

A Novel Experimental Study of Single–Plane Balancing Method of Crankshaft without Phase Angles Data

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Abstract

In real machines, rotating parts are balanced using data of both phase angles and vibration values. This paper presents an analytical solution of a novel single–plane balancing method using only two test runs of trial mass without needing to use any data of phase angles. In addition, a comparative experimental study for each of the proposed method and other two methods known as the three test runs and four test runs has been performed. Throughout this study, an actual arrangement of a crankshaft of an electrical generator has been constructed for the test purposes, moreover, a computer program based on the proposed method analysis has been written using (C) Language to calculate the magnitude and location of the required balancing mass. This study showed that the proposed balancing method was very effective, practical and saving lots of time, cost and effort as it requires only two test runs. In addition, a small vibration meter of light weight and low cost has been used to achieve balancing.

Keywords: Single plane balancing, Balancing of crankshaft without phase angles.

دراسة عملية جديدة لطريقة موازنة بمستوي واحد لعمود مرفقي بدون بيانات زوايا الطور

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الخلاصة

في الماكائن الحقيقية، تتم موازنة الأجزاء الدوّارة باستخدام كل من بيانات زوايا الطور وقيم الاهتزاز. قَدّم هذا البحث دراسة تحليلية لطريقة موازنة جديدة بمستوي واحد باستخدام اختبارين تجريبيين اثنين فقط للوزن التجريبي دون الحاجة لاستخدام أي بيانات لزوايا الطور. إضافة إلى ذلك، تم إجراء دراسة عملية مقارنة لكل من الطريقة المقترحة وطريقتين أخريتين تعرفان بالطريقة ذات الاختبارات التجريبية الثلاثة والطريقة ذات الاختبارات التجريبية الأربعة. لقد تم خلال هذا العمل إنشاء ترتيب حقيقي لعمود مرفقي لمولد كهربائي لأغراض الاختبار، كما تم أيضا كتابة برنامج حاسوبي باستخدام لغة (C) معتمد على الأساس التحليلي للطريقة المقترحة لحساب مقدار وموقع الكتلة اللازمة للموازنة. لقد أظهرت هذه الدراسة أنّ طريقة الموازنة المقترحة كانت فعالة وعملية جداً وتوفّر الكثير من الوقت والكلفة والجهد كونها لا تتطلب سوى اختبارين تجريبيين اثنين فقط. إضافة إلى ذلك، فإن عملية الموازنة تمت باستخدام مقياس اهتزاز صغير الحجم، خفيف الوزن وواطي الكلفة.

الكلمات الدلّيلية: الموازنة بمستوي واحد، موازنة عمود مرفقي بدون زوايا الطور.

Nomenclature

V	Vibration velocity value due to initial unbalance	mm/sec
V_t	Vibration velocity value due to trial mass only	mm/sec
V_0	Vibration velocity value due to initial unbalance and trial mass at (0°)	mm/sec
V_{90}	Vibration velocity value due to initial unbalance and trial mass at (90°)	mm/sec
V_{120}	Vibration velocity value due to initial unbalance and trial mass at (120°)	mm/sec
V_{180}	Vibration velocity value due to initial unbalance and trial mass at (180°)	mm/sec
V_{240}	Vibration velocity value due to initial unbalance and trial mass at (240°)	mm/sec
V_{270}	Vibration velocity value due to initial unbalance and trial mass at (270°)	mm/sec
φ	Phase angle of initial unbalance	degree
M_t	Trial mass	gram
M_b	Balancing mass	gram

1. Introduction

Unbalance is the most common cause of excessive vibration in rotating machinery. It generates forces proportional to the square of the machine rotating speed. For this reason, lowering of such vibration through balancing is essential as it leads to achieve longer bearings and tools life especially in high-speed machinery.

Over the last four decades various balancing techniques have been introduced to minimize rotor vibration due to unbalance. Parkinson [1], Foiles et al. [2] and Zhou and Shi [3] gave some comprehensive reviews of rotor balancing. In general, the most common method of balancing is the influence coefficients. It is an experimental balancing approach aims to minimize the vibration of rotors using both vibration value and phase data measured from multiple locations at a single speed for rigid rotors and more than a speed for flexible rotors.

The initial research for this method was traced back to the 1930s. Thearle [4] and Baker [5] essentially formulated a two-plane, single-speed procedure using influence coefficients. Goodman [6] extended the influence coefficients procedure to include the least-squares method for balancing of flexible rotor-bearing systems using data of both amplitude and phase from multiple speeds and measurement locations. It was then refined by Lund and Tonneson [7] and evaluated by Tessarzik et al. [8]. Afterwards, Pilkey & Bailey [9] and Pilkey et al. [10] presented further extensions to the local constraint optimisation method. Kang et al. [11] presented a modified influence coefficients method for asymmetrical rotor-bearing systems to balance crankshafts using the soft-pedestal machines. Furthermore, the validity and accuracy of the modified approach are verified both by computer simulations and balancing experiments on practical crankshafts. All the examples indicate that the modified influence coefficients balancing method can yield better balancing quality than the conventional balancing method does. Sinha et al. [12] estimated a method for the state of rotor unbalance of a rotor-bearing-foundation system using amplitude and phase data. The estimation uses a prior rotor and bearing models along with measured vibration data at the bearing pedestals. The suggested method has been applied to a small experimental rig and the estimated results were excellent. Al-Tae [13] presented both graphical and analytical

solutions for the two-plane balancing of rigid rotors and another analytical solution for the three-plane balancing of flexible rotors using vibration velocity and phase data. Firstly he conducted three computer programs based on his analytical solutions then he checked their validity on an experimental test rig of long rotor.

However, in some applications, balancing should be performed without using of any phase data. This may be simply due to phase measuring devices being unobtainable or due to the machine rotating parts to be balanced being completely surrounded or not easily being reachable. In such cases, it is possible to yield a high balancing quality using vibration meter. Early researches regarding rotor balancing using vibration measuring technique without phase was carried out first by Karelitz [14] who used a three trial mass to balance turbine generators. Ribary [15] presented a single plane graphical construction using only amplitudes taken from four runs, one due to initial and three due to trial mass. This graphical construction has then been simplified by Somervaille [16] who called it the four-circle balancing method without phase. Wilcox [17] conducted a graphical construction for single plane balancing using only the amplitudes data taken from an initial unbalance and four trial mass runs. Mah et al. [18] checked experimentally the graphical solution for single plane balancing on a disk shaped rotor. Their technique takes four runs, an initial and three trial mass runs. The four-circle balancing method without phase has also presented by Jackson [19].

On the other hand, analytical studies of balancing using only vibration data without phase has had several extensions. Hopkirk [20] presented an analytical solution of two-plane balancing using only amplitude data. Nisbett [21] presented an analytical solution of two-plane balancing of simple rigid rotors using only amplitude data. Although his technique takes nine runs including initial one, however, he stated that it can be used in the field efficiently. Al-Tae [22] considerably simplified the graphical construction of the four test runs method through presenting a new analytical solution for the method, then he checked the validity of his solution through an experimental test rig of a disk shaped rotor. His procedure was satisfied and can be used in the field effectively. Ali et al. [23] enhanced the graphical and analytical solutions of the four test runs method of Wilcox [17] and Al-Tae [22] respectively by reducing the required test runs from four to three. They conducted a computer program to investigate the validity of their analytical solution on an experimental test rig [13]. They showed that vibration has dropped down with an amount of around (91%).

This paper aims to establish an analytical solution of the mass unbalance problem that affects rotating shafts using a novel two test runs single plane balancing method based only on vibration velocities data. Throughout this work, an actual arrangement of a crankshaft of an electrical generator has been constructed in order to be used for the unbalance test purposes. The structure of the paper is as follows, the theoretical analysis of the two, three and four test runs methods is introduced in section 2 whereas section 3 presents the experimental work. Results and discussion are presented in section (4) and finally the paper is concluded in section (5).

2. Theoretical Analysis

2.1 Two test runs method

The proposed two test runs is a new balancing method requires, in addition to initial unbalance value, only two test runs of trial mass at two angles at a time; (0°) and (180°), where the reference or zero degree is an arbitrary position chosen by the user. Using these

three measurements to construct a vector diagram that illustrated in figure (1), it will be possible now to determine the magnitude and location of the rotating shaft unbalance. However, it should be remembered that the two important values which must be known in order to achieve balancing are the magnitude of balancing mass and its location on the rotor.

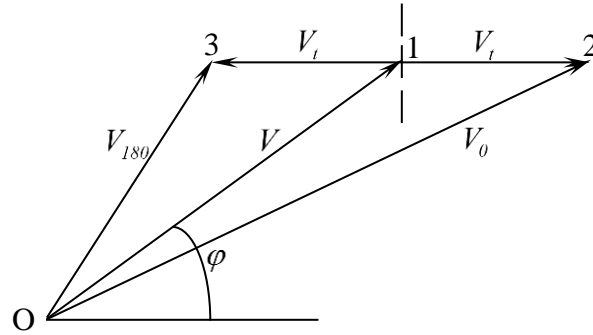


Figure (1). Vector diagram of two test runs method

With reference to figure (1), from triangle (O12), and using the cosine law, we obtain;

$$V_0^2 = V^2 + V_t^2 - 2VV_t \cos(180 - \phi) \tag{1}$$

or

$$V_0^2 = V^2 + V_t^2 + 2VV_t \cos\phi \tag{2}$$

Similarly, from triangle (O13), equation (3) below can be obtained;

$$V_{180}^2 = V^2 + V_t^2 - 2VV_t \cos\phi \tag{3}$$

Now, by subtracting equation (3) from equation (2) and rearranging to get the value of the phase angle (ϕ) due to initial unbalance as follows,

$$\phi = \cos^{-1} \left[\frac{V_0^2 - V_{180}^2}{4VV_t} \right] \tag{4}$$

The addition of equations (2) and (3) gives the value of the trial mass vibration velocity (V_t) directly as follows;

$$V_t = \sqrt{\frac{V_0^2 + V_{180}^2 - 2V^2}{2}} \tag{5}$$

Obviously, this method has the simplest and shortest mathematical analysis among other balancing methods, thus it is so easy to be programmed and implemented in site efficiently.

2.2 Three test runs method

This method requires three test runs of trial mass at three angles at a time; (0°), (120°) and (240°). Using these measurements, a related vector diagram shown in figure (2) can be constructed [23].

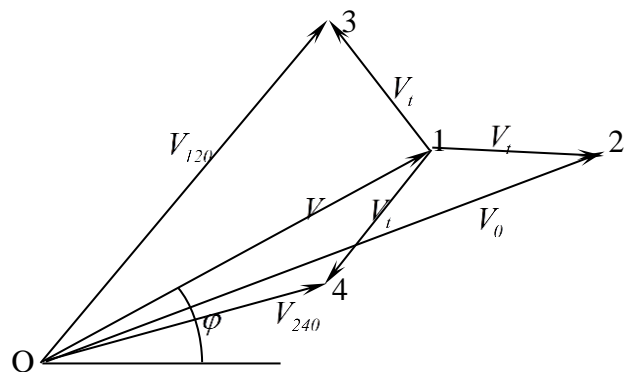


Figure (2). Vector diagram of three test runs method

From this diagram, the phase angle (φ) due to initial unbalance and trial mass vibration velocity (V_t), can be derived and written as follows,

$$\varphi = \tan^{-1} \left[\frac{V_{120}^2 - V_{240}^2}{2(V_0^2 - V^2 - V_t^2) \sin 60} \right] \quad (6)$$

$$V_t = \left[\frac{V_0^2 + V_{120}^2 + V_{240}^2 - 3V^2}{3} \right]^{1/2} \quad (7)$$

2.3 Four test runs method

This method requires four test runs of trial mass at four angles at a time; (0°), (90°), (180°) and (270°). Using these measurements, a related vector diagram shown in figure (3) can also be constructed [17, 21, 22].

From this diagram, the phase angle (φ) due to initial unbalance and trial mass vibration velocity (V_t), can be derived and written as follows,

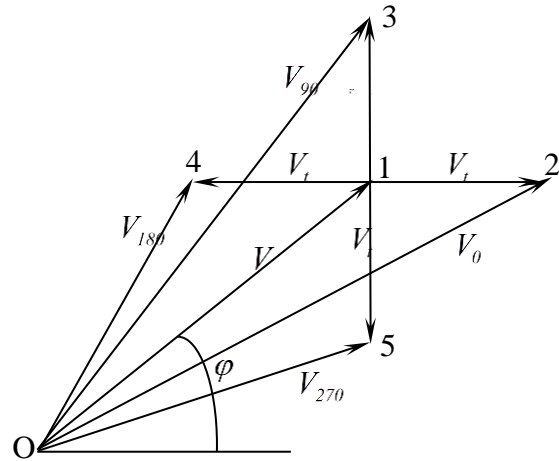


Figure (3). Vector diagram of four test runs method

$$\varphi = \tan^{-1} \left[\frac{V_{90}^2 - V_{270}^2}{V_0^2 - V_{180}^2} \right] \quad (8)$$

$$V_t = \frac{V_0^2 - V_{180}^2}{4 V \cos \varphi} \quad (9)$$

Regardless the method that is followed for balancing, it is possible now to find the magnitude of the balancing mass (M_b) as a function of the initial unbalance vibration velocity (V), trial mass vibration velocity (V_t) and trial mass (M_t) as follows [23],

$$M_b = (V / V_t) \cdot M_{trial} \quad (10)$$

In order to cancel the influence of initial unbalance, balancing mass should be put in an opposite direction to unbalance, i.e., it should be shifted by (180°).

3. Experimental Work

Figure (4) shows an actual arrangement of the electrical generator crankshaft that has been constructed and used for the test purposes. The generator which is commonly used for domestic purposes, is of (GAMMA) type with a speed of (1500 rpm) and output electrical power of around (5060 W). The arrangement consists of a single piston crankshaft, cast iron flywheel, two ball bearings, constant speed AC motor and steel frame. The crankshaft and flywheel weigh around (30 kg) together. The flywheel which represents the plane of balancing is of (280 mm) diameter and has (36) equally spaced holes at radius of (130 mm) to fix the trial and balancing masses. The motor has a power of (0.5 hp) and an approximated

speed of (1472 rpm). In order to compensate for any misalignment between the crankshaft and motor, an elastic tube has been used to connect them, which also helps to prevent any undesirable dynamic effects occurring in the motor being transmitted to the crankshaft. Furthermore, a proper mass has been attached to the flywheel in opposite direction to that of the crankshaft centre of gravity in order to compensate for the absence of both connected rod and piston. Appendix (A) clearly describes the experimental procedure that has been used to balance the crankshaft.

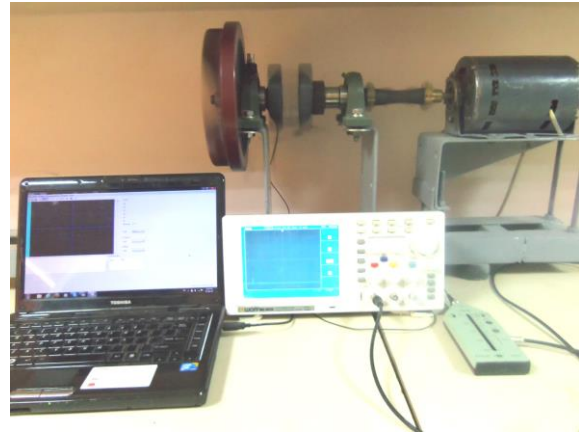


Figure (4). Crankshaft arrangement

4. Results and Discussion

Prior to balancing, it is necessary first to diagnose whether unbalance or not is the main cause of the crankshaft vibration since there are many different causes. The primary indication of unbalance is a high vibration level at the test speed which takes place here at (24.5 Hz, or almost equals to the test speed 1472 rpm) as shown in figures (5a) and (6a), the spectrum analyses of the right and left bearings respectively. With respect to the right bearing, the left one is the closest to the plane of balancing and has the highest vibration, thus it has been chosen for data collection. Table (1) gives the measured vibration velocities at the left bearing in the vertical direction with respect to the initial unbalance and trial mass at (0°) and (180°). In this work, the type of measurement that used to describe vibration was velocity, this is because of the crankshaft test speed which is in the range of (600 ... 60,000 rpm) or (10 ... 1000 Hz) [24].

Table (1). Measured vibration velocities at the left bearing

Test speed (rpm)	Test no.	Test symbol	($M_t = 10$ gram)	Vibration velocity (mm/sec)
1472	1	V	Initial unbalance	33
	2	V_0	Trial mass at (0°)	55
	3	V_{90}	Trial mass at (90°)	23
	4	V_{120}	Trial mass at (120°)	15
	5	V_{180}	Trial mass at (180°)	16
	6	V_{240}	Trial mass at (240°)	40
	7	V_{270}	Trial mass at (270°)	54

In table (1), it can be noticed that the adopted trial mass is (10 gram) which was originally calculated using ISO Formula [25] as shown in Appendix (B). However, as a rule of thumb, a trial mass should results in at least (30%) change in vibration amplitude and phase [26]. The measured vibration velocities of tests (1, 2, 5), tests (1, 2, 4, 6) and tests (1, 2, 3, 5, 7) recorded in table (1), have been fed as input data to the two test runs, three test runs and

four test runs computer programs respectively. Table (2) gives the required balancing masses and their phase angles for the three tested balancing methods.

Table (2). Required balancing masses and phase angles

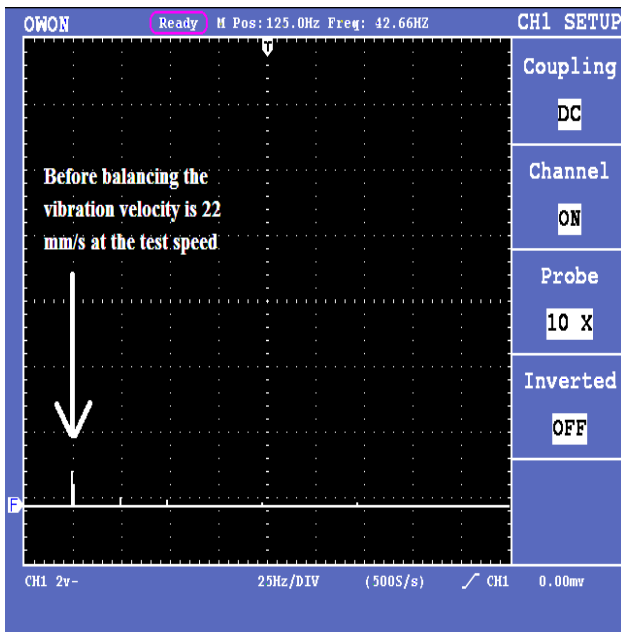
Balancing method	Balancing mass (gram)	Phase angle (degree)
Two test runs	14.3064414	153.2853751
Three test runs	14.6259656	150.5906893
Four test runs	12.1308635	139.2372141

The calculated balancing masses have been installed individually in the flywheel at their phase angles from reference (0°) in the counterclockwise direction. The crankshaft has then brought up to the test speed and the resulted vibration velocities were observed at both bearings. Table (3), gives a comparison of the vibration velocities at both bearings before and after balancing for the three tested balancing methods. Obviously, for the proposed two test runs method, the vibration velocities due to initial unbalance at the right and left bearings have fallen after balancing from (22 mm/sec) to (16.5 mm/sec) and from (33 mm/sec) to (8.5 mm/sec) respectively, which indicates that vibration has been significantly reduced especially at the left bearing.

Table (3). Vibration velocities at both bearings before and after balancing for the three tested balancing methods

Method of balancing	Right bearing (A)			Left bearing (B)		
	Vibration velocity (mm/sec)		Percentage improvt.	Vibration velocity (mm/sec)		Percentage improvt.
	Before balancing	After balancing		Before balancing	After balancing	
Two test runs	22	16.5	25	33	8.5	74.242
Three test runs		16.5	25		8.5	74.242
Four test runs		17.5	20.46		9.0	72.727

Figures (5) and (6) show the spectrum analyses before and after balancing for the three tested balancing methods of the right and left bearings respectively. Evidently, it can be stated that balancing has reduced a large amount of crankshaft vibration that results from unbalance which generally means that diagnosis takes an essential part in the solution of the problem of vibration.



(5a). Before balancing



(5b). After balancing using two test runs method

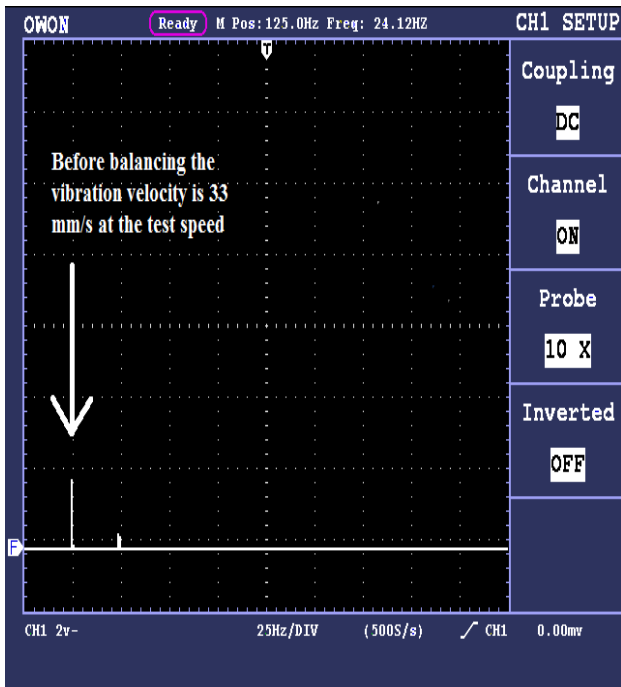


(5c). After balancing using three test runs method

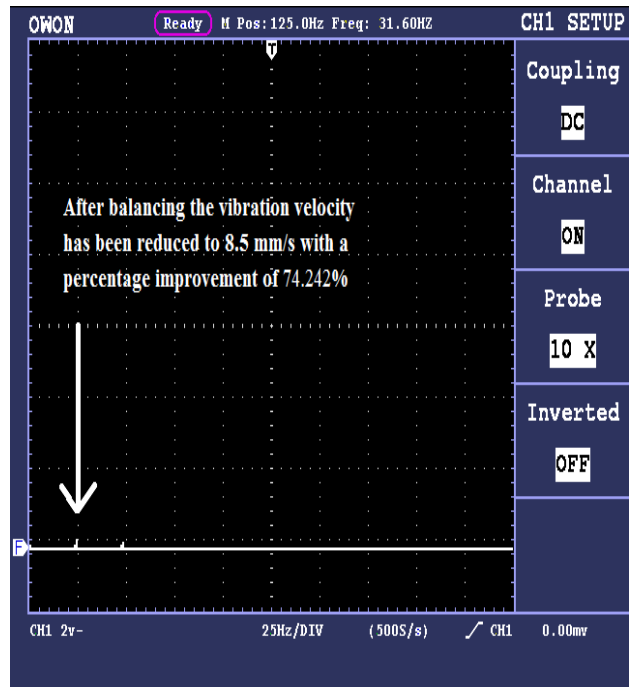


(5d). After balancing using four test runs method

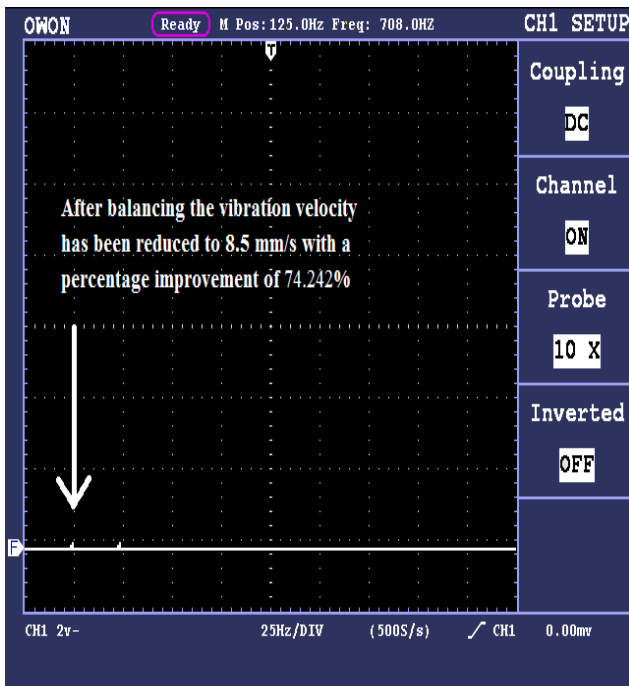
Figure (5). Spectrum analysis of the right bearing



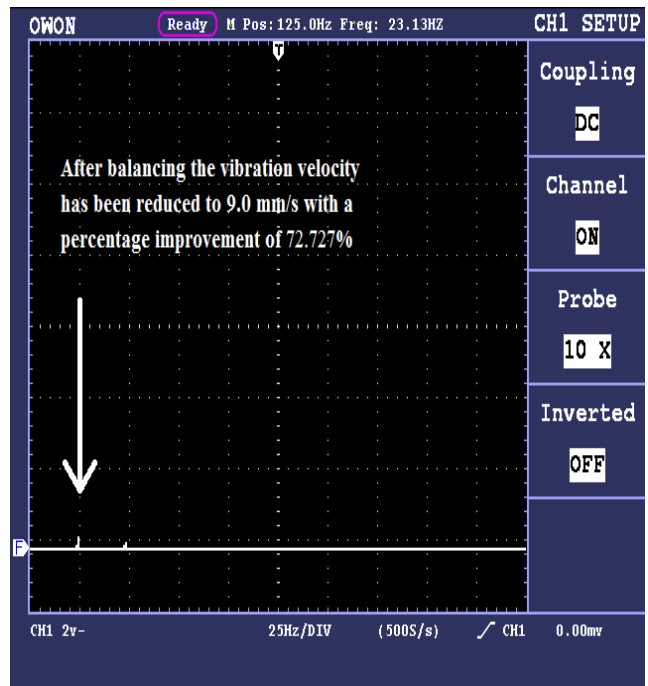
(6a). Befor balancing



(6b). After balancing using two test runs method



(6c). After balancing using three test runs method



(6d). After balancing using four test runs method

Figure (6). Spectrum analysis of the left bearing

5. Conclusions

The experimental comparison showed that the three tested single-plane balancing methods, namely two test runs, three test runs and four test runs are capable of achieving a good quality of balance of asymmetrical rotating shafts such as crankshaft using only vibration velocities data. Among these balancing methods, the proposed two test runs method was the best as just from two trial mass runs, an improvement equals to that of the three test runs method and better than that belongs to the four test runs method was gained, which leads to more saving time and cost, less shutdown runs and less physical efforts as well. For the proposed method and from only a single balancing process, an acceptable improvement of (25 %) at the right bearing and a good improvement of (74.242 %) at the left bearing have been obtained which permitted a safe operation for the crankshaft through its running speed. It is expected to obtain best results especially at the right bearing if another plane of balancing has been added which imposes to make a small redesign on the crankshaft so this point is currently under investigation.

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Appendix (A) Experimental Procedure

In this work, the crankshaft arrangement has a single plane of balancing (i.e., its flywheel) and a test speed of (1472 rpm) which is obviously very close to the (1500 rpm) crankshaft actual speed. However, this appendix describes the steps of the experimental procedure that has been followed for crankshaft balancing.

- 1) Attach the sensor to a suitable measuring point (usually the adjacent bearing to the plane of balancing) in the vertical or horizontal direction to measure the vibration velocity.
- 2) Run the crankshaft at the test speed for a few seconds to reach the steady state condition.
- 3) Due to only initial unbalance, measure the vibration velocity (V).
- 4) Stop the crankshaft and attach the trial mass to the flywheel at (0°).
- 5) Run the crankshaft again to the test speed and measure the vibration velocity (V_0).
- 6) Stop the crankshaft again and remove the trial mass.
- 7) Repeat steps (4, 5 and 6) for the other angles.

Appendix (B) Calculation of Trial Mass

For actual rotors, the value of the initial balancing mass (i.e., trial mass) could be estimated using one of the famous balance tolerance formulas (ISO 1940, API 610 or ANSI). The ISO 1940 Formula, which was used in this work to calculate the value of the trial mass, is defined as follows,

$$(9.54 * G * M) / (Rpm * r)$$

where,

(G)	= 6.3	Balance quality grade of the crankshaft (mm . rad/s).
(M)	= 30000	Mass of the crankshaft (gram).
(Rpm)	= 1472	Running speed of the crankshaft.
(r)	= 130	Radius of the trial mass (mm).

The most commonly used grades for normal applications are G2.5 and G6.3. The former is applicable to most turbomachinery rotors while the latter, which is the closest to the case of electrical generators, is applicable to fans, pumps, motors and general machinery. From ISO Formula, the magnitude of trial mass is (9.422 gram), however, (10 gram) has been adopted instead.

The work was carried out at the college of Engineering, University of Mosul