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The Effect of Unbalance on Bearing Life

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Abstract: *The life of bearing is closely related to its loads which are affected by the eccentric unbalances of rotational components in the structure system of the bearing. However, since the bearing structure in some of the machinery is complicated, the exact load calculations and the life prediction for this kind of bearing are difficult. The load and life calculation for the bearing are developed with considering the magnitude of eccentric unbalances. The influences of unbalances on the loads and life of the bearing are studied. The calculation and analysis results show that the radial loads on rolling element of the bearing fluctuate significantly under the actions of the unbalances of different parts of machines and the bearing life reduces regularly with the eccentric unbalances changing. In this article, I have focused not on machines that are "supposed" to vibrate as part of normal operation, but on those that should not vibrate: electric motors, rotary pumps and compressors, and fans and blowers. In these devices, smoother operation is generally better, and a machine running with zero vibration is the ideal.*
Keywords: *permissible residual unbalance; eccentricity ,maru, urr, pump bearing*

Nomenclature
E =Eccentricity
G = Balancing quality grade
U per = The maximum residual unbalance
F lbs. = Force due to unbalance
MARU =Minimum achievable residual unbalance test
URR =Unbalance reduction ratio
L10h= Basic rating life of the ball bearing (h)
P = Effective load (actual force applied to the bearing)
C = Published catalog load rating
mCf=the eccentric unbalance of the cooling fan (g · cm) in a pump
mCb=the eccentric unbalance of the driving wheel (g · cm) in a pump

I. Introduction

Research showed that in the automobile wheel and its shaft had to be in a state of balance, i.e. the mass had to be evenly distributed about the rotating centerline so that the resultant vibration was at a minimum. This had to be achieved during the manufacturing process so that maximum service life could be achieved from the system. Imbalance could be caused by manufacturing defects (machining errors, casting flaws) or maintenance issues (deformed or dirty fan blades, missing balance weights). As machine speed increases, the effects of imbalance become greater. Imbalance can severely reduce bearing life as well as cause undue machine vibration.

A level of unbalance that is acceptable at a low speed is completely unacceptable at a higher speed. This is because the unbalance condition produces centrifugal force, which increases as the speed increases. In fact the forces caused by unbalance increases by the square of the speed. If the speed is doubled, the force quadruples; if the speed is tripled the force increases by a factor of nine.

It is the force that causes vibration of the bearings and surrounding structure. Prolonged exposure to the vibration results in damage and increased downtime of the machine. Vibration can also be transmitted to adjacent machinery, affecting their accuracy or performance. Load determination and life prediction calculation of the bearing are the premises of a good bearing design and a reasonable working condition matching, and yet it is one of difficulties in the practical application. In some cases, vibration is inherent in machine design. For instance, some vibration is almost unavoidable in the operation of reciprocating pumps and compressors, internal combustion engines, and gear drives. In a well-engineered, well-maintained machine, such vibration should be no cause for concern. In simplest terms, vibration in motorized equipment is merely the back and forth movement or oscillation of machines and components, such as drive motors, driven devices (pumps, compressors and so on) and the bearings, shafts, gears, belts and other elements that make up mechanical systems.

From previous studies, it is not difficult to know that the internal load distribution of rolling element bearing is closely related to the external loads which are usually assumed to be invariable.

Machinery professionals intuitively know that by doing alignment and balancing jobs to tighter tolerances, and by reducing internal clearances in machinery, that vibration levels will be reduced with a corresponding increase in machinery reliability. However, it is often difficult to justify what needs to be done. Reliability and replacement costs for rolling element bearings are major concerns in most plants.

II. Literature Review

When man invented the wheel, he very quickly learned that if it wasn't
completely round and if it didn't rotate evenly about it's central axis, then he had a problem! Modern man still suffers from the same problem – only now the problem is amplified.

As machine first patent for balancing technology was filed by Henry Martinson of Canada in 1870, four years after the development of the dynamo by Siemens. Near the turn of the century, Akimoff (USA) and Stodola (Switzerland) attempted to develop Martinson's technology and apply it for industrial use. However, it was in 1907 when a modified version of the technology was patented by Dr. Franz Lawaczek, and offered to Carl Schenck, Darmstadt, Germany, for development. Schenck built the first industrial two-plane balancer, and subsequently bought exclusive world rights to the dynamic balancing machine in 1915.

Technology advancements gave way to improved sensitivity, frequency selectivity and plane separation capability. The development of electronics and mechanical/electrical transducers, greatly reduced balancing time and paved the way for modern balancing technology.

Today balancing equipment is used with confidence for a wide range of applications - from the smallest rotors for dental drill instruments to the largest steam turbines in the world. Precision balancing machines assure accurate, dependable rotor operation.

Now a days balancing industry provides a complete range of balancing, diagnostic and special equipment for the automobile industry, power generation industry, medical, aviation industries and an engineering staff that offers a broad range of experience for nearly any balancing application which involves different sizes of bearings starting from dental drill bearing sizes (very small) to aviation/power generation rotor bearing (extremely big) sizes.

III. Basic Bearing Life Equation

Examining the basic bearing life equation we find that speed, load and the type of bearing are factors:

$L10h = (16667 / rpm) \times (C / P)^r$

Where:

- L10h = 90th percentile of life in hours (the point at which only 10 percent of bearings in identical applications fail);
  Note: average life = 5 x L10h
- Rpm = Rotational speed of the bearing
- C = Published catalog load rating
- P = Effective load (actual force applied to the bearing)
- r = 3 for ball bearings
- r = 3 1/3 for other types of rolling element bearings

3.1 FACTORS AFFECTING BEARING LIFE

For those involved in predictive maintenance activities, especially vibration monitoring and analysis, two questions have always been present.

- What is the correlation between changes in vibration level and the corresponding impact on bearing life?
- What is the value in knowing this correlation if there is one?

  Few predominant factors affecting impact rolling element bearing life are:
- RPM of the shaft
- Design load rating of the bearing (as defined by the manufacturer)
- Type of rolling element bearing (ball or other rolling element type-cylindrical roller, spherical roller, needle roller, tapered roller)
- Actual load (force) applied to the bearing
- Lubricant ability
- Contamination level
- Operating temperature.

First, let's investigate the impact of rotational speed on bearing life. Reviewing the basic bearing life equation:

L10h = (16667 / rpm) x (C / P)r
The impact of increasing speed is obvious. Doubling the rotational speed (while maintaining a constant load) = L10h / 2 = 1/2 the original life.
Equation results:
2 x rpm = 1/2 life
3 x rpm = 1/3 life
1.25 x rpm = 0.8 life
Next, there is a need to investigate the impact of load on bearing life. Reviewing the basic bearing life equation again:
L10h = (16667 / rpm) x (C / P)r.
The impact of increasing load (force) is pronounced.
Doubling load (while maintaining a constant speed) = L10h / 8 or 1/8 life (1/2)3 for ball bearings.
The L10 bearing life is a published number and it is calculated as:
L10 = (16,667/RPM)*(rated load/actual load)3
The L10 life of a bearing is the life expectancy for 90% of the population, where a full load life is estimated at 1,000,000 revolutions. Guidelines for loading are as follows:

- Light loading at <6%
- Normal loading at 6% to 12%
- Heaving loading at >12%.

Typically, this works out to be a life expectancy from a few months to several years at continuous 365-day/24-hour usage.

TABLE-1

Other rolling element bearing types include cylindrical, spherical, tapered and needle bearings.

IMPACT OF INCREASED LOAD ON BEARING LIFE		
Percentage Life Decrease		
% Load Increase	Ball Bearings	Other Rolling Element Bearing Types1
5	14	15
10	25	27
15	34	37
20	42	46
25	49	52
50	70	74
75	81	85
100	87	90

vibration in motorized equipment is merely the back and forth movement or oscillation of machines and components, such as drive motors, driven devices (pumps, compressors and so on) and the bearings, shafts, gears, belts and other elements that make up mechanical systems.

TABLE-2

FORCES AND SOURCES OF VIBRATION		
Force Source	Type of Force	Reducible
Unbalance	Dynamic	Yes
Shaft Misalignment	Dynamic & Static	Yes
Belt / Drive Tension	Static	Yes, if Excessive Tension is Present
Looseness	Dynamic	Yes, if Excessive Looseness is Present
Rotor Weight	Static	No, Not Normally
Gear Reaction	Dynamic & Static	No
Process Forces	Dynamic & Static	No, Not Normally

Of these seven different forces, only the first four can normally be addressed by the maintenance department. The other three are machine design related and are not normally reducible.

TABLE-3

IMPACT OF VIBRATION REDUCTION ON BEARING LIFE (Assuming dynamic load is the major force component)		
Percentage Increase in Bearing Life		
% Reduction in Vibration	Ball Bearing Types	Other Rolling Element Bearing
5	17	19
10	37	42
15	63	72
20	95	110
25	137	161
30	192	228
40	363	449
50	700	908

IV. Effect Of Unbalanced Forces

Unbalance is one of the primary sources of machine vibration. The force produced due to unbalance can be calculated using either of the following formulae:
F lbs. = 0.062 x (rpm / 1000)2 x U gm. In. Where:
1 gm. in. = 1 gm. of mass @ 1 in. of radius from centerline of rotation
Because unbalance is a rotating load, the bearing’s inner race is zone loaded. This is a different type of loading compared to most of the other force sources.

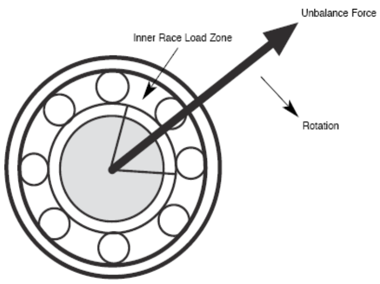


Fig 1

Because unbalance is a “rotating load or force”, the following conversion must be made to use this force in the bearing life equation:
P = F lbs. x fm
Where:
F lbs. = Force due to unbalance
fm = Factor of 1.0 to 1.5 according to the ratio of static force compared to the unbalance force on the bearing
(When this ratio is 1.0 then the factor is 1.333)

4.1 CALCULATION OF PERMISSIBLE RESIDUAL UNBALANCE

U per – the maximum residual unbalance permitted for a
rotor or in a correction plane[4].
U per = e per x m
where m = rotor mass
Calculation of permissible unbalance for a pulley of radius 152 mm running at 800 r.p.m, having 8.1 kg weight.
Speed, n = 800 rpm
Weight of pulley, P = 8.1 kg
Radius of pulley, R = 152 mm
(O.D. is 314mm)
Balancing grade, G = 4 (means vibration speed=4mm/s for pulley)[3}
Max. residual unbalance e=10XG/nx1000=10x4/800x1000=50µm
e=p x R/P
p=e x P/R =50X8.1/152=2.66KG

Therefore the Permissible Residual Eccentricity is 50µm & the Permissible
Residual Unbalance is 2.66g for this Pulley.

Another way to calculate permissible unbalance in balancing machines

U per (g-mm) = 9549 x G x W/N (W in kg)

G = Balance quality grade from Table 4

W = Rotor weight

N = Maximum service RPM

MARU (Minimum achievable residual unbalance test) and URR (unbalance reduction ratio) test are performed to test reliability of balancing machines.[1]

4.2 INFLUENCES OF THE UNBALANCE CHANGES ON THE BEARING LOADS

An imbalance in the motor, for instance, would most likely cause a radial vibration as the “heavy spot” in the motor rotates, creating a centrifugal force that tugs the motor outward as the shaft rotates through 360 degrees. Another key factor in vibration is amplitude, or how much force or severity the vibration has. The farther out of balance our motor is, the greater its amplitude of vibration. Other factors, such as speed of rotation, can also affect vibration amplitude. As rotation rate goes up, the imbalance force increases significantly.

Frequency refers to the oscillation rate of vibration, or how rapidly the machine tends to move back and forth under the force of the condition or conditions causing the vibration.

The calculation results of the radial loads on a pump were obtained under the given eccentric unbalance= m_{Cb} = 15 g cm of the pump driving belt wheel, and m_{Cf} = 30 g cm of the cooling fan. [11],[12].

To analyze the changes of the radial loads when the eccentric unbalances change gradually, two cases are discussed. One case is that three sets of m_{Cb}, i.e. 0 g cm, 15 g cm and 30 g cm are selected and the eccentric unbalance of the wheel keeps invariant with the value of zero. The other is that the eccentric unbalance of the fan is selected as 0 g cm, 30 g cm and 60 g cm respectively, and the eccentric unbalance of the wheel is reset to zero. The radial load variations on the two rolling element rows of the bearing in one rotational period of the cooling fan, with the eccentric unbalance of the fan changing, are shown in Fig. 3. In a similar manner, the radial load variations with the eccentric unbalance of the wheel changing in the same period are shown in Fig. 4.[12]

The calculation results show that for the given water pump bearing, the fluctuation range of the radial load on row I(impeller side) is enlarged with the eccentric unbalance increasing of the cooling fan, while the radial load on row II (fan side)increases proportionally.

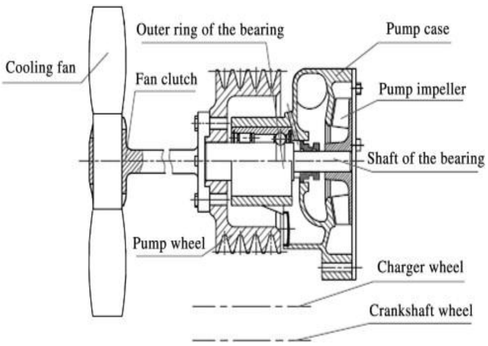
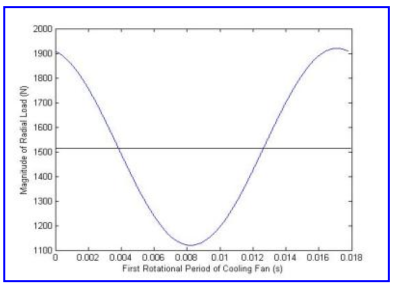


Fig. 2. Structure view of the water pump bearing.



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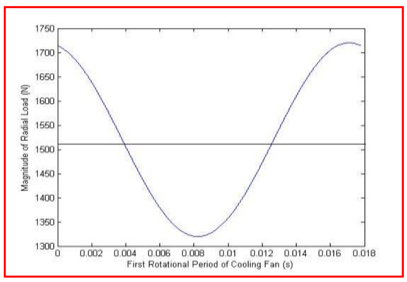

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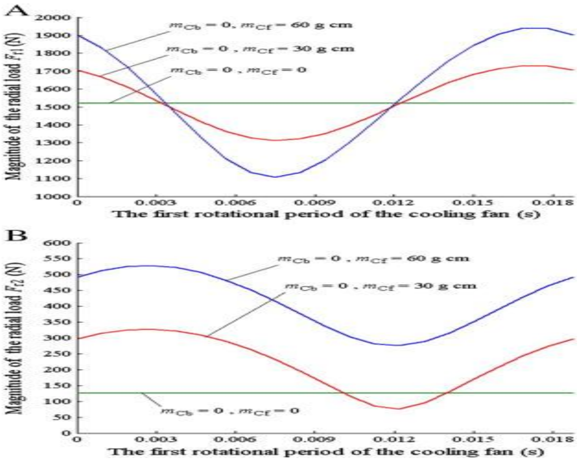

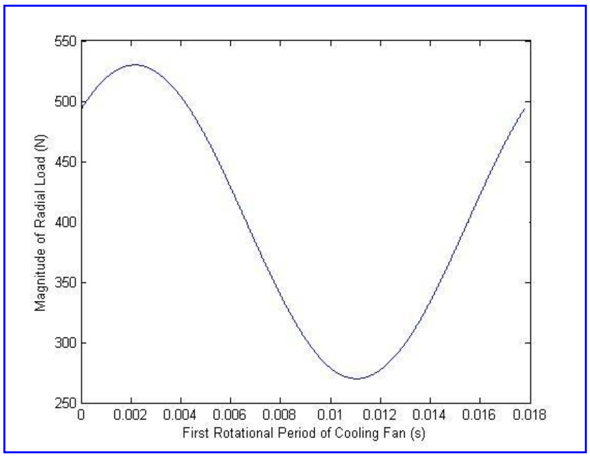

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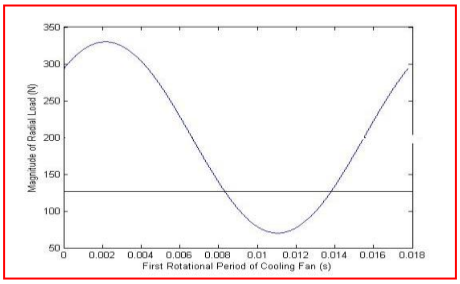

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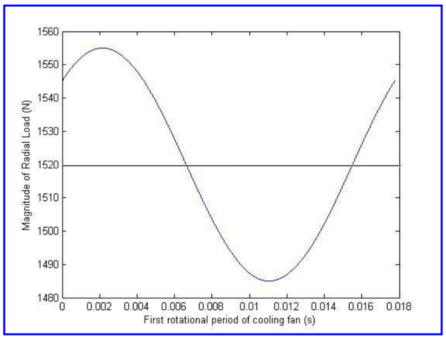
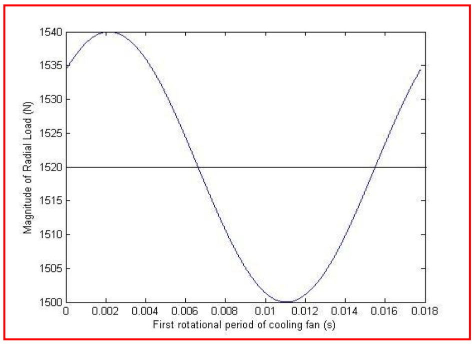
Fig. 3. Radial loads on two rolling element rows in one rotational period of the cooling fan: (A) F_{r1} under different fan unbalances; (B) F_{r2} under different fan unbalances.[12]





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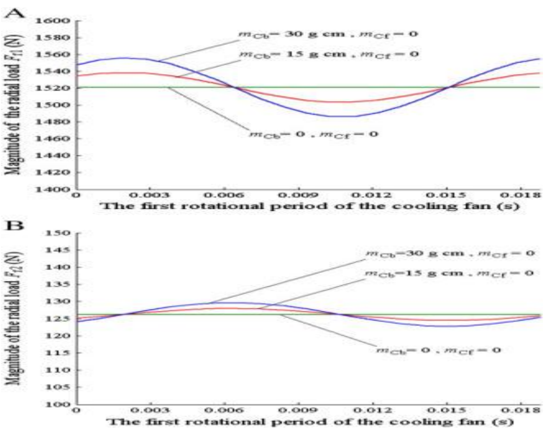

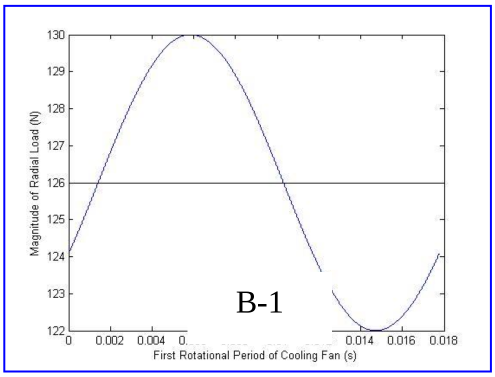

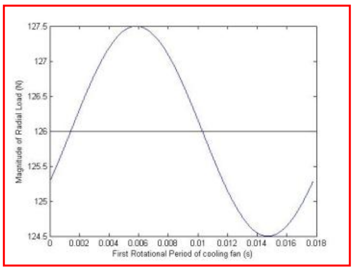

B-2

Fig. 4. Radial loads on two rolling element rows in one rotational period of the cooling fan: (A) F_{r1} under different wheel unbalances. (B) F_{r2} under different wheel unbalances.[12]

The mean dynamic equivalent radial loads are heavily influenced by the fan unbalance. On the contrary, the influences of the wheel unbalance on the mean dynamic equivalent radial loads are little. This may be caused by the fact that the cooling fan is located far away from the pump bearing and the driving wheel is near the pump bearing.

TABLE 4
QUALITY GRADES AS PER ISO-1940

Balance quality grade G	Rotor types
G 100	Crankshaft drives of large Diesel engines Complete engines for trucks and locomotives
G 40	Crankshaft drives for engines of trucks and locomotives
G 16	Parts of crushing machinery Parts of agricultural machinery
G 6.3	Fly-wheels Fans Aircraft gas turbine rotors Electrical armatures Process plant machinery Pump impellers
G 2.5	Machine-tool drives Turbo compressors Small electric armatures Turbine-driven pumps
G 1	Grinding machine drives Textile bobbins Automotive turbochargers
G 0.4	Gyroscopes Disk-drives Spindles for high-precision applications

Different quality grades are provided in I.S.O 1940 for a specific component depending upon Criticalness and accuracy required for his application.
Based on this suitable value of balancing tolerance can be calculated as per his application in different field as described above.[1],[2].

V. CONCLUSION

Vibration is a characteristic of virtually all industrial machines. When vibration increases beyond normal levels, it may indicate only normal wear – or it may signal the need for further assessment of the underlying causes, or for immediate maintenance action.

Understanding why vibration occurs and how it manifests itself is a key first step toward preventing vibration from causing trouble in the production environment. Reducing the forces caused by unbalance, looseness and misalignment will result in lower vibration levels for machines. Reducing excessive belt tension will also reduce machine forces but will not produce an appreciable reduction in vibration level. The vibrations themselves have only a minor impact on bearing life but the forces which cause these vibrations have a major impact on the actual bearing's longevity. In addition to improving reliability and reducing the cost of maintenance of machines, several more benefits are obtained by reducing vibration levels:

• Reduced Noise Levels
• Reduced Operating Costs (Utilities)
• Improved Operating Safety
• Improved Maintenance Technician Morale
• Increased Life for Related Machine Components (seals, housings, shafts, impellers, windings, etc.)
• Reducing vibration levels on machines by correcting common machine problems or applying tighter tolerances does indeed dramatically improve bearing life and reduce maintenance and operating costs.[9]

RESULT 1- Bearing life is inversely proportional to speed changes.
(1 / speed change ratio)
RESULT2- Increased load results in an inversely exponential reduction in life.
RESULT3- Unbalance is up to 50 percent more destructive to bearing life than other vibration sources producing equal vibration levels.
RESULT4- In a specific example of water pump the basic rating life of the bearing changes slowly when the fan unbalance is constant and the wheel unbalance increases from zero to 150 g cm.

When the wheel unbalance is constant and the fan unbalance varies in the range of 0–10 g cm, the basic rating life of the bearing decreases less than 900 h. The reduction is also very little. However, the reduction of the basic rating life is nearly 7000 h when the fan unbalance increases from 10 g cm to 30 g cm. The life of the bearing reduces much faster and the life variation reaches to 16 000 h when the fan unbalance varies from 30 g cm up to 60 g cm, [12] with keeping the wheel unbalance constant.

The above results show that small eccentric unbalances have small effect on the life of the water pump bearing, but if the eccentric unbalances, especially the fan unbalance, go beyond some threshold value, the life of the pump bearing would reduce sharply.[8]

In this paper the influences of the unbalance variations on the bearing life were studied. It was found that the unbalances will cause reduction of the bearing life [7]and this effect would be remarkable if the unbalances increased to a certain level.

The main component of the vibration is unbalance of the rotor and the rotor weight is nominal. Currently if the rotor is balanced to a tolerance of ISO G6.3. per plane then by adjusting this balancing tolerance to ISO G3.2 per plane, the bearing life should be extended by 700 percent. Since the vibration should be cut in half as a result of the improved balance tolerance,

5.1 Future scope

Further area of this study can be extended for analyzing the effect of unbalances on bearings of different industrial machines of different nature such as production machines of different application, mixer, grinder, axles in automobile etc. It will also be immensely helpful in the areas of increasing life of critical machines such as power generation machines (turbine and other moving parts),dental drills, aviation rotors, turbo chargers.

Sometimes these analysis are providing important tips for changing the design Of specific machine.(such as changing the bearing position inside the housing)

References

[1] ANSI S2. 19-1975 "Balance Quality Requirements of Rotating Rigid Bodies. "American National Standards Institute.
[2] ISO 1940-1:1986, Balance Quality Requirements of Rigid Rotors." British Standards Institution. BS 6861-1:1987,
[3] "Balance Quality Requirements of Rigid Rotors." International Organization for Standardization.. ISO 1940 STANDARD For Balancing Masses1. ISO 1940/1,
[4] "Balance Quality Requirements of Rigid Rotors." German Standards Institution. The Practical Application of ISO 1940/1, VDI 2060,
[5] DYNAMIC BALANCING HANDBOOK, "October 1990, IRD Mechanalysis Inc.
ISO 1925, "Balancing Vocabulary ."International Organization for Standardization.
[6] Dynamic capacity of rolling bearings G. Lundberg, A. Palmgren
Acta Polytech. Mech. Eng. Ser., 1 (3) (1947), p. 7
[7] Guide bearing probability load theory of large vertical pump , Mech. Mach. Theory, 42 (2007), pp. 1199–12097]
[8] Modern Pump Technology Handbook
(1st edition) China Astronautic Publishing House, Beijing (1995) B. Qiu, H. Lin, S. Yuan, X.F. Guan
[9] Reliability Magazine Reprinted courtesy of Reliability Magazine.
Article written by L. Douglas Berry in 12/95 issue
[10] Rolling Bearing Analysis (1st edition)John Wiley and sons, T.A. Harris,New York (1966)
[11] The influence of bearings on pump performance, World Pumps (September) (2004) 46–49. P. Burge
[12] The research on the life of auto water pump bearing considering the rigidity of bearing spindle
Koyo (136) (1989), pp. 51–63 Sakuragi