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Identification of imbalance of an alternating current motor and solution using bench balancing

Identificación de desbalance de un motor de corriente alterna y solución mediante balanceo en banco

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Abstract

Most dynamic equipment will have minimal vibration when it comes into operation; this can be influenced by everything from the poor placement of a screw to the uneven distribution of the body's mass. The vibration of a piece of equipment will generate a dynamic imbalance that will result in overheating, misalignment of the machinery, excessive wear, decreased useful life, high energy consumption, etc. This study evaluates the balancing of an alternating current motor and the important prior maintenance before performing a balancing, since it is extremely important not to confuse the imbalance with conditions that do not imply the irregularity of the rotor mass to proceed with carrying out dynamic balancing.

Dynamic Balancing, Mass, Maintenance

Resumen

La mayoría de los equipos dinámicos tendrán una mínima vibración al momento de entrar en operación, esto puede influir desde la mala colocación de un tornillo, hasta la distribución desigual de la masa del cuerpo. La vibración de un equipo generara un desbalance dinámico que traerá como consecuencias el sobrecalentamiento de éste, desajuste de la maquinaría, excesivo desgaste, disminución de la vida útil, alto consumo de energía, etc. En este estudio se evalúa el balanceo de un motor de corriente alterna y los mantenimientos importantes previos antes de realizar un balanceo, ya que es de suma importancia no confundir el desbalance con condiciones que no impliquen la irregularidad de la masa del rotor para proceder a realizar el balanceo dinámico.

Balanceo Dinámico, Masa, Mantenimiento

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Introduction

Dynamic unbalance is one of the most common causes of vibration in equipment, present in all machinery with rotating elements. This paper provides the essential information needed to solve most unbalance problems in rotating machinery using a bench balancing machine by means of the unbalance weight function. Vibration analysis by dynamic balancing is a predictive maintenance that has evolved over the years whose methodology has always been to analyse the dynamic state of the rotating component, as well as the parameters that generate a vibration of a machine and thus reduce its vibration through runs that will indicate the amount of unbalance it presents.

Vibration is not only caused by unbalance, there are other consequences that cause a machine to vibrate, such as bearing failure, eccentricity, pores in castings, resonance, tolerance clearances, these make up a small percentage when it comes to knowing that a machine has vibration. It has been found that the most common mechanical sources of vibration are unbalance and misalignment, these make up a large percentage of the vibration problems in rotating machinery.

Unbalance increases wear on both the equipment and its dynamic components, generating fatigue in the bearings, equipment housing and the foundation of the equipment causing the appearance of other problems such as mechanical looseness, bent shaft, bearing misalignment and eccentricity, as well as poor vibration transmission to the adjacent machine affecting the work and its performance.

The dynamic balancing on the bench is in charge of correcting and distributing the mass of the rotor that is not proportional to the axis of the centre of rotation, with the correction the centrifugal forces that are generated are eliminated and consequently the vibrations of the dynamic equipment are eliminated, allowing the equipment to operate correctly within the normative parameters helping in having a quality maintenance, as well as obtaining benefits in the equipment with long hours of operation, lower cost in maintenance, among others.

Unbalance

Unbalance is the unequal distribution of the mass of a rotor in reference to its centre of rotation line.

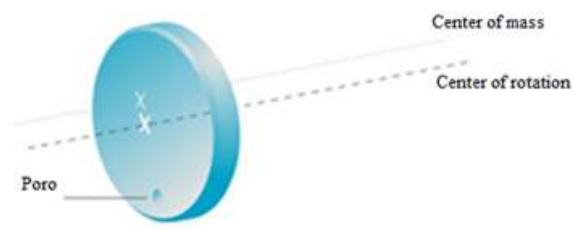


Figure 1 Unbalance of a dynamic body

According to the International Standards Organization (ISO), unbalance is that condition that exists in a rotor when its forces or vibratory motions are imparted to its bearings as a result of centrifugal forces.

Static Unbalance

The simplest case of unbalance, it is the condition where the principal axis of inertia is parallel to the axis of rotation, it occurs when there is a deviation of the centre of mass from the axis of rotation, but there is no angular deviation of the principal axis of inertia from the axis of rotation.

Figure 2 Static unbalance

Static unbalance in a rotating part can be detected by comparing the amplitude and phase of vibration of a bearing or shaft at the ends of the rotor.

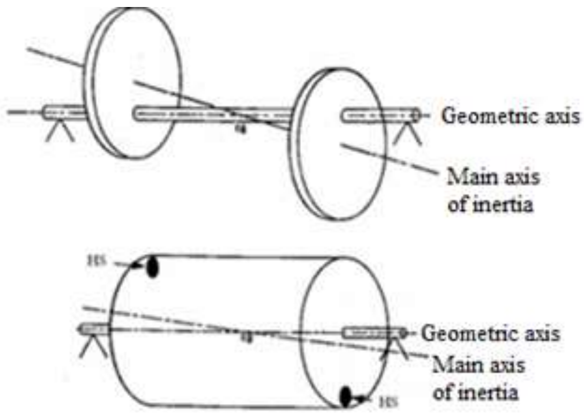
Torque unbalance

This type of unbalance occurs when the main axis of inertia intercepts the axis of rotation at the centre of gravity of the rotor.

A torque is two equal forces acting in opposite directions, not in the same plane.

ISSN-2531-2189

Article
Journal of Mechanical Engineering
December, 2023 Vol.7 No.20 1-8



Author: (Sosa, 2006

Figure 3 Force-torque imbalance

Quasi-static unbalance

The condition where the main centre line intersects the centre of rotation line, but not the centre of gravity of the rotor.

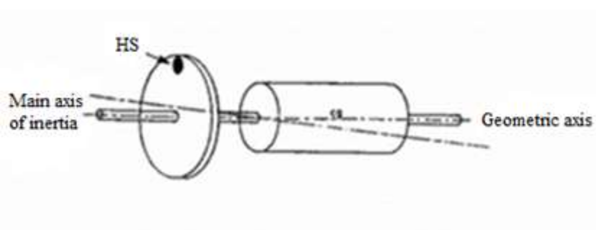


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Figure 4 Quasi-static unbalance

Dynamic Unbalance

Dynamic unbalance is the most common unbalance found in most rotors, occurring when the main axis of inertia does not intersect the axis of rotation, let alone run parallel to it.

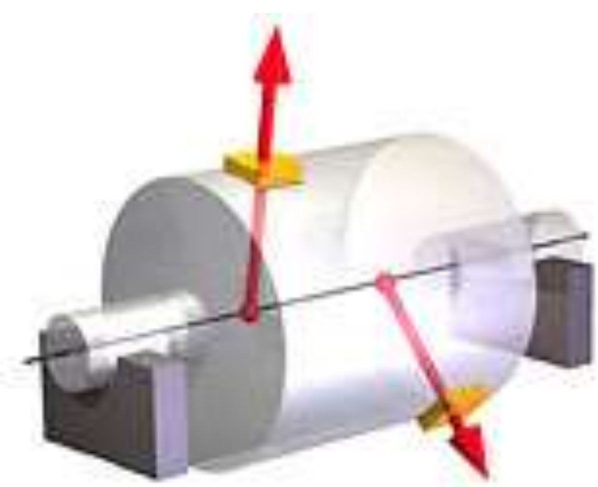

Figure 5 Dynamic unbalance

Dynamic unbalance shall be whenever static and torque unbalance are present, where static shall not be in direct line with torque.

Run Out

The run out technique allows us to verify the straightness of a shaft, it consists of placing the rotor in a horizontal lathe and centring it to zero at the points where it is verified, verifying the support point and then positioning the tailstock, subsequently a dial indicator is placed on the surface of the shaft and it is adjusted to 0, if there is deflection the needle will move clockwise indicating a high point, and counter clockwise indicating a low point.

It is important to know the state of the rotor shaft with this technique so as not to confuse rotor unbalance with any deflection of the rotor shaft.

Figure 6 Centring and verifying the straightness of the rotor on a vertical lathe

The NEMA (National Electrical Manufacturers Association) standard MG-1 4.9.7, indicates the tolerable values of run out.

DIMENSIONS IN INCHES		DIMENSIONS IN MILLIMETERS	
Shaft Diameter	Shaft Runout	Shaft Diameter	Shaft Runout
0.1875 to 1.625 incl.	0.002	4.76 to 41.3 incl.	0.051
Over 1.625 to 6.500, incl.	0.003	Over 41.3 to 165.1, incl.	0.076

Table 1 Permissible run out values for NEMA machines

The IEC (International Electrotechnical Commission) in standard 60072-1 C1.6 indicates its permissible run out values.

December, 2023 Vol.7 No.20 1-8

DIMENSIONS IN MILLIMETERS			DIMENSIONS IN INCHES		
Nominal Shaft Diameter		Shaft Runout*	Shaft Diameter		Shaft Runout*
Over	Up To		Over	Up To	
6	10	0.030	0.236	0.394	0.001
10	18	0.035	0.394	0.709	0.001
18	30	0.040	0.709	1.181	0.002
30	50	0.050	1.181	1.969	0.002
50	80	0.060	1.969	3.150	0.002
80	120	0.070	3.150	4.724	0.003
120	180	0.080	4.724	7.087	0.003
180	250	0.090	7.087	9.843	0.004
250	315	0.100	9.843	12.402	0.004
315	400	0.110	12.402	15.748	0.004
400	500	0.125	15.748	19.685	0.005
500	630	0.140	19.685	24.803	0.006

Table 2 Allowable run out values for NEMA machines

Balancing Machine

A dynamic balancing machine is essentially composed of a base plate assembly and an associated console or monitoring unit, two support pedestals (rigid or flexible) instrumented to load the rotating rotor at a constant speed, which is controlled by a motor and a transmission system.

Figure 7 IRD B-50 balancing machine

The effects caused by the unbalance are transmitted to the bearings of the bracket where the derived electrical signals are transmitted to the console unit, where the unbalance status is analysed and the calibrated correction values can be determined and displayed on the front panel.

Figure 8 IRD Model 295 balancing machine console

The balancing machine console is the control and analysis centre of the system, it must give the operator sufficient information to allow the permissible residual unbalance of the rotor to be balanced to be within the rotor tolerance. There are several suspension systems that are used to there are several suspension systems that are used to measure vibration, the most commonly used is the pendulum type because it provides a linear amplitude and frequency response.

Determination of Swing Plane

	Relationship L/D	Correction of the imbalance		
		One axis	Two axes	Multi-axis
	L/D<0.5	To 1000 RPM	Above 1000 RPM	Not applicable
	L/D>0.5 L/D<2	To 150 RPM	Above 150 RPM or above 70% of critical value	Above 2000 RPM or above 70% of its critical
	L/D>2	To 100 RPM	Above 100 RPM or above 70% of critical	Above 70% of its critical

Table 3 Choice of swing plane

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Single plane balancing is performed on narrow rotors, such as single runner turbines, fans, closed impellers, pulleys, etc. Two-plane balancing is performed on electric motor rotors, fans, blowers, semi-open and double suction impellers, turbochargers, etc. Multiplane balancing is often required on flexible rotors, rotors where two-plane balancing has proven to be insufficient, such as multi-stage pumps, multi-stage turbines, etc.

This procedure of doing the L/D ratio to find out the balancing plane is only a guide and should not be held as a true and should not be held to be true in all cases.

ISO 1940-1 Permissible Residual Unbalance

The ISO 1940-1 standard specifies the permissible residual unbalance that each rotating equipment has in order to have a reference value to be used when balancing the rotor.

$$Uper = \frac{G*6.015*w}{N}$$

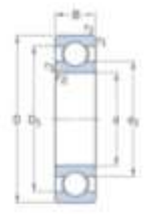
Where:
G=roll quality grade.
w=rotor weight in pounds.
N=speed at which the equipment operates.
6.015=equation constant.

ROTOR RATING (Degree of balancing accuracy)	ROTOR DESCRIPTION (Examples of general types)
G. 40	Automobile wheels, railroad wheels, cardan shafts. Sets of four-stroke engines (gasoline or diesel), with six or more cylinders; automobile, truck, locomotive engines.
G. 16	Automotive drive shafts, parts of agricultural machines and crushers, individual components of engines (gasoline or diesel) of automobiles, trucks or locomotives, parts of engines with six or more cylinders under special requirements.
G. 6.3	Special requirements drive shafts, processing machinery rotors, centrifugal drums, fans, flywheels, centrifugal pumps, machinery parts, general machinery and standard electric motor armatures.
G. 2.5	Gas and steam turbine, blowers, turbine rotors, turbogenerators, drive units for machines, tools, medium and large armatures for special requirement engines.
G. 1 PRECISION BALANCING	Jet and supercharger engine rotors, recorder and turntable drive units, special-purpose fractional horsepower engine armatures
G. 0.4 ULTRA-PRECISE BALANCING	Armatures, shafts and moldings of precision grinding machines.

Table 4 Degree of balancing accuracy

Verification of bearing fits and Run Out

A motor uses SKF 6314 bearings at both ends, non-locating side and coupling side, the dimensions are as follows:



D	5.905"	150 mm
d	2.755"	70 mm
D1	5.114"	129.9 mm
d1	3.738"	94.95 mm
B	1.377"	35 mm
$r_{1,2}$	0.082"	2.1 mm

Table 5 SKF 6314 bearing dimensions

As shown in the table the value of the bearing inner diameter is 70mm where ISO 286 states that the fit for normal and heavy load ball bearings in electric motors with shaft diameters between 18mm and 100mm is k5, this is equivalent to 0.015mm=0.0005", therefore the bearing will have an interference of half a thousandth with the trunnion.

Figure 9 Measuring points on the rotor

	Ø Outside	Ø Inside	Adjustment	Condition
Coupling side (A)	2.7555	2.755"	-0.0005	Acceptable
Coupling side (B)	2.7556	2.755"	-0.0006	Acceptable

Table 6 Electric motor bearing adjustment

Once the bearing supports have been adjusted and are in an acceptable condition, the next step is to check the straightness of the bearing acceptable conditions, the next step is to check the straightness of the motor.

The motor is mounted on the lathe and centred to zero with the dial indicator to check the straightness of the motor.

a	b	c	d	e
0.000	0.000	+0.001	0.0017	0.0012

Table 7 Run Out test readings

The readings at point d (free side of the rotor) have run out readings of 0.0017", however, these readings are within the permissible run out values for electric motors, according to Nema MG-1 4.9.9, according to Nema MG-1 4.9.7 and IEC 60072-1 C 1.6 standards, under these standards the rotor does not need to be machined the rotor does not need to be machined.

ISSN-2531-2189

Dynamic Balancing

Dynamic balancing is the process of reducing the vibration of equipment that generates excess mass at a point or points on the rotor, causing its centre of mass to move away from its geometric centre of rotation.

Before balancing an equipment rotor, whether it is a motor, pump, blower, etc., the rotor must be disassembled from the stator, as for balancing it is necessary to have only the rotor with its dynamic parts of the equipment accessories, fans, wedges, couplings, etc., as well as cleaning the components that rotate with it, following these steps will help to avoid obtaining erroneous amplitude readings and ensure quality maintenance.

– Total cleaning of the fan.
– Removal of rust where present.
– Tapping of the bores.
– Cleaning with solvent on the shaft to remove impurities.
– Sanding of the wedges to remove impurities.

Once this procedure has been carried out, the assembly of the components of the electric motor takes place. Such as shims, fan, bulkhead, etc. Once it has been verified that no part of the equipment is loose, the balancing machine is raised. In some cases it is necessary to compensate the mass of the wedges as it is a missing component, in this case it was not necessary to compensate the wedges.

Analysis of results

This balancing was performed in two planes, because the ratio of the length between the diameter is greater than 0.5 and less than 2. In some cases the balancing technician has the experience and knowledge to omit the balancing and knowledge capable of omitting the calculation to know the plane to be balanced.

$$\frac{L}{D} = \frac{9.75''}{5.25''} = 1.85$$

For the balancing of equipment it is necessary to add a test weight for each run that is given to the balancing equipment, this will help to rectify and verify if the unbalance readings are changing according to the weight added.

It is necessary to add a weight that the balancing machine can detect by means of the readings of each run, for that a formula is used to calculate the test weight:

$$m = \frac{Fc}{r * \left(\frac{n\pi}{30}\right)^2}$$

Where:
Fc=centrifugal force generated by the test weight.
r=correction radius of the test weight.
m=test weight in kg.
n=rolling speed in RPM.

The centrifugal force is calculated from the following data:

Rotor weight=70kg=154lb
Correction radius=4.75"=0.12065m

$$Fc = \left(\frac{70}{2}\right) * 9.81\frac{m}{s^2} * 0.1 = 34.33N$$

Rolling speed=600rpm

$$m = \frac{Fc}{r * \left(\frac{n\pi}{30}\right)^2}$$

$$m = \frac{34.33N}{0.12065m * \left(\frac{600\pi}{30}\right)^2}$$

$$m = 0.07207kg = 72gr$$

Equipment:
86kW squirrel cage electric motor.
Operating speed 1800 RPM
Rolling speed 600 RPM

Having the data we now substitute to calculate the permissible residual unbalance of the rotor:

$$Uper = \frac{G * 6.015 * w}{N}$$

$$Uper = \frac{2.5 * 6.015 * 154lb}{1800RPM} =$$

$$= 1.286\, oz.in * \frac{28.3gr}{oz} = 36.409gr.in$$

In this case that is balancing in 2 planes, which indicates that the permissible residual unbalance value is divided by 2, resulting in the following way:

Article
Journal of Mechanical Engineering
December, 2023 Vol.7 No.20 1-8

$$Uper = \frac{36.409\ gr.in}{2} = 18.204\ gr.in$$

To know the amount of allowable mass in both planes, simply divide the result of the allowable unbalance by the radius to be corrected..

$$Uper = \frac{18.204\ gr.in}{4.75\ in} = 3.832\ gr$$

Once the unbalance weight of the rotor is known, the rotor is mounted on the bench balancer and the equipment is installed in order to start the balancing runs.

Figure 10 Rotor mounted on machine and in full swing

Initial vibration readings of the equipment:
Left plane=0.343 mils 119°.
Right plane =1.30 mils 136°.

The 72 g plasticine test weight is placed on the left plane at 0° (in line with the reflective tape) and the second run is carried out:

Left plane=1.93 mils 307°.
Right plane=1.14 mils 135°.

The mass placed in the left plane is removed and placed in the right plane at 0° for the third run:

Left plane=0.494 mils 121°.
Right plane=1.13 mils 299°.

At the end of this run, the balancing brain IRD 290 or IRD 295 indicates the values and the weight of the imbalance in each plane, so that it can be corrected by removing or adding mass at the indicated angles:

Values given by the brain:
Left plane=4.36 gr 356°.
Right plane=13.4 gr 8°.

We proceed to add the mass and at the angle indicated in each plane to proceed to make one more run and verify the data:

Left plane=0.234 gr 269° Left plane=0.234 gr 269° Right plane=2.17 gr 269
Right plane=2.17 g 95°.

The readings obtained in this run are within the ISO 1940 norm for this rotor, however, 1 gr more could be added in each plane to further lower the rotor's unbalance, obtaining the following readings:

Left plane=0.449 gr 320°.
Right plane=1.12 gr 82°

Balance axes	Initial vibration readings (mils and grs)	Final vibration readings (mils and grs)	Value of the permissible residual unbalance of the equipment according to ISO 1940-1 standard	Percentage of initial to final vibration reduction comparison
Left axis	0.343 mils 119° 4.36 grs 356°	0.044 mils 77° 0.449 grs 320°	3.83 grs	87%
Right axis	1.30 mils 136° 13.4 grs 8°	0.108 mils 198° 1.12 grs 82°	3.83 grs	91%

Table 8 Dynamic unbalance

With these values obtained, the rotor is balanced and ready to be assembled and put into operation.

Conclusions

In the industrial field, vibrations are synonymous with mechanical problems whose symptoms can be reflected in various aspects such as wear, efficiency in the work of the equipment, noise, structural damage and in extreme situations, accidents in the area of operation of the equipment.

This work provides a real problem of an electric motor to know the diagnostics of unbalance with real aspects that must be considered. It is of great help to relate theory with practice, enriching the reader's knowledge and this method can be used with the balancing programmes that are currently on the market with bench balancers.

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December, 2023 Vol.7 No.20 1-8

In the end, the results were favourable and it was demonstrated that bench balancing with correction masses helped to reduce 85% of the vibration amplitude, with initial readings of 0.343 mils in the left plane and 1.30 mils in the right plane, with final results of 0.044 mils in the left plane and 0. 108 mils in the right plane, as well as in the unbalance mass readings with initial readings of 4.36 grams in the left plane and 13.4 grams in the right plane, thus reducing the readings to 0.449 grams in the left plane and 1.12 grams in the right plane, leaving the equipment within the standards according to ISO 1940-1.

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