© 2017 IJSRST | Volume 3 | Issue 2 | Print ISSN: 2395-6011 | Online ISSN: 2395-602X

National Conference on Advances in Engineering and Applied Science (NCAEAS)
16th February 2017
In association with
International Journal of Scientific Research in Science and Technology

Single and Multi-Plane Balancing of Rotor in Static and Dynamic Condition – A Review

Konal I. Singh , Saket Kolhe , Amit R. Bhende

Mechanical Engineering, St. Vincent Pallotti College of Engineering & Technology, Nagpur, Maharashtra, India

ABSTRACT

Static and dynamic balancing of a rotor plays an important role in deciding the life of any turbo-machinery. Vibrations generated due to unbalance rotor is considered as one of the major factors that can lead to accelerating degradation of machine components or even catastrophic failure especially at high speed where reliability is important consideration. Rotor unbalance may result due to error in manufacturing process, wear and tear of rotating component, thermal deformations, deposition of material on rotor surface etc. There are various static and dynamic balancing methods depending upon type of rotor. Different methods of rotor balancing requires different trial runs (number of times the machine has to stop) so as to place trial weights and measure vibration amplitude. This paper studies the number of trial runs required in various rotor balancing methods in static and dynamic condition. Any method which balances the rotor in less number of trials is considered to be the efficient one because stopping machine results in more energy consumption and reduction in machine life. It is normal human psychology to find and search for a method that solves the problem in least possible time and the work can be started again. The paper also includes the derivation of two balancing methods that has reduced the number of trial runs to a considerable number and also reviews the literature concerning the origin of various balancing techniques including the ones that use influence coefficient method, cradle balancing method, modal balancing method.

Keywords: Balancing, Rotor, Vibration, Phase Angle, Influence Coefficient Method

I. INTRODUCTION

Static and dynamic balancing of a rotor plays an important role in deciding the life of any turbo-machinery. A rotor is said to be unbalanced when axis of inertia of rotor axis of rotation of shaft are different. Balancing any rotor means coinciding (or try to coincide), the axis of inertial of the rotor and axis of rotation of the shaft. Theoretically, the inertia axis of the rotor coincides with axis of shaft, but in actual practice, this does not happen due to error in manufacturing process, wear and tear of rotating component, thermal deformations, deposition of material on rotor surface etc. Rotor unbalances results in centrifugal couple and unbalance centrifugal forces that are being generated and transmitted to the shaft support bearings. These centrifugal couple and unbalance centrifugal forces further increases the value of centrifugal force, which further increases the distance of center of gravity from axis of rotation. This effect is cumulative and ultimately the shaft fails. The bending of shaft not only depends upon the value of eccentricity but also depends upon the speed at which the shaft rotates. The speed, at which the shaft runs so that the additional deflection of the shaft from the axis of rotation becomes infinite, is known as critical speed. [1] The forces generate vibrations in the machinery and this is why unbalance rotor is considered as one of the major factors that can lead to accelerating degradation of machine components or even catastrophic failure especially at high speed where reliability is important consideration.

There are various static and dynamic balancing methods depending upon the type of rotor. The fundamental aspect of balancing process is to determine the amount and angular location of the mass concentration so as to either add an equal amount of mass to the opposite side of the mass concentration or remove mass at the mass concentration. Vibration amplitude is directly proportional to the amount of unbalance. Hence balancing process is accomplished by closely monitoring the vibration amplitudes and phase angle to determine the location of the unbalance.

A rotor can generate vibrations due to various reasons apart from unbalance such as faults in the bearing, misalignment of shaft, lose foundation bolts, transfer of vibrations from neighbouring machine etc. Now to diagnose the unbalance vibration signals, frequency analysis of vibration signal is carried out. If the vibration signals are due to unbalance of rotor, then prominent peaks can be seen at the rotational frequency in the frequency spectrum. Frequency analysis is carried out before and after balancing process to see the reduction in the vibration level due to balancing [6].

In this paper, various methods of static and dynamic balancing of rotor are reviewed and compared to various methods on their merits and demerits. For dynamic balancing, rotors are classified in two major categories, e.g., the rigid and flexible rotors. In fact, the same shaft of a rotor can be considered as rigid if it is operating much below its first critical speed and the flexible when it is operating near or above the first critical speed. That is why sometime it is also called the slow and high speed rotor balancing. [2]

The rotor balancing methods can be classified as off-line balancing methods and real-time active balancing methods. The off-line rigid rotor balancing method is very common in industrial applications. In this method, the rotor is modeled as a rigid shaft, which cannot have elastic deformation during operation. Theoretically, any imbalance distribution in a rigid rotor can be balanced in two different planes. Methods for rigid rotors are easy to be implemented, but they can only be applied to low speed rotors, where the rigid rotor assumption is valid. A simple thumb rule is that rotors operating under 5,000 rpm can are considered rigid rotors. It is well known that rigid rotor balancing methods cannot be applied to flexible rotor balancing. Therefore, researchers developed modal balancing and influence coefficient methods to off-line balance flexible rotors. Modal balancing procedures are characterized by the use of the modal nature of the rotor response. In this method, each mode is balanced with a set of masses specifically selected so as not to disturb previously balanced, lower modes. There are two important assumptions:

i. The damping of the rotor system is so small that it can be neglected; and

ii. The mode shapes are planar and orthogonal. [3]

The real-time balancing methods can be classified into passive balancing methods and active balancing methods, according to what kinds of balancing devices are used. Automatic Balancing uses Passive Devices. Very little research has been done on passive auto balancing devices. The selection of balancing method will depend on several factors such as unbalance configuration, length-to-diameter ratio, balance speed compared to operating speed, rotor flexibility and amount of cross-effect.

Different methods of rotor balancing requires different trial runs (number of times the machine has to stop) so as to place trial weights and measure vibration amplitude. Any method which balances the rotor in less number of trials is considered to be the efficient one because stopping machine results in more energy consumption and reduction in machine life. This paper reviews the various methods of static and dynamic balancing of rotor and effect of rotor speed on its balancing.

II. ROTOR BALANCING METHODS

This section of the paper gives an overview of previously published literature surveys on balancing of rotating machineries. These reviews are limited in scope, focusing on methods related to static and dynamic rotor type, and do not directly discuss application to rotor dynamic systems. The methods are reviewed in the order of their number of trial runs required to accomplish balancing process.

A. Single plane balancing:

In the actual practice location (radial as well as angular) of centre of gravity G point is unknown in the single plane rotors. The orientations of the point G can be obtained by keeping the rotor on frictionless supports and gently allow it to rotate freely without any external drive. The rotor becomes stationary after some time with heavy spot (G) acting vertically downwards. To confirm the orientation of the residual unbalance it can be marked by chalk or any other means. Now place a correction mass at 180˚ to the heavy spot and again allow rotor to rotate freely by gentle push of the rotor. If the marked heavy spot again comes vertically downwards that means correction mass m to be increased. If the marked heavy spot comes vertically upwards positions, means correction mass is more, and it has to be decreased. If heavy spot rests at some other position, means rotor is nearly balanced. Confirmed can be obtain by freely rotating the rotor again and finding whether it rests always at some indifferent equilibrium position. This is called the static balancing of rotor (disc) and is valid for a rotor with only one disc.

B. Two plane balancing (Cradle balancing machine)

Dr. R. Tiwari (2010) [2] describes and explains the two type of balancing methods i.e. hit and trial method and systematic method. He also explained the procedure to conduct the balancing of rotor on a cradle balancing machine mathematically and graphically. The rotor must be removed from the installation and mounted on the bearings of a cradle balancing machine. The author has described two procedure as (i) Hit and Trial method: A large no of measurement are required to obtain correction masses at two balancing planes. (ii) A systematic method: Requires only eight measurements to obtain correction masses at two balancing planes.

Hit & Trial Method

The cradle is supported on four springs and can be fulcrum about F_1 or F_2 to form a simple vibrating system about F_2 or F_1.The fulcrum are located on two chosen balance planes (i.e. I and II), where the correction mass to be added. The rotor is driven by a motor through a belt pulley arrangement. The spring system and natural frequency of the system should be in the range of motor speed. Fulcrum the cradle in plane I, by fixing F_1 and releasing F_2.Run the rotor to resonance, observing the maximum amplitude to the right of fulcrum F_2. The vibration is due to all the unbalance in plane II, since the unbalance in plane I has no moment about F_1. A trial mass is placed at a chosen and amplitude of vibration is calculated. A plot of this amplitude for different location of the same mass. Trial mass for correction is added at the location where the amplitude of vibration is minimum. Increase or decrease the trial mass at the same location, until the desired level of balance is achieved correction mass for plane II. Similar procedure is repeated by Fixing F_2 and releasing F_1 to get correction mass at plane I. This procedure is tedious and time consuming.

Systematic Balancing Method

This method is aimed to reduce the number of measurements while using the cradle balancing machine. Measurements are required in the cradle balancing machine and correction masses at plane I and II are obtained by fulcruming at F_2 and F_1, respectively, is based on only four observations. In first observation, vibration amplitude is taken without any addition of the trial mass to the rotor whereas the subsequent observations are taken with a trial mass at ɵ= 0˚, where is measured from a conveniently chosen location on the balancing plane, same trial mass at 180˚and same trial mass at ɵ=$\pm$90°.

The procedure is repeated for two cases (e.g. when fulcruming at F_1 and then for F_2). A graphical method is used for finding the unbalance vector. Fig. 2 shows the graphical method. The above method is derived further mathematically. Let, a_0, a_1, a_2 be the amplitude of vibration with residual imbalance, trial mass at 0° and 180° respectively. Let, 'x' is amplitude of unbalance then vector cosine law is applied to triangle OAB as shown in fig. 2

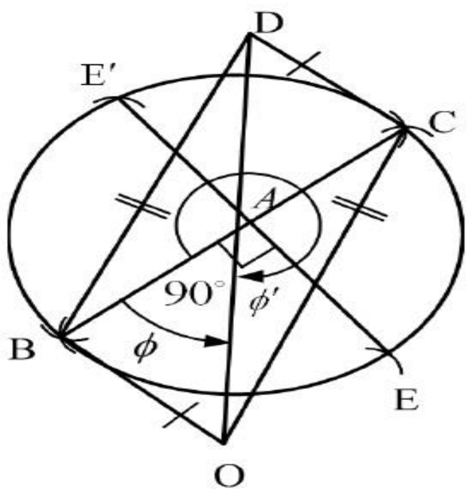

Figure 1: A construction procedure for finding the unbalance vector

So the equation for Ø becomes

$$\cos\emptyset = \frac{a_0^2 + \overline{AB}^2 - a_1^2}{2a_0\overline{AB}} - - - - - - - - (1)$$

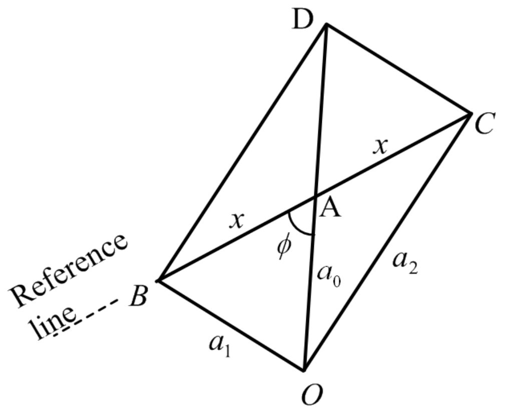

Figure 2: Geometrical constructions for determination

$$\cos(\pi - \emptyset) = \frac{a_0^2 + \overline{AC^2} - a_2^2}{2a_0\overline{AC}} - - - - - - (2)$$

$$\overline{AB} = \overline{AC} = x - - - - - - - - - - - - - (3)$$

Equating equation (1),(2),(3)

$$2a_0\overline{OB}\cos\emptyset = a_0^2 + \overline{AB}^2 - a_1^2 = a_0^2 + \overline{AC^2} - a_2^2$$

when simplified further

$$x = \pm\sqrt{\frac{1}{2}(a_2^2 + a_1^2) - a_0^2}$$

and

$$\cos\emptyset = \frac{a_2^2 - a_1^2}{4a_0 x}$$

C. Dual Plane Balancing

E. L. Thearleenet. al. (1934) [4], describes the various means of dealing with the various components of vibration occurring at running speed frequency. The author have also discusses about the different portable instrument used for balancing the rotating machineries.The author have used the technique for a two plane semi-graphical balancing procedure based on a linear rotor system to conduct experimental trial (using test rig shown in fig 3).The similar techniques named as Influence Coefficient Technique was originally proposed by Goodman (1964) [7], refined by Lund and Tonneson (1972) [8], and verified by Tessarzik and others (1972) [9].This technique comprised what we would now call a dual plane balancing; the balance computation included one speed and two vibration sensors (Generator & Contactor).

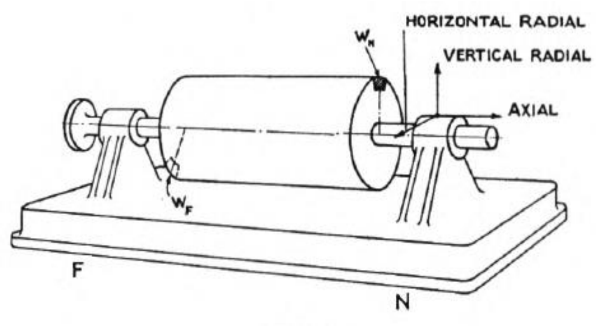

Figure 3: Test rig for dual plane balancing

The values of the correction mass and angle can be determined by plotting vector diagram as shown in Fig. 4. Vector diagram is plotted by taking the vibration amplitude and phase angle at no corrective weight being added, at reasonable amount of corrective weight W'_n is mounted on near end of rotor plane and corrective weightW'_f is mounted on far end of rotor plane. Fig. 4 shows the vector plot in dual plane balancing method.

The trial weights W'_n and W'_f and the final wreights W_n and W_f require a statement of both magnitude and position(direction) to specify each of them.Each final weight may be derived from its corresponding trial weight by a shift in angle and a

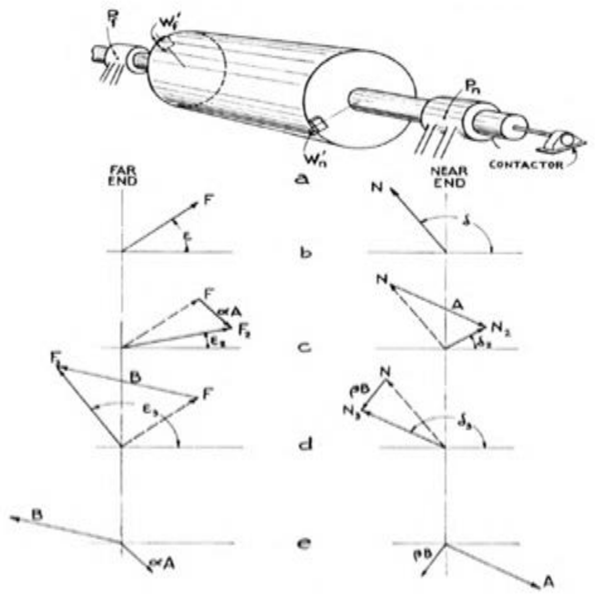

Figure 4: Vector plot in dual plane balancing method

multiplication. Thus two new vector operator θ and Ø may be introduced such that

$$W_n = \theta\, W'_n \;--------\;(4)$$

$$W_f = \emptyset\, W'_f \;--------\;(5)$$

The angles θ & Ø are the angle through which the near and far end trial masses should be shifted in counter clockwise direction. The values of correcting masses and their positions can also be calculated by plotting the vectors on a polar plot and the equations given below. Fig. 5 shows polar plot and balance data calculations.

The operators θ and Ø can also be calculated as

$$\theta A + \emptyset B = -N\;---------(6)$$

$$\emptyset B + \theta\,\alpha A = -F\;--------\;(7)$$

On solving equation (6) & (7)

$$\theta = \frac{\beta F - N}{(1-\alpha B)A}$$

$$\emptyset = \frac{\beta N - F}{(1-\alpha B)B}$$

Rearranged the above equation of θ and Ø as

$$\theta A = \frac{\beta F - N}{(1-\alpha B)}$$

$$\emptyset B = \frac{\beta N - F}{(1-\alpha B)}$$

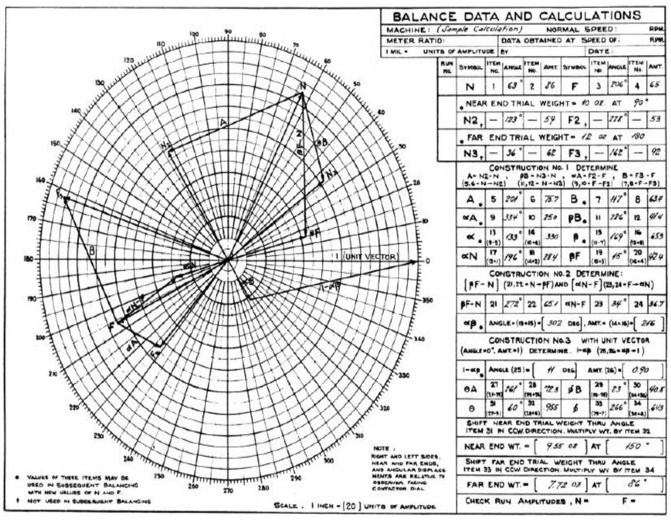

Figure 5: Vector Representation on polar plot

The value of θ can be calculated as θA-A& similarly the value of Ø = ØB - B and the corrective weight to be mounted at θ&Ø can be found from equation 4 & 5.

D. Modal Balancing

This is an off-line method in which rotor is to be mounted on a balancing machine. First run the rotor near first critical speed and measure vibration amplitudes. Select a suitable trial mass and mount it near to the hub of the rotor. Vibration amplitudes are measured at the same speed. Single plane balancing method can be applied to determine the correct mass and its location using the above two vibration measurements. After mounting the corrective mass, it can be observed that the vibration amplitudes are reduced considerably. Next, the rotor is run approximately at second critical speed. Vibration amplitudes are measured at this speed. Trial mass is mounted at 180°a part [10]. It has been observed that if rotor is balanced at first critical speed then it will not affect the balancing at other critical speeds [11]. Similarly correction mass is determined at second critical speed. Similarly rotor can be balanced up to higher modes [12] [13]. Kellenburger [14] suggested that the rotor should be corrected in N+2 planes, so as not to disturb the rigid body balancing.

III. CONCLUSION

Rotor balancing is very critical and time consuming process in industry. In off-line method, the rotor has to take out from the machine and put it on balancing machine or on some specially designed fixtures for the balancing. Whereas in on-line balancing, machine has to stop for several time to put the trial mass on the rotor and take the various vibration measurements. These stoppages to the machine lead to loss of energy, production loss, reduction in component life etc. Hence any method which accomplishes in less number of trial runs is considered to the efficient one. In this paper, various balancing methods have been reviewed. Out of these methods, the dual plane balancing method is found to be the efficient and accomplished in only three numbers of trial runs for first time balancing of a rotor. Any subsequent rebalancing process of the rotor requires only a single run provided there is no change in the machine such as speed, means of support.

All the methods explained in this paper are accomplished at constant speed. But in actual practice, machines are operated at various speeds. A rotor is balanced at one speed may get unbalanced at other speed. The design of rotor under various study are taken as uniform cross-section but in actual practice some rotor may have non uniform in cross section. The example of such type of rotor is conical rotor. Effectiveness of two plane balancing method is need to be studied the disadvantages of field balancing are production loss, component life, skilled manpower require. Frequent stoppages of machine for trial runs etc. These drawbacks can be reduced by using various numerical techniques such as finite element method. Hence this study is to be extended further to analyse the effect of rotor speed on its balancing.

IV. REFERENCES

Reference to a book:

[1]. R.S. Khurmi1, J.K. Gupta2, "Theory of Machines" 14th Edition, S. Chand& Co. Ltd., New Delhi, 2005.

Reference to a journal/conference article:

[2]. R. Tiwari, 2005, "Mechanical System and Signal Processing, Conditioning of Regression Matrices for Simultaneous Estimation of the Residual Unbalance and Bearing Dynamic Parameters", *ELSEVIER, Mechanical System and Signal Processing*.

[3]. ShiyuZho[1], Jianjun Shi[2], "Active Balancing and vibration control of Rotating Machinery: A Survey", *SVD-01-02*, May, 04, 2001, 1-30.

[4]. E. L. Thearle[1], Schenectady[2], N. Y[3], "Dynamic Balancing of rotating machinery in the field", *ASME Transactions, APM 56-19*, Vol. 56, 1934; 745-753.

[5]. T.P. Goodman, "A least-squares method for computing balance corrections", *Trans. American Society of Mechanical Engineers, J. Engineering for Industry*, 86(3) (1964), 273–279.

[6]. MacdaraMacCamhaoil, "Static and Dynamic Balancing of Rigid Rotors," *Bruel & Kjaer Application Notes*, 1-20.

[7]. Goodman[1], T. P[2], "A Least-Squares Method of Computing Balance Corrections", *ASME Transactions, Journal of Engineering for Industry*, Aug. 1964, 273-277.

[8]. Lund, J. W., and Tonneson, J., (, "Analysis and Experiments in Multi-plane Balancing of Flexible Rotors", *ASME Transactions, Journal of Engineering for Industry*, Vol. 94(1) (1972), Feb., 233

[9]. J. M.Tessarzik[1], R. H.Badgley[2], W. J. Anderson[3], "Flexible Rotor Balancing by the Exact Point-Speed Influence Coefficient Method", *ASME Transactions, Journal of Engineering for Industry*, Feb.1972, 148-157.

[10]. R.E.D. Bishop, "The Vibration of Rotating Shafts", Journal of Mechanical Engineering Science, Vol. 1, No. 1, 50.

[11]. R.E.D. Bishop, "On the Possibility of Balancing Rotating Flexible Shafts", Journal of Mechanical Engineering Science, Vol. 24, 1982, 215-220.

[12]. R.E.D.Bishop[1], and G.M.L.Gladwell[2], "The Vibration and Balancing of an Unbalanced Flexible Rotor," Journal of Mechanical Engineering Science, Vol. 1, No. 1, 1959, 66.

[13]. R.E.D. Bishop and A.G. Parkinson, "On the Use of Balancing Machines for Flexible Rotors," *ASME Transactions, Journal of Engineering for Industry*, Vol. 94, 1972, 561-576.

[14]. W. Kellenburger, "Should a flexible rotor be balanced in N or N+2 planes?", *Transactions of the American Society of Mechanical Engineers, Journal of Engineering for Industry*, Ind 94(2), 548-558.