ESTIMATION OF THE TEMPERATURE RISE IN HIGH-SPEED ELECTRICAL MACHINES WITH MAGNETIC BEARINGS

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ABSTRACT

The paper deals with the estimation of the temperature rise in high-speed electrical machines with active magnetic bearings. The calculation is based on a thermal network method. The input data for the model include the geometry, rotational speed, electrical power losses and cooling method of the motor and the bearings. The heat transfer coefficients and the friction losses are computed according to the input data.

The developed thermal model was tested in the case of a 36 kW and 50 000 rpm high-speed air compressor equipped with conical-type active magnetic bearings. The model seems to have a reasonable good accuracy if the roughness in the stator is taken into account.

INTRODUCTION

High-speed technology means small and powerfull machines in which the load is directly coupled to the motor. The idea is first to define the application, optimise its speed and power and after that design the driving motor. An adjustable speed is provided by a frequency convertor. When the shaft power of the motor is tens of kilowatts and the rotational speed is 30 000-100 000 rpm, contacless bearings have to be used. The choice between active magnetic, gas or hydrodynamic bearings depends mainly on the application.

The insulations for electrical machines are divided into classes A, B, F and H corresponding to the 60 K, 80 K, 105 K and 125 K temperature rise in the stator winding, respectively. If the continuous operating temperature is exceeded by 10 K from these values, the lifetime of the machine decreases to one half of the nominal value. The shaft power of electrical motors and the load capacity of magnetic bearings are limited by the temperature rise in the windings. A well designed motor has a good efficiency and operates near the maximum allowed temperatures. The temperature rise can be affected by minimising the losses and by an effective cooling of the machine.

This paper presents a computer program for estimating the temperature rise in high-speed electrical machines equipped with magnetic bearings. The thermal modelling is based on the thermal networks which are widely used for electrical motors. The principle of the method has been presented, for example, by Mellor et al [1] who developed a thermal model for totally enclosed fan-cooled induction motors. The difficulties in the thermal modelling of electrical machines are the calculation of the convective heat transfer and the contact resistancies appearing in the main heat flow paths.

In high-speed motors the surface speed of the rotor is typically between 150 and 400 m/s. At this range the friction losses are very high because they are proportional to the cube of the surface speed. The friction together with a high power-volume ratio are the reasons why high-speed motors often demand open circuir cooling. The most important aspects in the thermal modelling of high-speed machines are the heat transfer to the cooling air and friction losses

METHOD OF ANALYSIS

The starting point for an accurate temperature rise estimation is the knowledge of the power loss distribution in the motor and magnetic bearings. For high-speed motors with magnetic bearings this include both the electrical and friction losses, which have about the same order of magnitude.

Electrical losses

Due to the high, non-sinusoidal supply frequency and solid-rotor construction the electrical losses are difficult to define by using traditional theory and empirical formulae. In this study the electromagnetic analysis for the motor and active magnetic bearings have been done by numerical analysis programs based on the finite element method.

In the program the magnetic core of the machine is assumed to be two-dimensional. The endregion fields are modelled by constant end-winding impedances in the circuit equations of the winding. The time dependence of the magnetic field is modelled by the Crank-Nicholson method. The analysis programs have been reported by Arkkio [2] in more detail. The accuracy of the analysis method has been tested by normal 50 Hz motors supplied by a frequency convertor. The error limit for the calculation are studied to be less than ±10 % [3]. The error limits for active magnetic bearings are higher than for the 'more two-dimensional' electrical motors.

The most important result from the finite element analysis is the power loss distribution of the machine. In the case of the magnetic bearings the analysis method can be used to compute the forces caused by the electromagnets.

Friction losses

The analysis below concentrates on the power losses caused by the air flow in the airgap of a high-speed motor and active magnetic bearings. The Reynolds number for the tangential air-gap flow is given by the equation

$$Re = \frac{\rho\, \omega_r\, r_r\, \delta}{\mu} \tag{1}$$

where ρ is the density of the fluid, ω_r is the mechanical angular frequency, r_r is the radius of the rotor, δ is the radial air-gap length and μ is the dynamic viscosity of the fluid. A typical Reynolds number for the air-gap flow is 10^4–10^6. Thus, the flow is fully turbulent. The friction loss in the air gap is

$$P = c_f\, \pi\, \rho\, \omega_r^3\, r_r^4\, l \tag{2}$$

where c_f is the friction coefficient and l the length of the rotor. Yamada [4] have made measurements associated with the friction coefficient. He found the equation

$$c_f = \frac{0.0152}{Re^{0.24}} \tag{3}$$

valid for Reynolds number up to 10^5. Equation (3) applies for smooth rotor and stator surfaces. In addition to the air-gap friction, some losses are generated at the ends of the rotor. The Reynolds number for a rotating disc is

$$Re = \frac{\rho\, \omega_r\, r_r^2}{\mu} \tag{4}$$

Kreith [5] has collected information from several sources and gives the equation

$$c_f = \frac{0.15}{Re^{0.2}} \tag{5}$$

for the friction coefficient. The power loss for a rotating disc (one side wetted) is

$$P = \frac{1}{2}\, c_f\, \rho\, \omega_r^3\, (r_r^5 - r_{sh}^5) \tag{6}$$

where r_{sh} is the radius of the shaft.

Temperature rise

The thermal model for high-speed motors with magnetic bearings is based on the thermal networks. The structure of the developed model is presented in figure 1. The thermal model for the machine is put together from separate thermal networks for the electrical motor and magnetic bearings. The program has such a structure that the operator can choose first the cooling method (totally enclosed fan-cooled or open circuit cooling) and then the type of the magnetic bearings (conical or normal). All the separate thermal networks are combined into a matrix form

$$\begin{bmatrix} [G_{motor}] & & \\ & [G_{AMB}] & \\ & & [G_{cooling}] \end{bmatrix} \begin{bmatrix} T_1 \\ \cdot \\ \cdot \\ T_n \end{bmatrix} = \begin{bmatrix} [P] \\ [-P_c] \end{bmatrix}$$

where

G_{motor}	= conductance matrix for motor
G_{AMB}	= conductance matrix for magnetic bearings
$G_{cooling}$	= conductance matrix for cooling
T	= temperature rise vector (n nodes)
P	= power loss vector
P_c	= vector for power flows to cooling flow.

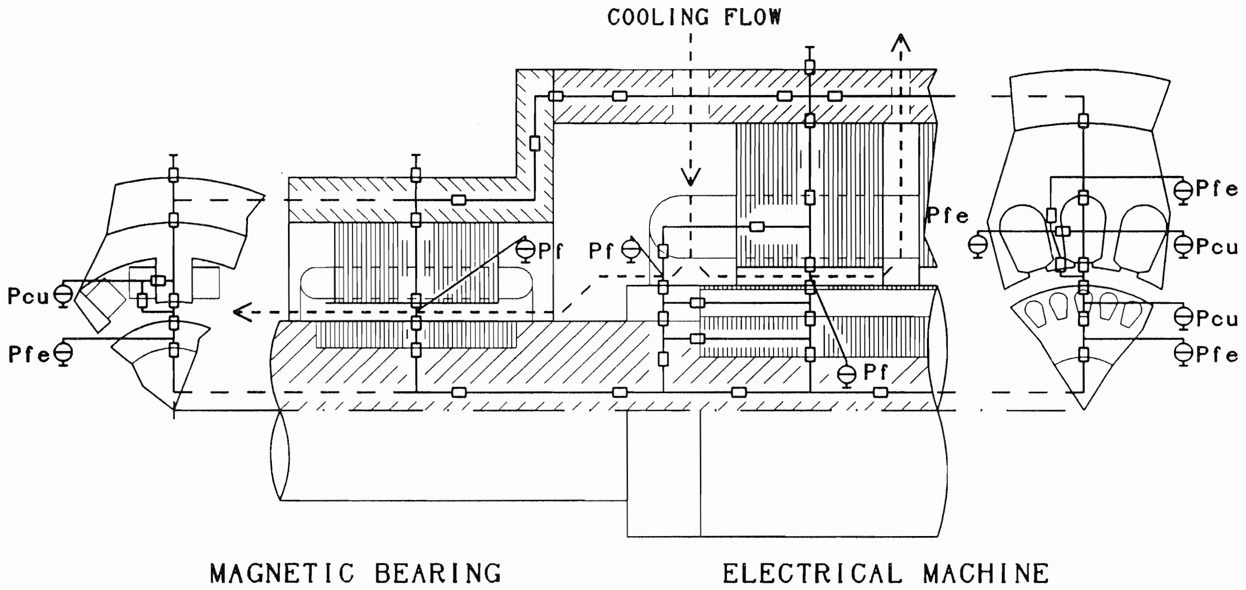

FIGURE 1: A simplified representation of the thermal network for high-speed motors with magnetic bearings. P_{cu}, P_{fe} and P_f are winding, core and friction losses, respectively. A typical number of nodes in a thermal model is 70.

The air space inside the motor and magnetic bearings is divided into 9 regions in which the temperature is homogenous. These are the middle channel, airgap of the motor, space between the endwinding and endring, endwinding space and airgap of the magnetic bearing. In the program the cooling air flow is modelled by using temperature controlled power sources.

Convective heat transfer

The convective heat transfer from the rotor and stator to the air-gap flow and from the stator endwindings to the surrounding air needs a special attention because of their location on the main heat flow paths. For the magnetic bearings the most sensitive is the heat flow from the rotor laminations to the airgap flow. In the thermal model the surfaces of the rotor and stator for the motor and bearings are assumed to be smooth. Based on the Reynolds analogy the Nusselt number for the rotor and stator has the form

$$Nu = \frac{1}{2} c_f Re \tag{7}$$

where the Reynolds number is defined in Eq(1) and friction coefficient in Eq(3). The heat transfer coefficient h can be calculated from equation

$$Nu = \frac{h\delta}{2k} \tag{8}$$

where k is the thermal conductivity and δ is the radial air-gap length. The part of the rotor located between the endring and magnetic bearing is treated as a rotating cylinder in free space. When the Reynolds number (Equation 4) is larger than 10^4, the Nusselt number is according to Kays and Bjorklund [6]

$$Nu = 0.095 Re^{0.667} \tag{9}$$

The heat transfer coefficient for a rotating cylinder in free space can be calculated from equation

$$Nu = \frac{hD}{k} \tag{10}$$

where D is the diameter of the cylinder. In the thermal network for the motor the endwinding is modelled as a toroid structure. The cooling air is blown radially against the endwinding. The heat transfer from the endwinding is modelled as a case of a cylinder in a crossflow. The Reynolds number for the endwinding is

$$Re = \frac{\rho v D}{\mu} \tag{11}$$

where D is the cross section diameter of the endwinding and v is the velocity of the air flow. The Nusselt number is

$$Nu = C Re^{m} Pr^{0.33} \tag{12}$$

where Pr is the Prandtl number. The values for the constants C and m depend on the Reynolds number. For instance, if $40<Re<4000$, C is 0.683 and m is 0.466. A complete table for the constants can be found from a book by Incropera and Witt [7]. The heat transfer coefficient h can be solved from Eq(11) and Eq(12).

Electrical motor
Shaft power P = 36 kW
Rotational speed n = 50 000 rpm
Electrical efficiency η = 94.7 %
Cooling air volume q_V = 2600 l/min

Magnetic bearings
Load capacity F = 200 N
Power input P = 100 W

Air compressor
Pressure ratio π = 1.7
Mass flow q_m = 0.510 kg/s
Efficiency η = 75 %

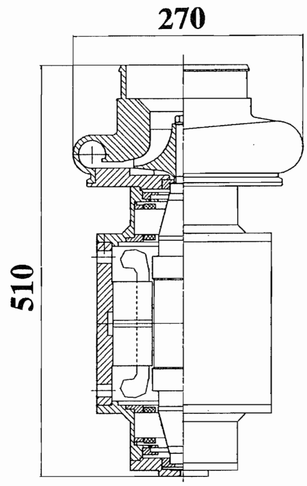

FIGURE 2: The test machine is a high-speed compressor equipped with conical-type active magnetic bearings. The rotor diameter is 71 mm corresponding the surface speed of 186 m/s at the rated rotational speed. The motor and the bearings have been developed in co-operation with High Speed Tech Ltd, Tampere, Finland.

RESULTS

Test motor

The accuracy of the developed thermal model was tested in a case of a high-speed air compressor. The main dimensions and rated values are presented in Figure 2. The electrical machine is an induction motor with a solid-rotor construction. The cooling of the test motor is provided by a external fan. The cooling air is blown into the motor through a radial ventilation duct in the middle of the stator core and taken out at the endwindings and magnetic bearings of the machine.

The motor is equipped with active magnetic bearings. The conical bearings are a robust construction where the axial and radial electromagnets have been combined. Because the conical-type magnetic bearings need less space than the normal ones, the length of the rotor is decreased thus increasing the first critical speed. The value of the first critical speed for the test motor is 1000-1200 Hz. A disadvantage in the conical bearings is their sensitivity to the thermal enlargement of the rotor. The reduction in the airgap length must be taken into account in the design of the control system.

Calculated and simulated temperature rises

The measurements were started by a retardation test. The power losses were defined from equation

$$P = J\omega_r \frac{d\omega_r}{dt} \tag{13}$$

where J is the inertia and ω_r is the mechanical angular frequency of the rotor. The losses calculated from Equation (13) are illustrated in Figure 3.

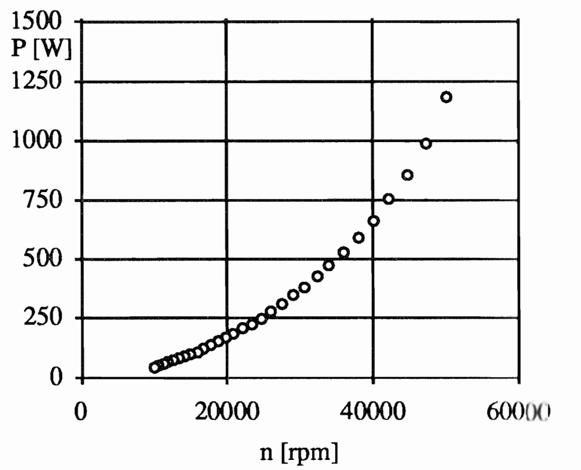

FIGURE 3: The power losses defined by a retardation test. The losses include the friction of the whole machine and the iron losses of the magnetic bearings. The iron losses were analysed to be between 30 - 70 W. The calculated friction losses (Equations (2) and (6)) are only about 50 % of the measured ones. The explanation for the high error is the roughness in the stator of the motor and especially in the magnetic bearings.

The measurements were continued by temperature-rise tests. The tests were performed by running the air compressor at four different speeds between 36 000 - 50 000 rpm. The flow rate of the cooling air was kept constant at 2670 l/min during all the tests.

Several thermocouples were installed in the electrical motor and magnetic bearings for the temperature-rise tests. The temperature of the stator winding was defined based on the traditional cooling down curves.

TABLE 1: Power losses used for estimating the temperature distribution in electrical motor and magnetic bearings of the air compressor. The electromagnetic losses are based on the finite element analysis.

Supply frequency (Hz) Rotational speed (rpm)	850 50 184	761 45132	693 41 070	611 36 186
Power input (kW)	38.9	27.4	20.0	14.8
Shaft power (kW)	35.6	25.0	18.1	13.4
Losses of the motor (W)				
Stator winding (core region)	320	239	179	143
Stator endwindings	628	470	382	281
Stator core (yoke and teeth)	193	180	132	105
Rotor bars (core region)	470	321	250	199
Rotor endrings	165	104	82	65
Rotor core (yoke and teeth)	284	189	147	104
Electrical losses of the motor	2060	1503	1172	897
Iron losses of the magnetic bearings	70	57	47	37
Friction losses	1110	793	653	503
Total losses (W)	3240	2353	1872	1437
Electrical efficiency	94.7	94.5	94.1	93.9
Total efficiency	91.7	91.4	90.6	90.3

In addition to the temperature rise the power input, voltage, current, $\cos\varphi$, supply frequency and rotational speed of the machine were measured. It took about one hour to reach the steady-state condition for one operation point. After the tests the electrical motor was analysed by using the measured slip and voltage. The results from the finite element analysis can be seen in Table 1.

The next step was to estimate the temperature rise in the motor and active magnetic bearings by the developed thermal model. The following figures present the calculated and measured temperature rise for some locations in the test machine.

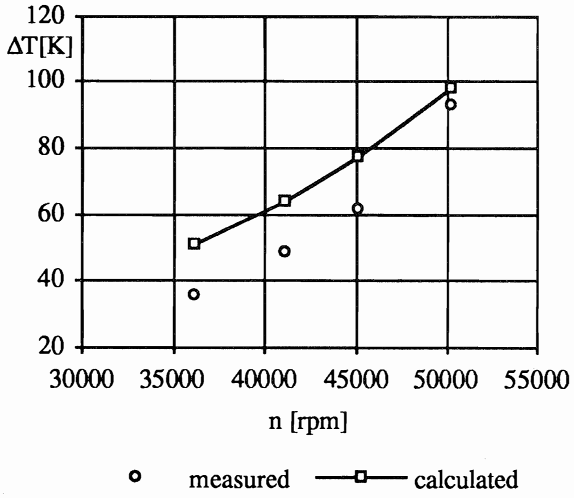

FIGURE 4: Average temperature rise in the stator winding. For the B-class insulation system the allowed temperature rise is 80 K, limiting the speed to 48 000 rpm with the used cooling. The relative error in the calculations is between ± 40 % and ± 5 %. and it is decreasing with the increasing speed.

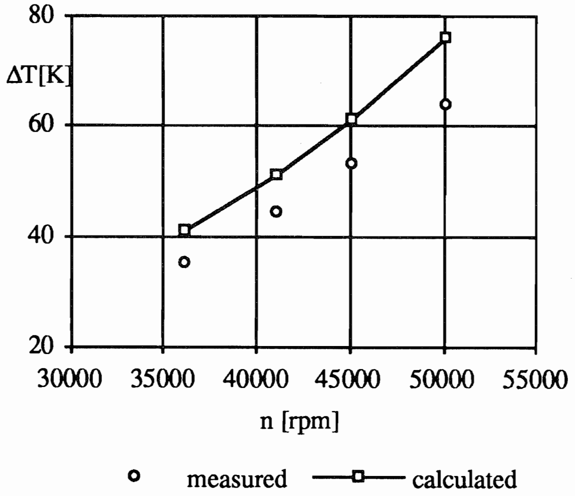

FIGURE 5: Temperature rise in the middle of the stator yoke. The value is strongly affected by the heat transfer from the stator core to the air-gap flow and from the endwindings to the cooling air. The relative error in the calculated temperature rise is ± 16 %.

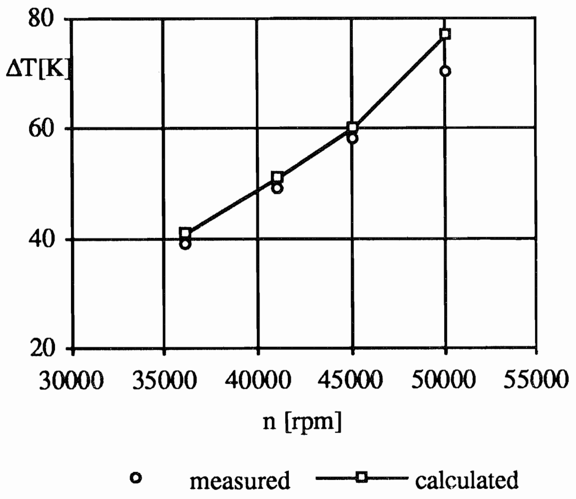

FIGURE 6: Temperature rise in the winding of the magnetic bearing. The calculated temperature rise is very close to the measured value. The electrical losses of the magnetic bearings are very low compared to the friction losses acting in their airgap.

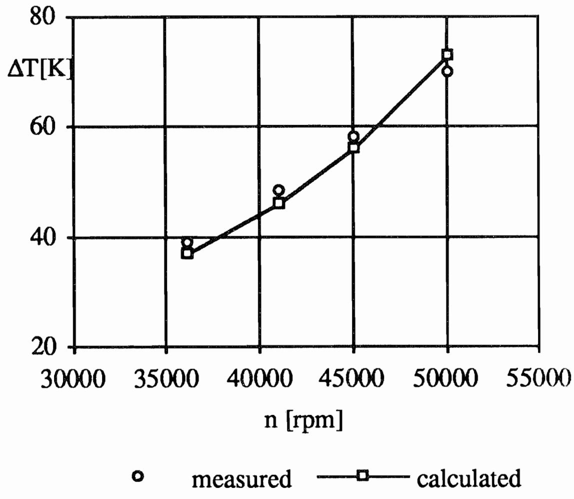

FIGURE 7: Temperature rise in the endwinding space. The good agreement between the measured and calculated temperatures indicates that the electrical and friction losses of the motor have correct values.

CONCLUSIONS

The paper deals a computer program for estimating the temperature rise in high-speed electrical motors equipped with magnetic bearings. The developed thermal model is based on thermal networks.

The high friction losses and convective heat transfer to the cooling air need a special attention in the thermal modelling of high-speed machines. In the developed program these values are calculated from the input data. The thermal model was tested for a 36 kW high-speed air compressor running at speeds between 36 000 - 50 000 rpm. After the temperature-rise tests the electromagnetic study for the machine was done by numerical analysis programs based on finite element method. The equations for smooth rotor and stator surfaces gave only about 50 % of the measured friction losses.

During the thermal simulations it was found that by assuming a smooth stator and rotor the heat transfer coefficient from the stator to the air-gap flow get a too low value. After doubling the coefficient, the calculated temperature rises showed some agreement with the measured results. It was also seen from the simulations that the total loss of the machine has a correct value.

The losses of the active magnetic bearings are very low compared to the losses of the electrical motor. Only the friction losses acting in the air gap of the bearings can cause some thermal problems.

REFERENCES

1. Mellor P.H., Roberts D. and Turner D.R., Lumped parameter thermal model for electrical machines of TEFC design, IEE Proceedings-B, Vol. 138, No. 5, September 1991, pp. 205-218.
2. Arkkio A., Analysis of induction motors based on the numerical solution of the magnetic field and circuit equations. Helsinki 1987, Acta Polytechnica Scandinavica, Electrical Engineering Series, No. 59, 97 p.
3. Arkkio A. and Niemenmaa A., Estimation of losses in cage induction motors using finite element techniques, in Proceedings of the International Conference on Electrical Machines, 15–17 September 1992, Manchester, UK. Vol. 2, pp. 317–321.
4. Yamada Y., Torque resistance of a flow between coaxial cylinders having axial flow, Bulletin of JSME, Vol. 5, No. 20, 1962, pp. 634–642.
5. Kreith F., Convection heat transfer in rotating systems, Advances in Heat Transfer, Vol. 5, Academic Press Inc., New York, USA, 1968, pp. 129–251.
6. Kays W.M. and Bjorklund I.S., Heat transfer from a rotating cylinder with and without crossflow, Transactions of the ASME, Vol. 80, January 1958, pp. 70-78.
7. Incropera F.P., De Witt D.P., Fundamentals of heat and mass transfer, 3rd edition, John Wiley & Sons, Inc., Singapore, 1990, 919 p.