Rotating Discs Balancing

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Abstract

In this paper, a single plane balancing method based on vibration amplitude readings only, has been adopted. This method is discussed in detail, including an analytical basis and a practical implementation procedure. Four runs of trial mass are required to determine the vibration amplitudes when a calibration mass of fixed value is placed in four different positions, each position being advanced (90^{0}) from its previous position. An experimental test rig is built and prepared to obtain the vibration measurements necessary to be fed as an input data to a written computer program, which calculates the magnitude and location of the weight required to balance the test rig rotor the. For the present study, the adopted method was successful and permitting safe operation for the rotor disc through its running speed.

Keywords: Balancing of Disc-Shaped Rotors and Single Plane Balancing Technique.

طريقة نظرية وعملية لموازنة الأقراص الدوّارة

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الخلاصة

في هذا البحث تـم تبني طريقة موازنـة ذات مستوي واحد تستند فقط الـى قراءات سعة الاهتزاز. لقد نوقشت هذه الطريقة بالتفصيل وبضمنها الاساس التحليلي اضافة الى اجراء تنفيذي عملي. تتطلب هذه الطريقة أربع دورات للأوزان التجريبية. ان الغرض من هذه الدورات هو تحديد السعات الاهتزازية عند وضع وزن معايرة ذو قيمة ثابتة في اربعة مواقع مختلفة وكل موقع تم تقديمه (٩٠) درجة عن موقعه السابق. بالنسبة للعمل الحالي تم بناء واعداد جهاز اختبار تجريبي للحصول على قياسات اهتزاز ضرورية يمكن تغذيتها كبيانات مدخلة الى برنامج

مطلوب يقوم بـ حساب مقدار وموقع الوزن المطلوب لموازنة دوار جهاز الاختبار. بالنسبة للدراسة الحالية، لقد كانت طريقة الموازنة ذات المستوي الواحد ناجحة وتتيح عملية تدوير آمنة للقرص الدوار خلال سرعته التشغيلية.

Notations:

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V Vibrational amplitude at the measuring point due to original unbalance of rotor.

V_t Vibrational amplitude at the measuring point due to a trial mass mounted at any angle on the balancing plane.

V_0 Vibrational amplitude at the measuring point due to the original unbalance plus a trial mass mounted at (0^0) angle on the balancing plane.

V_{90} Vibrational amplitude at the measuring point due to the original unbalance plus a trial mass mounted at (90^0) angle on the balancing plane.

V_{180} Vibrational amplitude at the measuring point due to the original unbalance plus a trial mass mounted at (180^0) angle on the balancing plane.

V_{270} Vibrational amplitude at the measuring point due to the original unbalance plus a trial mass mounted at (270^0) angle on the balancing plane.

$\vec{V}$ Vibrational amplitude vector at the measuring point due to original unbalance of rotor.

φ Phase angle due to original unbalance of rotor.

$\overrightarrow{M}$ Balancing mass vector.

$\vec{C}$ Correction factor vector applied to the trial mass.

M_x x component of $\overrightarrow{M}$.

M_y y component of $\vec{M}$.

C_x x component of $\vec{C}$.

C_y y component of $\vec{C}$.

M Magnitude of balancing mass.

Introduction:

Although vibration may be caused by a variety of conditions such as bent shafts, misaligned couplings or bearings, foundation failure, etc., unbalance is the most common source of high vibration that affect rotating equipments [1]. Consequently, precision balancing for rotating equipments is essential for machine performance. Sources of unbalance in rotating machinery may be classified as resulting from dissymmetry, nonhomogeneous material, distortion at service speed (blower blades in built-up designs), shifting of parts due to plastic deformation of rotor parts(windings in electric armatures), etc., [2]. In some practical problems it may be necessary to attempt field balancing without the use of phase measuring equipment. This may be due to the rotating parts of the machine to be balanced being completely enclosed, or simply due to no phase measuring equipment being available. In these cases it is possible to achieve a good degree of balance using amplitude measuring equipment (i.e., vibration meter). According to Foiles et al [3], Karelitz, Ribary, Hopkirk and Somervaille [4] were the early researchers who using the amplitude measuring method. Somervaille's construction is known as the four – circle method of balancing without phase. The four circle method, as it is generally used now [3], can be found in Jackson [5]. Wilcox [6] presented a graphical construction for single plane that balanced using only the amplitude taken from an initial run and four trial mass runs. Nisbett [7] performed a simple practical procedure of two – plane balancing using only amplitude measuring equipment. Nisbett found that most simple rotors can be balanced by applying this technique iteratively; this technique took eight trial mass runs, and it can be used in the field effectively. Mah et al [8] checked experimentally the graphical amplitude measuring method for single plane balancing on a narrow rotor. Their technique took four runs, an initial and three trial mass runs. They showed that vibration of the test narrow rotor was decreased at the bearing with a humble improvement of (29%).

The work in this paper aims to solve the problem of mass unbalance that affect disc-shaped rotors such as centrifugal pumps, fans, motors, grinding and turbine wheels, etc., achieve balancing on a test rig using vibration amplitude readings taken from an initial run and four trial mass runs. A novel theoretical analysis to develop mathematical equations and implement it in a computer program specifically written by the author. The computer program can be seen in Appendix (A).

Theoretical Analysis:

The basic assumption is that vibration is due to pure unbalance and that most vibration occurs at the machine running speed. Vibration measurements must be taken as close as possible to one of the machine's support bearings. However, once a location for taking the vibration measurements has been decided on, it is important that all vibration measurements are taken at exactly the same location and direction. Four trial mass runs are required, with the trial mass placed individualy on the rotor at (0°), (90°), (180°) and (270°), where zero degree is an arbitrary position chosen by the user. By using these measurements to draw a vector digram, it is now possible to calculate the residual unbalance in the rotor. Such a vector digram is depicted in figure (1). Two important variables must be known in order to achieve balancing, the amount of the balancing mass and its location on the rotor.

From figure (1), triangle (O12), and using cosine law, gets,

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

or

Similarly, from triangle (O13), using cosine law, the equation below

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

where $cos(90 + \varphi) = -sin\varphi$

can be obtained;

$$V_{90}^2 = V^2 + V_t^2 - 2V\ V_t\ cos(90+\varphi) \qquad (3)$$

or

$$V_{90}^2 = V^2 + V_t^2 + 2V\ V_t\ sin\varphi \qquad (4)$$

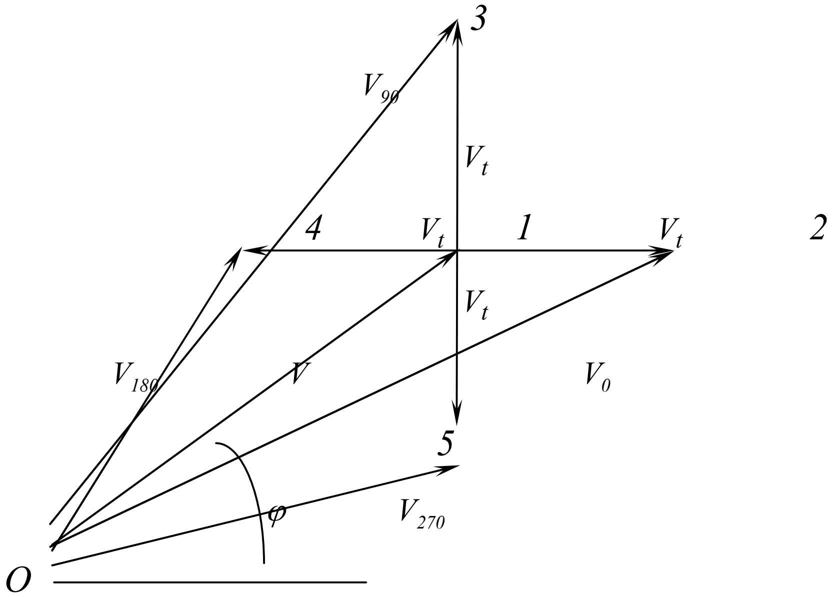

Figure (1). Vector diagram

In the same way, from figure (1), triangles (O14) and (O15) , the following equations can be obtained;

$$V_{180}^2 = V^2 + V_t^2 - 2V\ V_t\ cos\varphi \qquad (5)$$

$$V_{270}^2 = V^2 + V_t^2 - 2V\ V_t\ cos(90-\varphi) \qquad (6)$$

$$V_{270}^2 = V^2 + V_t^2 - 2V\ V_t\ sin\varphi \qquad (7)$$

Equation (6) can be rewritten as following,

where $cos(\,90-\varphi\,) = sin\varphi$

By subtracting equation (5) from (2) and equation (7) from (4), results,

$$V_0^2 - V_{180}^2 = 4V\ V_t \cos\varphi \tag{8}$$

$$V_{90}^2 - V_{270}^2 = 4V\ V_t \sin\varphi \tag{9}$$

By dividing equation (9) by (8) results,

$$\frac{\sin\varphi}{\cos\varphi} = \tan\varphi = \frac{V_{90}^2 - V_{270}^2}{V_0^2 - V_{180}^2} \tag{10}$$

$$\varphi = \tan^{-1}\left[\frac{V_{90}^2 - V_{270}^2}{V_0^2 - V_{180}^2}\right] \tag{11}$$

thus;

By re-arranging equation (8), the value of (V_t) can be obtained as following,

$$V_t = \frac{V_0^2 - V_{180}^2}{4\ V \cos\varphi} \tag{12}$$

From equation (9), the value of (V_t) can also be written as a function of (V_{90}) and (V_{270}) as shown in equation (13);

$$V_t = \frac{V_{90}^2 - V_{270}^2}{4\ V \sin\varphi} \tag{13}$$

The purpose of calculating (φ) and (V_t) values is ultimately to determine the balancing mass ($\vec{M}$) and its location on the rotor that will eliminate the original vibration at the bearing.

The following equation express the relationship between the trial mass and the final correction mass [7],

$$\vec{M} = \vec{C}\ M_{trial} \tag{14}$$

where vector ($\vec{C}$) is defined as the correction factor to be applied to the trial mass on the plane of balancing.

Equation (14) can be written in a component form as following,

$$M_x = C_x\ M_{trial} \tag{15}$$

$$M_y = C_y\ M_{trial} \tag{16}$$

It is desired to determine the correction factor ($\vec{C}$) which when applied to the trial mass on the balancing plane, will cause the total change in vibration at the rotor bearing to offset the original unbalance at the bearing itself. This can be expressed as following,

$$\vec{C}\ \vec{V}_t = -\vec{V} \tag{17}$$

Equation (17) state mathematically that the proper location of (M) on the balancing plane will produce vectors equal to, but opposite in direction to the original vibration (V) thus eleminating the vibration.

Resolving equation (17) as a function of its (x) and (y) components gives,

$$C_x V_t = -V \cos\varphi \tag{18}$$

$$C_y V_t = -V \sin\varphi \tag{19}$$

By re-arranging equations (18) and (19), (C_x) and (C_y) values can be obtained,

$$C_x = \frac{-V \cos\varphi}{V_t} \tag{20}$$

$$C_y = \frac{-V \sin\varphi}{V_t} \tag{21}$$

The two individual required balancing masses (M_x) and (M_y) can now be determined from equations (15) and (16) respectively. To place these masses at the correct locations, the following hints must be kept in mind,

(x) component placed at (0^0) position if positive, placed at (180^0) position if negative.

(y) component placed at (90^0) position if positive, placed at (270^0) position if negative.

The resultant balancing mass (M) can also be calculated from equation (22) shown below as a function of (M_x) and (M_y),

$$M = \sqrt{M_x^2 + M_y^2} \tag{22}$$

Experimental Work:

This section presents the experimental test rig, vibration meter and experimental procedure of balancing.

1. Experimental Test Rig:

An actual arrangement of a simple experimental test rig is shown in figure (2). The test rig consists of a thin plastic disc, and a clockwise A.C. motor. The whole apparatus was mounted on a rigid steel cradle by means of M 11.5 – threaded bolts. The cradle weighs (30 kg) and has dimensions of (1390 x 160 x 50 mm). Figure (3) shows the plastic disc which is (295 mm) in diameter, has (108) equally spaced (M4) holes, (72) holes at radius (130 mm) and (36) holes at radius (65 mm) to allow for the addition of balancing mass . The disc rotates counter-clockwise by the motor which has a power of (0.5 hp) and a maximum speed of (1425 rpm). The motor is provided with a small base-plate which is bolted to the steel cradle. To ensure the motor remained rigid, it has been tightened down on to its base-plate by (4 bolts). In this arrangement, the disc has been operated to a speed of (1425 rpm).

Figure (2). Experimental test rig

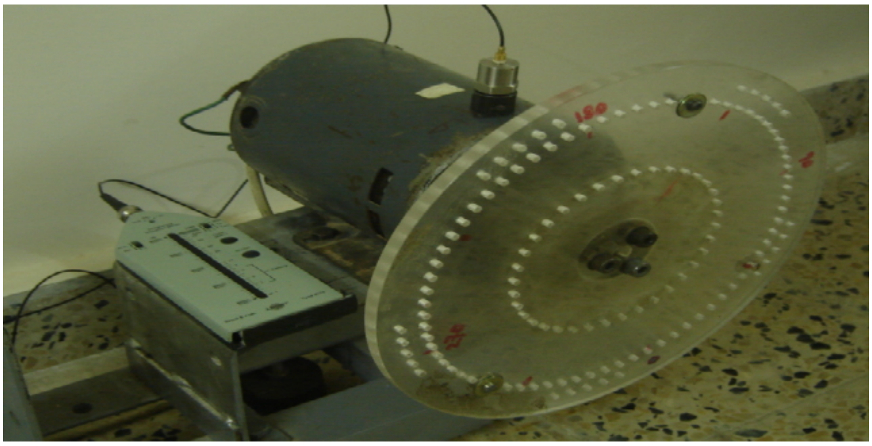

Figure (3). Plastic disc

2. Vibration Meter:

Figure (3) shows the vibration meter and piezoelectric accelerometer used for balancing process. The accelerometer is used for calculating the vibrational acceleration. This accelerometer weighs (103) gram with its magnetic base.

3. Experimental Procedure:

In this procedure, the motor speed of (1425 rpm) represents the running speed of the disc. Steps of the experimental balancing procedure of the disc are described in detail in Appendix (B).

Results and Discussion:

Table (1) shown below, gives the measured vibration levels in the measuring plane for the test speed corresponding to an original unbalance condition of the rotor, trial mass is placed in the balancing plane at (0^{o}), (90^{o}), (180^{o}) and (270^{o}) respectively and a radius of (130 mm).

Rotor speed (rpm)	Test number	Trial mass size and location (M_{trial} = 2.0 gram)	Vibrational acc. (m/sec^2)
1425		Original unbalance	2.90
	1	New original unbalance due to (6.0 gram) added at (150^{o})	6.05

	2	M_{trial} in balancing plane at (0^0)	5.50
	3	M_{trial} in balancing plane at (90^0)	7.70
	4	M_{trial} in balancing plane at (180^0)	7.20
	5	M_{trial} in balancing plane at (270^0)	4.90

Table (1). Measured vibration levels of the disc

The measured amplitude values of test (1 to 5) recorded in table (1), have been supplied as input data to the single – plane balancing program (See Appendix (B)), which is based on the amplitude readings. Test number (1) has been performed in order to creat a linearity for the test rig (i.e., original vibration is proportional to unbalance amount, where C is the proportionality constant). Table (2) shown below, gives the required balancing masses and their phase angles.

Balancing mass	Magnitude of balancing mass (gram)	Phase angle in the ccw direction (degree)
M_x	3.69532	0
M_y	6.03849	270
M	7.07946	301.465

Table (2). Calculated results of the rotor using computer program

The calculated balancing mass (*M*) was installed at its phase angle from the reference point at (0^0). Then the disc was brought up to the running speed of (1425 rpm) and the resulting vibrational acceleration was observed at the measuring point on the motor case.

Table (3) shown below, gives a comparison of vibration level of the disc before and after balancing.

Rotor speed (rpm)	Motor case		
	Vibrational acc. (m/sec^2)		Percentage improvement (%)
	Before balancing	After balancing	
1425	2.90	1.35	53.45

Table (3). Comparison of vibration level of the disc before and after balancing process

In commenting on table (3), the (2.90 m/sec^2) original unbalance value is reduced after balancing to the (1.35 m/sec^2) residual unbalance value which gives a good improvement in the form of reduction in the value of vibration at the test speed.

Conclusions:

The experimental test program used here has shown that a disc shaped rotor can be systematically balanced at its running speed. The single plane balancing method based on amplitude mesurements is found very effective, practical, successful and permitting safe rotor operation through its running speed. From just a single balancing run, an acceptable improvement of (53.45 %) has been obtained.

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Appendix (A)

Flow Chart of the Written Computer Program

In this appendix, figure (A) shown below represents a flow chart of the computer program written based on the adopted single-plane balancing method.

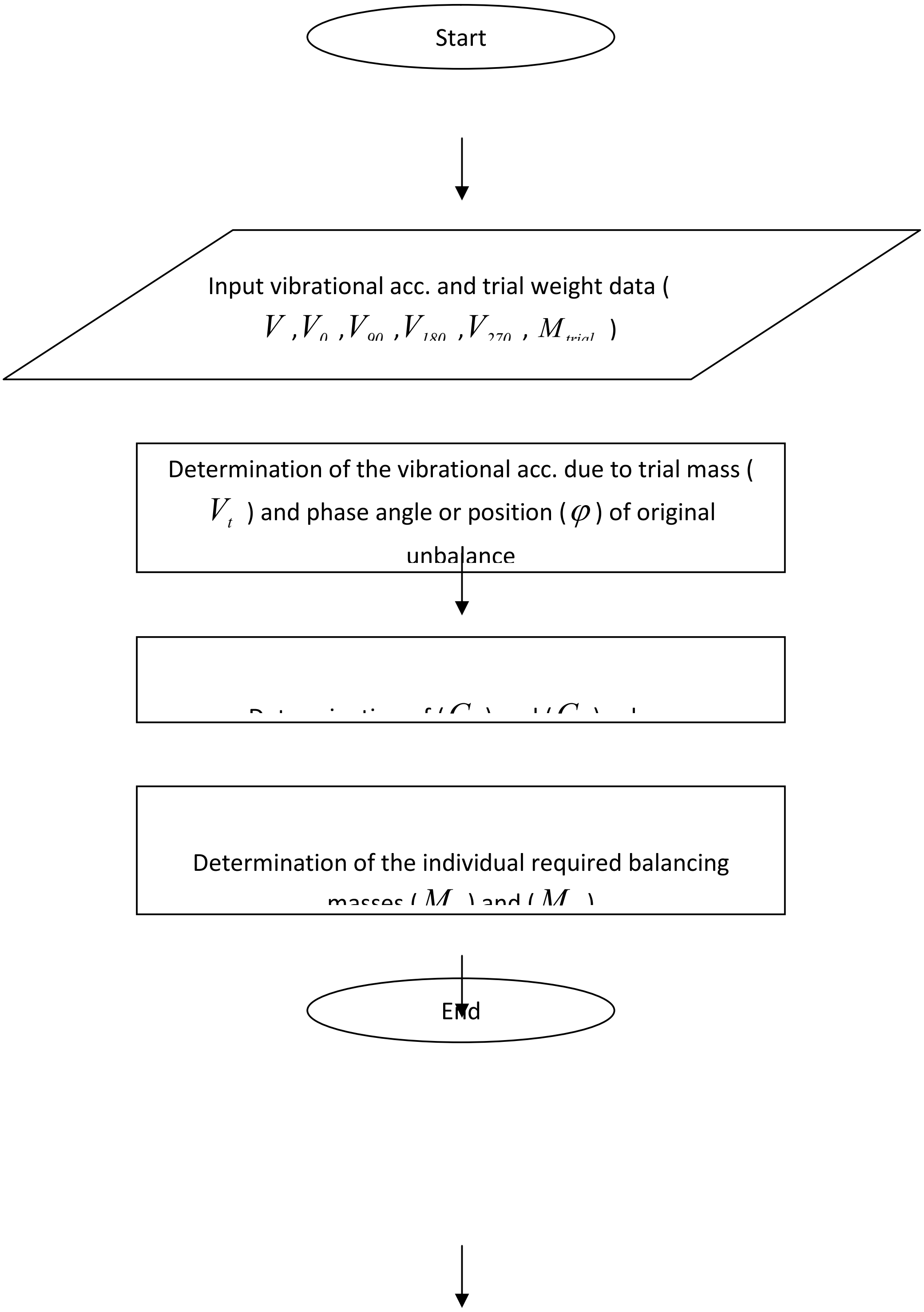

Figure (A). Flow Chart of Computer Program Based on the Single – Plane Balancing Method

Appendix (B)

In this appendix, steps of the experimental balancing procedure of the disc are described in detail as shown below.

a. The accelerometer was attached to the motor case in the vertical direction by means of a magnetic base (screwed to the accelerometer) as shown in figure (3).

b. First, in order to creat a linearity for the test rig along its rotation (i.e., original vibration is proportional to the amount of unbalance), a fixed mass of specific value (usualy greater than the trial mass value [6]) should be attached to the rotor along the balancing procedure at any arbitrary angle, thus a new original unbalance condition will be created (See test number (1) in table (1)).

c. The rotor was allowed to run to the test speed (i.e., 1425 rpm) for about five minutes to reach the steady state condition.

d. With only the new original unbalance condition, vibrational acceleration (V) was measured at the test speed.

e. The rotor was stopped, and the trial mass of (2.0 gram) was attached to the (0^{0}) position on the rotor. Zero degree is an arbitrary position chosen by the user. (The trial mass should be firmly attached to the rotor. It is a serious safety risk if the trial mass is not firmly attached and flies off while running the rotor).

For actual rotating part, the trial mass (i.e., initial balancing mass) value can be calculated using one of the well-known formulas (ISO1940, API 610 or ANSI), the following formula is defined by ISO1940 [9],

$$Balance\ Tolerance\,(gram - mm) = \frac{9.54\ *\ G\ number\ *\ Rotor\ mass\ (gram)}{RPM}$$

or

$$Trial\ mass\ \ value\,(gram) = \frac{9.54\ *\ G\ number\ *\ Rotor\ mass\ (gram)}{RPM\ *\ r\,(mm)}$$

where (G) is the balance quality grade of the rotor and (r) is the radius where the trial mass is fixed on the rotor.

f. The rotor was restarted and brought up to the test speed. Vibrational acceleration (V_0) was measured with the trial mass at (0^0).

g. The rotor was stopped and the trial mass was removed from the (0^0) position and reattached at the (90^0) position.

h. The rotor was restarted and brought up to the test speed. Vibrational acceleration (V_{90}) was measured with the trial mass at (90^0).

i. The rotor was stopped and the trial mass was removed from the (90^0) position and reattached at the (180^0) position.

j. The rotor was restarted and brought up to the test speed. Vibrational acceleration (V_{180}) was measured now with the trial mass at (180^0).

k. The rotor was stopped again and the trial mass was removed from the (180^0) position and reattached at the (270^0) position.

l. The rotor was restarted and brought up to the test speed. Vibrational acceleration (V_{270}) was measured with the trial mass at (270^0).

m. The rotor was stopped and the trial mass was removed from the (270^0) position.

The work was carried out at the college of Engg. University of Mosul