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Research Paper

TWO PLANE BALANCING OF A CONICAL ROTOR DRIVEN BY VERTICAL BELT SYSTEM DESIGNED TO REDUCE GYRO EFFECT

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Mounting and balancing of a conical rotor is taken up reducing the gyro component which plays down the general two plane balancing technique creating error in finding out the correcting masses. Faulty result in estimating the correcting masses may be because of horizontal belt drive or gear drive or even the direct drive with misalignment. The horizontal drive will be leading to bending of the shaft length wise sufficient to precess the rotating shaft vector culminating into the gyro action. Additional components of frequencies may intrude into the balancing procedure and may disturb the phase estimation. Hence, vertical belt drive nearer to the bearing is mooted to avoid the precession of the rotating shaft. Short length belt drive and compression of the rotor supporting bracket entail lesser displacement in the vertical direction ensuring smooth transmission of power with lesser vibration disturbance. With this precaution, the correcting masses have been calculated and the final vibrations were observed to be well within limits.

Keywords: Two plane balancing, Gyro effect, Trial masses, Correcting masses

INTRODUCTION

When the center of mass of a rotating element does not coincide with its axis of rotation a condition of unbalance exists. The force generated by this unbalance is proportional to the square of the rotational frequency. If the amount of unbalance exceeds permissible levels, even small increase in operating speed of the rotor can lead to significant increase in vibration levels. This condition can only be corrected by accurately measuring the vibration response of the rotor at its fundamental frequency and following a series of steps designed to determine the amount of unbalance and adding (or subtracting) an appropriate amount of compensating mass at the necessary locations.

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The first step in balancing is the definition of the number of balance planes, the maximum allowable vibration at each balance plane and the setup of vibration channels and tachometer input which is used to provide a measure of the rotor speed during balancing and also serve as a vibration phase reference (ISO Standard 8821:1989; ISO 1940/1; ANSI S2. 19-1975; BS 6861: Part 1; VDI 2060; Standard Paragraphs; MIL-STD-167-1 (SHIPS); Dynamic Balancing Handbook; ISO 1925).

Measurement of vibration channels may be defined in units of acceleration, velocity or displacement and the maximum allowable vibration limits for each balance plane may also be defined in any of these units. The tachometer has to be capable of jitter free triggering for the synchronous averaging of vibration required for accurate balancing runs.

The second step is to define the rotor geometry. Components making up the rotor being balanced can be defined as supports (usually bearing locations) along with rotors where the addition of correction masses occurs. The position of each component along the axis of rotation, a radius for each rotor and optionally, the number of pre-drilled holes available on the rotor for the addition of trial mass and which are used by the system for “weight splitting” must all be defined. In this experiment, two aluminum plates with same diameter and thickness have been interposed inside the rotor span to connect the trial mass at the assumed radii.

The final step is the balancing measurement. It is advantageous to view the measurements for all balancing planes in either the time or frequency domain along with the visualization of the addition of trial masses. Measurement concludes with a table of correction masses and locations for all the balance planes (Entek IRD #2049; and ISO Standard, 1925).

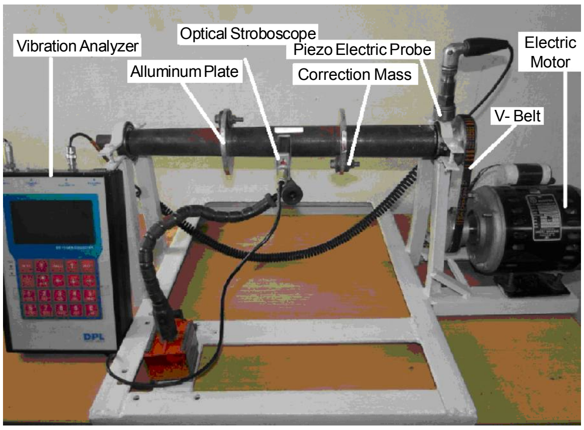
Dynamic unbalance is also referred to as two plane unbalance, indicating that correction is required in two planes to fully eliminate dynamic unbalance. A two plane balance specification is normally expressed in terms of correction weight and radius per plane and must include the axial location of the correction planes to be complete. Dynamic unbalance captures all the unbalance which exists in a rotor. This type of unbalance can only be measured on a rotating balancer since it includes couple unbalance.

Gyro effects may creep into if the drive provided creates precession of the spin axis and that is the reason mounting of the shaft and the drive to rotate the rotor is to be carefully designed not to create error in the estimation of the phase angle with the help of the optical stroboscope. Proper Tooling is necessary in the mounting design (Thearle, 1932; Genta *et al.*, 1999; and Derek, 2006) to ensure smooth rotation in the bearings without any gyro effect.

MOUNTING OF THE RIGID CONICAL ROTOR AND THE EXPERIMENTAL SET UP

The conical rotor (450 mm span, small end dia. 3.81 mm, big end dia. 5.08 mm, 6060 gm) is mounted on the centre of a base frame and brackets support the bearings at both ends fastened rigidly with nut and bolts. To avoid the gyro effect on the rotor and to ensure its smooth running a motor (with rated rpm of the motor at 1350) mounted on adjacent bracket in line with rotor is connected to it by short length vertical belt nearer to its big end which minimizes the displacement nearer to zero avoiding precession of the spin vector. A provision for fixing trial mass and correction masses around the rotor is made interposing two aluminum plates (10 mm outside diameter and 6 mm thickness) inside the rotor span in the respective two planes which are located 15 mm from each end. Mass of each ring is approximately 97.6 gm. DC-11 vibration analyzer is used to measure vibration phase and amplitude (Figure 1).

Figure 1: Experimental Set Up



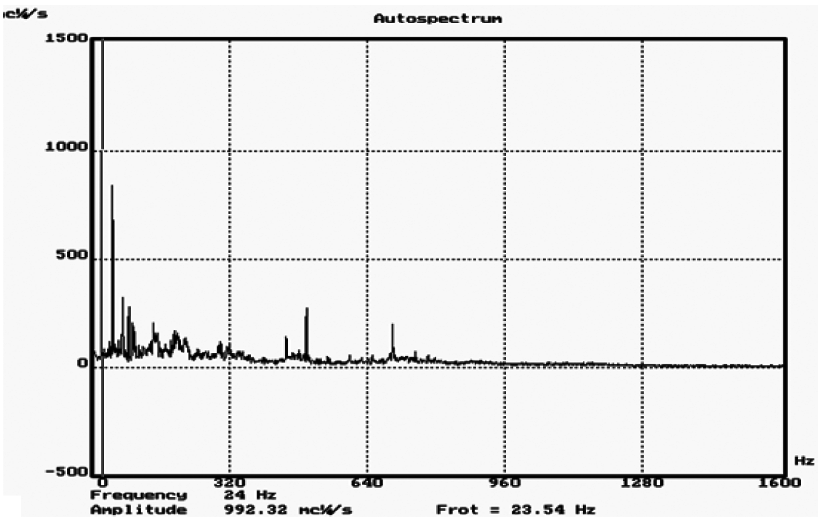
NOMENCLATURE

FFT	–	Fast Fourier transform
mV	–	milli volts
Hz	–	hertz
mm/s	–	milli meter per second
ms	–	milli seconds

EXPERIMENTATION

The trial mass is 5-10 times of maximum residual mass. The range of trial mass gives the flexibility for not considering the mass of lighter aluminum plates in calculations. Range of trial mass is 60-130 gm. A nut and bolt (8 mm dia.) with six washers (for symmetry) weighing 86.56 gm is used as trial mass. Locating trial mass is important as it leads to error in balancing. The radius of trial mass location should not be less than 35 mm for the assigned mass as calculated. Two holes

Figure 2: FFT at Small End of Rotor Before Balancing



(180 deg. apart) 8 mm diameter centered at 40 mm radii and at same position on each aluminum plate are drilled for connecting trial mass. The second hole on ring balances the mass loss of first hole. Accelerometer probe is positioned at bearing support at small (point 1) and big end (point 2) of rotor, the optical stroboscope is set for proper watch, speed of rotor is kept constant and thus simultaneously recording the Time Wave and FFT at each point (Figures 2-5). For the correction mass calculation, firstly the amplitudes (vibration) and phase angles in two planes are measured by attaching the probe at each point in vertical direction. Secondly, the trial mass is connected in plane 1 and the same are measured in the

Figure 3: FFT at Big End of Rotor Before Balancing

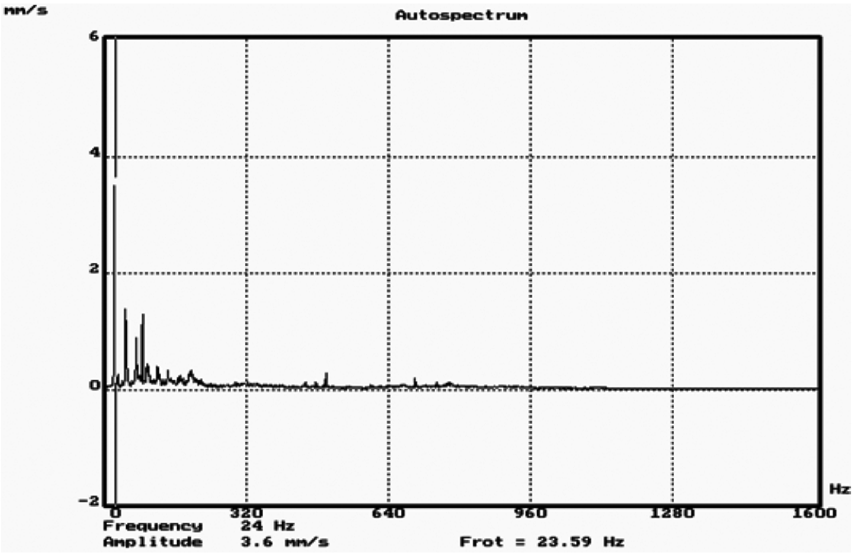


Figure 4: Time Wave at Small End of Rotor Before Balancing

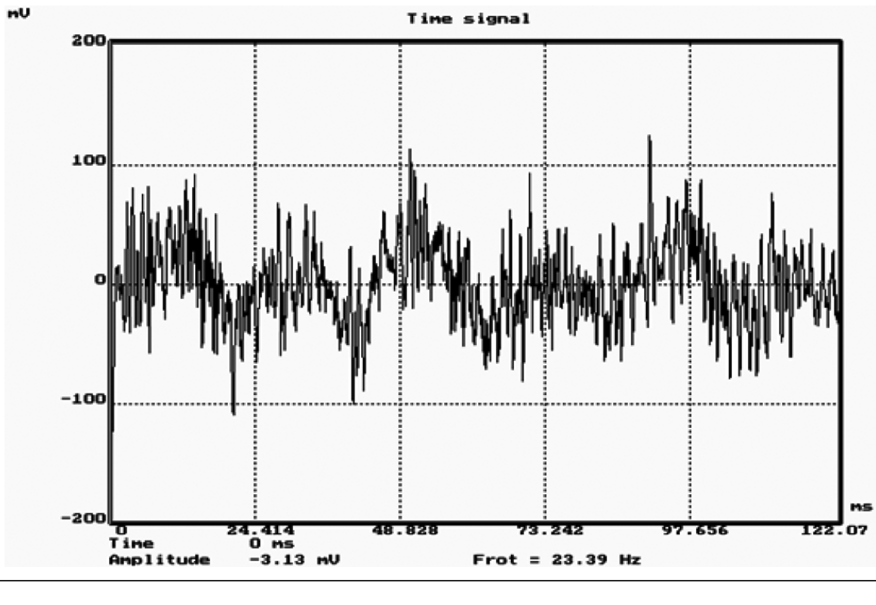
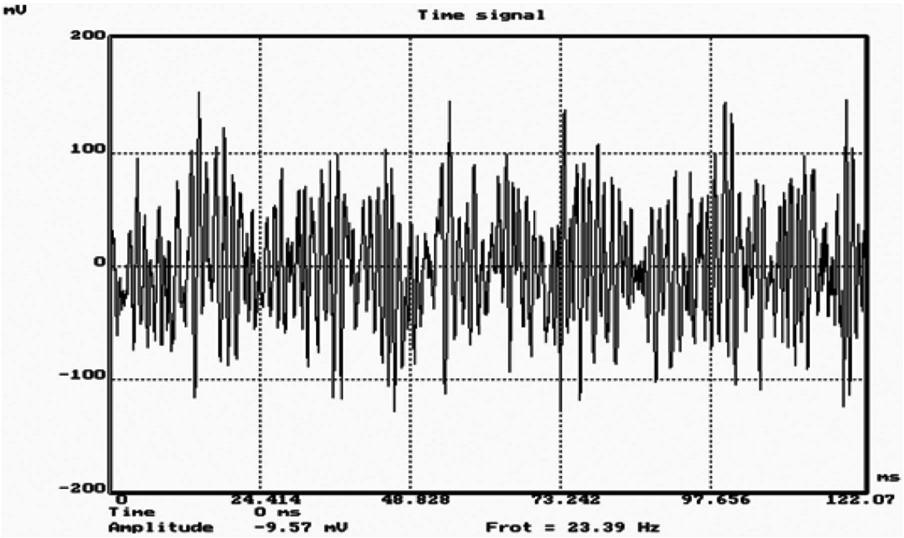


Figure 5: Time Wave at Big End of Rotor Before Balancing



two planes. Procedure is repeated for plane 2. Finally correction masses and the angles for two planes are calculated (Table 1). Standard balancing procedure equations are used in calculations. These results compare favorably with the results obtained using programmable calculator (Bruel and Kjaer, 1989).

Table 1: Before Balancing

Probe Point	Amplitude (mm/s)	Phase (deg.)
Without Trial Mass		
1.	4.195	49
2.	2.905	241
Trial Mass in Plane 1		
1.	12.124	272
2.	2.768	92
Trial Mass in Plane 2		
1.	9.031	342
2.	7.461	337

Plane	Correction Mass (gm)	Location from Trial Mass (deg.)
1.	56.61	304.42
2.	67.17	56.53

RESULT DISCUSSION

Holes (8 mm diameter centered at 40 mm pitch radius) are drilled in two plates to connect the correction masses at calculated locations. Mass loss of hole (0.8 gm approximately) is added to correction masses (Table 1). The correction masses thus used are nut and bolts with four washers each. Connecting the correction masses 57.15 gm in plane 1 and 67.899 gm in plane 2 in the respective locations and at constant speed and test run is made to assess the quality of balance of rotor. The Time wave, FFT are recorded in vibration analyzer (Figures 6-9) and phase angle, amplitude of vibration (peak to peak) are measured (Table 2). The vibration levels have reduced by 98.95% and 98.55% in two planes respectively.

The velocity plots of the vibration (FFT) before balancing indicate emergence of several synchronized and unsynchronized frequencies which will certainly affect the bearings at both ends with respect to time. Same is the case with the time waves recorded at the small end and big end of the rotor. Time wave at the small end shows vibration levels of nearly 100 mV and the time wave is of '*M*' type indicating couple imbalance. The time wave recorded at the big end further shows higher amplitudes of vibration exceeding 100 mV, which indicate impact on the bearing at the end because of conical volumetric geometry of the rotor. Figures 6 and 7 indicate smoother velocity curves of FFT indicating rectification of couple imbalance after correction masses attachment at the radii evaluated minimizing the amplitude of vibration at the rotating frequency to lowest

Figure 6: FFT at Small End of Rotor After Balancing

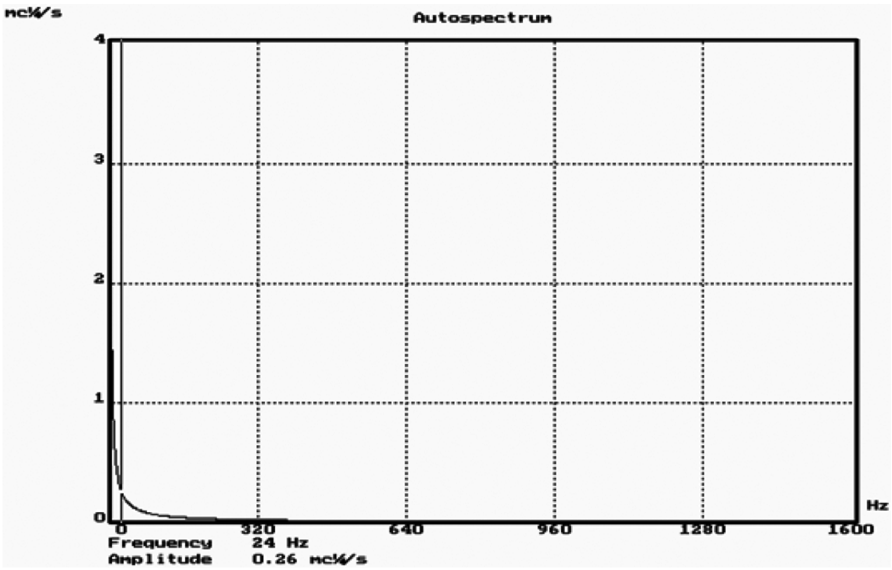


Figure 7: FFT at Big End of Rotor After Balancing

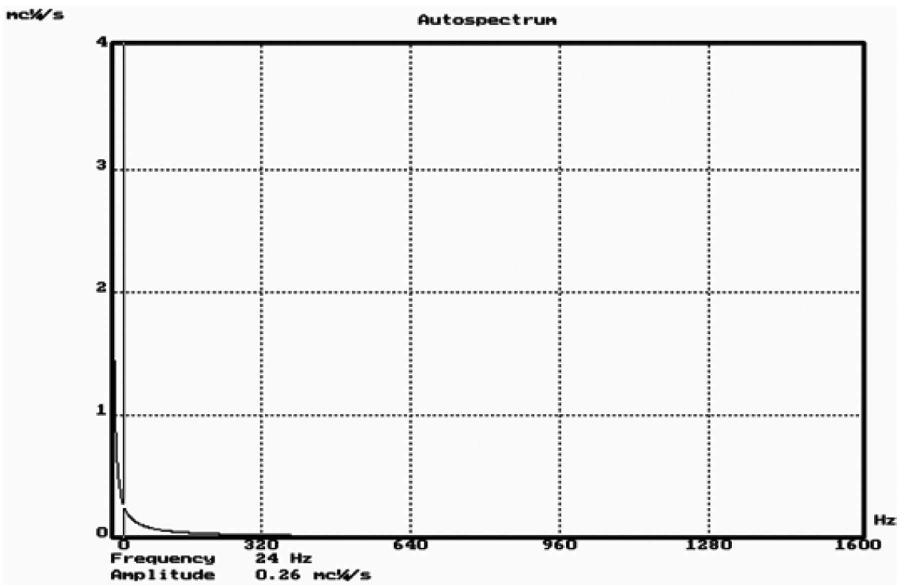


Figure 8: Time Wave at Small End of Rotor After Balancing

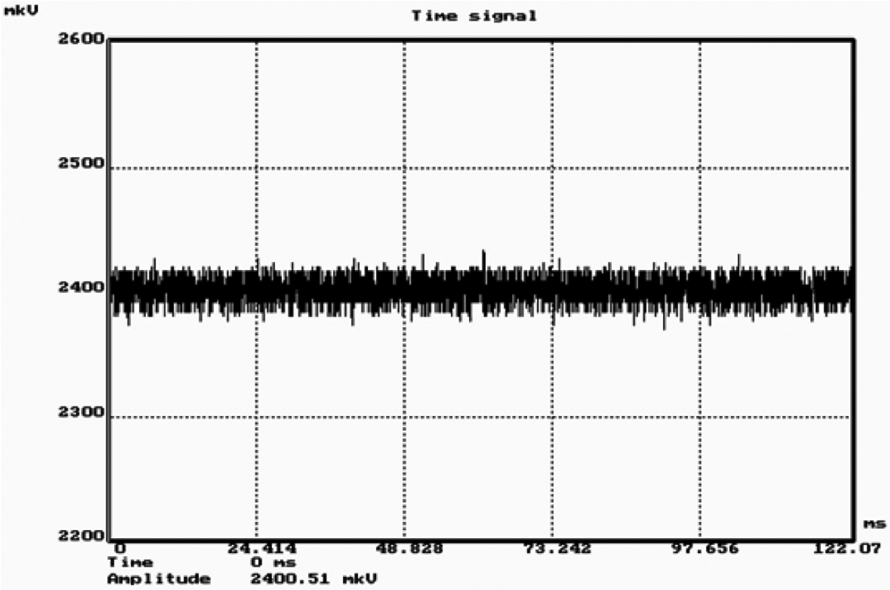
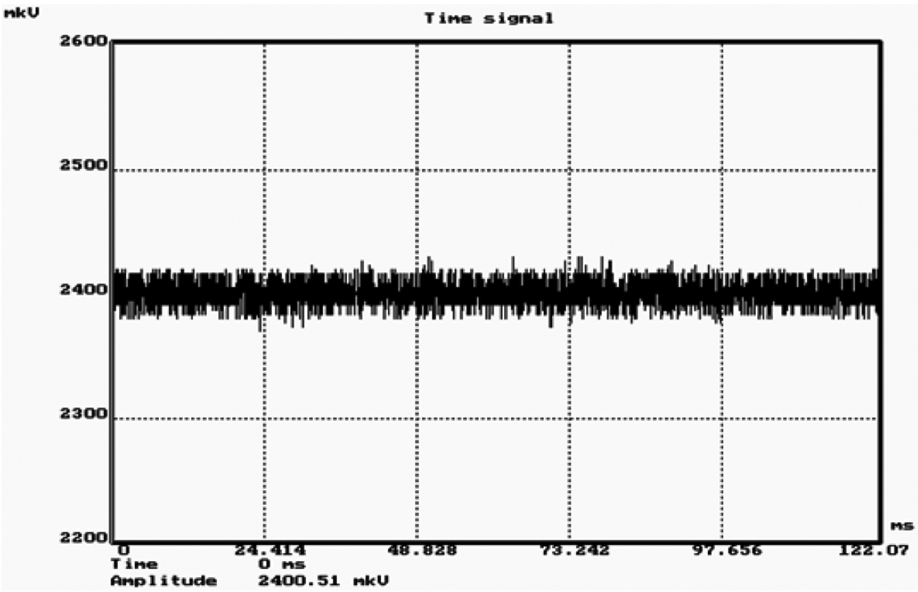


Figure 9: Time Wave at Big End of Rotor After Balancing



level. The time waves recorded after correction indicate micro volt amplitudes showing substantial reduction in vibration and the time waves encompass the belt vibration over a fixed level of vibration 2400 Micro Volts. The phase calculations of the vibration instrument

Table 2: After Balancing

Probe Point	Amplitude (mm/s)	Phase (deg.)	% Reduction of Vibration
1	0.044	308.4	98.95
2	0.042	299.8	98.55

are expected to be reliable now with the Gyro effect bringing down to minimum level with the kind of drive given to the rotor.

CONCLUSION

Considerable reduction in vibration levels shows that gyro effect on rotor is successfully reduced and rotor is almost balanced. Thus the mounting of rotor and drive to rotate rotor are properly designed to reduce gyro effect on rotor. The existing vibrations in rotor are seen as a black band with amplitude reversals in time wave represent belt drive vibration due to tensions change in the length of the belt.

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