TORSIONAL STEADY STATE ANALYSIS FOR THE COMPLEX TRAIN (VFD MOTOR- GEAR BOX –MULTI STAGE CENTRIFUGAL PUMP)

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ABSTRACT: *Centrifugal pumps are being used today in the various fields such as power generation, water pumping, nuclear power plants & oil and gases extraction etc. The efforts have been made to improve hydraulic efficacy day by day, to meet the human needs in the competitive world. In order to fulfil the pre requisites for high hydraulic efficiency, centrifugal pumps have been designed in a wide assortment of sizes and shapes.*

Torsional vibrations are common problems in any turbo machinery system where mechanical power is being transmitted. The most serious problem in power transmission is the resonance condition that may arise due to coincidence between the system's natural frequency and the frequency of excitation.

Chapter 1

INTRODUCTION

1.1. Centrifugal Pumps

A pump is a machine that bestows vitality to a fluid to build its weight and move it starting with one point then onto the next. We utilized the term fluid (not liquid) on the grounds that the word pump is all around used to allude to a machine that pumps fluid, while one that handles air, gas, or vapor, is particularly alluded to as pneumatic machine, vacuum pump, blower, blower, or fan.

Pumps assume an essential part in our day by day lives more than we most likely acknowledge it. At homes, pumps can be found in aquariums, swimming pools, clothes washers, and so forth. No matter how simple or complex some pumps are, their major parts are designed to perform specific functions regardless of their sizes and shapes.

The major parts are:

1. Casing
2. Impeller
3. Shaft
4. Seal
5. Bearing
6. Coupling

OBJECTIVE

Torsional vibration is an oscillatory rakish movement that causes relative curving in the turning individuals from a framework. The typical engineering objectives of torsional vibration analysis are listed below,

1. Predicting the torsional natural frequencies of the system.

2. Evaluating the effect of the natural frequencies and vibration amplitudes of changing one or more design parameters (i.e. "sensitivity analysis").

3. Computing vibration amplitudes and peak torque under steady-state torsional excitation.

4. Computing the dynamic torque and gear tooth loads under transient conditions (e.g., during machine startup).

5. Evaluating the torsional stability of drive trains with automatic speed control.

Chapter 2

SCOPE OF THE PRESENT WORK

The scope of the present work is to improve the mechanical dynamic system of the multistage centrifugal pump shaft (VFD motor-Gear box-Pump Shaft) in order to avoid failure under working conditions as per API.

- Predict the torsional critical speeds and determine the mode shapes of the gear train at those torsional critical speeds.

- Identify interference points within or near the operating speed range of a pump and motor speeds for all possible excitations as per API.
- Modify the mechanical system of the gear train to avoid the interference points as maximum as possible.
- Steady state (Forced response analysis) is performed to calculate the peak dynamic stress at the unavoidable interference points to make sure those stress are with allowable limits.

Methodology

The torsional dynamic analysis is performed for the great train (VFD motor-gear box-multistage centrifugal pump)..

The steps of the calculation are:

- The rotor modelling (geometry, masses, and inertia)
- Shaft line is made of elements describing geometry including mass and inertia.
- Natural frequencies calculation of the rotor
- Dynamic stress calculations for the gear train.

Chapter 3

TORSIONAL RESPONSE ANALYSIS OF MULTISTAGE CENTRIFUGAL PUMP

The primary objective of this work is to build an FE model of multi stage centrifugal pump-gear box-motor rotor in RBTS. The details about gear train, impellers, mechanical seal, central sleeve, couplings and winding mass are derived from an existing model. The following steps describes modeling, analysis and results extractions.

Torsional Modeling

The initial step comprises in discretizing the persistent firmness and mass of idleness dissemination to the torsional-spring (solidness) and torsional-mass components.

Fig. 3.1 presents the torsional mass-elastic system for the equipment arrangement with the gear box. The arrangement evacuated in a four branch system, who's low –speed branch operates at the motor's synchronous speed of 1832 rpm. Referring to this graph, the torsional elements are arranged sequentially from left to right. The balloons represent station-based inertia.

Material Data

Various steel grade materials are used for pump; motor, high speed shaft and low speed shaft are shown in table 5.1.

Table 3.1. Material properties

Component	Material Grade	Young's Modulus (Mpa)	Shear Modulus (Mpa)	Density (Kg/m3)	Yield Stress (Mpa)	Tensile Strength (Mpa)	Fatigue Strength (Mpa)
Pump shaft	Steel 4145	2.00E+05	8.50E+04	7.80E+03	7.58E+02	9.31E+02	4.65E+02
Motor Shaft	Steel 4145	2.00E+05	8.50E+04	7.80E+03	7.58E+02	9.31E+02	4.65E+02
HS Pinion	Steel 4145	2.00E+05	8.50E+04	7.80E+03	7.58E+02	1.10E+03	5.52E+02
LS Shaft	Steel 4145	2.00E+05	8.50E+04	7.80E+03	7.58E+02	1.07E+03	5.52E+02

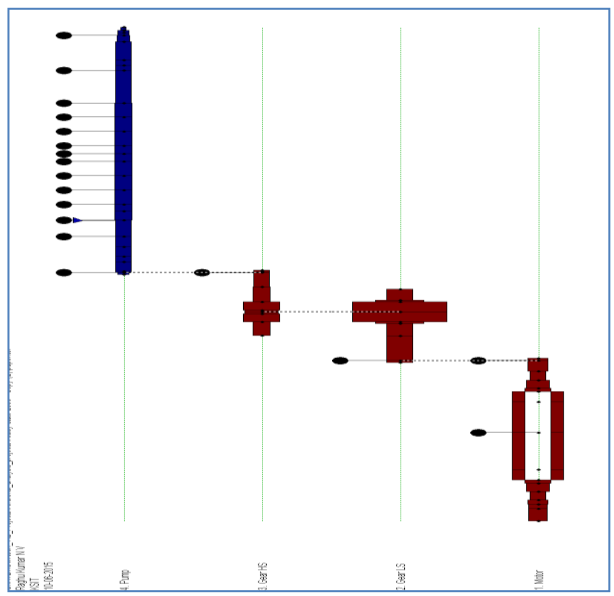

Fig.3.1. Modeling of multistage centrifugal pump casing – VFD Motor-Gear Box

Chapter 4

RESULTS AND DISCUSSIONS

4.1 Torsional Critical Speeds Calculation through RBTS

The torsional critical speeds are computed through RBT software. First 20 torsional critical speeds are obtained as shown in table 6.1. The first torsional critical speed is validated with theoretical calculations.

Table 4.1 Natural Frequencies of Undamped Rotor

Mode No	Frequency (RPM)	Mode No	Frequency (RPM)
1	2717	**8**	54080
2	5225	**9**	77349
3	17254	**10**	88256
4	29234	**11**	99236
5	29836	**12**	152631
6	35598	**13**	166439
7	50892	**14**	

The mode shapes at all natural frequencies are shown in Fig.6.1 and 6.13. The mode shapes are represented the deflection of the gear train at the various frequencies.

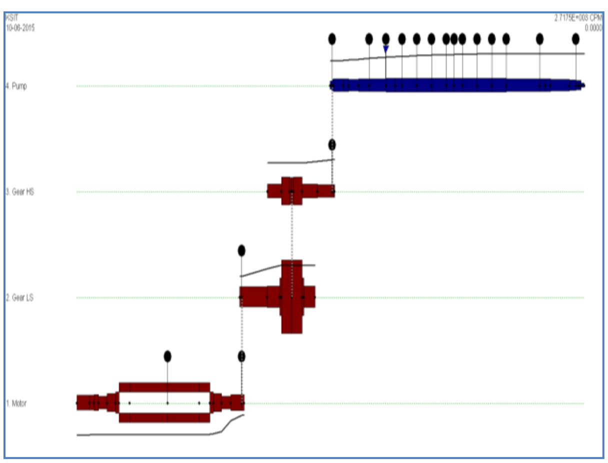

Fig. 4.1. Mode Shape at first Torsional Critical Speed

4.2. Torsional critical speed calculations using ANSYS

Torsional critical speeds are calculated using ANSYS software and these values are listed in table. 6.2.

Table 4.2. Torsional Critical speed

Mode No	Torsional critical speed (CPM)	Mode No	Torsional critical speed (CPM)
1	2919.60	8	58372.20
2	4305.00	9	75510.00
3	15607.80	10	90312.00
4	25630.80	11	100440.00
5	30895.20	12	137220.00
6	38372.40	13	162660.00
7	48220.80		

The mode shapes at all natural frequencies are shown in Fig.6.14 and 6.27. The mode shapes are represented the deflection of the gear train at the various frequencies.

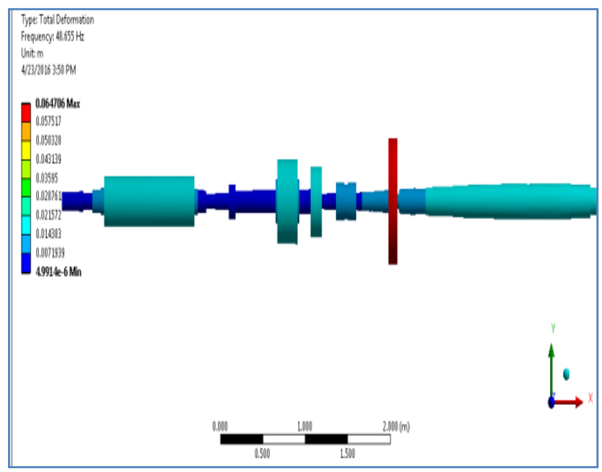

Fig.4.3. 1st torsional mode shape

4.2.1Interference Diagram (Campbell Diagram)

A representative diagram for a Campbell diagram is depicted in fig 6.28.The natural frequencies are plotted as horizontal lines and the operating speed range is designated by vertical lines. The upward slopping lines are harmonics of speed that represent the system's potential excitations.

Table 4.3: Natural frequencies at the 1x, 2x and 5x speed at 3600 RPM

	1x Pump Speed			2x Pump Speed			5xPump Speed		
S.No	Natural Frequency	Operating Speed	Gap(%)	Natural Frequency	Operating Speed	Gap(%)	Natural Frequency	Operating Speed	Gap(%)
1	2717	3600	-24.52	2717	7200	-62.26	2717	18000	-84.90
2	5225	3600	45.15	5225	7200	-27.43	5225	18000	-70.97
3	17254	3600	379.29	17254	7200	139.65	17254	18000	-4.14
4	29234	3600	712.06	29234	7200	306.03	29234	18000	62.41
5	29836	3600	728.79	29836	7200	314.40	29836	18000	65.76
6	35598	3600	888.83	35598	7200	394.41	35598	18000	97.77

Chapter 5

5.1 CONCLUSIONS:

In this present work finite element analysis is carried out to perform the torsional dynamic stability of the VFD motor driven centrifugal pump casing. The main conclusions are drawn from the above chapters,

- Torsional critical speeds are computed for entire gear train (VFD motor-Gear box-Multi stage centrifugal pump casings) to identify the resonance points. Detailed Campbell diagram is plotted for all the possible excitation; pump excitations, motor excitations, gear mesh and vane passing excitations.
- From the Campbell diagram, it is observed that there are interference points at the speeds of 5225 Cpm and 17254 Cpm, which are 2xpump speed and vane passing frequency.
- After doing many trails and errors, it is observed that coupling stiffness plays an important role to improve the torsional frequency away from the excitation. Thus, mechanical tuning phenomenon is used to eliminate the interference points.
- With the coupling design change, the 2nd mode is moved from 5225 Cpm to 5785 Cpm, which is more than the 10% of the 1x pump operating speed and interference point at the vane passing frequency is remains same.
- At the vane passing frequency, torsional steady state analysis is performed to compute the dynamic stress of the entire gear train. From the steady state analysis, it is observed that the dynamic stress at every step of the gear train is below the fatigue strength of the shaft material. Hence, the gear train under loading conditions is observed to be safe.
- In this present work, maximum dynamic stress is calculated based on dynamic torque. Effect of temperature conditions are not included in this work.
- Fatigue life of the component because of dynamic stress is not discussed in this work.
- Life of crack initiation and crack propagation for the entire gear box because of dynamic loading is essential to eliminate the catastrophic failures.

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