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Design of Transmission Housing

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Nomenclature

g acceleration due to gravity

l length of the rotor between any two rotors

q torsional stiffness = GI_p/l

u,v,w displacement in the three coordinates x displacement in axial direction

y shape function

E Young's modulus ($E=2.1\times10^5$ N/mm^2 *for steel*)

G modulus of rigidity I second moment of area

I_p *polar moment of inertia of the shaft*

J mass moment of inertia of the rotor

W_i *lateral deflection of the rotors*

Y displacement in lateral direction μ *weight per unit length of the beam* ω *angular frequency*

θ *angular displacement of rotor (torsional vibration)*

θ_x *rotation about x axis (for the plate)*

θy *rotation about y axis (for the plate)*

Abstract: The purpose of this paper is to design and analyse a two stage speed reduction gearbox housing. There has been hardly any rigorous design procedure adopted for gearbox housing during the design of a gearbox. This has led to a large number of failures in such systems and hence the need for a definite design procedure has risen. The gear train and the shafts were designed in accordance with standard design data. The gear train is theoretically mounted in a cuboidal box termed as the housing. The torsional critical speed of the rotor system and the lateral critical speeds of the shafts were obtained using ANSYS. A finite element modelling of the gearbox housing was carried out in ANSYS and the same was analysed to obtain its natural frequency and stress distribution. All the ANSYS solutions were extensively evaluated with different theoretical methods such as the Raleigh's energy method and were found to be consistent. A comparison of the natural frequency of the housing, the torsional critical speed of rotor system and the lateral critical speeds of each of the shafts with the exciting frequency clearly showed the former were far above the exciting frequency hence qualifying the gear box housing. A few industrial practices were taken up as case studies to understand their

implications on the safety of the gearbox. The degree to which the gearbox stability is forfeit due to the changes made in each of the three case studies is compared.

Key Words: Gearbox Housing; Critical Speed; Finite Element Method analysis concluded that the housing design plays a very significant role in the industrial scenario.

I. Introduction

A gearbox housing has certain functional requirements based on which it was designed through the years. There has been hardly any rigorous design procedure adopted for gear housing during the design of a gearbox. Owing to a large number of failures in such designs the need for a definite design procedure has risen. In this paper, the design of gearbox housing is attempted. The gear housing is first designed based on functional requirements considering the system to be statically indeterminate. The modal analysis is performed. In order to arrive at the optimized design, the factor that is of prime importance is the natural frequency of the configuration. It is necessary that it should lie sufficiently away from the exciting frequencies that arise from the gearbox, the exciting frequencies are mainly the lowest the torsional critical and lateral critical speed of the rotating shafts.

Ramamurti et al. [1] presented a design methodology for two-speed gearboxes using FEM and compared the results with the classical methodology.

Kostiæ and Ognjanoviæ [2] determined the structure of noise emitted by the gearbox walls and established the correlation between the housing vibrations and the noise emitted. Lee et al.

[3] studied the coupled vibration characteristics of a turbo-chiller rotor bearing system having a bull-pinion speed increasing gear, using a coupled lateral and torsional vibration finite element model of a gear pair, and to provide the mechanism of the characteristic changes.

Lim and Singh [4] discussed the analytical and experimental methodologies used for bearing dynamics, housing vibration and noise, mounts and suspensions and the overall gear and housing system. They have also proposed dynamic finite element analysis of real gearbox housing when both rigid and flexible mounting conditions are considered.

[5]. Ramamurti et al. [6] observed a decrease in the stress and displacement levels with an increase in thickness of the gearbox casing in both the fabricated and cast constructed gear boxes.

Choy et al. [7] developed a comprehensive procedure to combine the dynamics of the rotor-gear system with vibrations of the gearbox structure to determine the global system response.

Zang et al. [8] indirectly determined the generalized forces applied on a gearbox using numerical and experimental approaches.

Kostiæ and Ognjanoviæ [9] observed the importance of modal oscillation of gearbox housing walls and other elastic structures for the noise emitted by systems into the surroundings. Quite a number of publications on stress and dynamic analysis has been reported in the recent past [10-11].

II. Methodology

The methodology adopted for the design of gearbox housing involves the following stages: • Design of a two stage speed reduction gearbox • Determination of torsional critical speed of the gearbox • Computation of the first few lateral critical speeds of the shafts • Determination of the natural frequency of this configuration to make sure that the exciting frequency is sufficiently away from the natural frequency • To perform the static analysis of the gearbox.

III. Design of Gearbox

A two stage speed reduction gearbox was designed for an input power of 6kW at a running speed of 1440 rpm (This induction motor is common to most plate working and plate cutting machine tools in industry). The gears are all designed as spur gears. The reduction in the first stage is 5 and the reduction in the second stage is 4. Hence a net reduction of 20 has been achieved leaving the speed of the low speed end shaft as 72 rpm. The module for the first set of gears is 3mm and that for the second set of gears is 5mm. Based on the diameter of the gears and taking into consideration a certain allowance, a housing dimension of 700*450*200 been be achieved. A 6mm plate has been assumed to make the gearbox housing and the analysis is performed.

IV. Determination of Torsional Critical Speed:

During the determination of the torsional critical speed, the gearbox system from the motor to the pump is being treated as a single entity. The right end of the first shaft, the second shaft and the left end of the third shaft do not take any load and hence shall not be included in the torsional model.

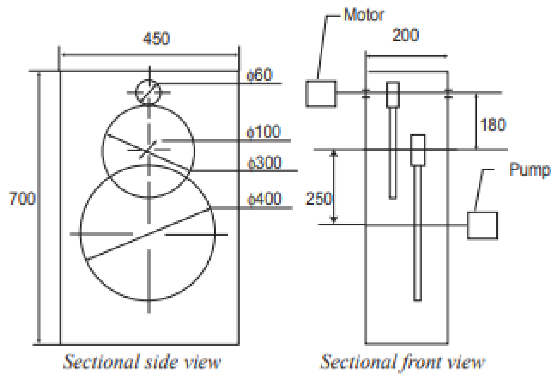

Fig.1: Final layout of the gearbox

Reducing the gearbox configuration to a torsional model as below with the high speed end as the reference shaft, the torsional critical speed of the system has been determined using ANSYS and verified subsequently using Rayleigh's energy method. For Rayleigh's energy method, the kinetic energy is given by

$$T= (\omega^2 /2)\ \Sigma J\theta^2 \qquad ...(1)$$

The strain energy by $U = \Sigma q\ (\theta_i{}^2 – \theta_{(i-1)})^2$ ...(2)

The procedure for using Rayleigh's method for torsion problems is illustrated in Example (5.2) of [4]. While modeling in ANSYS, the element chosen is a 3D beam. A minimum of one element is created for each diameter of the shaft and hence the FEA model of the torsional system has 8 nodes. All degrees of freedom except the rotation in the axial direction are constrained at the bearings. Each node has 1 degree of freedom and hence the system reduces to an 8 degree of freedom problem. A modal analysis is performed on the model and the fundamental frequency is recorded as the torsional critical speed. A torsional critical speed of 238 cps is arrived at using ANSYS and this value is found to be in close agreement with 232cps obtained using Energy method.

V. Determination of Lateral Critical Speed.

The basic assumption made in lateral critical speed associated with gear drives is that the contact between any gear pair does not influence the lateral stiffness. This leads to the procedure of computing the lateral critical speeds of the three individual shafts on two bearings treated as 2D beams each with masses located on them. The type and location of anti-friction bearing and the masses of the pinions and gears influence the critical speed irrespective of the speeds of the shafts.

B. ANSYS Modelling:

The pinion on one shaft and the corresponding meshing gear on the other have a line contact only, and hence it is assumed that one does not influence the other. Thus the shafts can be treated as separate entities. The lateral critical speed of each shaft is determined by modal analysis.

Constraints given at the bearing points: $u = v = w = 0$ $\theta_x = \theta_y = 0$

Constraints given at the rest of the nodes: $u = w = 0$ $\theta x = \theta y = 0$

VI. Gearbox Housing

For the analysis of the housing, ANSYS was chosen as the analysis software. The housing dimensions are 700 x 450 x 200 mm. Shell63 (4-noded thin shell) has been used since it resembles the required plate element (a feature which is classified under shells in ANSYS). Coarse mesh has been utilized here to avoid unnecessary rendition time. The thickness is set to 6mm. Linear elastic Isotropic material model is used.

A. Static Analysis:

A static analysis to calculate the effect of steady loading conditions on a structure was performed. A static analysis can, however, include time-varying loads that can be approximated as static equivalent loads (such as the static equivalent wind load). Static analysis determines the displacements, stresses, strains, and forces in structures or components caused by loads that do not induce significant inertia and damping effects. Steady loading and response conditions are assumed; that is, the loads and the structure's response are assumed to vary slowly with respect to time. The average stress in the vertical direction is 1.25MPa treating the bottom most cross section of the gear box to experience pure compressive stress due to the dead load above and 2MPa using ANSYS. Also, from ANSYS, the maximum stress on the housing is 11.6 MPa. Since the allowable stress of mild steel with a factor of safety of 3 is 35MPa, the design of the housing is safe.

B. Natural Frequency of the Housing

The natural frequency is that frequency with which an object vibrates independent of any and all external forces. For practical units this has been found to be ranging from as low as 1 cps (in passenger cars) to as high as 700 cps in (the case of short blades of high pressure steam turbines). When the exciting frequency is equal

Table 1: Lateral critical speed of shafts

Parameter		Frequemcy, cps	
		FE method	Energy method
Lateral	Shaft 1	1390	1256
Critical	Shaft 2	557	551
Speed	Shaft 3	607	472

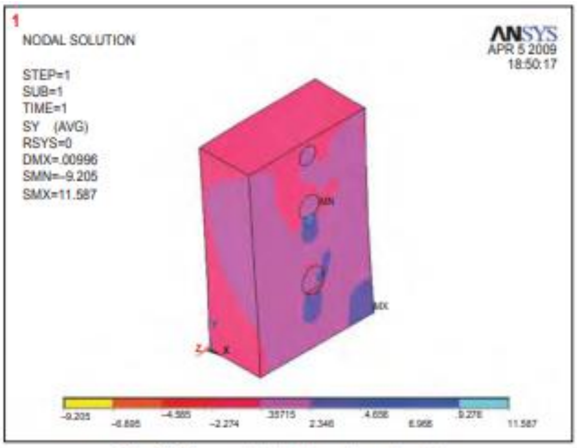

Fig.2: Stress distribution in y direction

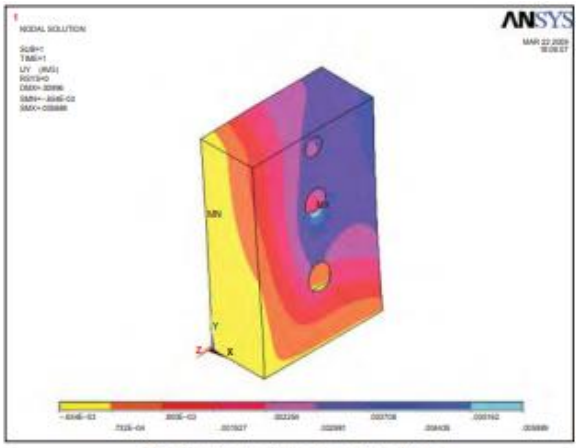

Fig.3: Mode shape in y direction

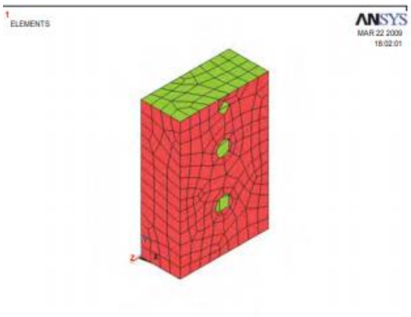

Fig.4: ANSYS model of the housing

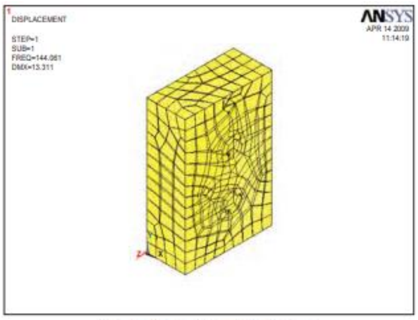

Fig.5: Mode shape of the housing

to the natural frequency, the phenomenon of resonance occurs. Thus in order to prevent resonance, we will calculate the natural frequency and confirm that the excitation frequencies are nowhere in this range. In the likely case that they are, necessary alterations to structural design would have to be made. Natural Frequency Determination by ANSYS The concentrated weights of the transmission elements on each of the shafts are treated as lumped masses (structural mass- MASS 21 in ANSYS) distributed at the bearing ends in the proportion of their distance from the bearing ends. A natural frequency of 144 cps was obtained using ANSYS.

VII. Case Studies

The analyses performed till now assumed independent bases for the motor and pump and also assumed the base of the housing to be fixed to the ground. But in the industrial scenario, these assumptions are often violated keeping in mind either functional requirements or for cost reduction. The effects of few such violations are studied in this paper.

Case 1: The motor and pump are assumed to be flange mounted.

Case 2: The housing as a whole is placed on an empty box in order to attain a desired elevation.

Case 3: Two sets of alternate sides of the elevating box are removed .

A. Case 1

Flange-mounted Motor on Shaft 1

A flange mounted motor is assumed to be fitted to shaft 1. The weight of the motor is very high comp red to the combined weight of the corresponding shaft and the other rotors placed on it. This system becomes an overhanging beam problem with a heavy weight at the overhanging end. The lateral critical speed of the new system is computed. A drastic reduction in the critical speed can be noted due to an increase in the mass of the system. Computation using ANSYS shows that the lateral crtical speed of the new system drops to 1827 rpm compared to its actual lateral critical speed (83400 rpm). This new lateral critical speed is very close to the exciting speed (1440 rpm).

Pump Mounted on Shaft 3

A pump is assumed to be fitted to shaft 3.

The weight of the pump is very high compared to the combined weight of the corresponding shaft and the other rotors placed on it. This system becomes an overhanging beam problem with a heavy weight at the overhanging end. The lateral critical speed of the new system is computed. A drastic reduction in the critical speed can be noted due to an increase in the mass of the system. Computation using ANSYS shows that the lateral crtical speed of the new system drops to 7140 rpm compared to its actual lateral critical speed (36420 rpm).

B. Case 2

The housing as a whole is placed on an empty box

The whole housing is assumed to be placed over anempty box with the same dimensions as that of the housing ie. 450 ×200×700 mm and a thickness of 6 mm.The total mass of the housing is taken as 900N and acts on the empty box with an area of cross section 450×200 mm^2 . The modal analysis is done on the entire system and a natural frequency of 99.6 cps is achieved. It is to be noted that there is a fall in the natural frequency of the system from 144 cps to around 100 cps because of the elevation. The mode shape of the empty box is shown in the Fig. 6.

C. Case 3

Two sets of smaller sides of the elevating box removed

The empty box on which the gearbox housing is placed is assumed to be with two rectangular cut outs. The mass on the top of the empty box is 900N. Fig.7 shows the empty box with two sides not present along with the housing placed on top of it. The modal analysis is done and the value of natural frequency is obtained as 7 cps. The mode shape is shown in fig.8.

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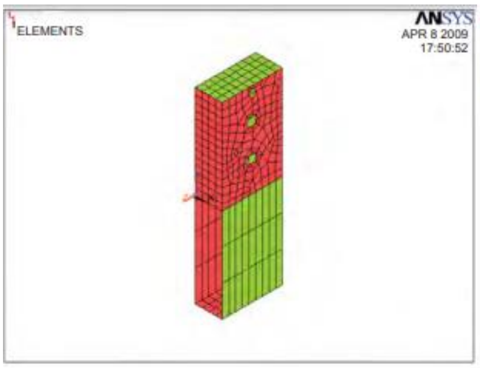

Fig. 7: The empty box with two smaller sides removed

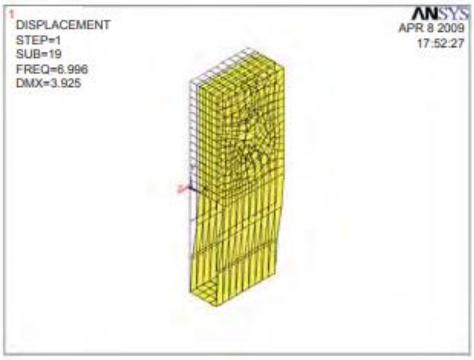

Fig.8: Mode shape when the two smaller sides removed

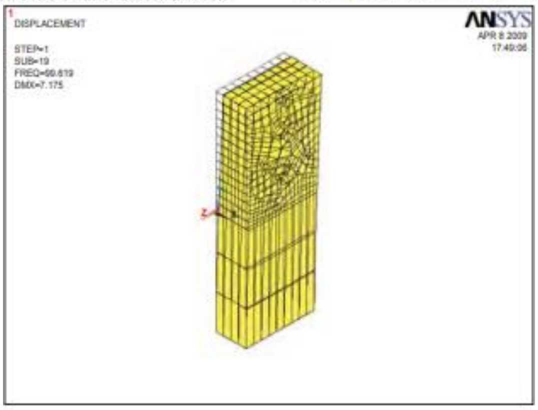

Fig.6: Mode shape of the empty box

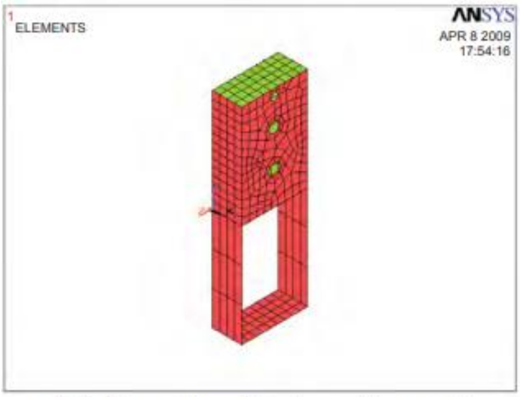

Fig.9: The empty box with two larger sides removed

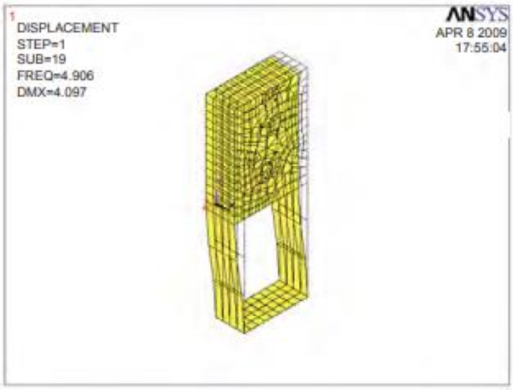

Fig.10: Mode shape when the two larger sides removed

Two sets of sides of the elevating box removed The empty box on which the gearbox housing is rested is assumed to be without the two faces. The mass on the top of the empty box is 900N. Fig. 9 shows the empty box with two larger sides not present with the housing placed on top of it. The modal analysis is done and the value of natural frequency is obtained as 5 cps. The deformed shape is shown in Fig.10.

Table 2: ANSYS results of the complete analysis

Parameter		Frequency, cps
Exciting Frequency		24
Torsional Critical Speed		238
Lateral Critical Speed	Shaft 1	1390
	Shaft 2	557
	Shaft 3	607
Natural Frequency Of Housing	144	

VIII. Results :

A gearbox was first designed and then a complete analysis of the gearbox housing was performed. This analysis included static analysis, natural frequency determination of the housing, torsional and lateral critical speed determination. The results of the analyses performed using ANSYS has been summarized as follows: The common practices followed in industries in view to meet the functional requirements (elevation) or reduction in material and hence their costs have been taken up as case studies and the reasons for their failure have been analysed. The results of their analyses are given in Table 3: of dynamic analysis of gear housing. It has been observed that many transmission housings fail due to exciting frequencies being very close to either the natural frequency of the housings or the critical speeds of the rotors housed inside. Even though the example chosen is a small size gearbox with a 6kW drive motor, the typical range of such units are from 1 to 6,000 kW and speeds from 3 to 6,000 rpm. An exhaustive study of the widely varying parameters is needed to address this problem.

Table 3: Summary of case studies

Case Studies		Frquency, Cps
Lateral Critical Speed Of Shaft 1	Foot Mounted Motor	1390
	Flange Mounted Motor	30.46
Lateral Critical Speed of Shaft 3	Foot Mounted Pump	607
	Flange Mounted Pump	119
Natural Frequency of Housing	Housing Alone	
	Housing Placed on Empty Box	144 100
	Smaller Sides of the Empty Box Removed	7
	Larger Sides of the Empty Box Removed	5

IX. Conclusion

The main objective of this article is to highlight the role.

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