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Cooling of Automotive Traction Motors: Schemes, Examples and Computation Methods– A Review

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***Abstract*—This paper presents a comprehensive overview of the latest studies and analyses of the cooling technologies and computation methods for the automotive traction motors. Various cooling methods, including the natural, forced air, forced liquid and phase change types, are discussed with the pros and cons of each method being compared. The key factors for optimizing the heat transfer efficiency of each cooling system are highlighted here. Furthermore, the real life examples of these methods, applied in the latest automotive traction motor prototypes and products, have been set out and evaluated. Finally, the analytical and numerical techniques describing the nature and performance of different cooling schemes have been explained and addressed. This paper provides guidelines for selecting the appropriate cooling methods and estimating the performance of them in the early stages of their design.**

Index Terms—Automotive applications, cooling, traction motors, thermal analysis, numerical analysis.

NOMENCLATURE

A	Cross section area of heat path (m^2).
A_l,	Linear current density (kA/m).
A_i,A_o	Inlet and outlet cross section areas (m^2).
c_p	Specific heat capacity (J/kg).
D	Diameter (m).
f_s,f_r	Friction loss factor (dimensionless).
g	Gravitational attraction force (m/s^2).
Gr	Grashof number (dimensionless).
H	Fin extension (m).
h	Heat transfer coefficient (W/m^{2}K).
h_l	Latent heat (kJ/kg).
k	Loss coefficient (dimensionless).
J	Current density (A/mm^2)
L	Length of the surface (m).
N	Number of fins (dimensionless).
$Nu_{Natural}$	Natural *Nusselt* number (dimensionless).
Nu_{Forced}	Forced *Nusselt* number (dimensionless).
Δp	Pressure drop (Pa).
Pr	*Prandtl* number (dimensionless).
R	Convection thermal resistance (K/W).
Re	*Reynold* number (dimensionless).
Re_r	Rotational *Reynold* number (dimensionless).
T_w, T_f	Wall and fluid temperatures (K).
ΔT	Temperature difference (K).
S	Fin pitch (m).
V	Axial velocity (m/s).
V_r	Tangential velocity (m/s).
μ	Dynamic viscosity (Pa·s).
λ	Thermal conductivity (W/m·K).
ρ	Density (kg/m^3).
β	Coefficient of the expansion (dimensionless).
σ	Tangential stress (kPa).

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I. INTRODUCTION

WHILE operating an electric motor, heat is generated due to the electromagnetic losses, mechanical power losses and other stray losses that take place in various components within an electric motor. Through conduction, convection and/or radiation, the thermal energy is transferred to a cooling medium [1] on the basis of a temperature difference between the hot and cold bodies. However, a detailed thermal management is essential during critical operating conditions, such as overload running, phase changing and/or asymmetric faults, to avoid failures that are usually due to the local hot spot formation, and material degradation [2-5]. Furthermore, the topic of magnetic losses and heat generation governs the performance of the electromagnetic efficiency and longer life expectancy. Firstly, excessively high temperatures can cause accelerated insulation aging [6] and deterioration within some essential components, such as winding conductors [7]. Secondly, the remanence and coercivity of the rare earth magnets are inversely proportional to the temperature. As a result of which, partial or full demagnetization at higher temperatures may occur [8, 9]. In case of the ferrite and recycled magnets [10, 11], the lower rotor temperatures may

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significantly boost the torque density and / or efficiency of the motor, as for the remanence sensitivity of these magnet to the temperature is 200%-300% higher than the conventional rare earth magnets. In addition, the electrical resistivity of the winding conductors is in a direct proportion to the temperature. This process can lead to a positive feedback in which an accelerated loss and temperature rises occur in the windings [12]. Finally, the thermal impact on the geometrical dimensions of the motor's physical structure, such as a narrowing within the airgap, may alter the motor's nominal performance, or, in serious cases, result in faults and failures [13]. To tackle the thermal challenges in an electric motor, alongside the minimization of the magnetic losses as sources of heat, one needs to carefully address the heat dissipation mechanisms for a given design in order to obtain a balanced heat distribution across different components.

Generally speaking, a thermal design uses a closed or an open cooling circuit to achieve a critical temperature balance within an electric motor. Heat from the inner components is conducted to the outer surface of the motor and then is subjected to the convective cooling. The former process is a kind of passive thermal design, which is affected by material properties, geometrical layout and contact interfaces. This process is considered economical and does not produce any additional parasitic effects such as acoustic noise. An alternative method is the so-called active cooling design in which an extra source of energy is applied to circulate a fluid with a high heat capacity in order to exchange and extract the heat from the hot surfaces [14]. This active method of cooling applies an external force created by a special device, e.g. pumps, fans to generate sufficient coolant flow to remove heat from the interior parts of a motor. This approach provides a high convection heat transfer capacity but the extra provisions are required for diminishing not only friction losses, but also risks of short circuit faults and corrosion [15]. Table I lists the typical values for the tangential stress σ, linear current density A_l, current density J and heat transfer coefficients h of different cooling methods.

A diversification of cooling approaches have been pursued to meet the cooling demands placed on various applications. However, the rapid growth of aerospace and traction industries have brought about increased requirements for electric motors such as compactness, high speed and high power density. This leads to significant rises in temperature in cases where miniaturized motors are involved, thus necessitating a more sophisticated and complicated cooling system to keep the working temperature within a safe range [16, 17].

In this paper, a detailed analysis of the active type cooling: the natural [21-28], forced air [29-42], forced liquid [43-63] and phase change types [18, 66-68], are reviewed in Section II. On this basis, a comprehensive summary of the convection methods as applicable to the automotive traction motors cooling contexts have been provided with the advantages and disadvantages of each method being compared. The essential elements for optimizing the cooling performance of each method together with the leading applications are specifically highlighted. In addition, the latest automotive traction motor prototypes and products [69-75] employing these methods, have been set out and evaluated in Section III. In Section IV, the use of the analytical lumped-circuit and the computational fluid dynamics techniques [90-94] for calculating the cooling performance are proposed and discussed. Section V sets out the conclusion in which a number of recommendations concerning the future developments and applications is offered. The aim of this paper is to familiarize and equip the electrical machine designers with a concise knowledge of the thermal background to improve upon the overall performance of their products.

TABLE I
THE TYPICAL VALUES FOR DIFFERENT COOLING METHODS [18-20].

Cooling method		σ, kPa	A_l, kA/m	J, A/mm^2	h, W/m^2 K
Natural convection		-	-	1.5-5	5-30
Forced gas cooled	Air	<15	<80	5-10	20-300
	Hydrogen	<25	70-110	7-12	100–1000
Forced liquid cooled	Indirect	20-60	90-130	7-20	100-10000
	Direct	60-100	100-200	10-30	200-25000
	Phase change	-	-	-	500–50000

II. COOLING METHODS

A. Natural passive cooling

Natural cooling uses the on-site energy, combined with the configuration of motor components to dissipate heat. The housing is the main path through which the heat is removed from inner components to the ambient environment. The design of the housing needs to be optimized in order to maximize the rate of convective heat dissipation.

In practice, correlations of convective heat transfer (HTC) h have been developed for natural cooling to show that the Nusselt number Nu mainly depends on the Grashof number Gr and the Prandtl number Pr [21], defined as equations (1-4):

$$Nu = h \cdot L/\lambda \tag{1}$$

$$Nu_{Natural} = f(Gr, Pr) \tag{2}$$

$$Gr = \frac{g\beta\rho^2(T_w - T_f)L^3}{\mu^2} \tag{3}$$

$$Pr = \mu \cdot c_p/\lambda \tag{4}$$

A suitably designed finned housing can improve the heat transfer coefficient value as compared to a non-finned housing. The cooling fins are normally placed on the surface of the housing, and are oriented in such a way as to not disturb the natural airflow. There are two types of fins branching off in different directions relative to the motor shaft: one being a radial fin array [22], the other an axial fin array [23].

The heat transfer rate from fins to the ambient environment may rise either by increasing the heat transfer coefficient and/or the fin surface area. However, the natural convection heat transfer coefficient depends on the ambient conditions. A common practice for improving the natural convection heat transfer is to extend the fin area; however, this increases the resistance of the air flow which in turn diminishes the gain

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factor. The optimization of the fin extension H, fin pitch S and the number of fins N which are illustrated in Fig. 1, are the main ways to increase the natural cooling performance [24-29]. The key design objective must be to maximize the rate of the heat dissipation, while minimizing the weight and volume of the cooling fins.

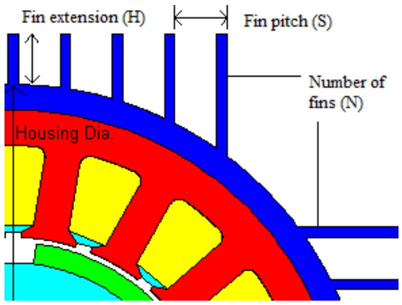

Fig. 1. Fin configuration geometry.

The natural cooling approach, however, is appropriate only for low or medium-power motors or large electric motors with sufficient heat transfer areas.

B. Forced cooling

Forced cooling is a more popular approach than the passive type discussed in Section A, as more power dense and compact motors are being introduced to the market. As compared to the natural cooling, the forced cooling uses an external device and source of energy to create sufficient coolant flow to exchange and extract the heat from the hotter components. The Reynolds Number Re is used to determine the flow patterns, so-called flow regimes, under different cooling media and architectures, and can be analytically estimated by (5). The heat transfer based on the forced convection method can be defined as a function of Re and Pr in accordance with (6) [21],

$$Re = \frac{\rho D V}{\mu} \tag{5}$$

$$Nu_{Forced} = f(Re, Pr) \tag{6}$$

1) Forced air

In a forced air cooling system, a fan or a blower is employed to generate the continual passage of air through a motor or over its exterior. Depending on the enclosure of a motor, forced air can be divided into two different varieties: an enclosed fan cooled (EFC) motor and an open fan cooled (OFC) motor.

An EFC motor consists of an inner and an outer flow circuit. These are displayed in Fig. 2 [30, 31]. The recirculating air from the inner circuit brings heat from the inner motor to the housing frame, where an outer flow circuit functions as a heat sink. The EFC configuration prevents a free exchange of air between the inside and the outside of the motor. An internal fan, either integral to the rotor or mounted on the shaft, circulates air inside the enclosure which promotes the heat transfer to the frame. An exterior fan, makes the surrounding air pass over the housing, thus removing heat to the ambient environment. However, the efficiency of a shaft-mounted fan is limited by the speed of shaft. Hence an external fan or blower is employed to generate the optimal level of air pressure independently of the shaft speed. For an EFC approach, the recirculated air is often cooled via the ambient air through the external frames in case of the small motors or by an air-to-water heat exchanger in case of the large motors. The key benefit of the EFC scheme is that the interior parts are better protected against pollutants which may block the ventilation ducts, with the risk of impeding the airflow. Furthermore, the cooling performance can be improved by replacing the air with a suitable gas that has higher heat conduction and higher specific heat capacity than air, e.g. hydrogen [32]. This is owing to the fact that the smaller and lighter gas molecules can result in a lower windage loss and better heat transfer than air.

An OFC motor ventilation structure is illustrated in Fig. 3. The coolant air is continuously drawn from the ambient environment into the motor enclosure, and not re-circulated. Since, in this method, the motor is exposed to the environmental contaminants, provisions such as using filtering or employing indirect air channels need to be in place to prevent particles or moisture from entering the motor [33]. Because of the accumulated pollutants, OFC motors are regularly dismantled for a clean-up operation once every two or three years [34].

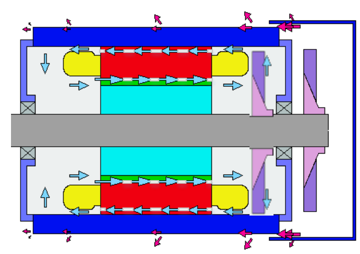
Fig. 2. The ventilation structure of an EFC motor.

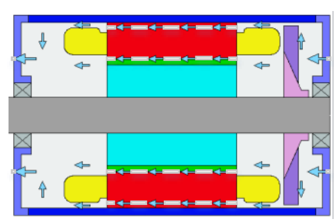
Fig. 3. The ventilation structure of an OFC motor.

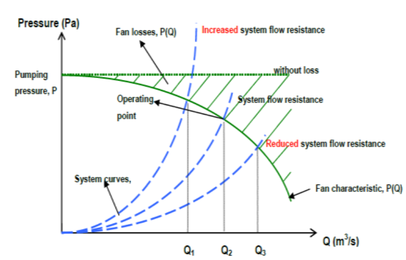

Fig. 4. A typical fan characteristic curve [35].

The cooling performance of the forced air motors strongly depends on how large the surface contact areas is between the coolant and the motor components. This can be improved by adding geometrical modification such as cutting multiple air slots into the shaft, rotor, or the stator core [36].

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In a fan based cooling system, the fan provides a differential pressure to make the coolant air flow. Fig. 4 shows the relationship between the fan characteristic and the motor enclosure system resistance curve, as well as the operating pressure and flow rate at the intersection point. The bending of the fan characteristic curve is due to the energy losses, and can be improved by optimizing the aerodynamic structure of the blades. A new kind of axial fan with forward-swept and inclined blades is employed in [37-39] to reduce the ventilation resistance inside an electric motor, as is illustrated in Fig. 5. Further enhancement in cooling can be achieved by various retrofit methods, such as adding internal air baffles to an EFC motor [40], or by interrupting any combination of flows from occurring especially at high rotor speed [35], as is shown in Fig. 6.

One of the major challenges associated with fan cooling is the emission of acoustic noise, especially at a high speed fan operation. Several noise mitigation methods have been proposed by the literature: a) using forward-swept inclined fans [37, 38]; b) using a better aero-foil shape blade cross-section [41]; c) using inlet bell-mouth entry [42]; d) using composite materials for blades [42]; e) reducing the number of blades [41, 42]; f) using irregular-pitch-blade fan [30, 42, 43]; g) using a mixed flow (both axial and radial) fan [41].

Fig. 5. The axial fan with forward swept and inclined blades [39].

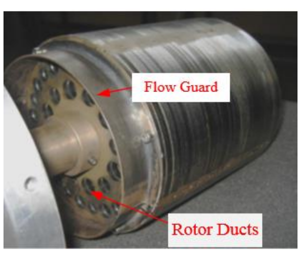

Fig. 6. Arrangement of flow guard [35].

2) Forced liquid

A forced liquid cooling solution is suitable in particular applications, especially for high-power electric motors, where the requisite outputs cannot be attained by EFC or OFC motors. Forced liquid cooling approaches such as those that are designed for electric motors are presented in Fig. 7. In such a cooling system, the forced liquid passes through the housing jacket, stator channels and/or rotor channels. However, the forced liquid cooling system suffers from a number of weaknesses, such as stains, corrosion, leakage and contamination. The remaining stains inside of cooling channels may lead to a significant rise in flow resistance, which causes a decrease in the cooling effectiveness. The most common liquid coolant in thermal management of electric motors is water. The reason why water is chosen is primarily due to the high relative heat capacity of this liquid. In addition, a number of components are available for commercial applications, such as ethylene glycol and water (EGW) 50/50 and engine oil.

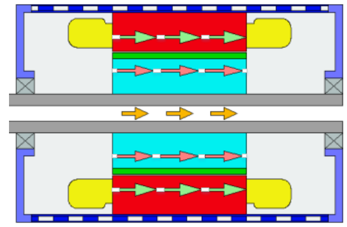

Fig. 7. The forced liquid cooling models.

a. Housing water jacket

The cooling via a housing jacket is the most common forced cooling approach. This is where the liquid flows through the cooling channels situated in a thermally conductive frame above the stator stack [17, 44-46]. The heat generated in the coils, as well as in the stator and rotor laminations, is initially transferred to the cooling housing through conduction, and is, then transferred to the ambient environment via convection in the coolant fluid. The efficiency of the liquid cooling technique heavily depends on the geometrical clearance and the resultant thermal resistance between the laminated stator core and the cooling housing.

The effects of different parameters on the contact thermal resistance between the laminated stator core and the frame, such as shrink fit pressure, thermal paste use etc., were experimentally investigated in [47]. A ferrite magnet motor design, [48], is used to verify the effect of the contact interfaces on the stator winding and magnet temperature. The results at 10000 rpm/55.5 kW, as is shown in Fig. 8, are based on an analytical method using Motor-CAD and these indicate that a poor contact between the stator and frame encourages an increase in the temperature of the winding and the magnet, with the maximum possible increase being about 40 ˚C .An alternative to water in a liquid cooling system, includes Shell Tellus oil premium 22 [49], and Statoil transmission oil [50], which are provided from the lubricating oil already applied to the gearbox system.

Whilst a housing jacket provides a sufficient heat transfer for the active part of the stator winding, it is, usually, inadequate to dissipate the heat from the end winding and rotor due to the high thermal resistance between the heat source and the coolant. This can be particularly problematic for motors with long end windings, such as distributed wound motors with few pole numbers.

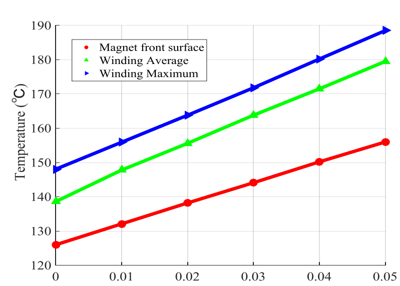

Fig. 8. Sensitivity analysis of the stator and housing contact interfaces on components temperature.

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b. Stator cooling

A further method of liquid cooling is via cooling slots cut directly into the stator laminations. These are located in the stator yoke [51-55] or the winding slots [56-59]. Even if the coolant gets closer to the windings and the laminations, resulting in a smaller thermal resistance, care must be taken to prevent the cooling slots from disturbing the magnetic flux in the stator core [60, 61]. In [52], the effects of an evenly spaced number of different water slots on the average temperatures of winding (T1), winding epoxy (T2), slot insulation (T3), iron core mover (T4), have been shown in Fig. 9. These effects are based on finite element (FE) simulations to demonstrate the effectiveness of the approach for this water-cooled permanent magnet linear motor. It is worth noting that the maximum number of water pipe slots should be chosen as a tradeoff between the ease of manufacture and the required temperature rise limitations.

Using a wet stator is a less commonly employed way of stator cooling. In this less common method, a fluid with a high heat transfer coefficient, such as hydraulic oil [62], is made to pass through the armature end windings and the stator laminations. A sleeve is introduced to prevent the liquid from entering the air gap in order to avoid windage loss, as illustrated in Fig. 10. This cooling approach provides the minimum thermal resistance between the coolant and the heat source, providing the best heat transfer capability [63].

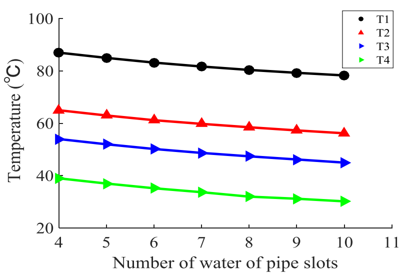

Fig. 9. Stator winding maximum temperature at various flow speeds with different water slots based on FE simulations [51].

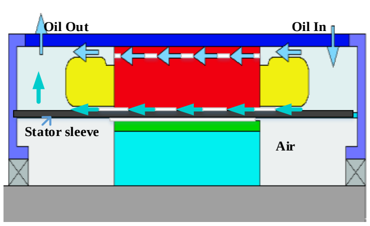

Fig. 10. Wet stator cooling system.

c. Rotor cooling

Having an internal rotor is usually associated with a poor heat transfer due to the airgap acting as an insulating material. Poor heat transfer leads to a loss of electromagnetic performance. To enhance the heat transfer, the enclosed air can be replaced by a high thermal conductivity medium, i.e. water or oil [62]. After that, the motor is totally flooded and the rotor and stator surfaces are directly flushed by a coolant. However, a direct liquid cooling method is not an economical and practical one. This is due to the extra provisions required for diminishing not only the friction losses, but also risks of short circuit faults and corrosion [62]. Reference [64] introduces an annular gap between the winding and the airgap for slotless motors. This is where the coolant is brought closer to the rotor. As a rotary part, the direct liquid cooling apparatus of a rotor is difficult to manufacture. As a results, a hollow shaft cooling system [65] as an indirect way of dissipating heat can be employed. Fig. 11 presents a shaft cooling system, where the coolant is introduced into the system via a coupling connected to a stationary inner tube and is driven back to the gap between the injection tube and the hollow shaft [16].

The heat transfer characteristics of the rotor cooling system are complex. A secondary flow will occur as a result of the rotation while the cold and dense fluid in the center tends to move radially to the wall due to the Centrifugal and Coriolis Effect. Consequently, the convection heat transfer coefficient correlations of a stationary case is invalid for rotor cooling [66] and the rotational Reynold number Re_r must be introduced to estimate the effect of the rotation. The typical form of the convection correlation for rotating flow is defined as (7) (8) [66]. Reference [65, 66] have experimentally and theoretically demonstrated that the rotational speed can significantly increase the convective heat transfer in the rotor cooling above the stationary condition.

$$Nu = f(Pr, Re, Re_r) \tag{7}$$

$$Re_r = \rho L V_r / \mu \tag{8}$$

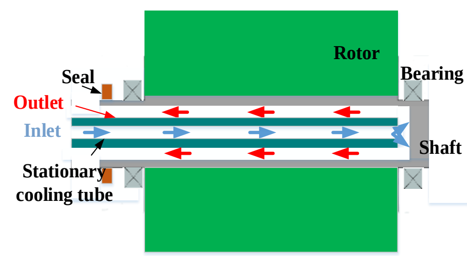

Fig. 11. Indirect rotor cooling scheme.

d. Phase change

In a phase change cooling system, a coolant is applied in such a way that at least a portion of the coolant is transformed into vapor upon heating. The principle of a phase change loop is illustrated in Fig. 12. A refrigerant is pumped into the heat source as an evaporator, at which point, after absorbing the thermal energy, is transformed into a gaseous state. The heated gas is then recycled back into the compressor, releasing thermal energy. The refrigerant at the liquid state is pumped back again as a new cycle begins. Phase change cooling systems provide high heat dissipation rates at higher operation temperatures with a considerably smaller working volume than a single phase [67]. The simplest phase change cooling scheme is a heat pipe. It is a closed pipe filled with a dedicated working fluid. In manufacturing terms, heat pipes can easily be installed in electric motors. However, the heat transfer capability of a heat pipe depends on the dimensions and the temperature drop between the two ends [19].

Spray cooling [68-70] is another cooling technique which involves a phase change. Spray cooling of the stator and rotor end turns is depicted in Fig. 13. A cooler liquid is sprayed onto the end-windings and/or rotor via nozzles. After moving over various surfaces on the inside of the motor, the coolant is

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partially evaporated into a gaseous state (phase change), which is condensed back into liquid in the condenser and drained out into the reservoir along with the unevaporated coolant. Spray cooling can be used to transfer large amounts of energy through the latent heat of evaporation and create uniform temperatures without reducing magnetic field fluctuations or creating electric noise. Despite these advantages, spray cooling is considered excessively complex insofar as it requires nozzles with high amounts of pressure [71]. The submerged double jet impingement method [72] and experimental approach [14] are used for calculating the heat transfer coefficients of the spray cooling.

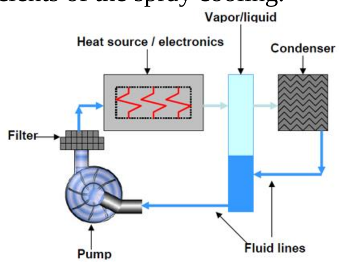

Fig. 12. Primary components of a phase change cooling loop.

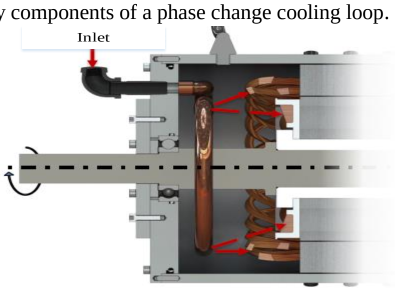

Fig. 13. A spray cooling on the stator and rotor end windings [73].

e. Hybrid approach

In most cases, an electric motor can work reliably under conditions of substantial overloading by applying a single method of cooling. However, for the high power density applications, such as traction motors, a single cooling method may not be sufficient. On this basis, a combined forced-air and liquid cooling system have been applied in [22, 46, 52, 74]. The results relating to the rise in temperature for 30 kW traction motor compare individual and coupled cooling systems, as is shown in Fig. 14. It is noted that the coupled cooling schemes are more effective whilst any suitability for the mass production remains a challenge due to the manufacturing complexity [16].

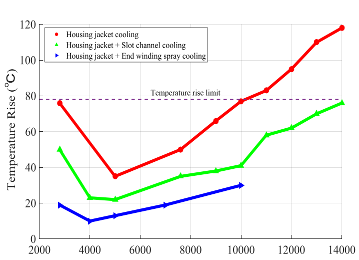

Fig. 14. The performance comparison for various cooling system [16].

III. Traction Applications

The Nissan-leaf electric motor shown in Fig. 15 (a) uses a water jacket for cooling an interior permanent magnet traction motor. Three cooling channels in Fig. 15 (b) are provided in the frame above the stator stack in parallel using EGW 50/50 as a coolant to ensure a sufficient cooling performance.

In Fig. 16 [57], a forced cooling fluid flows through the slot cooling tubes (with the option of being connected in series or parallel). The slot cooling tubes are placed inside the winding slots, adjacent to the coils.

A plurality of heat pipes are inserted in the stator slots in [60], as is displayed in Fig. 17 (a). All the heat pipes are extended into a cooling chamber that can be filled with oil or some other cooling fluid. Another example in [75], Fig. 17 (b) discloses a heat pipe that is located in the motor hollow axle, where a metallic plate as a heat exchanger is mounted at the end of the heat pipe.

Zytek Electric Traction Motor [76], as is shown in Fig. 18, is based on a dual cooling system, a self-ventilated cooling where internal forced air flows through the rotor axial ducts, as well as the airgap of the motor, in combination with a housing water jacket. The recirculating air brings heat from the inner motor to the heat exchanger.

A list of the various cooling methods installed in the latest traction motors has been provided in Table II .

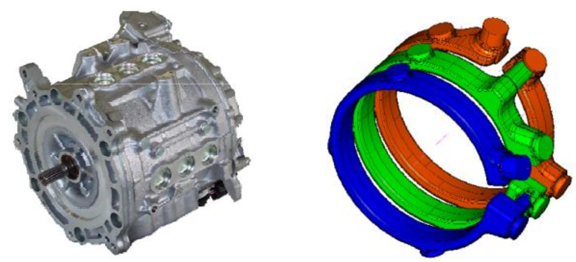
Fig. 15. a) Nissan-leaf electric motor b) Model of cooling water


Fig. 16. Slot duct cooling for the whole motor (left) and single teeth prototype (right) [55].

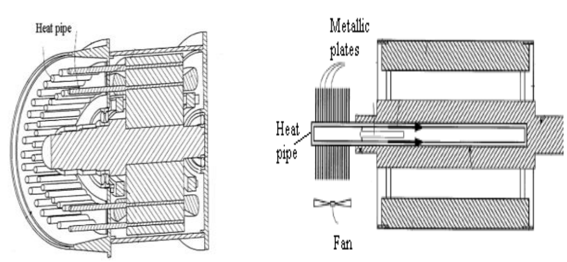

Fig. 17. Motor assembly with heat pipe cooling system [71, 73].

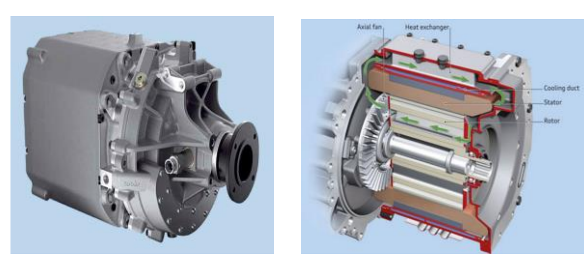
Fig. 18. ZYTEK high power density PMSM with dual cooling system [76].

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TABLE II
THE VARIOUS COOLING METHODS APPLIED FOR AUTOMOTIVE TRACTION MOTO

Motor type	Max torque (Nm)	Max Speed (rpm)	Torque density: Mass/Volume (Nm/kg, Nm/L)	Peak Power (kW)	Power Density: Mass/Volume (kW/kg ,kW/L)	Cooling methods	Reference
2010 Toyota Prius IPMSM	205	13,500	5.5 / 16.4	60	1.6 / 4.8	Housing jacket cooling with oil	[77]
2011 Sonata PMSM	205	6000	7.5 / 20.5	30	1.1 / 3.0	Housing jacket cooling with oil	[78]
2012 Tesla Roadster AC IM	370	14,000	6.98 /-	215	4.05 /-	Inner forced air + Fined housing + outer fan	[79]
2012 Nissan Leaf IPMSM	280	10,390	4.8 / 15.1	80	1.4 / 4.2	Housing jacket cooling with WEG	[80]
2013 Tesla S60 AC IM	430	14800	-	225	-	Housing Jacket cooling+ Shaft cooling	[1]
2015 Newcastle University SRM	280	10,500	5.2 / 15.9	80	1.5 / 4.5	Housing Jacket cooling with water	[81]
2016 BMW i3 IPMSM	250	11,400	6.0 / 18.2	125	3.0 / 9.1	Housing Jacket	[82]
GE Global Research IPMSM	180	14,000	>5.1 / >18.6	55	>1.5 / >5.7	Housing jacket + End winding spray + Rotor cooling	[16]
Zytek PMSM	460	12,200	6.1 /-	170	2.3 /-	Housing jacket + Forced fan cooling	[76]

IV. COMPUTATION METHODS

An accurate understanding of the cooling performance in an electric motor is a prerequisite of an accurate and efficient thermal design. The key parameters to achieve this goal are the convection heat transfer, flow resistance, and fan performance in case of fan cooled systems, etc., while the common approaches include analytical lumped-circuit and numerical methods.

The analytical approach can be subdivided into two main calculation types: heat-transfer and flow-network analyses. Both of them are based on readily available empirical correlations in the thermal analytical literature and have the advantage of being fast. In the lumped-circuit thermal network, the convection heat transfer between the surface of the motor components and the coolant is described by convection thermal resistance defined as (9). The temperature of the motor components allow predictions based on the convection thermal resistances for a given power distribution. In the flow network, a drop in pressure takes place due to the flow restrictions (e.g. friction, expansion, contraction). Any loss in pressure is usually quantified with an empirical loss coefficient (k) based upon the flow of kinetic energy, defined as (10). The dimensionless correlations used for calculating the convection heat transfer coefficient and the flow resistance coefficient are reported in Table III and Table IV respectively.

$$R = 1/(h \cdot A) \tag{9}$$
$$\Delta p = k \cdot \frac{1}{2}\rho V^2 \tag{10}$$

The numerical approaches based on the time step Finite Element Analysis (FEA) and Computational Fluid Dynamics (CFD) methods are commonly applied to the complicated cases (e.g. in case of a rotor duct, or a mounted fan system). However, the model setup and computations can be highly time-consuming especially when 3D modelling in necessary.

The FEA is a highly accurate tool for modelling the solid conduction heat transfer. However, in case of the cooling by connecting two different media, CFD needs to be employed to predict the coolant velocity distribution and pressure drops in the cooling ducts as well as the coolant heat transfer coefficient at the boundaries through the complex shape area. CFD is based on the finite volume technology with the aim of simulating 3-D laminar or turbulent flow with a high degree of accuracy.

TABLE III
THE FLOW RESISTANCE COEFFICIENT.

Friction loss coefficient		
Stationary pipe $k = f_s \cdot L/D$	$f_s = \begin{Bmatrix} 64/Re; Re < 2300 \\ 0.316/Re^{0.25}, 4000 < Re < 10000' \end{Bmatrix}$;	[83]
Rotating shaft $k = f_r \cdot L/D$	$f_r/f_s = \begin{Bmatrix} 1; V_r/V < 0.35 \\ 0.579(V_r/V)^{-0.52}; 0.35 \le V_r/V \le 0.8 \\ 0.47(V_r/V)^{-1.42}; 0.8 < V_r/V) < 1.2 \end{Bmatrix}$;	[84]
Air gap $k = f_r \cdot L/D$	$f_r/f_s = \{1 + (7/8)^2 (Re_r/2Re)^2\}^{0.38}$	[85]
Rotor ducts $k = f_r \cdot L/D$	$f_r/f_s = \begin{Bmatrix} 0.5{Re_r}^{0.16} Re^{-0.03}; 900 < Re < 9880 \\ 0.842{Re_r}^{0.023} Re^{0.002}; Re > 9880 \end{Bmatrix}$	[86]
Sudden expansion, contraction loss coefficient		
Stationary pipe	$k = (1 - A_i/A_o)^2$	[83]
Entrance of air gap	$k = 0.1(V_r/V)^2 - 0.06(V_r/V);\ V_r/V > 1$	[87]
Entrance of Rotor ducts	$k = 0.234(V_r/V)^2 - 0.043(V_r/V); V_r/V > 0.5$	[88]

Reynolds Averaged Navier-Stokes (RANS) equations are mostly used to deal with the 3-D turbulent flow and heat transfer of cooling systems. Six Reynolds stresses need to be calculated in order to close the RANS equations. A common method is to use the eddy viscosity model via the Boussinesq hypothesis which relates to the Reynolds stresses as a function of turbulent viscosity. In order to calculate turbulent viscosity, various turbulent models are employed to solve one or more transported quantities: for example, in case of the modified turbulent viscosity (Spalart-Allmaras model); the turbulent kinetic energy k and turbulent dissipation rate ε (k-ε model),

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TABLE IV
HEAT TRANSFER COEFFICIENT OF VARIOUS COOLING SYSTEM.

Natural cooling			Reference
	Cylinder housing	$Nu=\begin{cases} 0.525(Gr\cdot Pr)^{0.25}; Gr\cdot Pr<10^9 \\ 0.129(Gr\cdot Pr)^{0.33}; Gr\cdot Pr>10^9 \end{cases}$	[21]
	Finned housing	$Nu=\begin{cases} 5.22\cdot 10^{-3}(Gr\cdot Pr\cdot NS/L)^{0.57}(S/L)^{0.412}(H/L)^{0.656}; 10^6<Gr\cdot Pr\cdot NS/L<2.5\cdot 10^7 \\ 2.78\cdot 10^{-3}(Gr\cdot Pr\cdot NS/L)^{0.57}(S/L)^{0.412}(H/L)^{0.656}; 2.5\cdot 10^7<Gr\cdot Pr\cdot NS/L<1.5\cdot 10^8 \end{cases}$	[89]
Forced cooling			
	Cylinder housing	$Nu=\begin{cases} 0.664Re^{0.5}\cdot Pr^{0.33};\ Re<5\cdot 10^5, 0.6<Pr<50 \\ (0.037Re^{0.8}-871)\cdot Pr^{0.33};\ Re>5\cdot 10^5 \end{cases}$	[21]
	Finned housing	Nu=$0.03Re^{0.8}\{1-0.23(L/S)^{0.5}(L-H/L)^{1.5}$; For turbulent flow}	[23]
	Housing jacket	$Nu=\begin{cases} 3.66+0.668Re\cdot Pr\cdot D/L\cdot/\{1+0.04(Re\cdot Pr\cdot D/L)^{0.667}\}; 2300<Re \\ 0.125m(Re-1000)\cdot Pr/\{1+4.49m^{0.5}(Pr^{0.667}-1)\}; 3000<Re<5\cdot 10^6 \end{cases}$ $m=(0.79\ln Re-1.64)^{-2}$	[90]
	Rotational hollow shaft	$Nu=\begin{cases} 0.019Re^{0.93}+8.51\cdot 10^{-6}Re_r^{1.45}; 0<Re<3\cdot 10^4, 1.6\cdot 10^3<Re_r<2.77\cdot 10^5 \\ 2.85\cdot 10^{-4}Re_r^{1.19};\ Re_r>2.77\cdot 10^5 \end{cases}$	[66]
	Spray cooling	$Nu=Pr^{0.4}\{0.785Re^{0.5}\cdot L/D\cdot A_r+0.0257Re^{0.83}\cdot L/L^*\cdot(1-A_r)\}; A_r=\frac{\pi(1.9d)^2}{L^2}, L^*=\frac{0.5\cdot(1+\sqrt{2})L-3.8d}{2}$	[72]
	Heat pipe	$Nu=4.728\cdot 10^{-7}Re^{1.986}\,c_p\Delta T/h_l$	[91]

TABLE V
THE APPLICATIONS AND COMPARISONS OF TURBULENCE MODELS.

Model	Applications	Comments	Time cost
Standard k-ε	[15] addresses the flow velocity, the drop in pressure and the HTC of water in the water jacket cooling channels.	Robust industry standard model, and only valid for fully turbulent flows. Performs poorly for complex flows involving separation and strong streamline curvature.	Low
Realizable k-ε	[92] studies the HTC between air and stator of an air-cooled machine; [93] investigates the characteristic curve of a fan.	Performance generally exceeds the standard k-ε model. Performs well for complex flows with large strain rates, but still suffers from the inherent limitations of an isotropic eddy-viscosity model.	↓
Standard k-ω	[94] investigates the HTC of end windings for an EFC motor.	The k-ω models improve performance for boundary layers as compared to k-ε. Suitable for wall-bounded boundary layer, free shear, separated and low-Reynolds number (i.e. transitional) flows. But sensitivity to freestream	↓
Shear stress transport (SST) k-ω	[95] studies the flow, thermal and windage losses characteristics of fan blades;[65] addresses the HTC associated with a shaft-cooling of the traction motors.	Performs well for swirling flows without requiring sublayer damping and improves the separation flow prediction. Less sensitive to freestream than the standard model.	↓
RST	[96] investigates the heat transfer on the outward corrugated tube.	The RST model has the potential to predict complex flows more accurately than eddy viscosity models because the transport equations for the Reynolds stresses naturally account for the effects of turbulence anisotropy, streamline curvature, swirl rotation and high strain rates.	High

the specific dissipation rate ω (k-ω model). In addition, the Reynolds stress transport (RST) model known as a second. In addition, the Reynolds stress transport (RST) model known as a second class model is based on a direct calculation of the stress-tensor components, requiring 5 additional transport equations. The applications and comparisons of the most used turbulent models are shown in Table V.

The friction coefficients and heat transfer coefficients are need to be predicted near the wall. Thus, it is crucially to choose the right combination of near wall mesh resolution and wall treatment. In addition, the turbulent flows are significantly affected by the presence of walls. The k-ε and RST models are mainly valid for turbulent core flows that occur away from the walls; and hence they are coupled with wall functions to bridge them with the calculated variables in the viscosity-affected region. The k-ω models are designed to be applicable throughout the boundary layer with a sufficient near-wall mesh. Moreover, the accuracy of the CFD analysis also depends on the quality of the data input by the user, e.g. the mesh size, the boundary condition and the material properties.

V. CONCLUSION

A summary of practical methods and apparatuses for various types of cooling of automotive traction motors has been presented. Depending on the capacity and installation conditions of an electric motor, a single or a number of cooling techniques can be applied accordingly. Besides the efficiency of a cooling system in enhancing the electromagnetic performance, the reliability, manufacturing complexity and maintenance costs of any proposed cooling architecture, are amongst the factors that must be considered in the design stage. In the future, a highly effective cooling and ventilating system is essential to meet the cooling

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demands placed on automotive traction motors with both small packaging as well as high torque and high power density.

Although the analytical approach is often limited to the geometrics and topologies and while the accuracy depends on a number of empirical correlations, this approach can still offer a fast and simple preliminary solution. If we use FEA effectively in modelling the solid conduction of heat, it is essential that a predicted heat transfer boundary needs to be defined. The CFD may provide detailed insights to into the kinds of outputs achieved from the cooling system, while being computationally laborious and time intensive. A combination of the empirical formulations and the computational technologies can benefit from both analytical and numerical approaches, whereupon a relatively simple and more accurate thermal model might be obtained.

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