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Design & Analysis of Functional Evaluation System for Air Turbine

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***Abstract*—Energy associated with high speed air in certain regions is being utilized to generate electrical power with the aid of air turbines. Primary drawback of air turbine is the generation of electrical power with low speed and high torque. But this needs to be converted to high speed and low torque power so as to meet the conventional norms imposed by Indian electric power system using a gear system. Gear system will be installed at elevated height in an air turbine and its maintenance becomes tedious. In such cases, predictive maintenance is most preferred than general or breakdown maintenance and data supplied by the manufacturer is not sufficient to maintain the gear system.**

In this work, an air turbine is considered for functional evaluation of its gear system. A solid model of test rig representing all components and sub-systems is generated and analyzed using suitable software and dimensions of test rig are proposed. Using this test rig, the life of any air turbine can be predicted.

Keywords*— *Air turbine, electrical power, gear system, 3D CAD, ANSYS

I. 1. INTRODUCTION

An air turbine as shown in fig. 1 is a device that converts kinetic energy from the air into electrical power. Air mills are manufactured in a wide range of vertical and horizontal axis types. The turbines which are small in size are used for battery charging, auxiliary power for boats, traffic warning signals large turbines are used for domestic power supply and can be connected to grid. Air farms containing large turbines have become important source of renewable energy in order to reduce the dependency on fossil fuels

Fig. 1. Air Turbine.

II. LITERATURE REVIEW

With a service life of about 20 years, wind turbine power trains are subjected to a very diverse spectrum of dynamic loads. Due to the high number of load cycles which occur during the life of the turbine, fatigue considerations are of particular importance in wind turbine design. A. Heege et. al. [1] proposed fatigue procedure relies on a complete mechatronical wind turbine model, which includes a detailed gearbox model, and load transients are extracted from the global model for each gearbox component and fatigue cycle counting is performed individually for each power train component. Mathematical analysis for the life of gears was carried out by Gagandeep Singh [2] and further determined the noise levels of gears by changing the material from C-45 to 19mncr5. In another work on gearbox reliability, Musial and S. Butterfield [3] proposed to develop several tools such as multi-body dynamic analysis to model wind turbine loading coupled to internal loading, deformations of gearbox and test configuration. James F. Manwell et. al. [4] analyzed the design of gearbox in ESI-80 wind turbine based on computer codes and machine design methods. They focused on the failure of specific component in the integrated planetary gearbox

Performance evaluation was carried out by Hyoungwoo Lee and Kibong Han [5] which includes vibration analysis, noise analysis, strength of gear teeth, roller bearing life, and journal bearing design as per API standard. Physical configuration and material selection of the planetary gearbox design was conducted by Jeffrey Mobley [6] using different material and material processes and optimized to suit to light weight applications. Further he validated the calculation methods of gearbox load ratings and expected life.

MTBF, a frequently used value to quantify reliability of electronic component and systems of an air turbine. Two different methods were proposed to determine MTBF by Dr. Gerhard G and Antony Neugart [7]. Timothy Krantz and Brian Tufts [8] proposed a new designed alloy for improved properties of steel to improve the power density of gearing. Kevin Harrington et. al. [9] focused on the challenges of wind turbine lubrication by using advanced synthetic oils in the wind turbine gearbox and discussed the lubrication of gearing along with other contributing factors such as variable load, speed, temperature, bearing failure and downtime of gearbox. Jiri Tuma [10] reviewed practical techniques and procedures employed to quiet gearboxes and transmission units. He proposed a new enclosure to solve gear noise problem by improving the gear design.

All the research works are based on the modifications and improvements of wind turbines. Moreover, the relevant and reliable information is not available to predict the useful life of wind turbine. Based on these observations, there is a great need of a system which evaluates functionality of turbine. Therefore, in the present work, it is proposed to design a test rig which is useful to evaluate the functionality of the wind turbine. The wear and tear to which turbine is subjected when it is installed in the real time environment can be predicted and subsequently useful life of turbine.

III. DESIGN OF FUNCTIONAL EVALUATION SYSTEM

Based on the design philosophy a configuration is identified. All the subsystems of the configuration are to be designed taking functional loads into account. While designing the subsystems various mechanical design aspects are considered. The outcome of the structural design would be solid models of all subsystems of the intended system. The total design process is concluded with mention of details for the design, which will be ascertained further by finite element method. The objective of the project is to design a functional evaluation system for air turbine which means a system simulating air turbine at input of the gear box. Complexity involved in this requirement is that air turbine will not run at constant speed all the times rather it varies from very low to high value. To simulate this situation in ground level (Before the gear box is attached to air turbine) a drive is required which can feed motion to gear box. Only possibility of achieving this is through use of DC motor as such drive as speed needs to be varied. But to run the gear box, motor with very high energy (Power) is required particularly at high speed which makes the intended functional evaluation system expensive. Hence an additional means is required between motor and gear box which can substantiate for high energy requirement at high speed so as to accomplish the task with motor with low energy capacity. Hence this paper basically aims at design of such device which can boost up the enrgy requirement with a feature of variable moment of inertia with varying speed. Proposed design with all subsystems is shown in figure 2.

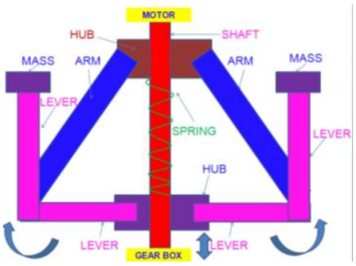

Fig. 2. Proposed Test Rig

As shown in figure 2 the proposed design consists of a shaft which will be connected to two hubs on extreme ends. Top hub will be connected to shaft rigidly whereas bottom hub will be connected to shaft such a way that it rotates along with the shaft but it can slide up and down relative to shaft. A spring will be connected between top and bottom hubs. Two arms connects top hub to levers through pivot which will be extended from bottom hub. Lever will be in L- shape bottom part of which will be connected to bottom hub and top part will house a mass positioned on its top. This lever can pivot such that mass comes close or goes away from shaft axis.

A. *PRINCIPLE OF WORKING*

Working principle of the proposed design is described in two phases as given below.

i. *When gear box has to be operated at high speed*

In this phase as shown in fig.3 first shaft speed & hence angular velocity increases, centrifugal force dominates spring force , spring compresses, arms rotates out through levers, masses will be pulled up (Through angular motion), bottom hub moves up towards top hub, moment of inertia increases, energy increases.

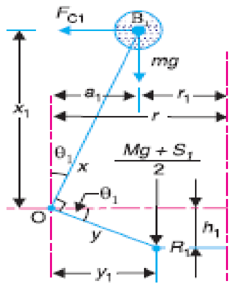

Fig. 3. First phase

ii. When gear box has to be operated at low speed

In this phase as shown in fig.4 first, motor speed reduces, angular velocity is less, centrifugal force is less than the spring force, bottom hub will be pushed down in tension by spring, masses are close to shaft.

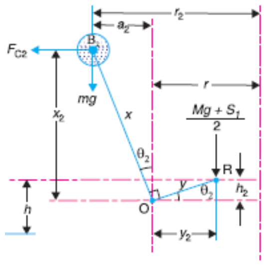

Figure.3. second phase

B. *DESIGN INPUTS*

In this work, air turbine HQ1650 is considered. The Manufacturer's data is, minimum speed =11RPM, maximum speed = 20 RPM, rated speed = 18 RPM, power of rotor =1.65 MW, time taken to attain maximum speed =0.001 sec,, mass positioned on lever = 65 Kg, shear modulus of spring material (Steel) = 84 GPa, spring index = 20, shear strength of shaft material = 133 N/mm^2, number of arms = 2, bending strength of arm = 316 MPa,

C. *DESIGN OF CONFIGURATION*

Minimum angular velocity can be expressed as follows

$$\omega_1 = \frac{2\,\pi\,N_1}{60}$$

In which

N_1: Minimum speed = 11 RPM

Substituting this we get ω_1 = 1.1519 rad/sec

In the same equation substituting N_2 = Maximum speed = 20 RPM

We get maximum angular velocity ω_2 = 2.09 rad/sec

We know that

$$\text{Power, } P = \frac{2\,\pi\,NT}{60}$$

In which
P: Power = 1.65 MW = $1.65x10^6$ W
N: Speed
T: Torque
In the above equation by substituting N_1 we get maximum torque,
$T_2 = 1.43x10^6$ N-m
And by substituting N_2 we get minimum torque,
$T_1 = 7.878x10^5$ N-m
Angular acceleration can be expressed as

$$\alpha = \frac{\omega_2 - \omega_1}{t}$$

Where
t: Time taken to attain maximum speed = 0.001 sec
From which
$\alpha = 942.5$ rad/sec^2
Torque can be expressed as
T = I x α
In which
I: Moment of inertia
In the above equation by substituting T_1 we get minimum moment of inertia,
$I_1 = 835.9$ Kg-m^2
And by substituting T_2 we get maximum moment of inertia,
$I_2 = 1519.8$ Kg-m^2
Moment of inertia can also be expressed as
I = m x r^2
Where
m: Mass positioned on lever = 65 Kg
By substituting I_1 we get
r_1: Minimum radius of rotation of mass as 3.5861 m
And by substituting I_2 we get
r_2: Maximum radius of rotation of mass as 4.83 m
Distance of pivot for lever from axis of rotation when the device in mid position can be expressed as

$$r = \frac{r_1 + r_2}{2}$$

From which r = 4.2 m
Energy that can be stored by the device can be expressed as

$$E = \frac{1}{2} I \omega^2$$

By substituting I_1 and ω_1 we get
E_1: Minimum energy as 554.6 Joules
And by substituting I_2 and ω_2 we get
E_2: Maximum energy as 3333.3 Joules
And increase in energy storage capacity by adopting variable moment of inertia feature in the proposed design is found to be
3333.33 – 554.6 = 2778 N
X: Length of the vertical part of the lever is taken as 1.0909 times that of r_1 which is found to be 3.91 m
Y: Length of the horizontal part of the lever is taken as 0.909 times that of r_1 which is found to be 3.26 m

From the decelerated position shown in figure 4 i.e. when the radius of rotation changes from r to r_1, the compression of spring or the lift of bottom hub, h_1 can be expressed as

$$\frac{h_1}{Y} = \frac{a_1}{X} = \frac{r - r_1}{X}$$

From the accelerated position shown in figure 3 i.e. when the radius of rotation changes from r to r_2, the compression of spring or the lift of bottom hub, h_2 can be expressed as

$$\frac{h_2}{Y} = \frac{a_2}{X} = \frac{r_2 - r}{X}$$

Combining both the equations we get

$$h = (r_2 - r_1)\frac{Y}{X}$$

From which we get h = 1.041 m
And

$$h_1 = \frac{h}{2} = 0.52\,m$$

Further for decelerated position
$a_1 = r - r_1 = 0.624$ m

$$X_1 = \sqrt{X^2 - a_1^2} = 3.86\,m$$

$$Y_1 = \sqrt{Y^2 - h_1^2} = 3.22\,m$$

Further for accelerated position
$a_2 = r_2 - r = 0.624$ m
$X_2 = X_1 = 3.86$ m
$Y_2 = Y_1 = 3.22$ m
Centrifugal force can be expressed as
$F_c = m\, r\, \omega^2$
In which substituting r_1 and ω_1 we get
Minimum centrifugal force F_{C1} as 309.3 N
And substituting r_2 and ω_2 we get
Maximum centrifugal force F_{C2} as 1378.7 N

For the decelerated position figure 4, taking moments about point O, we get

$$\frac{Mg + S_1}{2} y_1 = F_{c1} X_1 - m g a_1$$

In which
S_1: Spring force exerted on bottom hub at ω_1
(However 'M' is neglected)
Substituting all we get $S_1 = 494.8$ N

For the accelerated position shown in figure 3 taking moments about point O, we get

$$\frac{Mg + S_2}{2} y_2 = F_{c2} X_2 + m g a_2$$

In which
S_2: Spring force exerted on bottom hub at ω_2
(However 'M' is neglected)
Substituting all we get $S_2 = 3061.6$ N

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D.DESIGN OF SUBSYSTEMS:

From the figure 2 the following components are identified and the components are, spring, shaft, hub, arm, lever, mass, bearing. Standard mechanical engineering design procedure [ ] are implemented to design all sub systems

i. SPRING

As the spring will be subjected to cyclic loading i.e. for repeated operation it should be designed while taking fatigue into account. Helical springs subjected to fatigue loading are designed by using the Soderberg line method. Relation between factor of safety, variable stress and mean stress can be used as starting point for design of spring. Wire diameter, d = 25 mm. Mean coil diameter, D = 507 mm. Number of turns of the spring is 14,Assuming the ends of spring to be squared and ground, the total number of turns of the spring is 16, Maximum deflection of the spring be 1.83 m ,Free length of the spring=1.24 m, Pitch of the coil can be 122 mm. If the ratio between free length and mean coil diameter is > 4 buckling occurs. But for the present design buckling ratio is 3.61 and hence design is free from buckling

ii. SHAFT

As no bending moment is anticipated on shaft it is designed based on torque alone. Diameter of the shaft d_s = 380 mm.

iii. HUB

Normally outer diameter of the hub will be taken as *twice* the diameter of shaft. Hence outer diameter of hub = 2 x 380 = 760 mm, length of hub = 2 x Diameter of shaft = 2 x 380 = 760 mm, From this length of the shaft is estimated as = Length of spring + Length of top hub + Length of bottom hub + Diameter of shaft = 3734 mm, (Projection of shaft on either side of hubs is taken as half the diameter of shaft)

iv. ARM

Bending moment expressed by the arm can be expressed as Length of arm (Which is worked out to be 4 m from the configuration design which was already accomplished), n: Number of arms = 2, r: Outer radius of hub = 380 mm, M = 6.48×10^5 N-m , Z = 2.05×10^{-3} m^3, size of the arm (C/Sis taken as circular) d=275 mm

v. LEVER

Dimensions of lever are already worked out in configuration design as given below. X: Length of the vertical part of the lever = 3.91 m. Y: Length of the horizontal part of the lever = 3.26 m. Same cross section (Circular) evolved for arm is considered for lever also. Hence size of lever = 275 mm.

vi. MASS

To suit the mass chosen i.e. 65 Kg a steel disc of 400 mm dia. and 66 mm thick is considered as mass.

vii. BEARING

For the proposed design single row deep grove ball bearings are chosen as they are best suitable for taking both radial and thrust loads. Reaction on the bearing = Maximum load on spring = 3061 N.N speed=20 RPM, d: Diameter of bearing = 380 mm, D: Outer diameter of bearing = 480 mm, B: Axial width of bearing = 46 mm, No choice for number of balls

IV. STRUCTURAL ANALYSIS

Structural analysis is carried out for the solid model as shown in figure 6 against the self weight and functional loads i.e. due to movement of the system. Maximum Von Misses stress is obtained from the analysis. Maximum stress thus obtained is compared with allowable stress and obtained the available factor of safety.

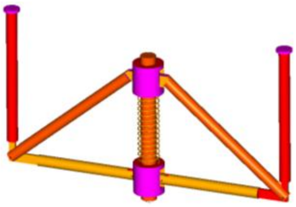

Fig .5. Solid model of the test rig.

A. CRITERIA

i. *Static analysis:* Minimum available factor of safety should be more than the desired factor of safety (1.5).

ii. *Modal analysis:* First natural frequency should be above the frequency associated with operating speed of rotor i.e. 0.33 Hz (20 RPM).

To begin with geometric model of the system is built in 3D CAD software from its dimensions evolved as an outcome in the previous chapter. However load bearing members are only considered for analysis. Then geometric model is converted into FE model by discretizing with elements in commercial FEM software package ANSYS. All the subsystems are discretized with Linear beam (BEAM4) elements.

B. MATERIAL PROPERTIES

Table1.Material Properties

Material	Young's modulus (E)	Poisson's ratio (μ)	Density (ρ)
Spring steel	2.1×10^{11} Pa	0.3	7850 Kg/m^3

BOUNDARY CONDITIONS:

Both extreme sides of shaft are constrained for all DOF except rotation about shaft axis.

C. LOAD:

Table 2. Loads

1. STATIC ANALYSIS
Self weight with 1g acceleration
2. STATIC ANALYSIS
• Angular velocity : 2.09 rad/sec (Calculated from maximum speed of rotor)
3. DYNAMIC ANALYSIS
Modal analysis

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FE model with boundary conditions is shown in figure

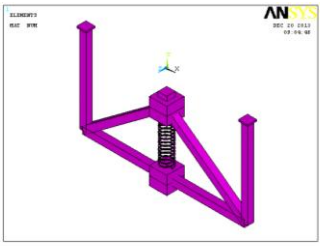


Fig. 6. FE model

D.Static Analysis – Self Weight

The FE model is then solved for Von Misses stress using ANSYS software. Maximum stress plot is shown in Figure in which maximum stress location is visible in red color.

Maximum Von misses stress = 67 MPa

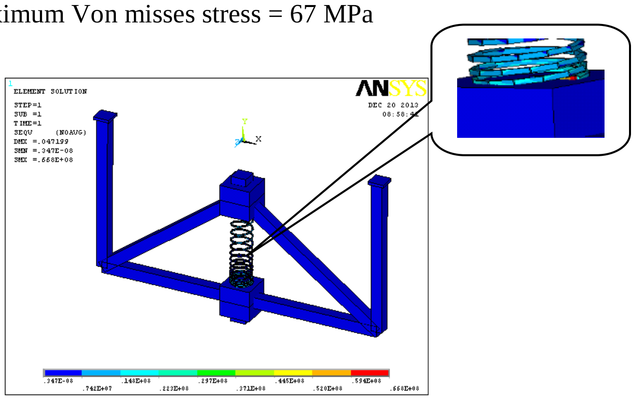


Fig. 7. Stress plot

i. Observations

Maximum Von Misses stress is observed to be 67 MPa. Available factor of safety is observed to be (>10) by comparing the maximum stress with that of allowable stress (Yield) of steel material i.e. 770 MPa. As the available factor of safety (>10) is more than minimum desired factor of safety (1.5) the design is safe.

E. STATIC ANALYSIS – SELF WEIGHT & FUNCTIONAL LOADS

Same FE model is then solved for Von Misses stress using ANSYS software. Maximum stress plot is shown in Fig.9 in which maximum stress location is visible in red color.

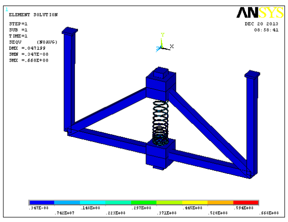


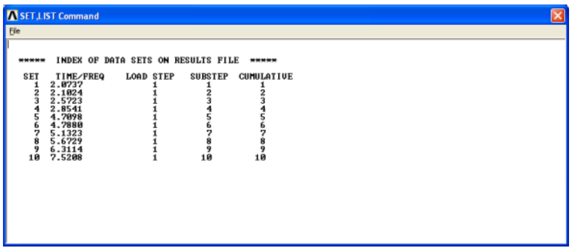
Fig. 8. Stress plot

i. Observations

Maximum Von Misses stress is observed to be 67 MPa. No change is noticed in the value of stress compared to that of analysis with self weight which conveys that influence of functional load i.e. angular velocity is negligible compared to that of self weight. Available factor of safety is observed to be (>10) by comparing the maximum stress with that of allowable stress (Yield) of steel material i.e. 770 MPa. As the available factor of safety (>10) is more than minimum desired factor of safety (1.5) the design is safe.

F. DYNAMIC (MODAL) ANALYSIS

Modal analysis is the study of the dynamic properties of structures under vibration excitation. Modal analysis is the field of measuring and analyzing the dynamic response of structures and or fluids when excited by an input. In structural engineering, modal analysis uses a structure's overall mass and stiffness to find the various periods that it will naturally resonate at. If a structure's natural frequency matches an earthquake's frequency, the structure could continue to resonate and experience structural damage. A modal analysis calculates the undamped natural modes of a system. These modes are given in decreasing order of period and are numbered starting from 1.The analysis calculates the natural modes of the discretised model, not those of the real continuous system. Same FE model used for static analysis is extended for modal analysis. The list of frequencies is given below.



SET,LIST Command

***** INDEX OF DATA SETS ON RESULTS FILE *****

SET	TIME/FREQ	LOAD STEP	SUBSTEP	CUMULATIVE
1	2.0737	1	1	1
2	2.1024	1	2	2
3	2.5723	1	3	3
4	2.8541	1	4	4
5	4.7098	1	5	5
6	4.7880	1	6	6
7	5.1323	1	7	7
8	5.6729	1	8	8
9	6.3114	1	9	9
10	7.5208	1	10	10

Fig.9. List Frequencies

First bending mode shape is shown in Fig. 11.

First bending mode: Frequency = 2 Hz

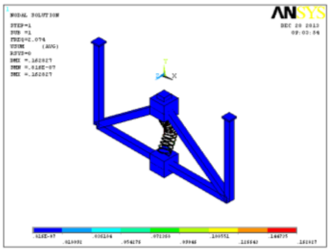


Fig.10 First mode shape

ii. Observations

Frequency of the intended system corresponding to first bending mode is found to be 2 Hz. First natural frequency (2 Hz) is much above the frequency associated with operating speed of rotor i.e. 0.33 Hz (20 RPM). Hence system doesn’t experience resonance

V. RESULTS AND DISCUSSION

A. SUMMARY OF DESIGN CALCULATIONS

Table.3. Summary of design parameters pertaining to configuration

Sl.No	Design Parameter	Value
	Configuration	
1.	Minimum Angular Velocity	1.159rad/sec
2.	Maximum Angular Velocity	2.09rad/sec
3.	Minimum Torque	7.878×10^5 N-m
4.	Maximum Torque	1.43×10^6 N-m
5.	Angular Acceleration	942.5 rad/sec^2
6.	Min Moment of Inertia	835.9 kg-m^2
7.	Max Moment of Inertia	1519.8 kg-m^2
8.	Min Radius of Gyration	3.5861 m
9.	Max Radius of Gyration	4.83 m
10.	Distance of pivot for lever from axis of	4.2 m
11.	Min energy that can stored by the	554.6 joules
12.	Max energy that can stored by the	3333.3 joules
13.	Increase in energy storage capacity	2778 N
14.	Length of the vertical part of the lever	3.91 m
15.	Length of the horizontal part of the lever	3.26 m
16.	Compresion of spring	1.041m
17.	Min centrifugal force	309.3 N
18.	Max centrifugal force	1378.7 N
19.	Minimum spring force exerted on bottom	494.8 N
20.	Maximum spring force exerted on	3061.6 N

The design parameter pertaining to subsystems are summarized in table 4.

Sl.No	Design Parameter	Value
Spring		
1.	Spring Index	20
2.	Wahl's stress factor	1.07
3.	Wire diameter	25 mm
4.	Mean coil diameter	507 mm
5.	Number of turns	16
6.	Stiffnes	2465.6 N
7.	Max. Deflection	1.24m
8.	Free Length	1.83 m
9.	Pitch of the Coil	122 m
Shaft		
1.	Diameter	380 mm
2.	Length	3734 mm
Hub		
1.	Outer Diameter	380 mm
2.	Inner diameter	760 mm
3.	Length	760 mm
Arm		
1.	Bending Moment	6.48x105 N-m
2.	Section Modulus	2.08x10-3 m3
3.	Cross-Section	Circular
4.	Size	275 mm
5.	Length	4 m
Lever		
1.	Cross-Section	Circular
2.	Length of the vertical Part	3.91 m
3.	Length of the horizontal Part	3.26 m
Mass		
1.	Mass	65 kg
2.	Diameter	400 mm
3.	Thickness	66 mm
4.	Material	Steel
Bearing		
1.	Outer Diameter	480 mm
2.	Width	46 mm
3.	Designation	61876 MA

B. SUMMARY OF FE ANALYSIS

i. STATIC ANALYSIS:

a. Allowable stress =770 MPa
b. Factor of Safety >10.

ii. Maximum stress = 67 MPa (Self Weight).

a. Maximum Stress=67MPa (Functional Load)

iii. DYNAMIC ANALYSIS:

a. Critical frequency = 0.33 Hz
b. First natural frequency=2 Hz

VI. CONCLUSIONS & RECOMMENDATIONS

Design of a functional evaluation system for air turbine is done which provides variable moment of inertia which enhances energy storage capacity of motor.

A.Static analysis – Self weight: Maximum Von Misses stress is observed to be 67 MPa. Available factor of safety is observed to be (>10) by comparing the maximum stress with that of allowable stress (Yield) of steel material i.e. 770 MPa. As the available factor of safety (>10) is more than minimum desired factor of safety (1.5) the design is safe.

B.Static analysis – Functional load: Maximum Von Misses stress is observed to be 67 MPa.No change is noticed in the value of stress compared to that of analysis with self weight which conveys that influence of functional load i.e. angular velocity is negligible compared to that of self weight. Available factor of safety is observed to be (>10) by comparing the maximum stress with that of allowable stress (Yield) of steel material i.e. 770 MPa. As the available factor of safety (>10) is more than minimum desired factor of safety (1.5) the design is safe.

C.Modal analysis: Frequency of the intended system corresponding to first bending mode is found to be 2 Hz. First natural frequency (2 Hz) is much above the frequency associated with operating speed of rotor i.e. 0.33 Hz (20 RPM). Hence system doesn't experience resonance.

D.RECOMMENDATIONS: It is recommended to incorporate the functional evaluation system for testing gear box of air turbine proposed in this project and enhance the energy storage capacity of the motor. It is further recommended to extend the design for higher capacity air turbines.

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