

FACULDADE DE ENGENHARIA DA UNIVERSIDADE DO PORTO

Enhancing Electric Motor Shaft Alignment: A Comprehensive Study and Tool Development in Flameproof Electric Motors

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Mestrado em Mecânica Computacional

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Abstract

This dissertation investigates the optimization of motor shaft axial alignment and consequently the alignment of the magnetic field, with a specific focus on flameproof electric motors. Positioned as the culmination of the Computational Mechanics Master's program, the study aims to address the challenges associated with the axial misalignment in motor shafts relative to the stator.

The methodology involves a detailed evaluation of existing motor designs, notably those manufactured by WEG, a leading industry manufacturer. The analysis of WEG's motor specifications will guide the study, concentrating on developing a tool to align the rotor. This thorough analysis will provide insights into the complexities of current designs and serve as a basis for proposing alternative solutions.

Furthermore, the dissertation aims to conduct a detailed evaluation and simulation the design alternative, taking into account factors such as efficiency and durability. The development of a computational simulation procedure will enable a detailed analysis of the proposal alternative, facilitating assessment of its strengths and weaknesses. This approach ensures a holistic exploration of potential solutions and establishes the foundation for proposing and validating an alignment tool tailored to the specifics of the selected motor design.

Keywords: Motor shaft axial alignment, Flameproof electric motors, Simulation of the design alternative.

Resumo

Esta dissertação investiga a otimização do alinhamento axial do eixo do motor e consequentemente o alinhamento do campo magnético, com um foco específico em motores de atmosfera explosiva. Posicionada como a culminação do programa de mestrado em Mecânica Computacional, o estudo visa abordar os desafios associados aos desalinhamentos axiais nos eixos dos motores em relação ao estator.

A metodologia envolve uma avaliação detalhada de projetos de motores existentes, principalmente aqueles fabricados pela WEG, uma principal fabricante da indústria. A análise das especificações dos motores da WEG orientará o estudo, concentrando-se no desenvolvimento de uma ferramenta para alinhar o rotor. Essa análise abrangente proporcionará insights sobre as complexidades dos designs atuais e servirá como base para propor soluções alternativas.

Além disso, a dissertação visa realizar uma avaliação detalhada e simulação da alternativa de design, levando em consideração fatores como eficiência e durabilidade. O desenvolvimento de um procedimento de simulação computacional permitirá uma análise detalhada da alternativa proposta, facilitando uma avaliação dos pontos fortes e fracos. Essa abordagem assegurará uma exploração holística de soluções potenciais e estabelece a base para propor e validar uma ferramenta de alinhamento adaptada às especificidades do design do motor selecionado.

Palavras-chave: Alinhamento axial do eixo do motor, Motores de atmosfera explosiva, Simulação da alternativa de design.

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I am grateful to my internship colleagues for their companionship throughout this journey where we shared many great moments together. Last but not least, I want to thank my parents for giving me the opportunity to pursue this Master's degree and for their unwavering support, as well as my friends for their constant encouragement.

Gonçalo Barbosa dos Santos

*“Character may be manifested in the great moments,
But it is made in the small ones.”*

Phillips Brooks

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Abbreviations

f	Frequency: Number of cycles per second
$U_{\max}$	Maximum Voltage: Peak voltage in a cycle
$I_{\max}$	Maximum Current: Peak current in a cycle.
RMS	Root Mean Square: Voltage and Current (U and I). Continuous values equivalent to DC power.
$\cos \phi$	Power Factor: Cosine of the phase angle between voltage and current
T_m	is the mechanical torque in Nm
P_m	is the power developed by the motor in watts
N	is the motor speed in revolutions per minute (RPM)
η	Efficiency
PF	Power Factor
Nu	Nusselt number
Re	Reynolds number
Pr	Prandtl number
Gr	Grashof number
FEA	Finite Element Analysis
FEM	Finite Element Method
CFD	Computational Fluid Dynamics
f_n	Natural frequency
k	Stiffness Matrix of the system
m	Mass of the system
ζ	Damping ratio
$x(t)$	Displacement of the system at time t
X	Amplitude of the excitation
ω	Angular frequency of the excitation
ϕ	Phase angle
EI	Flexural rigidity of the beam
w	Deflection of the beam along the x-axis
$q(x)$	Distributed load on the beam
k_f	Foundation modulus
$\ddot{x}$	Second derivative of displacement with respect to time (acceleration)
$\dot{x}$	First derivative of displacement with respect to time (velocity)
P	Applied load
$M(x)$	Bending moment at position x
$20^\circ C$	twenty degrees celsius
nP	number of poles
WWW	World Wide Web

Chapter 1

Introduction

I started working as an intern in the Product Research and Development Department at WEG's Factory 2 in Santo Tirso, Portugal. The internship duration spans from February 19, 2024, to June 30, 2024. The focus of my internship aligns with the theme of my dissertation, "Enhancing Electric Motor Shaft Alignment: A Comprehensive Study and Tool Development in Flameproof Electric Motors".

This internship offers a unique opportunity to delve into the practical aspects of electric motor development, contributing to WEG's commitment to technological advancement and sustainability.

1.1 Company Presentation

Founded on September 16, 1961, in Jaraguá do Sul, Brazil, by Werner Ricardo Voigt, Eggon João da Silva, and Geraldo Werninghaus, WEG has evolved into one of the world's largest manufacturers of electrical equipment. Initially producing electric motors, the company expanded its portfolio in the 1980s to include electrical and electronic components, products for industrial automation, power and distribution transformers, liquid and powder coatings, and electro-insulating varnishes.

The trajectory of the organization, propelled by the vision and determination of its founders, is marked by success. The fearless, dynamic, and grand essence of the company sustains its journey towards triumph. The commitment to values, beliefs, and ideals upheld by the founders remains ingrained in the organization, guiding its victorious paths throughout history.

The overarching vision at WEG is to be a global reference in electric motors, generators, transformers, and electric drives, offering a comprehensive integrated range for electrification, automation, and digitization. This strategic direction positions WEG as a global brand known not only for delivering efficient and complete solutions but also for fostering strong client relationships.

WEG's core values encompass People and the Environment, Participative Management, Governance, Efficiency with Simplicity, Innovation and Flexibility, as well as Leadership and Culture. These values underscore the company's commitment to professional and personal development, environmental stewardship, participatory decision-making, transparent governance, efficiency, innovation, flexibility, leadership development, and a strong organizational culture.

As part of its commitment to sustainable and ethical business practices, WEG has articulated policies addressing sustainability, quality, environmental conservation, energy efficiency, health and safety, and social responsibility. These policies underscore WEG's dedication to upholding legal standards, eradicating child and forced labor, promoting employee development, fostering diversity and multiculturalism, preventing harassment, respecting the right to association, and supporting the communities where it operates.

For WEG, technological innovation revolves around the development of new technologies, products, materials, and tests aimed at increasing efficiency, reducing costs, improving quality, and minimizing raw material usage. In 2021, the company invested R\$ 548 million in Research and Development, showcasing a robust commitment to innovation.

WEG follows structured programs, such as the Program of Technological Development (PDT), to organize and systematize innovative projects related to new products and processes. These programs ensure that all ideas are methodically organized within the framework of the WEG Quality and Productivity Program (PWQP)

In Portugal, WEG has been a prominent player since 2002, initially acquiring a factory in Maia and later expanding its operations with a sophisticated facility in Santo Tirso in 2018. The main solutions in Portugal are: Motors & Energy, Automation, Digital Solutions, Transmission & Distribution, Services and Coatings. The company more recently constructed a new 22,680-square-meters facility in Santo Tirso, consolidating its operations and reinforcing its commitment to the region.

This expansion aims to enhance production capacity, particularly in large electric motors, and involves relocating the Maia unit to Santo Tirso. This investment was significant to modernizing operations, introducing medium and high-voltage motors, electrical panels, automation solutions, and technical support services to Santo Tirso. The move also aligns with plans to broaden the range of electric motors, especially in larger sizes and high voltage [1].

1.2 Contextualization of the Study

The present agreement aims to associate in the Computational Mechanics program, with a tenure at the Company to undertake the dissertation, in accordance with the stipulations of its regulations, as outlined in Article 9 in the Contract for the Completion of a Master's Dissertation in a Business Environment. This collaboration provides an invaluable opportunity for the student to delve into a real-world industrial setting, specifically within the domain of the Company's Product Research and Development Department.

1.2.1 General Objective

The primary objective of this collaboration is to provide an immersive experience within the Company, fostering the execution of a dissertation aligned with the regulations of the Computational Mechanics program. The undertaking is designed to contribute substantively to the academic and professional growth. Additionally, it includes the development of a practical tool for electric motor shaft alignment, and consequently, the magnetic field, briefly summarized below.

Tool Summary: The developed tool will primarily focus on providing an effective and efficient method for minimizing shaft misalignment, due to vibrations during the motor rotation and external forces, between the rotor and stator in electric motors, more precisely in flameproof motors where the shaft is supported in sleeve bearings.

Integrating specific parameters and practical functionalities, the tool aims to facilitate the axial alignment process, ensuring optimized performance, motor durability and minimize the manual alignment time. In the context of sleeve bearings, it's essential to address the limitations they impose on axial loads. Unlike ball bearings, which can support axial loads due to their design, sleeve bearings primarily handle radial loads. This limitation makes precise axial alignment even more critical, as any misalignment can lead to increased wear and tear on the bearings, potentially leading to premature failure. Additionally, blocking the shaft axially, a common technique used in some motor alignments, isn't feasible in flameproof motors due to safety concerns. Sleeve bearings are preferred in flameproof motors for their intrinsic safety, robustness, and cost-effectiveness. Their simpler design reduces the risk of sparking in hazardous environments and makes them less susceptible to damage from contaminants. Additionally, sleeve bearings require less maintenance and are more economical to manufacture and replace compared to ball bearings.

The misalignment of the rotor in flameproof motors primarily occurs due to the unique support mechanism provided by sleeve bearings. Unlike ball bearings, which offer more rigid support, sleeve bearings suspend the shaft in oil, making the rotor more susceptible to axial displacement. This susceptibility is exacerbated by the presence of fans in the motor, which apply varying axial forces on the rotor. These forces leads to rotor misalignment, causing deviations in the magnetic field and affecting motor performance. Since the type of fans used can vary between motors, the magnitude of the axial force also differ, further contributing to misalignment issues.

Additionally, it is crucial to note that this tool is designed for use in manufacturing courses.

1.2.2 Specific Objectives

- **Study of Bearing Motor Projects:** Investigate and analyze existing projects related to sleeve bearings electric motors, specifically focusing on flameproof electric motors.
- **Evaluation of Project Alternatives:** Assess and compare various project alternatives concerning sleeve bearing motor designs, considering parameters such as efficiency and cost. Also it is important to ensure that any other components will be damaged during the tool operation and does not modify the tolerances and requirements of the motor when using the developed tool.

- **Development of Simulation Procedure:** Formulate a comprehensive simulation procedure, identifying and integrating relevant parameters and input variables. With that it is expected to implement modal, internal pressure and static simulations.
- **Comparative Analysis of Project Alternative:** Apply the developed simulation procedure to evaluate and compare the performance of project alternative with the original solution.
- **Proposal and Final Validation of Alignment Tool:** Develop and validate a tool for motor shaft alignment, culminating in a final proposal that integrates practical applications within the studied motor projects.

1.3 Motivation for the Choice of the Topic

The selection of the dissertation topic is a crucial decision that aligns with academic and professional interests. In the context of pursuing a Master's degree in Computational Mechanics with a focus on mechanical simulations, two distinct internship proposals were presented for consideration:

1.3.1 Study of the Influence of Complete Outer Machining of the Stator on the Thermal Performance of Flameproof Motors

The first proposal involves an in-depth examination of the thermal performance of flameproof motors concerning the complete outer machining of the stator. This study would entail exploring the impact of machining processes on heat dissipation, with potential implications for motor efficiency and safety.

1.3.2 Development of a Tool for Shaft Axial Aligning of Large Electric Motors

The alternative proposal centers around the development of a tool for aligning the rotor shaft axially and its magnetic field in flameproof electric motors. This project aims to enhance the alignment precision between the rotor and stator, potentially optimizing motor performance, longevity and reducing the manual time for aligning it.

1.3.3 Choice Rationale

The chosen topic for the dissertation is the "Development of a Tool for Shaft Axial Aligning of Large Electric Motors." This decision is motivated by several factors:

- **Mechanical Precision:** The alignment of large electric motors involves intricate mechanical processes. The chosen topic allows for a detailed exploration of mechanical precision requirements, including tolerances, which is crucial for the efficient operation of these machines.

- **Technical Drawing and Machine Design:** The development of an alignment tool necessitates a thorough understanding of technical drawing and machine design principles. This project offers an opportunity to apply and advance skills in these areas, contributing to a comprehensive knowledge of machine design.
- **Practical Application of Computational Mechanics:** The alignment tool's design and optimization can be effectively explored through computational simulations. This allows for a practical application of Computational Mechanics techniques in the realm of mechanical design and simulation.
- **Industry Relevance and Innovation:** Addressing the challenge of aligning large electric motors is not only academically stimulating but also holds significant practical implications for various industries. The research contributes to the innovation and improvement of mechanical systems within the realm of electric motors.

1.4 Research Objectives

By the points explained in [1.3.3](#), we aim to bring precision to the motor alignment process and, consequently, optimize overall motor performance.

1. **Understanding the Landscape:** This involves diving into the literature to understand the tools and techniques currently employed and their impact on motor performance. Also understanding how an electric motor works and all of its components.
2. **Mechanical Insight:** Conduct a meticulous mechanical analysis, considering factors such as tolerances, precision requirements, and the technical intricacies involved in the alignment process. This step also involves a close examination of technical drawings and principles of machine design.
3. **Simulating Possibilities:** Employ Computational Mechanics techniques to simulate the alignment process. This allows us to explore different design parameters and understand how they influence alignment precision under various conditions. This involves iteratively adjusting simulation parameters to enhance the tool's performance and precision, ensuring its effectiveness in a wide range of operational scenarios. Also investigate the impact of boundary conditions, loads, and various simulation parameters on the alignment tool's performance. Fine-tune the computational model to optimize precision by adjusting factors such as environmental influences, mechanical loads, and other critical parameters.
4. **From Concept to Reality:** Transform our theoretical insights into a tangible solution by developing a prototype of the alignment tool. This prototype will embody practical features aimed at simplifying the alignment of the magnetic center in large electric motors.
5. **Putting it to the Test:** Validate our creation through experimental testing and simulations. This step involves assessing the effectiveness and precision of our alignment tool, comparing

real-world results with our simulated expectations and also understand how to modify some requirements or inaccuracies during the creation process.

6. **Sharing Knowledge:** Document our entire research journey comprehensively, from the initial literature review to the development of the prototype and testing. The report will not only encapsulate the findings but also provide guidelines for applying the tool and suggestions for future improvements.

1.5 Dissertation Structure

The dissertation follows a coherent structure, Fig. 1.1, commencing with an introduction in Chapter 1 that outlines company details, contextualizes the study, and sets specific objectives. Chapter 2 conducts a comprehensive literature review on electric motors, bearing categories, existing alignment tools and misalignment and even into the study of flameproof motor projects, laying the groundwork for Chapter 3, which explores the theoretical foundations and applications of numerical simulations. This theoretical framework informs subsequent chapters, with Chapter 4 evaluating design alternatives, and Chapter 5 focusing on the development and validation of a simulation procedure and the alignment tool. Final considerations, limitations, and suggestions for future research are discussed in Chapter 6, leading to the dissertation's conclusion in Chapter 7, summarizing key objectives and conclusions.

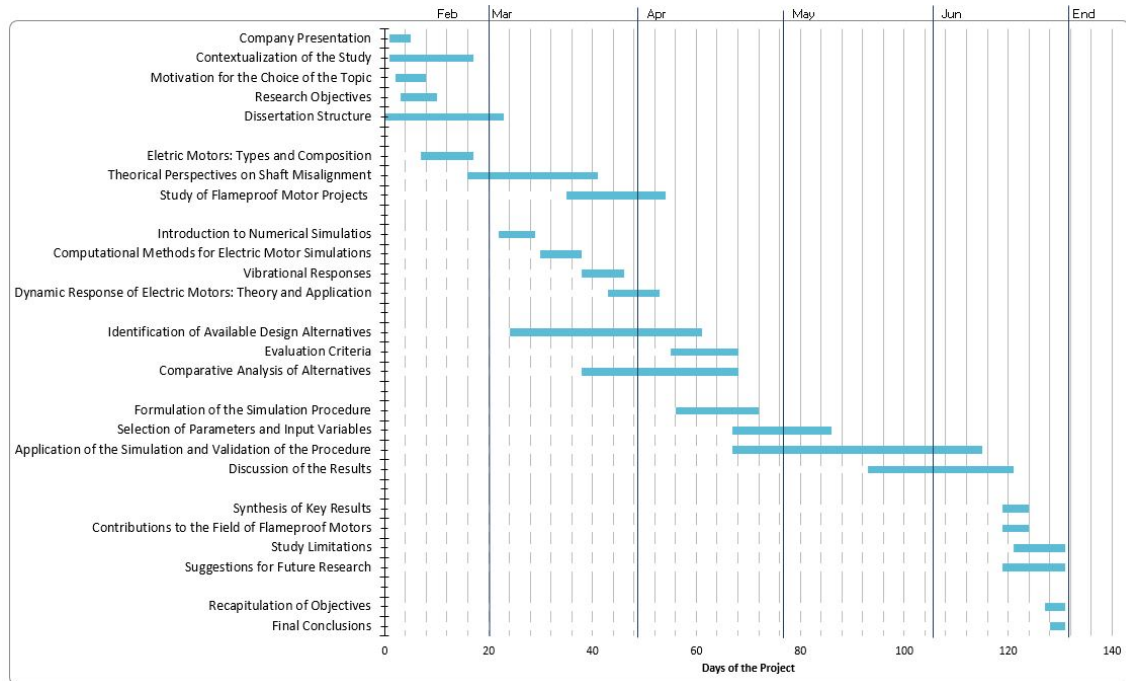


Figure 1.1: Gantt plan for the organization of the dissertation

Chapter 2

Literature Review in Induction Motors

This chapter provides a comprehensive review of some existing literature on induction motors, performance evaluations, and alignment tools characteristics. It serves as a crucial foundation, offering theoretical insights and references essential for advancing our primary research focus. We explore various types and compositions of electric motors, gaining a fundamental understanding of their characteristics.

2.1 Electric Motors: Types and Composition

An electric motor (Fig. 2.1) is a device that operates on the principle of electromagnetic induction to convert electrical energy into mechanical energy. When an electric current flows through a wire in the presence of a magnetic field, a force is generated, causing the wire to move. In an electric motor, this principle is utilized to produce rotational motion.

The basic components of an electric motor include a stator (stationary part) and a rotor (rotating part). The stator typically consists of coils of wire wound around a core, creating a magnetic field when an electric current passes through them. The rotor, usually made of a series of conductive bars or coils, is placed within the magnetic field of the stator.

When an electric current is supplied to the stator windings, a magnetic field is generated, which interacts with the magnetic field of the rotor. This interaction creates a force that causes the rotor to rotate, thus converting electrical energy into mechanical energy. The rotation of the rotor can be used to drive various mechanical devices or systems [2].

2.1.1 Torque

The torque or moment, denoted as C , measures the effort required to rotate a shaft. It is the product of force (F) and the distance from the center of the shaft (E):

$$C = F \cdot E \quad [\text{Nm}] \quad (2.1)$$

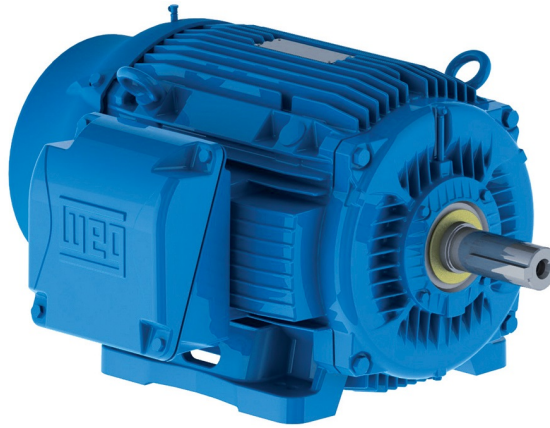


Figure 2.1: AC Electric Motor WEG

In practical experience, the force needed to lift a weight using a crank mechanism depends on the length of the crank. The torque is crucial, defined as the product of force and distance.

2.1.1.1 Work (W)

Work (W) is the measure of the energy transferred to or from an object when a force (F) is applied to move it through a distance (d). The work done is calculated by multiplying the force applied by the displacement:

$$W = F \cdot d \quad [\text{Nm}] \quad (2.2)$$

2.1.1.2 Power (P)

Power is the rate at which energy is applied or consumed. It is calculated by dividing the work done by the time taken:

$$P = \frac{W}{t} \quad [\text{Watts}] \quad (2.3)$$

For circular motion, torque is given by $C = F \cdot r$. Power is then calculated using angular velocity (n) in revolutions per minute (rpm):

$$P_{\text{mec}} = \frac{F \cdot d \cdot \pi \cdot n}{60} \quad [\text{cv}] \quad (2.4)$$

2.1.2 Electrical Energy and Power

For resistive loads, electrical power is given by:

$$P = U \cdot I \quad [\text{Watts}] \quad (2.5)$$

For resistive loads in AC circuits, power is given by:

$$P = 3 \cdot U \cdot I \cdot \cos \phi \quad [\text{Watts}] \quad (2.6)$$

The power factor is the ratio of active power to apparent power:

$$\cos \phi = \frac{P}{3 \cdot U \cdot I} \quad (2.7)$$

2.1.3 Basic Operating Principle

All electric motors operate based on the same fundamental principle. Two magnetic fields are formed inside the motors – one on the stationary stator and the other on the rotating rotor. The interaction between these magnetic fields generates a torque on the rotor, causing it to turn [3].

2.1.3.1 Direct Current (DC) Motor

The rotation of a DC motor is achieved by the force produced on current-carrying conductors in the magnetic field created by the stator. There are two stationary magnetic fields in a DC motor – the stator magnetic field produced by magnets or a field winding, and the rotor magnetic field produced by the current in the rotor conductors. The rotor magnetic field remains stationary due to brushes and commutators, ensuring continuous rotation.

2.1.3.2 Alternating Current (AC) Motor

AC motors differ from DC motors in that they exploit the force between two rotating magnetic fields. Both the stator and rotor magnetic fields rotate, generating a constant force between them, leading to continuous rotation. The rotating magnetic field on the stator is typically created by three-phase currents.

2.1.4 Types of AC Motors

There are two main types of AC motors:

1. **Synchronous Motor:** The rotor magnetic field is generated by a permanent magnet or a field winding powered by a separate DC power supply. The rotor must rotate at the synchronous speed to produce torque.
2. **Induction Motor:** The rotor magnetic field is generated by AC power transferred from the stator through electromagnetic induction. The rotor speed is always less than the synchronous speed, making it asynchronous.

2.1.5 Comparison Between AC and DC Motors

- AC motors use alternating current, while DC motors operate on direct current, Fig. 2.2.
- DC motors are often used for speed and torque control due to their simplicity, where speed is proportional to voltage, and torque is proportional to current.

- AC motors, especially induction motors, are known for their continuous and robust operation [3].

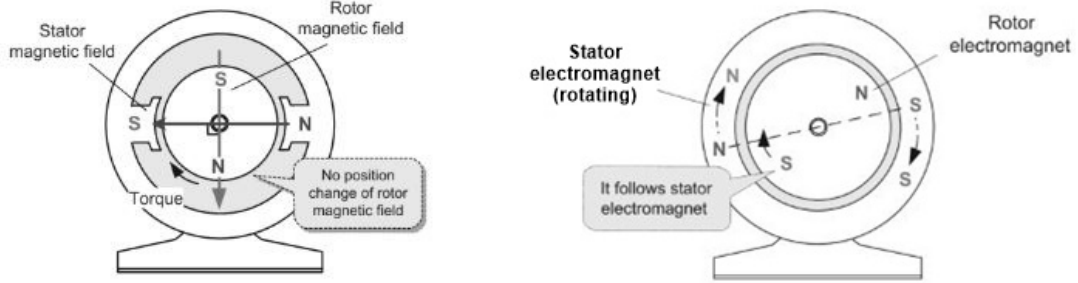


Figure 2.2: Operating system of DC motor, left. Operating system of AC motor, right. [3]

2.1.6 Motor Efficiency and Power Equations

The efficiency (η) of an electric motor is defined as the ratio of mechanical power output (P_u) to electrical power input (P_a). The relationship is given by:

$$\eta = \frac{P_u}{P_a} = \frac{736 \cdot P_{cv}}{3 \cdot U \cdot I \cdot \cos \phi} \quad (2.8)$$

Where P_{cv} is the mechanical power in horsepower, U is the voltage, I is the current, and $\cos \phi$ is the power factor [2].

2.1.6.1 Relationship between Torque and Power

For rotary motion, the power developed depends on torque (C) and rotational speed (n). The relations are:

$$P_{cv} = \frac{716 \cdot C_{kgfm} \cdot n_{rpm}}{7024} \quad (2.9)$$

$$P_{kW} = \frac{974 \cdot C_{Nm} \cdot n_{rpm}}{9555} \quad (2.10)$$

$$C_{kgfm} = \frac{716 \cdot P_{cv}}{n_{rpm}} \quad (2.11)$$

$$C_{Nm} = \frac{974 \cdot P_{kW}}{n_{rpm}} \quad (2.12)$$

2.1.7 Monophasic and Trifasic Currents in Electric Motors

2.1.7.1 Monophasic Current

Characteristics: Monophasic current involves a single alternating current waveform, denoted as $I(t)$ 2.13.

Use Cases: Commonly used in household applications and small-scale operations. Appliances like fans, refrigerators, and small power tools often operate on monophasic power supplies.

2.1.7.2 Trifasic Current

Characteristics: Trifasic current involves three alternating current waveforms, denoted as $I_1(t)$, $I_2(t)$, and $I_3(t)$, with a 120-degree phase difference between them.

Use Cases: Preferred for industrial applications and heavy-duty operations. Electric motors in manufacturing equipment, large pumps, and conveyor systems are often designed for trifasic power supplies.

Trifasic currents provide a more balanced and continuous power delivery in contrast to the pulsating nature of monophasic currents. Additionally, trifasic motors typically demonstrate higher efficiency and power output. The balanced characteristics of trifasic power contribute to a reduction in torque fluctuations, ensuring a more even distribution of power throughout the motor's rotation.

In comparison, monophasic systems are characterized by their simplicity and cost-effectiveness, making them suitable for smaller-scale applications. On the other hand, trifasic systems are more aptly designed for larger and more complex industrial operations where higher power and enhanced reliability are imperative [4].

Monophasic Current:

$$I(t) = I_{\max} \sin(\omega t + \phi) \quad (2.13)$$

Trifasic Current:

$$\begin{aligned} I_1(t) &= I_{\max} \sin(\omega t + \phi) \\ I_2(t) &= I_{\max} \sin(\omega t + \phi - 120^\circ) \\ I_3(t) &= I_{\max} \sin(\omega t + \phi + 120^\circ) \end{aligned} \quad (2.14)$$

2.1.8 Series and Parallel Connections

For two loads in a single-phase system, connections can be in series or parallel, affecting voltage and current distribution.

Three-phase systems involve three voltages with a 120° phase difference. Connections can be delta or star, affecting voltage and current distribution, Fig. 2.3.

For three loads connected in delta, the line voltage (U) is the system's nominal voltage.

For three loads connected in star, the phase voltage (U_f) is the system's nominal voltage [4].

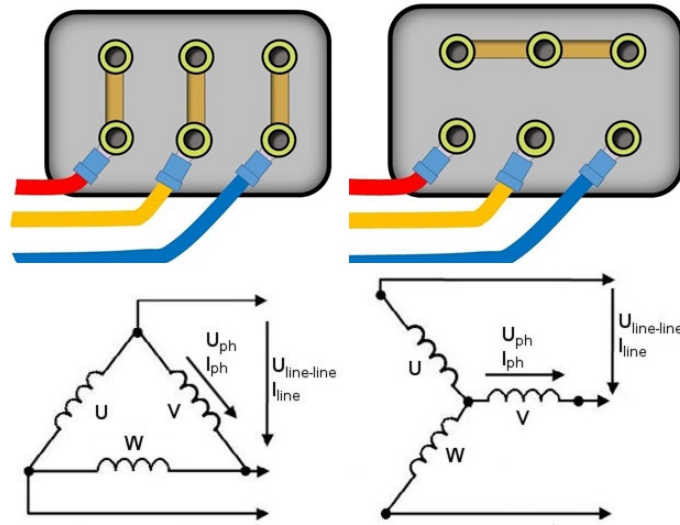


Figure 2.3: Delta connection, left. Star connection, right.

2.1.9 Induction Motor

The induction motor comprises a stator and rotor. Stator parts include the casing, laminated core, and three-phase winding. Rotor parts include the shaft, laminated core, and short-circuit bars [5].

2.1.9.1 Principle of Operation - Rotating Magnetic Field

A rotating magnetic field is created by the three-phase stator winding, inducing a current in the rotor bars. The interaction produces a rotating magnetic field in the rotor, generating torque for motor rotation.

2.1.9.2 Synchronous Speed (n_s) and Slip (s)

The synchronous speed is determined by the number of poles and frequency:

$$n_s = \frac{120 \cdot f}{p} \quad (2.15)$$

Slip is defined as the difference between synchronous and rotor speeds:

$$s(\%) = \frac{n_s - n}{n_s} \times 100 \quad (2.16)$$

2.1.9.3 Nominal Speed

The nominal speed of the motor is the operating speed at full load, influenced by slip and synchronous speed.

2.1.10 Operating Characteristics

Electric motors exhibit specific operational characteristics that significantly influence their performance and longevity. Understanding these features is crucial for proper usage and maintenance.

One of the key aspects influencing motor performance is the heating of the winding during operation. Monitoring and maintaining temperatures within specified limits ensure optimal performance.

The lifespan of a motor is influenced by diverse factors, including operational conditions and adherence to maintenance guidelines. Rigorous monitoring and following specified practices contribute to extending the motor's useful life.

Motors operate under various regimes, each with distinct characteristics. These include continuous, short-time, and intermittent operation. Some regimes involve features like electric braking or load/speed changes. Understanding these operational modes is vital for selecting the appropriate motor for a given application.

To prevent overheating, motors require effective thermal protection mechanisms. These safeguards ensure that the motor operates within safe temperature limits, enhancing reliability and preventing damage due to excessive heat.

Nominal power represents the maximum continuous power a motor can deliver under specified conditions. Proper consideration of nominal power is essential to prevent overheating and optimize the motor's efficiency.

For loads with small inertia, the equivalent power (P_m) can be calculated using the formula:

$$P_m = \frac{\sum P(t)}{T} \quad (2.17)$$

where P_m is the equivalent power requested from the motor, $P(t)$ is the variable power with time, and T is the total cycle duration.

The duty cycle factor (FS) indicates the permissible continuous load under specified conditions. A FS value of 1.0 means the motor is not designed for continuous operation above its nominal power [2].

2.1.11 Environmental Characteristics

To assess the feasibility of using an electric motor in a specific application, several parameters must be taken into consideration, including:

- Altitude at which the motor will be installed.
- Temperature of the cooling medium.

According to IEC 60034-1 standards, the usual operating conditions are defined as follows:

1. Altitude not exceeding 1,000 meters above sea level.

2. Cooling medium (usually ambient air) with a temperature not exceeding 40°C and free of harmful elements. .

Within various altitude and ambient temperature values, considered normal conditions, the motor should deliver its rated power without overheating.

Considering the combined effects of temperature and altitude variations, the motor's power dissipation capacity can be obtained by multiplying the rated power by the multiplication factor.

Table 2.1: Example of Multiplication Factor for Power Dissipation

T (°C)	Altitude (m)	Multiplication Factor (α)
55	2,000	0.83

For instance, a 100 hp motor with insulation class F, operating at an altitude of 2,000 meters with an ambient temperature of 55°C, would have a multiplication factor (α) of 0.83, resulting in the motor providing only 83 % of its nominal power [2]. Noticing that this characteristics above are related to WEG's electric motors.

2.1.12 Explosive Atmospheres

An atmosphere becomes explosive when the proportion of gas, vapor, dust, or fibers is such that a spark from an electrical circuit or the heating of a device causes an explosion. To initiate an explosion, three elements are necessary: Fuel + Oxygen + Ignition Source = Explosion [2].

2.1.12.1 Classification of Risk Areas

According to IEC 60079-10-1 standards, risk areas are classified as follows:

- Zone 0: Continuous presence of flammable/explosive mixture, e.g., inside a fuel tank.
- Zone 1: Probability of flammable/explosive mixture occurrence associated with normal equipment and process operation.
- Zone 2: Occasional and improbable presence of flammable/explosive mixture, associated with abnormal equipment operation, losses, or negligence.

2.1.12.2 Temperature Classes

The maximum temperature on the internal and/or external surface of electrical equipment must always be lower than the ignition temperature of the gas or vapor. Gases are classified into temperature classes T1 to T6 based on their ignition temperature.

2.1.12.3 Enclosure Types

Different enclosure types provide protection in explosive atmospheres, such as explosion-proof, increased safety, intrinsic safety, encapsulation, non-incendive, oil immersion, pressurization, sand filling, and dust ignition protection.

2.1.13 Sizing standards

WEG electric motors follow standardized dimensions according to the International Electrotechnical Commission - IEC-60072 standards. The basic dimension for the standardization of mounting dimensions for rotating electrical machines is the height from the base plane to the center of the shaft tip, denoted by H , as shown in Fig. 2.4.

Each H height is associated with a C dimension, the distance from the center of the foot hole on the shaft tip side to the plane of the shaft tip support. Multiple B dimensions (distance between foot hole centers) can be associated with each H dimension, allowing for the creation of both "longer" and "shorter" motors.

Dimension A , the distance between foot hole centers in the frontal direction, is unique for H values up to 315 mm but can take multiple values starting from H equal to 355 mm [2].

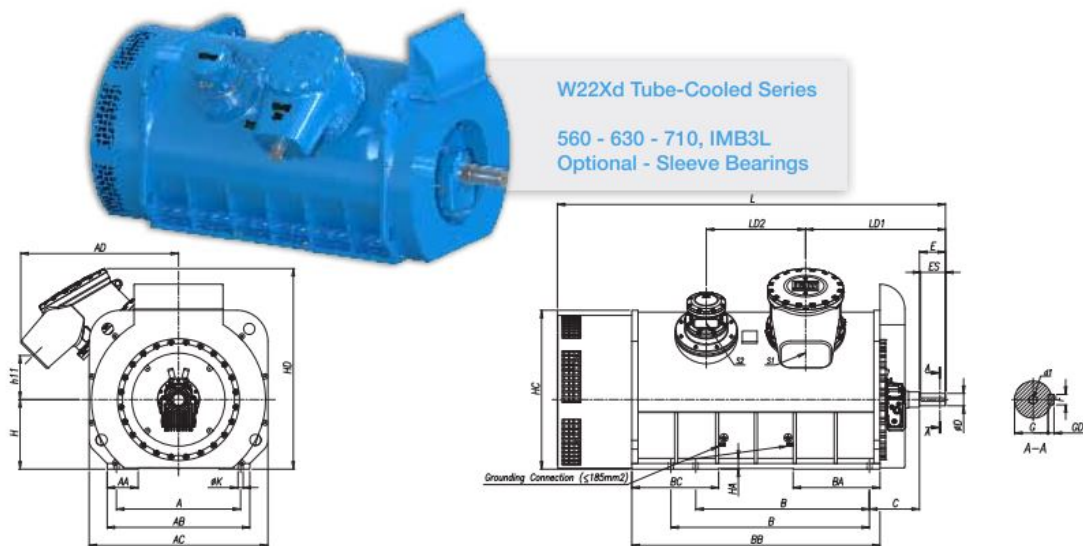


Figure 2.4: Dimensional Representation of an Electrical Motor WEG W22Xd Tube-Cooled Series [6]

Constructive forms refer to the arrangement of machine construction parts concerning fixation, bearing disposition, and shaft tip. These are standardized by DIN 42950, and NEMA MG 1-4.03. The position of the motor's connection box, as specified by IEC 60072, is determined with respect to the motor's top, 10 degrees below the horizontal center line when viewed from the drive side.

2.1.13.1 Composition

The composition of an electric motor varies in each type of configuration, but it is possible to give an overall understanding by illustration an exploded view of an example:

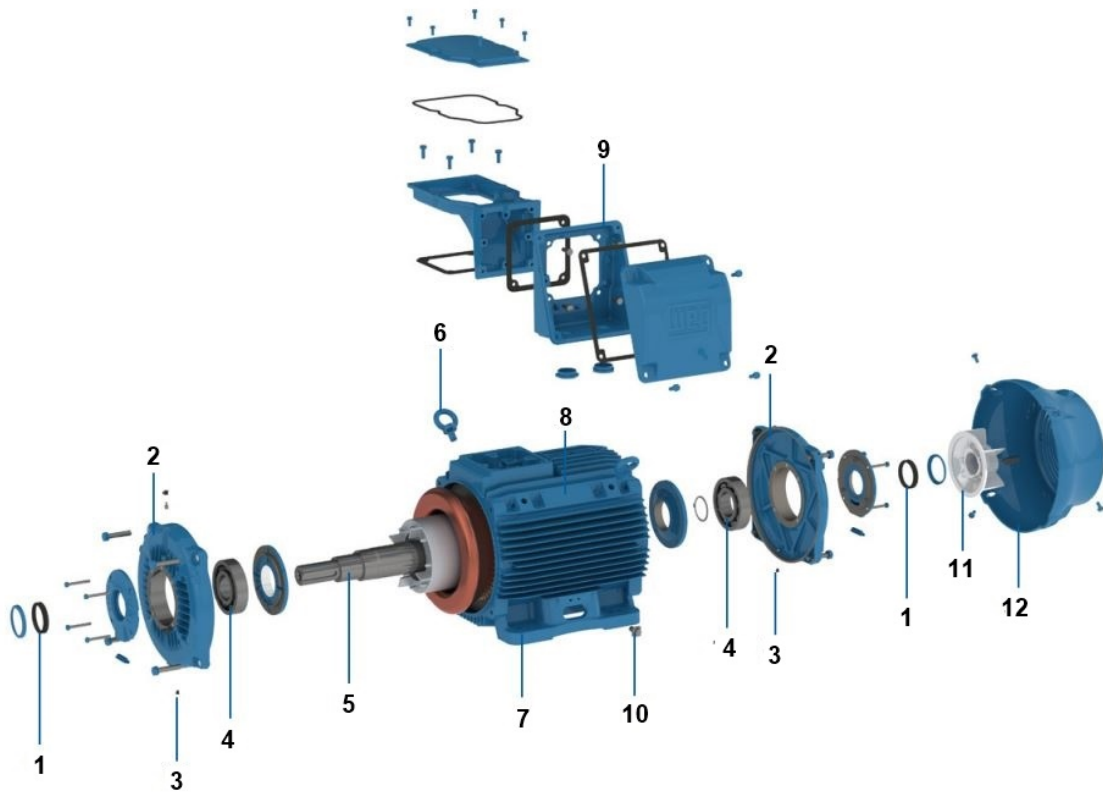


Figure 2.5: Exploded view of a W22 electric motor [7]

Table 2.2: Components identification in an electric motor constitution, Fig. 2.5

1	Enclosure	2	Endshield	3	Drainage system
4	Bearings	5	Shaft	6	Eye bolts
7	Housing	8	Induction plate	9	Junction box
10	Grounding terminals	11	Ventilation system	12	Deflector cover

2.2 Theoretical Perspectives on Shaft Misalignment

Shaft misalignment remains a pervasive challenge in electric motors, leading to issues like increased vibration, premature bearing wear, and decreased overall efficiency. This section provides a theoretical exploration of shaft misalignment, examining its causes, effects, and implications for motor operation. Fundamental principles of alignment theory, including the impact on torque transmission and bearing loads, are discussed. By establishing a strong theoretical foundation on the consequences of shaft misalignment, this section contributes to the rationale behind the need

for precise alignment and lays the groundwork for the subsequent development of an effective alignment tool.

The theory of shaft misalignment is based on the idea that the deviation of the relative positions of the rotation axes of two or more machines can cause operational problems. Shaft misalignment occurs when the central rotation lines of the shafts are not aligned with each other or when the shaft is not aligned axially with the stator. This can occur due to various factors, such as manufacturing errors, wear, vibration, deformation, or thermal movement of the machines. Shaft misalignment can lead to a range of problems, such as excessive vibration, premature bearing wear, increased energy consumption, excessive noise, and premature component failures, and in the case of electric motors, the misalignment of the magnetic field. Therefore, it is important to perform proper shaft alignment to ensure the smooth operation and extended lifespan of the machines. There are different methods and techniques for performing shaft alignment, such as the use of dial indicators, lasers laser alignment systems or even the production of a tool that provide the alignment of the rotor. The goal is to adjust the position of the shafts, minimizing misalignment and its negative effects. It is important to note that shaft misalignment can occur in both flexible and rigid couplings. Additionally, misalignment can be classified into different types, such as parallel, angular, or combined misalignment, depending on the direction and magnitude of the deviation of the shafts [8].

2.2.1 Shaft Misalignment in Electric Motors

Shaft misalignment in electric motors can be caused by various factors, and these causes can vary across different motor types and operating conditions. Here are some of the primary causes of shaft misalignment [8]:

- **Thermal Expansion:** Temperature changes in the motor and its surrounding environment can cause the shafts to expand or contract, leading to misalignment. This can occur due to friction in the bearings or thermal changes in the process liquids and gases.
- **Manufacturing Errors:** Inaccurate machining or assembly during the manufacturing process can result in misalignment. This can include errors in the alignment of the motor shaft and the driven equipment, as well as misalignment in the coupling or other connecting components.
- **External Forces:** External forces acting on the motor, such as vibrations, shocks, improper handling, or even the airflow fan force, can cause misalignment. These forces can disrupt the alignment of the motor shaft and the driven equipment, leading to misalignment.
- **Improper Installation:** Incorrect installation procedures, such as improper alignment techniques or inadequate torquing of fasteners, can result in shaft misalignment.

The variation of these causes across different motor types and operating conditions can depend on factors such as the motor design, size, load characteristics, and environmental conditions. For example, high-speed motors may be more sensitive to thermal expansion, while motors operating in harsh environments may be more prone to wear and tear. It is essential to consider these factors when addressing shaft misalignment in electric motors to ensure optimal performance and longevity of the equipment.

Another reason for the axial misalignment is the induction motors may not be perfectly balanced, for various reasons, resulting in some voltage discrepancies that impact the magnetic field, ultimately causing the rotor to move closer to the mechanical center. Upon motor startup, the rotor undergoes axial movement, wherein the magnetic field connecting the rotor and stator across the air gap becomes stretched if the rotor is axially misaligned with the stator. A restoring force then acts to realign it, resisting movement in the opposite direction as well. Consequently, the rotor oscillates forward and backward axially until it reaches a steady or balanced state. This equilibrium point, where these forces counterbalance each other, is termed the "magnetic center" due to the magnetic nature of these forces, although other forces may also be present. In ball bearing motors, the magnetic center is already established and fixed, with minimal axial float equivalent to the clearance of the thrust bearing. However, in larger sleeve bearing motors, the end play or axial movement can reach significant values, such as 16 mm or more [9].

2.2.2 Challenges associated with detecting and correcting shaft misalignment in electric motors

Detecting and correcting shaft misalignment in electric motors, especially in industrial settings with large and complex machinery, can pose several challenges. These challenges arise due to the nature of the equipment, the complexity of the alignment process, and the potential impact of misalignment on motor performance. Here are some of the key challenges associated with detecting and correcting shaft misalignment [8]:

- **Accessibility:** In industrial settings with large and complex machinery, accessing the motor and its components for alignment measurements can be challenging. The motors may be located in confined spaces or in areas with limited visibility, making it difficult to accurately measure the alignment parameters. This can require specialized tools and techniques to overcome accessibility issues
- **Precision Measurement:** Achieving accurate and precise measurements of shaft positions is crucial for detecting misalignment. However, in industrial settings, achieving high precision measurements can be challenging due to factors such as vibrations, environmental conditions, and the size of the machinery. Specialized alignment tools and techniques, such as laser alignment systems, may be required to achieve the necessary level of precision

- **Correction Complexity:** Correcting shaft misalignment in industrial settings can be complex due to the size and complexity of the machinery. The correction process may involve adjusting the position of the motor or the driven equipment, which can require specialized tools and expertise. In some cases, modifications to the mounting or coupling arrangements may be necessary to achieve proper alignment. The complexity of the correction process can increase the time and effort required for alignment
- **Impact on Production:** In industrial settings, any downtime or disruption to production can have significant financial implications. Detecting and correcting shaft misalignment often requires temporarily shutting down the equipment, which can impact production schedules. Minimizing the downtime and optimizing the alignment process is crucial to mitigate the impact on production

2.2.3 Alignment Techniques in Electric Motors

Proper alignment helps minimize vibration, reduce energy consumption, and prevent premature wear and tear on the motor and its components. There are several alignment techniques commonly used in electric motors, in a general way [8]:

- The first technique, the Straight Edge Method, is a basic alignment technique involving the use of a straight edge or a taut string to visually align the motor shaft with the driven equipment shaft. The goal is to ensure that the two shafts are parallel and in the same horizontal or vertical plane, Fig. 2.6.
- Another method, the Dial Indicator Method, utilizes a dial indicator to measure the misalignment between the motor and driven equipment shafts. The dial indicator, mounted on the motor shaft, measures the radial and axial displacement, and adjustments are made until the misalignment is within acceptable limits.
- Laser Alignment Method is a more advanced technique that uses laser technology to accurately measure and align the motor and driven equipment shafts. Laser alignment systems provide real-time measurements and graphical representations of the misalignment, facilitating precise adjustments.
- The Reverse Indicator Method involves using a reverse indicator to measure the angular misalignment between the motor and driven equipment shafts. Adjustments are made until the angular misalignment is minimized by measuring the difference in rotational position between the two shafts.
- For aligning motors with belt-driven equipment, the Rim and Face Method are commonly used. This technique measures the misalignment between the motor pulley and the driven

equipment pulley using a straight edge or a laser alignment system. Adjustments are made to ensure alignment both radially and axially.

- **Thermal Growth Compensation** techniques address the thermal expansion or contraction experienced by electric motors during operation. By measuring the temperature of the motor and driven equipment, adjustments are made to compensate for thermal expansion or contraction.

2.2.4 Axial Alignment

Aligning the magnetic field of an electric motor involves aligning the axial displacement of the shaft, which refers to the deviation of the motor shaft's relative position from its desired centerline. Because of the non axial forces support by the sleeve bearings, proper alignment ensures the rotor is centered within the stator field, enhancing motor performance. To achieve this alignment, can be followed these steps [8]:

1. **Uncouple the Motor:** Begin by disconnecting the motor from any driven equipment or components to allow independent operation and easy observation of axial displacement.
2. **Run the Motor Solo:** Operate the motor without any load or external forces after uncoupling. Apply power and observe the shaft's behavior.
3. **Observe Axial Movement:** When power is applied, the motor shaft may exhibit axial movement, indicating electromagnetic forces attempting to center the rotor. Allow the shaft to settle and observe its final position.
4. **Determine Axial Displacement:** The settled position represents the axial displacement, showing the deviation from the desired centerline. Note that this displacement may vary based on motor design.
5. **Adjust Axial Spacing:** Adjust the axial spacing between the motor and driven equipment based on the determined displacement. Proper spacing ensures the motor operates within its parameters and minimizes misalignment issues.

By aligning the axial displacement, the magnetic field of the electric motor is effectively aligned, enabling electromagnetic forces to center the rotor within the stator field. This optimization enhances motor performance and reduces wear or damage potential.

The objective of this project is to create a tool capable of accurately measuring axial displacement due to vibrations and after that measurement, machining the principal Endshield of the motor (the frame covers) to precise thicknesses, and with that, to assure the correct rotor axial location. This tool will facilitate the alignment of shaft axial displacement by ensuring that the Endshields fit perfectly, thereby optimizing the alignment process.

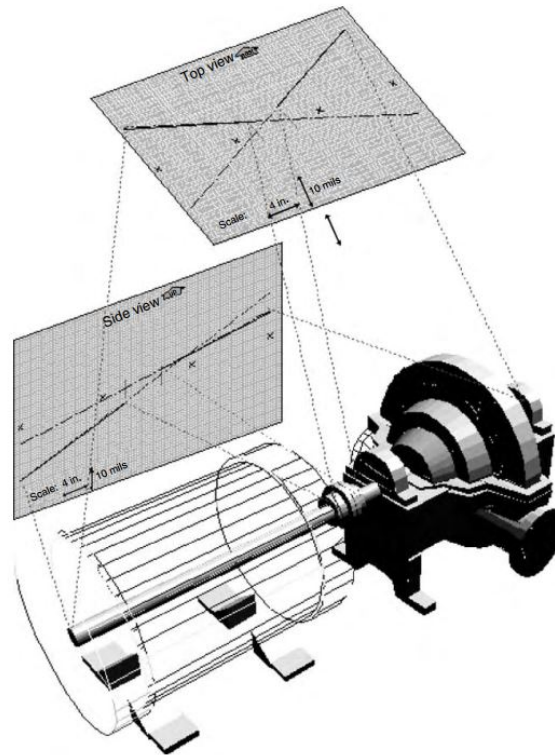


Figure 2.6: Shaft misalignment on an induction motor [8]

2.2.5 Energy consumption in rotating machinery caused by misalignment

The correlation between vibration level and energy consumption in rotating machinery is a critical aspect highlighted in the research article. This correlation serves as a key indicator of the efficiency and health of machinery systems, providing valuable insights into the relationship between mechanical faults, such as misalignment, and energy inefficiencies. By monitoring and analyzing vibration levels alongside electrical power consumption, the study aimed to uncover how misalignment influences energy losses in rotating machinery [10].

Vibration analysis plays a pivotal role in this correlation, as changes in vibration levels can indicate potential issues within the machinery, such as misalignment, unbalance, or bearing defects. These mechanical faults can lead to increased energy consumption as the machinery operates under suboptimal conditions. By establishing a direct correlation between vibration levels and energy consumption, industries can proactively identify and address these issues to optimize energy usage, reduce operational costs, and enhance machinery performance.

By leveraging vibration analysis techniques, organizations can gain valuable insights into the health of rotating machinery, identify potential energy inefficiencies, and optimize maintenance practices to ensure the long-term sustainability and efficiency of machinery systems.

To create an environment to analyse the misalignment of the shaft, several strategies can be used to study the energy consumption of the motor. One of them are the use of Shims. Shims

are used to create misalignment in the rotating machinery setup. Specifically, shims of 0.25 mm thickness are inserted under various conditions at the corners of the motor to induce misalignment in two ways: parallel misalignment and angular misalignment. By strategically placing shims under different conditions, the experimental setup simulates misalignment scenarios in the rotating machinery, allowing for the measurement of vibrations in terms of acceleration in axial and radial directions [10].

Also to control the displacement it can be used a strategy by adding shims of varying thickness behind the stator, the angle of the stator relative to the rotor can be adjusted. Similarly, shims are inserted between the rotor backing plate and the rotor shaft to induce dynamic angular or axial misalignment. The shims are cut to specific dimensions and inserted behind the rotor plate to create a controlled misalignment in the rotor component. Shims are used to align the motor by inducing misalignment in the rotor and stator components [11].

2.2.6 Vibration diagnostic challenges

Some scenarios are related to the behavior of electric motors and can be observed during vibration analysis.

One possibility is the presence of rotor balance and electromagnetic components with a frequency difference equal to the slip frequency. The slip frequency is the difference between the synchronous speed of the motor and the running speed. When these two components have similar amplitudes and a small frequency difference, they can create a phenomenon called a beat. This beat can be perceived as an audible humming. Another scenario is the presence of a 2FL (twice line frequency) vibration component and a twice-running-speed component caused by misalignment. In this case, the two components have similar amplitudes and a small frequency difference. The 2FL component is related to the line frequency, while the twice-running-speed component is caused by the misalignment of the motor. If between the two components the amplitudes are significantly different, the resulting plot will show a clear distinction between the two components. However, if the amplitudes are similar, the resulting plot will show a combination of the two components [12].

Usually, a table that provides valuable information about the different vibration components and their characteristics in electric motors is used. The table includes various parameters and observations related to vibration, such as changes with ambient temperature, large difference between uncoupled and loaded conditions, causes of bearing damage, oil film instability (oil whirl), and rotation frequencies. It mentions that some frequencies can cause bearing damage and instability in the oil film. Additionally, the table provides insights into the behavior of vibration components under different conditions, such as load, seal or bearing rub, and bearing misalignment. It describes the frequencies at which these components can be observed and their characteristics, such as erratic behavior and sudden disappearance at lower speeds.

To solve some of vibrations challenges, it's usual to apply some methods as:

1. In the case of support base incompatibility, a detailed dynamic analysis of the entire drive system can be used. This analysis aids in identifying the root cause of the vibration problem. It is crucial to determine whether the vibration originates from the motor or elsewhere in the drive system.
2. The importance of obtaining vibration levels and phase angles at discrete frequencies, including those equal to one, two, and even up to ten times the rotational speed is another technique. Analyzing the vibration spectrum provides valuable insights into the nature of the excitation and helps identify potential issues.

2.2.7 Shaft misalignment due to Vibrations

Vibrations can be a key indicator of shaft misalignment in rotating machinery. When shafts are misaligned, it can lead to uneven distribution of forces and loads within the system, causing vibrations to occur. These vibrations can manifest as increased noise, higher levels of mechanical stress on components, and reduced efficiency of the machinery [13].

Shaft misalignment can result in dynamic forces that generate vibrations at specific frequencies, which can be detected through vibration analysis techniques. By monitoring and analyzing the vibration signatures of a system, engineers can identify the presence of shaft misalignment and take corrective actions to realign the shafts.

The misalignment-induced vibrations typically exhibit specific frequency components that are distinct from vibrations caused by other factors such as unbalance or mechanical defects. These frequency components are related to the rotational speed of the shaft and can be identified through vibration analysis techniques.

Several corrective actions can be taken to realign shafts effectively:

1. **Soft Foot Correction:** Check for any soft foot conditions where the machine's feet do not sit evenly on the base. Correcting soft foot conditions can help in achieving proper alignment.
2. **Alignment Tools:** Utilize precision alignment tools such as laser alignment systems or dial indicators to accurately measure and adjust the alignment of the shafts.
3. **Adjusting Couplings:** If misalignment is detected at coupling points, adjust the coupling elements to ensure proper alignment between the connected shafts.

2.3 Study of Flameproof Motors Projects

This literature review extends to the concepts and applications of motors utilizing sleeve bearings and even more precisely, flameproof electric motors, contributing to a deeper comprehension of their operational principles.

2.3.1 Working Principles

The fundamental characteristic of flameproof motors is their ability to prevent the ignition of flammable gases or vapors in hazardous environments. These motors are designed with special enclosures that can withstand and contain any internal explosion, preventing it from igniting the surrounding atmosphere. The enclosure is constructed in such a way that it can withstand the pressure generated by an internal explosion and prevent the flame from escaping. This ensures the safety of the motor and the surrounding area. Flameproof motors are commonly used in industries where there is a risk of explosive atmospheres, such as oil and gas, chemical, and mining industries [6].

The key working principles of flameproof motors include:

- **Explosion-proof Enclosure:** Flameproof motors are equipped with a specially designed enclosure that can withstand and contain any internal explosion. This containment prevents the pressure from causing damage to the surrounding equipment or environment. This is achieved through the use of constructive joints and gaps that guarantee flame propagation between the motor's interior and the external atmosphere. These joints and gaps are designed to limit the size and number of openings in the enclosure, ensuring that the flame does not escape.
- **Total Enclosure Sealing:** The enclosure of flameproof motors is designed to provide a high level of sealing to prevent the entry of flammable gases or vapors into the motor. This ensures that any potential source of ignition within the motor is isolated from the surrounding explosive atmosphere.
- **Cooling Efficiency:** To improve cooling efficiency, flameproof motors may incorporate symmetrical air internal chambers with a ripple design. Additionally, tube-cooled motors may use stainless steel tubes mounted in the frame to dissipate heat effectively.
- **Environmental Considerations:** Flameproof motors are designed and manufactured with a focus on energy efficiency and reducing carbon footprint. The waste produced during the manufacturing process is recycled, contributing to the preservation of natural resources.

2.3.2 Sleeve Bearings Motors: Concepts and Applications

Sleeve bearings are used on flameproof motors to provide support and reduce friction between rotating parts. In the context of flameproof motors, sleeve bearings are specifically designed to meet the requirements of hazardous environments where the risk of explosive atmospheres exists. These type of bearings are not the only ones used on flameproof motors, but they are the ones

that are going to be studied on this project. The main purpose of sleeve bearings in flameproof motors is to ensure smooth and reliable operation while minimizing the risk of ignition. These bearings require less maintenance than roller bearings and have longer lubrication intervals, which contributes to their longer lifetime [6].

Sleeve bearing motors are motors that utilize sleeve bearings as their primary bearing type to support the motor rotor. Sleeve bearing motors are commonly used in high-power industrial applications, direct-drive systems, vertical motors, and applications with minimal external loads. They are suitable for scenarios where low-speed operation, minimal maintenance requirements, and specific load-carrying capabilities are essential [14].

The sleeve bearing is self-aligning during the assembly process to ensure proper wear patterns and contact between the shaft and bearing. These motors typically have oil seals, also known as shaft seals, to prevent oil leakage and maintain proper lubrication within the bearing.

The lubrication intervals for sleeve bearings are determined based on the recommended guidelines and standards. Flameproof motors typically use mineral oils as the standard lubricant for sleeve bearings. The oil viscosity value is defined according to ISO 3448 or SSU Standards [6].

Depending on the thrusts and rotation speed, sleeve bearings can be equipped with self-lubrication (natural cooling) or an oil circulation system (external oil cooling). In some cases, there may be a subsequent conversion from self-lubrication to forced lubrication. The available sleeve bearings for rib-cooled and tube-cooled motors are specified in Tables 2.3 and 2.4. These tables provide information on the different sleeve bearing options for specific motor configurations.

Sleeve bearings are a type of plain bearing that consists of a cylindrical inner surface (bearing) and a shaft that rotates within it without any rolling elements, Fig. 2.7. These bearings are commonly used in induction motors rated above 500 kW, where the shaft peripheral speed exceeds the design limits of rolling element bearings. Sleeve bearings are known for their reliability due to the absence of mechanical contact between the shaft and bearing, making them more stable compared to rolling element bearings [15].

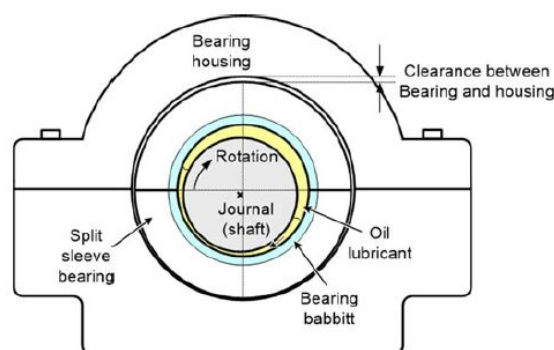


Figure 2.7: Structure of oil lubricated split sleeve bearing [15]

Some common types of sleeve bearings include [16]:

1. **Hydrodynamic Bearings:** These bearings rely on the relative movement of sliding surfaces and the lubrication gap between the journal and the bearing sleeves to create a lubrication wedge. This type of bearing is commonly used in various applications due to its efficiency in reducing friction and supporting loads.
2. **Fluid Bearings:** Fluid bearings use a fluid, such as air or liquid, to support the rotating shaft. They can be further categorized into aerostatic, aerodynamic, hydrostatic, and other types based on the type of fluid used for lubrication.
3. **Magnetic Bearings:** Magnetic bearings use magnetic fields to levitate and support the rotating shaft. They offer advantages such as low friction, high speed capabilities, and maintenance-free operation.
4. **Hybrid Bearings:** Hybrid bearings combine different types of bearings to leverage the benefits of each type. For example, a hybrid bearing could be a combination of fluid and magnetic bearings to achieve specific performance requirements.

Poles	Frame Size	Lubrication Method	Sleeve Bearing	Lubricant	Specification	Lubrication interval (hours)	Oil Quantity (liters)	Viscosity Range	
								ISO 40°C	SSU 100°F
2P	315L	Self-Lubrication	9-80	MOBIL DTE 24	Mineral Oil ISO VG32	8000	2.8	28.8-35.2 [cSt]	137-164 [s]
	355M/L								
	355A/B								
	400L/J								
	400G								
	450K/H								
	500K/H								
	560H/G								
4P+	315L		11-110	MOBIL DTE 25	Mineral Oil ISO VG46		4.2	41.4-50.6 [cSt]	193-235 [s]
	355M/L								
	355A/B								
	400L/J								
	400G		11-125						
	450K/H								
	500K/H								
	560H/G								

Table 2.3: Sleeve bearings lubrication data for rib-cooled motors [6]

Poles	Frame Size	Lubrication Method	Sleeve Bearing	Lubricant	Specification	Lubrication interval (hours)	Oil Quantity (liters)	Viscosity Range	
								ISO 40°C	SSU 100°F
2P	560J/H	Self-Lubrication	9-90	MOBIL DTE 24	Mineral Oil ISO VG32	8000	2.8	28.8-35.2 [cSt]	137-164 [s]
	560H/G		11-100				4.2		
	630K/J						Oil Circulation		
	630J/H	4.2							
	710K/J	11-110	4.2						
4P+	560J/H	Self-Lubrication	11-125	MOBIL DTE 25	Mineral Oil ISO VG46	8000	4.2	41.4-50.6 [cSt]	193-235 [s]
	560H/G		14-140				8		
	630K/J						14-160		
	630J/H		8						
	710J/H		8						
	710H/G		8						

Table 2.4: Sleeve bearings lubrication data for tube-cooled motors [6]

2.3.3 Sleeve bearings beneficts and limitations

Sleeve bearings are typically lubricated with oil, and the viscosity of the oil is selected based on operating temperature ranges and other factors like heat generation and bearing properties. Properly maintained sleeve bearings are theoretically considered to have an infinite life, making them reliable in long-term operations [14].

Mechanical looseness in sleeve bearings refers to a common problem where there is excessive clearance between the components of the sleeve bearing system. This looseness can occur due to various factors such as loose bolts, structural damage, improper fits, or increased clearance between the bearing components [15].

The most typical causes of mechanical looseness in sleeve bearings are the excessive clearance between the bearing inner surface and the shaft and the excessive clearance between the sleeve bearing outer surface and the bearing housing inner surface.

When there is mechanical looseness in sleeve bearings, it can lead to increased vibration, wear, and potential damage to the bearing components. The presence of mechanical looseness can disrupt the stable operation of the sleeve bearings, causing deviations in the shaft's stable operating condition and leading to issues such as unbalance, misalignment, or excessive/cyclic loading.

Sleeve bearings play a crucial role in vibration control within mechanical systems. By providing stable support and reducing friction between moving parts, sleeve bearings help dampen vibrations that can lead to system instability. Active sleeve bearings, equipped with control mechanisms, can actively adjust stiffness levels and damping properties to counteract vibrations effectively. This capability is essential in applications where precise control over vibrations is necessary to ensure smooth and reliable operation [16].

One of the key advantages of sleeve bearings is their adaptability to changing operating conditions. Active sleeve bearings can dynamically adjust their parameters in response to varying

loads, speeds, and environmental factors. This adaptability allows the bearings to maintain optimal performance across a wide range of operating conditions, enhancing system reliability and longevity. By continuously monitoring and adjusting their characteristics, sleeve bearings can ensure consistent and efficient operation under diverse circumstances.

2.3.4 Endshield

Flameproof induction motors require specific endshield designs to ensure safe and efficient operation in potentially hazardous environments, Fig.2.8. These endshields prioritize thermal management, structural integrity, and ease of maintenance [6].

The drive-end endshield features a finned design for superior heat dissipation, which lowers bearing operating temperatures and extends lubrication intervals. In contrast, the non-drive endshield prioritizes minimal airflow resistance for optimal motor cooling. Both designs utilize steel construction for maximum strength and rigidity, while minimizing welding to optimize the manufacturing process and reduce potential weak points. Internal ribbing further reinforces the endshields' structural integrity.

A key feature is the careful design of the fixing bolt housings. These housings protect the bolts from accidental impacts and water accumulation, safeguarding the motor's structural integrity. Some housings are even threaded to allow for easier mounting, dismounting, and handling of the endshields during maintenance or installation. The smooth external design of the endshields simplifies cleaning procedures, further reducing maintenance time and effort. Additionally, both drive and non-drive end shields can be supplied with flat areas for the easy placement of vibration probes in either vertical or horizontal orientations. Standard M8 threaded nipples are provided for vibration probe attachment, with the option for customization of threaded hole sizes to accommodate specific needs. Vibration probes can also be supplied along with the motor upon request.



Figure 2.8: W22Xd tube-cooled motors endshield [6]

2.3.5 Analysis of the Characteristics and Specifications of Flameproof Motors

The W22XdbT series, which is the one who will be studied on this project, designated with the letter "T" featuring tube cooling, and designated with "X" for flameproof capability, comprises a line of electric motors within the W22Xdb family. These motors stand out due to their tube cooling mechanism, utilizing longitudinal tubes in the casing to facilitate forced air circulation, ensuring efficient heat dissipation compared to traditional fin-based cooling systems. Spanning shaft heights from 560 to 900 millimeters, these motors are designed to meet diverse industrial needs. To enable optimal airflow within the motor, the rotor is equipped with axial channels, typically achieved through strategically drilled holes in the rotor plate. This feature enhances temperature regulation and overall motor performance [6].

The electric motor that was analyzed has 3400 kW, 4 poles, a shaft height of 900 mm, and operates at 50 Hz.

In the motor under study, axial misalignment is observed due to various factors such as vibrations, manual assembly errors, and the force exerted by the external fan. It is important to note that the motor has two internal fans; however, since they are symmetrically positioned, the forces they exert cancel each other out. Therefore, only the force from the external fan causes axial displacement. Consequently, it is essential to ensure the axial alignment of the rotor with its magnetic center to achieve optimal performance.

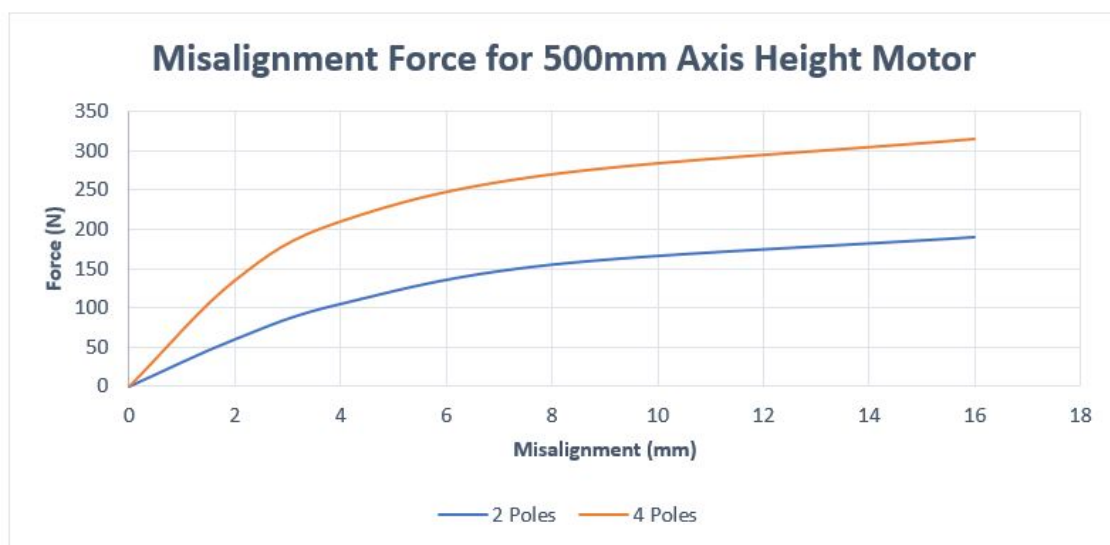


Figure 2.9: Misalignment of the Rotor due to Axial Force

For a motor with a shaft height of 500mm, WEG Brazil conducted an analysis on the misalignment of this motor depending on the axial force applied (Fig. 2.9).

Although there are no specific results for the motor under study, we can infer the misalignment values based on the motor's polarity. Since the motor in question is a 4-pole motor, it requires a

greater force to achieve significant misalignment. This suggests that the 4-pole motor is more resilient to axial displacement, ensuring better alignment under typical operational forces.

Chapter 3

Numerical Simulations and Dynamic Responses

3.1 Introduction to Numerical Simulations

Numerical simulations and simulation techniques are indispensable tools for predicting motor performance and optimizing design parameters. This section introduces theoretical models grounded in electromagnetic theory, thermodynamics, and fluid dynamics, which are utilized to simulate motor behavior. Discuss the role of computational tools such as finite element analysis (FEA), computational fluid dynamics (CFD), and multi-physics simulations in theoretical motor optimization.

3.1.1 Finite Element Analysis

Finite Element Analysis (FEA) is a numerical technique used to approximate solutions to complex engineering problems involving varying shapes, boundary conditions, and loads. Despite providing approximate solutions, FEA is widely utilized in engineering due to its diversity and flexibility as an analysis tool. With advancements in computer hardware technology, FEA has gained significant attention, as computational resources are essential for its application. Several commercial software packages such as STAAD-PRO, GT-STRUDEL, NASTRAN, ABAQUS, and ANSYS facilitate the analysis of complex structures using FEA [17].

Originally developed for stress analysis in aircraft design, FEA has expanded its applications beyond solid mechanics to encompass fluid flow, heat transfer, electric and magnetic fields, among others. Civil engineers extensively employ FEA for analyzing various structures including beams, frames, plates, shells, foundations, and rock mechanics problems. FEA handles both static and dynamic problems and finds applications in the design of ships, aircraft, electric motors, and heat engines.

Engineering problems often involve unknown field variables whose determination predicts the behavior of the entire structure. These field variables, such as displacements, velocities, potentials, or temperatures, are infinite in a continuum. FEA tackles this by reducing the unknowns to a finite number through discretization into small elements and expressing the field variables using approximating functions within each element. These approximating functions, often referred to as shape functions or interpolation functions, are defined in terms of field variables at specified points known as nodes.

The steps involved in FEA can be summarized as follows:

1. Select suitable field variables and elements.
2. Discretize the continuum.
3. Choose interpolation functions.
4. Determine element properties.
5. Assemble element properties to obtain global properties.
6. Impose boundary conditions.
7. Solve the system equations to determine nodal unknowns.
8. Make additional calculations to obtain required values.

In FEA, the relationship between element stiffness matrix $[k]_e$, nodal displacement vector $\{\delta\}_e$, and nodal force vector $\{F\}_e$ is expressed as:

$$[k]_e \{\delta\}_e = \{F\}_e \quad (3.1)$$

where k_{ij} represents the force in coordinate direction i due to a unit displacement in coordinate direction j .

3.1.1.1 Natural and Local Coordinate Systems

The concepts of natural and local coordinate systems are fundamental in the field of finite element analysis (FEA). These coordinate systems play an important role in defining the position and behavior of elements within a larger system, aiding in the accurate representation of deformation and response in structures and mechanical systems [18].

The natural coordinate system, Fig. 3.1, also known as the intrinsic coordinate system, is specific to each element. It is defined such that the sides of the element are located at particular values of the natural coordinates. For instance, in a triangular element, the natural coordinates α and β are chosen such that the sides of the triangle are positioned at $\alpha = 0$, $\beta = 0$, and $1 - \alpha - \beta = 0$. These natural coordinates allow for the convenient expression of shape functions and element behavior.

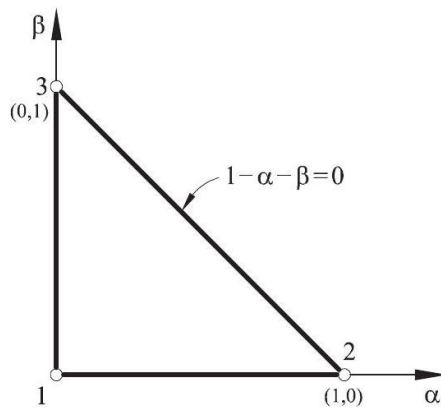
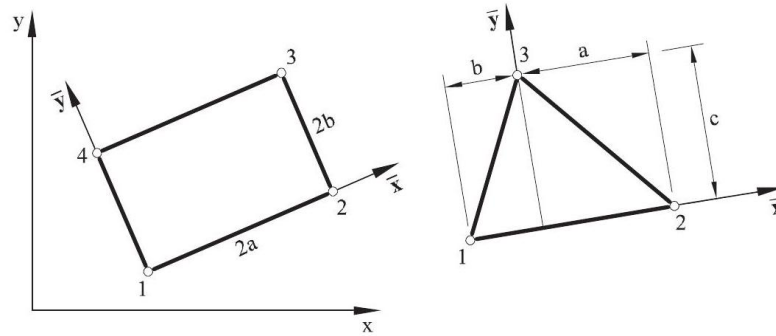


Figure 3.1: Natural coordinates for a triangular element [18]

The local coordinate system, Fig. 3.2, on the other hand, is defined with respect to a particular element. It is typically used to describe the position and behavior of the element within the global coordinate system. The local coordinate system is often aligned with the element's geometry or orientation. For example, in a rectangular element, the x -axis can be aligned with the longer side of the rectangle, while the y -axis aligns with the shorter side.

Figure 3.2: Local coordinate system $\bar{x}, \bar{y}$ for the analytical computation of integrals over triangular and rectangular elements [18]

The relationship between the natural and local coordinate systems can be established through coordinate transformation matrices.

$$\begin{bmatrix} x \\ y \end{bmatrix} = T \begin{bmatrix} \xi \\ \eta \end{bmatrix} \quad (3.2)$$

These matrices facilitate the transformation of quantities defined in one coordinate system to the other. For instance, a 2×2 coordinate transformation matrix T can be used to transform quantities from the local coordinate system (x, y) to the global axes.

3.1.1.2 Continuity

Continuity is a fundamental concept in the context of the Finite Element Method (FEM) that ensures smooth and consistent behavior of the solution within and between elements. It refers to the requirement that the displacement field and its derivatives should be continuous across the domain of analysis. In FEM, the displacement field is typically approximated using polynomial functions within each element. The concept of continuity can be understood in three different conditions: continuity of displacement within each element, continuity of derivatives of the displacement, and continuity of the shape functions [18].

The displacement field must be continuous within each element. This means that the displacement values at any point within an element should not have any sudden jumps or discontinuities. To achieve this, polynomial approximations are used to represent the displacement field. These polynomial approximations ensure that the displacement field is continuous within each element.

The derivatives of the polynomial approximation should exist up to the order of the derivatives appearing in the element integrals. This means that if the element integrals involve first-order derivatives of the displacement, the shape functions should be at least first-order polynomials. This condition ensures that the derivatives of the displacement field are also continuous within each element.

Shape functions are mathematical functions that describe the deformation of an element. They are crucial in Finite Element Analysis (FEA) as they interpolate the values of unknowns, such as displacements or temperatures, within the element based on known values at the element nodes. In the context of triangles, shape functions are often expressed as polynomials of the natural coordinates.

The natural coordinates, denoted as α and β , provide a coordinate system specific to each triangle. They are defined such that the triangle's sides lie along the lines $\alpha = 0$, $\beta = 0$, and $1 - \alpha - \beta = 0$. These coordinates enable expressing the shape functions of triangular elements in terms of the natural coordinates, particularly useful for deriving isoparametric triangular elements.

For a 3-noded triangle, the shape functions can be written as:

$$N_1 = 1 - \alpha - \beta$$

$$N_2 = \alpha$$

$$N_3 = \beta$$

These shape functions are linear polynomials and describe the variation of the unknowns within the triangle. The values of α and β range from 0 to 1, and their sum should be less than or equal to 1.

The area coordinates of the triangle, denoted as L_1 , L_2 , and L_3 , coincide with the natural coordinates α and β , respectively. Specifically, $L_1 = 1 - \alpha - \beta$, $L_2 = \alpha$, and $L_3 = \beta$. These area coordinates are used to define the position of the nodes within the element.

These shape functions determine the distribution of the displacement values within each element. To ensure continuity, the shape functions must satisfy certain conditions. For example, in 1D elements, the shape functions are typically chosen as linear or higher-order polynomials. These shape functions should satisfy the condition of nodal compatibility, which ensures that the shape functions have a value of 1 at the corresponding node and 0 at all other nodes within the element.

For a cubic rod element with four nodes at $\xi_1 = -1$, $\xi_2 = -1/3$, $\xi_3 = 1/3$ and $\xi_4 = +1$ in 3.3, the shape functions are:

$$\begin{aligned} N_1 &= \frac{(\xi - \xi_2)(\xi - \xi_3)(\xi - \xi_4)}{(\xi_1 - \xi_2)(\xi_1 - \xi_3)(\xi_1 - \xi_4)} = -\frac{9}{16} \left(\xi^2 - \frac{1}{9} \right) (\xi - 1) \\ N_2 &= \frac{(\xi - \xi_1)(\xi - \xi_3)(\xi - \xi_4)}{(\xi_2 - \xi_1)(\xi_2 - \xi_3)(\xi_2 - \xi_4)} = \frac{27}{16} \left(\xi - \frac{1}{3} \right) (\xi^2 - 1) \\ N_3 &= \frac{(\xi - \xi_1)(\xi - \xi_2)(\xi - \xi_4)}{(\xi_3 - \xi_1)(\xi_3 - \xi_2)(\xi_3 - \xi_4)} = -\frac{27}{16} \left(\xi + \frac{1}{3} \right) (\xi^2 - 1) \\ N_4 &= \frac{(\xi - \xi_1)(\xi - \xi_2)(\xi - \xi_3)}{(\xi_4 - \xi_1)(\xi_4 - \xi_2)(\xi_4 - \xi_3)} = \frac{9}{16} (\xi + 1) \left(\xi^2 - \frac{1}{9} \right) \end{aligned} \quad (3.3)$$

3.1.1.3 Differentiability

Differentiability is a fundamental concept in calculus that describes the smoothness and continuity of a function. A function is said to be differentiable at a point if it has a derivative at that point. The derivative represents the rate of change of the function at that point. Mathematically, a function $f(x)$ is said to be differentiable at a point $x = a$ if the following limit exists [18]:

$$f'(a) = \lim_{h \rightarrow 0} \frac{f(a+h) - f(a)}{h} \quad (3.4)$$

It represents the derivative of the function $f(x)$ at the point $x = a$. The concept of differentiability can be understood in terms of the tangent line to the graph of a function at a specific point. If a function is differentiable at a point, it means that the tangent line at that point exists and is well-defined. The derivative of a function provides important information about its behavior. It gives the slope of the tangent line at each point and can be used to determine the rate of change, the direction of the function, and the concavity of the graph. Differentiability is closely related to continuity. A function must be continuous at a point in order to be differentiable at that point. However, not all continuous functions are differentiable. Discontinuities, sharp corners, and vertical tangents are examples of situations where a function may not be differentiable. Differentiability also has important implications for optimization and approximation. In optimization problems, the derivative is used to find the maximum or minimum values of a function. In approximation problems, the derivative is used to estimate the behavior of a function near a specific point.

Differentiability is vital in the Finite Element Method (FEM) as it ensures the accuracy of solutions obtained. It guarantees smoothness in displacement fields, essential for accurately representing structural behavior. FEM approximates these fields using polynomials, where differentiability

is crucial for precision. Integration and differentiation, inherent in FEM, rely on this property for accurate calculations. Additionally, differentiability aids in error estimation and convergence assessment, allowing for refined mesh and more accurate results. It also ensures compatibility with mathematical models and physical laws, enhancing result reliability.

3.1.1.4 Computation of Stiffness Matrix

The stiffness matrix plays a crucial role in Finite Element Method (FEM) analysis, representing the relationship between applied forces and resulting displacements in structural or mechanical systems. It is essential for solving the system of equations governing the system's behavior under external loads.

The stiffness matrix is derived from the strain-displacement relationship and the material properties of the system. It is a square matrix relating the applied forces (or stresses) to the resulting displacements (or strains), and its size depends on the number of degrees of freedom in the system [18].

The general expression for the stiffness matrix is given by:

$$[K] = \int [B]^T [D] [B] dV \quad (3.5)$$

where:

- $[K]$ is the stiffness matrix,
- $[B]$ is the strain-displacement matrix,
- $[D]$ is the material elasticity matrix,
- dV is the differential volume element.

The strain-displacement matrix $[B]$ relates the strains in the system to the displacements and is obtained from the derivatives of the shape functions describing the element's deformation. The material property matrix $[D]$ encapsulates elastic properties like Young's modulus and Poisson's ratio.

The integral represents integration over the element's volume, typically performed numerically using techniques like numerical quadrature.

The stiffness matrix is assembled by summing contributions from each element in the system. The resulting system of equations is then solved to determine displacements and stresses.

3.1.1.5 Rigid Body and Deformation

This section delves into the fundamental concepts of continuum mechanics, particularly focusing on the quantification of displacements and deformations of objects [19].

Displacements are characterized by changes in the positions of points within a material body. In continuum mechanics, rigid body displacements refer to motions where the object moves without undergoing any internal deformation, Fig. 3.3. Mathematically, a rigid body displacement is represented as:

$$\mathbf{x} = \mathbf{X} + \mathbf{u}$$

where $\mathbf{x}$ denotes the deformed position, $\mathbf{X}$ represents the undeformed position, and $\mathbf{u}$ is the displacement vector.

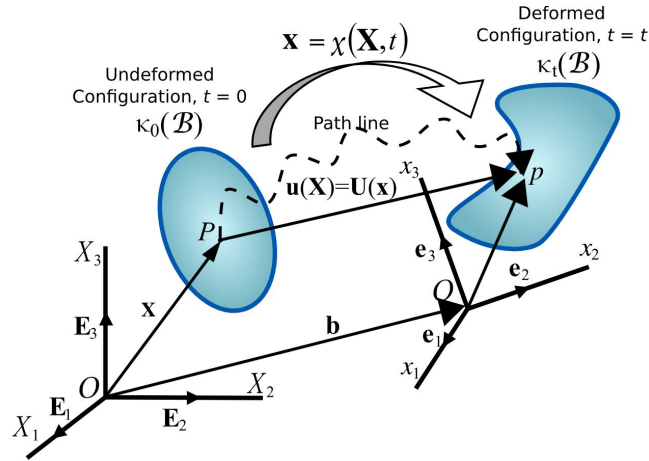


Figure 3.3: Displacement of a continuum body, from a reference configuration to the current configuration [19]

Deformations involve changes in the shape and size of a material body, including bending, twisting, stretching, and shearing. Unlike rigid body displacements, deformations generate stress within the material. The primary challenge in continuum mechanics is to accurately quantify deformations and separate them from rigid body motions.

The deformation gradient ($\mathbf{F}$) is a fundamental tensor that quantifies the local deformation of a material body. It is defined as the derivative of each component of the deformed position vector ($\mathbf{x}$) with respect to each component of the undeformed position vector ($\mathbf{X}$). Mathematically, the deformation gradient is expressed as:

$$F_{ij} = \frac{\partial x_i}{\partial X_j}$$

or equivalently,

$$\mathbf{F} = \frac{\partial \mathbf{x}}{\partial \mathbf{X}}$$

In tensor notation, this can be written as:

$$\mathbf{F} = \begin{bmatrix} \frac{\partial x_1}{\partial X_1} & \frac{\partial x_1}{\partial X_2} & \frac{\partial x_1}{\partial X_3} \\ \frac{\partial x_2}{\partial X_1} & \frac{\partial x_2}{\partial X_2} & \frac{\partial x_2}{\partial X_3} \\ \frac{\partial x_3}{\partial X_1} & \frac{\partial x_3}{\partial X_2} & \frac{\partial x_3}{\partial X_3} \end{bmatrix}$$

or more compactly,

$$\mathbf{F} = \begin{bmatrix} F_{11} & F_{12} & F_{13} \\ F_{21} & F_{22} & F_{23} \\ F_{31} & F_{32} & F_{33} \end{bmatrix}$$

The deformation gradient is central to separating rigid body motions from deformations, making it a key concept in continuum mechanics.

Rigid body rotations are a subset of rigid body displacements where the object undergoes rotational motion without any deformation. Mathematically, a rigid body rotation can be represented by a rotation matrix ($\mathbf{R}$). The presence of rigid body rotations complicates the interpretation of the deformation gradient, as they can erroneously appear as deformations if not properly accounted for.

3.1.2 Computational Fluid Dynamics (CFD)

Computational Fluid Dynamics (CFD) analysis is a numerical technique used to study fluid flow and heat transfer phenomena in various engineering applications. The core of CFD analysis involves solving the Navier-Stokes equations, which are fundamental equations governing fluid motion. These equations consist of the continuity equation, momentum equation, and energy equation [20].

1. Continuity Equation: The continuity equation represents the conservation of mass and is given by:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0 \quad (3.6)$$

where: - ρ is the fluid density, - t is time, - $\mathbf{u}$ is the velocity vector, - ∇ is the del operator representing the divergence of the velocity field.

This equation ensures that mass is conserved within the fluid domain, accounting for the inflow and outflow of mass at any point in the system.

2. Momentum Equation: The momentum equation describes the conservation of momentum and is expressed as:

$$\frac{\partial(\rho \mathbf{u})}{\partial t} + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) = \nabla \cdot (-p \mathbf{I} + \mu (\nabla \mathbf{u} + (\nabla \mathbf{u})^T) + \rho \mathbf{g}) \quad (3.7)$$

where: - p is the pressure, - μ is the dynamic viscosity, - $\mathbf{I}$ is the identity matrix, - $\mathbf{g}$ is the gravitational acceleration vector.

This equation accounts for the changes in momentum within the fluid due to pressure forces, viscous forces, and gravitational forces.

3. Energy Equation: The energy equation represents the conservation of energy and is given by:

$$\frac{\partial(\rho C_p T)}{\partial t} + \nabla \cdot (\rho \mathbf{u} C_p T) = \nabla \cdot (k \nabla T) + \dot{q} \quad (3.8)$$

where: - C_p is the specific heat at constant pressure, - T is the temperature, - k is the thermal conductivity, - $\dot{q}$ represents heat sources or sinks.

This equation accounts for the energy transfer within the fluid domain due to conduction, convection, and heat sources or sinks.

By discretizing these equations, applying appropriate boundary conditions, and utilizing numerical methods, CFD analysis provides detailed insights into fluid flow patterns, temperature distributions, and other important parameters, aiding in the design and optimization of processes, Fig. 3.4.

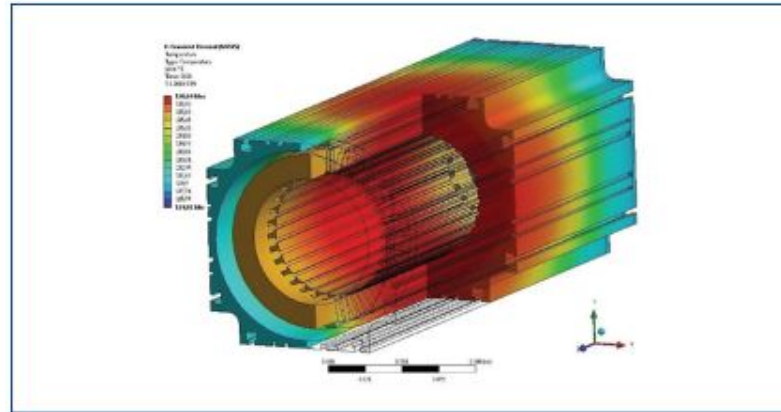


Figure 3.4: CFD simulation of temperature distribution on a stator [21]

3.2 Computational Methods for Electric Motor Simulations

Finite Element Analysis (FEA) is commonly used to analyze the structural and thermal behavior of motors by dividing them into smaller elements for detailed analysis. This method helps in predicting temperature distribution, stress levels, and other critical parameters within the motor components. By utilizing FEA, designers can optimize the motor's design for enhanced performance and reliability by identifying areas of high stress or temperature gradients [21].

On the other hand, Computational Fluid Dynamics (CFD) is essential for studying the flow of air or coolant within the motor to ensure effective cooling. By analyzing heat transfer mechanisms,

such as convection, CFD aids in predicting temperature distribution on motor surfaces. Through simulating airflow patterns and heat dissipation, CFD assists in optimizing the motor's cooling system to prevent overheating and ensure efficient operation.

Moreover, Multi-physics simulations combine FEA and CFD to study the interaction between structural, thermal, and fluid flow aspects of motors. These simulations provide a comprehensive understanding of how different physical phenomena influence each other in the motor system. By integrating multiple physics models, designers can optimize motor performance by considering the coupled effects of structural integrity, thermal management, and fluid dynamics .

3.2.1 FEM and ANSYS Maxwell

One of the primary computational methods extensively utilized in electric motor simulations is the Finite Element Method (FEM). The FEM is a numerical technique that discretizes the motor components into smaller elements, facilitating the analysis of electromagnetic fields, structural dynamics, and other physical phenomena within the motor system. By solving partial differential equations for each element, the FEM provides detailed insights into the electromagnetic interactions, mechanical stresses, and thermal behavior of electric motors [22].

Moreover, computer-aided design (CAD) modeling plays a crucial role in the simulation process, Fig. 3.5, by enabling the creation of detailed motor models for analysis. CAD models, developed using software like SolidWorks, serve as the foundation for conducting finite element analysis of electric motors, allowing researchers to explore the performance characteristics and behavior of the motor under different operating conditions.

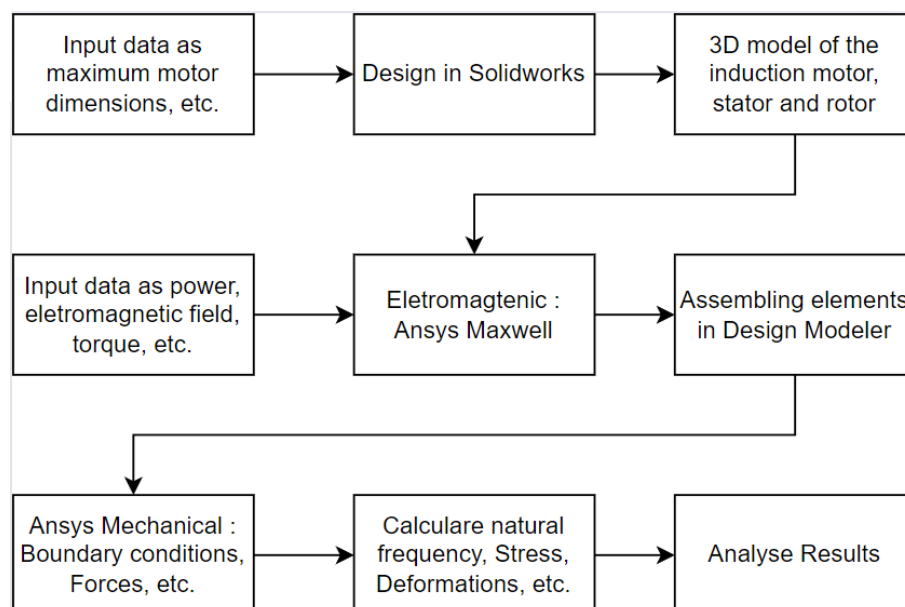


Figure 3.5: Flowchart of a finite element computational simulation routine

Ansys Maxwell employs the Finite Element Method (FEM) to conduct detailed electromagnetic field analyses of electric motors, Fig. 3.6, breaking down motor components into smaller elements for precise calculations. This approach allows for a comprehensive examination of electromagnetic interactions, current densities, and magnetic flux distributions within the motor system.

By simulating electromagnetic interactions and losses, engineers can fine-tune motor parameters to enhance performance and efficiency. The software's integration with CAD tools facilitates the creation of detailed motor models for comprehensive analysis and optimization, ensuring that the designed motors meet specific performance requirements [23].

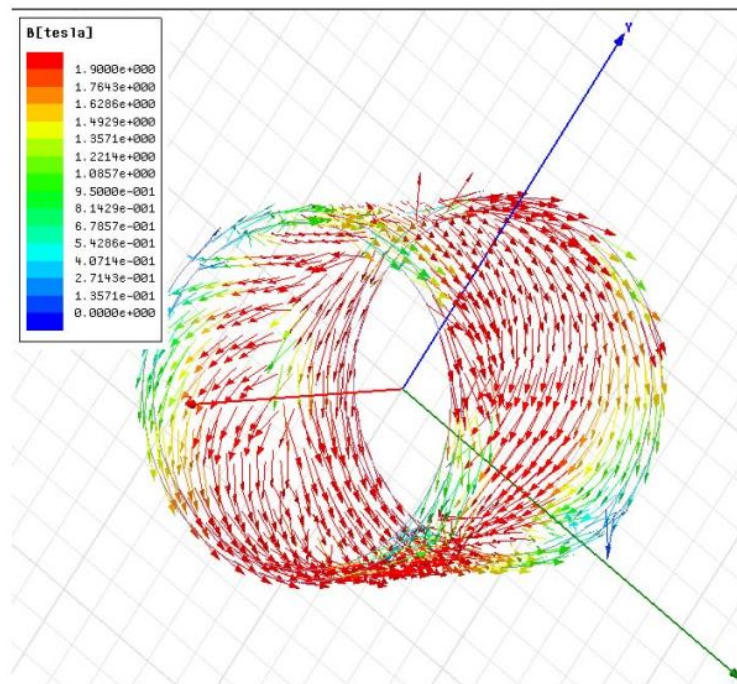


Figure 3.6: The direction of the vector lines of the magnetic field in the stator [23]

The software conduct modal analysis of motor components, identifying natural frequencies and potential resonance issues. This analysis is crucial for ensuring the structural stability and reliability of electric motors during operation.

3.2.2 Thermal and Electromagnetic Energies

One of the primary techniques utilized for thermal analysis is Finite Element Analysis (FEA), which enables researchers to discretize motor components and simulate heat transfer phenomena accurately. By leveraging FEA simulations, researchers can calculate temperature profiles, thermal gradients, and heat flow paths within the motor system. This allows for the assessment of different cooling strategies and the optimization of thermal management solutions to ensure optimal operating temperatures and mitigate thermal stresses [24].

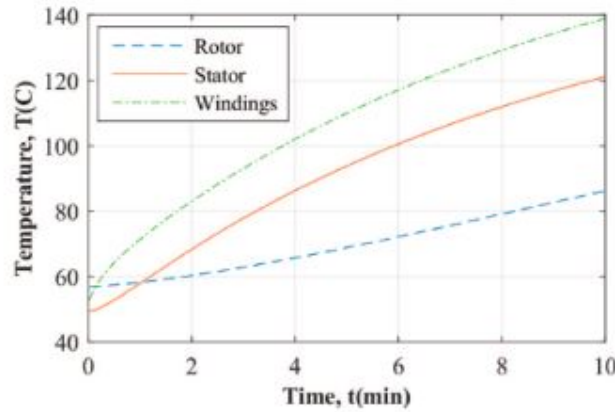


Figure 3.7: Rise in temperature under instantaneous load scenario [24]

To calculate temperature profiles and cooling flow within the motor, researchers set up detailed simulations in Ansys, defining material properties, boundary conditions, and heat transfer mechanisms. By running simulations with varying operating conditions and cooling strategies, researchers can analyze temperature distributions across motor components, optimize cooling system designs, and ensure efficient heat dissipation.

3.2.3 Deformation and Vibration

One of the primary techniques used for vibration analysis is the Experimental Modal Technique (EMT), which assists in locating deformations in the motor structure and identifying areas sensitive to vibrations. EMT allows researchers to predict vibration velocities and modes of vibration, providing insights into the dynamic behavior of the motor under different operating conditions [25].

Vibrational Analysis (VA) is another key technique utilized to forecast vibration velocities experimentally and assess the impact of vibrations on motor performance. By conducting VA on the motor system, researchers can identify excessive vibration levels that may lead to reduced lifespan or operational issues. VA helps in understanding the vibration characteristics of the motor and guides the redesign process to optimize structural integrity and reduce vibration levels.

The primary findings from the vibration analysis conducted on the induction motor are outlined as follows:

1. Vibration Velocity Reduction:

- Prior to any modifications, the vibration velocity within the induction motor was recorded.
- After implementing adjustments, a significant reduction in vibration velocity was observed.

- Experimental analysis indicated a noticeable decrease in vibrations post-adjustment.

2. Comparison with Existing Studies:

- A comparative analysis with previous literature on vibration reduction techniques was conducted.
- The methodologies employed in this study exhibited noteworthy effectiveness compared to other approaches.

3. Fault Identification:

- The vibration analysis facilitated the detection of various faults within the induction motor.
- Precise fault localization was achieved through the employed techniques.

4. Structural Adjustments:

- Structural modifications can be implemented based on the analysis outcomes.
- These adjustments aimed to mitigate vibrations and improve motor performance.

5. Enhancement of Reliability:

- By addressing vibration issues, the overall reliability of the motor can be enhanced.
- The modifications can contribute to meeting industry standards and reducing operational costs.

Also Finite Element Analysis (FEA) is commonly employed for simulating structural deformations and predicting vibration modes in electric motors. By setting up detailed FEA simulations, researchers can model the motor structure, apply dynamic loads, and analyze the response of the system to vibrations. FEA enables researchers to calculate vibration velocities, deflections, and modes of vibration, providing valuable insights into the structural behavior of the motor under dynamic conditions [26].

3.2.4 Interference and Structural Stress

In the realm of structural analysis for machines like Induction Machines (IMs) and Switched Reluctance Machines (SRMs), simulations play a pivotal role in understanding the impacts of interference fits and structural stresses. Simulating interference fits is crucial as it is a prevalent method for attaching shafts to rotor cores in high-speed machines. The process typically involves creating a simplified model (example in Fig. 3.8) using Finite Element Analysis (FEA) software such as ANSYS, where two disks representing rotor components are defined with specific dimensions and material properties. The difference between the diameters of these disks signifies the interference dimension, a critical parameter in the simulation. By calculating radial and tangential stresses induced by the interference fit through equations and FEA simulations, engineers can assess the effects of this attachment method on the rotor [27].

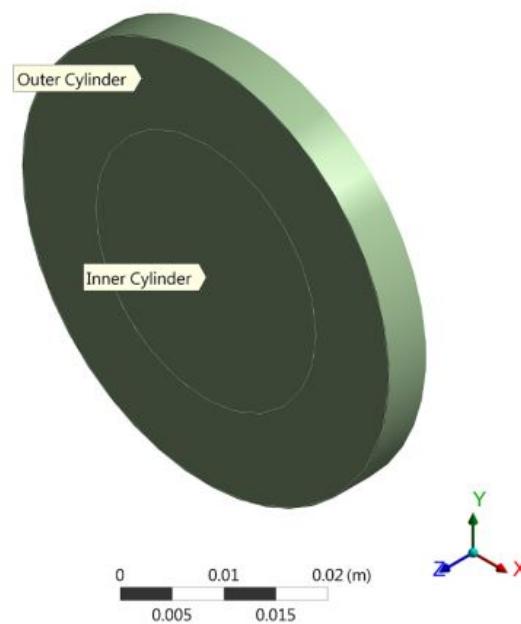


Figure 3.8: Simplified model of rotor components [27]

On the other hand, simulating structural stresses is essential for evaluating the structural integrity of machine components influenced by factors like thermal expansion, centrifugal forces, and operational speeds. This process entails developing a detailed model of machine components in FEA software, considering material properties, geometries, and boundary conditions. Thermal stresses, centrifugal forces, and other structural loads are then applied to the model to analyze resulting stresses and deformations in the components. Verification of simulation results against theoretical calculations and experimental data is crucial to validate the accuracy of the stress analysis and ensure the model's predictive capabilities.

3.3 Vibrational Responses

Mechanical vibrations, which refer to the oscillatory motions of mechanical systems, are fundamental in various engineering applications, including machinery, structures, vehicles, and musical instruments. Understanding mechanical vibrations is crucial for designing and analyzing systems to ensure their stability, performance, and durability [28, 29].

In the context of mechanical engineering, the study of vibrations involves analyzing the dynamic behavior of systems subjected to external forces or disturbances. This analysis helps engineers predict and control the response of the system, ensuring its safe and efficient operation.

The study of mechanical vibrations covers a wide range of topics, including understanding the fundamental principles of vibration, modeling and analyzing mechanical systems using mathematical equations such as differential equations, analyzing single-degree-of-freedom systems charac-

terized by parameters like natural frequency, damping ratio, and response amplitudes, extending the analysis to multi-degree-of-freedom systems using modal analysis and matrix methods, studying forced vibrations and their effects on mechanical systems, exploring techniques for mitigating or controlling vibrations in mechanical systems, and finding practical applications of mechanical vibrations in various fields such as automotive engineering, aerospace, civil engineering, and manufacturing. Understanding and controlling vibrations are essential for ensuring the performance, safety, and reliability of these systems.

3.3.1 Degrees of freedom

The concept of degrees of freedom refers to the number of independent coordinates needed to fully describe the motion of a physical system. In other words, it is the number of different ways in which an object or system can move independently [29].

In the context of mechanics, degrees of freedom are used to describe the kinematics and dynamics of a system. For example, in a system with a single mass point, such as a particle, only one coordinate is needed to describe its position in three-dimensional space. Therefore, this system has one degree of freedom.

However, in more complex systems, such as a double pendulum or a system of coupled springs, more independent coordinates are needed to fully describe the motion. Each coordinate represents a variable that describes the position or state of a part of the system. The total number of degrees of freedom is equal to the number of independent coordinates needed.

For example, a system with two mass points coupled by springs may have two degrees of freedom. This means that two independent coordinates are needed to fully describe the system's motion. Each coordinate represents the position of one of the mass points along a specific direction.

Degrees of freedom are important in the analysis and design of mechanical systems as they determine the number of equations of motion needed to describe the system's behavior, Fig. 3.9. Additionally, they influence the system's response to external forces and the possible configurations the system can assume.

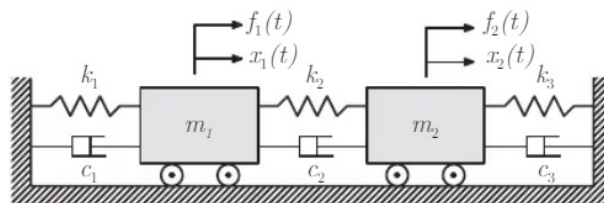


Figure 3.9: Simplified scheme of two degrees of freedom [29]

3.3.2 Fundamentals of Vibratory Systems

The response of a vibration refers to the behavior and characteristics of a vibrating system when subjected to an external force or excitation. It describes how the system reacts and moves in response to the applied force. The equation of motion for a vibrating system is commonly used to describe the response of the system. In the case of a single-degree-of-freedom system, the equation of motion can be written as [29]:

$$m\ddot{x} + c\dot{x} + kx = F(t) \quad (3.9)$$

This equation represents a damped harmonic oscillator subjected to an external force $F(t)$. In this equation kx denotes the force exerted by the spring, where k is the stiffness of the spring and x is the displacement of the object from its equilibrium position. $c\dot{x}$ represents the force exerted by the damper where c is the damping coefficient, quantifying the dissipative forces opposing the motion and $\dot{x}$ represents the velocity of the object, which is the first derivative of displacement x with respect to time. $m\ddot{x}$ is the exerted inertia of the system where m represents the mass of the object undergoing oscillatory motion and $\ddot{x}$ denotes the acceleration of the object [28, 29].

3.3.2.1 Static Deflection

The static deflection of a simple mass-spring system is the deflection of spring k as a result of the gravitational force acting on the mass. It represents the displacement from the equilibrium position when the system is in a static or unchanging state, usually under the influence of gravity alone [30].

In mathematical terms, the static deflection (δ_{st}) can be calculated using the formula:

$$\delta_{st} = \frac{mg}{k} \quad (3.10)$$

In a structural view, the deflection of a beam is typically calculated by integrating the bending moment equation twice. The relationship between the bending moment and the beam deflection is given by the differential equation [31]:

$$\frac{d^2v}{dx^2} = \frac{M(x)}{EI} \quad (3.11)$$

For example, the deflection at the center of a simply supported beam with a point load applied at the center can be calculated using the formula:

$$\delta = \frac{48EI}{PL^3} \quad (3.12)$$

Engineers typically use methods like the method of initial conditions, requiring them to solve complex differential equations to determine the deflection, rotation, bending moment, and shear force at different points along the beam with more complex shapes. Boundary conditions, like fixed or simply supported ends, are applied to solve this equation and find the deflection profile.

Using the method of initial conditions involves determining deformation and force parameters at one end of the beam based on boundary conditions at the other [32].

3.3.2.2 Damping and Resonance

Damping and resonance are pivotal concepts in vibration and dynamic systems analysis. Damping refers to energy dissipation within a vibrating system, while resonance occurs when an external force frequency aligns with the system's natural frequency [28, 29].

Damping significantly influences a system's response control. Absence of damping near resonance may result in destructive oscillations due to amplified amplitudes. However, damping curtails vibration amplitudes by dissipating system energy, thus averting excessive oscillations.

Resonance, conversely, involves substantial vibration amplification when the external force frequency matches the system's natural frequency. This heightened vibration can lead to structural stresses, potentially causing damage or failure.

Damping's role in controlling system response near resonance is crucial, Fig. 3.10. By introducing damping, either through damping materials or other means, system response can be regulated, preventing excessive vibration amplitudes. Practical applications often necessitate damping introduction to manage system response, especially in scenarios where complete resonance avoidance is impractical, such as variable-speed electric motors or internal combustion engines.

The requisite damping level for effective response control hinges on several factors, including system natural frequency, forcing frequency, and desired vibration control level. Adequate damping is essential to limit vibration amplitudes and transmitted forces near resonance, while avoiding excessive force transmission during operation.

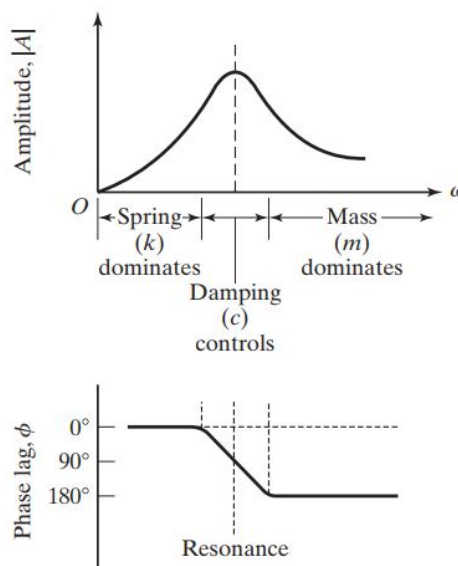


Figure 3.10: Damping and Resonance [28]

Vibration measurement is conducted using instruments known as accelerometers, capable of detecting and quantifying the acceleration of objects or structures undergoing vibratory motion. These devices convert acceleration into an electrical signal, facilitating recording and analysis. Alongside accelerometers, sensors for displacement, velocity, and force are employed for vibration measurement, depending on the system type and analysis objectives.

3.3.3 Description and formulation of vibration problems

Its formulation involves the use of mathematical equations and formulas to analyze and solve engineering vibration problems. The approach applies the knowledge of mathematics, including differential equations, matrix algebra, vector methods, and complex numbers, as well as principles of mechanics and dynamics [28].

In terms of formulas, a wide range of topics related to vibration problems are covered. Some of the commonly used formulas in the field of mechanical vibrations include:

1. **Natural frequency formula:** For a single-degree-of-freedom system:

$$f_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \quad (3.13)$$

The natural frequency, a cornerstone of mechanical vibrations, denotes the frequency at which a system naturally oscillates in the absence of external forces. Its importance is evident in various critical aspects [28, 29]:

In terms of system stability, matching the excitation frequency with the natural frequency can lead to resonance, potentially inducing system failure due to excessive displacements and stresses. Thus, engineers diligently regulate natural frequencies to avert resonance and uphold system stability.

When delving into design considerations for structures and machinery, engineers meticulously factor in natural frequencies to ensure operations fall within safe vibration thresholds. By optimizing these frequencies, engineers bolster system performance and reliability.

In efforts to sidestep resonance and its adverse effects on system functionality and longevity, engineers may fine-tune damping mechanisms or adjust system stiffness.

Furthermore, natural frequencies serve as a connection in the analysis of a system's dynamic behavior and response to external stimuli. Through scrutinizing natural frequencies and mode shapes, engineers pinpoint potential issues and optimize system performance.

2. **Damping ratio formula:** For a single-degree-of-freedom system:

$$\zeta = \frac{c}{2\sqrt{km}} \quad (3.14)$$

The damping ratio stands as a cornerstone in the realm of mechanical vibrations, playing a pivotal role in shaping the stability, responsiveness, and overall behavior of vibrating systems. Its significance emanates from its multifaceted impact on various aspects of system dynamics and performance [29].

At its core, the damping ratio serves as a critical determinant of system stability. By regulating the rate at which vibrations decay over time, it governs the system's ability to return to equilibrium following disturbances. A proper damping ratio ensures stable and controlled oscillations, preventing erratic behavior and potential instability.

3. **Forced vibration response formula:** For a single-degree-of-freedom system subjected to harmonic excitation:

$$x(t) = X \sin(\omega t + \phi) \quad (3.15)$$

Forced vibration response refers to the oscillation or vibration that occurs in a system when it is subjected to an external force, often a repeating type of force. This can occur in various mechanical systems, such as engines or structures, and is an important aspect to consider in the analysis and design of these systems [29].

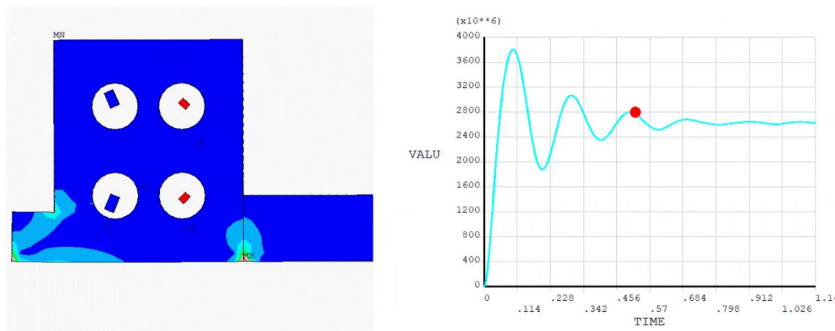


Figure 3.11: Simulation of the start-up of a scrap conveyor operating under forced vibration. Source: ISQ¹

3.3.4 Computational aspects including the use of MATLAB-based examples

MATLAB is a widely used software tool in engineering and scientific fields, known for its powerful computational capabilities, data analysis, and visualization features. It provides a user-friendly environment for solving complex mathematical problems and analyzing dynamic systems. In the context of mechanical vibrations, MATLAB offers numerous advantages. It allows users to efficiently solve vibration problems, analyze system responses, and visualize the results. MATLAB's matrix notation is particularly useful in vibration analysis, as it treats scalars, vectors, and matrices as complex-valued matrices. This enables users to handle various types of data and perform computations effectively [28, 29].

¹ Available at: <https://isq.pt/br/analise-de-vibracao-forcada/>. Accessed on April 4, 2024.

To provide a more detailed explanation, let's take a look at one of the MATLAB-based example plotting a Free-Vibration response [28], (Example 5.17):

The main objective of this example is to analyze and visualize the free-vibration response of two distinct systems over a specified time interval. By calculating and plotting the displacement of each system over time, the program allows users to observe and understand the dynamic behavior of the systems undergoing free vibrations.

The general equation used in the program can be expressed as:

$$x_i(t) = A_i \cdot \cos(\omega_i t) + B_i \cdot \cos(\Omega_i t) \quad (3.16)$$

In the program, these equations are used to calculate the displacements x_1 and x_2 of the two systems at each time step $t(i)$. The time vector t is created with 501 equally spaced points ranging from 0 to 20, and then each system's response is calculated using the general equation. Finally, the responses are plotted against time in separate subplots for visual analysis (Example in Fig. 3.12.

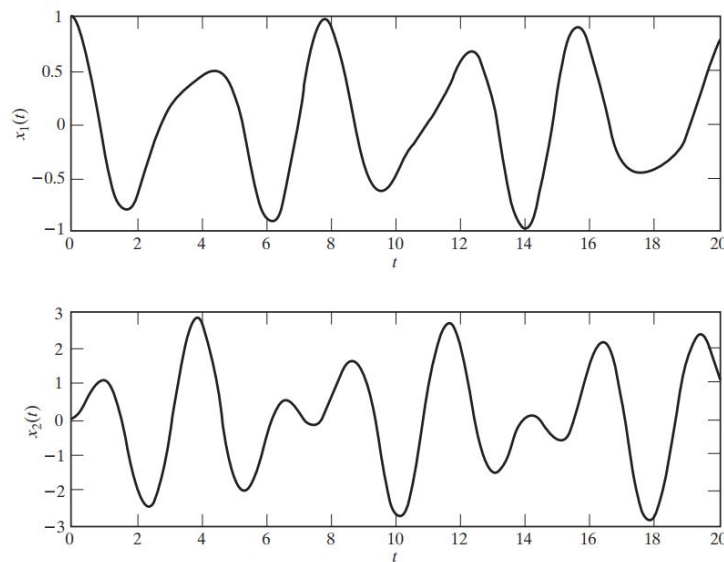


Figure 3.12: plotting a Free-Vibration response [28]

3.4 Dynamic Response of Electric Motors: Theory and Applications

The dynamic response of electric motors encompasses a multifaceted realm of study, delving into the intricate interplay between the motor's inherent characteristics and its response to varying operating conditions. This exploration involves analyzing transient responses and dynamic behaviors, offering insights into how electric motors adapt and perform in real-world scenarios. Understanding these dynamics is crucial for optimizing motor performance, enhancing efficiency,

and ensuring reliability across a spectrum of applications. This study aims to unravel the complexities of electric motor dynamics, shedding light on their theoretical underpinnings and practical implications in diverse fields of application.

3.4.1 Overview of Rotordynamics and Finite Element Analysis

Rotordynamics encompasses the study of dynamic behavior exhibited by rotating machinery, particularly rotors. It delves into the intricate analysis of vibrations, stability, and response exhibited by rotors under diverse operational scenarios. Rotordynamics provides comprehensive insights into various facets of this field, covering rotor elements, monorotor and coaxial multirotor models, as well as the integration of finite elements with stiffness matrices [33].

Rotor elements refer to the components or parts of a rotor system that play a significant role in its dynamic behavior. These elements include the rotor disk, shaft, bearings, and mass unbalance. Understanding the characteristics and behavior of these elements is crucial for analyzing and predicting the dynamic response of the rotor.

The equation that represents the rotation of the rotor in free motion is:

$$\begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix} \begin{bmatrix} \ddot{q}_1 \\ \ddot{q}_2 \end{bmatrix} + \Omega \begin{bmatrix} 0 & -a \\ a & 0 \end{bmatrix} \begin{bmatrix} \dot{q}_1 \\ \dot{q}_2 \end{bmatrix} + \begin{bmatrix} k & 0 \\ 0 & k \end{bmatrix} \begin{bmatrix} q_1 \\ q_2 \end{bmatrix} = 0 \quad (3.17)$$

That equation is obtained from:

$$\begin{aligned} m\ddot{q}_1 - a\Omega\dot{q}_2 + kq_1 &= 0 \\ m\ddot{q}_2 + a\Omega\dot{q}_1 + kq_2 &= 0 \end{aligned} \quad (3.18)$$

The application of 3.21, with the kinetic energy given by addition of 3.19, the strain energy given by 3.20, and no bearing influence, gives the equations 3.18 [33].

The kinetic energy equation describes the energy associated with the motion of the rotor. It is calculated based on the mass and velocity of the rotor elements.

$$\begin{aligned} T &= \frac{1}{2}m(\dot{q}_1^2 + \dot{q}_2^2) - \Omega a \dot{q}_1 q_2 \\ T_u &= m_u \Omega d f(l_1) (\dot{q}_1 \cos \Omega t - \dot{q}_2 \sin \Omega t) \end{aligned} \quad (3.19)$$

The strain energy equation represents the energy stored in the rotor due to deformation or strain. It is calculated based on the stiffness and displacement of the rotor elements.

$$U_s = \frac{1}{2}k(q_1^2 + q_2^2) \quad (3.20)$$

Lagrange's equations of motion are a powerful tool in describing the dynamics of mechanical systems. They are derived from the principle of least action and provide a concise formulation for determining the equations of motion for a system. The equations are given by:

$$\begin{aligned}\frac{d}{dt} \left(\frac{\partial T}{\partial \dot{q}_1} \right) - \frac{\partial T}{\partial q_1} + \frac{\partial U}{\partial q_1} &= F q_1 \\ \frac{d}{dt} \left(\frac{\partial T}{\partial \dot{q}_2} \right) - \frac{\partial T}{\partial q_2} + \frac{\partial U}{\partial q_2} &= F q_2\end{aligned}\quad (3.21)$$

The terms $\frac{\partial T}{\partial \dot{q}_i}$ represent the generalized momentum, while $\frac{\partial T}{\partial q_i}$ and $\frac{\partial U}{\partial q_i}$ represent the generalized forces.

These equations express the fundamental principle that the difference between the time derivative of the generalized momentum and the generalized forces for each coordinate must be equal to the external forces applied to that coordinate, multiplied by the corresponding generalized coordinate.

3.4.2 Finite Elements used for Induction motors

The use of finite elements for modeling rotors provides an overview of the main equations and methods used in the analysis. Finite element analysis is a numerical method used to analyze the behavior of complex structures by dividing them into smaller, simpler elements. In the context of rotordynamics, FEA is used to simulate and predict the dynamic response of rotor systems to various operating conditions and loads [33].

For the disk, the shaft, the bearings and the mass unbalanced are used several equations in order to determinate various parameters about the behaviour of the rotor [33].

For the disk, to understand the nodal displacement, the Lagrange's equations are used where the first matrix is the classical mass matrix and the second one is the gyroscopic (Coriolis) matrix.

$$\frac{d}{dt} \left(\frac{\partial T}{\partial \dot{\delta}} \right) - \frac{\partial T}{\partial \delta} = \begin{bmatrix} M_D & 0 & 0 & 0 \\ 0 & M_D & 0 & 0 \\ 0 & 0 & I_{Dx} & 0 \\ 0 & 0 & 0 & I_{Dx} \end{bmatrix} \begin{bmatrix} \ddot{u} \\ \ddot{w} \\ \ddot{\theta} \\ \ddot{\psi} \end{bmatrix} + \Omega \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & -I_{Dy} \\ 0 & 0 & I_{Dy} & 0 \end{bmatrix} \begin{bmatrix} \dot{u} \\ \dot{w} \\ \dot{\theta} \\ \dot{\psi} \end{bmatrix} \quad (3.22)$$

Looking for the shaft, there are several equations that describe its behavior. The shaft stiffness matrix is given by:

$$K = \frac{\rho S L}{420} \begin{bmatrix} 156 & 0 & 0 & -22L & 54 & 0 & 0 & 13L \\ 0 & 156 & 22L & 0 & 0 & 54 & -13L & 0 \\ 0 & 22L & 4L^2 & 0 & 0 & 13L & -3L^2 & 0 \\ -22L & 0 & 0 & 4L^2 & -13L & 0 & 0 & -3L^2 \\ 54 & 0 & 0 & -13L & 156 & 0 & 0 & 22L \\ 0 & 54 & 13L & 0 & 0 & 156 & -22L & 0 \\ 0 & -13L & -3L^2 & 0 & 0 & -22L & 4L^2 & 0 \\ 13L & 0 & 0 & -3L^2 & 22L & 0 & 0 & 4L^2 \end{bmatrix} \quad (3.23)$$

The Force applied on the bearings is related with the displacement with the matrix:

$$\begin{bmatrix} F_u \\ F_\theta \\ F_w \\ F_{u'} \end{bmatrix} = - \begin{bmatrix} k_{xx} & 0 & k_{xz} & 0 \\ 0 & 0 & 0 & 0 \\ k_{zx} & 0 & k_{zz} & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} u \\ \theta \\ w \\ \psi \end{bmatrix} - \begin{bmatrix} c_{xx} & 0 & c_{xz} & 0 \\ 0 & 0 & 0 & 0 \\ c_{zx} & 0 & c_{zz} & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} \dot{u} \\ \dot{\theta} \\ \dot{w} \\ \dot{\psi} \end{bmatrix} \quad (3.24)$$

The first matrix is a stiffness matrix and the second a viscous damping matrix. These matrices are generally asymmetric (i.e. $k_{xz} \neq k_{zx}$ and $c_{xz} \neq c_{zx}$) and the terms of the matrices can vary significantly as a function of the speed of rotation. equations gives.

According to mass unbalance, the Lagrange's equations gives:

$$\frac{d}{dt} \frac{\partial T}{\partial \dot{\delta}} - \frac{\partial T}{\partial \delta} = -m_u d \Omega^2 \begin{bmatrix} \sin \Omega t \\ \cos \Omega t \end{bmatrix} \quad (3.25)$$

with $\delta = [u, w]^T$

The expression 3.25 corresponds to a mass unbalance situated on the z axis at $t = 0$. When industrial rotors are studied, the influence of several mass unbalances acting simultaneously has to be considered.

Finite element analysis on induction motors involves discretizing the motor structure into smaller elements to analyze its dynamic response to various excitations. By representing the complex geometry and material properties of the motor components through finite elements, researchers can simulate and predict the natural frequencies, mode shapes, and vibration characteristics of the motor under different operating conditions [34].

In the context of the induction motor study, finite element modeling allows for a detailed analysis of the effects of different substructures (such as the stator, casing, support, and end-shields) on the overall vibration behavior. By comparing the calculated natural frequencies and mode shapes obtained from the finite element model with experimental modal testing results, researchers can validate the accuracy of the model and gain insights into how each component influences the motor's vibration response.

FEA of induction motor vibrations offers insights into natural frequencies, mode shapes, frequency response, damping, sensitivity analysis, and optimization recommendations, aiding in issue identification and performance enhancement.

Chapter 4

Evaluation of Design Alternatives

4.1 Identification of Available Design Alternatives

The Design alternative delves on the improvement of the process explained in 2.2.4 with the production of a tool that measures the exact axial misalignment of the rotor.

In non-flameproof motors, alignment is typically achieved through grooves and a needle for manual measurement, Fig. 4.1. Motors equipped with sleeve bearings feature three marks on the shaft end, with the central mark indicating the magnetic center and the two outer marks indicating the allowable limits for axial movement of the rotor. It fully move the shaft forward to ensure correct measurement of axial clearance (3 to 4 mm).

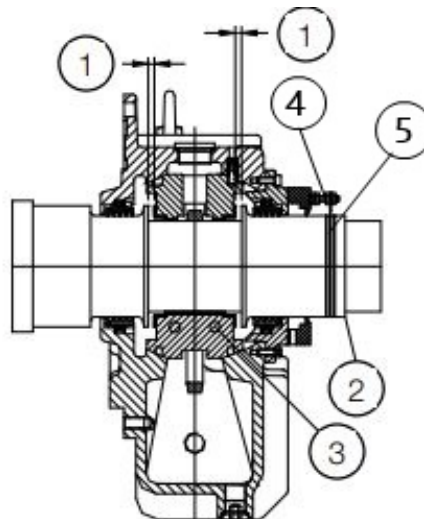


Figure 4.1: Clearance of the sliding bearing. [1] Axial Clearance. [2] Shaft. [3] Bushing. [4] Needle. [5] Grooves for measurement [35]

Carefully aligning the shaft ends introducing shims between the endshield and the bearing to adjust the axial position, illustrated in Fig. 4.2, and, whenever possible, use a flexible coupling, leaving a minimum axial clearance of 3 to 4 mm between the couplings [35].



Figure 4.2: Shims on non-flameproof motors for axial alignment

There are several challenges within manual alignment as imprecise measurement due to human error and subjectivity, but the most critical one is the time-consuming process leading to extended downtime. The process of adjust the axial displacement can take a lot of assemblies from the sleeve bearings to introduce the correct amount of plates until achieve the precise axial position of the rotor.

Even with that method, in flameproof motor it can't be done because introducing that plates on the sleeve bearings, it would introduce safety concerns and invalidate the flameproof criteria of the motors. In flameproof motors what can be done to align axially the rotor, is manually, with several attempts, adjust the position until it stabilizes on an accepted location. This method is time-consuming due to the repeated assembly and disassembly of the endshield, along with the need to reposition the rotor or, in critical situations, even the stator.

Also, proper axial alignment of the rotor is critical for the lifespan of sleeve bearings. Sleeve bearings rely on a thin film of oil for lubrication, separating the shaft from the bearing material, causing no wear on the sleeve bearing. Any misalignment creates unwanted side loads due to electromagnetic forces. These forces can push the shaft against the edge of the bearing, causing face friction. This friction not only leads to premature wear of the bearing but also reduces motor performance. In that cases, when axial misalignment is too big and the gap between the shaft and the sleeve bearing is not enough, it is mandatory to move the position of the stator, which is a really complex, meticulous, and sometimes dangerous method.

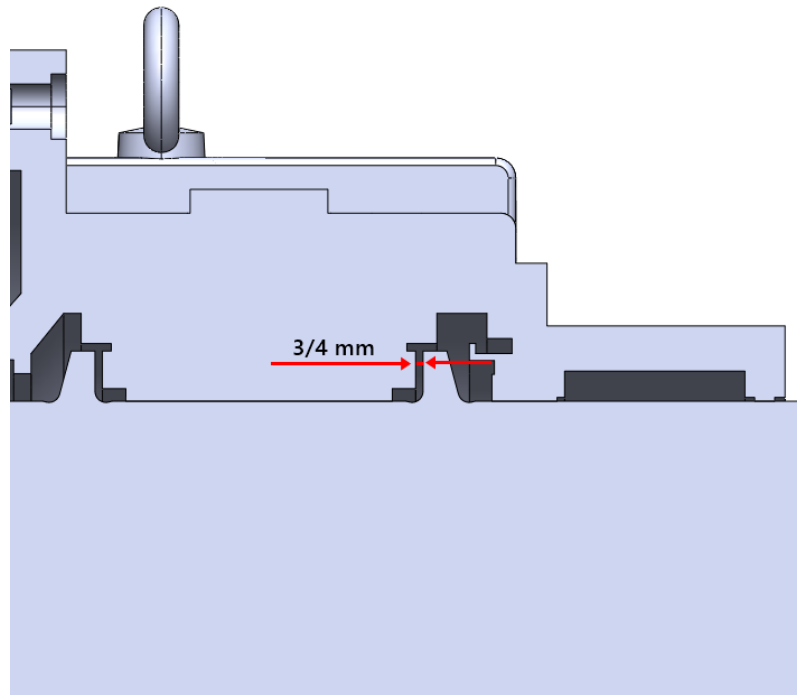


Figure 4.3: Detail of the Gap between the shaft and the Sleeve bearing (3mm to 4mm)

4.1.1 Proposed Alternative

To address this challenge, it's crucial to recognize the necessity of delivering fully machined endshields to clients, meticulously assembled with flameproof standards. In this context, precise machining of the endshield's rib thickness is paramount, ensuring optimal contact between the endshield and the sleeve bearing.

By precisely machining the thickness of the endshield's contact surface with the frame rib, we can regulate the axial displacement, thus ensuring the desired rotor positioning, Fig. 4.4. However, a pertinent question arises: how do we ascertain the precise axial position?

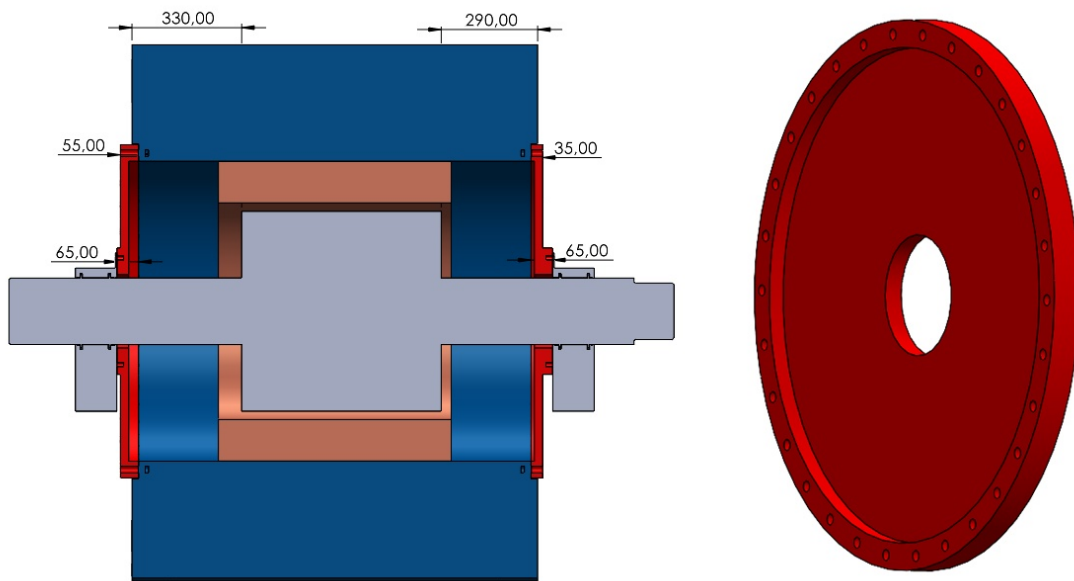


Figure 4.4: Simplification of the machined Endshield

To address this issue, an auxiliary endshield was developed, dividing the original one in two parts with one fix part and other moving part, in a way to replace the original one and determine the correct axial displacement during the functioning of the motor (**at WEG's factory, during the production process**) and, after that, place the original machined endshield. This tool was divided in four components:

- **External endshield:** This component mirrors the blue part depicted in Figure 2.8, securely fastened to the motor frame using multiple screws.
- **Flange:** Functioning as a movable part, it interfaces with the sleeve bearing and resides within the interior of the external endshield. Consequently, the shaft rests upon the two sleeve bearings, providing support for the rotor.
- **Slide Bushing:** Ensuring smooth sliding between the external endshield and the flange, this component prevents any interference friction on the surfaces, facilitating unrestricted movement between the two parts.
- **Shims:** These plates serve to measure axial displacement accurately.

With this system in place, the flange moves within the slide bushing, and during motor operation, the force of the magnetic field positions the rotor precisely axially. Shims are inserted between the flange and the external endshield to measure axial displacement and maintain motor rotation without the need to pause its movement. Screws are then employed between the flange and the endshield to prevent rotation of the flange and the sleeve bearing and to position the shims properly.

Once the precise axial displacement is determined, the real endshield is machined to match the measured thickness using the shims. This ensures alignment with the sleeve bearing face and the frame rib, guaranteeing the rotor's perfect axial position upon assembly with the motor frame.

Due to the tool's symmetrical design, the rotor's movement is mirrored. As the rotor slides outward on one side of the frame by a specific distance, an equal displacement occurs inwards on the other side. To limit this travel distance, both flanges incorporate stops that restrict movement to a maximum of 48mm. While this travel range may seem excessive, it can be particularly useful in extreme cases and allows for greater flexibility in the system. Additionally, this extended range ensures that the tool can accommodate a wide variety of scenarios, providing robustness and adaptability in different operational environments. Testing conducted with previous flameproof induction motors has demonstrated that this travel range is sufficient, thereby validating the design.

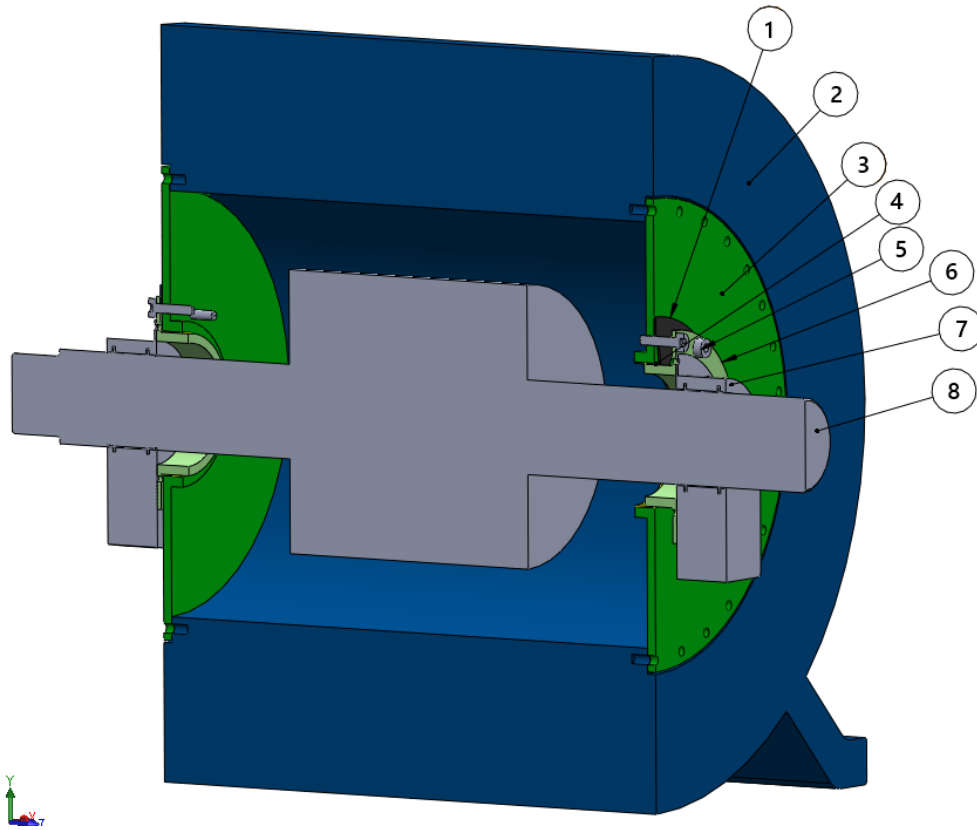


Figure 4.5: Simplified electric motor with the simplified alignment tool

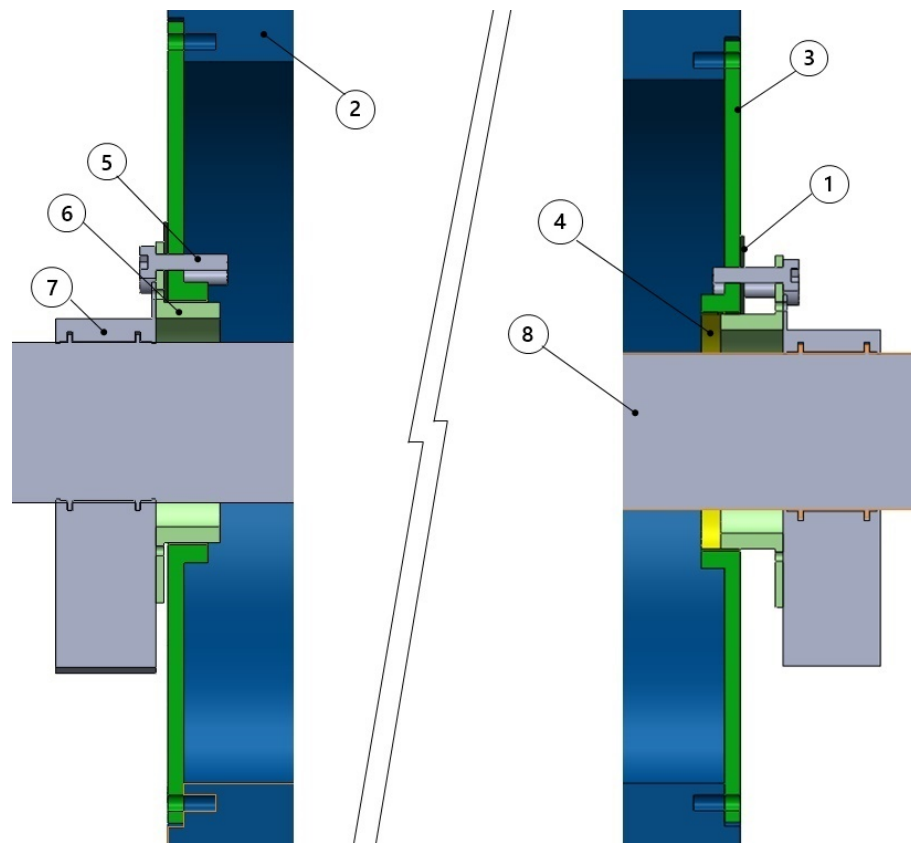


Figure 4.6: Lateral Cut view from a simplification of the alignment tool

1	Shim	2	Housing	3	External Endshield	4	Sliding Bush
5	Positioning screw	6	Flange	7	Slide Bearing	8	Shaft and Rotor

Table 4.1: Components identification of the alignment tool of Fig. 4.5 and Fig. 4.6

Operationally, it is crucial to note that for the initial start-up of the motor, a symmetrical number of shims must be inserted on each side to completely fill the space and ensure the stability of the motor. During the axial displacement process, shims are progressively removed from one side and added to the other. This adjustment continues until the magnetic field no longer pulls the rotor axially, and further shim repositioning is unnecessary. This careful balancing ensures that the motor maintains optimal performance and alignment throughout its operation.

For simulation purposes, it is essential to consider the most extreme scenarios where the maximum displacement is achieved.

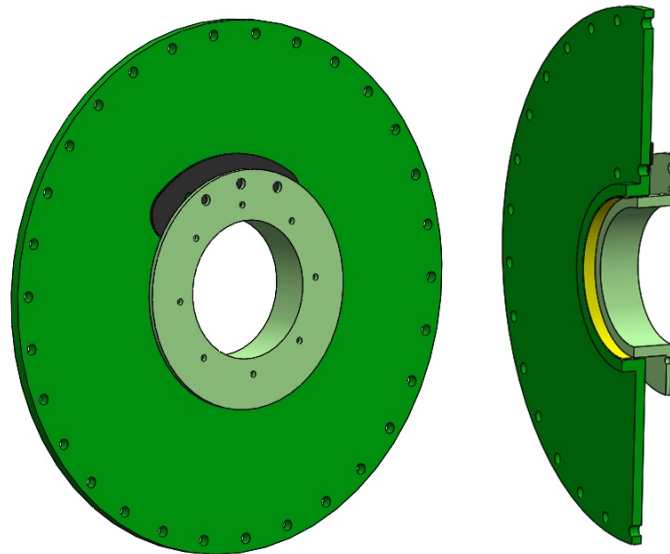


Figure 4.7: 3D view from a simplification of the alignment tool

An alternative solution could involve producing the identical product but omitting the slide bush, allowing the external endshield to slide directly on the flange. This option will be evaluated at a later stage [4.3](#).

4.2 Evaluation Criteria

The evaluation of the developed tool encompasses various aspects crucial for its functionality and effectiveness in minimizing shaft misalignment in flameproof motors. These criteria were assessed through computational simulations using Ansys Mechanical. The following evaluation criteria were considered:

- **Convergence Analysis:** A convergence analysis was conducted to determine the optimal number of elements for the mesh. This analysis aimed to ensure both accuracy and computational efficiency. Various mesh densities were tested, and the results were analyzed to find the point where increasing the number of elements no longer significantly improved the accuracy of the results.
- **Static Loads:** Within the convergence analysis, static load analysis was performed to ascertain whether the weight of the components would be adequately supported by the tool. This evaluation ensures the structural integrity and stability of the tool under operational conditions. Static load simulations were conducted to analyze the stress and deformation under the weight of the motor components.

- **Machining of the Endshield:** The influence of temperature differences during machining and the effects of clamping on the dimensions were evaluated. This analysis focused on understanding how machining processes may affect the performance and precision of the tool. Simulations were run to model the thermal effects and mechanical stresses induced during machining. The impact of these factors on the dimensional accuracy of the endshield was then assessed.
- **Modal/Axial Displacement Analysis:** Modal analysis and axial displacement were conducted to compare the vibration values of the motor with and without the tool, ensuring the tool does not introduce significant vibration issues. Modal analysis was performed to identify the operational frequencies of the system. This analysis aimed to ensure that the operating vibration levels of the motor with the tool in place are within acceptable limits relative to the rotor's natural frequencies.
- **Unbalance Analysis:** An analysis was conducted to evaluate if the difference in thickness of the original endshields would result in significant unbalance and affect motor vibrations. Simulations assessed the impact of the varying endshield thickness on the motor's balance and vibration characteristics.
- **Flange Bending:** The face contact between the flange and the endshield may experience bending due to the force exerted by the magnetic field. Therefore, it is crucial to ensure that the thickness of this face is sufficient to withstand the bending stress and maintain structural integrity.
- **Lifting Analysis:** A lifting analysis could be considered, specifically focusing on the tool's response to lifting forces. Simulations of lifting examine the potential deformations and stresses induced by lifting the tool. Within the company, lifting analysis is typically performed on the motor housings, not on the endshields, primarily due to the weight of the components. In this context, the external endshield is lighter than the original endshield. Given that the deformation caused by lifting the original endshield has already been analyzed by the company, it is reasonable to conclude that the lighter external endshield would not experience greater deformation than its predecessor. Consequently, performing an additional lifting analysis on the new external endshield is deemed unnecessary.
- **Fatigue and Rupture Analysis:** Given that the tool is intended for short cycle use within the factory, fatigue and rupture analysis were deemed unnecessary. The tool's operation is limited to the factory setting and short durations, where fatigue and rupture are not critical factors.
- **Pressure Resistance:** As the motor will not be subjected to pressure, an evaluation of pressure resistance was not relevant. The operational environment does not expose the motor to pressure conditions, eliminating the need for pressure resistance simulations.

In conclusion, utilizing this tool is expected to significantly improve production efficiency by streamlining the alignment process. With the tool, it is only necessary to assemble it once, disassemble, machine the original endshields to the determined thickness, and then assemble the machined endshield onto the motor—thus reducing assembly time and enhancing overall production efficiency. This is particularly beneficial for special motors that are not produced in series, where each endshield would need to be machined individually regardless.

4.3 Comparative Analysis of Alternatives

The evaluation of using two (one on each side of the motor) SKF POM composite straight bushings in the project reveals several significant advantages that align well with the project's requirements. These bushings, with a bore diameter of 300 mm, outside diameter of 305 mm, and width of 100 mm, are designed for a variety of movements including oscillating, rotating, and linear motions, and can accommodate substantial radial loads.

Advantages of Using SKF POM Composite Bushings:

- **Load Capacity:** The bushings have a basic dynamic load rating of 3,550 kN and a basic static load rating of 7,350 kN, which far exceeds the maximum weight of the rotor (approximately 50 kN). This ensures that the bushings can handle the operational loads without any issues, providing a high safety margin [36].
- **Smooth and Frictionless Sliding:** The SKF POM composite bushings allow for smooth and low-friction sliding between the endshield and the flange. This facilitates easy and precise axial movement, ensuring that the magnetic force can position the rotor at its magnetic center without any resistance apart from the rotor's inertia. Without the plain bearing the of the rotor would cause a lot of friction between the endshield and the flange face.
- **Minimal Maintenance:** These bushings are optimized for minimal maintenance, even under tough operating conditions. They are designed to operate with marginal lubrication, thanks to the grease reservoirs on the sliding surface. Initial grease fill is sufficient to ensure long periods of satisfactory operation, which is particularly beneficial for the project's need for reliable and low-maintenance components.
- **Durability and Cost-Effectiveness:** SKF POM composite bushings are known for their durability and cost-effectiveness. Their compact design and the ability to handle heavy loads contribute to a long service life, reducing the frequency and cost of replacements.
- **High Performance:** The bushings exhibit very good frictional properties and can accommodate high sliding velocities. This is crucial for ensuring the motor operates efficiently and without excessive wear on the components.

The plain bearings will be press-fitted into the external endshield with a clearance fit to the flange (tolerances can be found in [A](#)) This configuration allows for easy introduction of grease through a hole in the external endshield, ensuring smooth sliding on the flange. The self-lubricating properties and grease reservoirs extend the service life and reduce the need for frequent maintenance.



Figure 4.8: SKF POM composite plain bearing [37]

Another possibility to consider is the use of a brass bushing. Brass is a soft, low-cost material with excellent wear resistance and good machinability, making it an attractive option for applications requiring low friction. However, while brass bushings offer economic advantages and are easy to replace, they may not withstand the substantial loads involved in this project as effectively as SKF POM composite bushings. Brass bushings generally have lower load capacity and might require more frequent maintenance or replacement, which could compromise the reliability and efficiency of the motor.

Given the critical requirements for handling high loads and ensuring minimal maintenance, the SKF POM composite bushings are the more suitable choice for this application. Their superior load capacity, durability, and performance characteristics align better with the project's needs, ensuring the motor operates smoothly and reliably. However, the use of brass bushings could be further analyzed in future studies to explore potential cost savings for applications with lower load demands.

Chapter 5

Numerical Analysis of the Developed Tool

5.1 Formulation of the Simulation Procedure

The simulation procedure was developed using Ansys Mechanical, leveraging various simulation types tailored to specific aspects of the analysis. The primary objective of the simulations was to evaluate the mechanical robustness and functional efficacy of the developed tool, ensuring it does not cause interference, resonance during rotational vibrations, or any other mechanical issues.

5.1.1 Simulation Types and Their Applications

- **Static Structural Analysis:** Utilized for convergence analysis and static force evaluation. This analysis helps in understanding the stress distribution and ensuring the structural integrity of the tool under static loads.
- **Steady-State Thermal and Static Structural Analysis:** Applied to the machining process to evaluate the influence of temperature variations during machining and the potential impact of clamping forces on the dimensions.
- **Modal Analysis:** Conducted to assess the vibration characteristics and evaluate the impact of potential unbalance due to varying endshield thicknesses. This analysis helps in identifying operational frequencies and ensuring the tool does not induce resonance during operation.

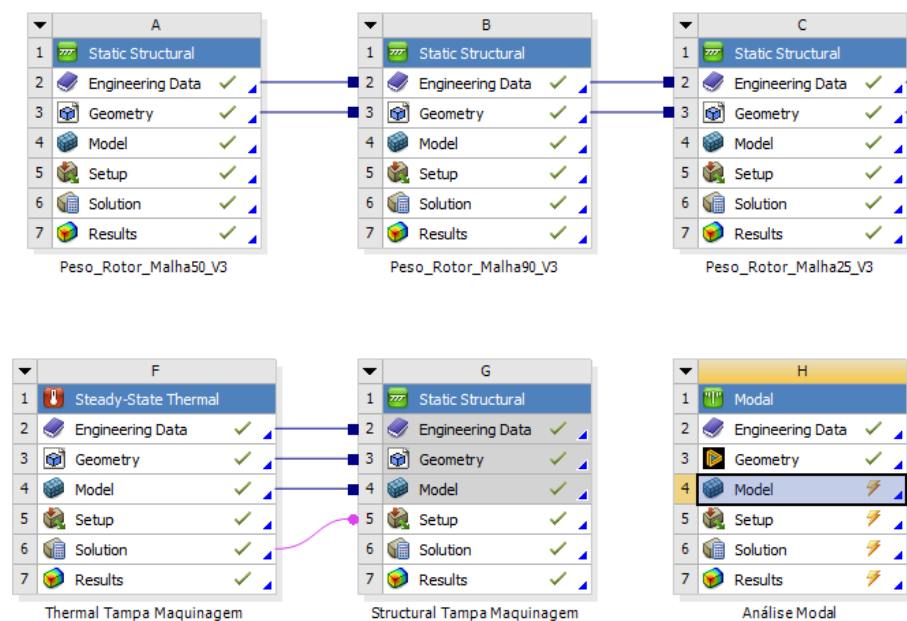


Figure 5.1: Ansys Workbench

5.1.2 Methodology

The methodology involved analyzing the tool both separately and in conjunction with other motor components. This comprehensive approach ensured a thorough evaluation of the tool's performance in real-world conditions.

1. Simplification of the Geometry of the Existing Motor Model:

- Unnecessary holes and slots were removed.
- Detailed features of the bearings were simplified.
- The shaft was simplified, and the stator was represented as a simple cylinder.
- Fans were removed and it was considered its mass distributed accordingly.
- A critical focus was on the casing, where unnecessary features were removed while retaining essential elements that influence structural integrity, such as internal plates that support cooling tubes.

2. Mesh refinement:

- Mesh refinement zones were strategically chosen based on contact areas, Fig. 5.2. In the original motor, the focus was on the contact between the bearing and the shaft. In the motor with the tool endshield, additional attention was given to the contacts between the bearing and the flange, and between the flange and the external endshield.

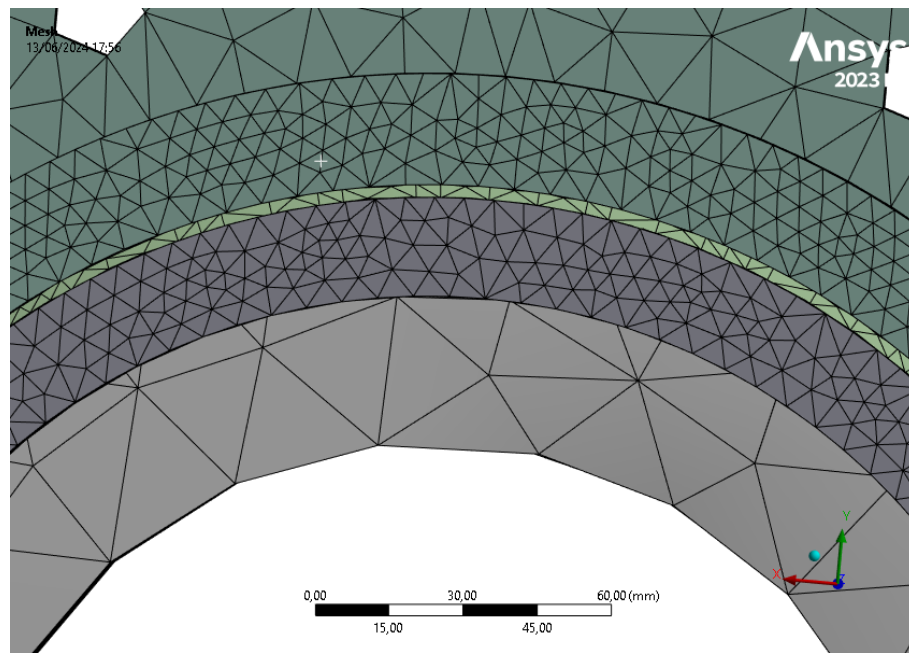


Figure 5.2: Detailed Mesh on Contact Points

Table 5.1: Mesh Data for Simulation Process

Element Size	25 mm
Nodes	3774343
Elements	1963453
Transition Ratio	0,272 mm
Min. Edge Length	$6,136 \cdot 10^{-2}$ mm

5.1.3 Simulation Objectives

The primary objectives of the simulation included:

- Verifying the mechanical development of the tool.
- Ensuring the tool does not cause mechanical interference or resonance during rotational vibrations.
- Verifying that the tool maintains structural integrity under static loads.
- Assessing the impact of machining processes on tool dimensions and overall performance.
- Evaluating the vibration characteristics and potential unbalance effects due to varying end-shield thicknesses.

By following this methodology, the simulation procedure was designed to provide a comprehensive evaluation of the tool, ensuring it meets all mechanical and operational requirements.

5.2 Selection of Parameters and Input Variables

The selection of parameters and input variables for the simulations depends heavily on the type of analysis being conducted, whether it's modal, static, thermal, or other specialized analyses. Each type requires specific considerations to accurately model the behavior of the motor and the tool under various operating conditions.

To begin with, defining the material properties for each component is crucial. Structural steel is chosen for the frame due to its strength and suitability for withstanding static loads, while SKF's POM composite is selected for the sliding bushing to ensure smooth operation and minimal friction under oscillating and linear movements.

Establishing contacts between components is a complex task that varies based on the type of analysis:

In static analysis, all connections can typically be defined as "bonded" to simplify the solution, focusing on the gravitational load direction caused by the motor's weight. This simplification ensures that the analysis accurately reflects the static equilibrium and stress distribution within the structure.

However, modal analysis requires a different approach where freedom of movement is essential for accurate simulation. Contacts such as the external endshield face to the casing flange groove and the stator outer face to the casing ribs can be considered "bonded" since they are static relative to the axis of the model. In contrast, the contact between the flange face and the bearing face should be defined as "no separation." Despite being fixed together by screws, they are not fixed relative to the axis of the model and must allow for relative movement during operation.

Another critical consideration is the contact between the internal face of the bush and the external face of the flange. Here, the flange slides freely on the grease film of the bush, necessitating a "frictionless" contact definition to accurately simulate the interaction between these components. Also the rotation of the shaft on the oil film of the bearing must be considered as "frictionless"

To simulate the limited axial movement (48mm) of movable elements such as the rotor, bearings, and flanges, a virtual spring with negligible resistance is applied to the flange face, Fig. 5.3, that's an Ansys technique for that specific characteristics on modal analysis. This constraint is vital for maintaining proper alignment and operational stability of the motor during dynamic conditions.

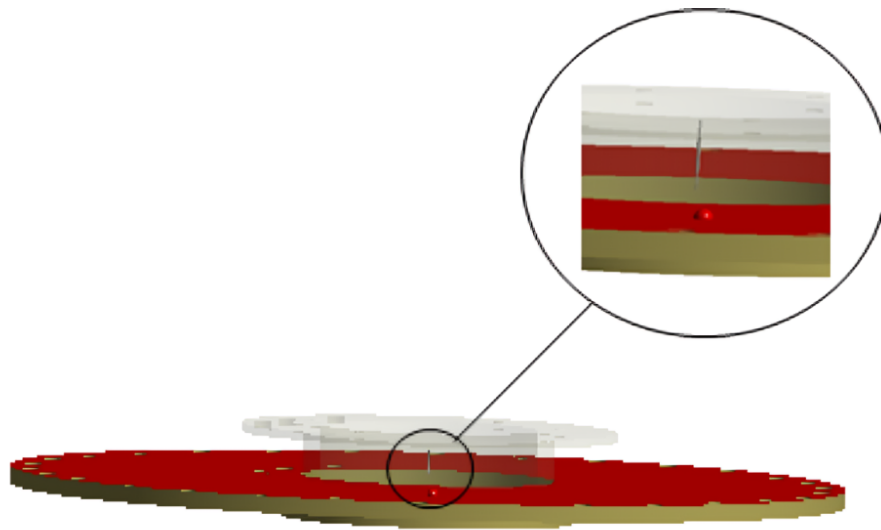


Figure 5.3: Spring consideration for axial displacement limitation

Additional boundary conditions include specifying the motor speed accurately. For instance, if the motor operates at a specific rpm (4-pole motor, 50 Hz), this rotational speed must be precisely applied in simulations to study dynamic effects and operational characteristics effectively.

Moreover, modeling the motor's mounting points on the ground with "fixed supports" ensures a stable base. To prevent deformation of the frame feet, "compression-only supports" should be applied at the base faces of the feet. This condition guarantees that the motor is firmly grounded without introducing damping effects that could alter the structural integrity under load.

In thermal simulations, ambient temperature serves as the external temperature boundary condition, while internal temperature distribution along the tool, Fig. 5.4, especially the central temperature during machining, provides insights into thermal gradients and their impact on component performance and longevity.

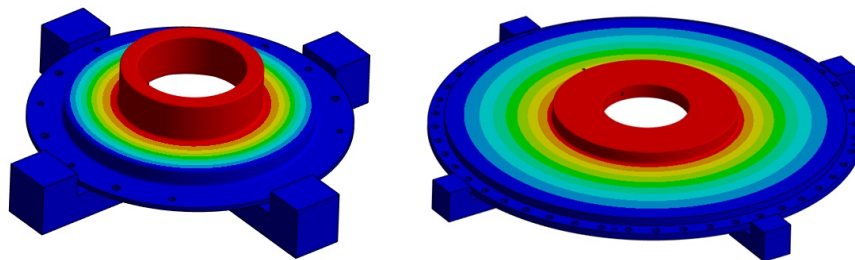


Figure 5.4: Thermal variation during endshield machining

In factory tests conducted on other motors, the axial displacement force exerted by the magnetic field on the rotor was observed to be significantly less than 1000N . Consequently, we will

simulate the bending of the flange face under an applied load of 1000N to determine its ability to withstand this force without compromising its structural integrity. This simulation will help ensure that the flange face thickness is adequate to handle the stress induced by the magnetic pull.

For the unbalance analysis, the maximum variation in the thickness of the endshield was considered, adhering to flameproof standards. Specifically, the analysis examined a scenario where there was an increase of 15mm on one side and a decrease of 15mm on the other, as illustrated in Figure 4.4. That increase and decrease is better explained on 6.3.

The unbalance created by these thickness variations can significantly impact the rotor's dynamic behavior, leading to potential vibration issues, using the Ansys modal analysis. Therefore, the analysis aims to evaluate the impact of such thickness discrepancies on the motor's operational stability. By assessing the unbalance, we can ensure that the motor maintains acceptable vibration levels, comparing with the one with both endshields with same thickness, which is crucial for both performance and safety in flameproof environments.

5.3 Application of the Simulation and Validation of the Procedure

The developed simulation procedure was applied to evaluate the design alternative. The criteria for evaluation its performance were based on optimizing mechanical performance, minimizing vibration, comparison with the original motor, and ensuring compliance with safety standards. The design alternative was subjected to detailed simulations explained in 5.1 , to assess its feasibility and performance under operational conditions.

5.3.1 Convergence Analysis and Static Loads

In this analysis, the objective is to understand the deformation of the rotor in a static structural analysis where the only acting force is its own weight. Simultaneously, it aims to determine whether the rotor's weight influences the deformation of the endshield at its maximum axial displacement position. As this is a relatively simple analysis that does not involve many boundary conditions, it serves as an excellent opportunity to determine the optimal number of elements to consider for future simulations. This is achieved through a convergence analysis by varying the number of mesh elements and observing when the results stabilize.

To demonstrate the deformation without disclosing proprietary company data, the displacement is expressed as a percentage in Figure 5.5. The air gap is the empty space between the stator and the rotor, representing the "usable" space within which the rotor can flex. Clearly, it is undesirable for the rotor to flex significantly. Thus, the graph depicts the rotor's flexion as a function of the air gap. It is noted that the deformation of the endshield is approximately one-eighth of the deformation experienced by the rotor.

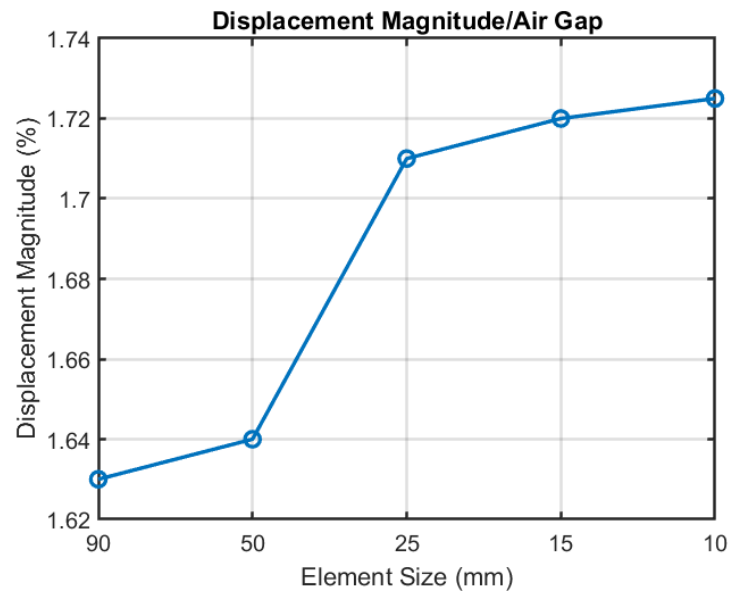


Figure 5.5: Convergence and Static Analysis

While the deformation analysis shows convergence with an element size of 25mm (Fig: 5.5), the stress analysis indicates that convergence is achieved with an element size of 50mm (Fig: 5.6).

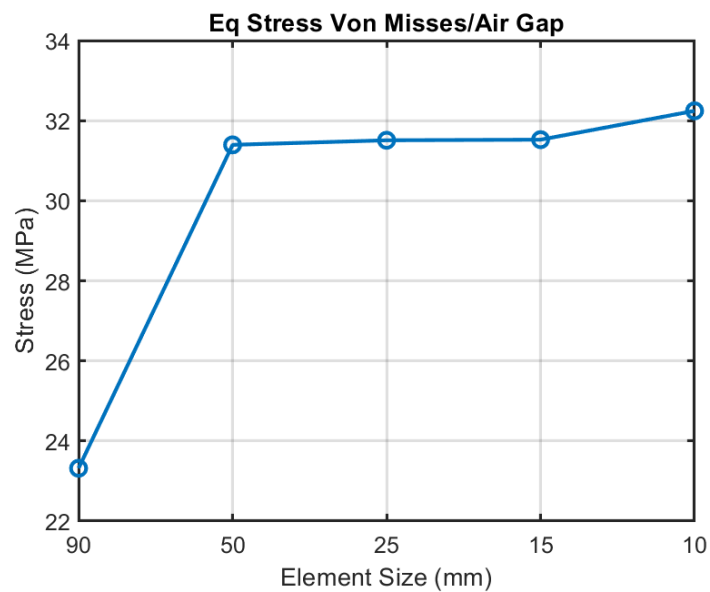


Figure 5.6: Convergence and Static Analysis - Stress

Regarding convergence, the graph in Figure 5.5 shows that with an element size of 25mm, the displacement function converges to a value of less than 1.74%, with negligible variation for smaller element sizes. This indicates that using a 25mm mesh element size ensures accuracy and stability in the simulation results. Although a larger element size of 50mm also shows convergence

in stress analysis, it does not provide the same level of precision as the 25mm mesh. Therefore, to achieve more accurate and reliable results in future simulations for the same motor model, a mesh with an element size of 25mm will be considered.

5.3.2 Machining of the Endshield

For the machining analysis, we will consider an ambient temperature of 20°C and the temperature within the endshield as measured in the laboratory during machining, as the force applied by the clamping machine. Thus, for the deformation analysis of the endshield during machining, we can take into account the temperature variation along the components, including both the endshield and the flange. The same external and internal temperatures were considered for both components.

This temperature variation in the Ansys analysis will be linked to the deformation of the components to achieve deformation results that closely approximate reality.

To obtain accurate results, it is important to observe deformations in different areas of the endshield, to understand the directional variation of deformation along edges, contact faces with other elements, etc (Fig. 5.7). This analysis is crucial because there are various fits and clearances that must be considered during machining. Predicting possible excessive or insufficient material removal during machining is essential for achieving precise and careful machining.

Moreover, by integrating thermal effects with structural deformation, we can better simulate the actual operating conditions and anticipate issues that might arise due to thermal expansion or contraction. This comprehensive approach helps ensure the reliability and accuracy of the tool's performance during actual use. Additionally, considering the mechanical properties and behavior of materials under thermal stress further refines our simulation, providing insights into potential weaknesses or areas for improvement in the tool design.

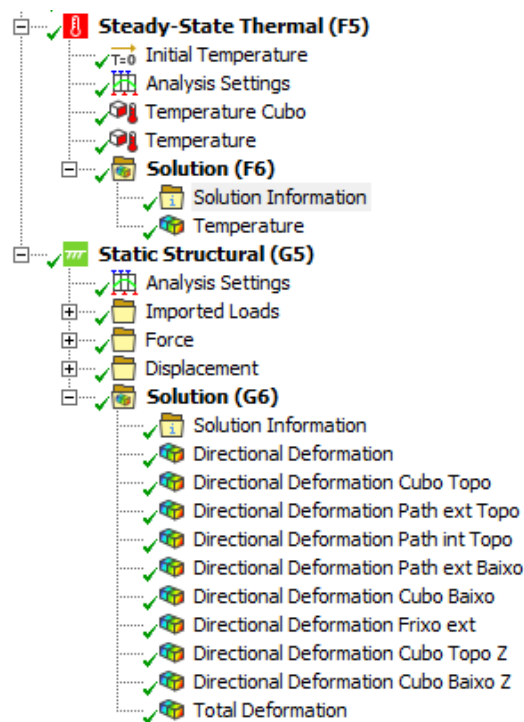


Figure 5.7: Ansys solutions for deformation with thermal analysis

In Figure 5.8, we can observe an almost negligible deformation in the rib of the endshield hub, which will come into contact with the outer face of the bushing. Therefore, it is crucial to ensure the tolerance between these two elements, making sure that there is no excessive material removal from the face and edge to allow the bushing to fit according to the manufacturer's tolerance [A](#). According to the manufacturer, we must ensure a hole with H7 tolerance. From the reference book [\[38\]](#), this indicates that there should be a maximum clearance of $52 \mu m$ and a minimum of 0.

Ensuring these tolerances is critical for maintaining the integrity and performance of the assembly. Excessive or insufficient clearance could lead to improper fitting, increased wear, or even failure during operation. The precision in machining and the subsequent fit of the bushing within the endshield hub is essential for the overall reliability of the tool.

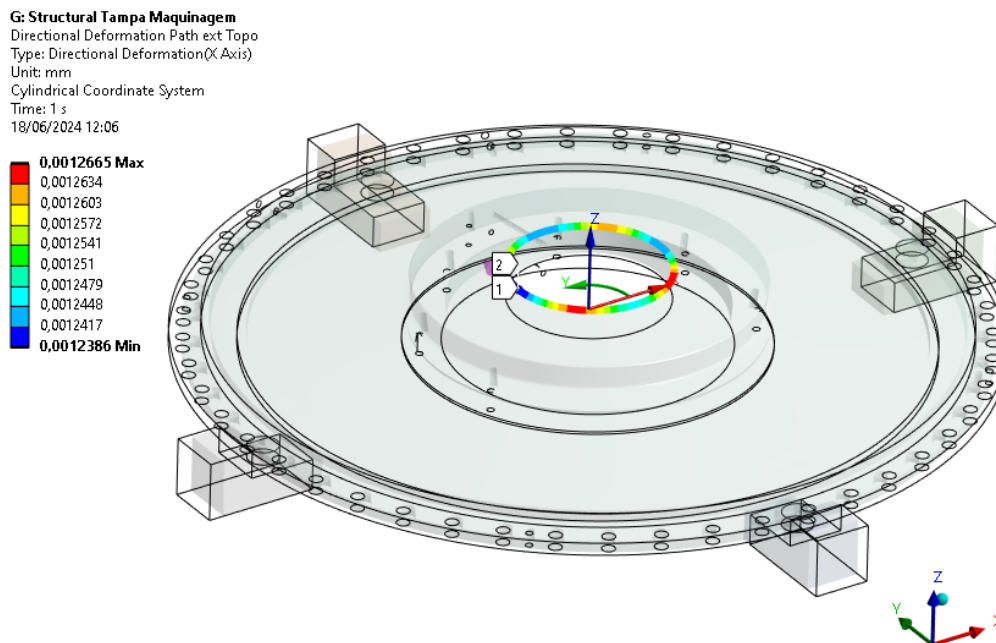


Figure 5.8: Rib deformation of the endshield through edge path

In Figure 5.9, based on the data extracted from ANSYS, we can observe in greater detail how the deformation of the endshield rib occurs during the clamping process. A sinusoidal variation is evident, with four peaks representing the positions of the clamps and where they perform the maximum force.

This detailed analysis highlights the localized deformation caused by the clamping forces, which is critical for understanding the mechanical behavior of the endshield during machining. The sinusoidal pattern indicates that the deformation is not uniformly distributed, but rather concentrated at the points of contact with the clamps. This non-uniform deformation can have significant implications for the precision of the machining process and the final fit of the components.

By strategically positioning the clamps or using additional support structures, we can achieve a more balanced force distribution, thereby reducing the deformation peaks. This will ensure that the endshield maintains its structural integrity and dimensional accuracy during machining.

Despite the endshield being symmetric in two planes, the peaks in the graph are not symmetrical. This asymmetry can be attributed to the presence of two holes for grease entry into the bush. These holes introduce small variations in the structural integrity of the endshield, causing uneven distribution of clamping forces during the machining process. As a result, the deformation peaks observed in the graph are influenced by these localized structural differences, leading to the observed asymmetry.

The results obtained are minimal and do not compromise the confidentiality of the company's data.

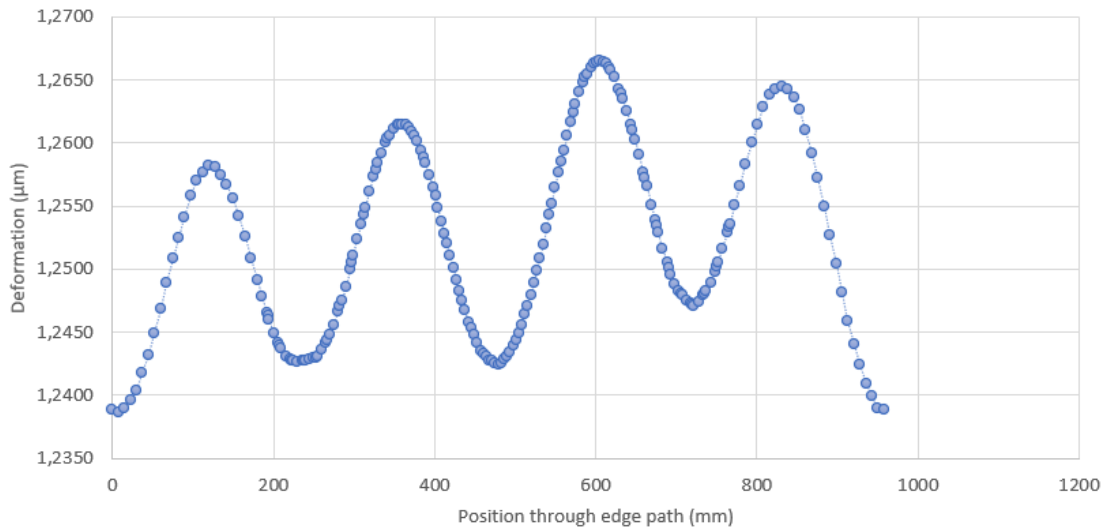


Figure 5.9: Plot of directional deformation of the endshield through edge path

5.3.3 Modal/Axial Displacement Analysis

Modal and vibration analysis is crucial for this type of project to ensure the mechanical integrity and performance of the motor and its components under operational conditions. Understanding the vibrational behavior helps in identifying potential resonances that could lead to excessive vibrations, which can cause noise, wear, and even failure of the components.

For the modal analysis, several factors had to be considered. Initially, the vibrational behavior of the original motor with the original endshield was simulated across five vibration modes. Subsequently, the same simulation was conducted with the tool endshield, incorporating considerations for axial displacement. For the modal simulation of the auxiliary endshield, it is essential to account for the axial freedom of the rotor and to restrict this displacement, as demonstrated in Figure 5.3. To achieve this, the "frictionless" option must be included instead of "bounded" for the relevant components, such as the contact between the external face of the flange and the internal face of the bushing, where the former will have axial displacement freedom.

In the case of the motor with the original endshield, the entire assembly is bounded, meaning the axial vibration mode will have a greater impact. In contrast, with the tool endshield, the rotor's freedom reduces its vibration, resulting in almost a 36% decrease in the first vibration mode, as shown in Figure 5.10.

In the second vibration mode, which tests the frequency of vibration due to rotation, a 19% decrease is also observed. Similar decreases in vibration frequency are noted in subsequent modes, although they are less significant.

It is important to note that the values in all subsequent graphs and data do not present the actual values obtained from the simulations to protect the company's data, it was used a scale factor in each data. However, the percentages accurately reflect the relationships between the data.

This modal analysis highlights the significant reduction in vibrational impact when using the tool endshield compared to the original endshield. It is important to note that the tool endshield is intended solely for use within the factory for axial alignment of the motor and will not be sent to the customer. Consequently, while the tool endshield demonstrates superior performance in reducing vibrations during alignment, it is not designed to replace the original endshield in operational environments. It is interesting to note that despite not being an endshield that will be shipped with the original motor to the customer, the fact that the vibration is lower than with the original endshield will not imply problems of excessive vibration. However, it is necessary to evaluate the possible presence of resonance, which will be addressed in Figure 5.12.

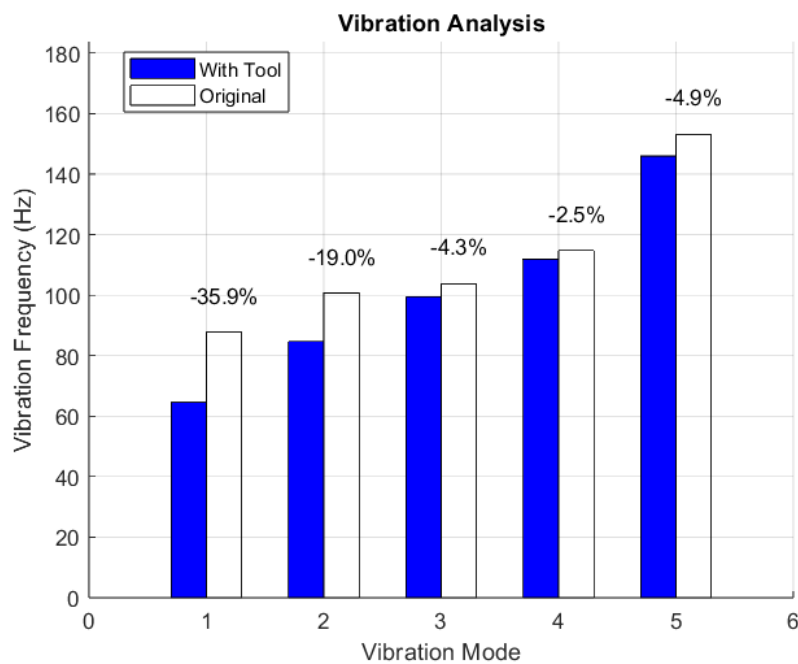


Figure 5.10: Vibration Frequency in Modal Analysis

In the graph shown in Figure 5.11, we can observe that even though the vibration frequency of the motor decreases when using the tool endshield, the deformation tends to increase. This outcome is expected, particularly in the first mode of vibration, due to the axial freedom. The software calculates the displacement based on the reference position, and with the axial freedom introduced by the tool endshield, an increase in axial deformation is anticipated. This result emphasizes the need to carefully consider the axial movement during the alignment process within the factory setting. Furthermore, the increase in deformation does not necessarily imply structural failure or operational inefficiency; instead, it highlights the importance of precise control and monitoring during the alignment phase.

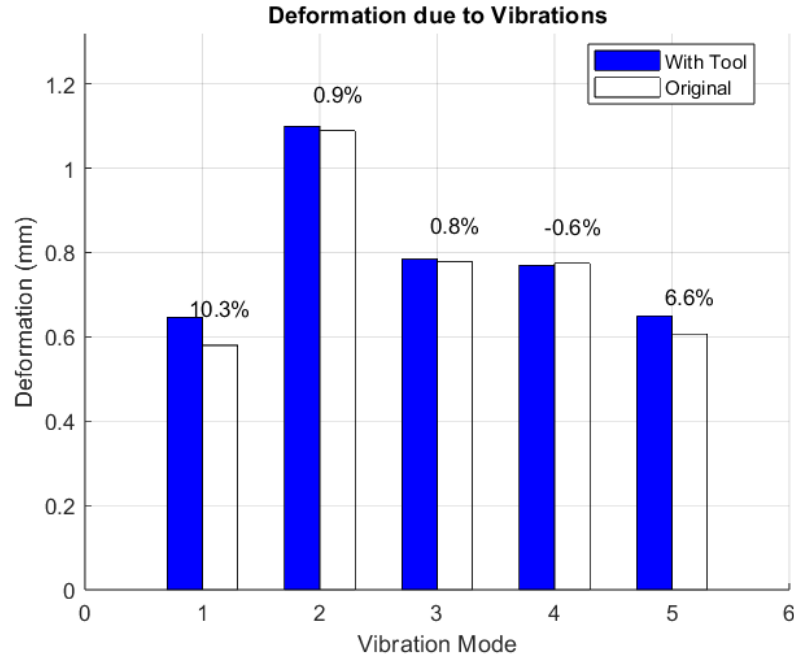


Figure 5.11: Deformation in Modal Analysis

The final aspect to consider in the modal analysis comparison between the original endshield and the tool endshield is the potential for resonance. For this purpose, we refer to the graph in Figure 5.12. Using the Mathcad tool, we calculated the natural frequency of the rotor, represented by the green line in the graph. The power network is represented by a constant line in the graph, and we can determine when a mode of vibration might be excited depending on the frequency value. If the vibration frequency is above the power network, it indicates that there is no possibility of excitation in that respective mode because the energy required to excite the mode exceeds the available power input, effectively damping the vibrations. Thus, with the original endshield, we can observe that no vibration modes are excited as the frequencies of all vibration modes are above the power network threshold.

For resonance verification, it is important to note that this analysis was conducted without axial displacement limitation. Due to previous factory tests on potential axial displacements, it was determined that in a normal situation, the displacement would never exceed 48 mm. Therefore, it is expected that the vibration frequency would approach zero due to the unlimited axial freedom. Consequently, the primary concern is in the second mode of vibration, which involves rotation.

The graph uses a frequency in rpm because the natural frequency of the rotor was calculated in rpm. To simplify the analysis, the following conversion was used in the Matlab code to represent the graph:

$$N_{rpm} = \frac{120 \times f}{nP} \quad (5.1)$$

In this equation, nP is the number of poles, which in our case is 4. The second mode of vibration shows values of 1690.5 rpm for the motor with the tool endshield and 1989.75 rpm for the natural frequency of the rotor (data with the scale factor). According to the API 541 standard used in the company [39], the operational vibration frequency of the motor should be within 15% above or below the natural frequency. Calculating the percentage difference, we find that the operational frequency of the motor with the tool endshield is 15.039% below the natural frequency, which complies with the standard.

The other modes of vibration are already above the power network, so they will not be excited. This analysis confirms that the motor with the tool endshield operates within safe limits and is unlikely to encounter resonance issues, ensuring reliable performance during its use within the factory.

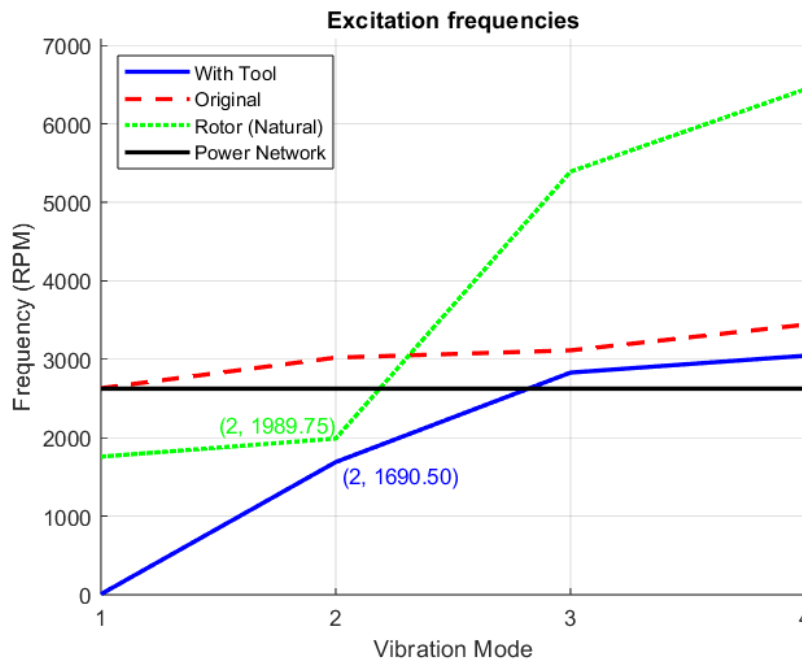


Figure 5.12: Comparison of Frequencies for excitation

5.3.4 Unbalance Analysis

This unbalance analysis aims to understand the impact of the maximum possible thickness difference between the two endshields. Utilizing the axial alignment tool and adhering to the explosion-proof standards for motor endshields, it is possible to reduce the thickness of one endshield by up to 15 mm and increase the thickness of the other endshield by up to 15 mm. It is essential to assess the impact of this change on the motor's vibration and deformation during operation.

To conduct the simulation, two modified endshields were designed: one with a thickness reduced by 15 mm and the other with a thickness increased by 15 mm compared to the original. A

modal analysis was then performed to evaluate the vibrational behavior of the motor with these modified endshields. This approach ensures that the simulation accurately reflects the potential impacts of the thickness variations on the motor's performance.

From the graph in Figure 5.13, it is evident that this variation in the motor's vibration is not significant compared to using endshields with the same thickness. The minor difference in vibration levels indicates that the motor can maintain its stability and operational efficiency even with the maximum allowable thickness variations, ensuring that the motor's performance remains within acceptable limits.

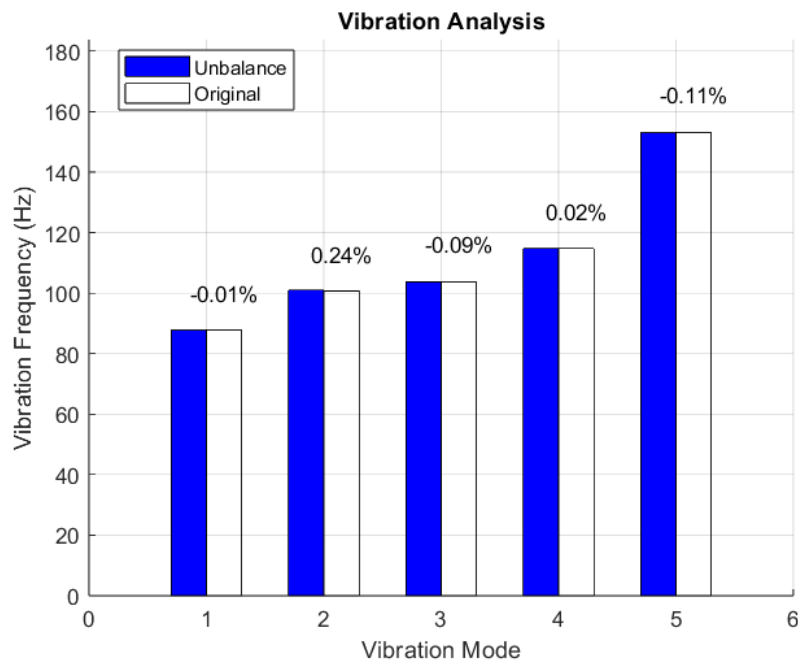


Figure 5.13: Unbalance simulation in modal analysis

Regarding deformation, as shown in Table 5.2, the difference in deformation is also insignificant when compared to using endshields of equal thickness. This negligible difference suggests that the structural integrity of the motor is not compromised by the thickness variation. The endshields, even with their differing thicknesses, provide adequate support and stability, ensuring the motor's reliability and longevity.

Table 5.2: Deformation Comparison - Original and Unbalance Endshield

Endshield	Deformation 1 (Axial)	Deformation 2 (Rotation)
Original	0,57962 mm	1,08927 mm
Unbalanced	0,57906 mm	1,08964 mm

5.3.5 Flange Bending

In the analysis of the flange face bending, the use of shims in the upper area of the flange was considered, as this is where they will be inserted. Consequently, the 1000N distributed load will not be evenly distributed across the entire face but will be more concentrated in the contact area with the shims and subsequently with the endshield, mirroring the actual operational conditions. For the boundary conditions, the face of the hub was fixed to simulate the real attachment of the flange during operation. This focused load distribution more accurately reflects the real-world scenario, ensuring the simulation results are both realistic and reliable.

The stresses observed in this analysis were found to be insignificant, indicating that the flange can withstand the operational conditions without compromising structural integrity or performance.

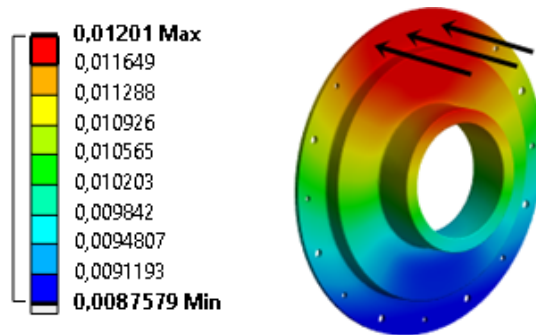


Figure 5.14: Bending Deformation of Flange Face

The results of the deformation obtained in Fig: 5.14 are minimal and do not compromise the confidentiality of the company's data.

5.4 Discussion of Results

This section analyzes and interprets the simulation results, highlighting significant trends and insights. The alignment between the simulation results and the initial objectives and expectations is evaluated. Discrepancies and unexpected results are addressed, providing possible explanations and considerations for future work. The limitations of the simulation procedure and the assumptions made are discussed. Finally, the effectiveness of the simulation procedure in achieving the project objectives and its potential for future applications are reflected upon.

5.4.1 Convergence Analysis and Static Loads

The convergence analysis and static load simulation aimed to understand the rotor's deformation under its own weight and its impact on the endshield's deformation at maximum axial displacement. Expressing displacement as a percentage in Figure 5.5, we noted that the rotor's flexion should be minimal within the air gap. The analysis showed that with a 25mm element size,

displacement converges to less than 1.74% which is acceptable according company standards, and making it the optimal size for future simulations.

The concentration of stress in the model varies significantly depending on the mesh element sizes used. Predictably, the maximum stress should occur at the support contact between the shaft and the sleeve bearing, where the reaction force is exerted. This expected stress concentration is illustrated in the left image of Figure 5.15. However, in the right image, where a larger element size is used, the maximum stress unexpectedly appears at the bottom of the motor frame. This deviation highlights the importance of selecting appropriate element sizes to accurately capture stress distributions.

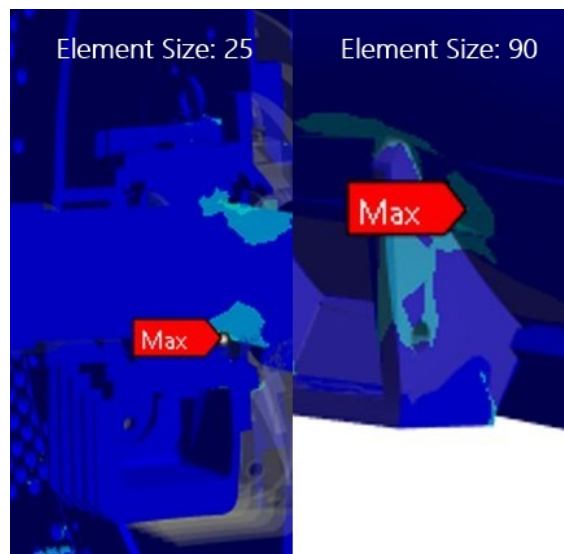


Figure 5.15: Observation of different points of maximum stress depending on mesh element size

However, the simulation process is not without limitations. One significant constraint is the time required for each simulation. Even with an element size of 10mm, the simulation took 8 hours to complete. Reducing the element size further for increased accuracy and converge even more the plot would result in prohibitively long simulation times. Conversely, using larger element sizes can make it challenging to identify refinement spots where the software cannot properly mesh due to excessively large elements. Despite these challenges, an interesting convergence was observed, indicating that the chosen element size of 25mm strikes a balance between accuracy and computational efficiency.

Overall, the results align with initial objectives, providing insights into rotor and endshield deformation. The analysis underscores the importance of appropriate mesh sizing for accurate and efficient simulations, with potential improvements in computational optimization for future studies.

5.4.2 Machining of the Endshield

The results, depicted in Figures 5.7 and 5.8, show that deformation, considering the variation of temperature, is almost negligible in the rib of the endshield hub, which contacts the outer face of the bushing. Ensuring the tolerance between these elements is crucial to avoid excessive material removal, thus maintaining the integrity and performance of the assembly. According to the manufacturer's tolerance (H7), there must be a maximum clearance of $52\text{ }\mu\text{m}$ and a minimum of 0, according to [38]. The precision in machining and the subsequent fit of the bushing within the endshield hub are essential for overall tool reliability.

In the machining simulation, we observed that the inner hub slightly moves inward, indicating that more material might be removed than initially anticipated due to the clamping forces. The simulation shows that this deformation of the inner hub is approximately $1.3\text{ }\mu\text{m}$ 5.8, which is not a significant variation. When drafting the endshield design, it is essential to include a tolerance to ensure the fit between the internal face of the endshield (hole - H7) and the external face of the bushing. Similarly, for the flange, the external face (shaft - h8) must be loose relative to the internal face of the bushing to allow for grease film deposition [37] and ensure free sliding. Additionally, the external face of the shaft should have a surface roughness of RA 0.8. These tolerances can be found in Appendix A.

Regarding the machining simulation results, with a maximum observed deformation of $1.3\text{ }\mu\text{m}$, the clamping forces and temperature variation from 30 to 20 degrees, obtained in laboratory experiments, do not significantly impact the machining process. The same analysis was conducted for other faces and edges of the endshield, and similarly for the flange, where the observed deformation was within the same order of magnitude.

5.4.3 Modal/Axial Displacement Analysis

- **Operational Implications:** The tool endshield is intended solely for use within the factory for axial alignment of the motor and is not designed for operational environments. However, its use in the factory setting provides valuable insights. The reduction in vibration, despite not being part of the final product, suggests that the tool endshield can help ensure precise alignment without introducing excessive vibrations, which could otherwise lead to alignment inaccuracies.
- **Deformation Analysis:** Interestingly, while the vibration frequency decreases with the tool endshield, the deformation tends to increase, particularly in the first mode of vibration. This increase is expected due to the axial freedom introduced by the tool endshield (Figure 5.11). The software calculates displacement from a reference position, and the added axial freedom allows for greater deformation. This result underscores the need to carefully consider axial movements during alignment to prevent excessive deformation that could affect the motor's performance.

- Resonance Considerations:** The final aspect considered was the potential for resonance. Using Mathcad, we calculated the natural frequency of the rotor and plotted it alongside the power network in Figure 5.12. The analysis indicates that with the original endshield, no vibration modes are excited as all frequencies are above the power network threshold. For the tool endshield, the primary concern is the second mode of vibration, involving rotation. Converting the frequencies to rpm, we found that the second mode of vibration for the motor with the tool endshield is 1690.5 rpm, compared to the rotor's natural frequency of 1989.75 rpm. According to the API 541 standard [39], the operational vibration frequency should be within 15% of the natural frequency. The motor with the tool endshield falls within this range, ensuring compliance with the standard and confirming that the motor is unlikely to encounter resonance issues.

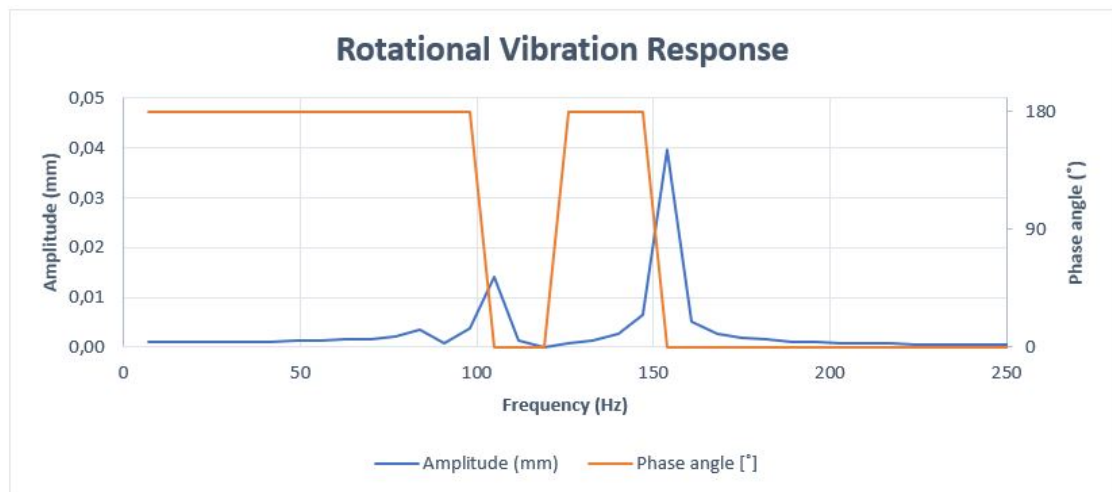


Figure 5.16: Rotational Vibration Response

In this graph 5.16, we observe the vibration response in the second mode (rotation) in relation to the phase angle. After changing the phase angle, two distinct peaks in deformation amplitude are observed. These peaks occur due to changes in the structural resonance of the system, influenced by the modification in the phase angle. This resonance can amplify certain vibration frequencies, resulting in increased deformation amplitudes at specific points within the structure.

It should be noted that the values presented in the graph are scaled according to the previously mentioned scaling factor.

5.4.4 Unbalance Analysis

- Impact on Vibration and Deformation:** The unbalance analysis demonstrates that the thickness variations allowed by explosion-proof standards do not significantly affect the motor's vibration and deformation. This result is critical for ensuring that the motor can operate safely and effectively with these modifications. The minor differences in vibration

levels indicate that the motor remains stable and efficient, even with the maximum allowable thickness variations.

- **Operational Efficiency:** The motor's ability to maintain operational efficiency with the modified endshields underscores the design's flexibility and robustness. This ensures that the motor can meet specific operational requirements without compromising safety.

5.4.5 Flange Bending

In the analysis of the flange face, which experiences the pull of the rotor induced by the magnetic field, we observe a maximum deformation of approximately 0.012 mm (see Figure 5.14). This deformation is not significant enough to raise concerns about the designed thickness of this face.

Chapter 6

Final Considerations

6.1 Synthesis of Key Results

This section resumes the discussion of results highlighting significant trends and insights. The alignment between the simulation results and the initial objectives and expectations is evaluated.

6.1.1 Convergence Analysis and Static Loads

- **Deformation Insight:** The convergence analysis indicated that a 25 mm element size for meshing is optimal, ensuring minimal rotor flexion within the air gap and acceptable displacement levels under company standards.
- **Stress Concentration:** Stress concentrations were accurately identified at the support contact points with appropriate mesh sizing. Deviations were noted with larger element sizes, underscoring the importance of proper meshing for accurate stress distribution.
- **Simulation Limitations:** The simulation time is a significant constraint, with smaller element sizes requiring prohibitive computation times. The chosen element size of 25 mm balances accuracy and computational efficiency.

6.1.2 Machining of the Endshield

- **Minimal Deformation:** Machining simulations showed negligible deformation ($1.3\ \mu\text{m}$) due to clamping forces, ensuring that the machining process does not significantly affect the endshield's integrity.
- **Tolerance and Fit:** Ensuring proper tolerance (H7) between the endshield hub and the bushing and tolerance (h8) between the flange and the internal face of the bush is crucial. The simulations confirm that the endshield can maintain necessary clearances without excessive material removal.

6.1.3 Modal/Axial Displacement Analysis

- **Operational Implications:** The tool endshield reduces vibration and ensures precise axial alignment in the factory, preventing alignment inaccuracies.
- **Deformation Analysis:** Increased deformation in the first mode of vibration is noted due to the axial freedom introduced by the tool endshield, highlighting the need for careful consideration of axial movements.
- **Resonance Considerations:** The motor with the tool endshield operates within the acceptable range of natural frequency per API 541 standards, confirming no resonance issues.

6.1.4 Unbalance Analysis

- **Impact on Vibration and Deformation:** Thickness variations allowed by explosion-proof standards do not significantly impact the motor's vibration and deformation, ensuring safe and effective operation.
- **Operational Efficiency:** The motor maintains efficiency and stability even with modified endshields, highlighting the design's robustness and flexibility.

6.1.5 Flange Bending

- **Deformation Impact:** The result demonstrates that the thickness designed for the flange face is adequate to withstand the axial loads expected during motor operation. The small magnitude of this deformation confirms the robustness of the design and reinforces confidence in the motor's ability to maintain the necessary axial alignment for efficient and safe performance.

6.2 Contributions to the Field of Flameproof Motors

This work significantly contributes to the development and validation of specialized tools designed for precise axial alignment of flameproof motors. These tools are essential in industrial settings, ensuring accurate motor alignment during installation and maintenance processes. The study addresses the specific challenges of aligning motors in hazardous environments, where precise axial alignment is crucial for maintaining safety standards and ensuring reliable motor performance in potentially explosive atmospheres. The tools not only ensure accurate axial alignment with the motor's magnetic field, improving overall performance, but also reduce production time by eliminating the trial-and-error method traditionally used to achieve optimal alignment. Moreover, they prevent contact between the shaft and the lateral face of the sleeve bearing, reducing friction and wear, thus enhancing motor performance.

6.3 Study Limitations

Despite the tool's ability to displace up to 48mm, the thickness of the original endshield's rib can only be reduced by 15mm. This limitation is due to certification and flameproof standards and the current thickness of the raw material, which currently restrict the maximum reduction in thickness. The endshield was designed to support extreme displacements in highly unbalanced assembly situations. However, this maximum reduction limits the tool's application range, potentially necessitating the development of another tool with less axial displacement for less extreme conditions. Furthermore, while the current tool is effective for both extreme misalignments and normal situations, developing a tool specifically for normal conditions would be advantageous for a broader range of scenarios, significantly enhancing the efficiency and safety of the alignment process across diverse operational environments.

6.3.1 DFMEA

Design Failure Modes and Effects Analysis, is a systematic methodology used to identify, evaluate, and prioritize potential failure modes within a product design. The primary goals of DFMEA are to enhance the reliability and quality of the design by identifying potential failure, analyzing causes and effects, prioritizing risks and implementing mitigation.

6.3.1.1 Risk Analysis

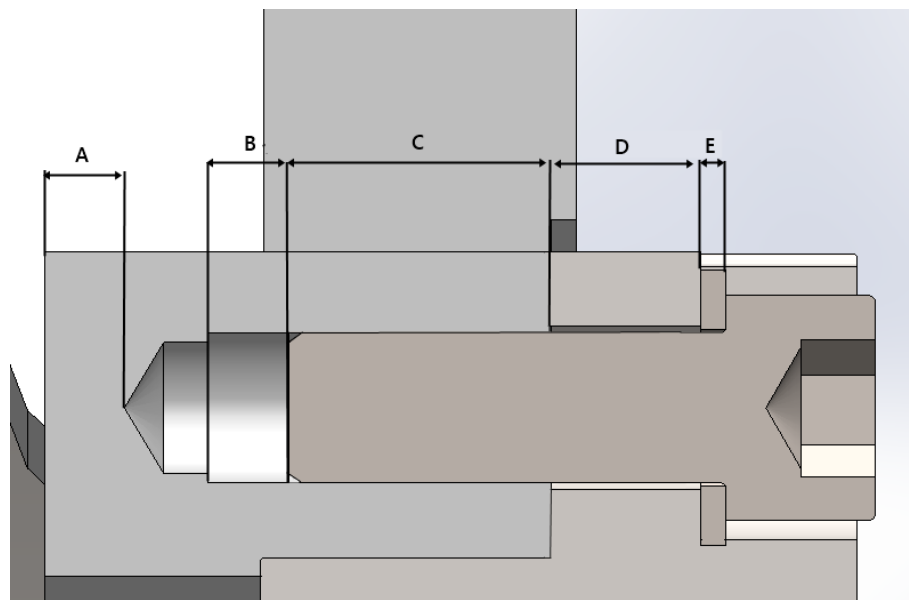


Figure 6.1: Screw Thread and Hole dimensions

When altering the thickness of the endshields (Dimension **D** in Fig. 6.1), careful consideration must be given to the screws used between the housing and the endshield. According to the IEC

60079-1 standard followed by the company, it is necessary to ensure a minimum screw thread engagement length in the housing (Dimension **C** in Fig. 6.1), a minimum distance between the drilled hole and the internal wall of the housing (Dimension **A** in Fig. 6.1), and adequate clearance between the maximum screw length and the end of the threaded hole (Dimension **B** in Fig. 6.1).

The screw used is an internal hex M24-70 with a 3 mm pitch. For this screw, the thread length, dimension (b) Fig. 6.2, is 60 mm.

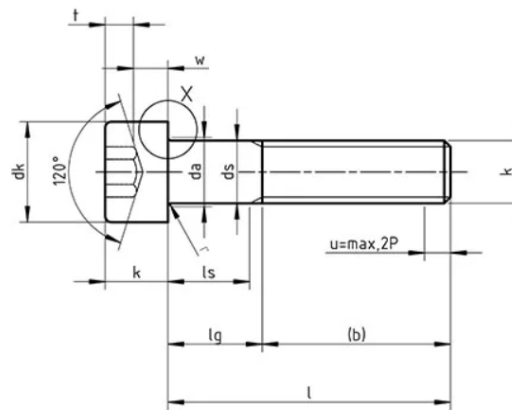


Figure 6.2: Screw Standard Dimensions. Source: ITA Fasteners¹

Best construction practices, as detailed in [38], suggest a minimum thread engagement length in the housing (Dimension **C** in Fig. 6.1) of at least 1x the metric diameter of the screw. In this case, since it is an M24 screw, we need to ensure a minimum thread engagement of 24 mm in the housing. According to the IEC 60079-1 standard for flameproof environments, the minimum thread engagement must be 5 times the pitch. With a pitch of 3 mm, the minimum engagement is 15 mm. We must always adhere to the greater value, hence adopting a minimum thread engagement of 24 mm.

The standard also specifies that the minimum distance between the drilled hole and the internal wall of the housing (Dimension **A** in Fig. 6.1) should be at least the metric diameter divided by 3. For an M24 screw, this distance must be at least 8 mm. Additionally, the clearance between the maximum screw length and the end of the threaded hole (Dimension **B** in Fig. 6.1) should be 1x the pitch, i.e., 3 mm. Since we are adding a washer with a thickness of 4 mm (Dimension **E** in Fig. 6.1), this value must be added to the clearance between the maximum screw length and the end of the threaded hole, making the required clearance at least 7 mm.

Thus, it is necessary to evaluate whether with the alteration of the endshield thickness by ± 15 mm, it is possible to use the same screws, reduce their length, or if it is necessary to use screws of different dimensions.

¹Available at: <https://itafasteners.com/products-socket-head-cap-screws.php>. Accessed on June 20, 2024.

Table 6.1: Screw Thread Limits for Endshield Thickness Variation, Fig. 6.1

Dimension	Original Dimension (mm)	Maximum Limit (mm)	Minimum Limit (mm)
A	13	-	8
B	13	-	7
C	45	60	24
D	-	+15	-15
E	4	4	4

6.3.1.2 Solution Avaluation

• Option 1:

In a brief analysis, we found that with the same hole length, it is not possible to reduce the endshield thickness by 15 mm (Dimension **D** in Fig. 6.1) because the screw length would exceed the minimum clearance of 7 mm between the maximum screw length and the end of the threaded hole (Dimension **B** in Fig. 6.1). Therefore, we would need to increase the hole depth by 9 mm to ensure the 7 mm clearance. However, increasing the hole depth by 9 mm would reduce the minimum distance between the drilled hole and the internal wall of the housing (Dimension **A** in Fig. 6.1) to 4 mm, which is below the minimum limit of 8 mm.

• Option 2:

To ensure the possibility of increasing or decreasing the endshield thickness by 15 mm, we need to shorten the screw length and increase the hole depth by 5 mm, reducing the dimension from 13 mm to 8 mm (Dimension **A** in Fig. 6.1), which is the minimum distance between the maximum hole depth and the internal wall of the housing. By using an M24-65 screw, we can ensure compliance with the minimum values specified by the standards:

- **Endshield increased by 15 mm in thickness:** The threaded length (Dimension **C** in Fig. 6.1) would be 25 mm, and the clearance between the maximum screw length and the end of the threaded hole (Dimension **B** in Fig. 6.1) would be 38 mm.
- **Endshield decreased by 15 mm in thickness:** The threaded length (Dimension **C** in Fig. 6.1) would be 55 mm, and the clearance between the maximum screw length and the end of the threaded hole (Dimension **B** in Fig. 6.1) would be 8 mm.

While this approach ensures compliance with the standard dimensions, it is also essential to consider other factors such as manufacturing defects, welded constructions, etc., which may require us to increase some criteria within the factory to maintain overall quality and safety.

With the change of dimension **A** in Fig. 6.1, some tests should be performed in the factory to ensure that the endshield would withstand the established internal pressure. These tests are critical to validate the structural integrity and ensure that the endshield meets safety standards under various operational conditions.

Additionally, for this case, instead of modifying the hole of the housing, it is possible to increase the raw material of the endshield, as explained in 6.4.1. This approach involves increasing the external thickness of the endshield to compensate for the reduced internal thickness, thereby maintaining structural integrity without altering the housing, or the opposite.

It is important to note that this solution applies only to the maximum adjustment scenario. For adjustments up to 10 mm of axial displacement, it is not necessary to modify the hole, simplifying the implementation and ensuring compliance with existing standards without additional modifications.

- **Option 3:**

The final option is to use different screws depending on whether the endshield thickness is increased or decreased, while maintaining the original drilling.

- **When increasing the endshield thickness by 15 mm:** An M24-85 screw can be used, where the dimensions **C** and **B** in Fig. 6.1 remain the same, as 15 mm is added to both the endshield thickness and the screw length.
- **When decreasing the endshield thickness by 15 mm:** The same principle applies in the opposite direction. An M24-55 screw would be used, ensuring that the dimensions **C** and **B** in Fig. 6.1 remain consistent.

However, this solution presents a significant risk of human error during modifications by the customer. If the customer decides to disassemble the endshield and replace the screws, there is a possibility of incorrect screw placement, compromising the flameproof certification of the motor and posing potential safety risks.

Therefore, we can conclude that the most favorable and practical choice is Option 2.

6.4 Suggestions for Future Research

6.4.1 Endshield Thickness

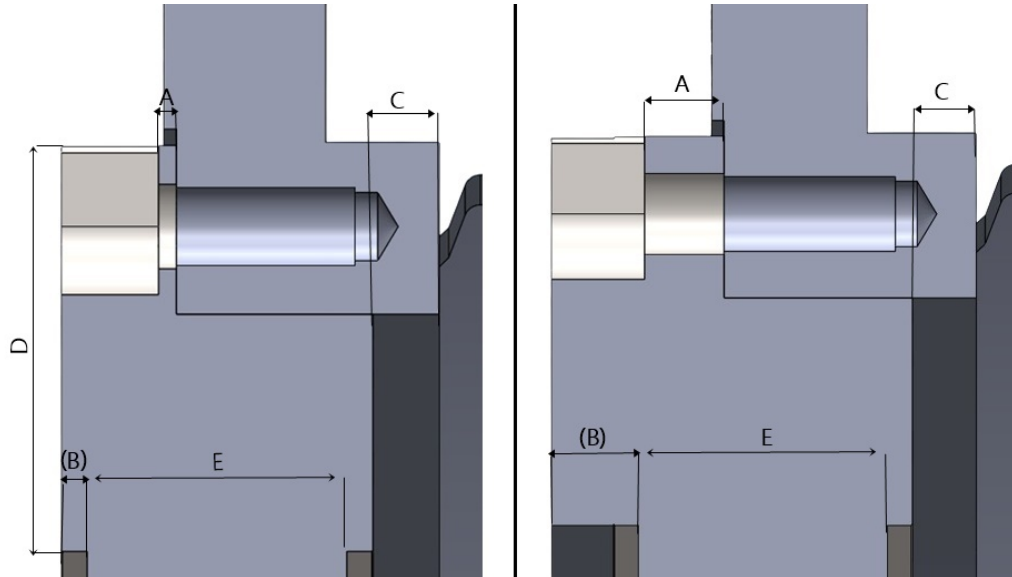


Figure 6.3: Suggestion for expanding the endshield rib

Even though reducing the thickness of the rib by only 15mm is currently feasible, a greater axial displacement could be beneficial for addressing larger potential misalignments. This could be achieved by altering the construction of the original endshield, increasing the thickness of the raw material before machining. By doing so, the external thickness of the endshield rib could be increased, allowing for a greater reduction in the internal thickness of the rib.

The image on the left depicts the original design, while the one on the right shows the proposed modification. In this proposal, dimensions **A** and **B** in Fig. 6.3 would extend equally throughout dimension **D** in Fig. 6.3, which represents the entire external thickness of the rib. Dimension **C** in Fig. 6.3 would remain unchanged. This approach would enable a reduction in dimension **A** in Fig. 6.3 beyond the current 15mm limit as the rib's thickness on the opposite side is increased. Dimension **E** must remain constant, as it ensures the pressure resistance of the endshield.

It is also important to note that the screws could need to be of different lengths on each side, which could pose certification challenges. This issue would need to be evaluated, as it was in the DFMEA 6.3.1.2.

By expanding the raw material, it is also possible to increase the dimension **A** in Fig. 6.1, making Option 2 in the DFMEA more feasible. This approach would eliminate the need for using two different types of screws, thereby simplifying the assembly process and reducing the risk of human error during modifications.

6.4.2 Estimation of Expenses

An analysis of costs, including the price of components, machining costs, production time, and fasteners, can be performed. This assessment helps in understanding the economic feasibility and cost-effectiveness of implementing the tool in manufacturing processes. The detailed cost analysis considered the price of the slide bushing, machining costs, production time, screws, and other components. By evaluating the price of the POM SKF bush, we can determine if it is cost-effective. If the price is deemed too high, we may consider switching to a brass bush. However, this alternative would necessitate additional simulations to ensure its suitability.

6.4.3 Machine Learning and Optimization

Developing a machine learning program to automatically adjust the tool's dimensions based on the specific motor being used offers significant potential. This approach would involve creating an algorithm that can analyze various motor specifications and operational requirements to determine the optimal settings for the axial alignment tool. The program would be trained on a comprehensive dataset, including different motor models, sizes, and alignment scenarios. By leveraging machine learning, the tool could provide precise adjustments of the dimensions of the endshield and the flange and even the proper bush for the case.

It would also be beneficial to optimize the tool's dimensions to eliminate the need for design changes for each motor shaft height and bearing type. Standardizing the tool design to be adaptable for various combinations would result in significant industrial and cost efficiencies. By optimizing the model to allow a single tool to be reused for different combinations, we can achieve substantial industrial and cost benefits. This optimization could be achieved through a machine learning process, which would analyze each possible shaft height/bearing combination and adjust the tool dimensions accordingly. Additionally, a dimensional analysis process could be employed to reduce the number of tools required to cover all possible combinations. This approach would streamline the alignment process, making it more versatile and cost-effective.

6.4.4 Less axial displacement

Investigating the feasibility and effectiveness of a tool designed to allow less axial displacement is another promising area for future research. Such a tool would be tailored for scenarios where extreme alignment adjustments are unnecessary, focusing on the majority of alignment tasks that fall within a normal range. By limiting the axial displacement capacity, the tool could be simplified, potentially reducing manufacturing costs and improving ease of use. Additionally, it would minimize the risk of over-adjustment, which could lead to unnecessary wear or damage.

Chapter 7

Conclusion

7.1 Recapitulation of Objectives

The objectives of this work focused on the development of an axial alignment tool for a flameproof motor with sleeve bearings that do not support axial loads, allowing for the axial displacement of the rotor and subsequent misalignment. This misalignment is observed due to vibration, the force exerted by the external fan, manual errors during the construction and assembly of the motor, and other factors. The primary issue lay in the time-consuming manual alignment of the rotor to ensure optimal alignment and to prevent the rotor shaft walls from contacting the bearing, which would generate friction.

The goal of this project was to create a design proposal to solve the existing problem and to evaluate the alternative through computational mechanical simulation processes. This approach aimed to understand the tool's influence on motor performance during the alignment process in the factory. After achieving the correct rotor alignment in the factory, the objective is to machine the thickness of the original flameproof endshields, which will be mounted on the motor for the customer, to the correct thickness obtained by the alignment tool. This ensures the rotor is axially aligned with its magnetic field.

In summary, the key objectives of this thesis were:

- **Development of an Axial Alignment Tool:** Create a specialized tool designed to facilitate precise axial alignment of the flameproof motor rotor, compensating for the inability of the bearings to support axial loads.
- **Reduction of Manual Alignment Time:** Address the problem of lengthy manual alignment processes by providing a tool that ensures quick and accurate alignment, minimizing the chances of rotor shaft walls contacting the bearing and causing friction.

- **Mechanical Simulation and Evaluation:** Use computational mechanical simulations to evaluate the proposed design and understand its impact on motor performance during the alignment process.
- **Implementation in Factory Settings:** Ensure the tool can be used in factory settings to align the rotor accurately, after which the thickness of the flameproof endshields will be machined to match the alignment achieved by the tool, ensuring proper axial alignment of the rotor with its magnetic field.
- **Customer-Specific Customization:** Machine the endshields to the correct thickness for each motor, as determined by the alignment tool, to provide a custom solution for each customer's motor, ensuring optimal performance and compliance with safety standards.

This comprehensive approach not only addresses the immediate problem of misalignment in flameproof motors but also enhances the overall efficiency and reliability of the alignment process in various operational environments, ensuring both safety and performance standards are met.

7.2 Final Conclusions

In conclusion, the tool was developed for application to a specific motor and sleeve bearing to align its rotor with the magnetic field, receiving validation through comprehensive computational simulations that accounted for a range of parameters, including industry standards and internal criteria. This rigorous validation process ensures the tool's reliability and effectiveness in real-world applications.

Throughout the project, various mechanical computational simulation methods were explored and mastered. These methods enabled a detailed analysis of the tool's behavior under both typical and extreme conditions, providing valuable insights and enhancing our ability to predict and mitigate potential issues.

One of the major difficulties encountered during the project was learning the techniques and tricks of the software, as well as understanding which simulations were appropriate for this type of tool. Additionally, there were challenges in integrating the simulation results with practical engineering constraints and ensuring the alignment of theoretical models with real-world manufacturing conditions.

The tool's implementation within the factory is expected to significantly improve production times and address potential misalignments that could otherwise damage the sleeve bearing. By ensuring precise axial alignment, the tool helps maintain optimal motor performance and extends the lifespan of critical components.

While the initial design includes a significant range for axial movement to accommodate extreme cases, future improvements could focus on limiting this displacement for more controlled scenarios. This approach allows for greater flexibility during the testing phase while providing room for refinement based on operational feedback.

Additionally, there is substantial potential for industrial optimization by creating a more versatile tool capable of accommodating various combinations of sleeve bearing and shaft heights. This would reduce the need for multiple specialized tools, leading to cost savings and increased efficiency. Tool's adjustments based on specific motor requirements could further enhance this versatility, enabling precise, data-driven refinements that ensure the tool's adaptability across a wide range of applications.

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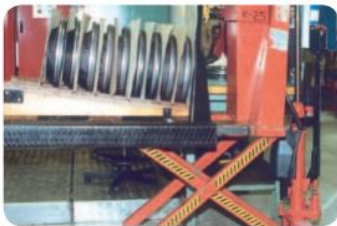
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Appendix A

SKF POM composite bushings, thrust washers and strips



Optimal combination of minimal maintenance under tough operating conditions

SKF POM composite plain bearings are referred to as prelubricated because they require only a trace of lubricant to operate satisfactorily for long periods. The bearing material is designed to operate with marginal lubrication and effectively fills the gap between fully lubricated bearings and dry sliding bearings. The SKF assortment of POM composite plain bearings consist of a wide range of straight bushings, thrust washers and strips.

Straight bushings are manufactured to dimensions in accordance with ISO 3547-1.

Advantages of SKF POM composite plain bearings include:

- maintenance-free operation
- very good frictional properties
- high load carrying capacity
- high sliding velocity
- small operating clearance



Material

SKF POM composite plain bearings are suitable for applications that require minimal maintenance under difficult operating conditions. As a result of the lubricant retention pockets on the sliding surface, SKF POM composite plain bearings are especially well-suited for applications in contaminated environments where lubricant cannot be supplied continuously or frequently.

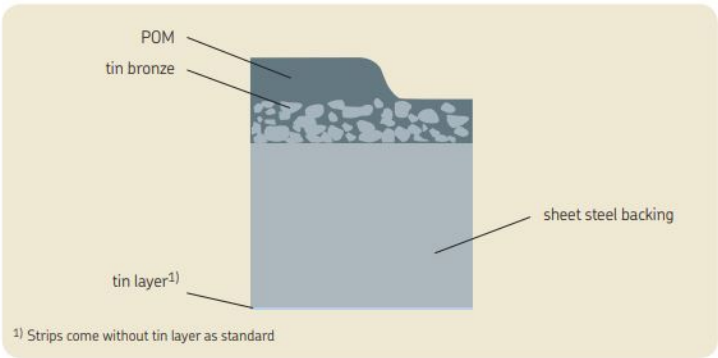
Main applications¹⁾

SKF POM composite plain bearings are suitable for applications where there are heavy loads and where the self-lubricating properties of the material are a prerequisite.

Applications include:

- agricultural equipment
- construction machinery
- material handling equipment
- home appliances and consumer goods

Cross section of SKF POM composite plain bearings



¹⁾ The performance of SKF POM composite plain bearings depends on the interaction of load, lubrication, surface roughness, sliding velocity, and temperature encountered in specific applications.

Lubrication

SKF POM composite plain bearings are designed to operate under marginal lubrication conditions. The sliding surface of these plain bearings contains grease reservoirs which should be filled prior to installation. It is not necessary to relubricate these plain bearings, but the presence of a lubricant can extend the plain bearing service life considerably. To protect the mating surface against corrosion, grease can be applied periodically.

Characteristics	
Permissible load (dyn/stat), N/mm ²	120 ($v \leq 0,02$) / 250
Permissible sliding velocity, m/s	2,5 ($p \leq 1,0$) ¹⁾
Friction coefficient μ	0,02 .. 0,20
Temperature range, °C	-40 .. +110
Application recommendations	
Shaft tolerance	f7 – h8
Housing tolerance	H7
Shaft roughness R_a , μm	0 .. 0,8
Shaft hardness, HB	150 – 600
¹⁾ See page 20	

Further details and comprehensive specifications can be found in the SKF technical documentation [37].