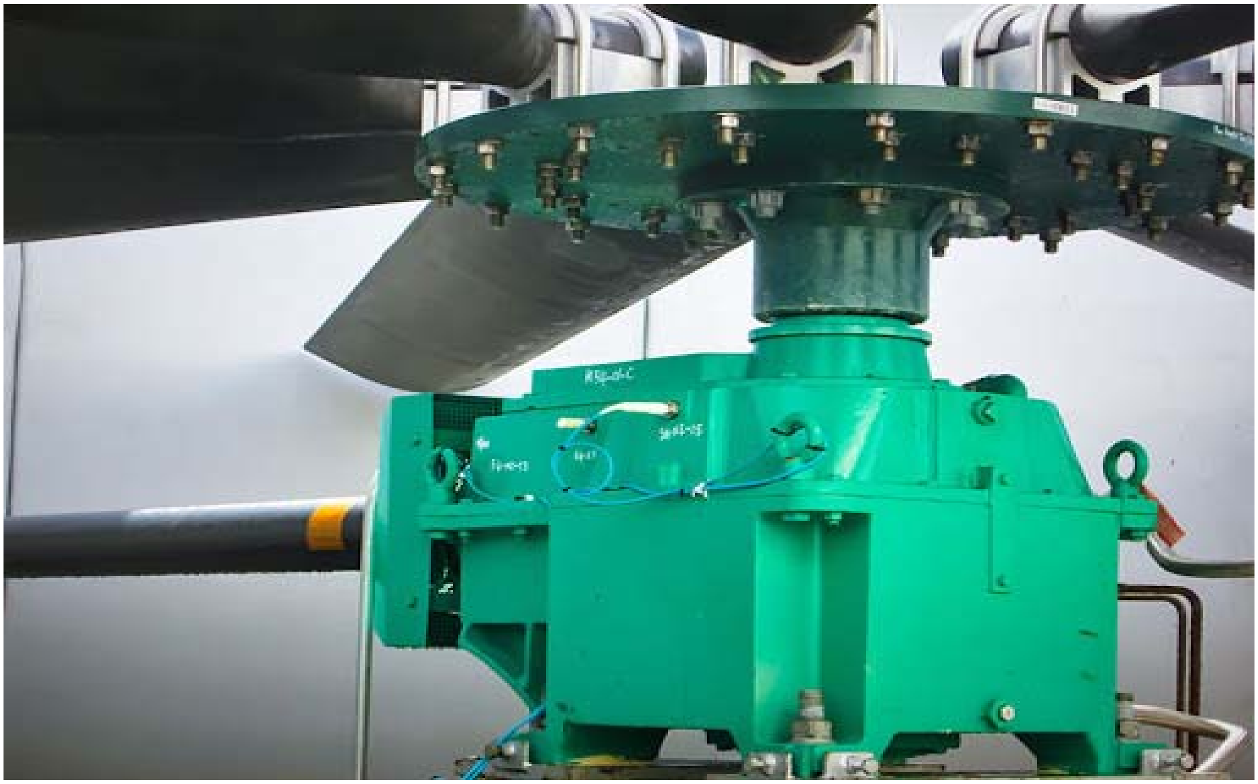
Failure Analysis: Petrochemical Plant 185 KW Cooling Tower Fan Gearboxes Year 2010

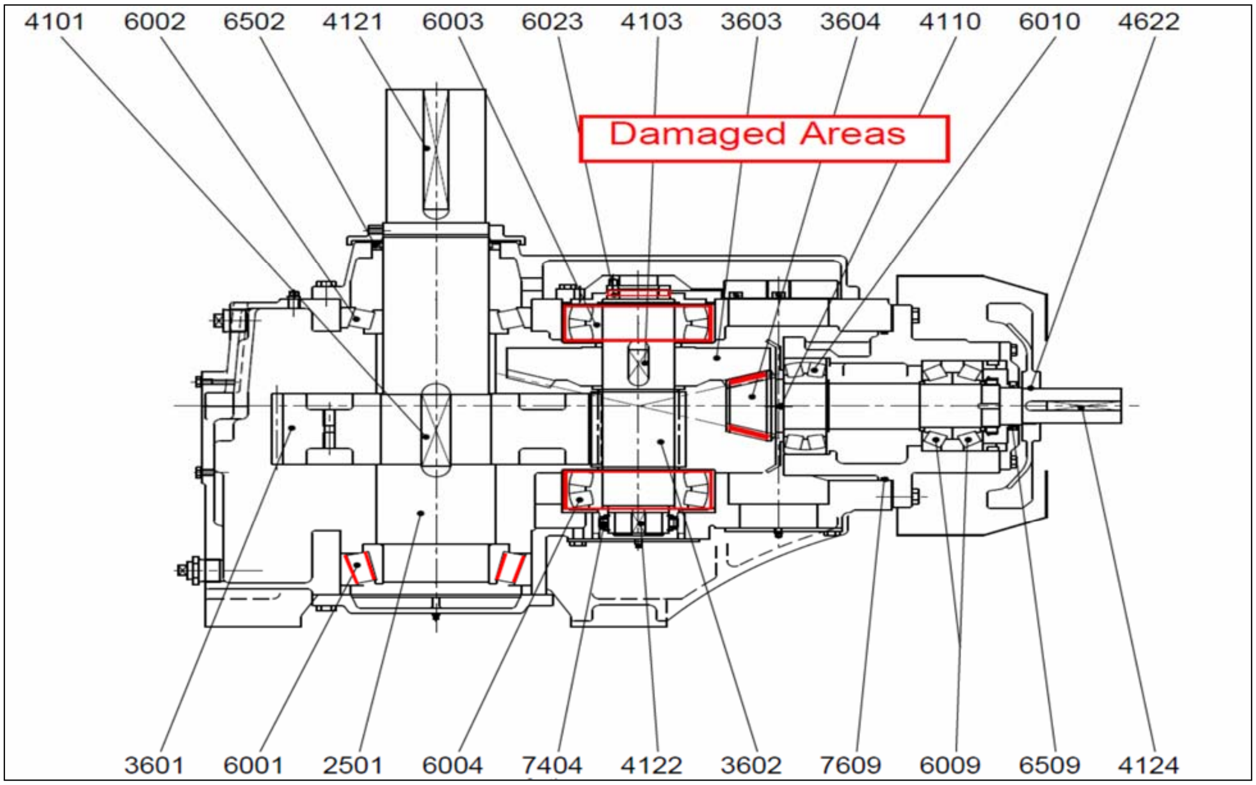
By Abdulrahman Alkhowaiter



Introduction

The Petrochemical plant operates with several cooling towers that utilize a total of approximately 50 cooling fan gearboxes with 1.5 years' actual average operating time before failures occurred. These gearboxes were manufactured by a well-respected Japanese gearbox manufacturer. They are rated at 185 KW rating, with approximately 120 rpm low speed shaft and driven by electric motors at 1750 rpm. The gearboxes are double reduction spiral helical and helical type gears with rolling element bearings and splash lubrication using EP-320 viscosity oil recommended by OEM.

After 1.5 years of installation six gearboxes were removed due to high internal noise, vibration, and excess pinion backlash. These were the first recorded failures since commissioning. As a result, the Mean Time Between Failures MTBF so far was 1.5 years which is not acceptable for a cooling tower gearbox application and four years is the minimum average expected lifetime. Therefore, these gearboxes had reached less than 40% of the desired design life. The failures are not a wear out mode but a true mechanical failure. Note that the gearboxes have not been opened before by the facility. They were being maintained by a full detailed lubrication PM lubrication program and monthly vibration monitoring.



Findings: Analysis of the installation on the same cooling towers led to the following

1. The cooling tower structural design is excellent, as witnessed by the very little structural vibration while fans are operating, about 0.12 inch/s rms. This also confirms that fans are properly balanced. No high gearbox noise is evident
2. The motor vibration in both radial and axial direction was also low on all operating units, about 0.12 inch-s rms maximum indicating excellent alignment of motor to gearbox, fan balance, and motor balance.
3. Lube oil levels and color on the failed gearboxes were found to be correct including oil piping.
4. The motor to gearbox coupling spacers are lightweight carbon fiber reinforced composite type, with well-designed rubber grommet type coupling hubs, all in good condition. No bent spacers, corroded hubs, damaged or worn rubber bushings were found on any of the five units.

Inspection and Failure Analysis

Disassembly of three gearboxes at maintenance shop led to the following findings:

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1. No signs of lube oil contamination were found on any of the three units we surveyed. All gears and bearing races were found to be shining and free of corrosion at disassembly. Oil in all units found at proper level, with few particulates, acceptable color and viscosity as per OEM. There were no signs of oil starvation or scuffing on the loaded bearing races or gear teeth. Lubrication failure is ruled out.
2. Gearboxes had correct fit of both coupling hubs on input and output shafts. No looseness.
3. The fan hub to shaft hub carbon steel coated disk is corroding severely on the bolt flange holes' location and will not operate for more than one year without repair.
4. On all units the small roller thrust bearing on top of intermediate shaft is found heavily damaged. **Failure Analysis Error**: **In 2010 we thought it is not a design defect but OEM said the bearing is weak and changed it to a heavy duty Bronze thrust bushing in 2011.**
5. On all units, the pinion gear teeth are worn out and damaged with only 50 to 60% maximum contact with mating spiral helical gear. This is due to incorrect assembly at factory as all gears should have about 80% tooth contact pattern. This led to overloading of teeth and surface fatigue damage.
6. Remarkably the pinion bearings on all units are found in excellent condition; this means that these bearings are well rated for the forces and are of good design.

7. There is no sign of tooth breakage or teeth root fatigue on any gears, only overloading damage on tooth surface of pinion and intermediate spiral helical gears. Gear manufacturing and metallurgical quality is high.
8. The LS shaft bottom bearing on one unit has damaged, flaking tapered roller bearings (not the race) at the upper portion. This is considered a result of axial overload from excessive preloading by shims at factory.
9. The intermediate shaft on two units has both the upper and lower main spherical roller bearings damaged with surface fatigue on rollers. Again, the root cause is incorrect preloading at factory.

Design and Manufacturing Review:

The Five Root Causes of Machinery Failures are:

1. Design Defects
2. Manufacturing Defect: Assembly, Machining, Welding, Casting, Material
3. Incorrect Specification or Installation-System Defect
4. Incorrect Operation
5. Incorrect Maintenance

Intermediate Shaft: Notice Contact Pattern of 50 to 60%

By reviewing the different observations on disassembly of the machines and including a field analysis of similar gearboxes at operating mode, we have ruled out the following: There are no maintenance related deficiencies found on the six gearboxes removed or the healthy units. The gears were never opened before by the Petrochemical plant. There was no installation related defects found at the cooling towers, and no incorrect operation as these are simply on-off operation. There are no specification defects either, as the gearboxes are under-loaded at present operating conditions compared to rated power, and are specifically designed for cooling tower applications.

From a design point of view: The gearbox nameplate rating is 185 KW, while motor amperage on all units is averaging 230 Ampere +/- 4 ampere, as measured on several machines in April 2010. The motor's data sheet shows a rating of 185 KW at 278 amperes. Therefore, the transmitted shaft power to gearbox is only:

Motor Shaft Output KW= Motor Eff. x 230/278 x 185 KW = 146 KW. No Overloading is Evident.

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The following are the main design findings:

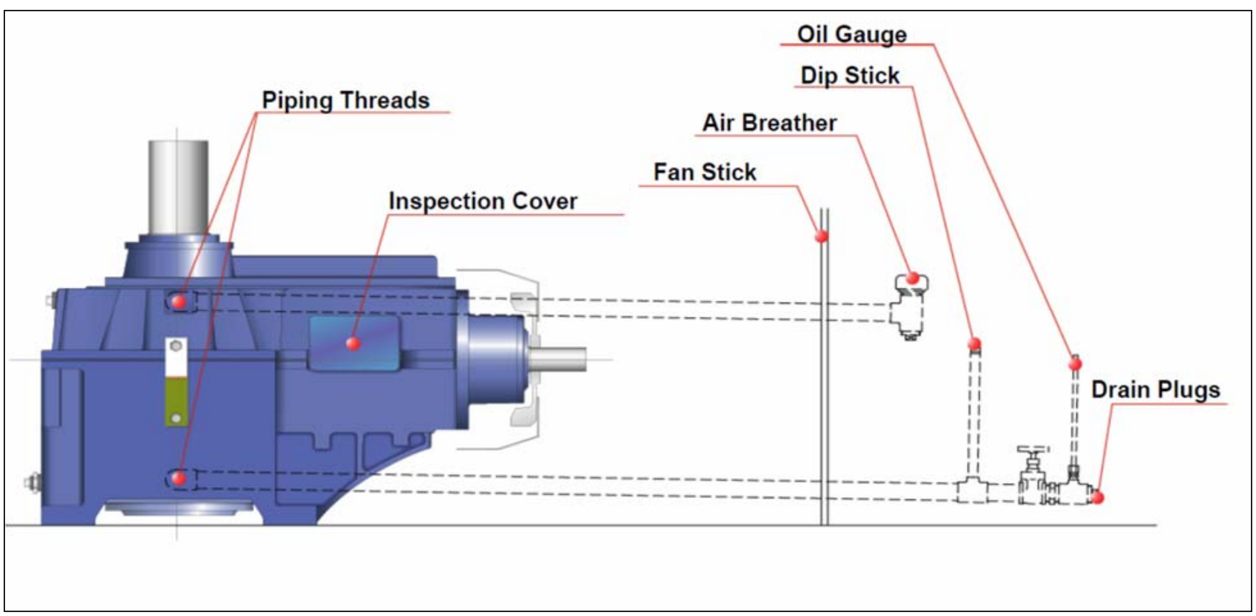
1. The gearbox bearings are well sized for this unit capacity so no obvious design errors here. The pinion bearings withstood the damaged pinion movement and high backlash without failing. **Failure Analysis Error**: **In 2010 we thought it is not a design defect but OEM said the intermediate shaft top thrust bearing is weak and changed it to a heavy duty Bronze thrust bushing in 2011. This additional failure root cause did not rule out the other defects found.**
2. The gear teeth are well sized and surface finish quality is acceptable.
3. The gearbox design is well sealed from the environment. The housing is stiff.
4. The shafting design and material has proper diameters to withstand torque loadings.
5. There is a cooling fan on the pinion shaft. This can be removed as this is a cool service application at all times and excess cooling is not desired.
6. The lube oil viscosity of EP-320 by OEM is not optimum for bearing lubrication at operating oil temperatures of 65 to 75 DegC and DN-values. See NTN bearing handbook which recommends ISO-VG-150 viscosity oil. We recommend EP-220 gear oil as optimum solution for both gears and bearings in this service. **Note: Gearbox OEM did not approve oil viscosity change so it was maintained as ISO-VG-320.**

The remaining premature failures causes in our opinion are Manufacturing-Assembly Defects. The following are the checks and defects found with solution for problems:

1. **The intermediate gear-shaft height was set too low at factory on the three failed units**, with incorrect lower bearing housing shim thickness. By this error the helical spiral gear low position does not allow penetration of the pinion gear toward gear center to achieve desired 80% contact pattern. This resulted in overloading of pinion teeth and failure in service. The assembly workers at factory installed the pinions with axial shims to the correct backlash, but this step was incorrect as the intermediate shaft must be set at exact height first by shimming. Next the top bearing housing cover is bolted down. Then a gear contact paste or blue contact paste such as Prussian Blue is placed on spiral gear teeth to test and confirm minimum 80% contact pattern with pinion. Finally, the pinion axial shims are applied to achieve correct OEM backlash figures. It is possible that the specified OEM design preload setting is incorrect, but we do not have these figures.
2. **The LS shaft lower bearing on one unit was damaged** because the LS shaft bearing was excessively preloaded, leading to early fatigue failure of the bearing. This means that the thickness of the top shims was excessive. Solution: **The measured shaft float on this LS rotor should be 0.002 to maximum of 0.003 inch.** Use crane to lift LS shaft upwards, using dial indicator for measurement. This is to allow for axial differential thermal growth of casing and gear shafts in vertical direction. Our calculations have shown that the LS shaft [Length between bearings = 54 cm] grows axially on average about 0.002 to 0.003 inch greater than the gearbox, so this is an additional operating preload. Use lead wire crush test on top cover of all LS bearing housings to measure exact clearances during assembly, and install new shim thicknesses.
3. **The intermediate shaft main top/bottom bearings were damaged on two units** due to the same incorrect excessive bearing preload by shims at factory. **The intermediate gear shaft [Length between bearings = 37 cm] must be set with a shaft float of 0.002 inch to 0.003 inch.** Before installing shims, use lead wire crush test on the top bearing housing. After assembly, lift shaft upwards and insure 0.002-inch float. The same applies to the small thrust bearing at top of intermediate gear. This has its own small shims and springs, but the incorrect larger shims on top forced these bearings fully metal to metal, thus defeating the purpose of the protective spring. This overloaded the small bearings since high thrust forces were transferred from the main shaft.
4. **There was no evidence found of incorrect machining** accuracy of bearing bores in housings.

Recommendations to Solve the Premature Failures

1. Assemble all overhauled gearboxes using the correct assembly procedures explained above. All gears must be blue contact tested for minimum 80% contact during assembly by QA technician. QA shall also witness and record lead wire crush preload measurements on shaft bearing top covers. **Recommendation was accepted by gearbox manufacturer.**
2. It is not necessary to use OEM bearings. Other quality manufacturers such as SKF, FAG, TIMKEN or NTN with bearings of equivalent rating and number can be used as option. We do not recommend other lower quality bearings due to possible low reliability.
3. Change gearbox lubrication to EP-220 oil. **Not applied; Gearbox vendor did not accept this.**
4. Replace all gearboxes piping from the existing galvanized steel to 316 stainless steel using flanged fittings, not threaded. Threaded fittings allow leakage and gearbox loss of oil.
5. Machine out the corroded sections on all fans hubs bolt flange circle, and weld 316 stainless steel rings in place, of 20 mm thickness and 125 mm ring width. Use 316 welding rods. Electrolytic corrosion will not occur because the Anode here, the steel hub, is of large mass with respect to the small mass stainless steel component. Apply epoxy paint to weld zone.
6. **Design Note: By year 2011, the OEM had recognized a design defect in the intermediate shaft thrust bearing and replaced all units with Bronze thrust bushings.**
7. **In Conclusion: The gearbox life improvement was due to the OEM finding and this report.**



Attachment 1: SKF Bearing Failure Analysis Manual

Flaking caused by axial compression

Cause	Action
Incorrect mounting, which results in axial loading, e.g. excessive preloading of angular contact ball bearings and taper roller bearings.	Check adjustment when mounting the bearings.
The non-locating bearing has jammed.	Check the fit and lubricate the surfaces.
Axial freedom of movement has not been sufficient to accommodate the thermal expansion.	If temperature differential between shaft and housing cannot be reduced, provide greater freedom of movement.


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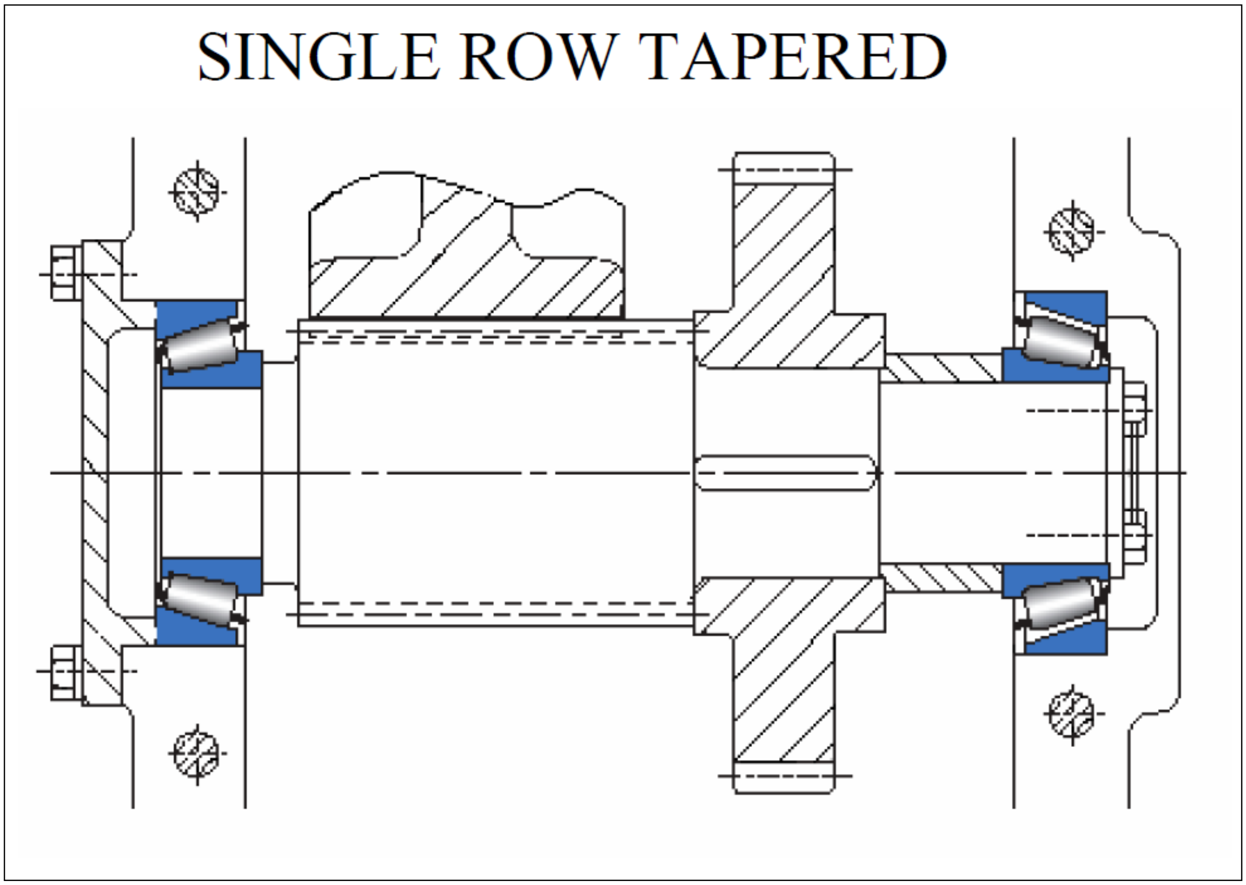
Fig 58 Outer ring of a self-aligning ball bearing subjected to excessive axial loading. Flaking in the load zone

Fig 59 Flaked inner ring of a spherical roller bearing. The extent of the flaking around one entire raceway indicates that the axial load has been very heavy in relation to the radial load

SKF 31

Attachment 2: Tapered Single Roller Bearing Typical Detail on General Gearboxes



Attachment 3: NTN Bearing Manual Preload Basics

8.4 Preload

Normally, bearings are used with a slight internal clearance under operating conditions. However, in some applications, bearings are given an initial load; this means that the bearings' internal clearance is negative before operation. This is called "preload" and is commonly applied to angular ball bearings and tapered roller bearings.

8.4.1 Purpose of preload

Giving preload to a bearing results in the rolling element and raceway surfaces being under constant elastic compressive forces at their contact points. This has the effect of making the bearing extremely rigid so that even when load is applied to the bearing, radial or axial shaft displacement does not occur. Thus, the natural frequency of the shaft is increased, which is suitable for high speeds.

Preload is also used to prevent or suppress shaft runout, vibration, and noise; improve running accuracy and locating accuracy; reduce smearing, and regulate rolling element rotation. Also, for thrust ball and roller bearings mounted on horizontal shafts, preloading keeps the rolling elements in proper alignment.

The most common method of preloading is to apply an axial load to two duplex bearings so that the inner and outer rings are displaced axially in relation to each other. This preloading method is divided into fixed position preload and constant pressure preload.

8.4.2 Preloading methods and amounts

The basic pattern, purpose and characteristics of bearing preloads are shown in Table 8.11. The definite position preload is effective for positioning the two bearings and also for increasing the rigidity. Due to the use of a spring for the constant pressure preload, the preload force can be kept constant, even when the distance between the two bearings fluctuates under the influence of operating heat and load.

Also, the standard preload for the paired angular contact ball bearings is shown in Table 8.12. Light and normal preload is applied to prevent general vibration, and medium and heavy preload is applied especially when rigidity is required.

8.4.3 Preload and rigidity

The increased rigidity effect preloading has on bearings is shown in Fig. 8.5. When the offset inner rings of the two paired angular contact ball bearings are pressed together, each inner ring is displaced axially by the amount δ_o and is thus given a preload, F_o, in the direction shown. Under this condition, when external axial load F_a is applied, bearing I will have an increased displacement by the amount δa and bearing II's displacement will decrease. At this time the loads applied to bearing I and II are F_I and F_{II}, respectively.

Under the condition of no preload, bearing I will be displaced by the amount δ_b when axial load F_a is applied. Since the amount of displacement, δ_a, is less than δ_b, it indicates a higher rigidity for δ_a.

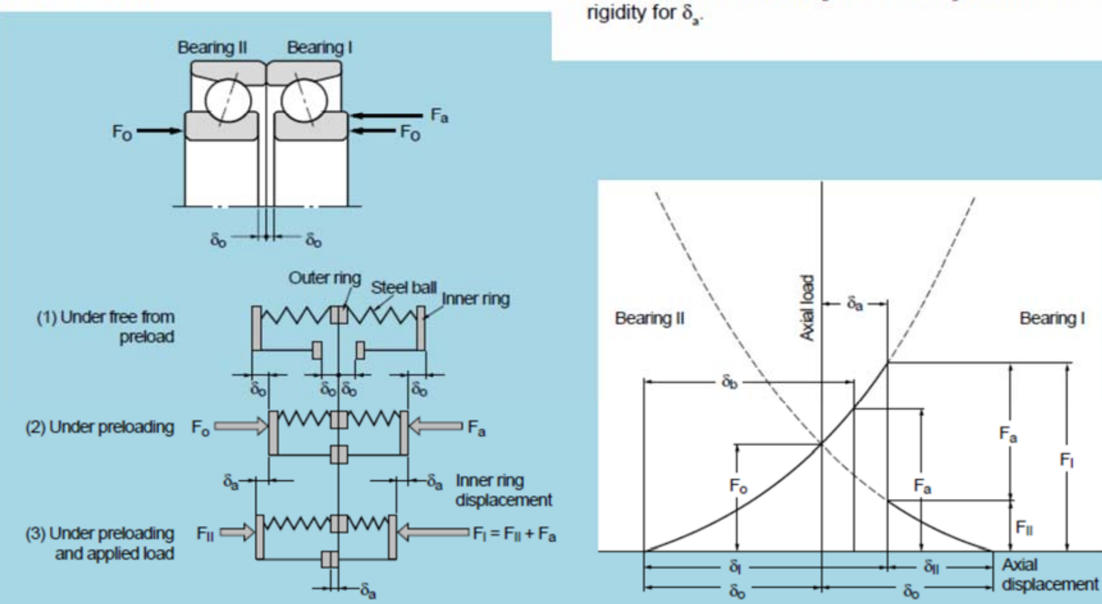


Fig. 8.5 Fixed position preload versus axial displacement

Attachment 4: NTN Bearing Manual Spherical Roller Bearing Preload Guideline

Table 15.1 Installation of tapered bore spherical roller bearings

Units mm

Nominal bearing bore diameter *d*		Reduction of radial internal clearance		Axial displacement drive up				Minimum allowable residual clearance		
				Taper, 1:12		Taper, 1:30				
over	incl.	Min	Max	Min	Max	Min	Max	CN	C3	C4
30	40	0.02	0.025	0.35	0.4	—	—	0.015	0.025	0.04
40	50	0.025	0.03	0.4	0.45	—	—	0.02	0.03	0.05
50	65	0.03	0.035	0.45	0.6	—	—	0.025	0.035	0.055
65	80	0.04	0.045	0.6	0.7	—	—	0.025	0.04	0.07
80	100	0.045	0.055	0.7	0.8	1.75	2.25	0.035	0.05	0.08
100	120	0.05	0.06	0.75	0.9	1.9	2.25	0.05	0.065	0.1
120	140	0.065	0.075	1.1	1.2	2.75	3	0.055	0.08	0.11
140	160	0.075	0.09	1.2	1.4	3	3.75	0.055	0.09	0.13
160	180	0.08	0.1	1.3	1.6	3.25	4	0.06	0.1	0.15
180	200	0.09	0.11	1.4	1.7	3.5	4.25	0.07	0.1	0.16
200	225	0.1	0.12	1.6	1.9	4	4.75	0.08	0.12	0.18
225	250	0.11	0.13	1.7	2	4.25	5	0.09	0.13	0.2
250	280	0.12	0.15	1.9	2.4	4.75	6	0.1	0.14	0.22
280	315	0.13	0.16	2	2.5	5	6.25	0.11	0.15	0.24
315	355	0.15	0.18	2.4	2.8	6	7	0.12	0.17	0.26
355	400	0.17	0.21	2.6	3.3	6.5	8.25	0.13	0.19	0.29
400	450	0.2	0.24	3.1	3.7	7.75	9.25	0.13	0.2	0.31
450	500	0.21	0.26	3.3	4	8.25	10	0.16	0.23	0.35
500	560	0.24	0.3	3.7	4.6	9.25	11.5	0.17	0.25	0.36
560	630	0.26	0.33	4	5.1	10	12.5	0.2	0.29	0.41
630	710	0.3	0.37	4.6	5.7	11.5	14.5	0.21	0.31	0.45
710	800	0.34	0.43	5.3	6.7	13.3	16.5	0.23	0.35	0.51
800	900	0.37	0.47	5.7	7.3	14.3	18.5	0.27	0.39	0.57
900	1,000	0.41	0.53	6.3	8.2	15.8	20.5	0.3	0.43	0.64
1,000	1,120	0.45	0.58	6.8	8.7	17	22.5	0.32	0.48	0.7
1,120	1,250	0.49	0.63	7.4	9.4	18.5	24.5	0.34	0.54	0.77

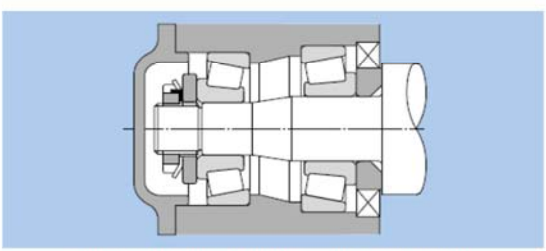


Fig. 15.12 Axial internal clearance adjustment

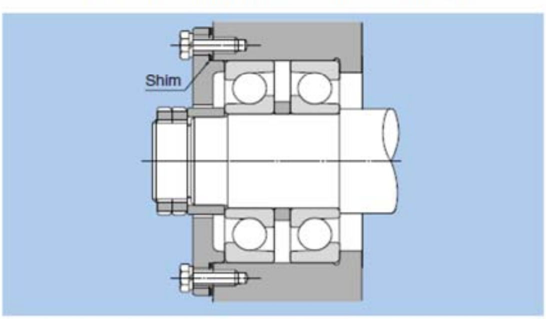


Fig. 15.14 Internal clearance adjustment using shims

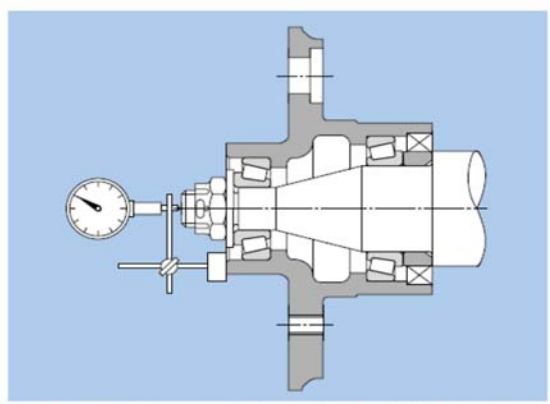


Fig. 15.13 Measurement of axial internal clearance adjustment

Attachment 5: Pinion Gear Tooth Contact Pattern Verification Procedure

The input shaft spiral helical pinion and gear were found to have a contact pattern in the 55% to 60% range only on the three gearboxes opened. Contact found as “High Pattern”, Figure 5.42.

5 Assembly and Installation

3. Rotate the ring gear FORWARD and BACKWARD so that the 12 gear teeth go past the drive pinion six times to get the contact patterns. Repeat if needed to get a more clear pattern.
4. Look at the contact patterns on the ring gear teeth. Compare the patterns to Figure 5.41, Figure 5.42 and Figure 5.43.

   The location of good hand-rolled contact patterns for new gear sets is from the central toe, center, to the central heel area of the gear tooth and in the center between the top and bottom of the tooth. Figure 5.41.

   When the carrier is operated, a good hand-rolled pattern will extend approximately the full length of the gear tooth. The top of the pattern will be near the top of the gear tooth. Figure 5.44.

   The location of a good hand-rolled contact pattern for an old gear set must match the wear pattern in the ring gear. The new contact pattern will be smaller in area than the old wear pattern.

   A high-contact pattern indicates that the drive pinion was not installed deep enough into the carrier. A low-contact pattern indicates that the drive pinion was installed too deep in the carrier.

   - **If the contact patterns require adjustment:** Continue by following Step 5 to move the contact patterns between the top and bottom of the gear teeth.
   - **If the contact patterns are in the center of the gear teeth:** Proceed to Step 6.

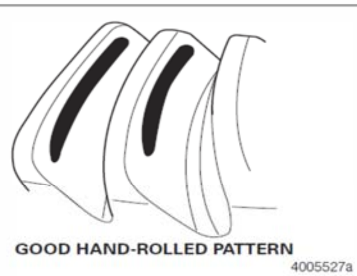


Figure 5.41

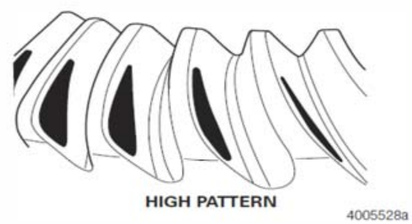


Figure 5.42

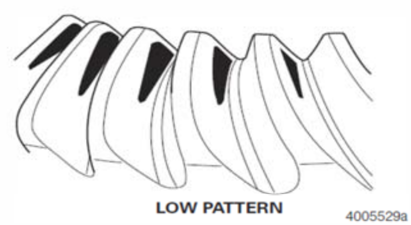


Figure 5.43

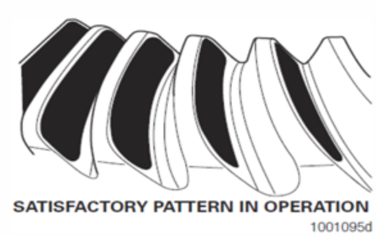


Figure 5.44

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Attachment 6: NTN Bearing Manual Lubrication Guideline

Table 11.4 Selection standards for lubricating oils

Operating temperature of bearings °C	*dn*–value	Viscosity grade of lubricating oil		Bearing type
		Ordinary load	Heavy or Impact load	
–30 to 0	Up to the allowable revolution	22 32	46	All type
0 to 60	Up to 15,000	46 68	100	All type
	15,000 to 80,000	32 46	68	All type
	80,000 to 150,000	22 32	32	Except thrust ball bearings
	150,000 to 500,000	10	22 32	Single row radial ball bearings, cylindrical roller bearings
60 to 100	Up to 15,000	150	220	All type
	15,000 to 80,000	100	150	All type
	80,000 to 150,000	68	100 150	Except thrust ball bearings
	150,000 to 500,000	32	68	Single row radial ball bearings, cylindrical roller bearings
100 to 150	Up to the allowable revolution	320		All type
0 to 60		46 68		Spherical roller bearings
60 to 100		150		

Notes: 1. In case of oil drip or circulating lubrication
2. In case the usage conditions' range is not listed in this table, please refer to NTN.

11.3.3 Oil quality

In forced oil lubrication systems, the heat radiated away by housing and surrounding parts plus the heat carried away by the lubricating oil is approximately equal to the amount of heat generated by the bearing and other sources.

For standard housing applications, the quantity of oil required can be found by formula (11.1).

$$Q = K \bullet q \cdots\cdots\cdots\cdots (11.1)$$

where,

- Q : Quantity of oil for one bearing cm3/min
- K : Allowable oil temperature rise factor (Table 11.5)
- q : Minimum oil quantity cm³/min (From chart)

Because the amount of heat radiated will vary according to the shape of the housing, for actual operation it is advisable that the quantity of oil calculated by formula (11.1) be multiplied by a factor of 1.5 to 2.0. Then, the amount of oil can be adjusted to correspond to the actual machine operating conditions. If it is assumed for calculation purposes that no heat is radiated by the housing and that all bearing heat is carried away by the oil, then the value for shaft diameter, d, (second vertical line from right in Fig. 11.12) becomes zero, regardless of the actual shaft diameter.

Table 11.5 Factor *K*

Temperature rise, °C	*K*
10	1.5
15	1
20	0.75
25	0.6

(Example)

For tapered roller bearing 30220U mounted on a flywheel shaft with a radial load of 9.5 kN, operating at 1,800 rpm; what is the amount of lubricating oil required to keep the bearing temperature rise below 15°C?

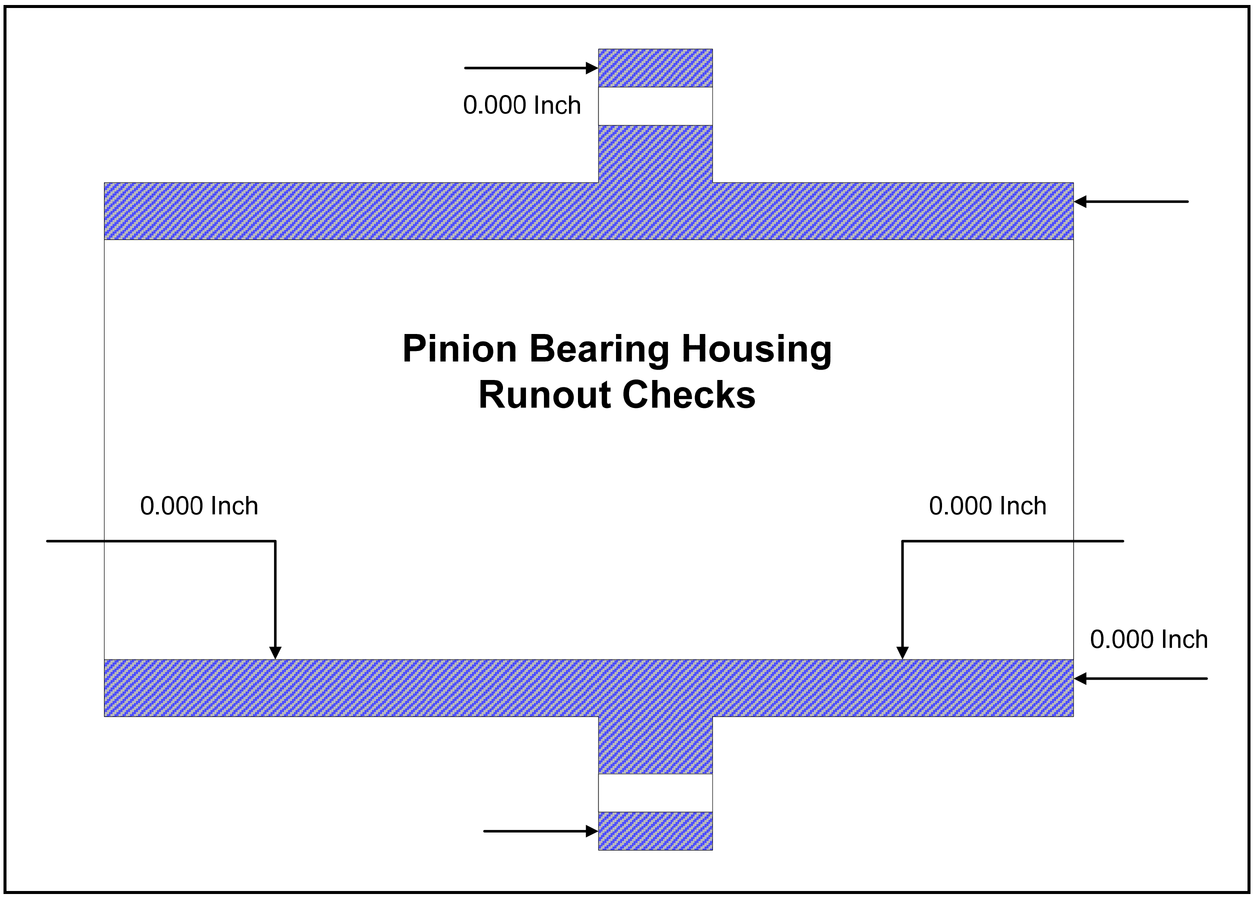
d=100 mm, dn=100×1,800=18×10⁴ mm r/min

from Fig. 11.12, q=180 cm³/min.

Assume the bearing temperature is approximately equal to the outlet oil temperature, from Table 11.5, since K=1, Q=1×180=180 cm3/min.

Attachment 7: Runout Checks at Shops to Check Factory Machining Accuracy

No machining errors found



Attachment 8: Typical Normal Vibration Readings at Motor to Gear Beam Structure

MEASUREMENT POINT	OVERALL LEVEL	Acceleration
BM-0601A - COOLING TOWER FAN	OVERALL LEVEL	1K-20 kHz
MOH - Motor Outboard Horizontal	.103 In/Sec	.289 G-s
MOV - Motor Outboard Vertical	.109 In/Sec	.378 G-s
MIH - Motor Inboard Horizontal	.107 In/Sec	.306 G-s
MIV - Motor Inboard Vertical	.043 In/Sec	.193 G-s
MIA - Motor Inboard Axial	.117 In/Sec	.807 G-s
BM-0601B - COOLING TOWER FAN	OVERALL LEVEL	1K-20 kHz
MOH - Motor Outboard Horizontal	.085 In/Sec	.254 G-s
MOV - Motor Outboard Vertical	.086 In/Sec	.440 G-s
MIH - Motor Inboard Horizontal	.075 In/Sec	.257 G-s
MIV - Motor Inboard Vertical	.044 In/Sec	.192 G-s
MIA - Motor Inboard Axial	.089 In/Sec	.762 G-s
BM-0601C - COOLING TOWER FAN	OVERALL LEVEL	1K-20 kHz
MOH - Motor Outboard Horizontal	.146 In/Sec	.881 G-s
MOV - Motor Outboard Vertical	.153 In/Sec	.536 G-s
MIH - Motor Inboard Horizontal	.059 In/Sec	.159 G-s
MIV - Motor Inboard Vertical	.122 In/Sec	.205 G-s
MIA - Motor Inboard Axial	.105 In/Sec	.243 G-s
BM-0601D - COOLING TOWER FAN	OVERALL LEVEL	1K-20 kHz
MOH - Motor Outboard Horizontal	.074 In/Sec	.076 G-s
MOV - Motor Outboard Vertical	.059 In/Sec	.428 G-s
MIH - Motor Inboard Horizontal	.049 In/Sec	.155 G-s
MIV - Motor Inboard Vertical	.071 In/Sec	.309 G-s
MIA - Motor Inboard Axial	.092 In/Sec	.492 G-s

Attachment 9: Lube Oil Sample from Failed Gearbox CT3B31-H Unit

ALHAMRANI FUCHS PETROLEUM SAUDI ARABIA LTD

FUCHS

CENT

Petrorabigh

Rabigh
Saudi Arabia

Condition Monitoring
Yanbu Blending & Grease Plant
P.O.Box 30439,
Yanbu Al Sinaiyah,
Saudi Arabia.
Telephone : + 966 4 396 2487
Fax : + 966 4 396 3783
http://www.cent.fuchs.com.sa

Unit :		Customer Ref:	CT3 B 031H
Equipment:		CENT Ref:	
Compartment:		Lubricant :	Petromin VG 320
Sample Point:		Equipment Location:	

Sample Number	Date Sampled	Date Received	Oil Hrs/Kms	Equip. Hrs/Kms	Oil-Topup (L)	Oil Changed	Filter Changed	Label Code	Lab. Code	Appearance	
1		17-Mar-2010							37662	Very Dark Brown	

Sample No.	KV40	Water	TAN	Crackle							
	D 445	D 1744	D 664	FPSA-1							
	cSt	ppm	mg KOH/g								
1	316.30	169.00	0.22	0.00							

Sample No.	Pb	Fe	Zn	Si	Al	Cu	Cr	Sn	Mg	Ni	Mo	Ca	P	Mn	Sb	Li	
	D 5185	D 5185	D 5185	D 5185	D 5185	D 5185	D 5185	D 5185	D 5185	D 5185	D 5185	D 5185	D 5185	D 5185	D 5185	D 5185	
	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	
1	0	128	33	24	5	0	1	0	2	0	0	6	117	0	0	0	

Sample Note :

Interpretation of results based on the Current Sample :

Wear metal detected (Fe), to be monitored further.
Dirt entry detected (Si & Al), Kindly check air intake system.

Date : **23-Mar-2010**

Signed By : **Saaid Ali Gamdi**

ISO 9001:2000
CERTIFIED
BSI FM-25054

ISO 14001:1996
CERTIFIED
BSI EMS-52991

CURRENT STATUS

CAUTION