

Kumar Abhishek*, Naman Kumar Jain, Siddharth Agnihotri

*Department of Mechanical Engineering,
Krishna Institute of Engineering and Technology, Ghaziabad, India
Department of Mechanical Engineering,
Krishna Institute of Engineering and Technology, Ghaziabad, India
SPXFLOW INC., Ghaziabad, India

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ABSTRACT

Vibration due to mass imbalance in rotating parts such as in pump impellers is an important engineering problem. It is a very important factor to be considered in modern machine design, especially where high speed and reliability are significant considerations. Due to mass imbalance excessive vibration occurs and may lead to decrease in fluid velocity and pressure which may cause cavitation in the pump. Reducing rotor vibrations generally increases the service life of the rotating machinery.

International Standard ISO 1940/1 is a widely accepted reference for selecting rigid rotor balance quality. This paper presents effect of balancing a Boiler Feed Pump (BFP) on two different grades G6.3 and G2.5 as specified by ISO 1940/1 and their effect on the pump performance considering the vibration as the parameter.

KEYWORDS: : ISO1940/1, Centrifugal pump, Vibrations, quasi static unbalance.

INTRODUCTION

Pumps are generally classified as Centrifugal pumps and Positive Displacement pumps. [1]

Dynamic pumps:

- Centrifugal pump
- Axial pump

Positive Displacement Pumps:

- Rotary pump
- Reciprocating pump

Classification of Centrifugal Pumps:

- Overhung type
- Between Bearing
- Vertically Suspended

Table 1 — Pump classification type identification

Pump type ^a		Orientation		Type code		
Centrifugal pumps	Overhung	Flexibly coupled	Horizontal	Foot-mounted	OH1	
				Centreline-supported	OH2	
			Vertical in-line with bearing bracket	—	OH3	
			Rigidly coupled	Vertical in-line	—	OH4
		Close-coupled	Vertical in-line	—	OH5	
			High-speed integrally geared	—	OH6	
	Between-bearings	1- and 2-stage	Axially split	—	BB1	
			Radially split	—	BB2	
		Multistage	Axially split	—	BB3	
			Radially split	Single casing	BB4	
				Double casing	BB5	
	Vertically suspended	Single casing	Discharge through column	Diffuser	VS1	
				Volute	VS2	
			Axial flow	VS3		
			Separate discharge	Line shaft	VS4	
		Cantilever		VS5		
		Double casing	Diffuser	—	VS6	
Volute			—	VS7		

^a Illustrations of the various types of pump are provided in 4.2.2.

Fig 1. Classification of pump

Applications of Centrifugal pumps:

- Thermal power plants
- Nuclear power plants
- Oil and gas Industry
- Marine Industry
- Paper, sugar, and cement plants
- Waste supply
- Agricultural purposes
- Desalination Industry[2]

A centrifugal pump converts input power to kinetic energy by accelerating liquid in a revolving device - an impeller. The most common is the volute pump - where fluid enters the pump through the eye of the impeller which rotates at high speed. The fluid accelerates radially outward from the pump casing and a vacuum is created at the impellers eye that continuously draws more fluid into the pump. The energy from the pumps prime mover is transferred to kinetic energy according the Bernoulli Equation. The energy transferred to the liquid corresponds to the velocity at the edge or vane tip of the impeller. The faster the impeller revolves or the bigger the impeller is, the higher will the velocity of the liquid energy transferred to the liquid be.

Rotating machinery is commonly used in mechanical systems, including machining tools, industrial turbo machinery, and aircraft gas turbine engines. Vibration caused by mass imbalance is a common problem in rotating machinery. Imbalance occurs if the principal axis of inertia of the rotor is not coincident with its geometric axis. Higher speeds cause much greater centrifugal imbalance forces, and the current trend of rotating equipment toward higher power density clearly leads to higher operational speeds. For example, speeds as high as 30,000 rpm are not rare in current high-speed machining applications. Therefore, vibration control is essential in improving machining surface finish; achieving longer bearing, spindle, and tool life in high-speed machining; and reducing the number of unscheduled shutdowns. A great cost savings for high-speed turbines, compressors, and other turbomachinery used in petrochemical and power generation industries can be realized using vibration control technology.[3]

Mass imbalance in rotors play a significant role in producing vibrations and affects normal operation of machine and subsystems. Due to excessive vibrations subsystems may fail if the vibration level reaches resonance value. Thus proper balancing of rotor becomes essential for proper functioning and longer life of rotating machinery.

Types of unbalance:

Static Unbalance:

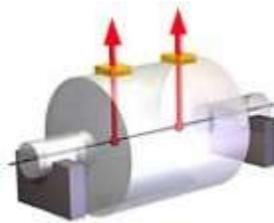


Fig 2.

Two unbalances (shown here as arrows in fig 2) can have the same size and angular position and can be the same distance from the centre of gravity. The same condition results for an individual, twice as large, unbalance that acts at the centre of gravity, i.e. in this case in the middle of the rotor. If such a rotor is supported on two knife edges, it would swing until the "heavy point" was facing downwards. This means this unbalance acts even without rotation; it is therefore called "static unbalance". It causes the centre of mass to shift away from the geometric centre, which in turn causes the rotor to swing oscillate parallel to its rotational axis when it is running.[4]

Couple Unbalances:

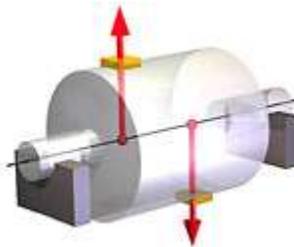


Fig 3

Two unbalances (shown here as arrows in fig 3) can have the same size, but their angular position is offset by precisely 180° to each other. This unbalance distribution cannot be detected by swinging, because the rotor does not take up a unique position at rest. The rotating rotor executes a wobbling movement about its vertical axis (perpendicular to the axis of rotation), because the two unbalances exert a moment. This type of unbalance distribution is called couple (moment) unbalance.[5]

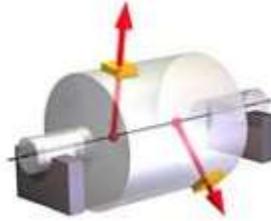


Fig 4.

A real rotor does not only have a single unbalance, theoretically it has an infinite number arbitrarily distributed along the axis of rotation. These can be replaced with two resulting unbalances (shown here as arrows) in two random planes, which generally have different sizes and angular positions. As this unbalance condition can only be fully determined during rotation, it is called dynamic unbalance. It can be broken down into a static unbalance and a couple unbalance, whereby one or the other can be the overriding unbalance.[6]

Quasi-Static Unbalance:

A special form of dynamic unbalance in which the static and couple unbalance vectors lie in the same plane. The central principal axis intersects the axis of rotation, but the mass center does not lie on the axis of rotation. This is the case where an otherwise balanced rotor is altered (weight added or removed) in a plane some distance from the mass center. The alteration creates a static unbalance as well as a couple unbalance. Conversely, a rotor with quasi-static unbalance can be balanced with a single correction of the right magnitude in the appropriate plane.[7]

Causes of Unbalance:

There are many reasons that unbalance may be present in a rotor. The most common causes are described briefly in following paragraph.

Blow Hole in Castings:

On occasion, cast rotors such as pump impeller will have blow holes or sand traps which result from the casting process. Blow holes may be present within the material, undetectable through normal visual inspection. Nevertheless, the void created may represent a truly significant unbalance.

Eccentricity:

Eccentricity exists when geometric centre line of a part does not coincide with its rotating centerline. The rotor itself may be perfectly round; however, for one reason or another the center of rotation has been located “off center”

Addition of Keys and Keyways:

Unfortunately, there are few industry wide standards regarding the addition of keys when component balancing. A motor manufacturer may balance his product with a full key or half key or perhaps no key at all. Thus, if a pulley manufacturer balances a pulley without a key, and a motor manufacturer balances his motor without a key; when the two components are assembled with a key, unbalance will result. Similarly if both were to balance their products with a full key, the assembled units would be unbalanced.

Distortion:

Although a part may be well balanced following manufacture, there are several influences which may serve to distort or otherwise change the shape of rotor to alter its original balance. Common causes of such distortion includes stress relief and

thermal distortion. Stress relieving is sometimes a problem with rotors which have been fabricated by welding. Actually any part that has been shaped by pressing, drawing, bending, extrusion etc, will naturally have high internal stresses. If the rotor or component parts are not stress relieved during manufacturing, they may undergo this process naturally over a period of time, and as a result rotor may distort slightly to take a new shape. Thermal distortion may require the rotor to be balanced at its normal operating temperature, even though it may have been well balanced when it was cold.

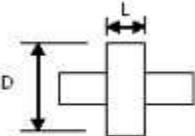
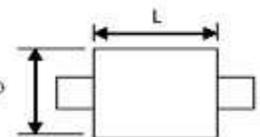
Types of Balance correction:

These correction methods are for re-distributing a rotor’s mass so as to better align the central principle inertia axis with the axis of rotation. The two most common methods employed for rigid rotors are Right-Left and Force-Couple. A balance computer will normally display balance corrections in one or both of these methods. When calculated correctly, both methods will have identical effects on a rigid rotor. Any condition of unbalance can be corrected by applying or removing weight at a particular radius and angle. The magnitude of a balance correction is correctly stated in terms of a weight, w, at a radius, r. The product of weight and radius are unbalance, U.

$$U = w \cdot r$$

The strategic addition or removal of weight redistributes the mass, altering the mass properties to better align the mass center and the central principal axis with the axis of rotation.

Single and Double Plane balancing:

Rotor	L/D Ratio	Balance Correction	
		Single Plane	Two Plane
	Less than 0.5	0 - 1,000 RPM	Above 1,000 RPM
	More than 0.5	0 - 500 RPM	Above 500 RPM

Note: RPM stands for revolutions per minute, is machine operating speed.

Fig 5.[8]

Single plane balancing is approved to be satisfactory for rotors of thin rigid disc type but unreliable for rotors of elongated rigid body type. For rotors of latter type double plane balancing is recommended. Another name for double plane balancing is dynamic balancing since it results in the cancellation of both unbalanced forces and moments.

If the thickness of rotor is more then two plane balancing i.e dynamic balancing is done because effect of adding or removing the mass from one correction plane is not so pronounced on its adjacent correction plane so mass has to be removed or added to both the correction planes.

If the thickness of rotor is less i.e. rotor is thin then single plane balancing is preferred i.e static balancing is done because being thin rotor, effect of adding or removing mass from one correction plane is appreciable on its adjacent correction plane. So mass is added or removed from one correction plane only.

High speed machines such as multistage centrifugal pumps, compressors, steam and gas turbines may require balance corrections be made in several planes and are often designed with multiple correction planes.

LITERATURE REVIEW

Amit Kalmegh and Santosh Bhaskar in their paper “Dynamic Balancing of Centrifugal Pump Impeller” showed experimentally how to balance a centrifugal pump impeller and keep its unbalance within specified limits in grade 6.3 as specified by ISO 1940/1 .[9]

Earl M. Halfen in his paper “Shop Balancing Tolerances A Practical Guide” compared balancing a typical shop rotor of 1500 lbs at service rpm of 4000 in API 617, MIL-STD-167-1 (U.S. NAVY), ISO 1940/1 (INTERNATIONAL STANDARDS ORGANIZATION), and API (AMERICAN PETROLEUM INSTITUTE). He found that API has more stringent tolerances followed by US Navy, API 617 and ISO 1940/1.[10]

Randall L. Fox in his paper “Dynamic Balancing” discussed various causes of balancing and how the same can be reduced. He discussed various methods of balancing , Single Plane Balancing, Single Plane Vector method of Balancing and Dynamic Balancing in the specified grade.[11]

Stefan Florjancic and Arno Frei in their paper “Dynamic loading on pumps- causes for vibrations” showed that there are two major causes responsible for pump vibration- Mechanical and Hydraulic. Out of which mechanical unbalance forces increases linearly with the shaft speed and the rotor mass and hydraulic forces increase with square of shaft speed. Dynamic unbalance forces have to be taken into account at higher speeds and higher rotor masses.[12]

The proper limit is dependent on many factors including rotor mass, speed and the application in which the machine is used. It would be uneconomical to exaggerate balancing quality requirements. However, if the balancing quality is underestimated, it would reduce machine reliability and availability. Moreover, sometimes demanding overqualified balancing reduces machine availability by consuming necessary time in unnecessary balancing. So, the balancing quality limit should not be oversimplified and given through vibration readings only. This is especially true for new machines for which no pervious vibration experience exists.[13]

Vibration problems on a centrifugal pump can result from a multitude of possible parameters which are not easily identified. Yet, in order to find a remedy, the cause of the vibration must be first understood and found.

Generally there are two principal areas to be investigated- either the dynamic pump rotor-casing-base plate system is resonant or close to resonance(at a critical speed or at a bearing housing resonance) or the forces driving the vibrations are excessive. Hence based on the nature of the problem, either the system dynamics or the loading forces need to be identified and reduced.

Balancing on ISO 1940/1, G6.3 and G2.5:

The International Standards Organization (ISO) publishes several standards which are the global benchmark for industrial balancing.

The ISO standards contain detailed methods of calculating different static and couple unbalances tolerances that are dependent on the ratio of the part's diameter to its length.

The ISO also specifies a **Balance Quality Grade**. This is a term used to define the limits of residual unbalance. It represents the product of the eccentricity (in millimeters) times the operating frequency (in Hertz).

A balance quality grade of G6.3 is suggested for balancing pump impellers however at **Precision Balancing** we exceed these standards, and we offer **G2.5 grade** balancing in most cases.[14]

Boiler Feed Pump (BFP) rotor was balanced in two different grades G6.3 and G2.5 and after balancing unbalance left and vibration readings were analyzed for both the grades.

A **boiler feed water pump** is a specific type of pump used to pump feed water into a *steam boiler*.

Table 1 Balance quality grades for various groups of representative rigid rotors (From ISO 1940/1)

Balance Quality Grade	Product of the Relationship $(e \cdot \omega)^{1-2}$ mm/s	Rotor Types - General Examples
G 4 000	4 000	Crankshafts/drives ³ of rigidly mounted slow marine diesel engines with uneven number of cylinders ⁴
G 1 600	1 600	Crankshafts/drives of rigidly mounted large two-cycle engines
G 630	630	Crankshafts/drives of rigidly mounted large four-cycle engines Crankshafts/drives of elastically mounted marine diesel engines
G 250	250	Crankshafts/drives of rigidly mounted fast four-cylinder diesel engines ⁵
G 100	100	Crankshafts/drives of fast diesel engines with six or more cylinders ¹ Complete engines (gasoline or diesel) for cars, trucks and locomotives ²
G 40	40	Car wheels, wheel sets, wheel sets, drive shafts Crankshafts/drives of elastically mounted fast four-cycle engines with six or more cylinders ¹ Crankshafts/drives of engines of cars, trucks and locomotives
G 16	16	Drive shafts (propeller shafts, cardan shafts) with special requirements Parts of crushing machines Parts of agricultural machinery Individual components of engines (gasoline or diesel) for cars, trucks and locomotives Crankshafts/drives of engines with six or more cylinders under special requirements
G 6.3	6.3	Parts of process plant machines Marine main turbine gears (merchant service) Centrifuge drives Paper machinery rolls/print rolls Fans Assembled aircraft gas turbine rotors Flywheels Pump impellers Machine-tool and general machinery parts Medium and large electric armatures (of electric motors having at least 80 mm shaft height) without special requirements Small electric armatures, often mass produced, in vibration insensitive applications and/or with vibration-isolating mountings Individual components of engines under special requirements
G 2.5	2.5	Gas and steam turbines, including marine main turbines (merchant service) Rigid turbo-generator rotors Computer memory drums and discs Turbo-compressors Machine-tool drives Medium and large electric armatures with special requirements Small electric armatures not qualifying for one or both of the conditions specified for small electric armatures of balance quality grade G 6.3 Turbine-driven pumps
G 1	1	Tape recorder and phonograph (gramophone) drives Grinding-machine drives Small electric armatures with special requirements
G 0.4	0.4	Spindles, discs and armatures of precision grinders Gyroscopes

1 $e = 2e \cdot \omega = r \cdot \omega^2$, if r is measured in revolutions per minute and ω in radians per second.
 2 For allocating the permissible residual unbalance to correction planes, refer to "Allocation of U_{res} to correction planes."
 3 A crankshaft/drive is an assembly which includes a crankshaft, flywheel, clutch, pulley, vibration damper, rotating portion of connecting-rod, etc.
 4 For the purposes of this part of ISO 1940/1, slow diesel engines are those with a piston velocity of less than 9 m/s; fast diesel engines are those with a piston velocity of greater than 9 m/s.
 5 In complete engines, the rotor mass comprises the sum of all masses belonging to the crankshaft/drive described in note 3 above.

Fig 6.

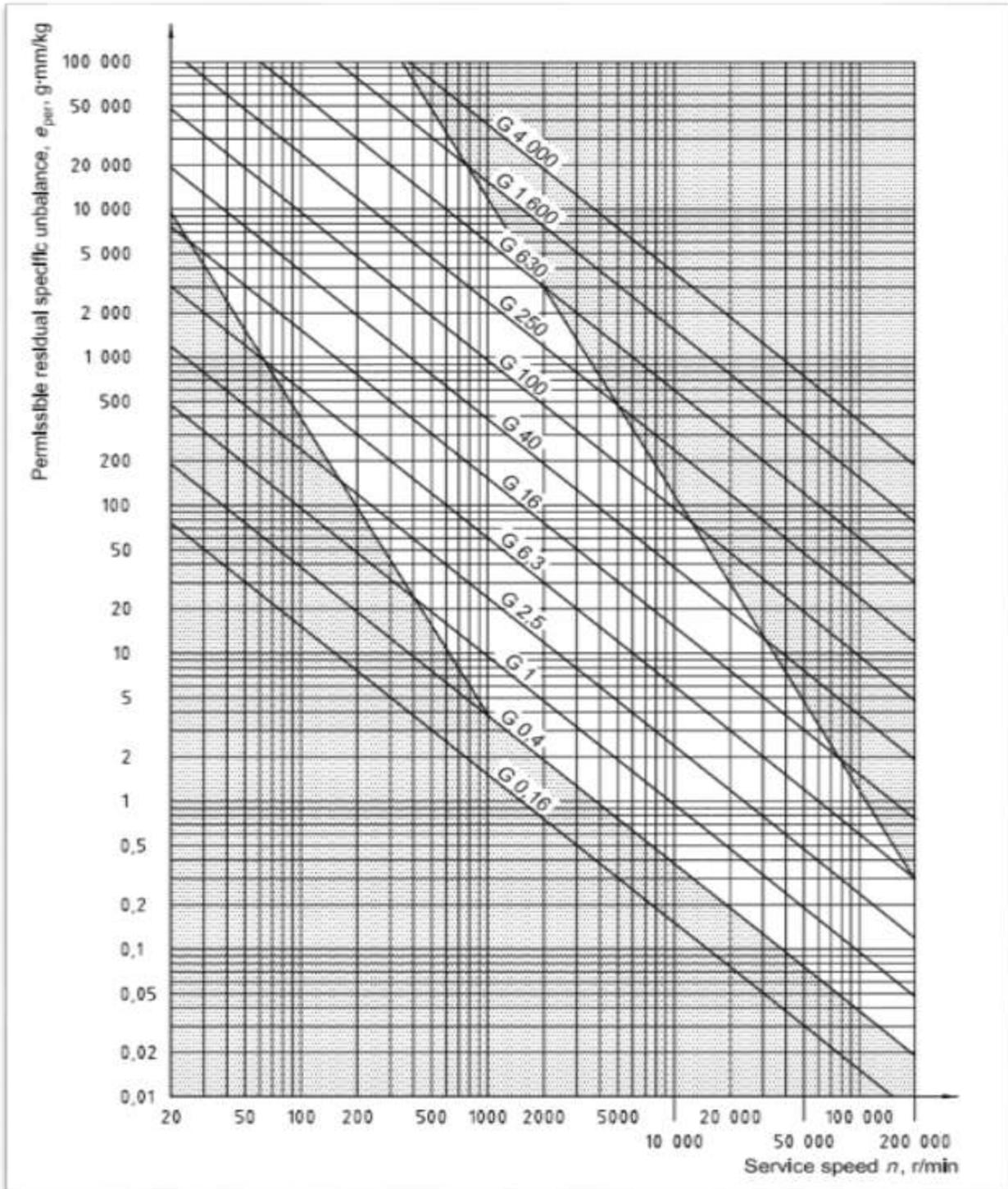


Fig 7.



Fig 8.Boiler feed pump rotor

RESULTS AND DISCUSSION

Balancing BFP rotor in G6.3 according to ISO 1940/1:

Rotor type data: Boiler Feed Pump

Balancing mode: Dynamic

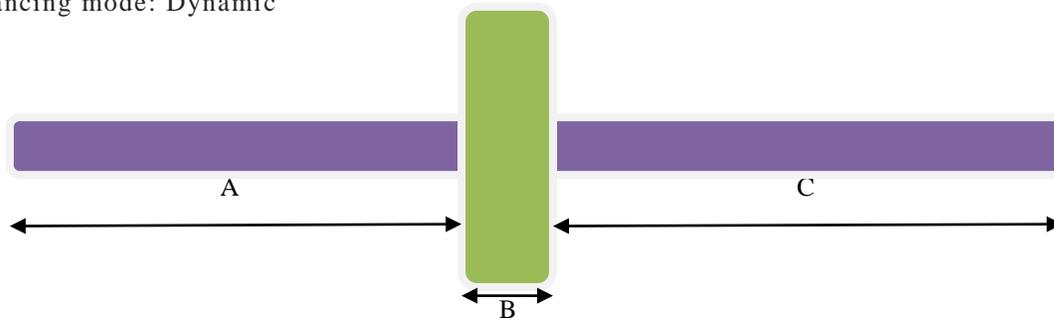


Fig 9.

A=900mm

B=240mm

C=890mm

Balancing radius (R1) = 160mm

Balancing radius (R2) = 160mm

Balancing Speed (RPM) = 6000

Tolerance (In correction planes) According to ISO 1940

Rotor mass: 490.0 kg

Quality grade: G6.3

Operating speed: 6000 RPM

Deviation (+/-): 0.00%

Tolerance allocation: symmetrical

Plane: Left Right

Distribution: polar polar

Correction method:

Plane: Left Right

Method: Masses Masses

Material: Remove Remove

Measuring results:

Run: 1 Measure speed: 718 RPM

Left: -12.6 g at 52°

Right: -14.5g at 313°

Total unbalance mass left on both balancing planes after balancing: 27.1g

Residual specific unbalance (e_{per}) =

$$\frac{(\text{Total unbalance mass left} * \text{Balancing radius})}{\text{Rotor mass}}$$

Rotor mass

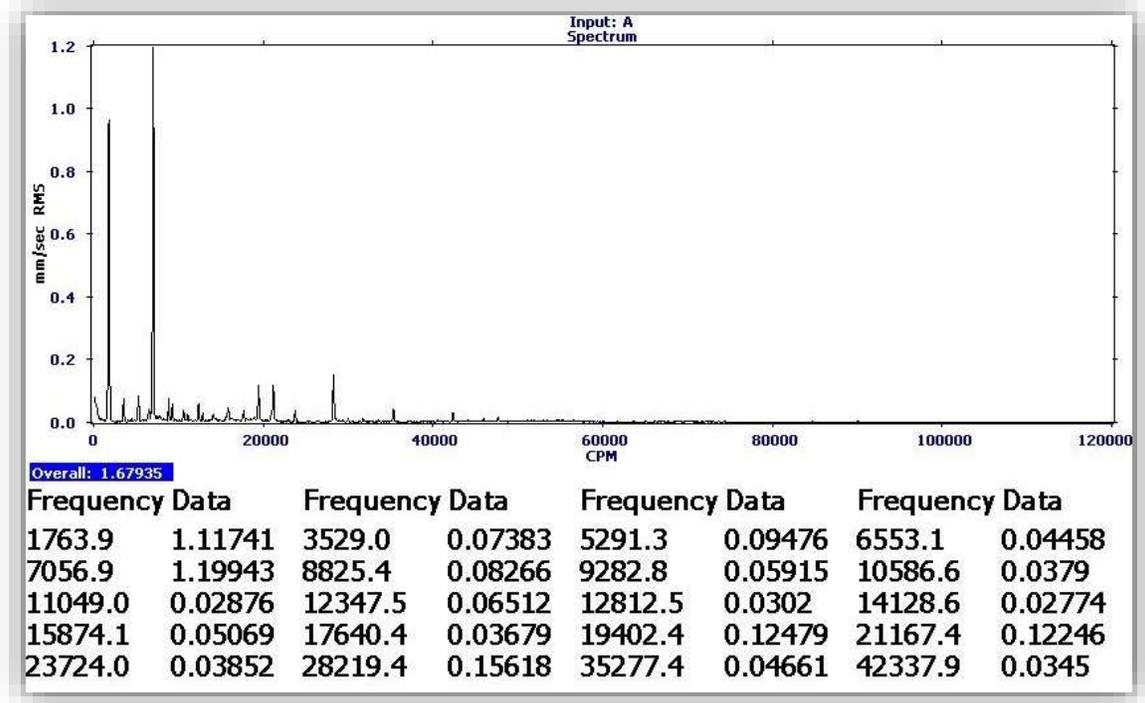
$$=(27.1*160)/490$$

$$= 8.84 \text{ g.mm/kg}$$

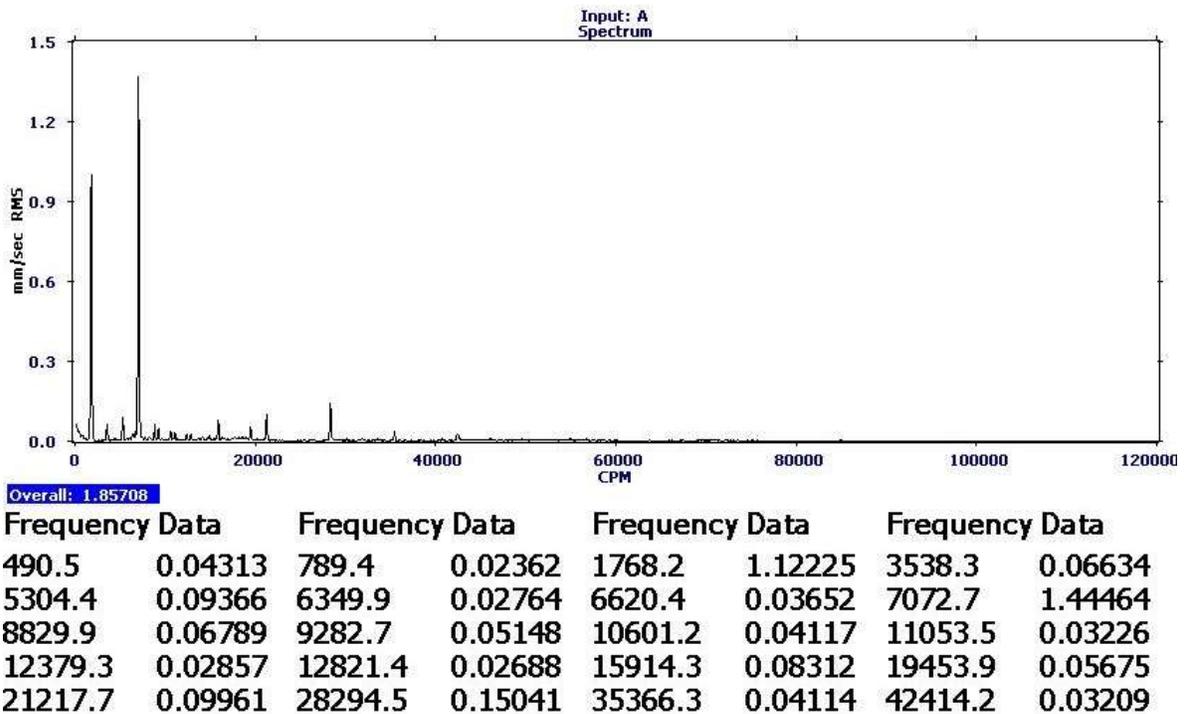
This is the residual specific unbalance left in BFP rotor after balancing.

Permissible residual unbalance (e_{per}) as per ISO 1940/1 G6.3 at the service speed of 6000RPM = 10 g.mm/kg

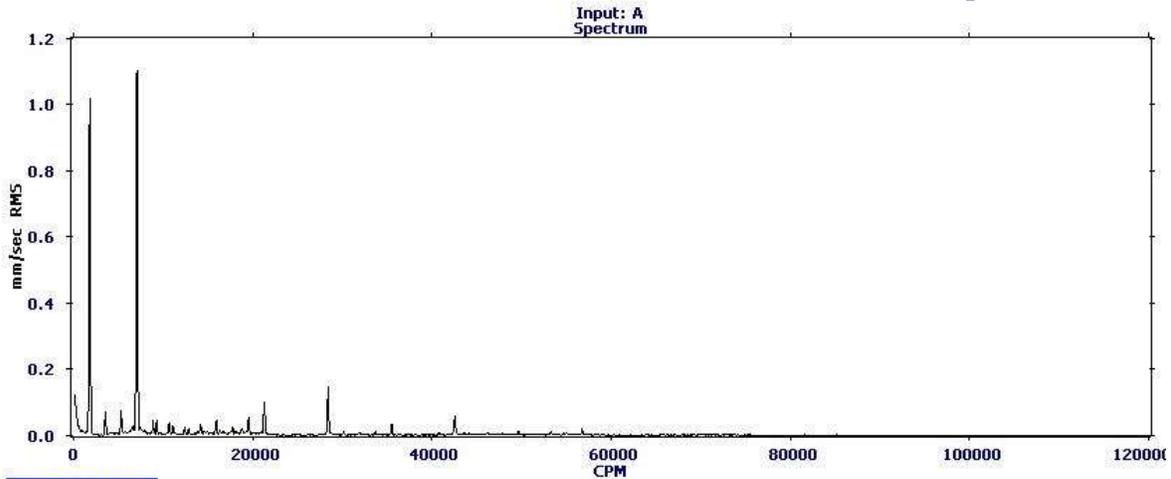
Vibration readings were taken with residual specific unbalance 8.84 g.mm/kg-



Graph 1



Graph 2



Overall: 1.73302

Frequency Data		Frequency Data		Frequency Data		Frequency Data	
1772.0	1.111	3544.5	0.07291	5316.6	0.07444	6637.4	0.03374
7088.3	1.28596	7469.2	0.02634	8869.6	0.04952	9276.3	0.04841
10620.2	0.03967	11052.4	0.03357	12403.5	0.02614	14185.1	0.03588
15945.9	0.05069	17730.7	0.02826	18726.3	0.02401	19486.3	0.05766
21263.7	0.1195	28354.8	0.14737	35441.3	0.0403	42527.8	0.06085

Graph 3

Balancing same rotor in G2.5 according to ISO 1940/1:

Rotor type data: Boiler Feed Pump

Balancing mode: Dynamic

A=900mm

B=240mm

C=890mm

Balancing radius (R1) = 160mm

Balancing radius (R2) = 160mm

Balancing Speed (RPM) = 6000

Tolerance (In correction planes) According to ISO 1940

Rotor mass: 490.0 kg

Quality grade: G2.5

Operating speed: 6000 RPM

Deviation (+/-): 0.00%

Tolerance allocation: symmetrical

Correction Position:

Plane: Left Right

Distribution: polar polar

Correction method:

Plane: Left Right

Method: Masses Masses

Material: Remove Remove

Measuring results:

Run: 1 Measure speed: 718 RPM

Left: -4.81 g at 40°

Right: -6.01g at 249°

Total unbalance mass left on both balancing planes after balancing: 10.82g

Residual specific unbalance (e_{per})

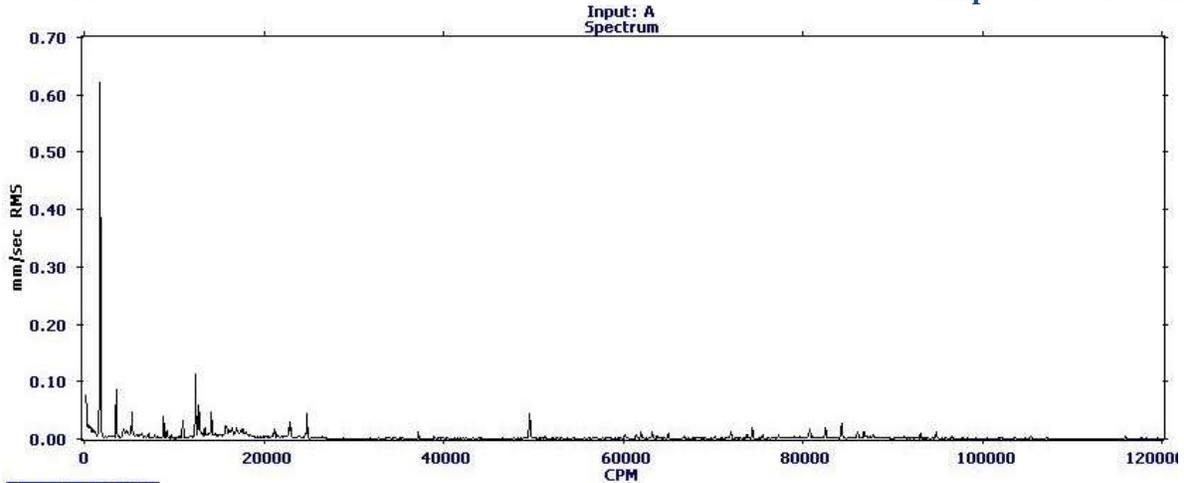
$$= (10.82 * 160) / 490$$

$$= 3.53 \text{ g.mm/kg}$$

This is the residual specific unbalance left in BFP rotor after balancing.

Permissible residual unbalance (e_{per}) as per ISO 1940/1 G2.5 at the service speed of 6000RPM = 4 g.mm/kg

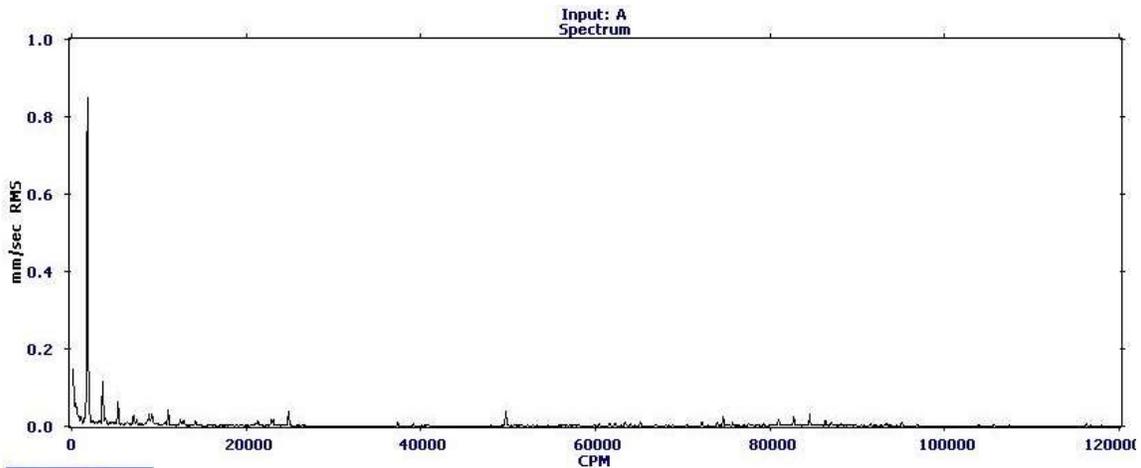
Vibration readings were taken with residual specific unbalance 3.53 g.mm/kg :



Overall: 0.73485

Frequency Data		Frequency Data		Frequency Data		Frequency Data	
416.3	0.02798	632.6	0.02406	1769.7	0.68828	3540.6	0.0887
5305.7	0.04889	8848.5	0.03951	10989.2	0.03606	12389.0	0.11564
12766.1	0.05935	13396.0	0.02073	14159.2	0.04894	15711.3	0.02553
15862.6	0.02726	16917.6	0.02259	22820.7	0.03214	23009.3	0.02062
24778.0	0.0473	49565.4	0.04486	82436.1	0.0208	84207.5	0.02792

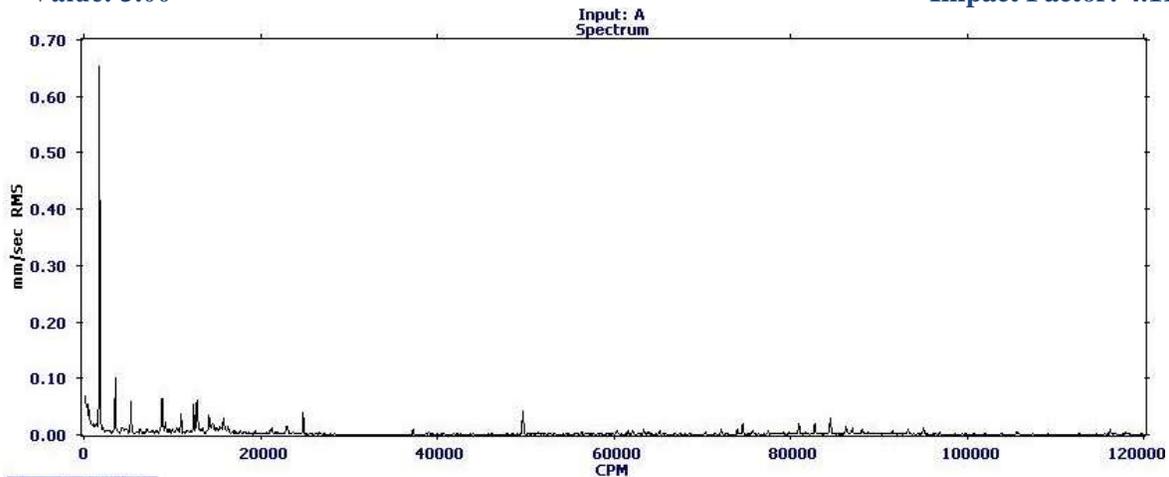
Graph 4



Overall: 0.95605

Frequency Data		Frequency Data		Frequency Data		Frequency Data	
418.8	0.06722	791.0	0.02959	1083.3	0.02643	1775.2	0.90929
3548.1	0.12342	3859.6	0.02285	5327.8	0.06163	7103.8	0.03036
7459.3	0.01882	8866.9	0.03179	9245.5	0.03288	11017.2	0.04256
12427.8	0.02035	22881.8	0.0187	24851.2	0.04316	49699.3	0.04203
74551.3	0.02454	80884.0	0.01931	82659.6	0.02365	84432.0	0.03247

Graph 5



Frequency Data		Frequency Data		Frequency Data		Frequency Data	
411.5	0.06286	573.5	0.0465	940.5	0.02092	1155.5	0.02073
1373.2	0.02201	1774.2	0.70259	3550.3	0.10808	5323.1	0.05992
8886.1	0.0752	9250.0	0.02439	11017.8	0.03614	12425.2	0.05848
12796.5	0.06634	14204.3	0.03842	14584.4	0.02271	15778.3	0.03129
24839.8	0.03962	49680.0	0.04574	74521.1	0.02177	84402.2	0.03301

Graph 6

CONCLUSION

Vibrations were considerably reduced after the rotor was balanced in G2.5 and unbalance amount left at the end of balancing was appreciably less than after balancing in G6.3. This shows that mass unbalance in rotor plays a significant role in pump vibration and it should be kept to a practical minimum. However there is not only the mass unbalance (mechanical cause) responsible for the vibration, hydraulic causes like cavitation have more severe impact (more than mechanical causes) but they were not considered here.

It is commonly perceived that the better the balance, lower the vibrations forces and deflections leading to better pump reliability and longer life. Indeed it is known that for grossly unbalanced rotors, vibrations are high and noticeable even without special measuring equipment and first reaction usually is to attempt to balance the rotor to as fine as possible, to eliminate the problem.

But there is no handy recognizable general relation between rotor unbalance and the machine vibrations. The unbalance response depends essentially on speed, the geometric proportions and mass distribution of rotor, as well as on the dynamic stiffness of the shaft, bearings and the foundation. Rotor unbalance is one of the causes of vibration but it may not be cause for every vibration that arises in rotating machinery.

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