Design and Analysis of a Mechanical Component (Shaft) in the Alternator

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***Abstract*—an Alternator is electrical generator that converts mechanical energy to electrical energy in the form of alternating current. Alternators are designed to meet the customer requirements in remote as well as urban areas for uninterrupted power supply. Large range of alternator ratings available in the market from 2KVA up to 5000KVA.Alternator consists of number components like Bearing , Shaft, Stator , Rotor, Windings, Damper bars , Support bars, Drive end and Non drive brackets , wedges, fan etc.Shaft is common component used in variety of applications across the industries. This is shaft used in Alternator which carries rotary components.The Alternator shaft is designed on the basis of static and dynamic loadings by using analytical method and validation of static design carried out with FEA solution for static and dynamic case.**

I. Introduction

An alternator is an electrical generator that converts mechanical energy to electrical energy in the form of alternating current.Alternators are well known as 'Basic Prime movers' for Engineering Applications. Alternators are available at different speed, sizes & output power. Due to its clean & noiseless operation, the Alternators are very popular drive [1].

The Alternator have two ends drive end and non drive end as shown in figure 1 . Different member of alternator like main rotor, excitor rotor, fan are mounted on the shaft. These members along with the forces exerted upon them causes the shaft to bending. In other words, one may say that a shaft is used for the transmission of torque and bending moment.

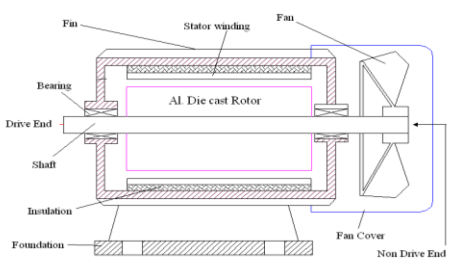


Figure 1. Principal parts of Alternator

II. Problem Definition

It is observed that in the industry like Cummins Alternator ranging from 100KVA to 300KVA they have using the shaft. Because of which the overall size of smaller rating alternator are increased. It is observed that 125KVA and 160KVA ratings Alternator are largest selling Alternator. So if one can redesign the shaft of 160KVA rating then one may get the marginal difference in safe diameter which results in reduction in weight and saving of cost.

The aim is to redesign and Analyze Shaft of 160 KVA Alternator. And forming design concept for reducing weight and size of Alternator

III. Objective

Evaluation of key material reduction opportunities for Key Components of Alternators like Shaft.Design and Optimization of Alternator shaft. Validation of the analytical calculations by the FEA. Concept creation for reducing the Weight and size of an Alternator.

IV. Stresses in the Shaft

The following stresses are induced in the shafts: Shear stresses due to the transmission of torque (i.e. due to torsional load, Bending stresses (tensile or compressive) due to the forces acting upon machine elements like main rotor, exciter rotor, Fan etc. as well as due to the weight of the shaft itself, Stresses due to combined torsion and bending loads

V. Design of Minimum Shaft Diameter

Design of minimum Shaft Diameter of 160 kVA Alternator is verifiedwhen Shaft is subjected to combined twisting moment and bending moment. It is observed that Shaft is safe even after reducing the shaft diameter by 10mm of 160KVA Alternator by strength point of view.

Shaft is redesigned according to newly calculated diameter. Next step is to analyse the shaft by doing FEA analysis to check for deflection and stresses induced.

Figure 2. Creo Model of Shaft

VI. Static Analysis

Static Analysis of shaft includes the calculation of maximum deflection and stresses induced in the shaft under static conditions which means when no torque is applied on the shaft.

A. Shaft deflection by using Macaulay's method

Macaulay's Method is a means to find the equation that describes the deflected shape of a beam. From this equation, any deflection of interest can be found Before Macaulay's paper of 1919, the equation for the deflection of beams could not be found in closed form. Different equations for bending moment were used at different locations in the beam.

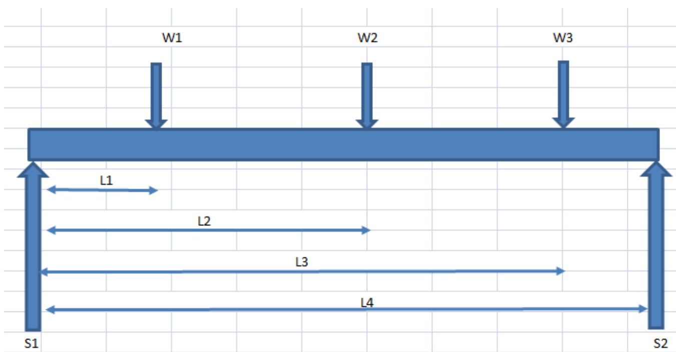


Figure 3. Free body Diagram of Shaft(1)

Where
W1= Weight of Excitor Rotor=120.46N
W2= Weight of Main Rotor + Self Weight +UMP= 1567N
W3= Weight of the Fan=54.24N
S1=Bearing Support
S2= Coupling Support

Therefore,Accourding to Macauley's method, The bending moment at any section distance X from bearing support B given by –

$$EI\frac{d^2y}{dx^2} = S1(x) - W1(x - L1) - W2(x - L2) - W3(X - L3)$$

W2 having large value as compared to W1 and W3.So it is obvious that deflection because of weight W2 is maximum.Therefore by intigrating above equation and substituting the value one can get the maximum deflection value.

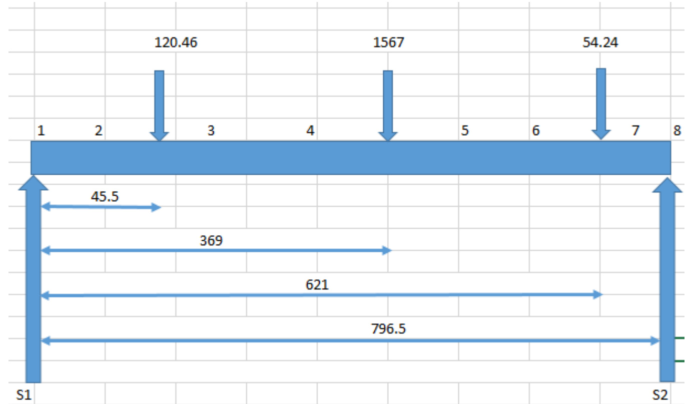


Figure 4. Free body diagarm of Shaft(2)

$$EI\frac{d^2y}{dx^2} = 967(x) - 120.46(x - 45.51) - 1567(x - 369) - 54.24(X - 621)$$

$$EI\frac{dy}{dx} = \frac{967x^2}{2} + C1 - \frac{120.46(x - 45.5)^2}{2} - \frac{1567(x - 369)^2}{2} - \frac{54.24(x - 621)^2}{2}$$

$$EIY = \frac{967x^3}{6} + C1Y + C2 - \frac{120.46(x - 45.5)^3}{6} - \frac{1567(x - 369)^3}{6} - \frac{54.24(x - 621)^3}{6}$$

Further by substituting the boundary conditions, one can get the values of C1 and C2. Maximum deflection is given by rotor region. To find maximum deflection, put X= L2=369
By analytically deflection is-
Ymaximum = 0.0544 (In rotor Region)

B. Stresses induced in shaft

Equivalent Torque calculate by using following formula

$$T_e = \sqrt{(kb.mb)^2 + (kt + mt)^2}$$

= 936792.267 N-mm
Using torsion theory

$$\tau_{max} = \frac{16T_e}{\pi d^3}$$

τ_{max}Induced= 11.30914 N/mm2
For given material: 40C8
Sut = 550 N/mm2;Syt = 290N/mm2
Assuming FOS = 2
τAllowable = 290/2 = 145 N/mm2
τAllowable = 145/2 = 72.5 N/mm2
τMaximum induced <<τallowable
Hence shaft is safe for given loading condition

C. Finite Element Approach

The FEA is a numerical procedure for analyzing structures of complicated shapes, which otherwise would be difficult by classical analytical methods. Analytical solution is a mathematical expression that gives values of desired unknown quantity at any location in a body or a structure and it is valid for an infinite number of locations in body or structure. But analytical solutions can be obtained only for simple engineering problems. It is extremely difficult and many a times impossible to obtain exact analytical mathematical solutions for complex engineering. Problems in such cases FEM is used which gives approximate solution

Static structural analysis of shaft by using ANSYS

Cad model

CAD model resembling the actual shaft which is modeled in Creo Parametric 2.0 and then exported as x*t file. This x*t is imported in ANSYS 11 and used for further FEA analysis.

Mesh Generation

After successful import of CAD model, elements were generated with SOLID 187 element of ANSYS.SOLID 187 is 3-D 10-Node Tetrahedral Structural Solid SOLID187 element is a higher order element. It has a quadratic displacement behavior and is well suited to modeling irregular meshes (such as those produced from various CAD/CAM systems).

Figure 5. Meshed Shaft

The element is defined by 10 nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions.

Number of elements in model: 355611

Number of nodes in model: 249372

Loads and boundary conditions

Loads like weight of the Excitor rotor, main rotor, Fan are applied on shaft, Apart from these load, two boundary conditions are applied at bearing location and coupling location.

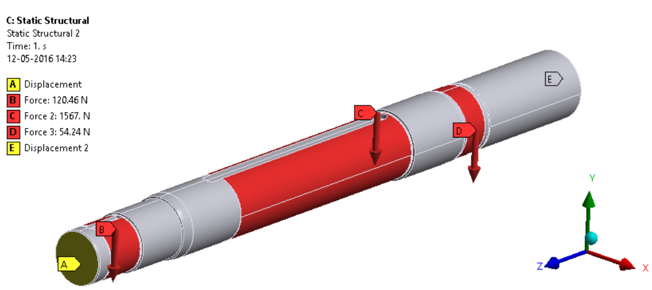


Figure 6. loads and boundry conditions of the shaft

Material Properties

Material: 40C8 Steel

Young’s Modulus of Elasticity: 210 GPA

Poisson’s Ratio: 0.3

Density: 7500 Kgf/m^3

Results:

Induced Deflection in the shaft

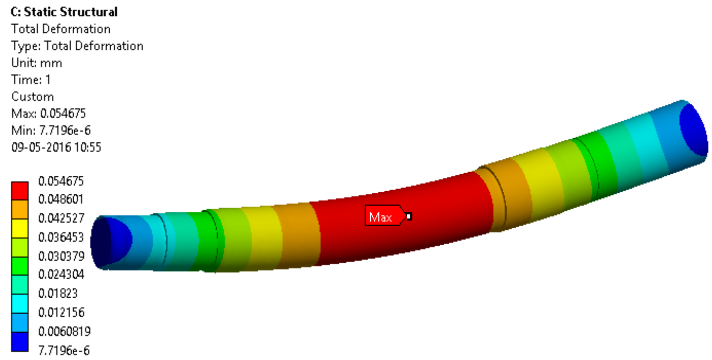


Figure 7. Maximum deflection of the shaft

Stressses induced

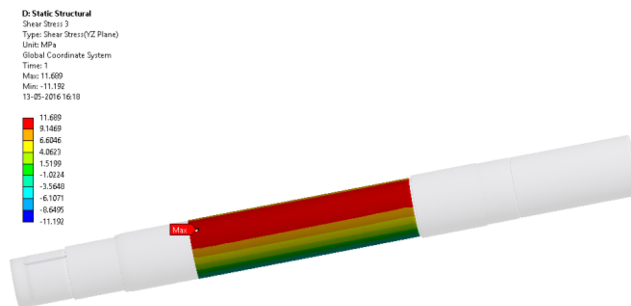


Figure 8. Maximun Stresses Induced in the shaft

Table 1. static result comparison

Item	Analytical	FEA
Stresse τ(N/mm^2)	11.309	11.689
Deflection Y(mm)	0.0544	0.0546

D. Acceptance Criteria for Maximum Deflection

Maximum deflection of Shaft should be less than 5% of the air gap between the Stator and the Rotor. 0.054 deflection value satisfies the criteria of maximum deflection. By analytically and by using FEM approach maximum deflection value of shaft is 0.054mm.

VII. Dynamic Analysis of Shaft

Dynamic analysis of shaft includescalculation of critical speed and calculation of Transverse natural frequency.

A. Critical speed and frequency of transverse vibration

The speed at which the shaft runs so that the additional; deflection of the shaft from the axis of rotation becomes infinite, is called as critical or whirling of shaft.

To determine the critical speed of a shaft which may be subjected to point loads one can find the natural frequency of transverse vibration which is equal to critical speed of a shaft in r.p.s. The Dunkerley's method may be used for calculating the frequency.

Dunkerley's method

The natural frequency of transverse vibration for a shaft carrying number of point loads is obtained from Dunkerley's empirical Formula.

According to this,

$$\frac{1}{(f_n)^2} = \frac{1}{(f_{n1})^2} + \frac{1}{(f_{n2})^2} + \frac{1}{(f_{n3})^2} + \cdots \frac{1}{(f_{ns})^2}$$

Where

$f_n =$ Natural frequency of transverse vibration of the shaft carrying point load or uniformly distributed load

f_{n1}, f_{n2}, f_{n3}= Natural frequency of transverse of the each point load

$f_{ns} =$ Natural frequency of transverse vibration of the uniformly distributed load (or due to mass of the shaft)

Let $\delta_1, \delta_2, \delta_3$ =deflection due to the load W1, W2, W3 etc when considered separately,

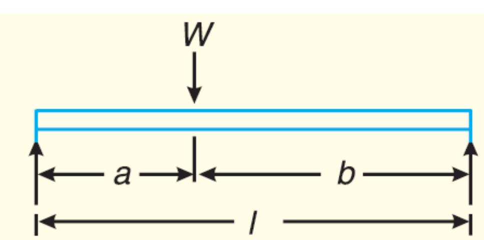
Natural frequency of transverse vibration is given by,

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{\delta}}$$

Which can be written as,

$$f_n = \frac{0.4985}{\delta}$$

B. Deflection of Simply supported beam with an eccentric load.



$$\delta = \frac{Wa^2b^2}{3EIL}$$

Deflection due to Excitor rotor = $\delta 1$ = 0.000789 mm

Deflection due to Main rotor+ UMP +Self weight = $\delta 2$ =0.05275 mm

Deflection due to Excitor rotor = $\delta 3$ =0.0011746 mm

C. Natural transverse frequency according to Dunkerlays Equation

$$f_n = \frac{0.4985}{\sqrt{\delta_1 + \delta_2} + \delta_3 + \frac{\delta_s}{1.27}}$$

By substituting the values one can get,

$f_n = 69.46\ Hz$

D. Dyanamic analysis of shaft FEA result

A dynamic analysis of shaft is the one in which result of modal analysis is used to calculate critical speed or transverse natural frequency.

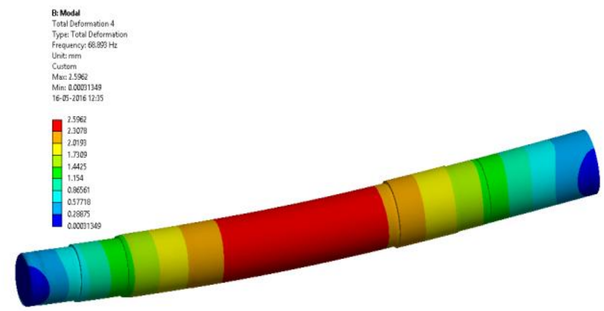


Figure 9. Modal analysis of shaft

Table 2 Dynamic result comparison

Item	Analytical	FEA
Transverse frequency (HZ)	69.46	68.89

E. Acceptance criteria for Transverse natural frequency

As natural frequency of 160 KVA 50Hz Alternator stator lies in between 10% of the given frequency that means in between 45Hz to 55Hz, so to avoid resonance condition shaft frequency should not lie between the above mentioned range. Calculated frequency is coming around 70Hz. So we can conclude that the Shaft is safe dynamically.

VIII. Concept for reducing the size of an Alternator

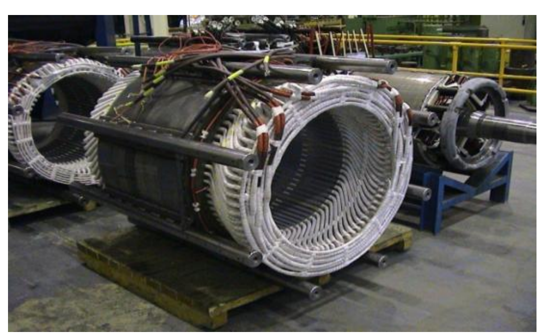


Figure 10. Stator with landing bars

Figure 10 indicates the present design of an Alternator in which the Landing bars are directly welded on stator core. Landing bars are 6 in numbers and having around 35-40 mm diameter which individually weights approximately 10kg.The thickness of the barrel is 1.5mm which wraps around the landing bars.

One can eliminate the landing bars by providing pads on stator core and which can press fit with the round shaped barrel having thickness around 4 mm .(Refer figure 11)

Stator core is made up stacking process of stator laminations. One can give the profile of pad on stator lamination itself to reduce the scrap.

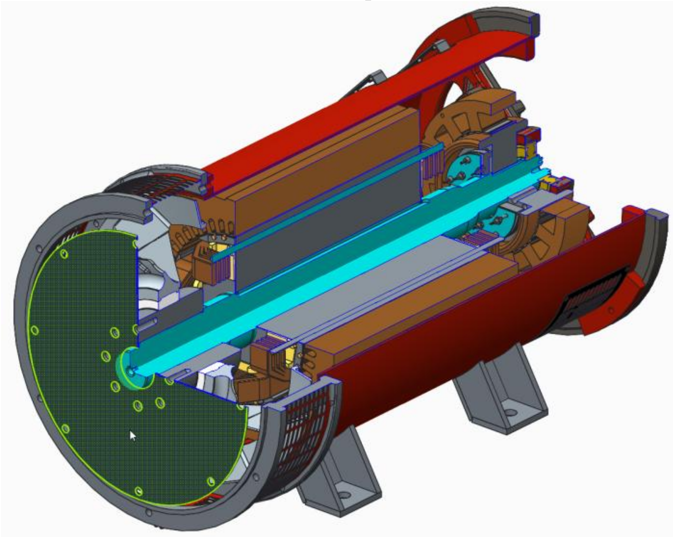


Figure 11. Press fit stator core and barrel

Results:

Pads are formed by stacking of stator laminations as one can give the profile on stator lamination. Scrap of steel sheet can also get reduced by making such arrangement. Weight and size will get reduced by doing such arrangement.

Approximately weight of landing bars with barrel= 60kg

Approximately weight of new press fit concept= 42kg

IX. CONCLUSION

The 160kva alternator shaft is redesigned. Design is validate analytically for static case and dynamic case. Deflection, stresses, transverse frequency are validate by FEA using ANSYS software. The maximum deflection and stresses in shaft is observed at main rotor location. Value of maximum deflection and natural frequency are within allowable limit.

Because of redesigning of shaft, Weight is reduced by around 8 kg which result in cost reduction.

Size of Alternator can reduce by eliminating landing bars. The advantage of making 4mm thick frame along with reduced size is one can directly weld foot to the barrel (frame). Barrel becomes more robust and strong.

X. REFERENCES

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