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Centrifugal Fan Design for Permanent Magnet Synchronous Motor in a Traction Application

Tino Jercic, Damir Zarko, Marijan Martinovic, Marinko Kovacic, Josip Juric,
Zlatko Hanic, Stjepan Stipetic
Department of electrical drives and automation
University of Zagreb, Faculty of electrical engineering and computing
Zagreb, Croatia
tino.jercic@fer.hr, damir.zarko@fer.hr, marijan.martinovic@fer.hr, marinko.kovacic@fer.hr,
josip.juric@fer.hr, zlatko.hanic@fer.hr, stjepan.stipetic@fer.hr

***Abstract*— The requirements of high torque density and high efficiency, which are particularly pronounced in electric traction applications, often result in substantial thermal loading of electric machines for driving trams, electric multiple units (EMU) or electric cars. Permanent magnet synchronous machines are suitable candidates for traction applications due to their inherently high torque density and high efficiency. At the same time they are sensitive to temperature rise, especially in permanent magnets, highlighting the need for implementation of efficient cooling system. The performance of the cooling system and its ability to remove heat directly affect the attainable torque and efficiency of the electric machine. In this paper, the selection and sizing of the cooling system for an interior permanent magnet motor designed to drive a low-floor tram is presented. The procedure for selecting the basic dimensions of the centrifugal fan according to the analytical formulas in combination with computational fluid dynamics (CFD) analysis are explained. In addition to the geometry of the centrifugal fan itself, the geometry of the passive system components (e.g. air flow router) which have a significant impact on the performance of the cooling system, are also considered. The results of computer aided CFD analysis, which is taken as a benchmark of system performance in the design stage of the cooling system, have been confirmed with measurements on the machine prototype.**

***Topic*—electric vehicle, traction, totally enclosed fan cooled motor, centrifugal fan, computational fluid dynamics**

I. Introduction

In industrially developed countries electric motors are the major consumers of electrical energy so the increase of their efficiency is in the focus of international standardization bodies [1], [2], motor manufacturers and final users. The increase of efficiency with minimum increase of size, weight and cost of the motor can be achieved through utilization of better materials and through demanding design process. Very often during the design process, attention is paid mainly to electromagnetic calculations while thermal calculations and design of efficient cooling system are placed in the background. Therefore, the design of the cooling system usually occurs after the final version of electromagnetic optimization is done in order to satisfy the thermal limits of the critical parts (e.g. permanent magnets and winding insulation).

The area of great importance which puts high demands on electric motors in terms of overall performance is electric traction, where the major challenge for designers and also the major aggravating factor is the limited available space [3]. Current demands and trends in traction applications put a synchronous permanent magnet motor into focus due to its inherently higher torque density and efficiency compared to the rival types of electric machines [4], [5]. Despite these advantages, the concentration of losses in a small volume creates difficulties for the heat removal. Moreover, due to temperature sensitivity of the magnets, which in the end results in reduced performance with temperature increase [6], greater attention needs to be paid to the cooling system design [7].

Different cooling methods have different effects on heat removal, cost and lifetime of the machine. The most common types of traction motors that can be distinguished regarding cooling methods are: open type self-ventilated [8], open type independently ventilated [9], cooled by natural convection, totally enclosed fan-cooled [10], and liquid cooled [11].

In the case of liquid cooling, two approaches are utilized: direct and indirect cooling. In the case of direct cooling [12] the coolant is in the direct contact with the stator or rotor, while in the case of indirect cooling the coolant flows through channels in the machine housing [13], [14]. The indirect method suffers from problems that arise due to thermal contact resistance and different thermal expansion rates of stator core and the surrounding housing which can deteriorate the cooling performance.

In the case of electric traction, the main factors affecting the selection of the cooling type are: price, limited space and working conditions. This paper discusses the selection and design of a cooling system for an interior permanent magnet (IPM) synchronous machine intended for driving the electric tram Končar TMK2200. The electromagnetic design and thermal analysis of this motor are described in [15]. The main idea or guiding principle of this design project is to replace the existing open type (IP20) induction motor (which currently drives the TMK2200) with a totally enclosed (IP54) IPM machine of equal volume, but with 50 % higher torque rating in order to replace six induction motors with four IPMs retaining the same performance of the tram. This increased torque density of the traction motor along with the vehicle's demanding driving cycle creates an additional thermal load [16] emphasizing the importance of integrating the cooling system design with thermal and electromagnetic analysis. For the cooling system design the SolidWorks Flow CFD tool was used [17].

II. Ventilation System

The ventilation system can be divided into an active element (fan) which produces air flow in the cooling system and passive elements which create a resistance to the produced air flow affecting the cooling performance of the machine. The main parameter of interest when designing a cooling system is the volume rate of the air flowing through the cooling channels of the machine. The volume flow rate in ventilation systems can be determined by intersecting the fan and system resistance curves as shown in Fig.1.

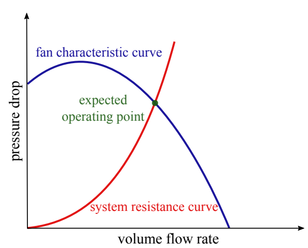

Fig. 1: Centrifugal fan and system resistance curves

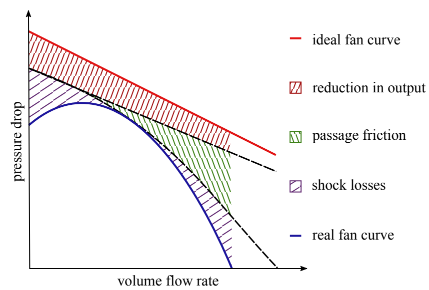

Fig. 2: Centrifugal fan losses

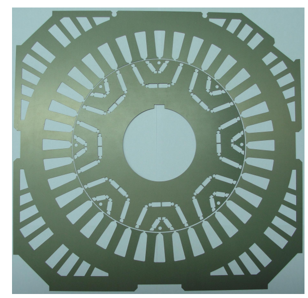

Fig. 3: Stator and rotor laminations of the IPM motor

The system curve, which is basically resistance to the flow produced by a fan, can be, for the known geometry of cooling channels, analytically calculated using the empirical formulas [18]. The obtaining of the fan characteristic requires either experimental measurements or the use of CFD analysis, which is a demanding process requiring the knowledge of fan design in advance.

By intersecting the fan and system resistance curves obtained independently a machine designer can acquire the air flow rate required for the subsequent thermal analysis. Since fan curve is generally obtained assuming uniform distribution of flow at the inlet and in reality the air inside the motor emerges from the cooling ducts located in the corners of the stator laminations (Fig. 3), this approach does not take into account losses arising due to uneven distribution of air flow at the fan entrance and additional losses throughout the system, such as losses at the entrance to the fan (clearance losses) and losses at the exhaust (diffuser losses), analysis of which is possible with the aid of CFD tools. The CFD analysis can be done by modelling the fan and the system geometry simultaneously thus providing more reliable results for the air flow rate in the design stage.

III. Centrifugal Fan Selection and Sizing

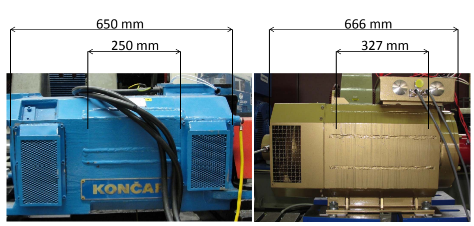

Fig. 4: Comparison of induction machine (left) and IPM machine (right)

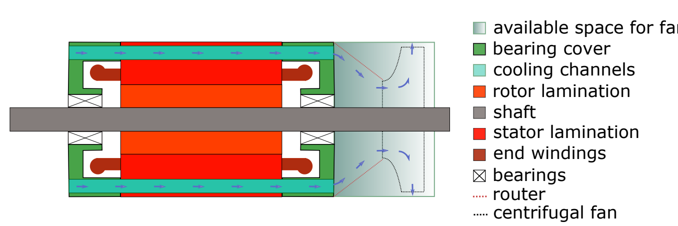

Fig. 5: Machine cross section

The rate of air flow in the cooling system can be influenced by altering the fan characteristic or the system resistance through intervention in the fan and system geometry. One of the major limiting factors when designing a ventilation system for the traction motor is available space on the vehicle bogie. Furthermore, the available space for the cooling system is determined after electromagnetic and mechanical analysis of the machine are conducted which yield the basic dimension such as: stator lamination outer diameter, lamination stack length, shaft diameter, end winding width and bearing cover width (Fig. 5). An additional requirement which restricts the ventilation system design is request for a complete enclosure of the IPM motor interior (IP54) in order to prevent the penetration of iron dust sticking to the rotor during operation. The iron dust particles are generated on the tracks by wear of the tram's wheels. Since height and width of the IPM motor and the existing induction motor must be the same, the dimensions and shape of the axial cooling channels on the periphery of the stator laminations of both motors are kept the same (Fig. 3). With predefined shape of the cooling channels and by taking into account the geometry of the stator laminations and bearing cover, the task of designing the cooling system reduces down to sizing of the fan and the router whose task is to direct the air flow from the cooling channels to the fan inlet. The router must be of such a shape to provide minimal resistance to the air flow while ensuring an even distribution of the air flow at the fan inlet and minimizing the possible occurrence of backflow and flow swirls. The uneven distribution of flow occurs because of the shape and distribution of the cooling channels on the periphery of the stator which reduces the fan capacity. The dimensions of the router chamber will, due to limited total available space for the cooling system, affect the fan dimensions and vice versa.

IV. Centrifugal Fan Selection and Sizing

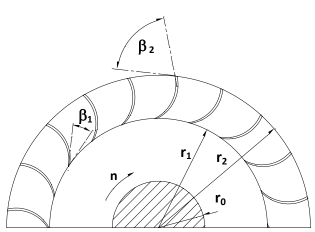

Fig. 6: Backward curved blade centrifugal fan (radial cross section)

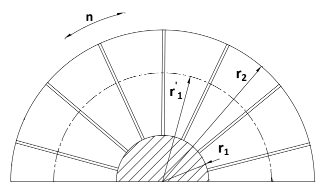

Fig. 7: Radial blade centrifugal fan (radial cross section)

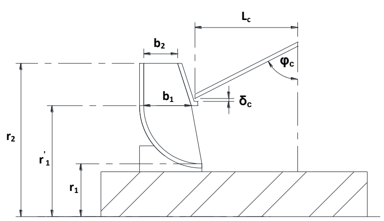

Fig. 8: Radial blade centrifugal fan (axial cross section)

There are three basic types of centrifugal fans distinguished by the shape of the impeller blades: radial, backward curved and forward curved. The choice depends on the application and the characteristics of a cooling system. Centrifugal fans with radial blades are generally of simple design with rugged construction and offer a minimum accumulation of dirt, do not require a scroll housing and have the same characteristics regardless of the direction of rotation. In the electric tram TMK2200 two motors are placed on a bogie rotating in opposite directions and driving two pairs of wheels via gearbox, where each pair of wheels is connected by a common shaft. Since the motor is close to the ground, it is exposed to dirty environment which requires a high level of robustness so centrifugal fan with radial blades is selected. For an infinite number of blades and neglecting the losses, the pressure vs. volume flow rate curve $\Delta p = f(V)$ of a centrifugal fan is according to Eck [19] given by

$$\Delta p_{th\,\infty} = \varrho u_2^2 - V \frac{\varrho}{g} \frac{u_2^2}{2\pi r_2 b_2 \tan\beta_2}, \tag{1}$$

where u_2 is the fan peripheral velocity $u_2 = (2\pi r_2 n)/60$, n is the rotational speed in rpm, ϱ is the density of the operating fluid, g is the gravitational acceleration, β_2 is the blade tip angle, r_2 is the impeller outer radius, and b_2 is the blade width at its tip. The centrifugal fan performance and its characteristic curve are affected by various types of losses occurring at the fan inlet, outlet and impeller itself. Among these, clearance losses, shock losses, and impeller friction losses along with reduction in output due to finite number of blades can be singled out as dominant factors affecting the fan curve (Fig. 2). The finite number of blades causes a drop in air guidance effectiveness resulting in reduction of theoretically attainable pressure. The calculation of change in pressure for finite number of blades is expressed with velocity coefficient ε, which is according to

[19] given as

$$\varepsilon = \frac{\Delta p_{th}}{\Delta p_{th\infty}} = \left[1 + \frac{1.5 + 1.1\beta_2/90}{z\left(1 - (r_1/r_2)^2\right)}\right]^{-1} \tag{2}$$

In order to minimize the reduction in output due to finite number of blades the ratio r_1/r_2 should be as small as possible. The reduction in output also depends on the number of blades z, for which [19] also provides guidelines for selection using

$$z = \frac{4\pi}{1,5}\frac{\sin\beta_2}{1 - r_1/r_2} \tag{3}$$

Shock losses can be divided into impeller entrance loss and guide vane loss. The shock losses at the inlet occur due to sudden change of flow direction (around 90°) as the air enters the eye of the impeller and are given as

$$\Delta p = \mu\frac{\varrho}{2}u_2^2\left(\frac{r_1}{r_2}\right)^2\left[\frac{V_x}{V} - 1\right]^2 \tag{4}$$

From (4) and Fig. 2 it is evident that the ratio of fan inlet radius to outer radius r_1/r_2 affects the slope of the fan curve in a way that longer impeller blades, i.e. smaller ratio r_1/r_2, produce flatter characteristic. Longer blades result in a larger fan mass and inertia and consequently larger windage losses of the electric machine. During exploitation of the analyzed machine, due to environmental conditions the clogging of the cooling channels is expected, thus increasing the system resistance. In order to maintain the flow rate at the satisfactory level even when cooling channels are clogged, it is desirable to have a flatter fan characteristic curve. According to (1), the larger is the peripheral velocity, which is proportional to impeller outer radius, the larger is the developed pressure of the fan resulting in higher flow rate. In order to maximize the fan performance, the outer radius r_2 should be selected as a maximum allowed value which is limited by available space for the motor. The impeller inner radius r_1 is indicated as the blade inner radius and usually coincides with the inlet radius as presented in Fig. 6. In order to obtain the flat fan characteristic, the largest impeller blade length is required, i.e. the minimum radius r_1 equal to the motor shaft radius should be used. If the impeller were designed according to Fig. 6, the fan with inlet radius equal to the shaft radius would be unable to produce any flow, which is avoided using the design according to Figs. 7 and 8 where inlet radius designated as r_1' is greater than the shaft radius.

The inlet radius r_1' also affects the dimensions of the router, namely the router angle φ_c, the increase of which results in decrease of the air flow resistance in the router. Longer router allows better developing of the flow, producing more even distribution of the flow at the fan inlet (Fig. 9) which results in better fan performance. However, due to limited space, the longer router length reduces the fan impeller width b_1 and b_2 which deteriorates the fan performance.

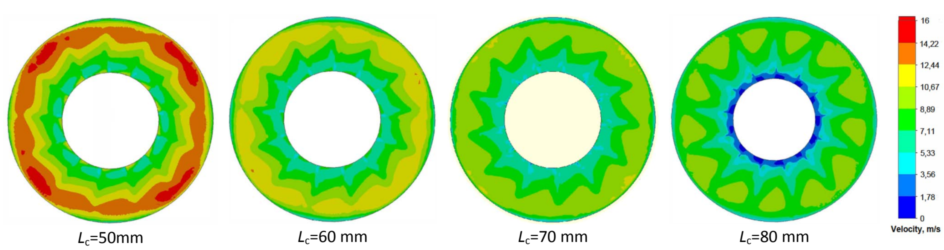

Fig. 9: Influence of router length L_c on the fan inlet flow/velocity distribution

V. Conclusion

This paper presents a two stage design of the centrifugal fan for an IPM traction motor. Analytical expressions are first used for basic sizing of the fan. Further optimization of the fan (and other cooling system segments) in order to produce the maximum flow rate was performed using the CFD analysis, resulting in dimensions presented in Table I according to which the fan prototype shown in Fig. 10 was built and installed into the IPM motor.

Unfortunately, due to production error the delivered router was not built according to the specifications given in Table I. Instead, the clearance gap between router and fan was $\delta_c = 8$ mm, resulting in the emergence of large clearance losses and

TABLE I: Centrifugal fan dimensions

Parameter	Value
r_1, mm	50
$r_1^{'}, mm$	105
r_2, mm	145
b_1, mm	45
b_2, mm	32
z	13
L_c, mm	70
δ_c, mm	1

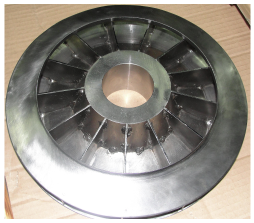

Fig. 10: Centrifugal fan prototype

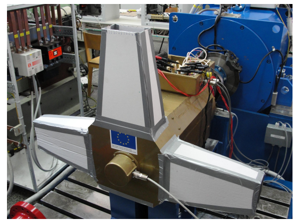

Fig. 11: Air flow measurement setup

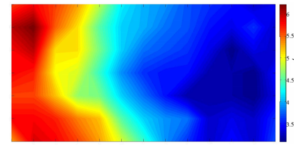

Fig. 12: Measured air velocity profile across the outlet surface

flow reduction. The measurement of air flow at the fan outlet resulted in the volume flow of $Q = 0.101\ \mathrm{m}^3/\mathrm{s}$ at the motor speed of $n = 1800$ rpm. The CFD analysis of the delivered machine with clearance gap of $\delta_c = 8$ mm resulted in the flow of $Q = 0.097\ \mathrm{m}^3/\mathrm{s}$, which compared to the measurements confirms the accuracy of the CFD analysis and justifies its use in the design of the selected cooling system. The design and optimization process of the centrifugal fan and other cooling system components (bearing cover, diffuser opening), along with results of the CFD analysis and measurements will be presented in more detail in the final paper.

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