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THERMAL MODELING OF A MINI ROTOR-STATOR SYSTEM

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ABSTRACT

In this study the temperature increase and heat dissipation in the air gap of a cylindrical mini rotor stator system has been analyzed. A simple thermal model based on lumped parameter thermal networks has been developed. With this model the temperature dependent air properties for the fluid-rotor interaction models have been calculated. Next the complete system has also been modeled by using computational fluid dynamics (CFD) with Ansys-CFX and Ansys. The results have been compared and the capability of the thermal networks method to calculate the temperature of the air between the rotor and stator of a high speed micro rotor has been discussed.

NOMENCLATURE

c_f Friction coefficient
G Thermal conductance matrix
h Convective heat transfer coefficient
k Thermal conductivity
Nu Nusselt number
P Vector of power losses
P_f Power loss due to fluid friction
Re_r Tip Reynolds number
Re_δ Couette Reynolds number
R_{ia} Axial thermal resistance
R_{ir} Radial thermal resistance
r Rotor radius
T Vector of nodal temperatures
Ta Taylor number
Tr Rotor outer surface temperature
Ts Stator inner surface temperature
δ Air gap
Ω Rotation speed
μ Dynamic viscosity
τ Shear stress
ρ Density
θ_i Temperature of the i th node

INTRODUCTION

Recently, there has been a trend to develop mini rotating machinery which operates at high speeds. However as the rotation speed increases, the heat dissipation due to air friction and the temperature increase in the air gap between rotor and stator becomes more significant. Fig.1 illustrates the simple rotor-stator with the air gap in between. Friction losses in the air gap of a rotor stator system are resulting from viscous flow. Air friction loss is determined by the velocity field and air properties. The fluid velocity field is calculated by Navier-Stokes and continuity equations. These equations can be solved analytically for simple geometries in laminar flow.

However for turbulent flow which is generally observed in high speed mini rotating machinery, these equations become more difficult to solve. Therefore numerical methods and semi-empirical correlations are frequently used to solve turbulent flow equations.

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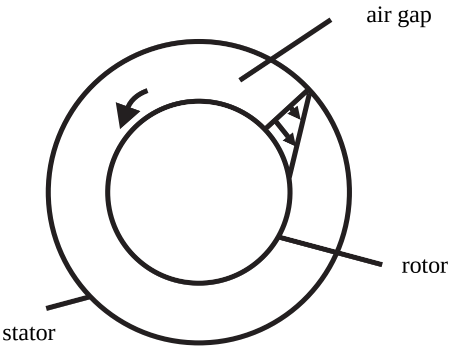

Figure 1. ROTOR-STATOR AND AIR GAP

In this analysis a thermal model based on thermal networks method has been developed to calculate the steady state temperature of the air between rotor and stator. Then a more complex analysis has been done by using commercially available tools. The lumped parameter thermal networks method has been extensively used for a long time for thermal analysis of electric motors and generators. Advantages of this simple model are: thermal networks are easy to construct, and it requires less computation time compared to other methods. The thermal networks involve nodes describing the mean temperature of each component and resistances between them. Each component is modeled by independent axial and thermal networks and heat generation in the component is applied to the node describing the mean temperature of the component.

In this study the rotor and stator have been modeled by thermal networks at steady state, and the air in between has been described by a node. At each rotation speed the heat dissipation due to air friction has been calculated via empirical friction coefficients which are functions of Couette-Reynolds number and Taylor number. Then heat dissipation is applied to the node representing the air in the gap between rotor and stator. In this way the temperature of the air has been calculated at different rotation speeds. For CFD analysis the air gap is modeled by using Ansys CFX and the rotation speed and estimated steady state temperature of the rotor and stator surfaces are applied as boundary conditions. The temperature in the gap and the convective heat transfer coefficients between the air-rotor and air-stator are calculated at each speed with initially assumed rotor and stator surface temperatures. Then convective heat transfer coefficients are imported to Ansys and the steady state temperatures of the rotor and stator surfaces are calculated. The updated boundary conditions (rotor-stator surface temperatures) are imported to CFX and the air temperature is recalculated. This procedure is continued till results converge. Then the results are compared with thermal networks and the capability of thermal networks method to calculate the temperature of the air between the rotor and stator of a high speed micro rotor is discussed.

THERMAL NETWORKS

Air Friction Loss Calculation

For high speed rotating machinery, most of the total loss occurs as a result of friction with the surrounding air. It is important to estimate the air friction losses in order to determine the temperature distribution in the machine and design the rotor for maximum efficiency. The behavior of the gas flow depends on the inertia and viscous forces. The ratio of the inertia and viscous forces is the non dimensional Reynolds number. For a rotating shaft in free space, the relevant Reynolds number is called the tip Reynolds number and it is defined as:

$$Re_r = \frac{\rho \Omega r^2}{\mu} \tag{1}$$

where ρ is the density, μ is the dynamic viscosity, Ω is the rotational speed and r is the shaft radius. However the behavior of the flow in the air gap of a rotor-stator system is determined by the Couette-Reynolds number which is defined as:

$$Re_\delta = \frac{\rho \Omega r \delta}{\mu} \tag{2}$$

where δ is the air gap in radial direction. Due to the centrifugal force on the fluid particles, circular velocity fluctuations (Taylor vortices) appear in the air gap. At low speeds the flow is laminar and the creation of Taylor vortices is damped by frictional forces [1]. Taylor vortices occur when the critical Taylor number of 1700 is exceeded [2]. The Taylor number is defined as:

$$Ta = \frac{Re_\delta^2 \delta}{r} = \frac{\rho^2 \Omega^2 r \delta^3}{\mu^2} \tag{3}$$

If $Re_\delta < 2000$ and $Ta < 1700$, the laminar two-dimensional Couette flow theory is valid, when $Re_\delta < 2000$ and $Ta > 1700$, the flow is still laminar, but three-dimensional Taylor vortices are present; if $Re_\delta > 2000$, the flow is turbulent. Due to high rotation speeds, the turbulent regime is widely observed in the air gaps of mini rotating machinery. The shear stress is difficult to solve in turbulent flows. Therefore empirical friction coefficients are defined and used to calculate the shear stress and power loss due to friction. Correlations for empirical frictional coefficient are defined as a function of Reynolds number. The friction coefficient and power loss due to friction are:

$$c_f = \frac{\tau}{\frac{1}{2}\rho \Omega^2 r^2} \tag{4}$$

$$P_f = c_f \pi \rho \Omega^3 r^4 l \tag{5}$$

There has been a great number of studies available in the literature for calculation of the friction coefficient of a rotating cylinder. Saari [3] made a literature review of friction losses and heat transfer between concentric cylinders. One of the initial studies about the friction torque of a rotating cylinder in free space is made by Theodorsen and Regier [4]. In another study, Bilgen and Boulos [5] have measured the friction torque of smooth concentric enclosed cylinders. They developed the following correlations for friction coefficient as a function of Couette-Reynolds number.

$$c_f = 0.515\frac{\left(\frac{\delta}{r}\right)^{0.3}}{Re_\delta^{0.5}} \qquad (500 < Re_\delta < 10000) \tag{6}$$

$$c_f = 0.0325\frac{\left(\frac{\delta}{r}\right)^{0.3}}{Re_\delta^{0.2}} \qquad (10000 < Re_\delta) \tag{7}$$

In our study, the rotor and stator surfaces are assumed to be smooth ignoring the roughness effects on the friction. The correlations above are used to determine the heat generation due to air friction for the simple rotor-stator system. The Couette-Reynolds number is calculated at each rotation speed, then the friction coefficient and power dissipation due to air friction are computed.

Thermal Analysis of a Cylindrical Rotor-Stator

The thermal networks method is applied in this study due to its advantages over other methods [1]:

- Less computation time is required
- Thermal networks are easy to build
- Equations for the friction losses and the convection heat transfer coefficients can easily be implemented

Thermal networks are widely applied for thermal analysis of electric motors and generators [6–9]. Perez and Kassakian [10] modeled each component of a high speed synchronous machine in terms of a thermal node that approximates the mean temperature of the component. Mellor [11] et al. described a similar thermal model for both steady-state and transient analysis. Kylander [12] presented a thermal model for enclosed electric motors. The thermal networks involve nodes describing the mean temperature of each component. All the heat generation in the component is applied to the node describing the component. Each component is modeled with independent axial and radial thermal networks with the resistances for conductive and convective heat flow. Then thermal networks representing each component are assembled in order to perform a thermal analysis of the complete system. Fig. 2 illustrates independent axial and radial thermal networks for a general cylindrical component [11]. The heat transfer equations have been written for each node as:

$$\begin{aligned}
&\frac{1}{R_{1a}}\theta_3 + \frac{1}{R_{2a}}\theta_4 + \frac{1}{R_{3a}}\theta_m - \left(\frac{1}{R_{1a}} + \frac{1}{R_{2a}} + \frac{1}{R_{3a}}\right)\theta_5 = 0\\
&\frac{1}{R_{1r}}\theta_1 + \frac{1}{R_{2r}}\theta_2 + \frac{1}{R_{3r}}\theta_m - \left(\frac{1}{R_{1r}} + \frac{1}{R_{2r}} + \frac{1}{R_{3r}}\right)\theta_6 = 0\\
&\frac{1}{R_{3a}}\theta_5 + \frac{1}{R_{3r}}\theta_6 - \left(\frac{1}{R_{3a}} + \frac{1}{R_{3r}}\right)\theta_m = -Heat
\end{aligned} \tag{8}$$

These equations have been written in matrix form and the vector of the nodal temperatures has been obtained from the equation:

$$\mathbf{P} = \mathbf{GT} \tag{9}$$

where G is the matrix of thermal conductances. P is the power loss vector and T is the temperature vector.
The thermal networks method and applications has been explained in detail in many studies [9–14].

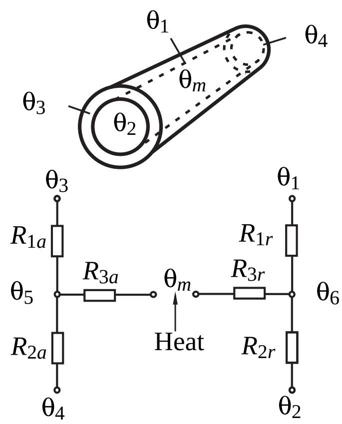

Figure 2. AXIAL AND RADIAL THERMAL NETWORKS

The conductive resistances in the structure are constant, however the convective resistances between the air-rotor and air-stator surfaces change with the rotation speed. The convective heat transfer coefficient which is used for the calculation of convective resistances is given as [1]:

$$h = \frac{2kNu}{\delta} \tag{10}$$

where Nu is the Nusselt number and k is the thermal conductivity of the fluid. The Nusselt number for tangential air flow between concentric cylinders is given by Becker and Kaye [15] as a function of the Taylor number:

$$Nu = 0.128Ta^{0.367} \qquad (1700 < Ta < 10^4) \tag{11}$$

$$Nu = 0.409Ta^{0.241} \qquad (10^4 < Ta < 10^7) \tag{12}$$

In this study a thermal network corresponding to a simple rotor stator system has been constructed by using the component resistances developed by Saari [1]. A simple Matlab based code has been developed for calculation of air friction and temperature increase. The material properties, dimensions and rotation speed are inputs of the program. The convection heat transfer coefficients are calculated in the program and used for determination of the resistances between air-rotor and air-stator. Then heat flow equations are solved at each node [16] and the nodal temperatures are calculated. In this way the mean temperatures of the rotor-stator, air and heat generation due to air friction are calculated.

ANSYS-CFX COUPLED THERMAL ANALYSIS

The thermal analysis of the rotor-stator system has also been performed by using the commercial software packages ANSYS CFX and ANSYS Workbench. The rotor and stator are modeled in ANSYS Workbench and the air film in between has been modeled in ANSYS CFX. In order to calculate the steady state air temperature in the gap between rotor and stator, rotor outer and stator inner surface temperatures are required in ANSYS CFX. These boundary conditions are initially estimated in ANSYS CFX and then calculated in ANSYS Workbench. Since only one way coupling is possible between these packages initial assumptions for the rotor and stator surface temperatures have been made, heat generation due to air friction, heat transfer coefficients and air temperature are calculated in ANSYS CFX. Then heat transfer coefficients are transfered into ANSYS Workbench to calculate the assumed rotor and stator surface temperatures. The procedure continues till initial assumptions and final results agree each other. The procedure is shown in Fig.3.

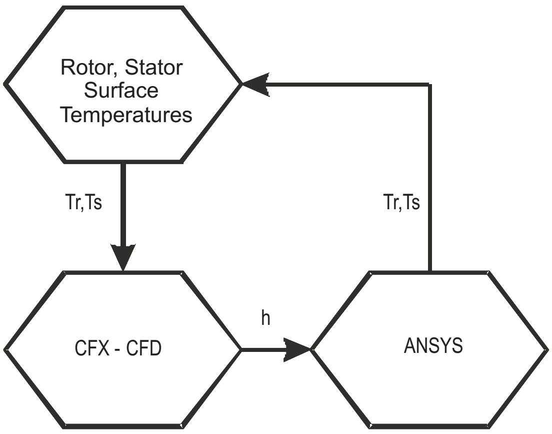

Figure 3. ANSYS-CFX COUPLED THERMAL ANALYSIS PROCEDURE

The fluid film has been modeled in ANSYS CFX as shown in Fig.4. The analysis has been run with different mesh sizes

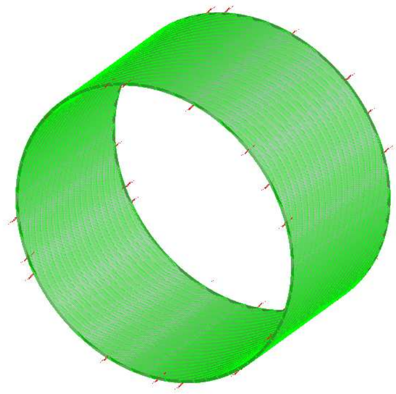

Figure 4. CFX MODEL FOR THE AIR

and suitable mesh size has been determined as the convergence achieved. The rotation speed of the rotor outer surface, the temperatures of the rotor and stator surfaces have been applied as the boundary conditions. The total energy formulation including the viscous terms has been used for heat transfer equations since it is suitable for flows with Mach number greater than 0.2. The k-ε turbulence model has been used since it is appropriate for internal flows and offers a good compromise between numerical effort and computational accuracy. Simulations have been performed at each rotation speed and the air temperature profile has been obtained. The convective heat transfer coefficients have been exported to ANSYS for further analysis to check initially assumed boundary conditions.

The rotor and stator have been modeled in Ansys Workbench (see Fig. 5). Thermal analysis of the structure has been done by applying the ambient temperature to the side walls and importing the convective heat transfer coefficients from the ANSYS CFX solutions. The temperature of the rotor and stator surfaces has been computed and the initially assumed temperatures used in ANSYS CFX have been updated.

SIMULATION RESULTS

The simulations have been performed by using both thermal networks and ANSYS CFX based CFD analysis for a rotor with a radius of 25 mm, length of 30 mm and air gap of 0.5 mm. Tab.1 compares the CFD analysis with the thermal networks method. The temperature profile in the air gap is computed in ANSYS CFX, then the average of the air temperature profile is calculated

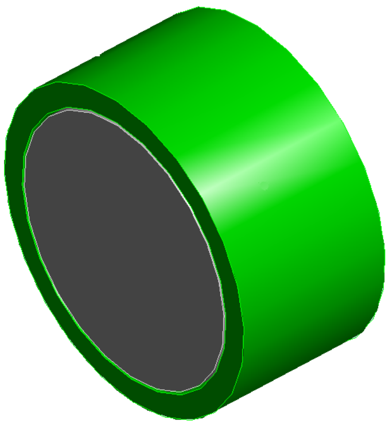

Figure 5. ROTOR AND STATOR

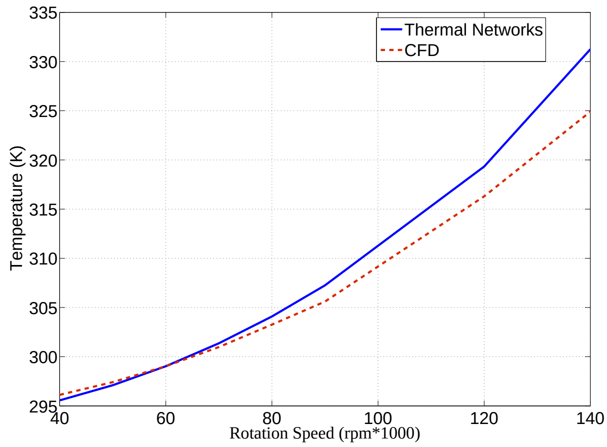

Figure 6. THERMAL NETWORK AND CFD RESULTS

Table 1. CFD vs THERMAL NETWORKS

CFD	THERMAL NETWORKS
Many DOF	One node for modeling the air
Local Temperature Distribution	Global Temperature distribution
Much Computation time	Less computation time

and compared to the results obtained by using thermal networks (see Fig.6). For the thermal networks, the air in the gap is modeled as a node and the computation time to calculate the temperatures corresponding to the mid-rotor, stator and air has been 0.06 sec. On the other hand the CFD model constructed using ANSYS CFX involves 3072 elements, 6144 nodes and the computation time has been 214 seconds.

There is fair agreement between both methods. The difference at higher rotational speeds is a further research issue. The thermal networks method gives reasonable estimates of the air temperature. The updated air temperature is used to renew the air poperties at the specific rotation speed for air-rotor coupled dynamic analysis.

CONCLUSIONS

Thermal analysis of a simple cylindrical rotor and stator system has been performed by using thermal networks and CFD. The thermal networks are simple to construct and can be easily coupled with other analysis methods. The temperature rise of the air in the gap between a mini rotor and stator due to air friction has been calculated by using thermal networks and more complicated CFD method. The simulations are performed and acceptable agreement between the results using both methods has been obtained.

Thus, thermal networks method seems to be appropriate to be implemented into fluid rotor interaction models to update temperature dependent air properties for further analysis.

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