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ANALYSIS OF TURBINE ROTOR RADIUS AND WING LENGTH IN TURBINE VIBRATION

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ABSTRACT
A vibration criterion is one of the most important factors for machine components working in heavy duty operations. It is well known that mechanical vibrations influenced machine components are continuously under mechanical loadings such as tensile, compress, bending, torsion and shear. Therefore the machine components affected by variable mechanical loadings produce fatigue compare to single mechanical loaded machine parts. In the present study, four wings turbine model was selected as machine component and turbine natural frequency was calculated by using ANSYS computer program. There different materials (steel, chromium and nickel) were selected as turbine materials. Moreover, ratios of turbine rotor radius (r) to wing length (h) like r/h = 1.25, 1.50, 1.75 and 2 have been investigated and analyzed.
Keywords: Turbine blade, natural frequency, modal analysis

1. INTRODUCTION
The major problem of wing turbine is bending and fatigue. It is known that the rotational force and movement of an unbalanced mass are generating vibrations transmitted to the construction elements mainly through the bearing points of the load. The disturbing frequency is equal to the rotation frequency of the turbine wing and the vibrating equipment. The calculation error of natural frequency can be the assumption of isotropy and the design natural frequency should be larger than the working frequency to prevent resonance contending to by Bo et al. [1]. A method of determining equivalent viscous damping ratio for different rotational speeds and modes as a function of displacement or strain at a reference point in a blade has been presented by Rao [2]. Lazan showed that the logarithmic decrement values increase with dynamic stress, with vibration amplitude, where material damping is the dominant mechanism. Damping plays an important role in determining this stress value accurately and experimentally [3]. Materials with lower density such as fiber aramid (Tecnora) have higher natural frequency and bigger deflection. The comparison of fiber glass S-type and fiber glass E-type shows increase frequency without eigenvalue change [4]. Euler-Bernoulli and Timoshenko theories for transverse vibration, in order to predict the natural frequencies of the structure made with the predictions of a finite element analysis of the complete structure, which is taken as the benchmark for accuracy [5]. The combination method of finite element analysis and optimum design has been presented in this work to obtain the elastic constants of measured by vibration testing. Under suitable selection of parameters in the present combination method, it is proved to be a fast and accurate method [6].

2. FREE VIBRATION-UNDAMPED
Free vibrations occur in a system in the absence of any external excitation as a result of a kinetic energy or potential energy initially present in the system. These vibrations are oscillations about one

of the system's static-equilibrium positions. The differential equation for free response of an undamped (conservative) system of order n is written as

$$m\ddot{x}. + kx = 0 \tag{1}$$

where m and k are coefficients specific to the system determined during the derivation of the differential equation. If one tries a solution of the form for equation 1, $x = \phi_i.e^{j.\omega_i.t}$, ϕ_i and ω_i must satisfy the eigenvalue problem

$$(K - \omega_i^2 M)\phi_i = 0 \tag{2}$$

because M and K symmetric, K is positive semi definite and M is positive definite, the eigenvalue ω_i^2 must be real and non negative. ω_i is the natural frequency and ϕ_i is the corresponding mode shape; the number of modes is equal to the number of degrees of freedom, n. Note that Equ.(2) defines only the shape, but not the amplitude of the mode which can be scaled arbitrarily. The modes are usually ordered by increasing frequencies $(\omega_1 \leq \omega_2 \leq \omega_3 \leq ...)$ [7]. From Equ.(2), we see that if the structure is released from initial conditions $x(0) = \phi_i$ and $\dot{x}(0) = 0$, it oscillates at the frequency ω_i according to $x(t) = \phi_i \cos \omega_i t$, always keeping the shape of mode i.

3. MATERIALS AND METHOD

Turbine blade model was occurred with ANSYS package programme and its shape was in Figure 1.

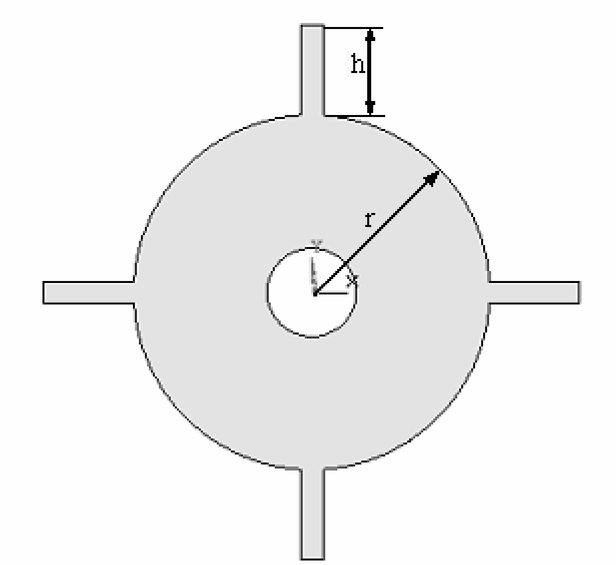

Figure 1. Turbine blade model shape

Three different material types were taken for turbine blade analysis. The properties of selected materials were shown in Table 1.

Table 1.The mechanical properties of materials

Material	Young Modulus (Pa)	Poisson Ratios	Density (kg/m^3)
Chrome	2.79e11	0.21	7.10-e3
Nickel	2.14e11	0.31	8.88-e3
Steel	2.11e11	0.29	7.87e-3

4. FINITE ELEMENT METHOD

Finite element calculations were performed by the commercial package of ANSYS 11. Shell63 element type was chosen for the analysis. SHELL63 has both bending and membrane capabilities. Both in-plane and normal loads are permitted. The element has six degrees of freedom at each node: translations in the nodal x, y, and z directions and rotations about the nodal x, y, and z axes figure 2. Stress stiffening and large deflection capabilities are included. A consistent tangent stiffness matrix option is available for use in large deflection (finite rotation) analyses.

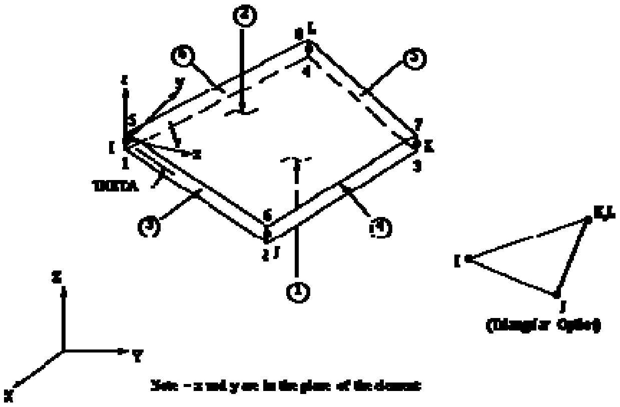

Figure 2. SHELL63 Elastic Shell

The obtained node and element number using finite element method according to r/h ratios were shown in table 2.

Table 2. Nodes and element numbers according to shell63 element type

r/h	1.25	1.5	1.75	2
Nodes	1194	1234	1262	1345
Elements	1076	1126	1157	1244

5. MODAL ANALYSIS

The first step in dynamic analysis is modal analysis. Modal analysis is an analysis method which does not consider nonlinear parameters such as plasticity, connection elements, etc. Natural frequencies and mode shapes of elements which constrained dynamically are obtained. Obtained modal parameters may exhibit differences according to material properties [8].

The following assumptions are made during modal analysis.

1. The structure has constant hardness and mass.
2. Damped is not existed, unless damped solution method is selected.
3. Time dependent forces are not included in the model. Displacements, pressures and temperatures are not applied. Natural frequencies belong to first 10 modes are achieved by modeling turbine blade, in table 3. First 10 natural frequencies of models are between 10.989-71.644 Hz. In figure 3, for steel materials and r/h=2, first mode and fifth mode are illustrated.

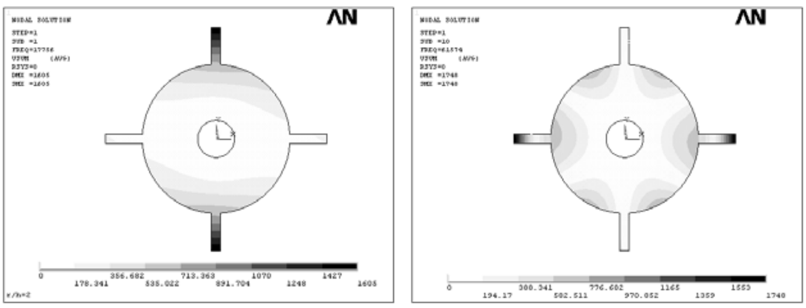

Figure 3. First and tenth mode shape for turbine blade

Table 3. Natural frequency of turbine blade

Material	Steel				Nickel				Chrome			
Mode No	٢	1.٧5	1.5	1.٢٥	٢	1.٧5	1.5	1.٢٥	٢	1.٧5	1.5	1.٢٥
1	17756	16434	14525	12099	16199	14986	13238	10989	20171	18725	16627	14031
2	17761	16442	14531	12195	16204	14994	13243	11107	20177	18736	16635	14039
3	18647	16965	14640	12201	17042	15405	13295	11113	20830	19355	17175	14283
4	18792	17261	15218	12699	17065	15767	13890	11580	22229	20069	17308	14446
5	28813	28807	28788	27444	26184	26178	26161	24996	33866	33859	33837	31624
6	37136	33802	30326	27460	33731	30726	27595	25011	43791	39604	35234	31644
7	37137	33816	30340	28146	33732	30739	27607	25655	43792	39621	35251	32229
8	40797	35827	31409	28823	37074	32583	28596	26193	47885	41797	36319	33880
9	53067	46250	40075	35085	48290	42088	36468	31922	61511	53589	46455	40737
10	61574	57553	54192	51345	56008	52333	49258	46652	71644	67161	63466	60340

6. CONCLUSIONS

- Although, nickel and steel young modulus and poisson ratios are near each other, natural frequency of nickel has obtained greater than natural frequency of steel in all modes. The reason for this is that the density of nickel is higher than that of steel.
- In analysis for all materials, chrome which has the lowest density and highest young modulus has been observed to have the highest natural frequency in all modes.
- As the blade length gets smaller, the natural frequency gets higher in all modes for three materials.
- Young modulus and mass have been observed to be the parameters affecting the natural frequency.
- For all selected material, the frequency has increased as the mode numbers rises.
- Natural frequencies are observed to be near each other in fifth mode for all values of r/h.
- Natural frequencies in sixth and seventh modes have changed very little.

7. REFERENCES

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