Stress Analysis for Rotor Shaft of Electric Motor

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Abstract

The intent of this paper is to study stress in rotor shaft of electric motor. Shaft is common component used in variety of applications across industries. This is typical shaft used in motor which carries rotary components. Analytical static design is well defined procedure whereas analytical dynamic design of shaft for has not been well established. In case of dynamic operations the behavior of shaft is different and stresses induced are multi-fold than static case. The motor rotor shaft is designed on the basis of static and dynamic loadings. The design is validated analytically for static case and pattern is observed.
In this paper a validation of static design carried out with FEA solution for static and dynamic case is done. A result is presented with the designs available for applications. Some recommendations are made regarding possible solutions considering various types of loading conditions. Stresses, deflections and safety factors of the shafts are checked by commercial finite element analysis software (ANSYS 13.0).
Results of validation of design carried out with FEA solution for static and dynamic conditions are presented. Results of finite element analysis software are within the limits as compared to analytical method.

Keywords: Shaft, Static Loading, Dynamic Loading, CAD and FEA

1. INTRODUCTION

Electric motors are well known as 'Basic Prime movers' for Engineering Applications. Electric motors are available at different speed, sizes & output power. As a general rule, conversion of electric power into mechanical power takes place in the rotating part of electric motor. Stresses in rotor shafts are complex, normally involving combinations of bending, torsion, and shear. Some stresses alternate or reverse in direction with each shaft revolution; others may be cyclically varying. Good design dictates holding maximum stress below the endurance or fatigue limit - the fairly low stress which can be sustained indefinitely through cycles of application.

Unlike strength, shaft stiffness (resistance to bending) is independent of steel composition. Shaft critical speed or resonance is a function of shaft deflection under the influence of gravity. Calculated shaft deflection, depends on rotor weight to be carried by the shaft, and upon the method chosen to allow for the non-uniform diameter of most shafts. The shaft-rotor assembly will assume a deflection curve shape by bending under its own weight. Also depending upon the couplings used, the shaft may be subjected to unacceptably high bending stress. When an existing motor is redesigned for higher horsepower, or is applied to a different type of load (particularly with a belt drive), a design analysis is necessary to ensure that increased stress does not risk shaft failure.

2. PROBLEM IDENTIFICATION

2.1 FAILURE CAUSES

The shaft is designed to have an infinite life. This will be the case unless the shaft is overloaded or damaged. In order to properly dimension the rotating system, the failure causes or modes have to be determined. Most failures are due to unpredictable factors and the cure is to create a robust design that can cope with the unknown. Predictable failures or lifetimes can be calculated although many parameters will vary and statistical considerations have to be taken into account.

Fatigue crack

The cause of a broken shaft is almost always fatigue. A crack starts at a stress concentration from a keyway or a sharp radius or in rare cases, from a material impurity. Flaws in the surface of a shaft, such as scratches, indents or corrosion, may also be the starting point of a fatigue crack. Loads that drive a crack are normally torsion loads from direct online starts or bending loads from the pulley or coupling end.

Plastic deformation

Plastic deformation can only occur in extreme load cases when debris is squeezed into a radial clearance that results in large deformations.

Defective shaft
Defective shafts that have passed the checks in the factory, checks for unbalance, as well as the check at test run are very rare. On site, however, it is important to prevent damage from corrosion, indents etc, by proper handling.

Joint
A poorly designed joint between the shaft and the rotor can be detrimental for the system both at mount (high mounting forces, impacts) and at run.

Loose connection
A loose connection can cause impacts that generate high loads and damage on the shaft.

Misaligned connection
This is normally hard to achieve but with incorrect mounting it may occur.

Bearing
A roller bearing does not last forever, sooner or later fatigue or wear, or lubricant deterioration will ultimately destroy the bearings ability to function properly.

2.2 SHAFT FUNCTIONS
- Provide an axis of rotation
- Used to transmit power
- Used to position / mount gears, pulleys, bearings etc.

SHAFT DESIGN ISSUES
- Geometry (stepped cylindrical geometry)
- Loading
- Stress and Deflection (Strength and Rigidity)

SHAFT DESIGN PROCEDURE
- Develop a static free-body diagram.
- Draw a bending moment diagram in two planes.
- Develop a torque diagram.
- Establish the location of the critical cross section.
- Perform a stress analysis for sizing.

SHAFT ASSEMBLY
- Shaft with two bearings at A and B and with resulting forces P1 and P2,
- Free body diagram of torque and forces from   assembly drawing;
- Moment diagram in x-z and x-y planes;
- Torque diagram.

SHAFT DESIGN GUIDELINES
- Keep shafts short and minimize simply supporter   designs.
- Hollow shafts have better stiffness / mass ratios, but   are more expensive.
- Configure shaft geometry to reduce stress concentrations.
- Remember that gears / pulleys can produce radial, tangential and axial loads.
- Be aware of maximum shaft deflection requirements of bearings.
- Shaft natural frequency should be as high as practical.

2.3 LOADS

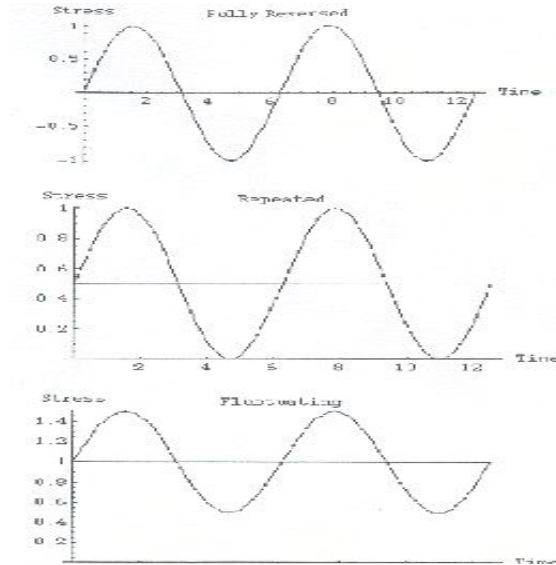| *Static Loading*	*Dynamic (Cyclic) Loading*
Radial	Fully reversed
Tangential	Repeated
Axial (Thrust)	Fluctuating

Figure.1 Stress pattern

STRESS EQUATIONS

- Stress due to Axial Loading

$$\sigma_x = \frac{4F}{\pi d^2}$$

- Stress due to Bending

$$\sigma_x = \frac{My}{I}$$

$$\sigma_{max} = \frac{32M}{\pi d^3}$$

- Stress due to Torsion

$$\tau_{xy} = \frac{16T}{\pi d^3}$$

STRESS CONCENTRATIONS

- Stress caused by a sudden change in form
- Fillets (on shoulders)
- Holes (for pins)
- Grooves (for snap rings)

$$K = \frac{\text{highest value of stress on "feature"}}{\text{Nominal stress on minimum cross section}}$$
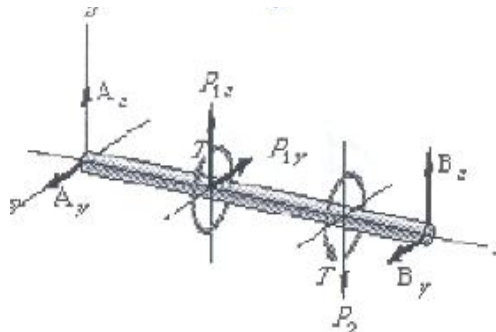
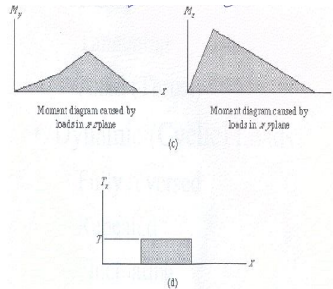

Figure.2 Force diagram of shaft

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Figure.3 Stress concentrations

3 .FINITE ELEMENT ANALYSIS

Computer-aided engineering (CAE) is the application of computer software in engineering to evaluate components and assemblies. It encompasses simulation, validation, and optimization of products and manufacturing tools. The primary application of CAE, used in civil, mechanical, aerospace, and electronic engineering, takes the form of FEA alongside computer-aided design (CAD).

In general, there are three phases in any computer-aided engineering task: Pre-processing is defining the finite element model and environmental factors to be applied to it Analysis solver is solution of finite element model Post-processing of results using visualization tools.

Pre-processing

The first step in using FEA, pre-processing, is constructing a finite element model of the structure to be analyzed. The inputs of a topological description of the structure's geometric features are required in most FEA packages. This can be in either 1D, 2D, or 3D form, modeled by line, shape, or surface representation, respectively, although nowadays 3D models are predominantly used. The primary objective of the model is to realistically replicate the important parameters and features of the real model. The simplest mechanism to achieve modeling similarity in structural analysis is to utilize pre-existing digital blueprints, design files, CAD models, and/or data by importing that into an FEA environment. Once the finite element geometric model has been created, a meshing procedure is used to define and break up the model into small elements. In general, a finite element model is defined by a mesh network, which is made up of the geometric arrangement of elements and nodes. Nodes represent points at which features such as displacements are calculated. FEA packages use node numbers to serve as an identification tool in viewing solutions in structures such as deflections. Elements are bounded by sets of nodes, and define localized mass and stiffness properties of the model.

Analysis (computation of solution)

The next stage of the FEA process is analysis. The FEM conducts a series of computational procedures involving applied forces, and the properties of the elements which produce a model solution. Such a structural analysis allows the determination of effects such as deformations, strains, and stresses which are caused by applied structural loads such as force, pressure and gravity.

Post processing (Visualization)

These results can then be studied using visualization tools within FEA environment to view and to fully identify the implications of the analysis. Numerical and graphical tools allow the precise location of data such as stresses and deflections to be identified.

4. ROTOR SHAFT ASSEMBLY MODEL VERIFICATION

When viewing the cantilever rotor shaft assembly it can be hypothesized that the motor body is analogous to the fixed base of a cantilever beam. Virtually all displacements will occur in the rotor shaft, the upper support collar and the rotor shaft section above the motor body. The combined length of the upper support collar and exposed rotor shaft is greater than the outside diameters of both the upper support collar and the rotor shaft. Thus, the assembly may be approximated as a cantilever beam with an additional end mass as shown.
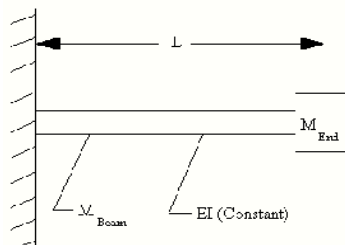

Figure.4 Cantilever Beam

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From Rao [8] it is known that this is easily treated as the following single degree of freedom spring-mass system with the following equivalent stiffness K and mass M values.
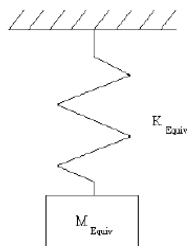

Figure.5 spring-mass system

$$K_{equiv} = \frac{3EI}{L^3}$$

$$K_{equiv} = 0.23 M_{BEAM} + M_{END}$$

E = Modulus of Elasticity
I = Area Moment of Inertia
L = length of beam

The problem with this approach is that the section modulus EI for the beam must remain constant. The upper support collar introduces a change in I for the system, thus violating this requirement. The decision was made to analyze the beam using I values for the upper support collar I1, the rotor shaft I2 and an equivalent I value weighted for I1 and I2. From Thomson [5] the first natural frequency of a cantilever beam may be found using the following relationship.

$$= 3.52 \sqrt{\frac{EI}{M_{BEAM} L^3}} \qquad y_{AB} = \frac{Px^2}{6EI} (3a-x)$$

$$y_{BC} = \frac{Pa^2}{6EI} (3x-a)$$

Where,
y = displacement, P = applied load

From Cheng [6] it is known that the static deflection of a cantilever beam due to an in-plane force perpendicular to the axial direction as shown may be found through the following relationships.
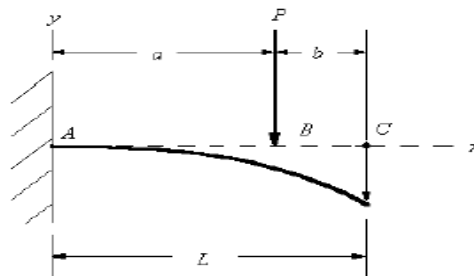

Figure.6 static deflection of a cantilever beam

5. CONCLUSION

Following conclusions are drawn from this paper: The motor rotor shaft is designed on the basis static and dynamic loadings. The design is validated analytically for static case and pattern is observed for dynamic case. As expected in dynamic loading stresses induced in shaft are higher than static case. The stresses induced in the shaft are validated by FEA using ANSYS software for both the cases.

The maximum bending moment in shaft is near right hand bearing support, at rotor location. The maximum equivalent stress is in keyway and bearing support area in both the cases. Problem in ANSYS were solved using SOLID elements. Moreover, Finite element analysis is also carried out to find deflection in the shaft, which is within allowable limit for both cases.

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