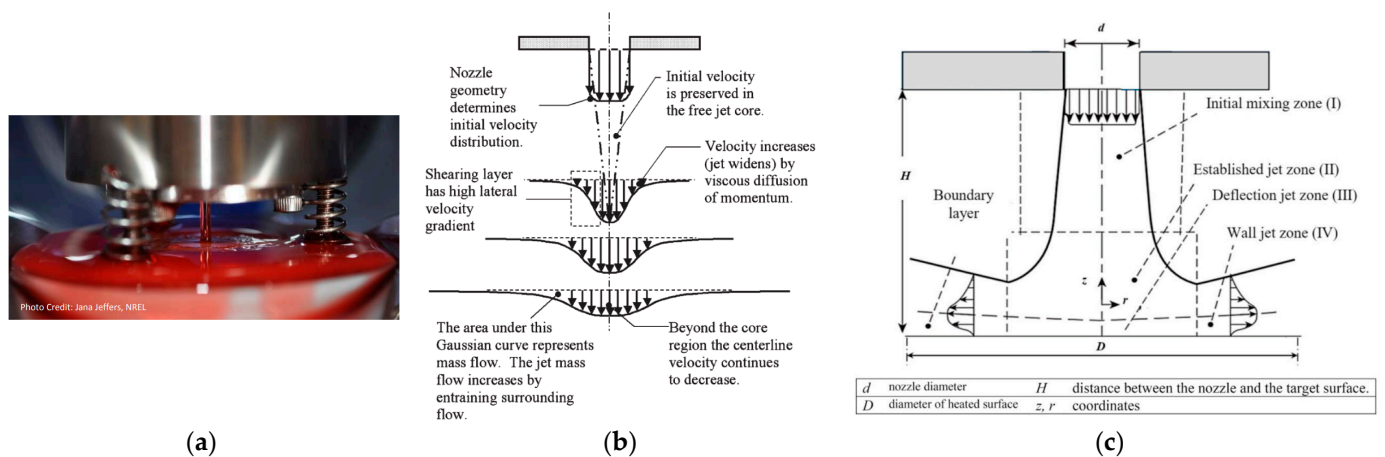


Figure 9. (a) Image of laboratory tests on oil jets Reprinted/adapted with permission from Ref. [49]. Copyright © 2015 by ASME; (b) development of the fluid jet in the free jet region; (c) sliding region along the solid wall.

Three distinct regions can be distinguished from experimental studies when a liquid jet hits a surface. The first is that of the free jet, which develops instantaneously at the nozzle outlet and remains throughout the injection process. This region can, in turn, be subdivided into further sub-regions in which the flow takes on ever-changing and consequential characteristic features Figure 9b. Thus, after exiting the nozzle bore, the first section is characterised by a velocity, temperature, and turbulence profile dependent on the upstream flow and, therefore, on the shape of the nozzle [50,51]. Finally, the oil jet impinges on the opposite surface, and it is deflected outward as illustrated in Figure 9c.

For example, a cylindrical nozzle's flow will have a parabolic velocity profile, including moderate turbulence. In contrast, a thin, flat nozzle will create a flow with a flat velocity profile and low turbulence. If the velocity profile presents spatial gradients, these give rise to shear stresses on fluid 'packets' present in the lateral edges of the jet.

This transfer (diffusion by viscosity) momentum outward from the jet, attracting additional fluid and increasing the mass flow of the jet. During such a process, the jet loses energy, and the velocity profile widens in spatial extension, decreasing its modulus along the edges of the jet itself. The “core” of the liquid column, in general, is not affected by momentum transfer and thus forms a central zone with a higher total pressure than the rest, with a velocity along the nozzle axis (U_m) almost equal to the velocity exiting the borehole (U_n). The extreme points of this zone have velocities equal to $0.95 \cdot U_n$ and thus allow it to be distinguished from the rest of the liquid column. It may occur that shear stresses also expand towards the “core” before the jet reaches the surface. The decay of the “core” itself thus begins.

Typically, this occurs at distances from the exit bore ranging from four to eight times the nozzle’s diameter (or width). Should the jet decay, the velocity in the central part decreases, and its profile becomes similar to a Gaussian curve. We are then in conditions of a fully developed velocity profile [52,53].

The second region is the impact region, where the interaction between the jet and the surface produces a strong flow deceleration. The fluid begins, thus, to flow in a direction parallel to the solid, forming a liquid layer that grows along the dimensions of the impact surface. This liquid “film” represents the third region in Figure 9c [50,52].

When the liquid first impacts the surface, it stagnates in a small region of the impact zone. It remains in this condition for a given period and only then begins to expand along the surface. This time frame, called “residence time” (t^*), can vary from a fraction of a second to a few minutes, depending on the conditions under which the experiments are carried out. At a time instant less than t^* , the surface temperature decreases slowly and almost at a constant rate, although there is a sudden drop at the moment of impact. At instants later than t^* , the liquid begins to expand, wetting the surface, consequently decreasing the temperature faster. The stagnation zone typically extends 12 times the diameter of the nozzle (in the case of circular jets) [50].

Droplets break away from the liquid layer when the turbulent oil jet impacts the solid wall. This phenomenon is called “splattering” (Figure 10a) and reduces the efficiency of the heat transfer process due to liquid loss.

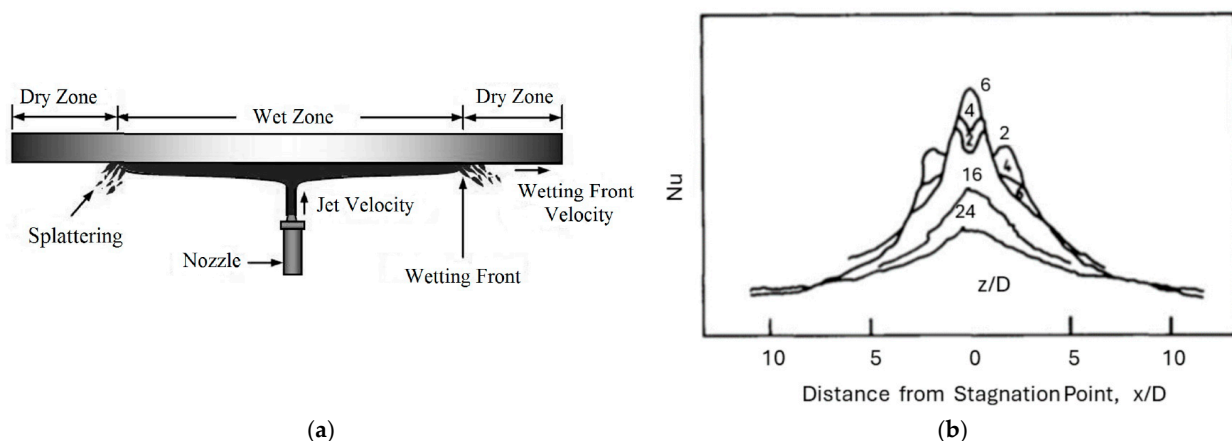


Figure 10. (a) Representation of the impact of an oil jet resulting in the splattering phenomenon [50]; (b) radial variation of the heat transfer coefficient, the curves are parameterized based on the z/D ratio [51].

The intensity of this event depends on the Weber number of the jet and the surface tension of the liquid (σ):

$$We_l = \frac{u^2 D \rho_l}{\sigma}$$

In case the regime is laminar, no “splattering” occurs [50]. The heat transfer of a jet striking a surface is expressed by the Nusselt number (Nu) and is a complex function of many parameters:

$$Nu = f\left(Re, Pr, \frac{z}{D}, \frac{x}{D}, \text{nozzle shape}\right) = \frac{hD}{\lambda}$$

where (z/D) is the dimensionless distance between the nozzle and surface and (x/D) is the dimensionless distance from the stagnation point. In addition, nozzle geometry, turbulence, and jet velocity also have significant effects. By studying this trend, it is possible to derive the value of the convective heat transfer coefficient of the jets.

Some analytical studies carried out in laminar jets have shown that:

$$Nu \propto U_m^{1/2}$$

This suggests that the Nusselt number should remain approximately constant in the “core” and decrease downstream. Furthermore, again, by observing Nu , it was seen that in the stagnation zone, along the jet axis and thus in the “core”, there is a point where heat exchange is maximum. This point also coincides with the maximum turbulence intensity. As we move away from the “core”, the heat transfer rate decreases due to ever-lower liquid velocities. However, this decrease is stopped for high turbulence levels, and an increase is shown [53]. This is until the drop in velocity is compensated for by the increase in turbulence. Figure 10b shows the radial variation of the heat transfer coefficient, obtained by [54], by measuring the Nusselt number of jets from a cylindrical nozzle. As can be seen, there is a local maximum at $x/D = 0.5$ for all injections with nozzle-surface distance $4 < z/D < 6$, while a second maximum, smaller than the previous one, occurs for values of $z/D \leq 4$, at x/D equal to 2.

The first peak is due to an acceleration of the radial velocity in the stagnation zone. As for the second peak, the only explanation suggested can be that at $1 \leq x/D \leq 2$ there is a transition from laminar to turbulent flow. In fact, at distances $x/D = 2$, vortices with a toroidal shape have been found by [55] to hit the surface. At greater distances x/D , the radial velocity decreases, thus lowering the efficiency of heat exchange. The previously mentioned vortices, however, are only present for distances $z/D \leq 4$. Above this value, these vortices tend to break down into smaller-scale vortices penetrating the “core.” This is why nozzle-surface distances more significant than four have a single maximum and a bell-shaped heat transfer coefficient distribution.

This cannot be applied in the case of fully developed jets or jets with large nozzle-surface distances, as these cause turbulent flow in the stagnation zone in the boundary layer of the wall. In [56], it has been shown that jets with $z/D = 50$ exhibit important ring, helical, and double-helix vortical structures [53]. The angle of impact also plays an essential role in heat transfer. Taking a more recent study into account, we can see how much the angle of the nozzle matters concerning the solid surface [57]. This study showed how the angle of inclination of the nozzle, about the impact surface, affects heat transfer (Figure 11). Specifically, keeping the liquid flow rate fixed, the inclination angle varied from 45° , 60° , and 90° , respectively.

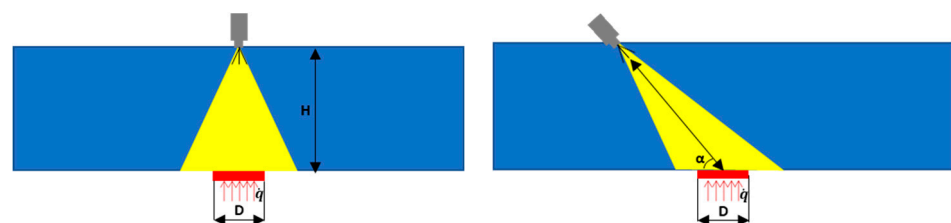


Figure 11. Representation of different jet angles impinging heated surface.

It has been found that the Nusselt number decreases when the angle of inclination becomes less than 90° . In fact, at smaller angles, only a part of the jet impinges on the surface, and the stagnation zone and wall flow development are very different from the classical 90° injection case [57]. For the 90° injection angle, there is the highest Nu, and the target surface isotherms are more uniform and symmetric.

All these data are typically obtained on flat geometries and limited dimensions; the purpose of the reported study is to understand, with the help of CFD, how these behaviours change when we work on nonplanar geometries in a confined environment, especially when rotation of the geometries is introduced.

4. Design and CFD Analysis of Motor Cooling System

Research shows that the temperature increase in the electric motor negatively affects the performance of the electric aircraft. In general, an increase in operating temperature of 30°C leads to a reduction in torque of up to 50%. Additionally, increased failure rates and a shortened life cycle reduce the overall efficiency and performance of the electric propulsion systems.

4.1. PM Electrical Machine Short Description

A complex hybrid propulsion system for a regional aircraft is being developed as part of the EU-funded ORCHESTRA project. Two electric machines are under development, a generator to convert the mechanical energy of a gas turbine into electrical energy and a motor to drive the propellers for distributed propulsion. Since the two machines have different requirements, mainly high power for the generators and low speed and high torque for the motors, one of the strategies outlined is to proceed with two separate designs. A 1MW generator was conceived by the University of Nottingham, the leading partner of the ORCHESTRA project, who provided the architecture as illustrated in Figure 12 and calculated the thermal loads reported in Table 4.

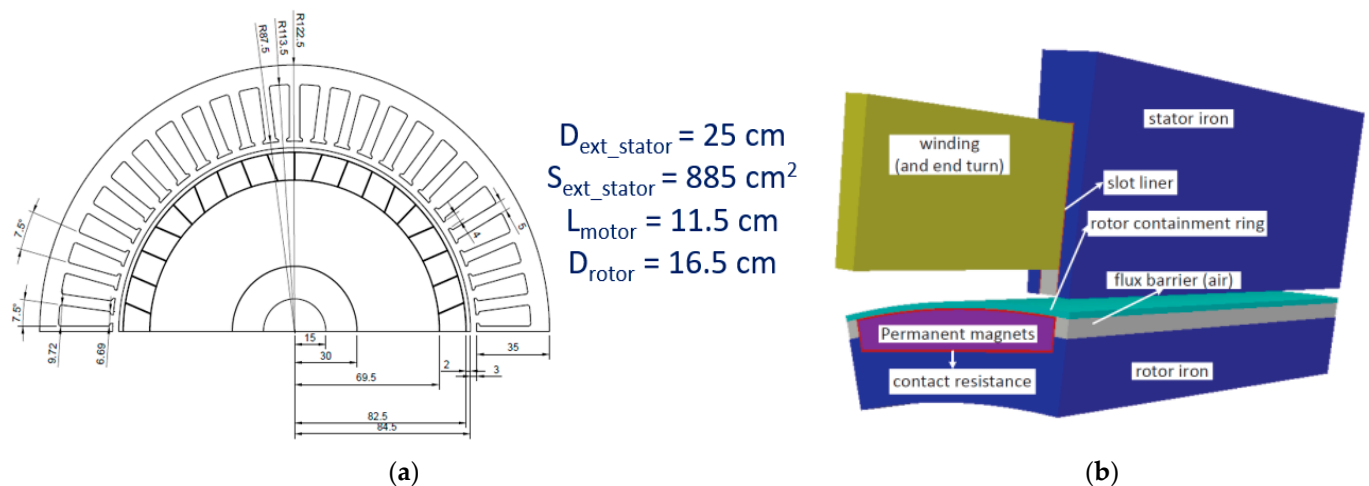


Figure 12. Electric machine design: (a) electrical machine dimensions under investigation; (b) PM motor segment components.

The generator is a 900 kW–20k rpm PM electric machine with 48 poles. It is made up of Recoma 33 material used for the Samarium Cobalt permanent magnets, NKM (slot liner insulation material) for the slot liner that keeps the surface magnets connected, Recoma 33 material used for the Samarium Cobalt permanent magnets, Vacoflux Cobalt Iron for the stator, copper with Litz type treatment for windings, steel for the crankshaft, and aluminium for the external case.

Table 4. Thermal loads of current electric machines.

Id.	Description	Value
Q1 [kW]	Thermal power—Winding	14.3
Q2 [kW]	Thermal Power—Stator	2.02
Q3 [kW]	Stator teeth Thermal Power	1.75
A1 [m ²]	Surface Winding	0.0002
A2 [m ²]	Surface Stator	0.000273
A3 [m ²]	Surface teeth	0.001449
V1 [m ³]	Volume winding	3.14×10^{-5}
V2 [m ³]	Volume stator	0.000027
q1 [MW/m ³]	Thermal power density winding	11.034
q2 [MW/m ³]	Thermal power density Stator	1.340
q3 [MW/m ³]	Thermal power density stator teeth	0.0252

4.2. TMS Design

By performing the thermal analysis of the electric machine for which a simple surface finning was used for the power density of Table 4, the maximum temperature was detected at the stator windings, with temperatures rising above 800 K. The design of the TMS for the current application started with collecting global data on different cooling technologies' power and global heat exchange coefficients. From this, it was observed that the power involved and the limited spaces were incompatible with classic cooling systems such as cooling jackets. It was therefore decided to focus on direct cooling with thermal oil, which remains liquid even at temperatures above 180 °C. Even if thermal oil has a specific heat and a density slightly lower than water, thanks to the potential of direct cooling and the effect of the impinging jets [58], a more efficient heat exchange is obtained locally and globally. Both the review of cooling techniques and the in-depth analysis of oil jet cooling have been discussed in Sections 2 and 3, respectively.

The design of the 1MW generator TMS was conducted to guarantee a maximum temperature of 523 K, preserve the compactness of the electric machine and the TMS group, and seek the maximum specific power for the latter.

With these objectives, a TMS was developed that exploited the cooling of oil jets. In essence, as sketched in Figure 13, the oil jets are directed mainly towards the ends of the windings with oil subsequently collected by gravity on the bottom of the case to be recirculated through a closed cooling circuit.

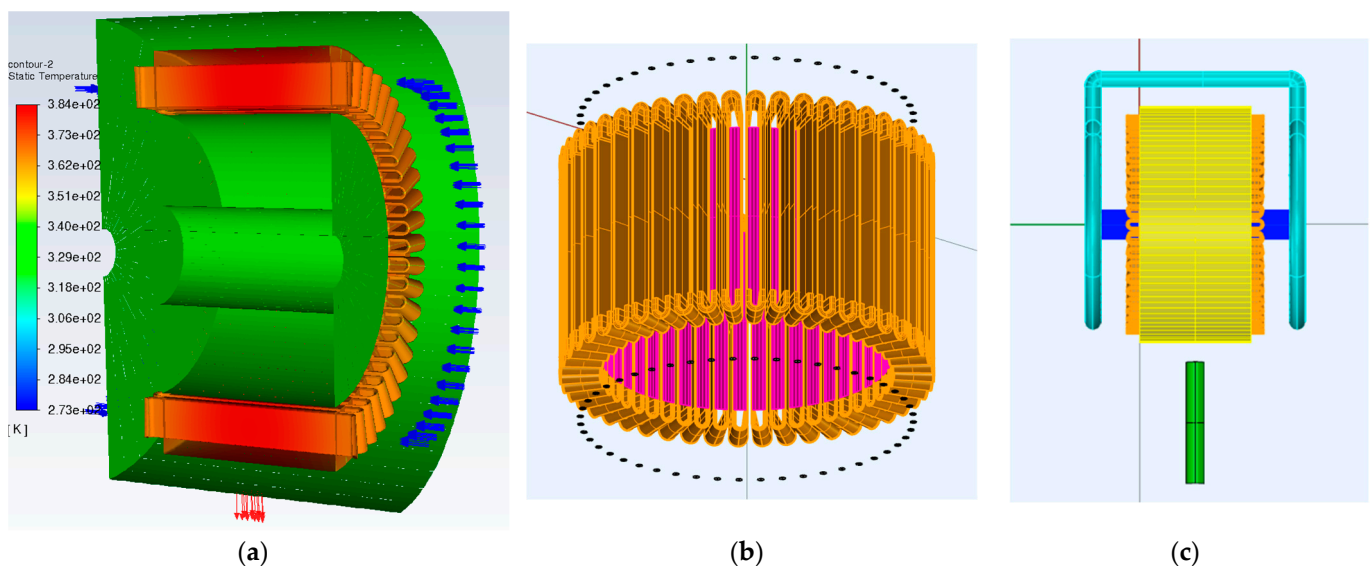


Figure 13. Oil cooling architecture schematic: (a) configuration selected for the electric machine; (b) oil cooling by more orifice; (c) detailed circular ring needed for oil jets.

Specifically, based on the geometry provided, an external case with a net internal diameter of 285 mm was assumed to have a gap with the external part of the stator of 17.5 mm on the radius (see Figure 14). This cavity allows the oil to flow and effect additional direct cooling, similar to a cooling jacket. The oil is conveyed inside the case by means of two circular pipes of the same diameter connected by a duct placed above the external case since the delivery pump is connected on only one side. The cooling oil that enters will then exit from the exit duct located in the centre of the bottom external case, as illustrated in Figure 15. The holes for oil impinging are located on a 16 mm diameter pipeline with a circular axis on both sides. They are positioned so that the oil jets go against the end windings; by taking into account the 48 motor poles, there are in total 96 holes equally spaced as in Figure 16a. Two hole sizes, 2 and 4 mm, were considered for CFD analyses. To obtain an exit velocity of at least 0.5 m/s, a mass flow rate of between 9 and 36 L/min on the 96 holes is required. Lower speed values would negatively affect heat transport. Furthermore, setting the distance between the orifice and the end-winding cusp to approximately 26 mm, as shown in Figure 16b, results in an H/D_{hole} ratio of 12 and 6 according to the two hole sizes. These values are optimal so that the liquid jets optimise the heat exchange, as seen in Section 3.2. Finally, resin to electrically isolate the copper windings from the stator is considered a design option.

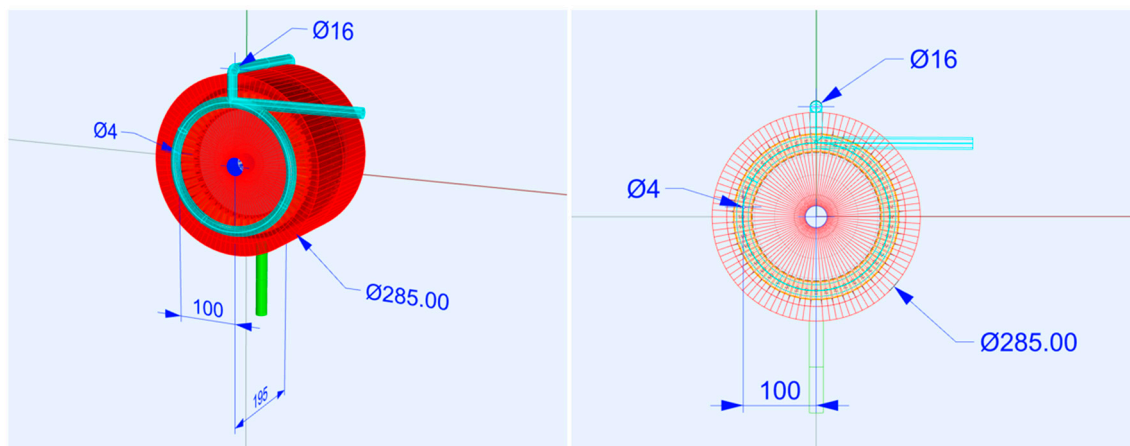


Figure 14. TMS oil cooling design.

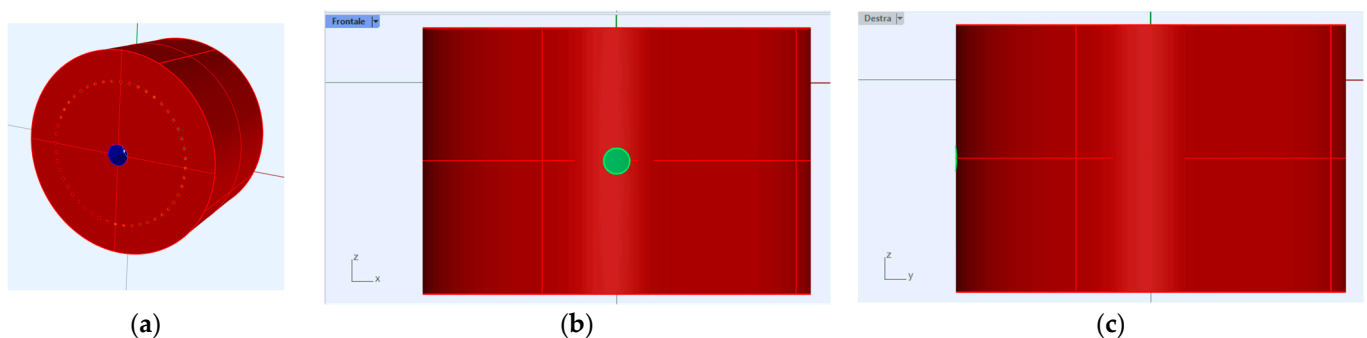


Figure 15. CAD of EM's outer case: (a) isometric view; bottom with exit hole (b) and top (c) views.

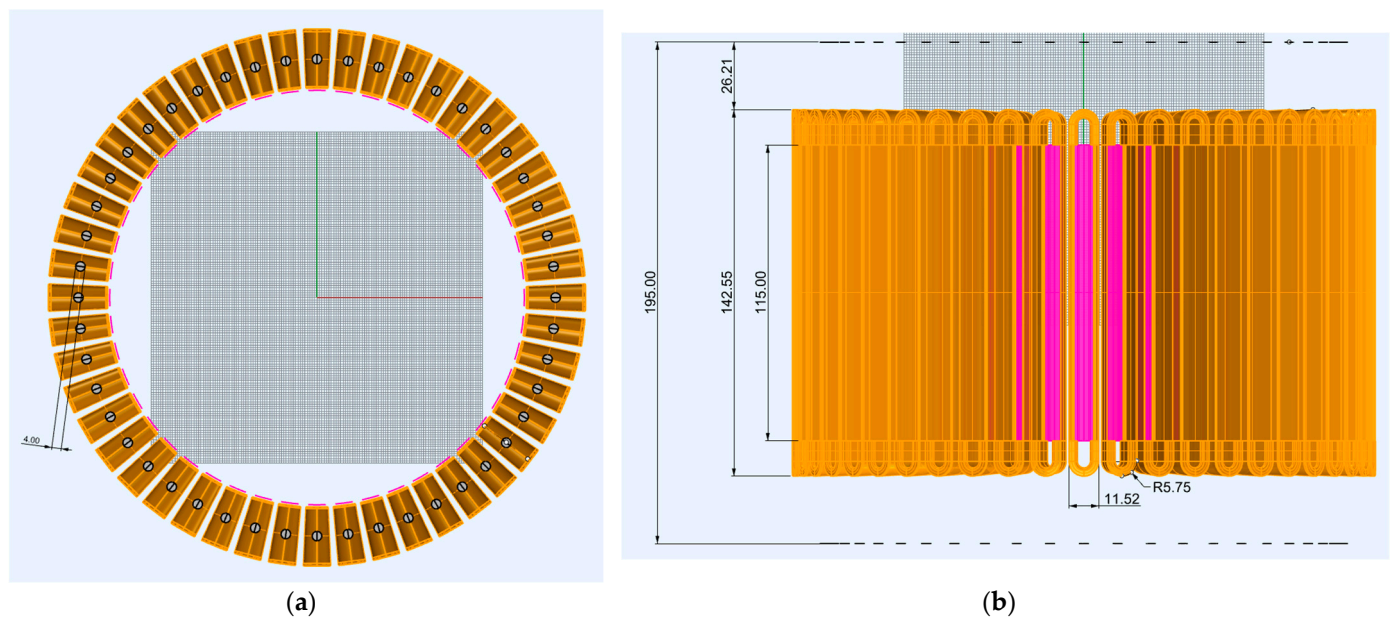


Figure 16. CAD inside EM: (a) end-winding with oil jet holes position; (b) top view of the stator with axial dimensions and distance to oil holes.

4.3. Simulations Setup

Once the TMS was preliminarily designed, verification and analysis were performed using Ansys Fluent[®] as a CFD tool. The CFD simulations conducted are in steady-state conditions, with the properties of the solid materials and oil constant with temperature to reduce the computational cost. In these calculations, the interface between fluid dynamics and structure has been considered from a thermal point of view; the energy equation is activated to consider the effects, among other things, of heating due to friction in the air gap, while the turbulence model used is the $k-\epsilon$ [59–61] with standard wall functions. Finally, CFD analyses do not consider the mechanical connections between the external case and the generator structure, assuming they would not interfere with cooling. Full three-dimensional analyses were performed considering the effects of both gravity and non-symmetry of the system (for example, the presence of a single exit duct on the bottom of the case and rotation). Further details about the simulation setup are provided in the following article subsections.

4.3.1. Governing Equations and Mesh Details

TMS analysis of the investigated electric machine requires the resolution of a conjugated conductive–convective problem.

Reynolds Averaged Navier–Stokes equations and turbulence models are applied to describe the fluid flow evolution using a coupled implicit approach, adopting the $k-\epsilon$ two-equations model for the turbulence fluctuations. In fluid regions, in particular, the transport equations have the following dimensional form:

$$\begin{aligned}
 \text{Mass equation} & \quad \frac{\partial}{\partial t}(\rho) + \nabla \cdot (\rho \vec{v}) = 0 \\
 \text{Momentum equation} & \quad \frac{\partial}{\partial t}(\rho \vec{v}) + \nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla p + \nabla \cdot (\bar{\tau}) + \rho \vec{g} + \vec{S} \\
 \text{Energy equation} & \quad \frac{\partial}{\partial t}(\rho h) + \nabla \cdot (\vec{v} \rho h) = \nabla \cdot (\kappa_f \nabla T) + S_h
 \end{aligned}$$

The above equations are discretised using a finite volume formulation and solved by the FLUENT[®] COUPLED algorithm associated with a well-assessed Algebraic Multigrid model. In conjunction with this, 2nd order spatial numerical upwind schemes have been adopted to discretise the spatial domain. All grids were generated considering the require-

ments that RANS calculations need near the wall. In solid regions, the energy transport equation used by FLUENT has the following dimensional form:

$$\frac{\partial}{\partial t}(\rho_s h) + \nabla \cdot (\vec{v} \rho_s h) = \nabla \cdot (\kappa_s \nabla T) + S_h$$

where:

ρ : density [kg/m³]

h : sensible enthalpy, $\int_{T_{ref}}^T c_p dT$ [kJ/kgK]

k : thermal conductivity [W/mK]

T : temperature

S_h : volumetric heat source.

Regarding the grid generation, five prismatic layers were generated on each wall in the fluid domain. To improve the accuracy of the heat transfer calculation in the air gap, the mesh in the radial direction was built with ten layers, five on the rotor wall and five on the stator wall, and two or three layers of tetrahedrons. The solid domains, like windings and parts in ferromagnetic materials, were discretised using tetrahedrons, refining the mesh in the zones where a greater temperature gradient was forecast. Figure 17a,b shows the views of the mesh at $z = \text{const.}$ and at $x = \text{const.}$, for which clustering is noticeable near the air gap and the end windings where higher temperature gradients are expected. Figures 18 and 19 show close-ups of the stator, teeth, windings and end-winding.

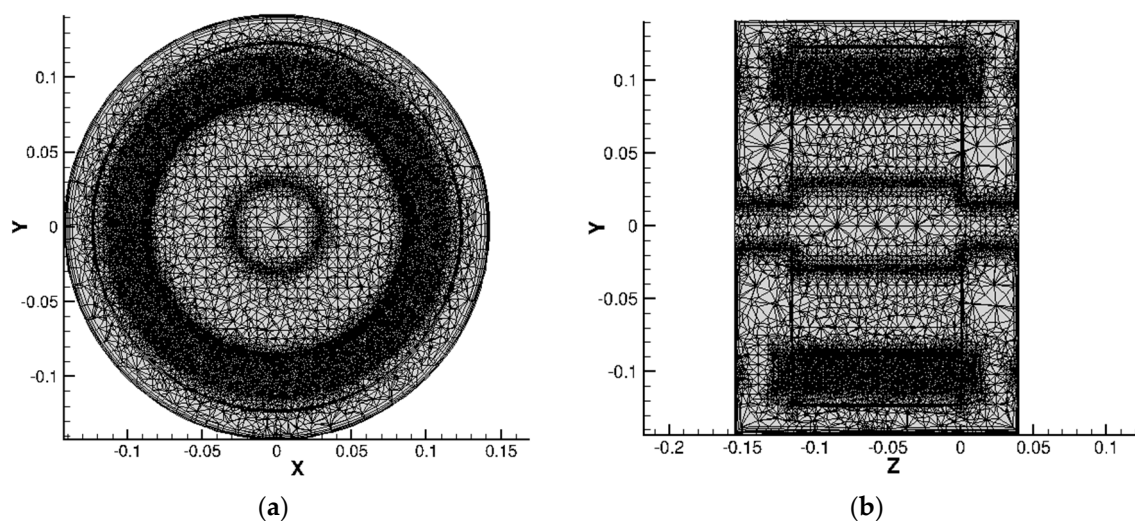


Figure 17. Details of the mesh, in the case with the presence of resin between the windings and $d = 2$ mm; (a) section at $z = \text{const.}$; (b) section at $x = \text{const.}$

Tables 5 and 6 summarise boundary conditions and mesh characteristics. A mesh independence study was conducted by creating three different levels of mesh refinement based on 8, 15, and 30 million cells, respectively. The main interest of the CFD analysis was to identify the maximum temperature reached in the solid zone. It was observed that the maximum temperature computed using the grid levels did not exceed 2%. The intermediate level was considered a good compromise for considering the computational effort and the accuracy of the results simultaneously.

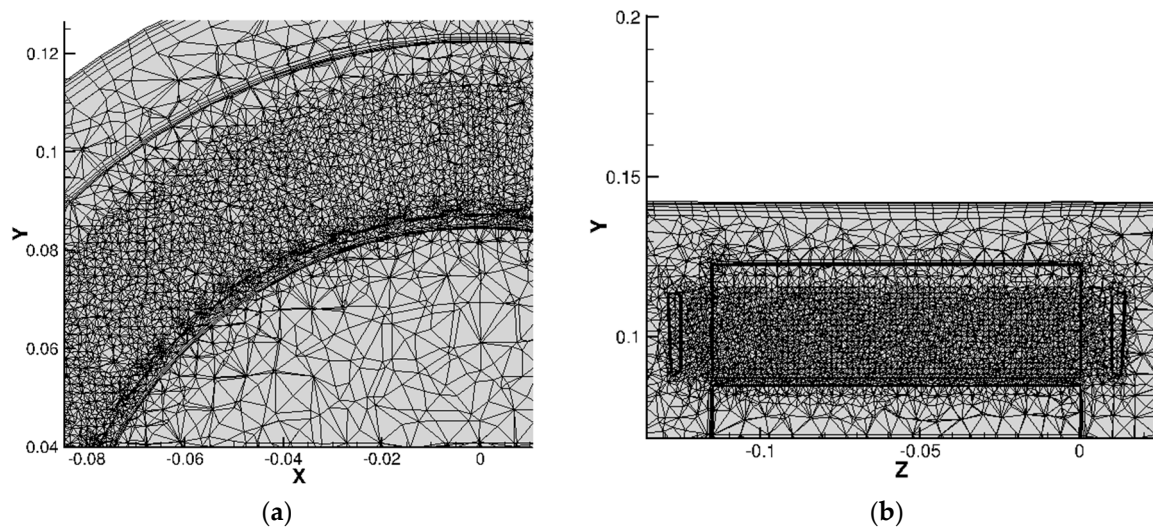


Figure 18. Mesh details: (a) stator; (b) windings along z-axis.

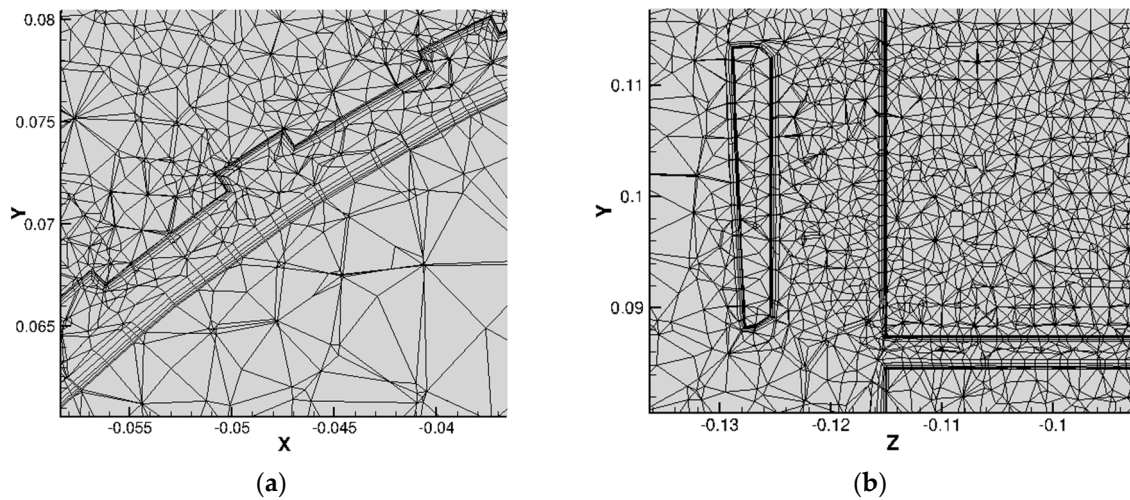


Figure 19. Mesh details: (a) teeth and air-gap; (b) end-windings.

Table 5. Reviewed boundary condition of CFD cases.

Boundary Conditions	Zones
MASS FLOW INLET	inlet holes
OUTFLOW	exit hole
MOVING WALL	external rotor surfaces
WALL/no Slip	the remainder
Constant heat flux	teeth
Uniform heat power source (W/m^3)	stator yoke, end-winding, copper

Table 6. Number of cells for various test cases.

Id.	Configuration Status 1	Configuration Status 2	Nr. Cells
Mesh 1	d = 2 mm	Without Epoxy Resin	20M
Mesh 2	d = 2 mm	With Epoxy Resin	15M
Mesh 3	d = 4 mm	Without Epoxy Resin	20M
Mesh 4	d = 4 mm	With Epoxy Resin	15M

4.3.2. Thermophysical Properties

The positioning of the various solid materials is schematically represented in Figure 20a; the details of the electrical machine with 48 slots and the filling of these slots with copper wires are included in Figure 20b. Table 7 shows the thermophysical properties of the materials which were used in setting up the CFD model.

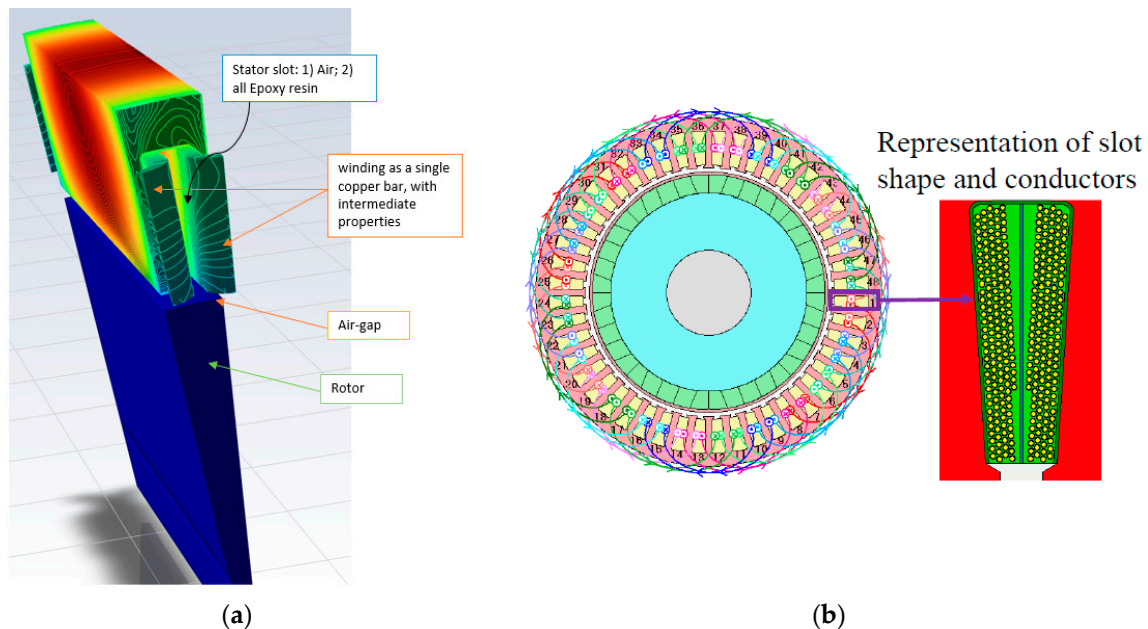


Figure 20. CFD domain and materials association: (a) detail of a stator slot; (b) representation of slot shape and conductor.

Table 7. Thermophysical properties of solid materials of the electric machine.

Solid Material Property	Value
<i>NKM—slot liner</i>	
Density [kg/m^3]	395
Cp [$\text{J}/(\text{kg}\cdot\text{K})$ —Specific heat]	1200
k [$\text{W}/(\text{m}\cdot\text{K})$ —Thermal conductivity]	0.14
<i>Recoma33—magnet</i>	
Density [kg/m^3]	8300
Cp [$\text{J}/(\text{kg}\cdot\text{K})$ —Specific heat]	350
k [$\text{W}/(\text{m}\cdot\text{K})$ —Thermal conductivity]	10
<i>Resin—insulator</i>	
Density [kg/m^3]	2500
Cp [$\text{J}/(\text{kg}\cdot\text{K})$ —Specific heat]	784
k [$\text{W}/(\text{m}\cdot\text{K})$ —Thermal conductivity]	0.8
<i>Steel—shaft</i>	
Density [kg/m^3]	8030
Cp [$\text{J}/(\text{kg}\cdot\text{K})$ —Specific heat]	502.48
k [$\text{W}/(\text{m}\cdot\text{K})$ —Thermal conductivity]	16.27
<i>Aluminum—external case</i>	
Density [kg/m^3]	2719
Cp [$\text{J}/(\text{kg}\cdot\text{K})$ —Specific heat]	871
k [$\text{W}/(\text{m}\cdot\text{K})$ —Thermal conductivity]	202.4

Table 7. Cont.

Solid Material Property	Value
<i>Vacoflux48 [CoFe]—stator</i>	
Density [kg/m ³]	8120
Cp [J/(kg·K)]—Specific heat	430
k [W/(m·K)]—Thermal conductivity	33
<i>Litz—copper wire</i>	
Density [kg/m ³]	8890
Cp [J/(kg·K)]—Specific heat	385
k [W/(m·K)]—Thermal conductivity	395

4.3.3. Test Matrix for CFD Analyses

Once the TMS was preliminarily designed, a test matrix was identified to analyse the influence of different parameters on the cooling used. These parameters and the ranges explored are listed below:

- Oil flow rate (0.5–1 m/s);
- Holes number;
- Holes position;
- Hole diameter (2–4 mm);
- Position and diameter of the oil outlet from the machine;
- Rotation speed (0 rpm < n < 20,000 rpm);
- Oil inlet temperature (37–70 °C).

Table 8 shows the test matrix and the setup of the boundary conditions for various configurations; in particular, the diameter of the holes, the rotation speed of the motor, the oil inlet speed, and the influence of the epoxy resin, which fills or does not fill all the stator slots, isolating them from the windings, were varied. The resin-free configuration is to be understood as using a thin insulating layer on the windings but still allowing the presence of a cavity in the stator slot, which should positively influence cooling. This configuration is intended as an improvement even if it requires greater processing and more expensive resins.

Table 8. Test matrix of CFD analyses.

Id. Case	d [mm]	Resin between Winding	Rotor Speed [rpm]	T _{inlet} [K]	V _{oil} [m/s]
Run 1	2	no	0	325	0.5
Run 2	2	no	20,000	325	0.5
Run 3	2	no	0	325	1.0
Run 4	2	yes	20,000	325	0.5
Run 5	2	yes	20,000	325	1.0
Run 5_bis	2	yes	0	325	1.0
Run 6	4	no	0	325	0.5
Run 7	4	no	20,000	325	0.5
Run 8	4	no	0	325	1.0
Run 9	4	yes	20,000	325	0.5
Run 10	4	yes	20,000	325	1.0
Run 11	4	no	20,000	325	1.0
Run 11_bis	4	no	10,000	325	1.0

4.4. CFD Results

Figures 21 and 22 show the temperature maps velocity fields, wall shear stresses, and heat transfer coefficient on the computational domain for the Run 4 and Run 10 cases, respectively. These two cases are to be considered real configurations, with differences in the flow rate of oil disposed of and the diameter of the oil orifices. In both cases, the maximum temperature is significantly lower than the design limit (523 K), which also allows us to have a safety margin on the maximum temperature, given some simplifications of the analyses, such as the thermal power generated by the bearings.

As can be seen, the maximum temperature is reached on the windings; on the front areas, there is maximum cooling efficiency, while on the midsection, there is the maximum operating temperature because the oil has difficulties entering that area. Nonetheless, this value is comfortably below the limit. The same effect is seen on the rotor Figure 21c due to heat transport from the stator windings via the air gap. In Figure 21e, however, a front section taken at $z/2$ of the solid domain of the motor is shown: a reduction in the temperature on the teeth near the air gap due to the Taylor convective motions generated by the rotation of the rotor is visible. The same trends can be noticed in Figures 21b and 22b.

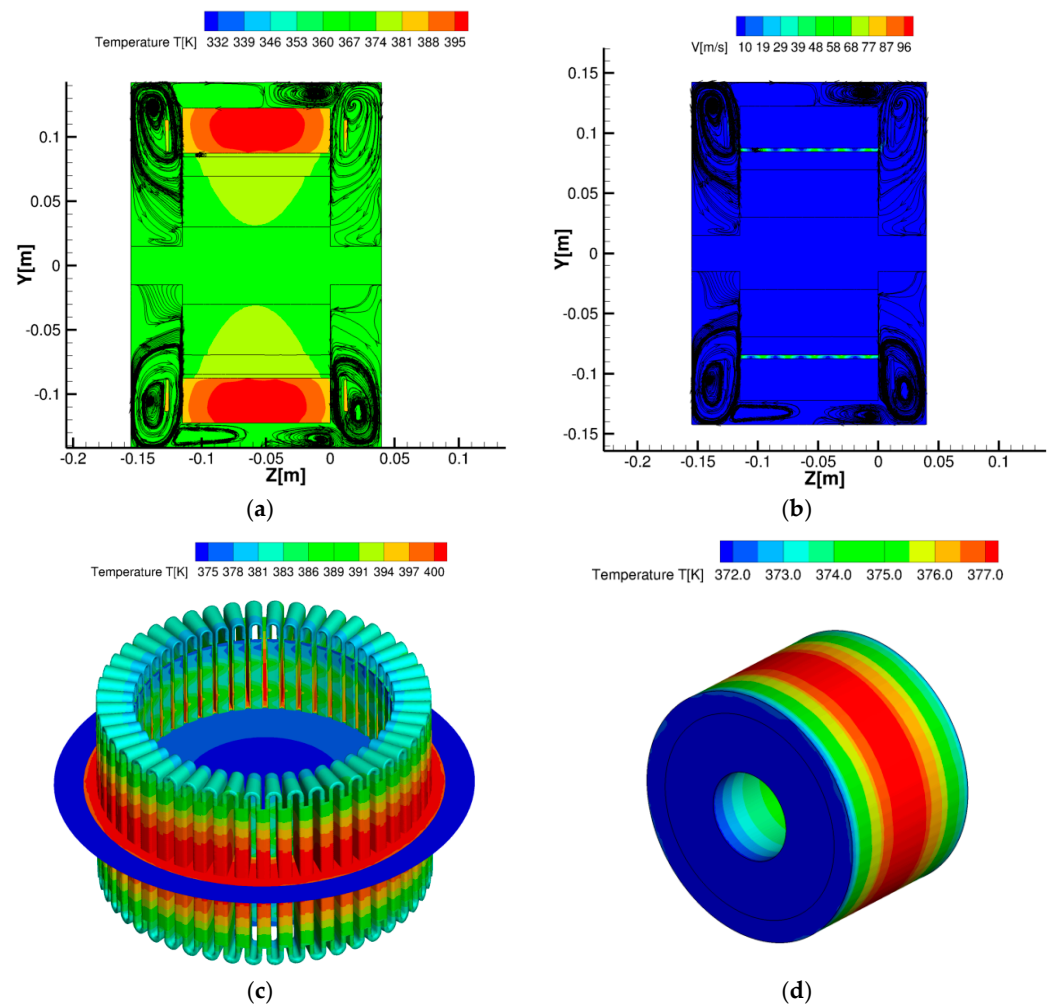


Figure 21. Cont.

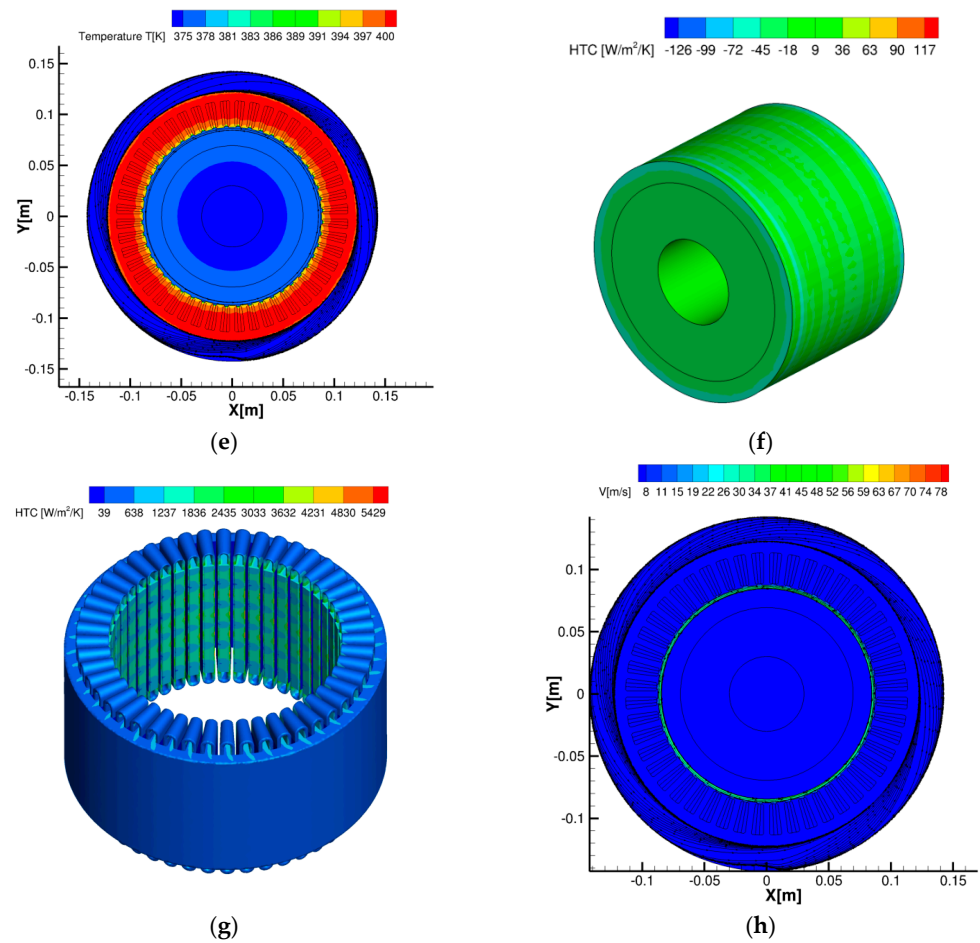


Figure 21. Contour map CFD domain EM—Run 4: (a) Temperature and streamlines $x = \text{cost.}$; (b) velocity magnitude; (c) temperature on windings; (d) temperature on rotor; (e) temperature at section at $z = \text{cost.}$; (f) heat transfer coefficient (HTC) on rotor; (g) HTC on complete stator; (h) velocity magnitude at section at $z = \text{cost.}$

Figure 22 summarises the results of the Run 10 case. The same considerations previously described can also be applied to this test case.

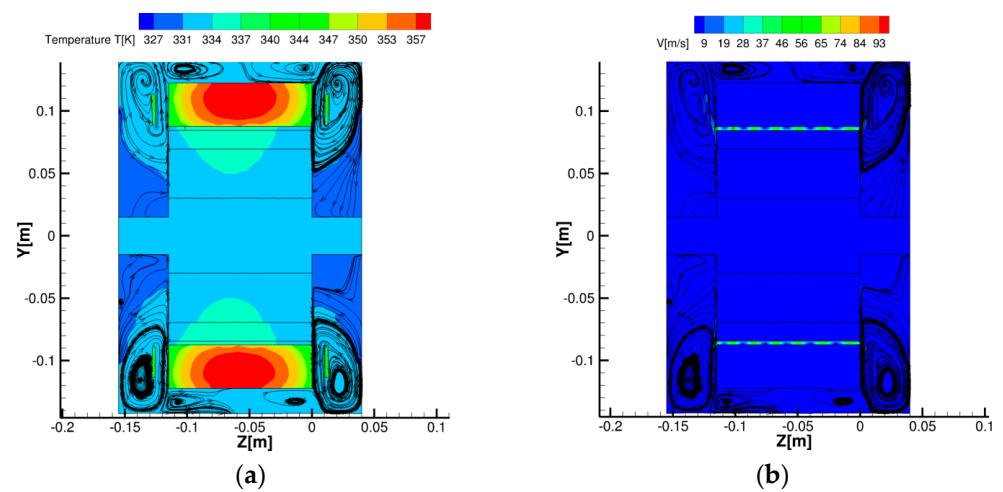


Figure 22. Cont.

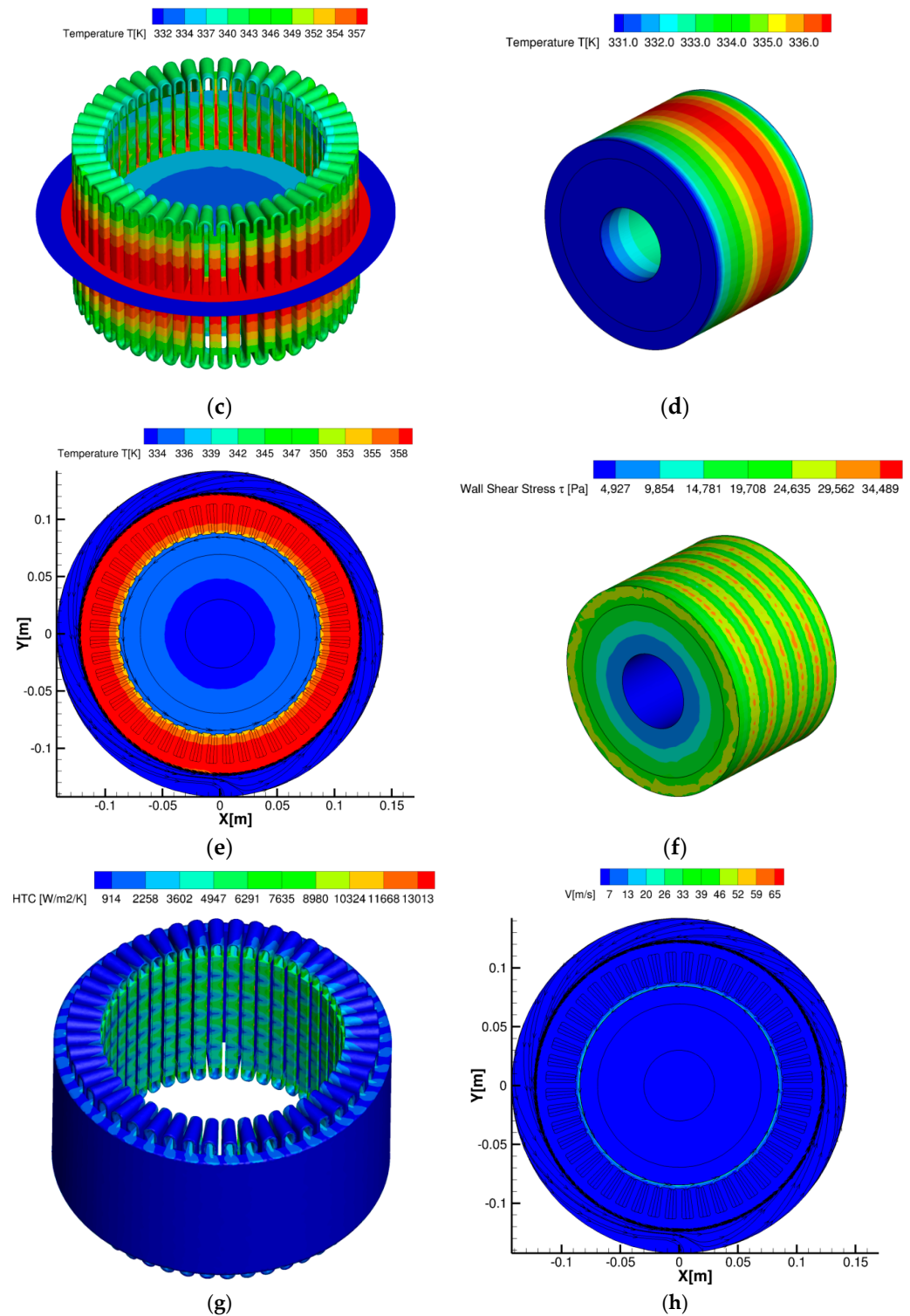


Figure 22. Contour map CFD domain EM—Run 10: (a) Temperature and streamlines $x = \text{cost}$; (b) velocity magnitude; (c) temperature on windings; (d) temperature on rotor; (e) temperature at section at $z = \text{cost}$; (f) WSS on rotor; (g) HTC on complete stator; (h) velocity magnitude at section at $z = \text{cost}$.

Figures 23 and 24 show the trends on the curvilinear abscissa of the convective heat exchange coefficient along the coil profile of a winding. In particular, two symmetrical coils are considered (indicated as upper and lower), one located near the oil outlet hole Y_{min} (lower) and the other symmetrical above (upper) Y_{max} .

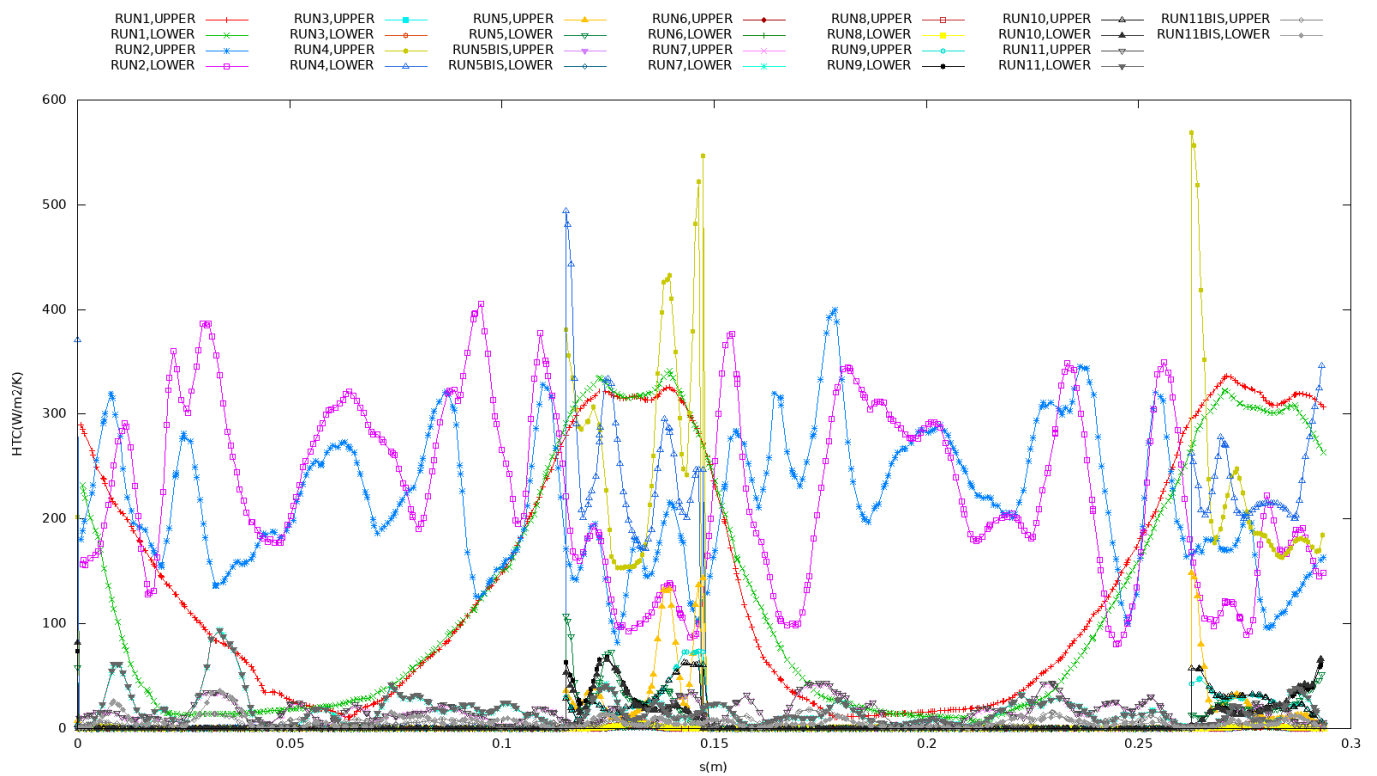


Figure 23. Trend of HTC on two stator windings, at top and bottom of oil outlet, for all test cases analysed.

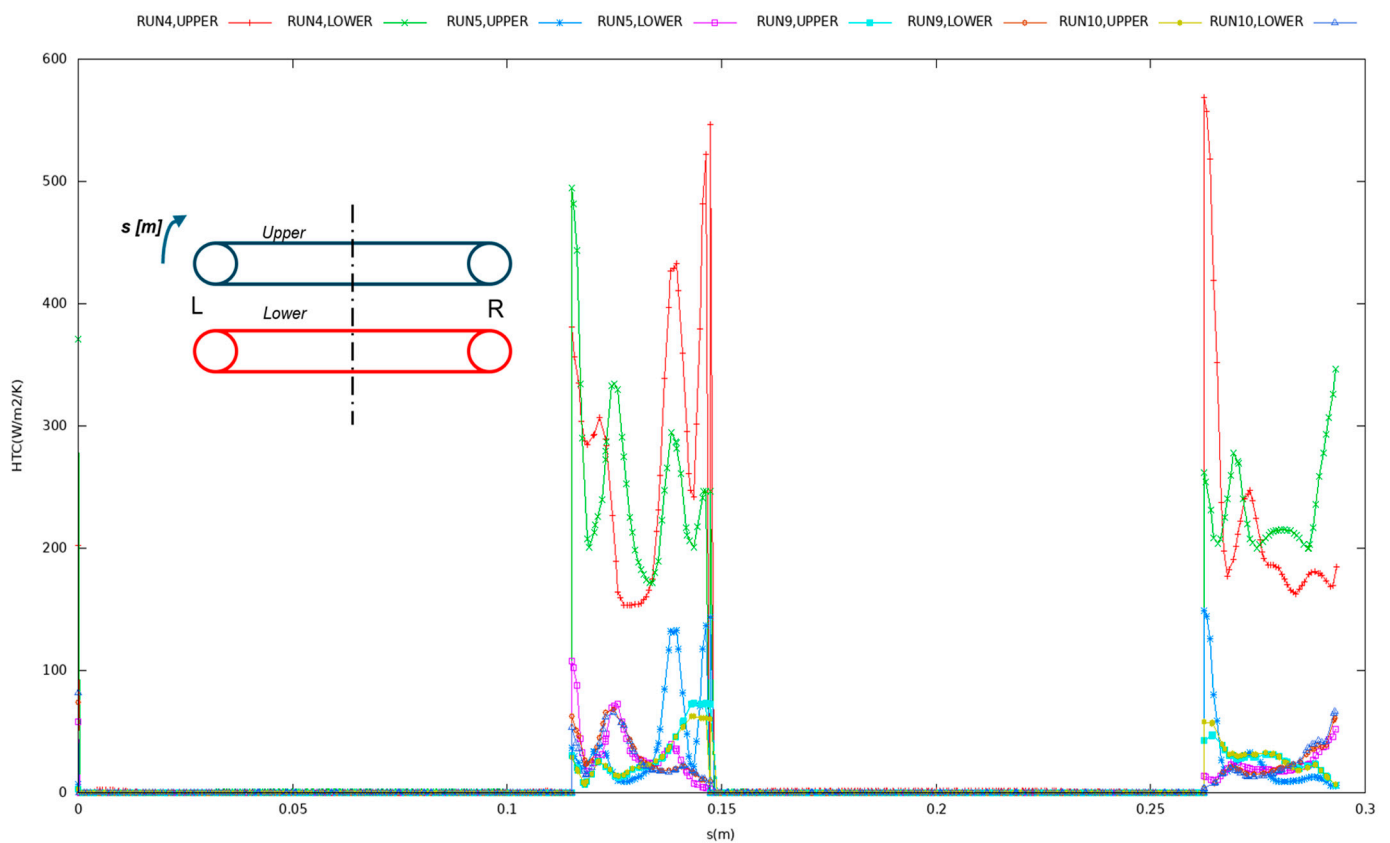


Figure 24. Trend of HTC on two stator windings, above and below oil outlet, for test cases with resin-filled stator slots and active rotation.

The lack of symmetries depends on the fact that the fluid dynamic field generated is not symmetrical due to the presence of the exit hole and the rotor's rotation speed.

The highest HTC values occur at higher cooling flow rates, especially at the maximum rotation speed of 20,000 rpm. Figure 24 shows the HTC trend in the case of a stator slot filled with epoxy resin as insulation. In this case, the refrigerant oil finds an obstacle; in the range of s [0.0, 0.12] and [0.145, 0.26], the HTC equals 0, and there is no exchange. There is, instead, a strong heat exchange along the zone exposed to the oil jet where the convection follows the curvature distribution of the end-winding. Indeed, it increases as the flow accelerates and moves away from the leading edge.

Table 9 reports the maximum and average windings, stators, and teeth temperature results. Furthermore, the values of the pressure drops obtained from the CFD software Ansys FLUENT 2021R2, the temperature and speed of the exit hole, useful data on the design of the radiator and oil pump, and the shear stresses on the rotor are reported for each case study analysed.

Table 9. CFD summary results.

Id. Case	V_{out} [m/s]	$T_{out, ave}$ [K]	Δp [Pa]	τ_{ave} [Pa]	τ_{max} [Pa]	T_{Max} (stat) [K]	T_{Ave} (stat) [K]	T_{Max} (EW) [K]	T_{Ave} (EW) [K]	T_{Max} (Teeth) [K]	T_{Ave} (Teeth) [K]	L/1'
Run 1	0.574	325	4.75	4.48×10^{-3}	0.33	564.7	537	564.7	533	565.6	539	9
Run 2	0.570	325	-1.38×10^3	5.45×10^3	41,962.1	388.9	384	388.9	383	388.9	377	9
Run 3	1.15	349	-1.38×10^3	0.00	0	491.7	465	491.7	462	492.7	465	18
Run 4	0.568	372	1.26×10^3	1.41×10^4	33,516.8	402.3	395	402.3	393	402.3	386	9
Run 5	1.15	325	-2.82×10	1.73×10^4	35,718.0	379.8	373	379.8	370	379.8	363	18
Run 5_bis	1.16	325	3.55	0.00	0.0	554.3	531	554.3	527	555.4	536	18
Run 6	2.18	330	-5.19×10^2	2.97×10^{-2}	0.84	483.1	454.7	483.1	451	484.3	456	18
Run 7	0.542	337	-1.11×10^3	1.12×10^4	41,561.2	355.2	351	355.2	349	350.9	343	18
Run 8	4.36	328	-5.04×10^2	1.15×10^{-1}	2.63	439.5	415.6	439.5	413	440.5	415	36
Run 9	2.24	337	-3.66×10^3	1.75×10^4	39,505.0	366.7	360	366.7	358	366.7	351	18
Run 10	4.49	331	-1.75×10^4	1.76×10^4	39,416.0	360.8	354	360.8	352	360.8	345	36
Run 11	4.36	331	-1.07×10^3	1.13×10^4	41,897.5	349.5	345	349.3	343	349.3	337	36
Run 11_bis	4.36	331	-2.48×10^4	0.3×10^4	11,856.7	356.8	352	356.8	350	356.8	343	36

Figure 25 shows the values of the maximum temperature reached by the motor in the various cases analysed. As can be seen, the values that exceed the limit are reached only in the Run 1 and Run 5_bis cases, which represent theoretical cases in which we apply the thermal power, but there is no motor rotation. For all the other cases, as the flow rate varies, the temperature is always below the imposed limit. It should also be noted that in the case with half the rotation speed, considering all the other conditions fixed, the maximum temperature increases by 6 K. This depends on the fact that the rotation speed has a prominent effect on cooling, which, combined with the TMS, makes everything efficient. Still, its influence tends to reduce between 5,000 rpm onwards, as analysed by the scientific literature.

Another important effect to note is the influence of the air gap on cooling, and in particular, the effect of EM rotation on the fluid dynamic field induced in this restricted area. Figure 26a and b show the maps of the HTC on the rotor and the speed in the air gap, respectively. In this area, heat transfer by convection comes from the stator windings towards the rotor. Still, other losses are generated due to friction and, therefore, to the shear stresses in the cavity.

Mechanical losses can be divided into friction losses and windage losses. Frictional losses depend on speed and occur, for example, in bearings. Ventilation losses occur in electric motors with non-round rotors and depend on speed. For example, switched reluctance motors (sr-motors) or separately excited synchronous motors have rotors that are not round.

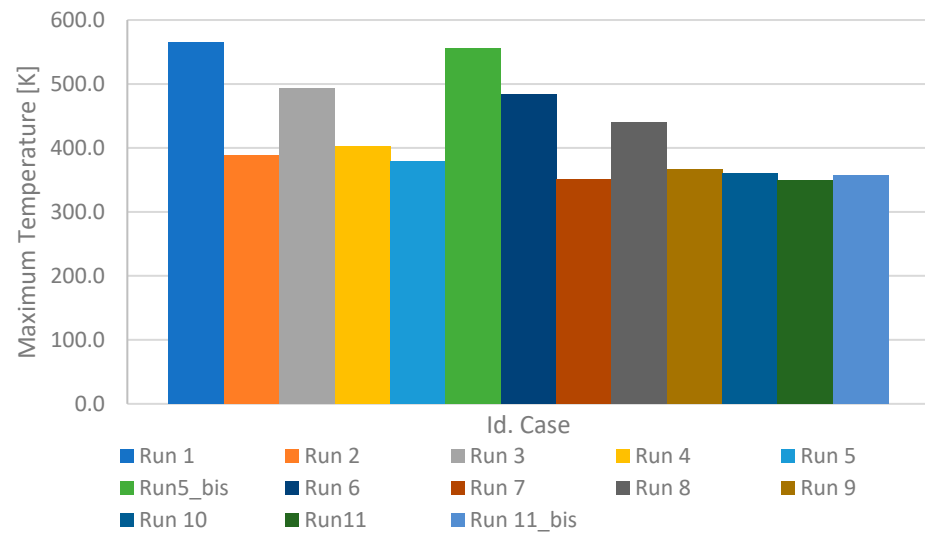


Figure 25. Maximum temperature value for various test cases.

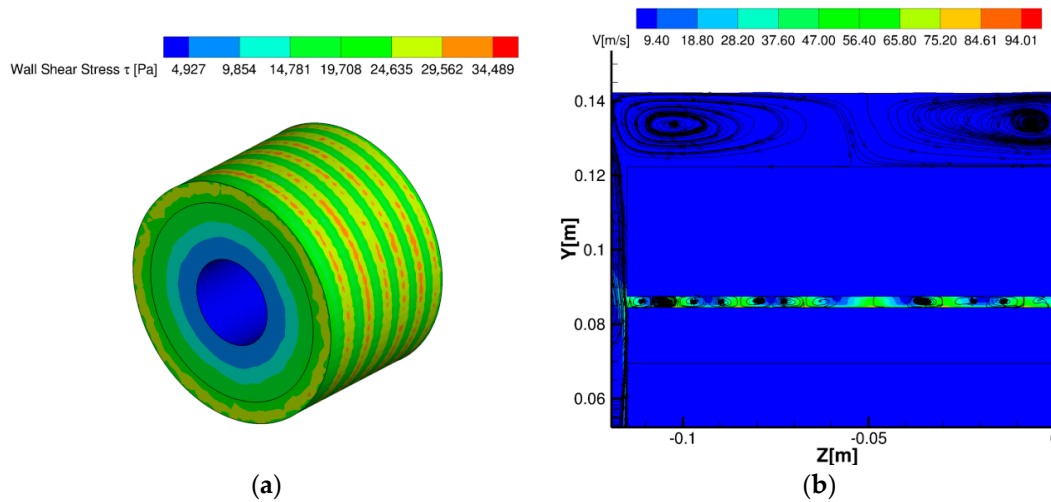


Figure 26. Detail contour maps—Run 10: (a) WSS on the rotor, alternating striations due to Couette–Taylor motion in the air-gap; (b) velocity field section along the axis, air-gap detail.

Windage loss in air-gap is relative to rotor movement, creating tangential velocity components. The friction of rotating air to surfaces and between its fluid layers creates significant heat dissipation. These losses are estimated with the following:

$$P_{air-gap} = k_r \cdot C_{M,air-gap} \cdot \pi \cdot \rho_{fluid} \cdot \omega^3 \cdot r_{rot}^4 \cdot L_{air-gap}$$

where k_r is the roughness coefficient, r_{rot} is the rotor radius, $L_{air-gap}$ is the length of the air-gap and $C_{M,air-gap}$ is the friction coefficient in the air-gap, defined as:

$$C_{M,air-gap} = 0.515 \cdot \left(\frac{e}{r_{rot}} \right)^{0.3} \cdot \frac{1}{\sqrt{Re_{air-gap}}}$$

where e is the air-gap thickness, and $Re_{air-gap}$ is the rotational Reynolds number relative to air-gap considering rotor speed ω , defined as:

$$Re_{air-gap} = \frac{\rho_{oil} \cdot e \cdot \omega}{\mu_{oil}}$$

In Figure 26a, stripes of higher WSS are alternating, as are for HTC. This effect is due to alternating vortices in the air-gap and the Couette–Taylor motion induced by the rotation. Therefore, this effect improves the heat exchange locally, especially in the central area where the oil has more difficulties entering directly.

5. Conclusions and Remarks

In general, better cooling translates into greater efficiency, longer life, and lower associated costs of electrical components. The design of the cooling system must be considered as early as possible and should be as important as structural and electromagnetic issues.

This research study presents the TMS designed by the Centro Italiano Ricerche Aerospaziali (CIRA) team, which is intended to equip the PM electric machine developed by the University of Nottingham (UNOTT) as part of the ORCHESTRA project. Therefore, the description of the electric machine and the definition of the requirements were the starting point for designing an effective TMS. In addition to maintaining the correct operating temperature, TMS is designed to add as little weight as possible so that the aircraft's overall performance is not affected by parasitic drag, resulting from compensating for any additional weight.

Diathermic oil jets directed at the end windings of the electric machine was the cooling technique chosen for a more compact and efficient TMS. A preliminary CFD analysis using cutting-edge tools allowed for obtaining the heat map of the electric machine, useful to define the baseline configuration of the TMS to be designed. The identified configuration introduces a coolant to each side of the generator, with 48 orifices per side spraying oil directly onto the end windings, which represent the most critical impact surface that needs to be cooled. Cooling is symmetrical along the motor axis by injecting on both sides. The plane orthogonal to the motor axis at the centreline is a critical area in heat dissipation. Still, from the simulations carried out, it was observed that the maximum temperature in that area is below the design limits. Different oil injection solutions were analysed to verify the influence of the flow rate on the cooling, rotation speed and temperature of the inlet oil ($9 \text{ L/min} < Q_v < 36 \text{ L/min}$, $25^\circ\text{C} < T_{oil} < 70^\circ\text{C}$, $0 < \omega < 20,000 \text{ rpm}$). The results show a dependence of the flow rate on the global cooling performance, increasing the oil inlet temperature and the average temperature of the solid domain by the same amount. A significant effect, however, is due to the rotation speed; in our full three-dimensional (3D) Reynolds Averaged Navier–Stokes (RANS) type analyses, considering a Moving Reference Frame of the rotor, rotation generates a distortion of the flow field, which is favourable for cooling. In fact, in Table 9, the maximum temperature difference between Run 1 and Run 2 (the conditions of the CFD runs are defined in Table 8) is a decrease of approximately 70 K. The influence of the rotation speed is significant up to 10,000 rpm, then decreasing for higher speeds, as can be seen from Run 11 and Run 11_bis. This is because, at higher speeds, the increase in shear stresses generates significant frictional heat, which limits their effectiveness. The maximum operative temperature of the motor is below the imposed limit of 523 K in all realistic conditions examined. Theoretical conditions, however, such as those with a stationary rotor, are considered only to verify the influence of the various parameters. Using the designed cooling system, for the motor nominal power, a weight/power ratio of approximately 14 is obtained, but for the peak power, which many catalogues adopt as a reference, it is possible to reach 20 kW/kg, while still maintaining the temperature below of the limit with large safety margins.

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Abbreviations

The following abbreviations are used in this manuscript:

3D	Three-Dimensional
ACARE	Advisory Council for Aviation Research and Innovation in Europe
ATF	Automatic Transmission Fluid
ATR	Avions de Transport Régional
CFD	Computational Fluid Dynamics
CIRA	Centro Italiano Ricerche Aerospaziali
CO ₂	Carbon Dioxide
EC	European Commission
EG	Electric Generator
EM	Electric Motor
EPS	Electric Power System
EU	European Union
GA	General Aviation
HTC	Heat Transfer Coefficient
ICE	Internal Combustion Engine
IPCC	Intergovernmental Panel on Climate Change
NO _x	Nitrogen Oxide
Nu	Nusselt number
ORCHESTRA	Optimised Electric Network Architectures and Systems for More-Electric Aircraft
PCM	Phase Change Material
PM	Permanent Magnet
RANS	Reynolds Averaged Navier–Stokes (equations)
RBCC	Reverse Bryton Cycle Cryo-Cooler
TMS	Thermal Management System
TRL	Technology Readiness Level
UNOTT	University of Nottingham
WSS	Wall Shear Stress

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