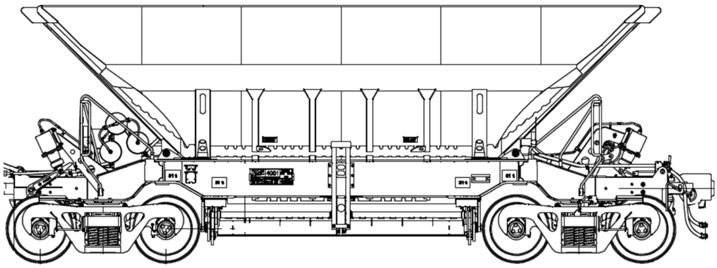
TECHNICAL REPORT

Inspection and Analysis of the Functioning of the Bearings Used on Railways

A study of the life of a bearing under real operating conditions



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Technical Report

Luleå University of Technology
Department of Civil, Environmental and Natural Resources Engineering
Division of Operation and Maintenance Engineering

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First of all, I thank God for allowing me to stay here and give me the health and fitness necessary to finalize this Master's Thesis.

Then, principally I would like to thank Professor Diego Galar for his specialised technical advice and recommendations, which were a great help in completing this thesis. Besides, I must thank the Vicente Macian and Bernardo Tormos professors, for giving their knowledge to support the elaboration of this work.

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Numan Perales Luna
March, 2014
Luleå, Sweden

ABSTRACT

The objective of the present work is to carry out an analysis of the performance and life of bearings used in railways transporting ore. The project determined the nominal life (L_{10}) for a single type of bearing commonly used in railways (SKF CTBU Class K and Timken AP-2^{TM}). To do so, it considered two methodologies. Methodology I is an alternative approach proposed by a few authors in Brazil, inspired by a procedure used to design rail shafts. Methodology II is based on well-established techniques proposed by the bearing manufacturers (as SKF and Timken).

After of have estimated L_{10}, is determined the bearing lifetime (L) using two correction parameters (one based on reliability (a_1) and the other based on the working conditions of the bearing (a_{ISO})). This in order to determine whether it was being replaced at the appropriate time, to improve its replacement periods, decrease maintenance costs, make the most of the life of the bearing and detect possible causes of faults it may present at any given time.

In addition, it discusses some Condition Monitoring Techniques that can help identify anomalies, faults and breakdowns in these bearings. It touches on the phenomenon of electric current through wheel bearings, the way to identify, detect and prevent it, and some measurement techniques. Finally, it explains in detail the bearings process reconditioning (mounting, dismounting and reconditioning) that accomplished some companies in order to maximize the life of these.

KEYWORDS: *Railway, Bearing, Nominal Life, Bearing Life, Maintenance, Ore, Wagons, Replacement Period, Costs.*

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Part I
SUMMARY

1

INTRODUCTION

1.1. Background.

Bearings are one of the most important components in a railway vehicle for safety, since a failsafe design is not available. Contact fatigue, internal clearance, corrosion, and contamination of the lubricating oil can cause bearing failures. Generally, failures show up as imperfections in the ball race, in the ball/roller or in the retainer. The more frequent defects are caused by contact fatigue [1].

The bearing life is characterized by the formation of small flaws in the surface of the track that tend to grow, resulting in the loss of performance of the bearing. Bearings are selected based on a theoretical life, defined as the nominal life L_{10}, or the lifetime reached by 90% of them. The characterization of the conditions of use is fundamental to estimate L_{10}. Several factors have an effect on bearing life, causing it to vary significantly. The relationship between the life after the appearance of the first defect and L_{10} is approximately 10% [1]. Motivated by this concern, the division of Operation and Maintenance Engineering (Investigation Centre at Luleå University of Technology) decided to determine the bearing life (L_{10}) and with help of correction factors, to determine the lifetime (L) of a bearing type most commonly used on the railways. The intention was to compare L with the period of replacement of the bearings, to improve spare periods, reduce maintenance costs and, above all, to use the maximum bearing lifetime. To ensure that many railway companies can benefit from the results, the study takes into account other important variables in the bearing's operation, such as rotational speed, temperature, lubrication, wear and friction, and even weather conditions.

The analytical methodology (Methodology I) is an alternative approach proposed by the authors [1], inspired by a procedure used to design rail shafts [2].

The purpose is to compare the obtained life using methodology I, with the life estimated by LKAB Company. The latter uses a calculation method well-established in manuals of bearing manufacturers such as SKF and Timken. This is to avoid a drawback observed in this type of approach, namely, the lack of rigour in defining terms associated with the mathematical model used to estimate L_{10}.

The research also touches on the process of reconditioning bearings, as it is much more profitable to recondition the large bearings instead of using new bearings. When a reconditioned bearing is used instead of a new bearing, this allows a reduction in inventory costs relating to maintenance

A final important point discussed in this thesis is the passage of electrical current through rail bearings. The technical report will explain how this phenomenon occurs and suggest some methods of measuring currents and voltages in wheel bearings. Since no current can pass through a bearing without creating voltage, the voltage bearing was chosen as an important parameter to study. Finally, knowing the distribution of the electric current along train can help determine whether by pass conductors will help.

1.2. Research goal.

To perform an inspection and analysis of the functioning of the bearings most commonly used on railways in order to estimate the bearings' life under real operating conditions.

Part II

THEORETICAL FOUNDATION

ANALYTICAL METHODS TO ESTIMATE THE LIFE L_{10}. 2

This section presents the methodology used to estimate L_{10}. It represents a refined procedure inspired by an approach used to design rail shafts [2] which describes in detail the loads applied to the bearings.

2.1. Methodology I.

As mentioned previously, the methodology to estimate L_{10} is based on procedures used to design rail shafts [JIS/4501]. The following steps are necessary for implementation [1, 2].

2.1.1. DETERMINATION OF THE LOADING ON THE BEARINGS.

In the design of rail shafts, two types of loading should be considered: (i) the static load W, characterized by the weight, and (ii) the dynamic loads F_a and F_r, generated by the vibratory response of the wagon due to variation of the wagon speed and irregularities of the rails. Such irregularities are provoked by vertical and horizontal undulations along the rails [1].

2.1.2. STATIC LOADING – W.

Considering the loading conditions on the wagon, the static load (W) can be calculated by:

$$W_c = \frac{M_c * g}{N_e}, \tag{1}$$

and

$$W_v = \frac{M_v * g}{N_e},$$ (2)

Where M is the mass of the wagon, *g* is the acceleration due to gravity, N_e is the number of shafts per wagon, and the subscripts c and v represent the conditions of full and empty wagons, respectively [1].

2.1.3. DYNAMIC LOADING – F_A AND F_R.

The horizontal (axial) and vertical (radial) components of the dynamic forces, respectively F_{ai} and F_{ri}, can be obtained by determining the equilibrium of forces and moments in the wagon [Eqs. (3) – (6)]. The free body diagram of the system can be observed in figure 1. Then:

$$F_{ac} = \alpha_{Lc} * \frac{M_c * g}{N_e * N_r},$$ (3)

$$F_{av} = \alpha_{Lv} * \frac{M_v * g}{N_e * N_r},$$ (4)

$$F_{rc} = \alpha_{Lc} * \frac{M_c * g * h_c}{N_e * N_r * j},$$ (5)

$$F_{rv} = \alpha_{Lv} * \frac{M_v * g * h_v}{N_e * N_r * j},$$ (6)

where *h* is the vertical distance from the axle to the centre of gravity of the wagon, *j* is the distance between the bearings, α_L is the coefficient of horizontal acceleration, and N_e and N_r are the number of axles and the bearings per axle, respectively [1].

The coefficient of horizontal acceleration, α_L, is a parameter used to quantify the effect of the dynamic loading on the wagon. Basically, this coefficient depends on the speed of the wagon and on the conditions of the railroad. As the rail system did not have this information, as reference, we have adopted the values given in [3] and reported in table 1.

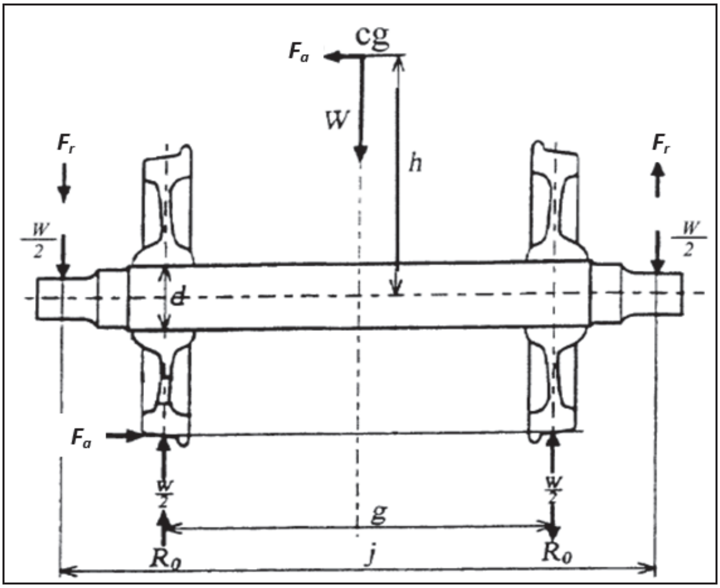


Figure 1. Defenition of the loads applied to the rail shaft [1].

Table 1. Coefficient of acceleration [3].

Railway system	Class	Velocity V (km h^{-1})	Vertical accelerationα_V	Horizontal accelerationα_L
System 1	SA	200–350	$0.0027V$	$0.030 \pm 0.00060V$
	A	150–200	$0.0027V$	$0.030 \pm 0.00085V$
System 2	A	60–160	$0.0027V$	$0.040 \pm 0.0012V$
		< 60	0.16	0.11
	B	60–130	$0.0052V$	$0.060 \pm 0.0018V$
		< 60	0.31	0.17

SA is the especially improved tracks for the velocity $>$ 200 up to 350 km h^{-1}
A in the system 1 is the usual high speed railway line for the velocity over 150 up to 280 km h^{-1}
A in the system 2 is the especially improved tracks in the conventional line
B is the conventional track line

2.1.3.1. Centre of gravity.

A vehicle's centre of gravity is the basis for an analysis of its behaviour; to determine the centre of gravity, it is necessary to know what loads it can take [9].

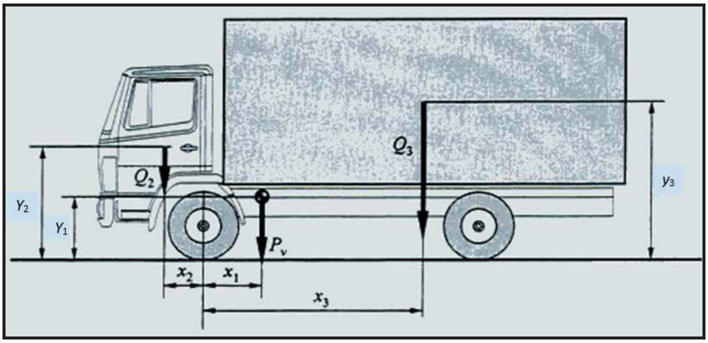


Figure 2. Analytical determination of the centre of gravity of a vehicle [11].

To calculate the centre of gravity, it is necessary to know the exact position of the various loads in the vehicle. Otherwise, the method will become experimental, not analytical. The curb weight of the vehicle is a point charge located at the centre of gravity of an unladen vehicle. The three-dimensional coordinates of the centre of gravity are obtained by the following expressions [9]:

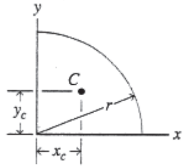
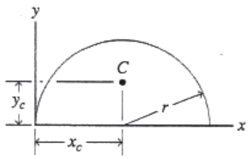
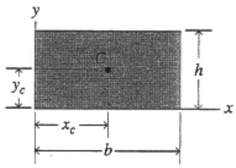
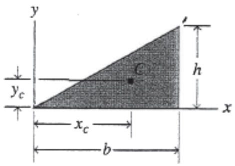
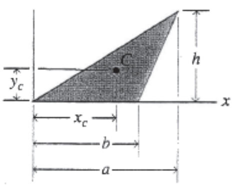
$$x_{cdg} = \frac{\sum_i Q_i * x_i}{\sum_i Q_i}, \qquad (7)$$

$$y_{cdg} = \frac{\sum_i Q_i * y_i}{\sum_i Q_i}, \qquad (8)$$

$$z_{cdg} = \frac{\sum_{i} Q_i * z_i}{\sum_{i} Q_i}, \tag{9}$$

When all loads are centred on the median longitudinal plane of the vehicle or vehicle composition (load symmetry), the coordinate "Z" disappears and is taken as nil [9].

Table 2. Area, center of gravity and moment of inertia [11].

Quarter circular arc $L = \frac{\pi r}{2}$ $x_C = \frac{2r}{\pi}$ $y_C = \frac{2r}{\pi}$	
Semicircular arc $L = \pi r$ $x_C = r$ $y_C = \frac{2r}{\pi}$	
Rectangular area $A = bh$ $x_C = \frac{b}{2}$ $y_C = \frac{h}{2}$	
Triangular area $A = \frac{bh}{2}$ $x_C = \frac{2b}{3}$ $y_C = \frac{h}{3}$	
Triangular area $A = \frac{bh}{2}$ $x_C = \frac{a+b}{3}$ $y_C = \frac{h}{3}$	

2.1.4. EQUIVALENT DYNAMIC BEARING LOAD.

The above information can be used to calculate the bearing load P. When the bearing load fulfils the requirements for the basic dynamic load rating C, i.e. the load is constant in magnitude and direction and acts radially on a radial bearing or axially and centrically on a thrust bearing, then $P = F$ and the load may be inserted directly into the life equations [5, 6].

In all other cases, the equivalent dynamic bearing load must be calculated first. The equivalent dynamic bearing load is defined as that hypothetical load, constant in magnitude and direction, acting radially on a radial bearing or axially and centrically on a thrust bearing which, if applied, would have the same influence on bearing life as the actual loads to which the bearing is subjected (see figure 3) [5, 6].

Radial bearings are often subjected to simultaneously acting radial and axial loads. If the resultant load is constant in magnitude and direction, the equivalent dynamic bearing load P can be obtained from the general equation [5, 6]:

$$P = \left(X * F_r\right) + \left(Y * F_a\right), \tag{10}$$

where,

P = Equivalent dynamic bearing load [KN].
F_r = Actual radial bearing load [KN].
F_a = Actual axial bearing load [KN].
X = Radial load factor for the bearing.
Y = Axial load factor for the bearing.

SKF applications related to the railway industry, where the equivalent bearing load to be used for the L_{10} basic rating life calculation is either as described above or based on specific customer load configurations, are the following [9]:

$P = F_r + \left(Y * F_a\right)$ for tapered roller bearings and spherical roller bearings. This means:

$$X = 1$$
$$Y = 2,55$$

and,

$P = F_r$ for cylindrical roller bearings.

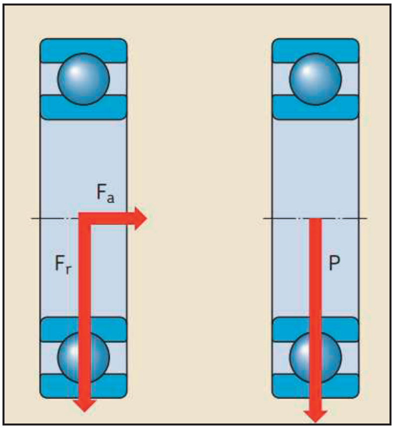


Figure 3. Equivalent dynamic bearing load.

2.1.5. DETERMINATION OF THE MEAN EQUIVALENT DYNAMIC LOAD ON THE BEARING.

Once the horizontal and vertical components of load on the bearing are determined, it is necessary to evaluate the equivalent dynamic load (P).

According to the bearing manufacturer [SKF] [5,6], this is determined by Eq. (10):

$$P = (X * F_r) + (Y * F_a), \tag{10}$$

Depending on the wagon load conditions Eq. (10) can be rewritten as:

$$P_c = X * \alpha_{Lc} * \frac{M_c * g * h_c}{N_e * N_r * j} + Y * \alpha_{Lc} * \frac{M_c * g}{N_e * N_r}, \qquad (10a)$$

and,

$$P_v = X * \alpha_{Lv} * \frac{M_v * g * h_v}{N_e * N_r * j} + Y * \alpha_{Lv} * \frac{M_v * g}{N_e * N_r}, \qquad (10b)$$

where the subscript v and c represent the conditions of full and empty wagons, respectively [1].

The mean equivalent dynamic load used to estimate the bearing's life, F_m, is given by:

$$F_m = \sqrt{\frac{F_1^3 * u_1 + F_2^3 * u_2 + \ldots + F_k^3 * u_k}{\sum_{i=1}^{k} u_i}}, \qquad (11)$$

Where F_i, i =1. . . K, are the possible loads applied to the wagon along the track, and u_i, are their respective periods of operation. Considering the specific conditions of this analysis, the mean equivalent dynamic load can be quantified by [1]:

$$F_m = \sqrt[3]{0{,}5 * \left\{ \frac{1}{f_{rc} + f_c} * \left[f_c * (P_c)^3 + f_{rc} * \left(\frac{W_c}{N_r} \right)^3 \right] + \frac{1}{f_{rv} + f_v} * \left[f_v * (P_v)^3 + f_{rv} * \left(\frac{W_v}{N_r} \right)^3 \right] \right\}}$$

(12)

Where f_{rc} and f_{rv} [Hz] are the frequencies of W_c and W_v, and f_c and f_v [Hz] are the frequencies of oscillation of the wagon along the track. In order to determine these frequencies, it is necessary to consider that (i) f_{rc} and f_{rv} are proportional to the rotation of the axle, and (ii) f_c and f_v depend on the conditions of the rail track and the train speed. Here is assumed that: f_c and f_v equal 0,5 Hz, and f_{ri} can be calculated by [1]:

$$f_{ri} = \frac{2 * V_i}{D_w}, \tag{13}$$

Where V_i [m/s] is the wagon speed and D_w [m] is the wheel diameter.

The L_{10} is calculated by:

$$L_{10} = \frac{\pi * D_w}{1000} * \left(\frac{C}{F_m}\right)^p * 1{,}5E^{+6}, \tag{14}$$

Where C is the dynamic load capacity of the bearing and p is an exponent for the life equation [1, 9].

3

CORRECTION FACTORS OF LIFE.

Using the catalogue, bearing manufacturers can calculate the life of a bearing under given load conditions, with a failure probability of 10%. However, this is not good enough; it is necessary to calculate the life with a reliability greater than 90%, for given working conditions, etc. So once the nominal life, L_{10}, of the bearing is calculated, it is necessary to apply a series of correction factors to calculate the lifetime L [4].

The bearing life (L) in ISO 281:2007 is given by the following expression:

$$L = a_1 * a_{ISO} * L_{10}\ , \qquad (15)$$

3.1 Reliability (a_1):

The nominal life of a bearing is calculated for a 90% probability of survival. There are applications in which this reliability is insufficient (equipment medical, nuclear, etc...). To account for greater reliabilities, a correction coefficient is included for reliability (a_1), as calculated in the following table:

Table 3. Reliability Factors [4].

Reliability %	a_1
90	1
95	0.64
96	0.55
97	0.47
98	0.37
99	0.25
99.2	0.22
99.4	0.19
99.6	0.16
99.8	0.12
99.9	0.093
99.92	0.087
99.94	0.080
99.95	0.077

3.2. Working conditions (a_{ISO}):

This factor has replaced the previously used (a_{23}). It is necessary to take the working conditions into account, especially inadequate lubrication. ISO 281:2007 allows each bearing manufacturer to define the procedure for obtaining this correction factor, usually expressed in terms of the relationship between the load and fatigue load limit bearing P_u (this never causes failure if lubrication is suitable), lubricant contamination, viscosity at the working temperature and dimensions of the bearing and speed [4].

However, the equation $L = a_1 * a_{ISO} * L_{10}$ does not provide reliable results when [4]:

- The load applied is very high (greater than 50% C_0 or C).
- The rotational speed is very low (less than 20 rpm) or very high (greater than the bearing's speed limit).
- The temperature is high (above 130 ° C).

- ➢ The lubricant is contaminated with water.
- ➢ The misalignment is excessive.
- ➢ Electric current is passing through the bearing.
- ➢ The bearing is subjected to high vibration.

It is can calculate the factor a_{ISO} that brings together the two previous effects by following a standard procedure [4]:

1. Figure # 4 obtains the relative viscosity (ν_1) based on the average diameter of a bearing (d_m); manufacturers use this to define the dynamic load capacities of their bearings [4].

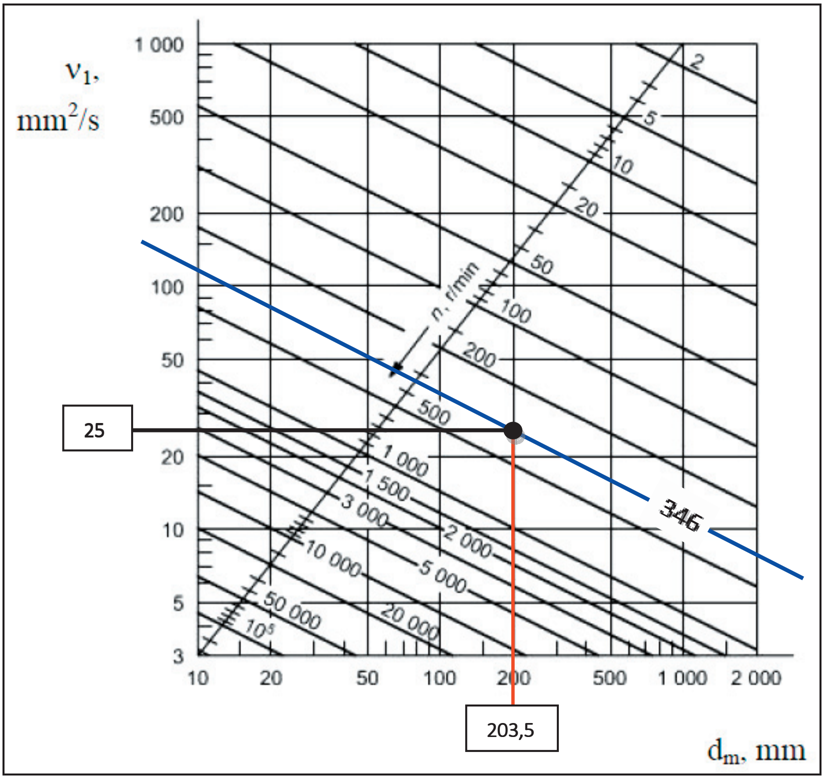


Figure 4. Relative Viscosity (ν_1).

2. Figures 5 and 6 show the viscosity (ν) of the oil at the operating temperature of a bearing [4].

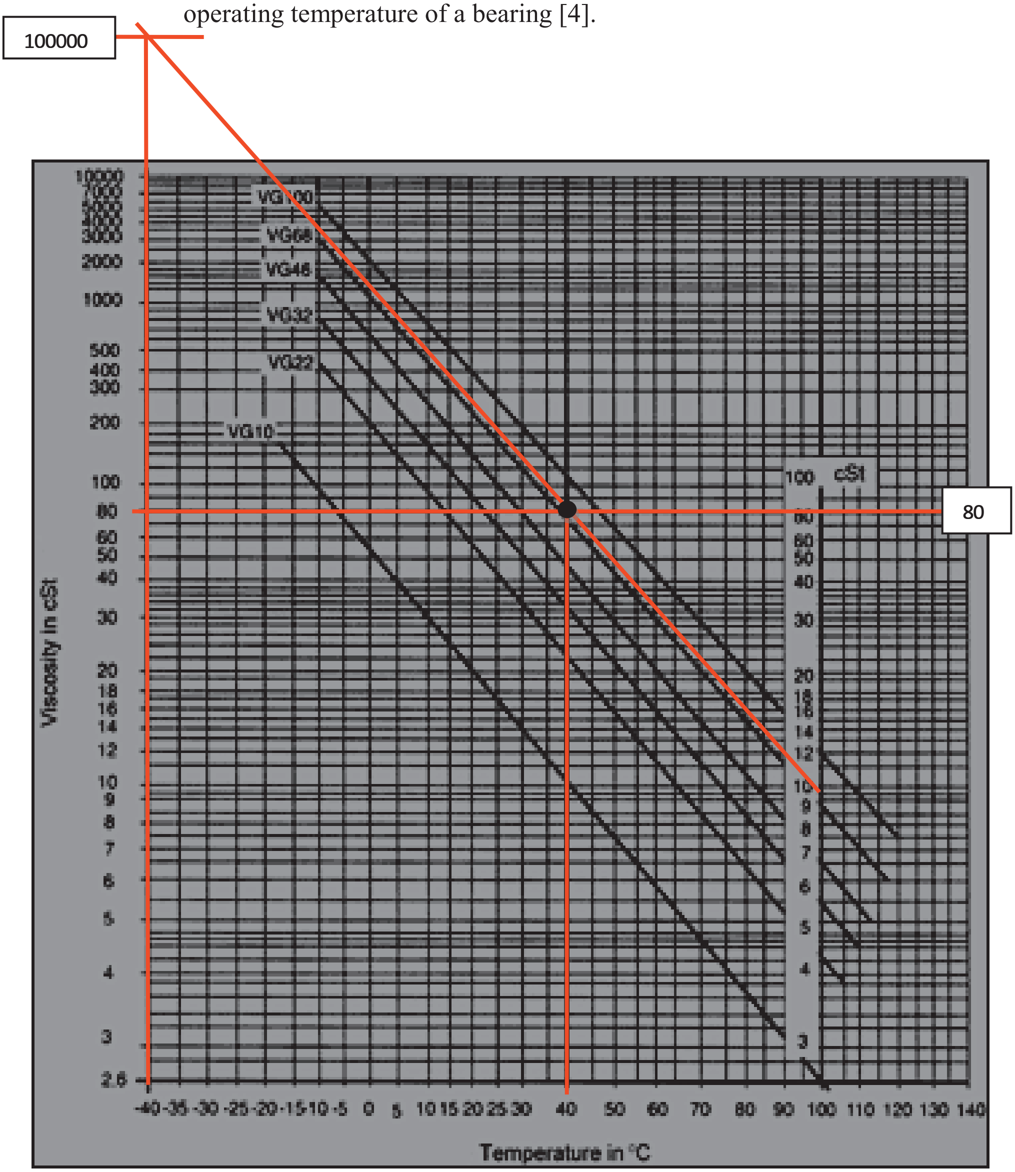


Figure 5. Variation of the kinematic viscosity of the lubricant with temperature (low temperatures) [20].

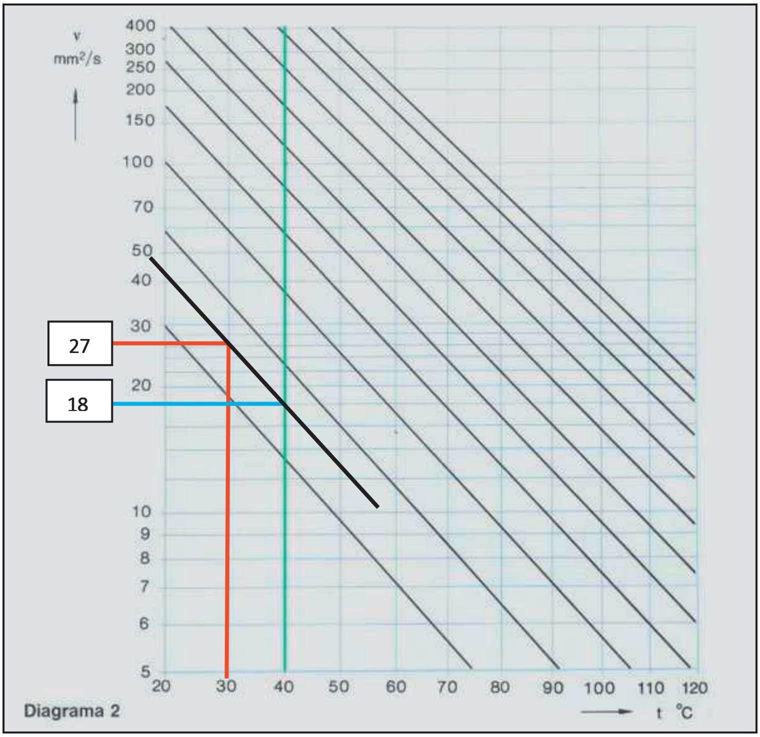


Figure 6. Variation of the kinematic viscosity of the lubricant with temperature.

3. The ratio of these two viscosities is identified by the letter kappa $\kappa = \nu / \nu_1$. If $\kappa < 0.4$, there will be contact between the solid elements (raceways and rolling elements) making the use of additives EP or solid lubricants (such as graphite and molybdenum disulfide) necessary. If these additives are used and the contamination level is not high, $\eta_c < 0.2$ $\kappa = 1$ is acceptable, but a_{ISO} must be limited to a maximum value of 3 [4].

4. The level of contamination of the lubricant provides the factor η_c; this factor can be obtained from a simplified form of table 4.

Table 4. Contamination Factor (η_c) [5].

Guideline values for factor η_c for different levels of contamination

Condition	Factor η_c [1)] for bearings with diameter $d_m < 100$ mm	$d_m \geq 100$ mm
Extreme cleanliness Particle size of the order of the lubricant film thickness Laboratory conditions	1	1
High cleanliness Oil filtered through extremely fine filter Conditions typical of bearings greased for life and sealed	0,8 ...0,6	0,9 ... 0,8
Normal cleanliness Oil filtered through fine filter Conditions typical of bearings greased for life and shielded	0,6 ... 0,5	0,8 ... 0,6
Slight contamination Slight contamination in lubricant	0,5 ... 0,3	0,6 ... 0,4
Typical contamination Conditions typical of bearings without integral seals, coarse filtering, wear particles and ingress from surroundings	0,3 ... 0,1	0,4 ... 0,2
Severe contamination Bearing environment heavily contaminated and bearing arrangement with inadequate sealing.	0,1 ... 0	0,1 ... 0
Very severe contamination (under extreme contamination values of η_c can be outside the scale resulting in a more severe reduction of life than predicted by the equation for L_{nm})	0	0

[1)] The scale for η_c refers only to typical solid contaminants. Contamination by water or other fluids detrimental to bearing life is not included. In case of very heavy contamination ($\eta_c = 0$), failure will be caused by wear, the useful life of the bearing can be shorter than the rated life

5. It is necessary to calculate the contamination-load relation $(\eta_c * P_u)/F$ for the type of bearing chosen to get the correct working conditions (a_{ISO}). The charts containing this information are the following [4]:

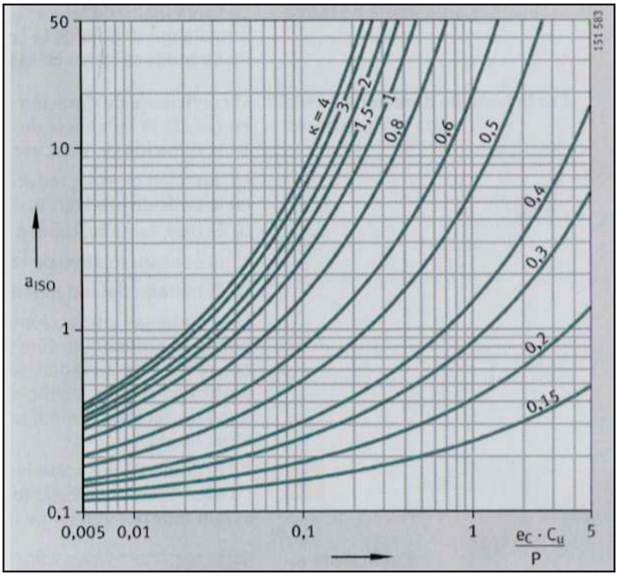


Figure 7. a_{ISO} factor for radial ball bearings [4].

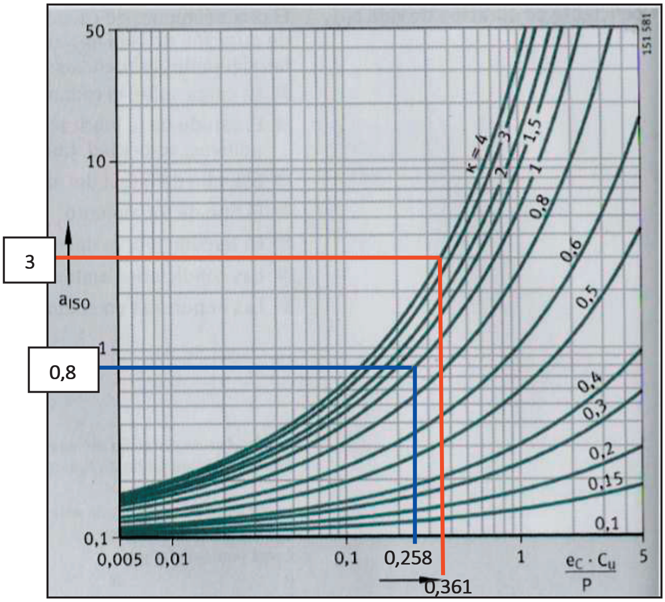


Figure 8. a_{ISO} factor for radial roller bearings [4].

3.3. Requisite rating life.

When determining bearing size, it is necessary to verify SKF's calculation of the rating life with the specification life of the application, if it is available. This usually depends on the type of machine and the requirements for duration of service and operating reliability. In the absence of previous experience, the guideline values listed in tables 5 and 6 can be used [5].

Table 5. Guideline values of life for different machine types [5].

Machine type	Specification life Operating hours
Household machines, agricultural machines, instruments, technical equipment for medical use	300 ... 3 000
Machines used for short periods or intermittently: electric hand tools, lifting tackle in workshops, construction equipment and machines	3 000 ... 8 000
Machines used for short periods or intermittently where high operational reliability is required: lifts (elevators), cranes for packaged goods or slings of drums etc.	8 000 ... 12 000
Machines for use 8 hours a day, but not always fully utilized: gear drives for general purposes, electric motors for industrial use, rotary crushers	10 000 ... 25 000
Machines for use 8 hours a day and fully utilized: machine tools, woodworking machines, machines for the engineering industry, cranes for bulk materials, ventilator fans, conveyor belts, printing equipment, separators and centrifuges	20 000 ... 30 000
Machines for continuous 24 hour use: rolling mill gear units, medium-size electrical machinery, compressors, mine hoists, pumps, textile machinery	40 000 ... 50 000
Wind energy machinery, this includes main shaft, yaw, pitching gearbox, generator bearings	30 000 ... 100 000
Water works machinery, rotary furnaces, cable stranding machines, propulsion machinery for ocean-going vessels	60 000 ... 100 000
Large electric machines, power generation plant, mine pumps, mine ventilator fans, tunnel shaft bearings for ocean-going vessels	> 100 000

Table 6. Guideline values of life for axle box bearings and units for railway vehicles [5].

Guideline values of specification life for axlebox bearings and units for railway vehicles

Type of vehicle	Specification life Millions of km
Freight wagons to UIC specification based on continuously acting maximum axle load	0,8
Mass transit vehicles: suburban trains, underground carriages, light rail and tramway vehicles	1,5
Main line passenger coaches	3
Main line diesel and electric multiple units	3 ... 4
Main line diesel and electric locomotives	3 ... 5

Part III
METHEDOLOGY

4

METHODOLOGY

This section will determine the nominal life L_{10} by applying the Methodology I described above. This value will be corrected using a correction factor adjusted to the working conditions and operation of the bearings to finally determine the life of the bearing (L).

4.1. Estimation of the bearing's nominal life (L_{10}) using the methodology I [1, 2].

As mentioned earlier, axles take two types of loads: (i) the static load, W, characterized by the weight, and (ii) the dynamic loads, F_a and F_r, generated by the vibration of the wagon.

4.1.1. CALCULATION OF THE STATIC LOAD – W.

To calculate this load, W, were considered two conditions: (i) with the wagon empty, W_v, and (ii) with the wagon full, W_c. Then, using Eqs. (1) and (2), and taking the data of M_v and M_c in appendix A.1., we obtain:

$$W_v = \frac{21{,}5\,\text{Ton} * 9{,}81\,\text{m/s}^2}{4} \quad \Rightarrow \quad W_v = 52{,}729\,\text{KN}$$

$$W_c = \frac{126{,}5\,\text{Ton} * 9{,}81\,\text{m/s}^2}{4} \quad \Rightarrow \quad W_c = 310{,}241\,\text{KN}$$

4.1.2. CALCULATION OF THE DYNAMIC LOADS – F_a AND F_r.

Continuing as suggesting in section 2.1.3., were calculated the loads F_a and F_r, i.e., the horizontal and vertical components of the dynamic forces, respectively. As before, were considered two conditions: (i) with the wagon empty, $F_{av} - F_{rv}$, and (ii) with the wagon full, F_{ac} - F_{rc}. These are obtained with the forces and moments in the wagon reach equilibrium (see figure 9 for a free body diagram of the system).

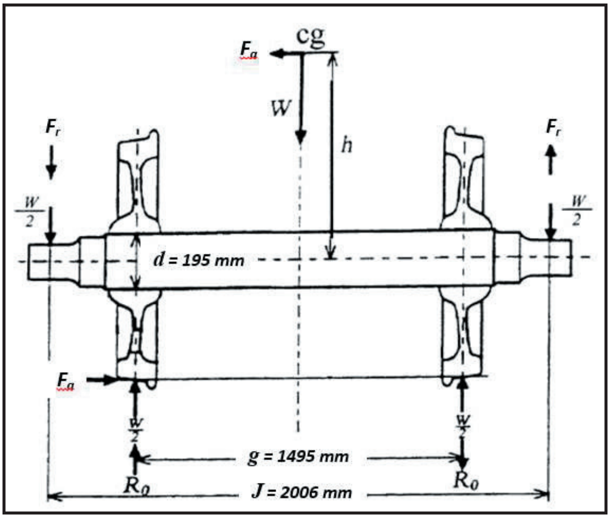


Figure 9. Free body diagram of the shaft-wheel system**.**

In addition, is used a coefficient of horizontal acceleration $(\alpha_L = 0{,}060)$ as shown in table 1, for Class B, System 2, in the conventional track line for velocities between 60 - 130 km/h.

Now, before calculating loads F_a and F_r, it is necessary to determine the wagon´s centre of gravity to find the distance *h*.

To estimate the vertical distance from the axle to the centre of gravity of the wagon (h), it is necessary to divide the train's wagon into two parts, i.e., the basket and the frame or support which includes the wheels and axles. At the same time, the basket is divided into three geometric sections (A, B and C) to facilitate calculation of its centre of gravity, as shown in figure 10.

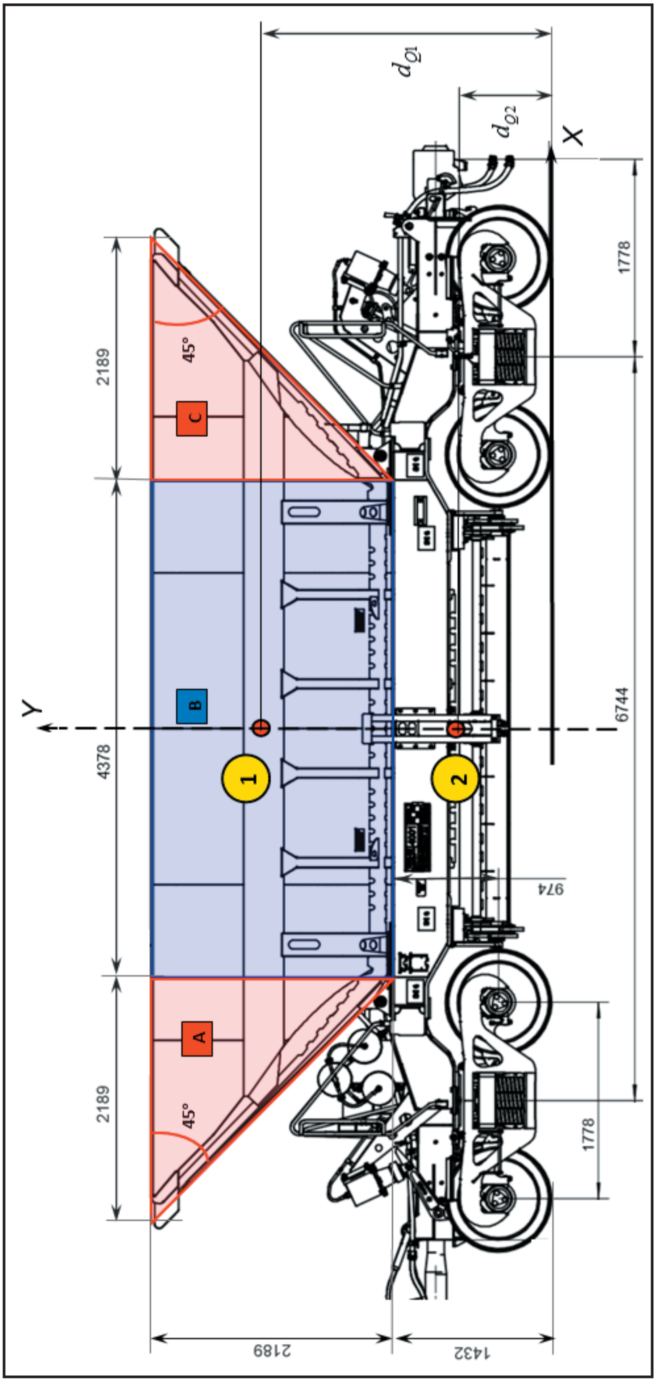


Figure 10. Wagon divided into two sections to facilitate calculation of the centre of gravity.

As indicated in figure 10, putting the reference axis in the centre of the wagon to determine the distances d_{Q1} and d_{Q2}, which represent the distance from the horizontal axis to the centre of gravity of each part of the wagon (part 1 and part 2); i.e., the points (marked in red) are the loads, Q_1 and Q_2, representing the weight of the basket (part 1) and the weight of the frame or support which includes the wheels and axles (part 2), respectively; see figure 11.

Then, using figure 10, is determined the centre of gravity of the basket (part 1).

To facilitate the calculation, is using the following table.

Table 7. Calculate the centre of gravity.

Geometrical figure	**Area (A) (m^2)**	$\overline{X}$ **(m)**	$\overline{Y}$ **(m)**	$\overline{X}*A$ **(mm^3)**	$\overline{Y}*A$ **(mm^3)**
Triangle A	2395,861	- 2,9187	2,891	- 6992,798	6927,152
Rectangle B	10371,482	0	2,527	0	26203,549
Triangle C	2395,861	2,9187	2,891	6992,798	6927,152
$\sum$	**15163,204**			**0**	**40057,853**

Now, knowing that:

$$\overline{X} = \frac{\sum\left(\overline{X}*A\right)}{\sum A} \quad \Rightarrow \quad \overline{X} = \frac{0}{15163{,}204\,m^2} \quad \Rightarrow \quad \overline{X} = 0$$

$$\overline{Y} = \frac{\sum\left(\overline{Y}*A\right)}{\sum A} \quad \Rightarrow \quad \overline{Y} = \frac{40057{,}853\,m^3}{15163{,}204\,m^2}$$

$$\Rightarrow \overline{Y} = 2{,}6418\,m \approx 2641{,}8\,mm$$

In this case, $\overline{Y} = d_{Q1} = 2{,}6418\text{m} \approx 2641{,}8\text{mm}$.

Next, is estimated the centre of gravity of part 2 considering the symmetry of the wagon, as shown in figure 11. The figure shows the distance d_{Q2}, representing the distance from the horizontal axis to its centre of gravity. This value is:

$$d_{Q2} = 0{,}716\,\text{m} \approx 716\,\text{mm}$$

Then, assuming all loads are centred on the median longitudinal plane of the vehicle or vehicle composition (load symmetry) the coordinate "Z" disappears (will be taken as nil) and:

$$Q_{1v} = 3{,}5\ \text{Ton}$$

$$Q_{2v} = 18\,\text{Ton}$$

$$Q_{1c} = 108{,}5\,\text{Ton}$$

$$Q_{2c} = 18\,\text{Ton}$$

Note: The distance to the centre of gravity, h, must be considered in cases when the wagon is empty and full; thus, the load Q_1 is expressed for both conditions, as shown above (Q_{1v} and Q_{1c}). Note also that the load Q_2 is the same for both cases.

Now, is calculating the wagon’s centre of gravity with the help of figure 11:

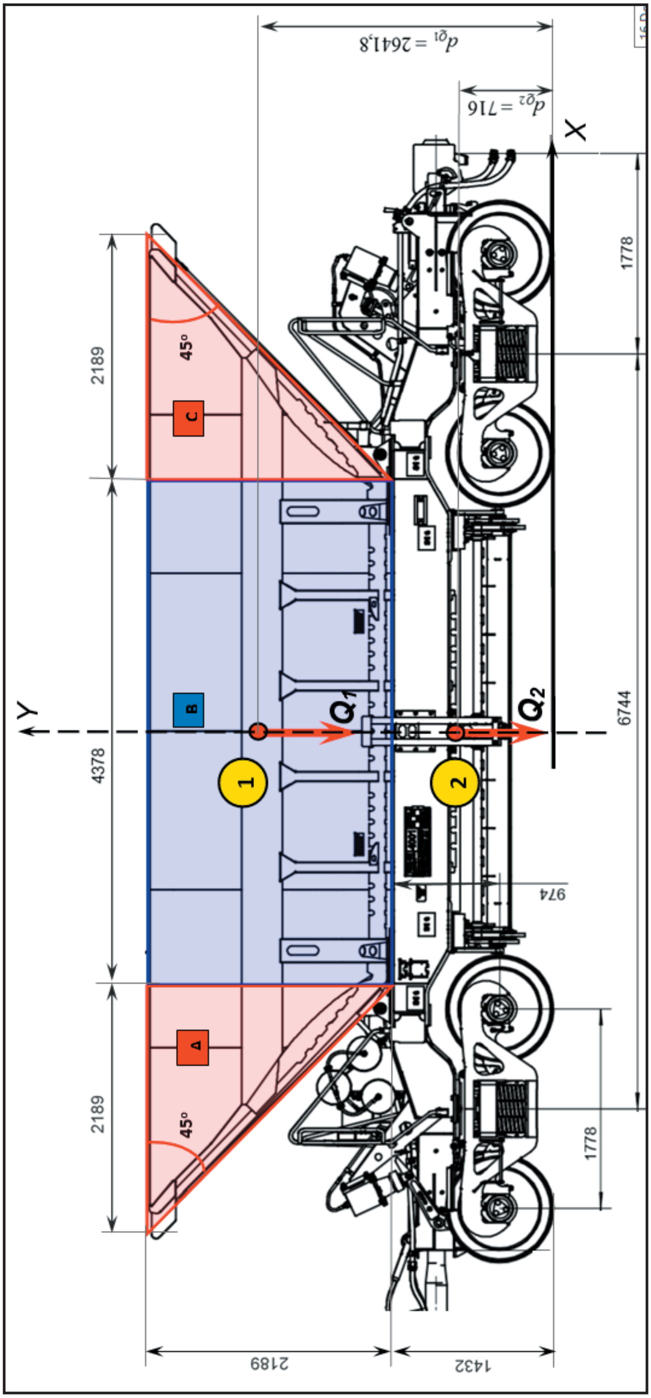


Figure 11. Diagram of free body of the wagon showing loads Q_1 and Q_2 in the centre of gravity of both parts of the wagon.

Using Eqs. (7) - (9), is obtained the following:

Table 8. Wagon Empty.

	Load (Q) (Ton)	x (m)	y (m)	x * Q (Ton.m)	y * Q (Ton.m)
Q_{1v}	3,5	0	2,642	0	9,246
Q_{2v}	18	0	0,716	0	12,888
$\sum$	**21,5**			**0**	**22,134**

$$x_{cdgv} = \frac{0}{21{,}5\,\text{Ton}} \Rightarrow x_{cdgv} = 0$$

$$y_{cdgv} = \frac{22{,}134\,\text{Ton} * \text{m}}{21{,}5\,\text{Ton}} \Rightarrow y_{cdgv} = 1{,}03\,\text{m} \approx 1030\,\text{mm}$$

$$z_{cdgv} = 0$$

Table 9. Wagon Full.

	Load (Q) (Ton)	x (m)	y (m)	x * Q (Ton.m)	y * Q (Ton.m)
Q_{1c}	108,5	0	2,642	0	286,635
Q_{2c}	18	0	0,716	0	12,888
$\sum$	**126,5**			**0**	**299,523**

$$x_{cdgc} = \frac{0}{126{,}5\,\text{Ton}} \Rightarrow x_{cdgc} = 0$$

$$y_{cdgc} = \frac{299{,}5233\,\text{Ton} * \text{m}}{126{,}5\,\text{Ton}} \Rightarrow y_{cdgc} = 2{,}37\,\text{m} \approx 2370\,\text{mm}$$

$$z_{cdgc} = 0$$

Then, distance *h* to the empty and full wagon, respectively, is:

$$h_v = y_{cdgv} - \frac{D_w}{2} = 1030\,\text{mm} - \frac{920}{2}\,\text{mm}$$

$$\Rightarrow h_v = 1030\,\text{mm} - 460\,\text{mm} \quad \Rightarrow h_v = 570\,\text{mm} \approx 0{,}57\,\text{m}$$

$$h_c = y_{cdgc} - \frac{D_w}{2} = 2370\,\text{mm} - \frac{920}{2}\,\text{mm}$$

$$\Rightarrow h_c = 2370\,\text{mm} - 460\,\text{mm} \quad \Rightarrow h_c = 1910\,\text{mm} \approx 1{,}91\,\text{m}$$

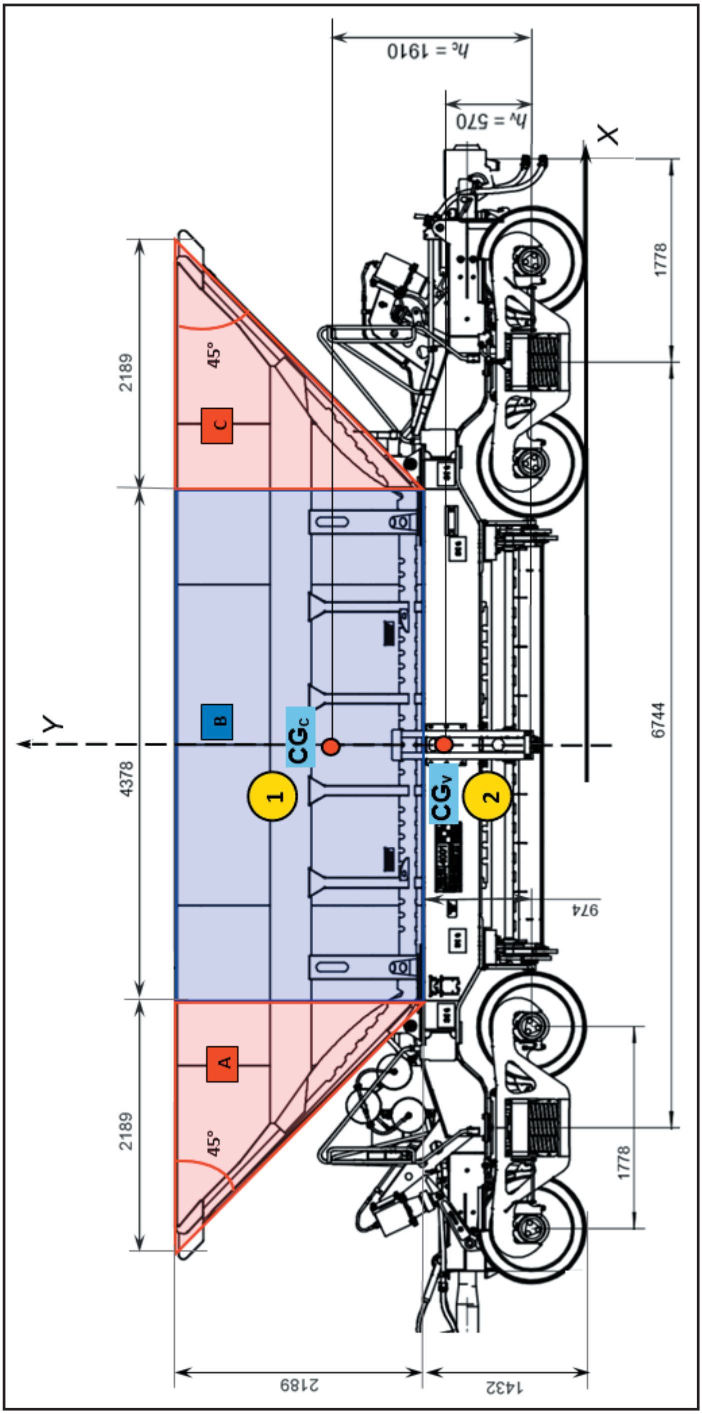


Figure 12. Diagram of the vertical distance *h* from the axle to the centre of gravity of the wagon.

Using Eqs. (3) – (6), is obtained:

$$F_{av} = 0{,}060 * \frac{21{,}5\,\text{Ton} * 9{,}81\,\text{m/s}^2}{4 * 4} \quad \Rightarrow \quad F_{av} = 0{,}791\,\text{KN}$$

$$F_{ac} = 0{,}060 * \frac{126{,}5\,\text{Ton} * 9{,}81\,\text{m/s}^2}{4 * 4} \quad \Rightarrow \quad F_{ac} = 4{,}654\,\text{KN}$$

$$F_{rv} = 0{,}060 * \frac{21{,}5\ \text{Ton} * 9{,}81\ \text{m/s}^2 * 0{,}57\,\text{m}}{4 * 4 * 2{,}006\,\text{m}}$$

$$\Rightarrow F_{rv} = 0{,}225\,\text{KN}$$

$$F_{rc} = 0{,}060 * \frac{126{,}5\,\text{Ton} * 9{,}81\,\text{m/s}^2 * 1{,}91\,\text{m}}{4 * 4 * 2{,}006\,\text{m}}$$

$$\Rightarrow F_{rc} = 4{,}431\,\text{KN}$$

4.1.3. CALCULATION OF THE MEAN EQUIVALENT DYNAMIC LOADS – F_m.

Before determining the mean dynamic equivalent load F_m, it is necessary to calculate the equivalent dynamic load P. Once the horizontal and vertical components of load on the bearing are determined, we evaluate the equivalent dynamic load P, using Eqs. (10a) and (10b) to obtain:

$$P_v = 1 * 0{,}225\,\text{KN} + 2{,}55 * 0{,}791\,\text{KN}$$

$$\Rightarrow P_v = 0{,}225\ \text{KN} + 2{,}017\,\text{KN} \qquad \Rightarrow P_v = 2{,}242\,\text{KN}$$

$$P_c = 1 * 4{,}431\,\text{KN} + 2{,}55 * 4{,}654\,\text{KN}$$

$$\Rightarrow P_c = 4{,}431\,KN + 11{,}868\,KN \qquad \Rightarrow P_c = 16{,}299\,KN$$

Then, is determined f_{rc} and f_{rv} [Hz], the frequencies of W_c and W_v, respectively. However, as mentioned, it is necessary to consider that f_{rc} and f_{rv} are proportional to the rotation of the axle. Therefore, f_r can be calculated by Eq. (13):

$$f_{rv} = \frac{2 * 70\,Km/h}{920\,mm} \Rightarrow f_{rv} = \frac{2 * 19{,}444\,m/s}{0{,}92\,m} \Rightarrow$$
$$f_{rv} = 42{,}27\,Hz$$

$$f_{rc} = \frac{2 * 60\,Km/h}{920\,mm} \Rightarrow f_{rc} = \frac{2 * 16{,}667\,m/s}{0{,}92\,m} \Rightarrow$$
$$f_{rc} = 36{,}232\,Hz$$

Finally, considering the specific conditions of this analysis as shown in section 2.1.5, we determine the mean equivalent dynamic load, F_m:

$$F_m = \sqrt[3]{0{,}5 * \left\{ \frac{1}{(36{,}232 + 0{,}5)Hz} * \left[0{,}5\,Hz * (16{,}299\,KN)^3 + 36{,}232\,Hz * \left(\frac{310{,}241\,KN}{4} \right)^3 \right] + \frac{1}{(42{,}27 + 0{,}5)Hz} * \left[0{,}5\,Hz * (2{,}242\,KN)^3 + 42{,}27\,Hz * \left(\frac{52{,}729\,KN}{4} \right)^3 \right] \right\}}$$

$$\Rightarrow F_m = \sqrt[3]{0{,}5 * \left\{ \frac{1}{36{,}732\,Hz} * \left[0{,}5\,Hz * (16{,}299\,KN)^3 + 1{,}691 * 10^7\,Hz * KN^3 \right] + \frac{1}{42{,}77\,Hz} * \left[0{,}5\,Hz * (2{,}242\,KN)^3 + 9{,}683 * 10^4\,Hz * KN^3 \right] \right\}}$$

$$\Rightarrow F_m = \sqrt[3]{0{,}5 * \left\{ \frac{1}{36{,}732\,Hz} * \left[2164{,}975\,Hz * KN^3 + 1{,}691 * 10^7\,Hz * KN^3 \right] + \frac{1}{42{,}77\,Hz} * \left[11{,}270\,Hz * KN^3 + 9{,}683 * 10^4\,Hz * KN^3 \right] \right\}}$$

$$\Rightarrow F_m = \sqrt[3]{0{,}5 * \left\{ \frac{1}{36{,}732\,Hz} * \left[16912164{,}975\,Hz * KN^3 \right] + \frac{1}{42{,}77\,Hz} * \left[96841{,}270\,Hz * KN^3 \right] \right\}}$$

$$\Rightarrow F_m = \sqrt[3]{0{,}5 * \left(460420{,}477\;\; KN^3 + 2264{,}234\,KN^3 \right)}$$

$$\Rightarrow F_m = \sqrt[3]{0{,}5 * \left(462684{,}711\,KN^3\right)} \Rightarrow F_m = \sqrt[3]{231342{,}356\,KN^3}$$

$$\Rightarrow F_m = 61{,}388\,KN$$

Using Eq. (14), L_{10} is calculated by:

$$L_{10} = \frac{\pi * 0{,}92}{1000} * \left(\frac{271\,KN}{61{,}388\,KN}\right)^{\frac{10}{3}} * 1{,}5E10^{+6}$$

$$L_{10} = 611854{,}719\ Km \approx 612.000\ Km$$

4.2. Calculation of correction factors of life.

As noted, the nominal life of a bearing is calculated for a 90% chance of survival (see section 4.1.), but there are applications where this reliability is insufficient, and greater reliability is required. Thus, a correction coefficient for reliability (a_1), and a correction coefficient for working conditions (a_{ISO}) are included.

4.2.1. ESTIMATION OF CORRECTION FACTOR OF LIFE BY RELIABILITY (a_1):

As is want to obtain a reliability that can adjust to the requirements favourable to the replacement of the bearings, were considered all the values of reliability shown in table 5 to calculate the bearing life (L); it is choose the reliability that best suits the needs of maintenance. In other words, is want to obtain a chart of life (L) as a function of reliability to determine the bearing's behaviour and its maximum lifetime.

Therefore, the calculations of each life (L) will estimate a correction factor of life (a_1) that corresponds to reliability.

4.2.2. ESTIMATION OF CORRECTION FACTOR OF LIFE BY WORKING CONDITIONS (a_{ISO}):

To establish the factor of working conditions (a_{ISO}) we must consider two (2) conditions, i.e., the operating temperature in winter (-40 ºC approx.) and in summer (+30 ºC approx.).

To obtain the correction factor a_{ISO} for winter conditions, we do the following:

1. Knowing the average bearing diameter (d_m) (Appendix A.2 [7]) and the rotation speed (*n*) with the help of the figure 4, is could obtain the relative viscosity (ν_1) (see figure 4):

$$d_m = \frac{D+d}{2} = \frac{(250-157)\text{mm}}{2} \qquad \Rightarrow d_m = 203{,}5\,\text{mm}$$

$$n = \frac{\text{Velocity of the railway}}{\text{Perimeter of the wheel}} = \frac{60\,\text{Km/h}}{\pi * D_w}$$

$$\Rightarrow n = \frac{\frac{60\,\text{Km}}{\text{h}} * \frac{1000\,\text{m}}{\text{Km}} * \frac{1\,\text{h}}{60\,\text{min}}}{\pi * 0{,}92\,\text{m}} = \frac{1000\ \text{m/min}}{\pi * 0{,}92\,\text{m}}$$

$$\Rightarrow n = 346\,\text{rpm}$$

Then,

$$\nu_1 = 25\,\text{mm}^2/\text{s}$$

2. As the viscosity (ν) of the lubricating oil at the operating temperature of the bearing is known, the base oil must meet the following requirements [15, 18]:

- A high viscosity index (VI).
- A low pour point.
- As low a base oil viscosity as possible at the required operating temperature.

Then, the lubrication used on the bearings has a viscosity (ν) of 80 cSt at 40 ^{o}C [16, 18]. When we enter the viscosity of the oil with respect to the temperature (see figure 5) into the graph, we obtain:

$$\nu_{-40} = 100.000\,\mathrm{mm}^2/\mathrm{s}$$

Note: At low temperatures, the viscosity greatly increases; as it is normally plotted on a logarithmic scale, the variations are large. Is should be remembered that the typical equations of experimental behavioural are always exponentials; that is, kinematic is viscosity based on units of cSt (mm^2/s).

3. Then, is determined the ratio of these two viscosities; the ratio is identified by the letter kappa *k*, as in the following:

$$\kappa = \frac{100.000\,\mathrm{mm}^2/\mathrm{s}}{25\,\mathrm{mm}^2/\mathrm{s}} \Rightarrow k = 4000$$

4. Now, the level of contamination of the lubricant provides the factor η_c. Knowing the average bearing diameter (d_m) and with the help of table 4, is obtained this factor. Assuming a level of contamination "Normal Cleanliness" to an average bearing diameter $\mathrm{d_m} \geq 100$, we get a value of:

$$\eta_c = 0{,}7$$

5. Finally, is determined the contamination-load relationship $\left(\eta_c * P_u\right)/F$, where

- Fatigue limit load (KN):

In this case the manufacturer does not provide the value of limit fatigue loads for the bearing; therefore the value will be approximated as in the following theoretical relations [21]:

$$P_u \approx \frac{1}{27} * C_o \ \ldots\text{for ball bearings,} \tag{16}$$

$$P_u \approx \frac{1}{35{,}5} * C_o \ \ldots\text{for self-aligning ball bearings,} \tag{17}$$

$$P_u \approx \frac{1}{8{,}2} * C_o \ \ldots\text{for other bearings,} \tag{18}$$

Where, $C_o = 69.000\,\mathrm{N}$ is the static load capacity, taken from the catalogue of the manufacturer. Then, knowing the type of the bearing to used, is taken Eq. (18), for the Tapered Roller Bearing:

$$P_u = \frac{1}{8{,}2} * 69.000\,\mathrm{K}N \;\; P_u = 8414{,}634\,\mathrm{N} \approx 8{,}415\,\mathrm{KN}$$

- Force Transmitted (KN):

$$F = P_c = 16{,}299\,\mathrm{KN}$$

Therefore,

$$\frac{(\eta_c * P_u)}{F} = \frac{0{,}7 * 8{,}415\,KN}{16{,}299\,KN} = 0{,}361$$

Now, with help of figure 8 is can derive the correction factor working conditions (a_{ISO}):

$$a_{ISO} = 3$$

Note: For a value kappa $\kappa > 4$ (as in this case) SKF susgest using the curve for $\kappa = 4$.

Now, once obtained the parameters for winter conditions, is follow the same procedure for summer:

$\nu_1 = 25\,mm^2/s$ $\qquad\qquad$ $\nu_{30} = 27\,mm^2/s$

$k = 1{,}08$ $\qquad\qquad$ $\eta_c = 0{,}5$ $\qquad\qquad$ $\frac{(\eta_c * P_u)}{F} = 0{,}258$

$$a_{ISO} = 0{,}8$$

4.3. Calculation of the bearing life (L).

Finally, when is obtained the correction factors of life by reliability (a_1) and per working conditions (a_{ISO}), is calculated the bearing life (L) for different reliabilities using Eq. (15). The results appear in section 8.

Table 10. Data obtained.

a_{ISO} - Winter	a_{ISO} - Summer	L_{10}
3	0.8	612,000.00

4.4. Estimate of the nominal life (L_{10}) of the axle box bearings.

Before estimating the nominal life (L_{10}) of the axle box bearings, it is necessary to briefly explain their function and design.

European railways use freight cars with two and four axles. For heavier loads, a four axle version is more suitable. In most cases, these cars are equipped with two Y25 type bogies. In principle, two different bogie frame designs can be used, based either on welding or on casting production techniques. The new Y25 bogie design is equipped with axle boxes guided by special wear plates and supported by steel coil springs on both sides [46].

In the past, there were several different axle box designs to accommodate spherical, cylindrical and tapered roller bearings. These are open bearings and must be protected against contamination and grease losses. A full bore housing needs to be applied with a sealing system located close to the wheel. On the opposite side, a front cover is fitted with several screws onto the axle box body. This design is relatively complex, and performance relies on proper mounting. Bearings must be greased manually and installed in a very clean environment to avoid contamination.

SKF has developed a completely new axle box design by incorporating the SKF Compact TBU. The axle box has the same mechanical interface between the body frame and the coil springs as earlier designs, making it possible to upgrade existing bogies with the new SKF axle box design (see figure 13).

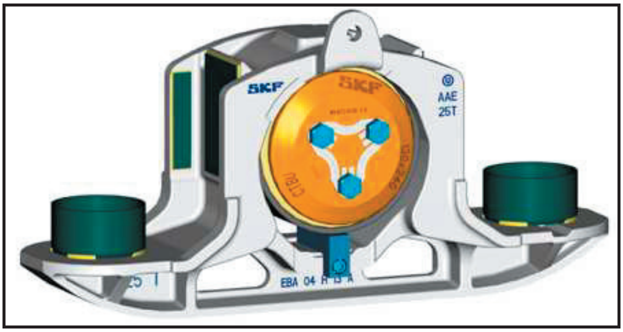


Figure 13. The new SKF Y25 axle box [46].

The key to a lower life cycle cost is design. The compact design of the TBU offers new opportunities for longer maintenance intervals and improved performance and safety.

Compared with previous designs for Y25 bogies, the main user benefits of the compact TBU bearing units are:

- ✓ Shorter axle length for new bogie designs, reducing axle bending under the bearing and unsprung weight.
- ✓ Improved safety and performance through the use of polyamide cages instead of the traditional steel cages, resulting in less wear and reduced grease contamination.
- ✓ Avoidance of fretting corrosion in the inner ring/backing ring contact zone, achieved by the use of a polymer spacer.
- ✓ Improved protection against contamination due to the incorporation of a newly designed low-friction contact seal that rides on the inner ring shoulder.
- ✓ Longer lubricant life, using a new, long-life grease.
- ✓ Simplified and easier mounting.

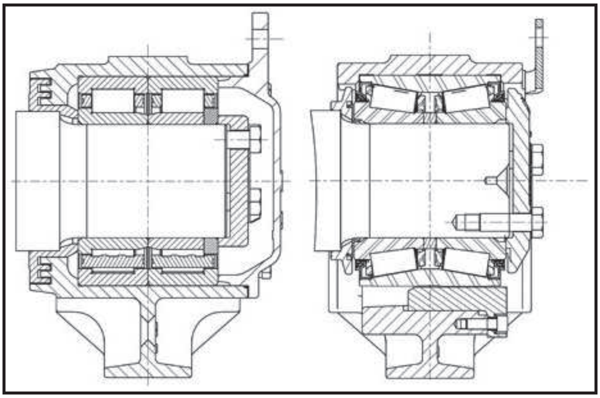


Figure 14. Existing axle box design based on open (unsealed) cylindrical roller bearings (left) and the new design equipped with compact TBU (right) [46].

The greasing process for previously used unsealed axle box bearings is done at the workshop of wheelset producers or overhaul facilities. Cleanliness, the proper grease quantity, and correct distribution are absolute requirements. In case of a sealed and greased bearing unit such as the CTBU, this process is incorporated into the bearing production where these requirements can be met with a very high reliability [46].

The main user benefits of the new SKF Y25 axle box design are [46]:

- ✓ Weight saving, around 20 kg per axle box or 160 kg for a four-axle wagon, in comparison with previous designs. This reduces the unsprung weight of the wheel set by around 4%, which contributes to the running performance and reduces wheel and rail wear.
- ✓ Simplification of the axle box assembly, fewer parts of the two main components – bearing unit and axle box housing.
- ✓ Simpler and easier mounting, disassembly and maintenance.
- ✓ Lower life-cycle cost due to longer maintenance intervals.
- ✓ Increased safety and performance.

Figure 15. Y25 axle box assembly [46].

With the same calculation methodology used previously (section 4.1) we proceed to determine the nominal life L_{10} of the bearings in the axle box, but considering 8 bearings per shaft, i.e., $N_r = 8$. The results are shown in table 11.

Table 11. Results obtained for the bearings of the axle box.

Static load (W)	$W_v = 52{,}729\,KN$ $W_c = 310{,}241\,KN$
Dynamic loads (F_a - F_r)	$F_{av} = 0{,}395\,KN$ $F_{ac} = 2{,}327\,KN$ $F_{rv} = 0{,}112\,KN$ $F_{rc} = 2{,}215\,KN$
Equivalent dynamic load (P)	$P_v = 1{,}119\,KN$ $P_c = 8{,}149\,KN$
Frequencies (f_{rv} –f_{rc})	$f_{rv} = 42{,}27\,Hz$ $f_{rc} = 36{,}232\,Hz$
Mean equivalent dynamic load (F_m)	$F_m = 61{,}386\,KN$
Nominal life (L_{10})	$L_{10} = 611921{,}170\ Km \approx 612.000\ Km$

5 RAILWAY BEARING CONDITION MONITORING

Condition monitoring is a mature technology that offers new capabilities in the railway industry, as it enables a proactive approach to achieve financial savings and meet safety targets [22]. The constant movement of trains over tens of thousands of miles has a relentlessly detrimental effect on many car components. One of the components most susceptible to catastrophic failure due to this constant rolling friction are the wheel bearings. Over time, these carefully machined roller bearings may crack, spall, and even seize with resulting negative effects on fuel economy and, most importantly, on train safety [23].

A number of techniques have been used to perform fault detection in railway vehicles, including advanced filtering, system identification and signal analysis methods. These techniques are used to detect and identify faults that deteriorate with time [24].

The practical application of condition monitoring to train dynamics is done by using track-based sensors or vehicle-based sensors. The track bed-based sensors are commonly used to monitor the condition of wheelset, while the rolling stock-based sensors monitor the rolling stock infrastructure. Modern rolling stock is fitted with high-capacity communication buses and multiple sensors which require advance processing units for data collection and management [25].

During operation, it is important to regularly inspect the condition of the bearing by performing basic condition monitoring measurements. These regular inspections will allow the detection of potential problems and help to prevent unexpected machine stoppages. Consequently, the machine maintenance can be planned to suit the production schedule, thus increasing the plant's productivity and efficiency [19].

This section explains some of the existing condition monitoring techniques applied to monitor railway vehicle dynamics and to detect and predict flaws and anomalies in railway bearings. It also discusses the future use of Smart Bearings in the railway industry.

A number of tools and techniques of condition monitoring can be used to check a series of properties. These are explained in the following section.

5.1. Vibration analysis.

One of the many forms of condition monitoring utilises the principles of vibration analysis. This system has been successfully utilised for many years to monitor equipment as a whole by measuring the vibration of the overall machine or by analysing the individual components. In many cases, measuring the vibration of individual components is the most effective method of monitoring the condition of rolling element bearings.

Vibration can be defined as the motion of a machine and its components from a resting position. External forces act to produce motion or movement inherent to a particular machine, its components and use. It can be measured by specific equipment, namely, an accelerometer, which converts the vibrating motion to an electrical signal in preparation for analysis. This motion takes a sinusoidal wave form, having characteristics of frequency, amplitude, wavelength and phase. When machine vibration is measured, several sinusoidal waveforms or motions are usually found; they combine to give an overall time wave form. To improve the use of this wave form, a Fourier analysis is performed using specialised equipment to convert the time wave form to an amplitude versus frequency spectrum. This equipment is known as a Fourier transform analyser or a Fast Fourier Transform (FFT) [26].

5.1.1. Vibration analysis applications.

One of the oldest, most reliable and widely used forms of bearing condition monitoring is vibration analysis. It is used in many industries and in various forms, such as fixed or portable units.

In general, to effectively utilise vibration analysis for bearing condition monitoring, the bearing of interest should be in service, under an applied load and rotating at operating speed, or conversely, a known speed, sufficient to enable a constant, accurate and timely reading to be taken. Additional bearing information, such as the type and manufacturer, are also required to successfully and accurately analyse the bearing's condition, as the variation between rolling elements within bearings differs between manufacturers.

Taking the readings requires a series of accelerometers and the components to record and send information to be processed and logged. Such equipment is supplied by the bearing manufacturing company SKF and can be readily obtained and adapted to the application. Accelerometers located on the loaded side of the bearing housings acquire the vibration readings of each bearing simultaneously, as the wheels are rotated at a pre-determined speed.

Consultation with an SKF representative reveals that the equipment best suited to an automated system is a combination of the following: first, a multi-parameter sensor, which measures both vibration and temperature; these are individually identified by the SKF product code CMPT 2310T (SKF website, 2008 [27]); second, transmitters, with product code CMPT CTU (SKF website, 2008 [27]), which process the signals and send them to the alarm display modules, with product code CMPT DCL (SKF website, 2008). These alarm display modules can be programmed to sound the alarm at specific points of concern. The transmitter can also be utilised to send information to a personal computer, where it can be analysed and logged using specific software. At a minimum, this system requires eight sensors, eights transmitters and sixteen display modules to monitor one eleven tonne cane bin. Figure 16 illustrates the design concept but only shows one cane bin. The testing apparatus can test two bins and a maximum of eight bearings simultaneously. The testing apparatus must be reconfigured to test the models of bins in service when required. That is, the positions of the wheel rollers and accelerometer adaptors will be altered to conform to specific bin type geometry.

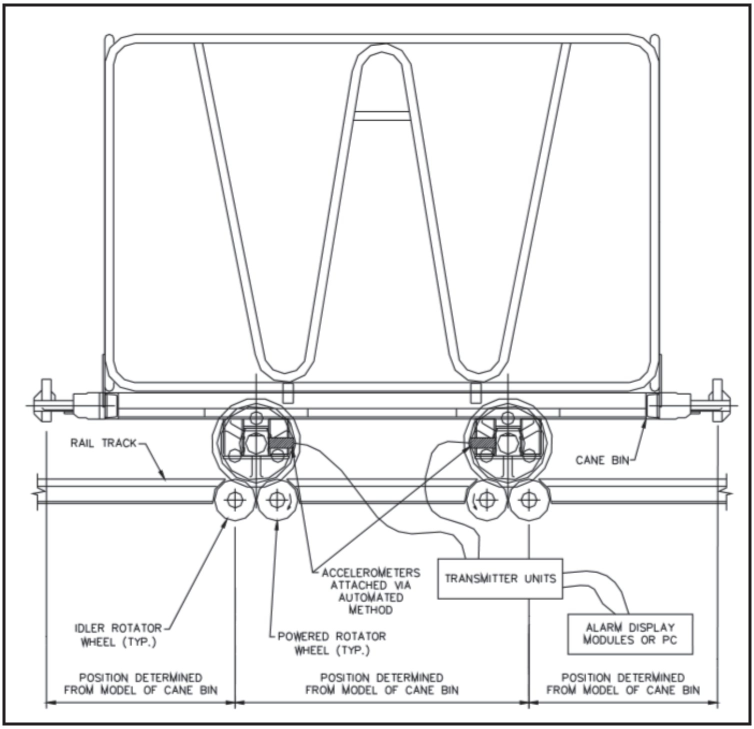


Figure 16. Conceptual design of vibration system [26].

One advantage of this type of system is that it can be adapted to be portable if required, allowing it to be re-located to alternative sites. It also facilitates the trending of bearing conditions and gives substantial prior warning of bearing failures.

The disadvantages of this vibration analysis system include the present lack of a bin identification system, which would permit bearing information to be partnered with the correct bin. The fact that there are presently several types of bearings in service from various manufacturers compounds the problem, as historic data relating to cane bins and their bearings have not been maintained; therefore, the correct identification of bearings per bin must be a known factor. The correct placement of the accelerometer, by means of an automated method, may pose difficulties, as incorrectly placed accelerometers will give erroneous readings. It is vital that a pre-determined test speed be known, as vibration frequencies are heavily reliant on correct speed identification. Other disadvantages include the speed at which it operates, as the available time in which the system may perform the monitoring function is under sixty seconds. In this time, the loaded bin must be located on the testing apparatus, accelerometers attached, the wheels rotated and the information gathered. The reversing of this procedure must also be kept within a sixty second time period, to allow the bin to continue to the unloading station without causing delays to the factory [26].

5.2. Acoustic monitoring.

Similar in many aspects to vibration monitoring, where a physical vibration is measured, acoustic monitoring involves the measurement of the audio signature emitted by components. The audio signature is derived from the impact frequencies associated with bearing defects. The concept behind acoustic monitoring is to capture these impact frequencies as they occur. Utilising specialised equipment, the microphones capture the audio signals and send relevant data to computers, where the signals are processed and differentiated using specifically designed software.

As with vibration analysis, the type of bearing, manufacturer and operating speed of the bearing must be known factors to obtain consistent results [29].

5.2.1. ACOUSTIC BEARING MONITORING APPLICATIONS.

Recently adopted technologies have introduced acoustic analysis to monitor bearings on rail wagons. This type of analysis identifies and utilises the acoustic signature emanating from operational wheel bearings, in an attempt to isolate defective or damaged components. The technology relies heavily on modern software to enable it to segregate and differentiate between the various acoustic signals obtained to accurately identify problems.

Vipac Engineers and Scientists Limited have developed the RailBAM© system for use in monitoring rail wagon bearings, using acoustic monitoring technology. Their system is presently being used nationally and internationally. The system has the ability to identify the various sounds emanating from passing train wheels sets, using specialised equipment and computer software.

The basic configuration is to have specially constructed microphone stations on both sides of the rail tracks; these are connected to sensors activated by approaching trains which, in turn, activate the microphones by opening the protective shutters covering them. The system also incorporates a radio frequency identification tag (RFID) reader, used to identify the passing wagons and wheel detectors, aiding the processing and location of faults. The microphone stations are illustrated in figure 17 [26].

Figure 17. Microphone station. (Vipac Engineers) [26].

The RailBAM© system has the ability to monitor wheel bearings at train speeds of between 30 km/h and 130 km/h in either direction. The system consistently differentiates between the various sounds, identifying bearing and wheel defects, wheel profile and tracking problems associated with wheel sets. Figure 18 illustrates an operational RailBAM© system.

The information collected is sent to an enclosed structure adjacent to the rail tracks, which houses the computer system. At this point, the information is analysed via proprietary techniques. Proprietary techniques differentiate between the various acoustic signatures from the numerous operating components and categorise the severity of bearing condition.

The data base can be utilised to trend wagon information spanning several years, through the compilation of historical information, which is useful for future reference. As the system is capable of identifying bearing defects at their earliest stages, it reduces the potential of bearings failing in service by offering an improved maintenance program [26].

Figure 18. An operational RailBAM© system. (Vipac Engineers) [26].

One of the primary advantages of this system is that it operates as a stand-alone unit; thus, it can be installed in an optimal position throughout the railway network in each sugar milling district [26].

5.2.2. ACOUSTIC BEARING FLAW DETECTION.

IEM also developed this patent-pending system to use acoustic analysis to detect and identify numerous faults in passing railcars, ranging from “slid flats” to bearing spalls, locked brakes, and truck faults.

Key Features and Benefits:

- Automatic Process: Automatically examines every bearing and other moving components on passing rolling stock, eliminating the need for manual inspections.

- More efficient than traditional Hot Box Detectors: Current “hot boxes” are only able to detect bearing flaws a short time before complete failure. Acoustic system provides sufficient time to plan for repairs.

- Ease of Set-Up and Use: Requires much less hardware and space than alternative acoustic systems.

- Instant Notification: Analysis is done in real-time and alerts are sent directly to the train and/or the maintenance shops. These alerts provide accurate information as to the nature, severity and location of the fault.

Automatic and accurate detection of bearing flaws well in advance of failure permits bearing maintenance to take place at normal scheduled maintenance shops, eliminating “hot box” stops and trackside replacements which are extremely time-consuming and expensive.

The Acoustic Fault Detection System can be installed outside train fouling zone, allowing fault detection while assuring the system is at no risk from dragging equipment, straps, and other common dangers that have destroyed other types of acoustic detectors [28].

5.3. Thermography.

The relatively new technology of thermography is being used in many industries and disciplines. The primary application of thermography is with electrical equipment; electrical joints and major components can be monitored effectively through the measurement and actual placement of temperature elevations; It has also been found useful in mechanical applications.

Figures 19, 20 and 21 illustrate thermal images of cane bins and their respective bearings whilst travelling along the rail track.

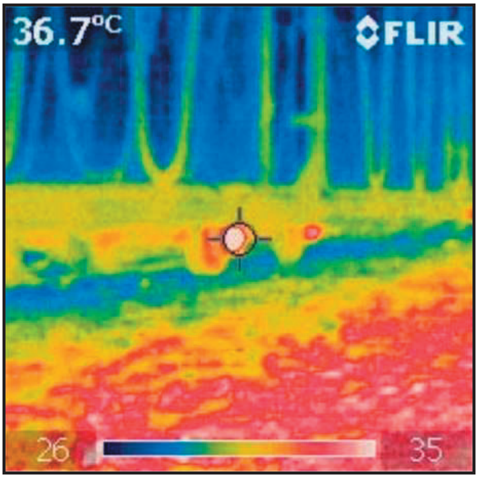


Figure 19. Bearing temperature image [26].

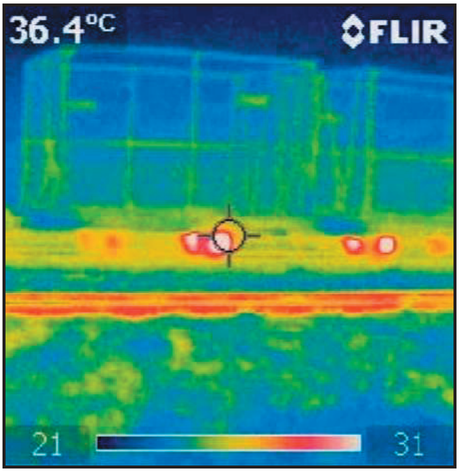


Figure 20. Bearing temperature image [26].

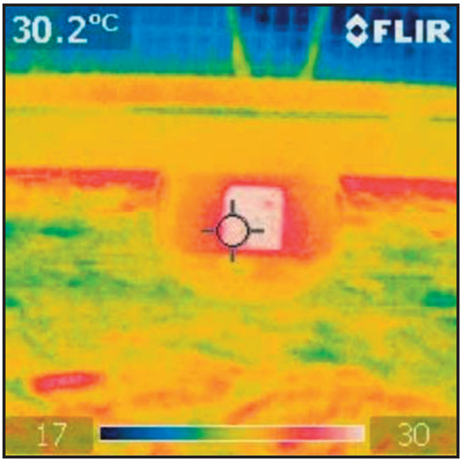


Figure 21. Bearing temperature image [26].

Producing results that are favourably comparable to other systems, the thermography camera can be sourced to other departments for use in alternative applications, maximizing its potential as a non-destructive testing tool. Thermography also delivers a greater level of accuracy in identifying components potentially requiring attention.

The system has several disadvantages as well. As a manually operated system, thermography requires the employment of two trained technicians, the camera operator and a second technician to record bin identification data. This raises the problem of the availability of human resources/staff who are appropriately trained in this system and who can be retained [26].

5.3.1. THERMOGRAPHY APPLICATIONS.

There are several manufacturers of thermography equipment, one of which is FLIR©, who produce many different models of thermography cameras for a wide range of applications and costs.

Whilst thermography has been tested in Europe by mounting a thermography camera to a rail wagon to monitor the temperature of specific rolling stock wheel bearings (FLIR© website, 2008, FLIR© Application Story), no definitive results have been found for its application to the monitoring of sugar cane bins bearings in Australia [30].

As both sides of the cane bins must be monitored to capture all the bearings, an additional thermography camera is required unless an adaptation can be found. This is in the form of the placement of the camera, that is, only one camera may be required if the camera is positioned at a 45 degree angle to the rail tracks and at a maximum distance of ten metres from the furthermost bearing to be monitored, as shown in figure 22. This configuration reduces the costs of the system. However, the ability of the thermography camera to measure the temperature of the furthermost bearing can be affected by the wheel obstruction, as well as the maximum speed at which the bins travel past the camera, therefore inhibiting the collection of accurate bearing temperature readings.

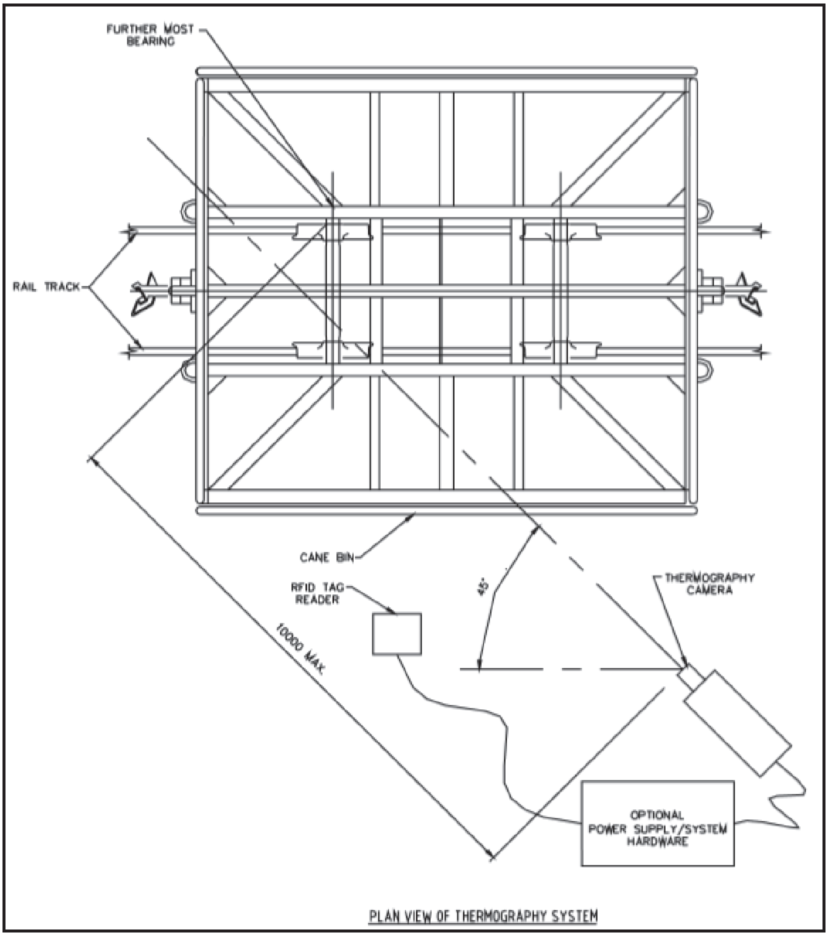


Figure 22. Conceptual design of thermography system [26].

The images shown in figures 19, 20 and 21 were taken by a FLIR© hand held thermography camera. Whilst this instrument is fulfils its task, it does not meet the requirements of being a stand-alone system and necessitates two operators. After consultation with the manufacturer, we determined that a FLIR© A320 thermography camera was best suited to the application. Figure 23 illustrates the FLIR© A320 camera [30].

Figure 23. FLIR© A320 thermography camera [30].

The possible configuration would see the camera mounted in a specially manufactured protective casing and connected to a power source, either 12 or 24 volts, produced via transforming mains power or an on-site power supply (batteries). The images may be processed by the accompanying computers and software installed to support the system and powered similarly to the camera. The data collated may be either manually retrieved from the unit, or additional technological equipment utilised to transmit the information to relevant departments for review. The possible portability of this system is advantageous, as various locations may be utilised to monitor a greater number of fleet units.

Factors that inhibit the accuracy of such a system include the train speeds, as speeds in excess of 10 km/h begin to distort the images produced, creating difficulties in accurate data processing. This occurs as the A320 camera utilises a microbolometer to measure temperature and has a response time of 12 milliseconds. The ends of the bearing housings are approximately 150 mm; this reduces to a target size of 75mm at the 45 degree design angle. At 10 km/h, the bin will move approximately 30mm in 12 milliseconds, producing a distorted image. External coatings of mud and or grease on the bearing housings can also impact the accuracy of this system, due to changes in emissivity [26].

5.4. Lubrication sampling.

Bearing lubrication sampling is another method to monitor bearings. This system undertakes periodic sampling of the grease lubricants used in the bearings. These may show evidence of microscopic wear particles; this occurs as bearing and seal wear escalates and may be indicative of potential bearing failure. The method has proved invaluable in certain applications, such as the monitoring of power transmission units and other hydraulic systems [26].

5.5. Four stages of bearing failure.

Stage 1. Earliest detectable indication of bearing failure using vibration analysis. Signals appear in the ultrasonic frequency bands around 250 KHz to 350 KHz. At this point, there is approximately 10 to 20 percent remaining bearing life.

Stage 2. Bearing failure begins to "ring" at its natural frequency (500 to 2,000 Hz), and a signal appears at the first harmonic bearing frequency. At this point, there is five to 10 percent remaining bearing life.

Stage 3. Bearing failure harmonics of the fundamental frequency are now apparent. Defects in the inner and outer race are apparent and visible upon vibration analysis of the noise signal. Temperature increase is also apparent. There is now one to five percent of remaining bearing life.

Stage 4. Bearing failure is indicated by high vibration. The fundamental harmonics begin to actually decrease, random ultrasonic noise greatly increases, and temperatures increase quickly. Remaining life is one hour to one percent [31].

5.6. Smart bearings railway.

Proper bearing analysis is the key to keeping equipment running efficiently, reliably, consistently and cost effectively. Monitoring for and preventing costly bearing damage can enhance productivity, ensure peak performance and affect the bottom line.

Use of “smart” bearing technology is one method manufacturers can use to monitor bearing operation. Smart bearings are supplied with sensors to provide information about their surrounding environment, including speed, direction, temperature, vibration, load, levels of debris and other factors. The integration of sensors and bearings gives smart bearings their name.

Further, in industrial applications, the data collected by smart bearings are often matched with condition monitoring programs where being aware of temperature and vibration levels is essential to prevent bearing failures. Once smart bearings gather the data, they feed them to a control unit used to monitor the particular bearing operation [32].

5.6.1. SMART BEARING TYPES.

The most popular smart bearings are found in automotive wheel applications.

Most automotive “hub unit” bearings include speed sensors which send wheel speed data to the ABS (anti-lock brake system) and traction control units on light vehicles. Figure 24 shows one such hub unit bearing.

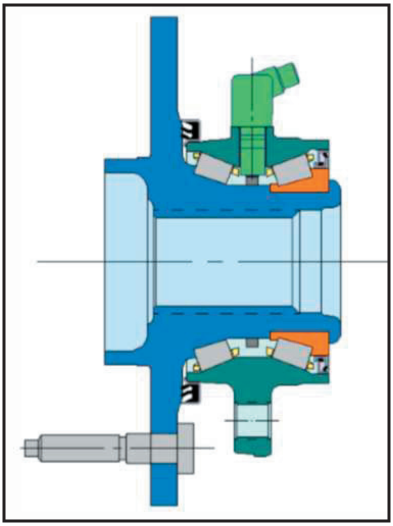


Figure 24. Gen 3 hub unit bearing with speed sensor – courtesy of the Timkem Company [32].

In the industrial market, housed bearing units can be equipped with sensors that monitor bearing speed, vibration, temperature or a combination of all three. Figure 25 shows a smart bearing that is using a speed pickup proximity switch. The speed pickup proximity switch senses the presence of two targets on a special collar or locknut inside the sensorised bearing housing. When a target comes into range, the proximity switch closes, allowing the supplied voltage to pass through. The time between the two pulses per revolution may be measured to determine the shaft speed.

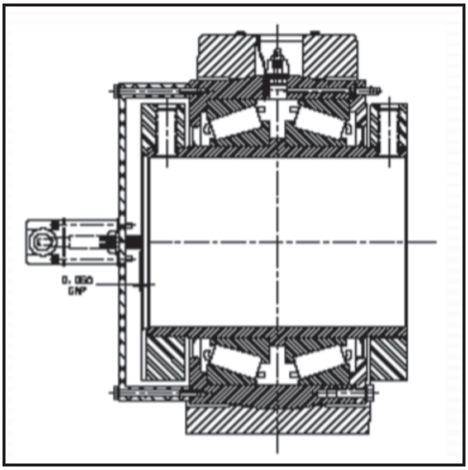


Figure 25. Smart bearing with speed proximity switch – courtesy Baldor – Dodge – Reliance [32].

5.6.2. AVAILABLE SENSORS.

The types of sensors that create smart bearings range in capabilities and usage. For industrial applications, sensors are available to measure speed, direction, temperature (thermocouple) and vibration (accelerometer) [32].

5.6.3. CONDITION MONITORING.

Manufacturers continue to explore the benefits and uses of smart bearings in specific applications. Currently, smart bearings are evolving to have the ability to measure bearing system performance and predict the remaining useful life.

Condition monitoring units are another option in predictive bearing maintenance technology and can be used in conjunction with smart bearing sensors. Just as sensors are being used to transmit data to a source, condition monitoring units are external devices that can receive data on the operating conditions of equipment to ensure peak performance. Together, these devices can communicate to an operator when critical machine elements have become worn, contaminated, damaged, improperly lubricated or experience a rise in temperature or vibration, all leading to potentially costly down time and repairs.

As industries continue to grow and develop, additional smart bearing sensor data are needed to more closely monitor proper bearing function, something essential to optimal operation. Advancements in bearing technology, including data sharing and maintenance tracking, will continue to be researched and developed for more applications [32].

6

DISMOUNTING - RECONDITIONING AND MOUNTING OF A RAILWAY BEARING

Currently, bearings AP-2™ and CTBU of the principal manufacturers TIMKEN and SKF, respectively, may be installed or removed with a bearing press, wheel press or portable fixtures, depending on production requirements.

To ensure that bearings are properly seated, bearing or wheel presses should be equipped with relief valves so that the specified pressure can be maintained for a short interval. Bearings may not be properly seated if the required pressure is not obtained during the surge of the press when the backing ring/seal wear ring of the bearing contacts the axle shoulder [37].

6.1. Introduction of TIMKEN make CTBU.

As the name implies, the Cartridge Taper Roller Bearing unit is used for the wheels of LHB coaches. There are two types of CTBUs/TBUs. One is manufactured by M/s TIMKEN and the other by M/s SKF. They are the only suppliers of CTBUs/TBUs for Indian Railways.

The cartridge bearing is a self-contained, pre-assembled, pre-adjusted, pre-lubricated tapered roller bearing unit; it is applied to and removed from the axle without exposing the bearing elements, or lubricant to contamination or damage. This pre-assembled cartridge bearing reduces the number of separate parts to be applied to the axle assembly to a minimum. The CTBU is designed and manufactured to meet high technical and safety standards [38].

6.2. Bearing or wheel presses.

Bearing presses or wheel presses should be checked with a load cell to ensure that the ram pressure, as indicated by the gauge, is correct in the tonnage range and for the piston travel required to apply roller bearings to axles [36, 37].

When roller bearings are applied in a bearing or wheel press, the following may be used [37]:

- Pilot sleeves fastened to the end of the axle and separate assembly sleeves, as shown in figure 26.

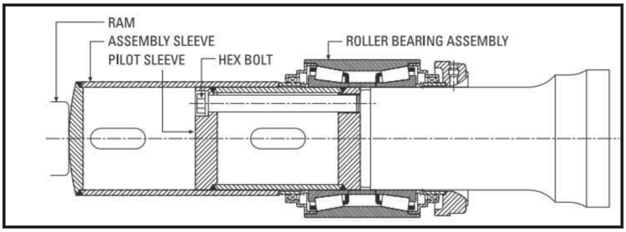


Figure 26. Separate sleeve method of applying roller bearings to an axle [36].

- Telescoping pilot and assembly sleeves fastened directly to the ram of the press, as shown in figure 27.

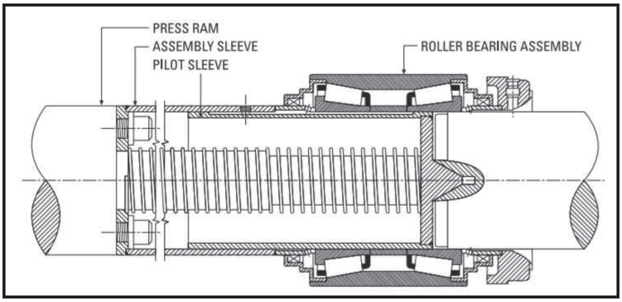


Figure 27. Telescoping sleeve method of applying roller bearings to axle [36].

- Assembly sleeve used to mount a bearing, shown in figure 28.

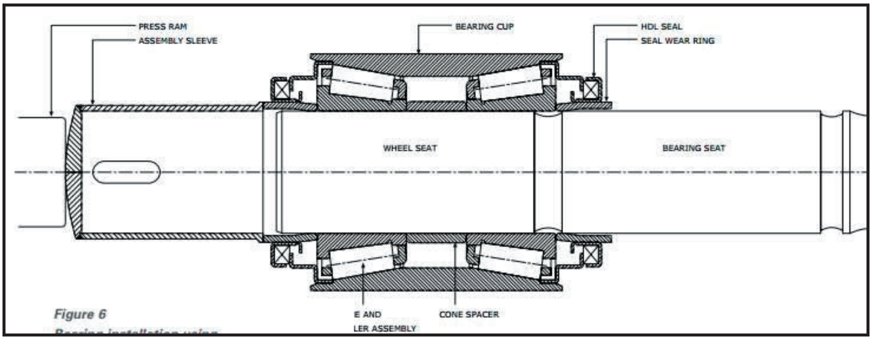


Figure 28. Bearing installation using assembly sleeve [37].

- Bearing presses may be double-ended to apply bearing assemblies to both ends of the axle at the same time (figure 29).

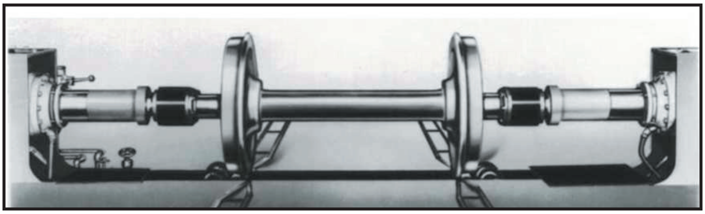


Figure 29. Applying bearing assemblies to both ends of an axle at the same time [36].

The fixtures required to remove a bearing in a bearing press or in a wheel press without removing the wheel are shown in figure 30.

The adapter shoe and reach rods required to attach the fixture to the bearing press or wheel press should be designed to suit the specific press conditions. The pulling shoe insert adapter must be held down in position behind the backing ring until the initial pressure has been applied to ensure proper contact with the backing ring [36].

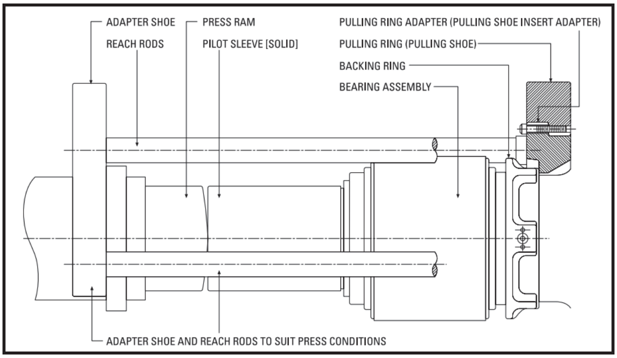


Figure 30. Fixture for removing a bearing with a bearing press or wheel press [36].

6.3. Portable fixtures.

Portable fixtures consisting of a pilot sleeve, assembly sleeve, pulling shoe, reach rods and a base plate may be used for bearing installation and removal. These fixtures can be operated by hand, air or electrically operated pumps and jacks, which are available commercially to suit production requirements. A self-contained portable machine equipped with an electrically operated pump for installing or removing roller bearings is shown in figure 31 [37].

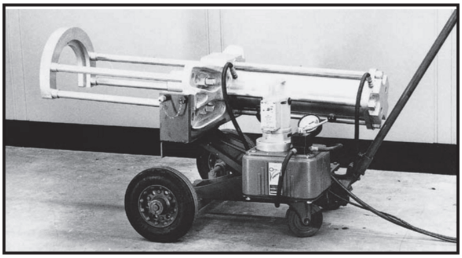


Figure 31. A self-contained portable machine equipped with a pump for installing and removing bearings [37].

6.4. Axles.

Before proceeding with the bearing installation, the axles should be checked to ensure the bearings can be applied without difficulty. Axle bearing seat diameters, length over backing shoulders and shoulder diameters should be checked to ensure that finished axle dimensions are within prescribed tolerance levels to properly mount the bearings.

Axle bearing seat diameters should have a smooth machined, smooth machined and rolled, or ground finish and must be free from sharp corners, burrs, nicks, tool marks, scratches or corrosion. They must be concentric with the wheel seat diameters.

A dial or digital snap gauge (with 0.0001” scale) is shown in figure 32 and 33.

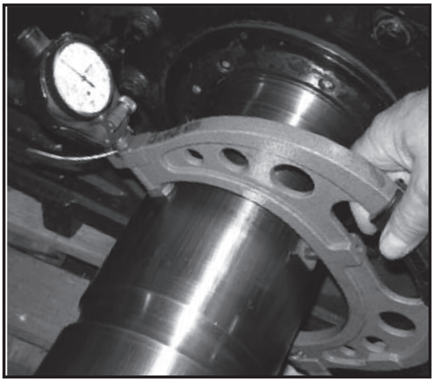


Figure 32. Dial snap gauge [37].

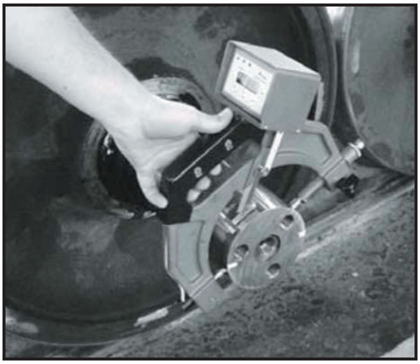


Figure 33. Dial snap gauge [36].

Master gauges must be the same temperature as the axle being measured unless appropriate compensation is made for the temperature difference between the master disc and the axle. Axle diameters should not be checked while the axles are heated for machining. All measuring equipment and axles should be at the same temperature [37].

6.5. Mounting the bearings.

- Roller bearing work should be confined to a specific area.

- Bearing assemblies should not be removed from the shipping package and the protective wrapping should not be removed until the time of application.

- The cardboard insert should not be removed from the bore of the bearing assembly. This insert is required to hold the cone spacer in alignment with the bearing cones when installing the bearing assembly.

- The bearing assemblies are shipped with a protective coating of grease over the vent fitting, if so equipped. Care should be taken that the grease is not wiped off when the bearings are applied to the axles [37].

- Cartridge bearings must be pressed on the axle. Heat must not be applied to the bearing cone assemblies to facilitate installation [38].

- The amount of press fit of the bearing on the axle is pre-determined by the dimensional tolerances of the axle and mounting parts. Bearings and the axle do not need to be selected for fit for any given class.

- A pilot sleeve should be used to keep the cone spacer in alignment with the bores of the cones and to guide the bearing assembly on the axle. The pilot sleeve may be fastened to the end of the axle or may be guided by the lathe centre hole in the end of the axle [37].

- The bore of bearing cones that have had previous service should be checked for acceptability before being pressed on the axle to ensure a suitable interference fit.

- Oversize bearing cones should be scrapped.

- The bearing seats of the axle should be coated with castor oil, heavy mineral oil or a molybdenum-disulphide and oil mixture.

- White lead should not be used; lead compounds may be detrimental to lubricating greases by acting as an oxidation catalyst.

- To minimize the risk of ingress of water through the backing ring contact area with the axle, a sealant (Anabond-685 or equivalent) should be applied to the backing ring/axle interface.

- A thin coating of a quick-drying rust preventative must be applied to the portion of the axle between the wheel hub and the bearing. The rust preventative used must not contain lead or other compounds which may be detrimental to lubricating greases [38].

- When the bearing assembly is slipped on the pilot sleeve and the cardboard insert is ejected, the outboard seal wear ring should be held in place to prevent it from riding out of the seal.

- If the seal wear ring does slip out of the assembly, it must be inserted into the seal correctly and carefully, end first, so that the outer lip of the seal does not turn under when the seal lips are expanded over the seal wear ring.

- No tool or other instrument should be inserted between the seal element lips and seal wear ring, as this may damage the seal element lips or scratch the seal wear ring, resulting in bearing lubricant leakage.

- A small lift or other bearing handling device may be used for handling larger bearing sizes [37].

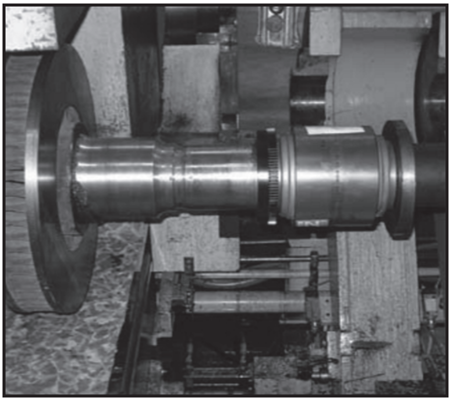


Figure 34. Bearing ready for mounting [37].

6.6. Pressing the bearing assemblies onto axles.

An assembly sleeve, which contacts the seal wear ring outer face and telescopes over the pilot sleeve, is used to press the bearing onto the axle (figures 26 and 27) [36].

- Place the wheel and axle assembly in a bearing press, in position to press the bearing assembly on to the axle.

- Measure and record the outer Dia of axle at three locations. Each location should have three readings at 120 ^{o}C. See the axle dimensions.

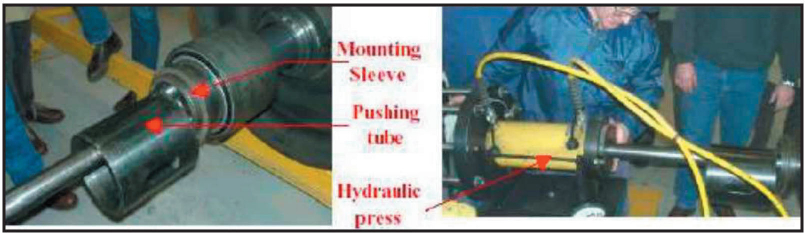


Figure 35. Pressing of bearing assembly on axle [38].

- Fit the pilot sleeve onto the end of the axle, using the screws to hold it in position. Slide the bearing assembly over the pilot as far as it will go and place the bearing assembly.

- Coat the bearing seats of the axle with oil (SAE 30).

- Apply pressure to the end of the assembly sleeve until the bearing assembly is correctly seated. Keep rotating the bearing by using both hands while mounting it on the axle. If it stops rotating, mounting should be stopped. After the bearing is removed from the axle it should be sent for refurbishment.

- Rotate the bearing assembly to make sure it will turn. Due to the rubbing type seals, the bearing assembly will not rotate freely at initial application. However, if the bearing is equipped with HDL seals, it may rotate more freely. New bearing assemblies are pre-adjusted at the factory. No adjustment is necessary at installation.

- To ensure that the bearing is firmly seated against the axle fillet, the pressure indicated on the gauge during pressing should be increased by 50%. This seating load pressure should be within the limits of 20-30 tonnes [38].

6.7. Checking bearing lateral play.

Check the bearing mounted lateral play with a dial indicator mounted on a magnetic base. To measure the bearing lateral play, place the magnetic base on the axle end and position the indicator stem against the face of the cap as shown in figure 36. Force the bearing cup laterally away from the indicator and pull the bearing cup toward the indicator. Oscillate the bearing while making the lateral measurement to ensure that the rollers are seated for an accurate measurement [36].

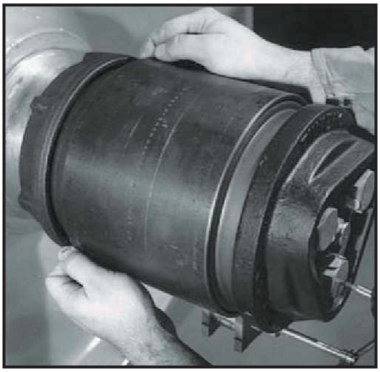


Figure 36. Bearing ready for mounting [36].

If bearing end play as indicated by the dial indicator is less than minimum "MEP at installation", remove the bearing assembly from the axle. Minimum and maximum end play values at the mounted end are from 0.025 mm to 0.330 mm for a new bearing and for an in-service bearing, 0.025 mm to 0.500 mm. If the bearing does not fall under the above MEP range and it can still be rotated by hand, it can be taken into service/operation [38].

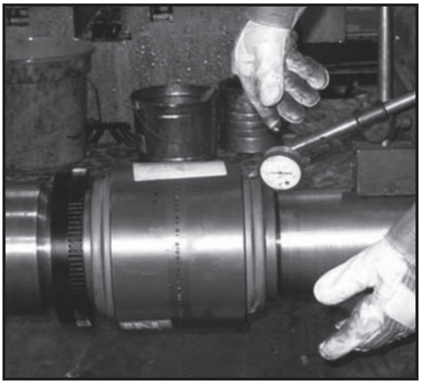


Figure 37. Bearing mounted lateral check using dial indicator [37].

Figure 38. Bearing mounted lateral check using dial indicator [38].

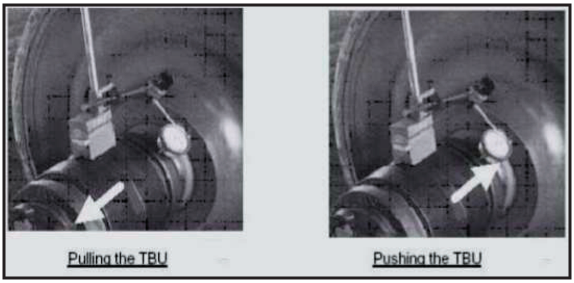


Figure 39. Bearing mounted lateral check using dial indicator [38].

6.8. Removing the bearing.

- After removing the wheel and axle assemblies from the truck, remove the adapters or housings from the bearings.

- Stop blocks, or cap screws and nuts used as stop blocks, must be removed from adapters so equipped, prior to removing the adapters from the bearings.

- If difficulty is experienced removing bearing housings, it may be desirable to modify the bores of the housings to eliminate this difficulty at subsequent housing removals (contact the manufacturer's representative).

- Otherwise, thoroughly clean the bores of the housings, remove all rust or corrosion and apply a heavy coating of grease to the bores of the housings.

- If a $2^{7/8}$ plug, retained with a locking plate, is used in the axle end cap, it must be removed before removing the cap screws.

- Bend the tabs of the cap screw locking plate away from the heads of the cap screws.

- Remove the cap screws, locking plate and axle end cap. It may be necessary to tap the end cap lightly for removal.

- If more than one bearing assembly is to be removed, the guide tube should be fastened to the hydraulic ram, if possible. Usually, a guide tube that is a size smaller than the axle size is used to eliminate alignment problems.

- If portable machines are used, be sure to tighten the guide tube against the ram head so the pressure required to remove the bearing will not shear or bend the connector pin (figure 40).

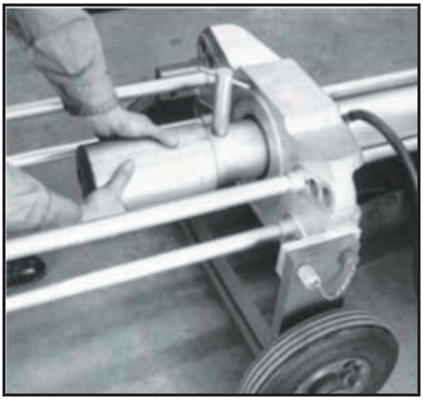


Figure 40. Portable machines [36].

- When portable machines are used and only one bearing is to be removed from an axle, and a bearing is to be immediately applied, fasten the guide tube of the proper size to the axle. It will then be in place for installing the bearing (figure 41).

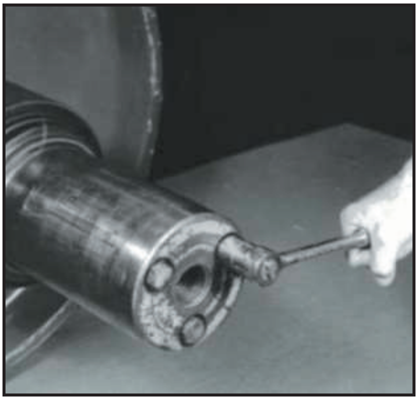


Figure 41. Installing the bearing [36].

- A pulling shoe and pulling shoe insert adapter, similar to that shown in figure 42, which fits behind the backing ring as shown in figures 42 and 43, is used to remove the bearings when it is desirable to remove the bearings without removing the wheels.

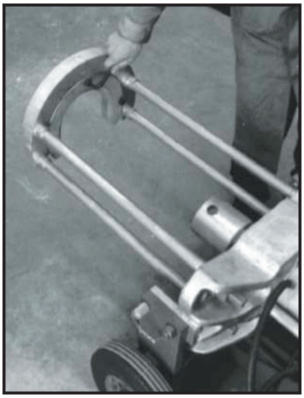


Figure 42. Pulling shoe insert adapter [36].

- Make sure the pulling shoe is of the correct size for the bearing to be removed. Proper contact with the backing ring and puller alignment is necessary for efficient bearing removal.

- Position the pulling shoe behind the backing ring. The pulling shoe contact surface of the backing ring is very narrow. Therefore, it is necessary to hold the pulling shoe down in position behind the backing ring as shown in figure 40. This must be done until the initial pressure has been applied, to ensure proper contact with the backing ring and to prevent distorting or bending the backing ring. Extend the ram to remove the bearing from the axle [36].

Figure 43. Pull shoe down in position behind the backing ring. [36].

Table 12. Sequence of proper dismounting and mounting [38].

Activities
Removal of Bearing from Axle: ✓ Bend the tabs of the bolt locking plate away from the heads of the bolts. ✓ Remove the bolts, locking plate and axle end cap. ✓ Fit the Pilot Sleeve to the axle end. ✓ Ensure the removal of bearing is done as specified in maintenance manual of M/s TIMKEN & SKF. The falling of the bearing freely on the ground after removal should be avoided by putting suitable rubber cushioning. ✓ After the bearing assembly is removed from the pilot sleeve, a card board to be inserted in the bore of the bearing assembly to hold the internal bearing parts in place. ✓ Ensure bearings are kept in a clean and dry place in a covered room. ✓ Ensure bearings are stacked properly and not thrown one over the other.
Mounting of Taper Bearing on Axle: ✓ Check journal dial at three points as described in case of new axle. ✓ Clean the journal with kerosene oil and inspect carefully. ✓ Check threads with the help of thread gauge. ✓ Apply lubricant on the journal. ✓ Tighten the pilot sleeve on axle with the help of bolts. ✓ Put the bearing assembly on pilot sleeve and refit the bearing on journal with 20-25 tonne seating load (for Timken Makes bearings). ✓ Rotate the bearing for ensuring free movement. ✓ Tighten the security plate & security disc with M-20 cap screw at 18-22 Kg-m torque with the help of torque wrench. ✓ Check mounting lateral play with the help of magnetic base dial indicator. It should be 0.025 mm to 0.33 mm for new bearings and 0.025 mm to 0.500 mm for in-service bearings 3.0 Seal the cap screw by bending tabs of the locking plate.
End Cap: ✓ Inspect end cap for cracks, breakage, wear and distortion of machined surface.
Cap Screw and Locking plate: ✓ Check cap screws for wear on threads and for stretching or elongation. ✓ Check that cap screw threads are properly cleaned and lubricated before fitment. ✓ Lock the bolts by bending all tabs of the locking plate flat against the sides of the bolt heads using adjustable rib-joint pliers. ✓ Ensure that cap screws are tightened with torque wrench. (Set at 18-22 Kg-m) ✓ Ensure that locking plates are not re-used.

6.9. Bearing reconditioning.

Bearings are core components of plant assets, and they take a lot of punishment. Bearings are usually replaced during planned maintenance when nearing the end of their operational life or after unplanned breakdowns. Depending on the bearing type, replacement can be expensive and may involve long lead times. In addition, scrapping "end of life" bearings may have a negative impact on a company's sustainability profile – an aspect that is increasingly important to investors and customers. How can one increase the service life of bearings in order to decrease downtime, reduce costs and reduce scrap? Remanufacturing is the key [39].

6.9.1. CLASSIFICATION OF BEARING REWORK.

Terms and definitions of reworked used rolling bearings are explained in the following.

According to the rate of prior use and the wear status, companies like SKF and TIMKEN divide bearing rework into five classes.

While the operations in the particular classes are numbered, the actual sequence of the work is not directly related to these numbers. Special agreements between the maintenance company and the operator must be respected [39].

- Class 0 – Inspection.

  Class 0 includes inspecting used bearings (or bearings stored for a long time) and comparing them with drawing/specification requirements. This process involves:

  1. Cleaning.
  2. Non-destructive testing.
  3. Visual/microscopic inspection.
  4. Dimensional inspection.

5. Providing a report.

Note: Usually a recommendation for appropriate treatment and suitable rework is given.

- Class I – Reclassification (requalifying, reclamation).

  Reclassification includes all operations of inspection (Class 0) and the following additional work:

6. Minor repair: buffing and minor polishing of inactive and active surfaces, grinding of scratches and grooves.
7. Demagnetization.
8. Reassembly (including cage riveting if necessary).
9. Dynamic testing (if required): bearing ring rotation to evaluate noise levels, determine torque characteristics and/or similar functional parameters.
10. Lubrication/preservation.
11. Packaging.

- Class II – Refurbishment (reconditioning).

  Refurbishment of bearings encompasses all operations of inspection (Class 0) and reclassification (Class I) plus one or more of the following:

12. Replacing rolling elements if necessary, installing oversize rolling elements (of the same nominal diameter (see further work).
13. Remanufacturing the cage or replacing it with an identical cage.
14. Interchanging used components (e.g., seals, snap rings, etc.).
15. Grinding or polishing and/or plating mounting surfaces as necessary to return to original drawing dimensions of the bearing outside surface and bore.

16. Polishing raceways (not to exceed 13 µm total metal removal per surface).

- Class III – Remanufacturing level 1.

  Remanufacturing level 1 for bearings encompasses all previous operations of inspection (Class 0) and reclassification (Class I) and, where appropriate, refurbishment operations (Class II) plus one or more of the following operations:

17. Grinding raceways – up to 75 µm per surface and if the outside diameter, D> 400 mm, up to 300µ.
18. Installing oversize rolling elements of a larger nominal diameter (see operation 13).
19. Installing the original reworked cage or a new one (see operation 14).
20. Changing or substituting components to create a different assembly identity (tuning: modifying in order to improve performance or properties).

- Class IV – Remanufacturing level 2.

  Remanufacturing level 2 for bearings involves reworking bearings of Classes I to III plus an additional operation:

21. Installing a new ring.

6.9.2. Reconditioning of bearings by the Duroc Company.

In order to know more details about the process of reconditioning bearings, we visited the Duroc Company, which performs maintenance on railway wheels for locomotives and wagons for Swedish and European railway operators; thus, the company is also responsible for performing maintenance tasks on and the reconditioning of bearings used in the railway sector.

In principle, the rework process performed by Duroc includes only activities identified in the reconditioning Class 0 and a few of Classes I and II. The other classes of reconditioning (III, IV) are conducted by larger companies, such as SKF and TIMKEN, whose processes of rail reconditioning facilities worldwide are certificated by international standard ISO.

To have a better understanding of the bearing reconditioning activities performed by Duroc Company, the different stages of the process are listed below:

1. Normally, the bearings are received in the company directly coupled to the wheels and are inside the axlebox, as shown in figure 44.

Figure 44. Wheels received by the company.

2. Initially, the wheel is placed on a rail for to be moved to a hydraulic lift that lifts it to an appropriate height (see figure 45) and then proceeds to dismount the axlebox.

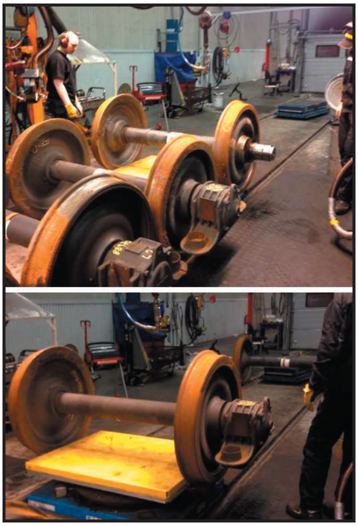


Figure 45. Rail and hydraulic lift.

3. The axlebox is unscrewed on both sides of the wheel with the help of a pneumatic screwdriver; it is then dismounted using an overhead crane and carried on a table for cleaning (see figures 46 and 47, respectively). Finally, using a hydraulic bearing press, the bearings are extracted from the axlebox (see figure 48).

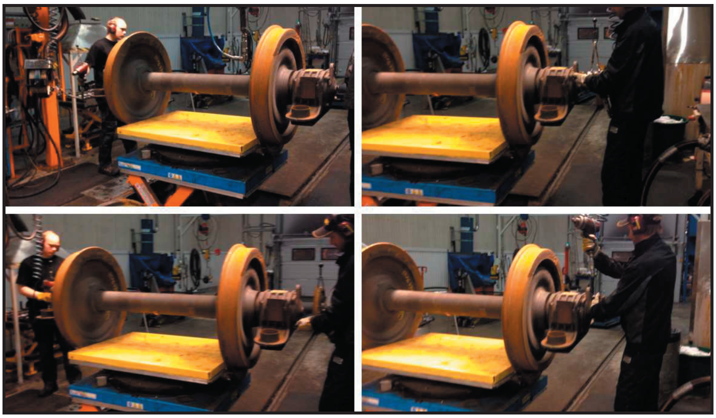


Figure 46. The bearings are removed with a pneumatic screwdriver.

Figure 47. Dismounting the axle box.

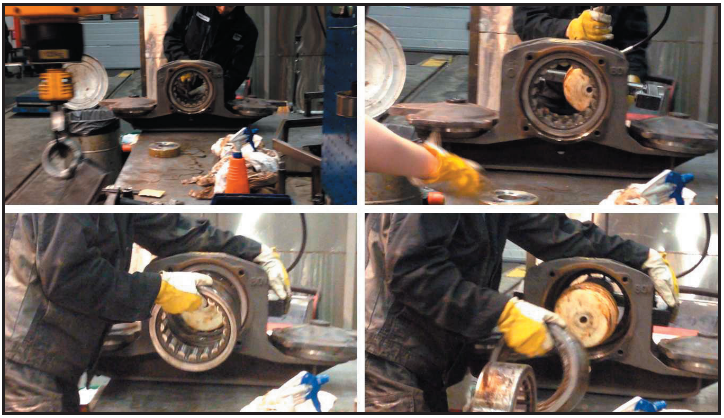


Figure 48. Using the hydraulic press bearings to dismount bearings of the axle box.

4. An inspection of the wheel is performed (see figure 49), and it the next activities are determined, i.e.: i) Carry the wheel directly to a machining process in a mechanical lathe to hone it (see figure 50 and 51), or ii) Move it to other station, where the bearings will be extracted (SKF CTBU or TIMKEN AP-2) (see figure 52).

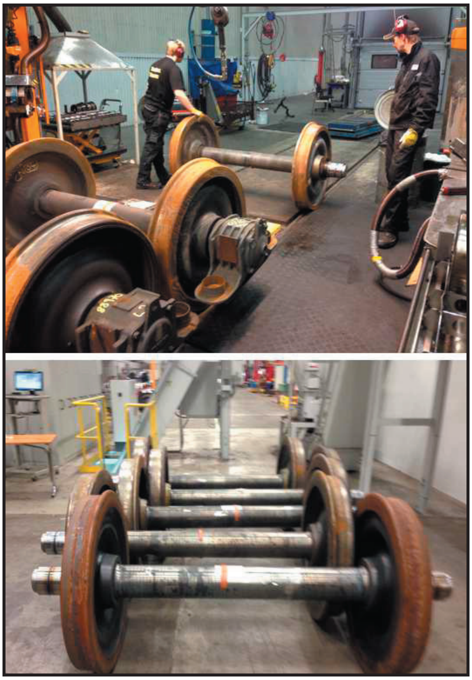


Figure 49. Inspection of the wheels.

Figure 50. Machining process (honing).

Figure 51. Wheels after machining process (honing).

Note: After honing, an anticorrosive or antirust is applied to the wheel, as shown in figure 52.

Figure 52. Machining process and anticorrosive aplication.

Figure 53. Inspection of the bearings before dismantling.

5. Bearings that are transferred to another station are dismounted from the wheel with the help of a portable fixture, as shown in figure 54.

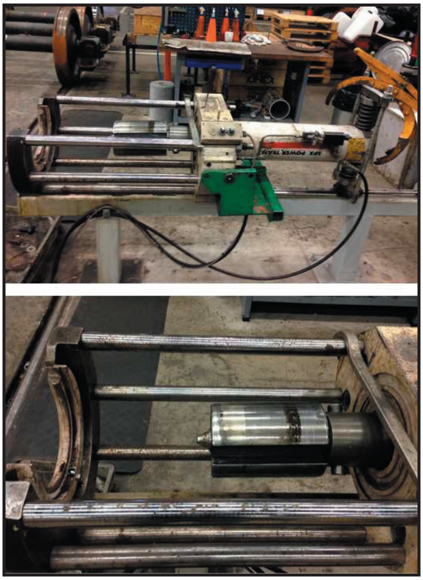


Figure 54. Portable fixture used in Duroc Company (Bearing press).

6. Once the bearings are dismounted, they are separated into their different parts: Inner ring, Outer ring, Bearings, Polymer cage, Snap rings, etc. However, we should mention an important feature in the design of these bearings; the outer ring has two possible configurations, i.e., one common outer ring (see figure 55), or two outer rings coupled together (see figure 56). This determines the conditions to which the bearing will be subjected.

Figure 55. Bearings with one common outer ring on the raceway.

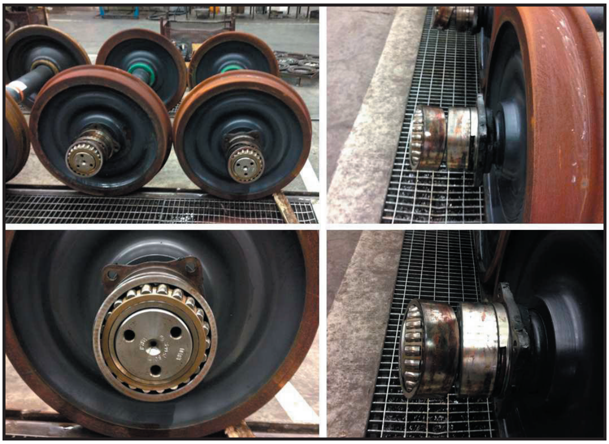


Figure 56. Bearings with two outer rings on the raceway.

Note: The axlebox is also separated into its respective parts (see figure 57), which will be inspected; depending on the physical conditions of wear, the used components may be replaced (i.e., seals, snap rings, cages and others). Also, the axleboxes for the wheels arriving at DUROC have different designs to accommodate spherical, cylindrical or tapered roller bearings (as mentioned in section 4.4).

Figure 57. Axle box parts.

7. The remanufacturing process begins with the "cleaning" of the outer ring, to remove all oxide on the surface (see figure 58). This "cleaning" is done with a machine consisting of rotating rollers, on which is placed the outer ring; while it rotates in one position, a wire brush (also turning) removes the oxide on the ring surface (see figure 59).

Figure 58. Outer ring of the bearings (presence of oxide on the surface).


Figure 59. Machine used to remove oxide.

8. The rings and rolling elements extracted from the axle box and from the CTBU-SKF and TIMKEN AP-2 are placed on frames and metallic baskets, as shown in figures 60 and 61, respectively.

Figure 60. Rings and rolling elements on metallic frame.

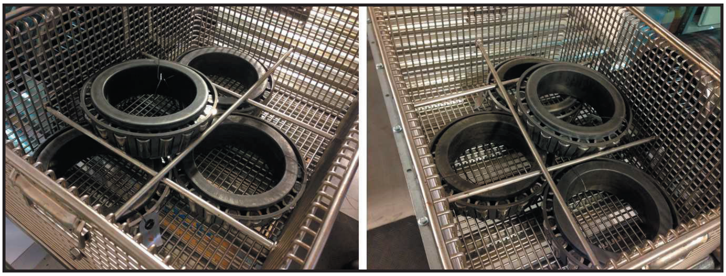


Figure 61. Rolling elements in metallic baskets.

9. Next, they are inspected in detail. Those in very poor condition are discarded (see figure 62), while the rest continue with the reconditioning process, beginning with a degreaser bath and alcohol (see figure 63).

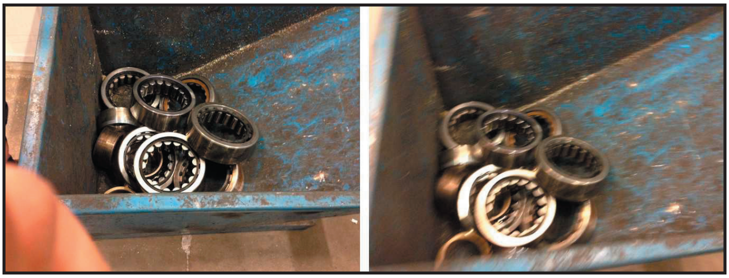


Figure 62. Discarded rings and rolling elements.

Figure 63. Machine for degreaser bath.

Note: Figure 63 shows the equipment used to perform the degreasing bath. The red circles indicate the entrance to and exit from the degreasing process. The alcohol containers are also shown.

10. After the degreasing bath, the parts are taken to a laboratory for replacement of used components, if necessary (see figure 64).

Figure 64. Different parts in the laboratory.

11. The process begins in the laboratory with a visual/microscopic inspection for cracks, scratches and defects. The equipment used to perform this task is shown in figure 65.

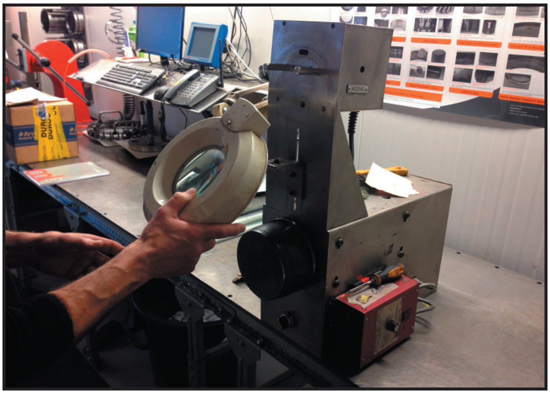


Figure 65. Equipment to perform visual/microscopic inspection.

12. The polymer cage is replaced, and using a press, the inner rings are positioned in the cage (see figure 66). Once these are installed, the polymer cage is assembled.

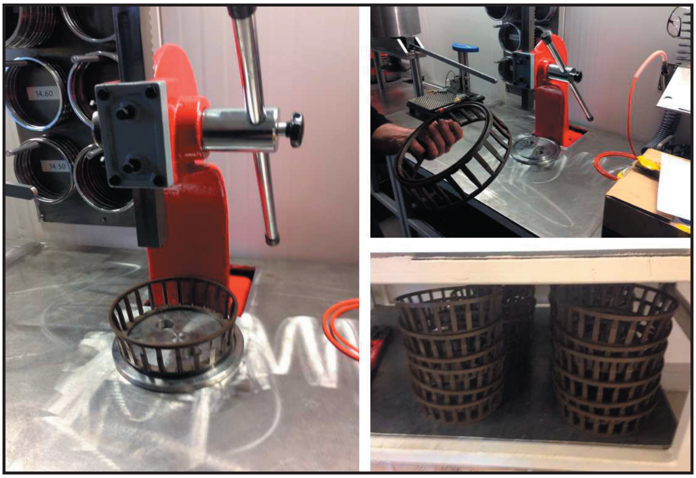


Figure 66. Replacement of the polymer cage.

13. As soon as the polymer cage and rolling elements are replaced, the snap rings, seals, etc. are installed. Figure 67 shows the parts mentioned.

Figure 67. A) Snap rings, B) Housing rings, C) Seals.

Note: The method of assembling these parts is not shown in detail, because at the time of our research, the company was not performing these tasks. Thus, we merely provide a brief explanation of the process.

14. Once the bearing assembly process is complete, it is ready to be lubricated. The lubrication is performed using a grease injection inserted in a hole on the lateral side of the bearing. The grease injection is performed with the fixture shown in figure 68.

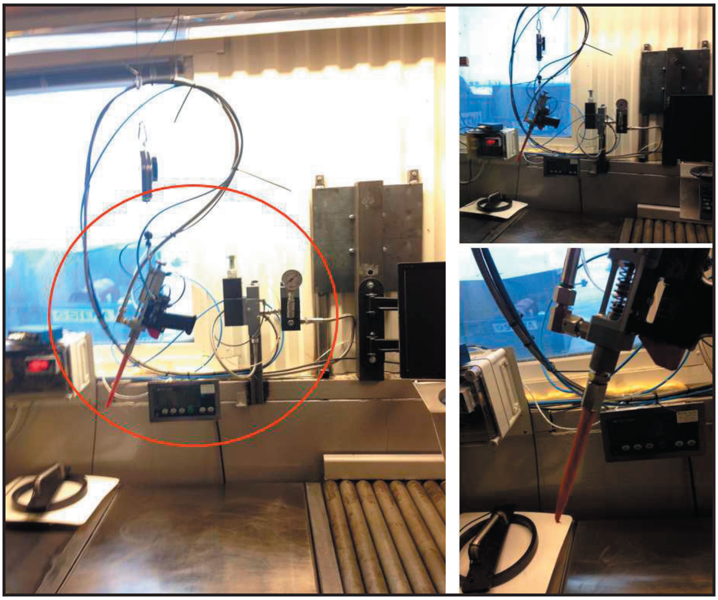


Figure 68. Grease injector.

15. Finally, after lubricating the bearing, a report is generated on the operations and replacements (see figure 69), and the bearings are mounted on wheels that have come out of the machining process. Once a wheel is assembled, it is carried to the last station (delivery / distribution (see figures 70 and 71)).

Figure 69. Providing a report.

Figure 70. Assembled wheel with axle box.

Figure 71. Last station (delivery / distribution).

7
CURRENTS IN WHEEL BEARINGS

Reliability is an essential requirement for modern bearings, together with extended maintenance intervals. For example, they must be robust enough to operate in electrical equipment with modern AC technology. The passage of electric current through rolling bearings can cause damage in a relatively short period of time.

To enable smooth, trouble-free operation, it is necessary to prevent electric current from passing through the bearing. The contact areas between the housing, the outer ring, the rolling elements, the inner ring and the shaft act as electric contacts.

Bearing damage can occur depending on the electrical regime, the bearing impedance and its tribological behaviour in the rolling contact. Electrical discharges in the form of arcs/sparks can damage rolling elements and raceways of bearing rings. This process is characterized by very high temperatures similar to those produced by the melting and the welding processes. Craters are typical damage and in a later stage, as a secondary effect, fluting or washboarding can occur [47].

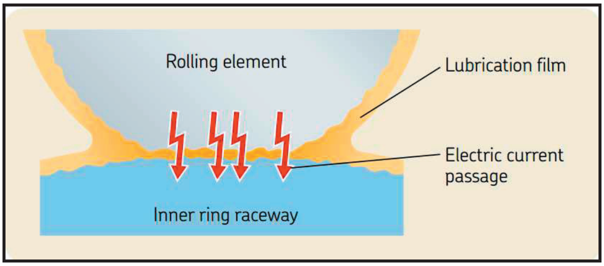


Figure 72. Principle of electrical current passage on the rolling contact area [47].

7.1. Electrical erosion process.

7.1.1. EXCESSIVE VOLTAGE.

When an electric current passes through a bearing by going from one ring to the other via the rolling elements, damage can occur. At the contact surfaces, the process is similar to electric arc welding where there is a high current density over a small contact surface.

The material is heated to temperatures ranging from tempering to melting levels. This causes the appearance of discoloured areas, varying in size, where the material has been tempered, re-hardened or melted. Craters are formed where the material has been melted.

Appearance: Single craters in raceways and/or rolling elements. Localized burns in raceways and on rolling elements [41, 42, 47].

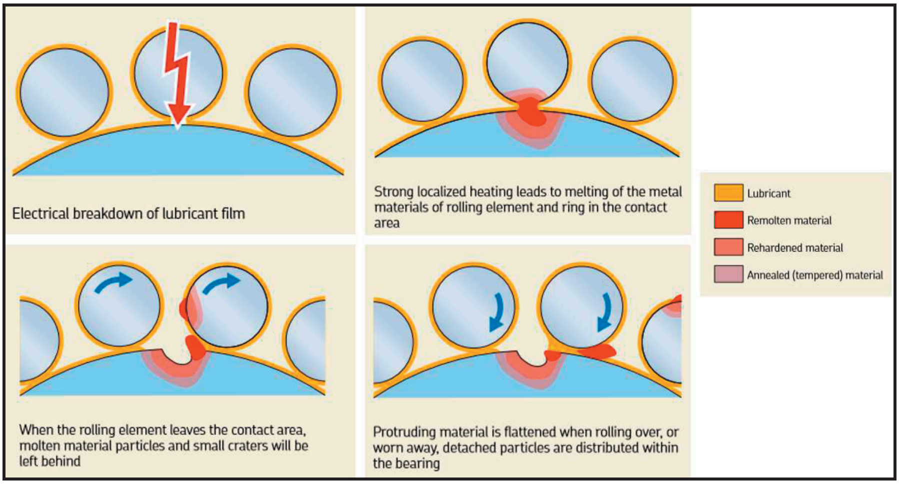


Figure 73. Bearing damege caused by electrical erosion [47].

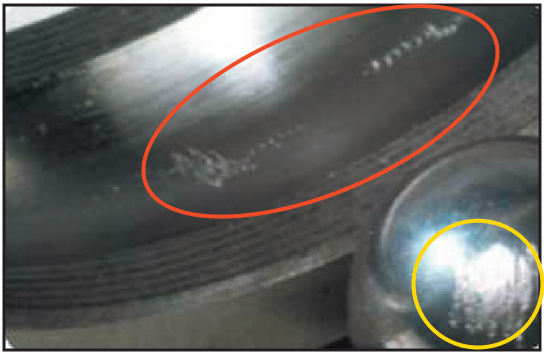


Figure 74. Excessive voltage. Raceway of an outer ring of a deep groove ball bearing. 1) Craters in a beadlike procession (red circle). 2) Zig-zag pattern in the outer ring and the ball (yellow circle) [47].

Figure 75. Excessive voltage. Ball bearing raceway with large spalls [47].

7.1.2. CURRENT LEAKAGE.

Where current flows continuously in the form of arcs through the bearing in service, even at low intensity, the raceway surfaces are affected by the heat; they erode as many thousands of micro-craters are formed, mostly on the rolling contact surface. These craters are closely positioned to one another and are small in diameter compared to the damage from excessive voltage. Flutes (washboarding) will develop from craters over time as a secondary effect and are found on the raceways of rings and rollers.

The extent of damage depends on a number of factors: type of bearing, bearing size, electrical regime, bearing load, speed and lubricant. In addition to bearing steel surface damage, the grease close to the damage might be carbonized, eventually resulting in poor lubricating conditions and consequently to surface distress and spalling.

Very similar electrical erosion damages appear in axle box applications and sometimes in gearbox applications. DIN VDE 0123 describes the current flow in railway vehicles in detail [47].

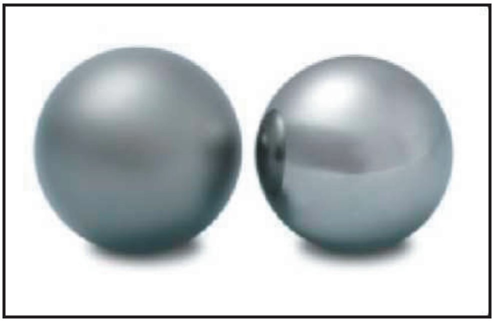


Figure 76. A dull grey surface of the rolling elements can be a sign of microcratering. Left: ball with dull gray surface. Right: new ball with mirrored picture of the ball to the left [47].

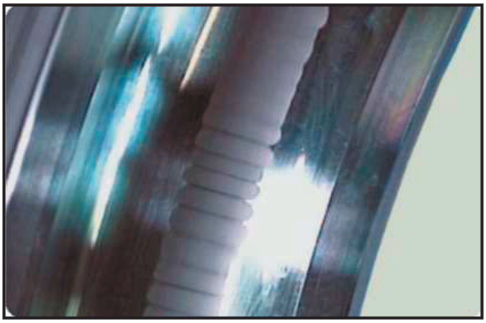


Figure 77. Fluting or washboarding in a raceway caused by the passage of damaging electrical current. Outer ring raceway of a deep groove ball bearing [47].

Figure 78. Fluting or washboarding in a raceway caused by the passage of damaging electrical current [47].

The passage of electric current frequently leads to the formation of fluting (corrugation) in bearing raceways (see figures 78 and 79). Rollers are also subject to fluting when balls have a dark discolouration. It can be difficult to distinguish between electric current damage and vibration damage. A feature of the fluting caused by electric current is the dark bottom of the corrugations, as opposed to the bright or rusty appearance at the bottom of the vibration-induced fluting. Another distinguishing feature is the lack of damage to the rolling elements of bearings with raceway fluting caused by vibrations.

Both alternating and direct currents cause damage to bearings. Even low amperage currents are dangerous. Non-rotating bearings are much more resistant to electric current damage than bearings in rotation. The extent of the damage depends on a number of factors: current intensity, duration, bearing load, speed and lubricant.

The only way to avoid damage of this nature is to prevent any electric current from passing through the bearing [41, 42].

Table 13. Features, causes and actions to take when damage is caused by the passage of electric current through bearings [42].

Appereance	Cause	Action
Dark brown or greyish black fluting (corrugation) or craters in raceways and rollers. Balls have dark discolouration only. Sometimes zigzag burns in ball bearings raceways.	Passage of electric current through rotating bearing.	• Re-route the current to by-pass the bearing. • Use insulated bearings.
Localised burns in raceways and on rolling elements.	Passage of electric current through non-rotating bearing.	• Re-route the current to by-pass the bearing. When welding, arrange earthing to prevent current passing through the bearing. • Use insulated bearings.

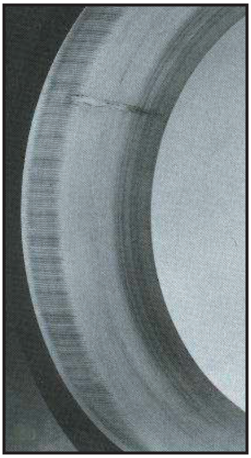


Figure 78. Fluting caused by the passage of electric current, in the outer ring of a spherical roller bearing [41].

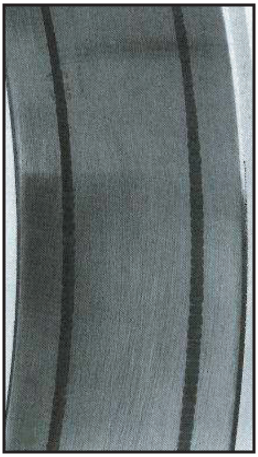


Figure 79. The outer ring of a self-aligning ball bearing damaged by electric current [41].

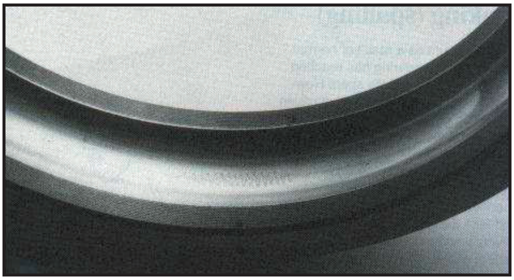


Figure 80. Deep grove ball bearing with electric current damage in zigzag pattern. It is assumed that burns of this configuration arise when the momentary passage of high amperage current is accompanied by axial vibration [41].

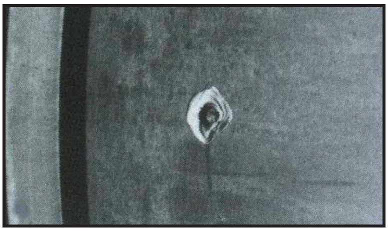


Figure 81. A railway axle box bearing damaged by the passage of high amperage current while the bearing was not running [41].

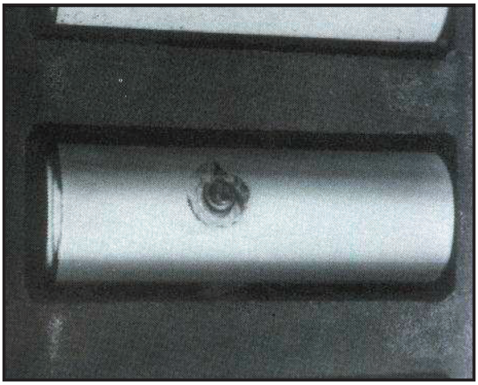


Figure 82. Roller of a railway axle box bearing damaged by electric current (same bearing as in figure 75) [41].

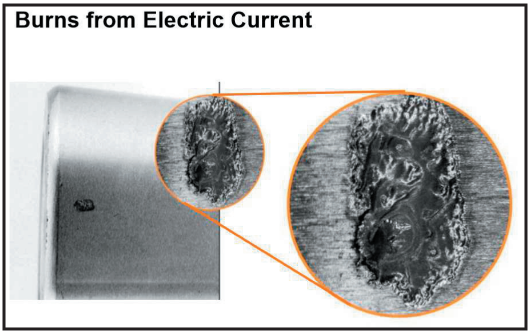


Figure 83. Pitting / small burns created by arcs from improper electric grounding in stationary bearing [43].

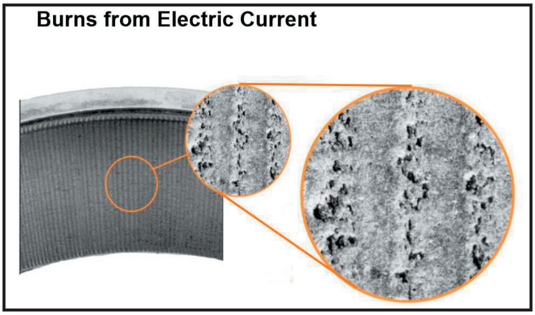


Figure 84. Fluting (small axial burns) caused by electric current passing through the bearing while it is rotating [43].

7.2. Current distribution along the train.

Previous investigations and measurements have shown that the coupling current is far above expected levels and the current distribution over a train is such that the bearings of the last part of the train tend to take more current than the bearings of the first part. Since the magnitudes of the currents are greatest closer to the engine, it could be advantageous to electronically isolate the couplings of the engine. This could reduce the total current into the train and, consequently, reduce the currents in the bearings [44].

7.2.1. CURRENT PATHS IN THE TRAIN.

As an aid, a short description of the possible current path from the engine connected to any car is given in figure 85. The figure shows only one engine for clarity. However, when used for ore transport, two engines are connected in series. Each locomotive has six motors, three axles supplied with carbon brushes to give a defined current path to the rail, two axles without carbon brushes, and one axle with carbon brushes connected to the chassis. In figure 85, a few possible current paths are indicated with arrows; the figure shows the six axles and the couplings to the cars (first, *n*th, and last). The situation described is as expected when current is flowing from the locomotive into the first coupling, and added through the bearing of the first cars, travelling through the couplings of the train; a portion of the current is distributed over the bearings of cars and the remainder exits through the bearings in the last car. The figure does not represent the situation when an autotransformer system (AT) is in front of the engines and the current is not going through the cars in the return path [44].

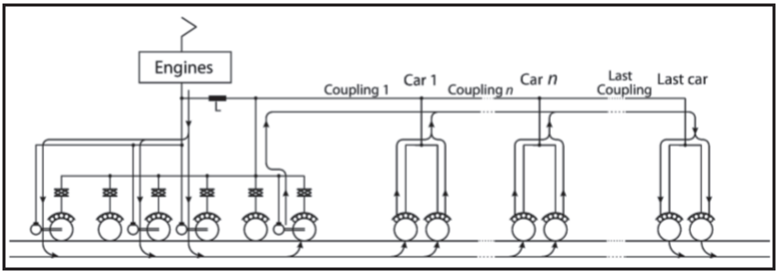


Figure 85. Schematic sketch of the current path in the engines, couplings, and cars. This is a ground protection choke. The engine has six engines in each of the two series electric locomotives. The cars have four axles but only two are shown [44].

7.3. Methods of measuring currents and voltages in wheel bearings.

7.3.1. Indirect method of measuring currents in wheel bearings.

The current flowing through the front and the rear coupler on the same car are simultaneously measured using Rogowski ampere meters, as shown in figure 86. The difference between the two measured currents is the current going through the wheel bearings.

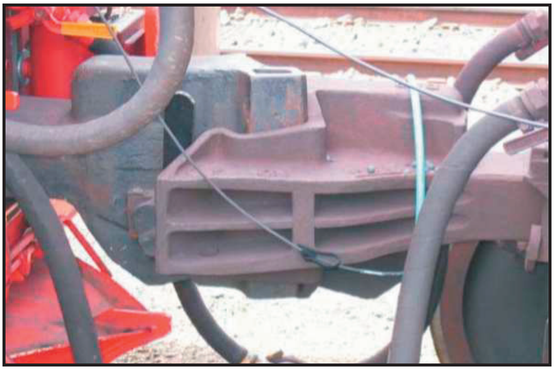


Figura 86. Rogowski current probe measuring the current in the coupling between the engine and the first car [45].

7.3.2. Direct method of measuring currents through wheel bearings.

One of the ore cars was specially prepared for measuring the electrical currents flowing through the wheel bearings by electrically insulating the wheel bearing boxes from their bogie-truck frames, using plastic sheets on the contact surfaces, and connecting a cable between the front left bearing box, nearest the front coupler, and its bogie frame and a second cable on the rear right bearing box, nearest the rear coupler, and its bogie frame, as shown in figure 87. Tong amperemeters were clamped around these cables to measure the current flowing through the cable and wheel bearing boxes, i.e. through the bearings.

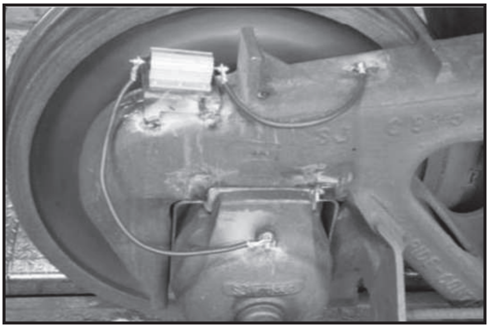


Figure 87. Axis prepared for measuring current from the boggie frame to the bearings. Electric isolation is wrapped around all eight bearing boxes of the car. A cable is connected between one isolated bearing box and the boggie frame. A tong ampmeter is attached around the cable. (The metallic component attached to the cable is a resistor, not used when measuring bearing currents) [45].

7.3.3. Method of measuring voltages over the wheel bearings.

All four wheel axles, the bearings and the bearing boxes were electrically isolated from the bogie truck frames by wrapping the contact surfaces in plastic sheets. One of the wheel bearing boxes was equipped with a carbon brush making contact at the end of the wheel axle. The voltage measured between the wheel axle and the bogie frame was considered to be equal to the voltage over the bearing, as shown in figure 88.

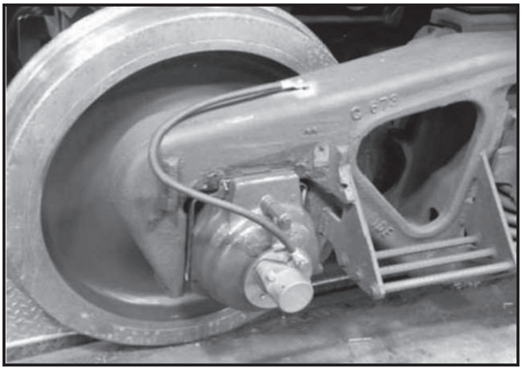


Figure 88. Detail of car prepared for measuring bearing voltage. A carbon brush contact is installed at the end of one axis for measuring the voltage between the carriage and the axis. The volt meter is connected between the boggie frame and the cable. The volt meter is not shown on this picture [45].

7.3.4. METHOD OF MEASURING PRIMARY CURRENTS.

The total current from the contact wire to the engines is denoted primary current. Initial checking showed that both engines in tandem always took equally big currents, due to the motor control systems. Therefore, it was considered enough to measure the primary current in one of the two engines. The primary current consumed by the engine could easily be measured and monitored by means of a tongamperemeter gripping a factory installed cable in the high voltage cabinet of the engine [45].

7.3.4.1. Storing measured data.

Using laptop computers and measuring cards, the current and voltage values were sampled at a rate of 10 kS/s, but only the calculated RMS-values were stored, once per second. In exceptional cases, in order to check waveforms and frequency spectra, short bursts of signals were stored at the full sampling rate, 10 kS/s.

7.3.4.2. Method of measuring time, position and speed.

A GPS receiver connected to one of the laptop computers continuously plotted position coordinates, speed and time. These GPS data were stored once per second. Since the currents and voltages in different positions along the train were measured by means of four different laptop computers, each computer with its own internal clock, these recordings had to be synchronized with respect to time.

7.3.4.3. Time synchronization.

Comparing starting and stopping times at well known locations along the track achieved time synchronization of the GPS data and the measured current and voltage data. Knowing the length of the cars gave the distance between the GPS receiver and the measuring points on the train [45].

Part IV
RESULTS AND CONCLUSIONS

RESULTS AND DISCUSSION 8

8.1. Bearing lifetime (L).

Using equation 15 presented in section 3, is estimated the value of life (L) given the climatic and working conditions.

Table 14. Bearing lifetime (L) for reliability differences.

Reliability	a_1	Life (L) - Winter (Km)	Life (L) - Summer (Km)
90%	1.00	1,836,000.00	489,600.00
95%	0.64	1,175,040.00	313,344.00
96%	0.55	1,009,800.00	269,280.00
97%	0.47	862,920.00	230,112.00
98%	0.37	679,320.00	181,152.00
99%	0.25	459,000.00	122,400.00
99.20%	0.22	403,920.00	107,712.00
99.40%	0.19	348,840.00	93,024.00
99.60%	0.16	293,760.00	78,336.00
99.80%	0.12	220,320.00	58,752.00
99.90%	0.093	170,748.00	45,532.80
99.92%	0.087	159,732.00	42,595.20
99.94%	0.080	146,880.00	39,168.00
99.95%	0.077	141,372.00	37,699.20

Now, with the values of lifetime (L), is obtained two graphs taking into account the climatic conditions of winter and summer.

Then, was made a logarithmic adjustment to these graphs to find an equation that better fit this curve, because the correction factor values cannot be interpolated; with this equation, it is possible to get a better trend to estimate and predict the behaviour of the curve to increase or decrease the reliability. It should be mentioned also that this logarithmic fit to the curve was approximately 98%.

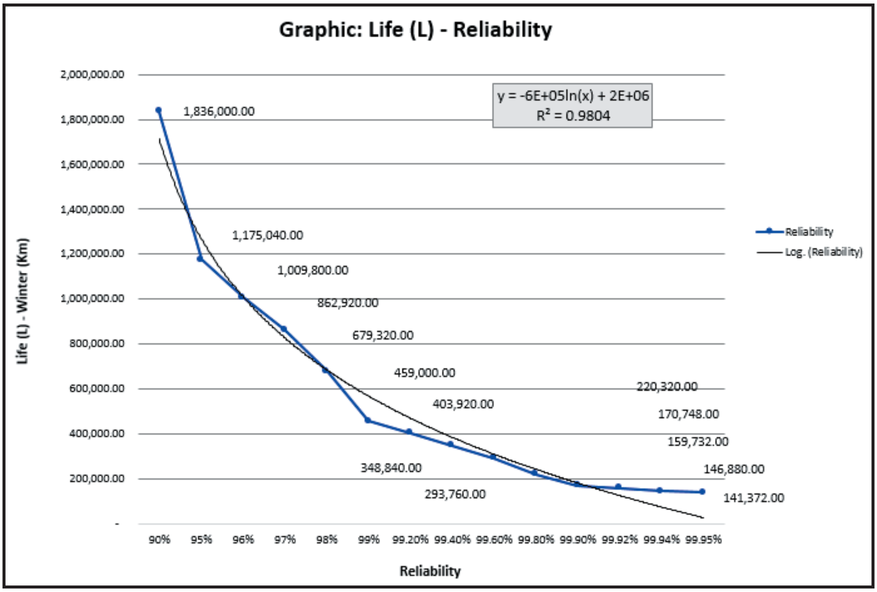


Figure 89. Chart of bearing life (L) as a function of the reliability for the conditions winter climatic.

The graph in figure 89 shows a decreasing trend with increasing reliability. This is logical because when the operating period is shorter, the reliability is higher.

Furthermore, in winter weather after 1,836,000 km, the bearing still has a reliability of 90%. This situation could be considered favourable from a maintenance standpoint, because the bearing lifetime is at the maximum, with a prolonged performance time. This also can be unfavourable, however, for railways that travel great distances, as “the bearings under the specific use conditions, possess superior failure probability to that specified by the manufacturer”.

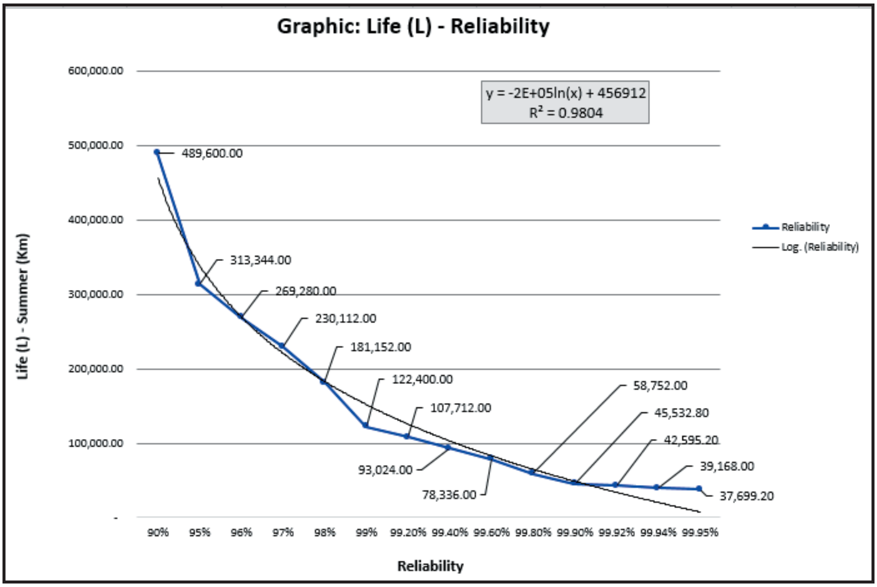


Figure 90. Chart of bearing life (L) as a function of the reliability in summer conditions.

The graph in figure 90 also has a digressing trend, but for summer the lifetime (L) of the bearings is less and its reliability diminishes for shorter operating times. In summer, the high temperatures diluted the oil (make it less viscous) and smooth the grease, restricting the lubricating ability. In addition, the normal operating temperature can be increased when heat is transferred by conduction from the shaft to the bearing or arrives by radiation in very hot environments.

Figure 90 also indicates that for a reliability of 90%, the bearing has only been working 489,600 km; this is just 25%, quite different from what happens in winter. The large decrease of the bearing lifetime (L) is consequence of what is mentioned above; exposing the lubricant to high temperatures for long periods under constant agitation and in contact with air will cause the oil to progressively oxidise; deposits from the oxidation of oil or grease can cause blockages in the oil passages and create a wedge between the rolling elements and spacers, thus interfering with the free movement of parts.

In order to solve this problem can be posed as solution the use in summer of more viscous lubricants or any special high temperature grease to improve this behaviour.

8.2. Comparing the bearing life (L_{10}) values.

Initially is compared the values of the bearing life (L_{10}) provided by LKAB-Kiruna Company with those obtained using Methodology I. The results are summarized below.

Note: As mentioned previously, the bearing life analysis was performed specifically for a class K bearing (TIMKEN).

Table 15. Results obtained.

The bearing life (L_{10})	
LKAB-Kiruna Company	Methodology I
803.000 Km	612.000 Km

As shown in table 15, the results differ, possibly because Methodology I considers several loadings applied to the bearings (it is a more refined procedure inspired by an approach used to design rail shafts). LKAB Company adopted the SKF and TIMKEN approach because of its common use in the industrial sector. But this type of approach has a drawback, namely, the lack of rigour in defining the terms associated with the mathematical model used to estimate L_{10}. Thus, Methodology I include each load on the axle, both the static loading and the dynamic loadings generated by the vibratory response of the wagon from variations in the wagon speed and rail irregularities.

Now, in the absence of previous experience, the guideline values listed in table 6 can be used. If is compared these values with results, in principle, is could say that the information about bearing life (calculated by LKAB company) coincides with these, because the methodology used in both cases was an established technique found in a manufacturer's manual for bearings (SKF or Timken).

UIC (International Union of Railways) sets a value of 800,000 km for freight wagons and railway vehicles; its specification is based on the continuous action of maximum axle load, i.e., the case study, and this value is similar to the results obtained by LKAB Company.

But if is compared the approaches (SKF and Timken) with Methodology I, is could say that the differences between the lifetimes (L_{10}) obtained usually depend on the type of machine and the requirements for duration of service and operational reliability. In some cases, the bearing size and the operations conditions of the machine (loads, kilometres travelled, working period, velocity, weather conditions, among others) have an influence as well.

8.3. Analysis of the nominal life value (L_{10}) obtained for the axlebox.

Table 16. Results obtained.

Nominal Life (L_{10})	$L_{10} = 611921{,}170\,\text{Km} \approx 612.000\ \text{Km}$

Initially, as shown in table 16, the nominal life value (L_{10}) obtained for bearings in the axle box is very close to the nominal life value (L_{10}) estimated for bearings (SKF and Timken CTBU AP-2^{TM}) for several reasons:

- When eight bearings per axle instead of four are considered, the dynamic loads (F_a and F_r) axial and radial respectively, together with the equivalent dynamic load (P) acting on the bearings is halved, but this does not significantly alter the value of the mean equivalent dynamic load (F_m), since generally, the dynamic forces can be the result of errors of the wheels and unbalanced rotating components; in addition, the require-ments for quiet running, axlebox are made to such a high level of accuracy that these forces are generally negligible, and not considered when making bearing calculations (in accordance with manufacturers). Moreover, only F_m is affected by the value of the static load, which is proportional to the mass of the wagon (empty or full) and the number of axles per wagon, taking into account that this methodology is based on a design of railway shafts. Is should consider that additional loads arising from the type and mode of operation of the machines can only be determined when the operating conditions are known. Their influence on the rating lives of the bearings is determined using an "operation" factor that takes shock loads.

- Another important point to consider is that the axle box saves around 20 kg per axle box or 160 kg for a four axle wagon. It simultaneously reduces the unsprung weight of the wheel set by around 4 %, contributing to the running performance and to reducing wheel and rail wear. Thus, it is normal that the nominal life (L_{10}) of all bearings in the wagon are similar, as shown in the obtained results. In simple terms, the axle box reduces the weight not supported by the suspension in the wheel and helps lighten the weight of the wagon; this, in turn, contributes to better performance of the bearings.

- Finally, if is compared the values of the bearing lifetime (L) (listed in table 14) with the values in table 6, that establish guideline values of life for axlebox bearings, is see that for an operating period of about 800.000 km (0.8 million km), the bearing has a reliability between 97% and 98%; this is considered favourable from a maintenance standpoint, because the replacement periods could be extended, making the most of the useful life of the bearings. But the calculated lifetime (L), although correctly done, is only one part of estimating the practical life of a bearing. The moderately loaded bearings may work for a long time even though signs of fatigue have appeared. Matters like wear and the failure causes mentioned above must also be considered. Too long a calculated lifetime can even shorten the practical life of a bearing.

9 CONCLUSIONS

The following conclusions can be derived from the research:

1. The methodology used by manufacturers to estimate L_{10} is well-established. However, methodology I has proved to be a more flexible and refined procedure, within an operational context, since it allows control of the parameters that are directly involved in the phenomenon of insufficient support (static loads, speed, dynamic loading, and the vibrational response of the structure).

2. Although the methodology proposed by the bearing manufacturer used by LKAB to estimate L_{10} is based on a standard procedure, it can be considered a generic approach. This can be demonstrated by comparing the two methodologies; between L_{10} = 803,000 km (LKAB - manufacturers SKF and Timken) and L_{10} = 612.000 km (methodology I) there is a difference of 191,000 km. Is could conclude that the approach based on the design of axles (methodology I) can be used for different types of trains (wagons), especially when we want to obtain a better approximation of the lifetime. Using this methodology, is could restructure, schedule and establish maintenance activities that generate large benefits in the industry.

3. Graphs showing the lifetime (L) as a function of reliability have a decreasing tendency, i.e., for greater reliability, the lifetime is shorter.

4. Because in summer the high temperatures dilute the oil and smooth the grease, the normal operating temperature can be increased, possibly due to the heat transferred by conduction

from the shaft to the bearing or arriving by radiation in very hot environments. Therefore, in summer, the lifetime (L) of the bearings is less and their reliability diminishes in fairly short operating times.

5. The axle box saves around 20 kg per axle box or 160 kg for a four axle wagon. It simultaneously reduces the unsprung weight of the wheel set by around 4 %, contributing to the running performance and to reducing wheel and rail wear. In simple terms, the axle box reduces weight not supported by the suspension in the wheel and helps lighten the weight of the wagon; this, in turn, contributes to better performance of the bearings.

6. The systems of condition monitoring that are currently being used to monitor bearings are the RailBAM© (acoustic monitoring), FLIR© (thermography). But the approaches of vibration analysis and thermography remain conceptual and require much more research. However, companies like SKF have developed condition monitoring technology that supports the railway industry, and this meets the increasingly stringent reliability and safety requirements while contributing to savings in the cost of maintenance, thanks to the ability to better predict failure of key components at an early stage when remedial work can be planned efficiently.

7. Condition monitoring units are another option in the technology of the predictive maintenance of bearings and can be used in conjunction with smart bearing sensors.

8. Reconditioning bearings allows companies to reduce production downtime, save costs, reduce the amount of waste and contribute to sustainability.

9. A good reconditioning process can improve the lifetime factor for a reworked bearing. Depending on the extent of

refurbishment, this factor of life may vary the life of a new bearing from 87% to 99%.

10. The Duroc company although it does not perform complete reconditioning can be considered to have a high quality repair program because it establishes detailed explanations of the scope of the work to be performed, has significant experience with bearings, replaces any component using established standards, offers outstanding service, a good warranty and the service to back it, and features products that matches the performance of a new bearing.

11. The current distribution over a train is such that the bearings in the last part of the train tend to take more current than the bearings in the first part.

12. Bearing damage can occur depending on the electrical regime, the bearing impedance and its tribological behaviour in the rolling contact.

13. The passing of electrical current through the bearings is characterised by very high temperatures similar to that produced by the melting and the welding processes. Craters are typical damage and in a later stage, fluting or washboarding occur as a secondary effect.

10

FURTHER RESEARCH

The results obtained during this work have been satisfactory in the estimation of the nominal life L_{10} . Methodology I turned out to be an approach that could be very specific in determining the life of a bearing. We can say it is a method with a refined operating context since it allows control of the following parameters: wagon weight, centre of gravity of the wagon, static loads, dynamic loads, railway speed, and the vibrational response of the structure. However, the following tasks should be developed in future work:

- One of interesting point would be the study of L_{10} (using methodology I) in other types of trains (wagons); we could study the influence of different characteristics of train, and determine whether methodology I is applicable elsewhere.

- Something that would be interesting which could generate significant results, is considered using more viscous lubricants in summer. This way would be conducted a study in order to observe the influence of the lubricant on bearing life. Of being positive these results would be extended the bearing lifetime and therefore would improve the replacement periods, bringing benefits to rail industry.

- Another point to be developed in the future is a stricter comparison; using the methods presented we could evaluate some parameters to quantify the lifetime (L). Reliability may be estimated using the Weibull distribution, and then compared with the reliability shown in table 14; this would give a more accurate estimate and we would have a better understanding of the life of the bearings following reliability criteria. We could also find a reliability that achieves better estimations than that presented in this thesis.

- Another suggestion is to extend the results obtained by estimating the useful life (L) of the axle box bearings, with a view to estimating its reliability in different working conditions, periods of replacements; we could also study in more detail the influence it has on the life of bearings CTBU or AP-2™.

- Work that would be very useful and would complement the present research is to measure the electric current in these bearings in real operating conditions. This would allow us to control this variable, to avoid any fault it might generate, especially unscheduled stops. It could be of great utility, because it would permit a detailed report of the measurement process, as in the reconditioning process conducted by Duroc.

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A.1. Data from the railroad cars supplied by the MTBA (i Kiruna AB Malmtrafik).

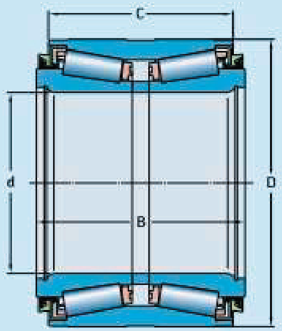
Generally	
Vehicle Profile	SJ-F (disp) and NSB U
Operating temperature	+30 °C to -40 °C
Transport speed up to	70 km/h
Transport speed laden	60 km/h
Release speed	0,25 m/s to 0,6 m/s
Last-/lossning terminal time	50 min
Technical lifetime frame	25-30 years
Technical lifetime Basket	10-15 years
Bogie	15-20 years
Mileage about	160.000 km/year
Curve radius coupled	$\leq$ 150m low speed
Curvature radius coupled	$\leq$ 290m in speed 70km/h
Curve radius disconnect / disassembly	$\leq$ 250m
Loads	
Gross Combination Weight (GCW)	8432 ton
Axle load max	31 ton
Pellet density	2,1 ton/m^3
Pellet weight per wagon	105 ton
Total weight basket (empty)	3,5 ton
Total weight basket (full)	108,5 ton
Total weight of carriage	18 ton
Tara weight (empty weight) (M_v) = Total weight basket + Total weight of carriage	21,5 ton
Full weight (M_c) = Pellet weight + Tara weight	126,5 ton

The trailer intended first care for: Pellets KBF MAF MAC		
The trailer intended second hand: Olivine Specialty products		
Basket		
Load Volume min		50 m^3
Frame height		3600 mm Basket Over SMOKE
Side height Basket		3800 mm Over SMOKE
Vehicle width max		3500mm
Angle of repose gable min		45°
Repose side		50°
Number of shafts per wagon, N_e		4
Frame		
Bogie spacing		6690 mm
Koppel Height		1040 mm
Koppel Distance		10,3 m
Distance adjacent shaft		1778 mm (minimum same as in the bogie)
Distance between the bearing, j		2,006 m
Bearings per axle, N_r		4
Clearance over SMOKE		130mm (minimum)
Wheel diameter	max Ø920 mm	
	min Ø860 mm	
Brake		
Mains		5 bar
Main Container		5,6 bar
P-brake loaded wagon		30% Slope (applied force 50 kg maximum) ??Nm

Track layout	
Luleå-Kiruna-Narvik Råtsi-Svapavara Gällivare-Vitåfors	
Standards	
SS-EN 12663 BSK-99	

A.2. Catalog of SKF. Type of bearing used [7].

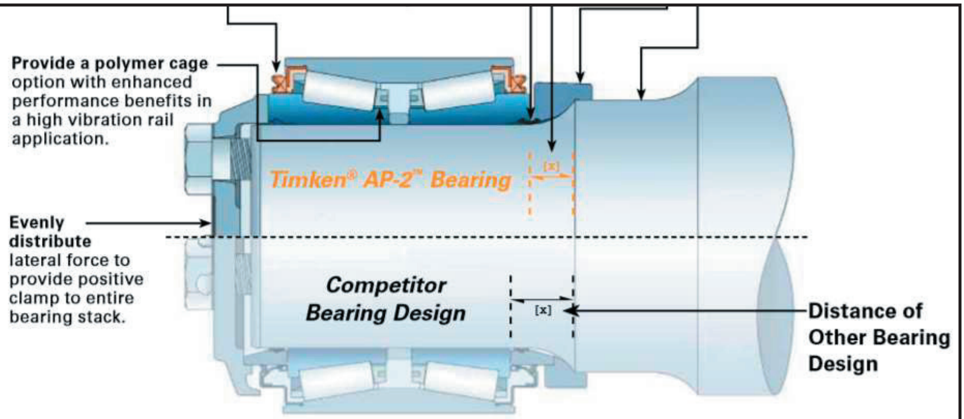
CTBU compact tapered roller bearing units



Size	Principal dimensions			
	d	D	B	C
Metric sizes	mm			
CTBU 100 x 175	100	175	130	120
CTBU 110 x 180	110	180	142	142
CTBU 130 x 210	130	210	148	132
CTBU 130 x 220	130	220	145	135
CTBU 130 x 230	130	230	166	150
	130	230	176	160
CTBU 130 x 240	130	240	172	160
CTBU 130 x 250	130	250	172	160
CTBU 140 x 220	140	220	150	140
CTBU 150 x 250	150	250	180	160
CTBU 160 x 280	160	280	195	180
Inch sizes	mm *(in)*	-	-	-
CTBU Class K 6 1/2" x 9"	157 *(6.1880)*	250 *(9.8420)*	181 *(7.1102)*	160 *(6.2992)*
CTBU Class G 7" x 9" and 7" x 12"	178 *(7.0005)*	276 *(10.8750)*	200 *(7.8740)*	186 *(7.3120)*

These figures are for information only. Contact SKF for detailed product specifications.

A.3. Catalogue of TIMKEN. Type of bearing used [12].



Weight Savings Comparison:
Timken® AP™ Bearing vs. Timken® AP-2™ Bearing

AP-2 Class K – 6½ x 9 in. (165.1 mm x 228.6 mm) for 286,000 lbs. (130,000 kg) Gross Rail Load Cars				
	Bearings (2)	Adapters (2)	Axle	Total
Class F Shrouded	223.5 lbs. (101.4 kg)	70.5 lbs. (32.0 kg)	1175.0 lbs. (533.0 kg)	1469.0 lbs. (666.3 kg)
Class K	178.0 lbs. (80.7 kg)	64.0 lbs. (29.0 kg)	1168.0 lbs. (529.8 kg)	1410.0 lbs. (40.0 kg)
Savings Per Wheelset				59.0 lbs. (26.8 kg)
Savings Per Car				236.0 lbs. (101.7 kg)

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A.4. Annexes.

Table 17. Bearing manufacturers for the railway.

Bearing Manufacturers for the Railway		
Manufacturer Name	**Bearing Type**	**Advantages and Benefits**
<u>**SKF**</u>	CTBU Class K	• Labyrinth-Lip (LL) seal: (i) Improved protection against contaminants. (ii) Longer grease life. (iii) Better and longer performance. • Energy Efficient (LL E2) seal: (i) Lower energy consumption of hauled rail vehicles because of a lower frictional moment compared to an LL seal, (ii) Reduced seal wear rate, better performance and longer maintenance intervals achievable, (iii) Better thermal conductivity, lower bearing operating temperature and longer grease life, longer maintenance intervals. • Polymer spacer: (i) Fretting corrosion avoided, (ii) Lower wear rate of the inner ring side face/backing ring contact zone, (ii) Longer performance and longer maintenance intervals achievable because of longer grease life. • Polymer cage: (i) Reduced friction and roller slip, reduced wear and lower operating temperatura, (ii) Improved safety and performance, (iii) Safe failure mode without seizing. • For high-speed and very high-speed trains, SKF offers a TBU design equipped with a labyrinth seal on both sides. For some selected applications, a special contacting seal like the LL E2 seal is used. • The TBU designs can be equipped with impulse wheels and sensors to detect operating parameters.

TIMKEN	AP-2™	• Reduce axle fillet damage by minimizing the potential for water ingress and resulting corrosion. • Reduce the likelihood of bearing damage due to increased contaminant exclusion. • Reduce bearing set outs by lowering seal operating temperature. • Reduce fuel cost due to low operating torque. • Provide a polymer cage option with enhanced performance benefits in a high vibration rail application. • Evenly distribute lateral force to provide positive clamp to entire bearing stack. • Reduce component wear rejection. The Timken design provides the shortest distance between the cone face and the dust guard. This design reduces the amount of movement and the resultant wear on the cone back face. • Unitize entire bearing assembly and backing ring to aid in installation procedure. • Decrease potential axle failure. The shorter axle journal design provides a longer and stiffer dust guard, which, in turn, reduces stress at the crucial axle fillet area. • Eliminate axle grooving. By removing the seal wear ring in the Timken design, axle grooving – and the resulting scrapping or expensive repairs of the axle – can be eliminated.
NTN-SRN	JT21	• Tapered roller bearings are commonly used for their ability to endure high moment loads. • They are able to support high thrust loads despite their discreet profile and are relatively inexpensive. • Face-to-face mounted bearings are seldom used today because they have low rigidity when a moment load is applied. • These are sealed tapered roller bearings used for axles of rolling stock. • Prelubricated with grease. • They are provided with a hermetic seal, which

		permits a simplified axle box structure. • They are designed and built to provide ease of use and long life.
NSK MOTION & CONTROL NSK	Type A – Class 150	• It can carry radial and axial loads simultaneously and therefore permits compact design of the bearing and its adjacent parts. • Ir requires precise internal clearance adjustment in order to perform properly. • For rolling stock axle applications where heavy moment loads are expected, the back-to-back arrangement provides a greater distance between load centres. • Improvements in surface roughness and contact geometry have virtually eliminated the friction problems associated with tapered roller bearings for axles. • This type of axle bearing can be designed with a sealed arrangement between the rear cover and the bearing box; it can have an internally sealed construction.
FAG FAG	TAROL – Class K	• Polyamide Cages: These have several advantages, ranging from low mass, longer grease life and excellent emergency running characteristics to longer bearing rating life, lower friction and lower running noise levels.

Table 18. Differences between the various bearing manufacturers (depending on type).

Differences between the various manufacturers
1. SKF, TIMKEN and FAG offer a longer grease life, while NTN-SNR is pre-lubricated with grease (the long grease life isn't guaranteed).
2. SKF and TIMKEN seals design offers improved protection against contaminants in comparison with NTN-SRN, SNK and FAG.
3. SKF and TIMKEN are equipped with "Polymer cages" but these have different functions.
4. NSK and NTN-SRN bearings are designed to endure high moment loads, i.e., their dynamic and static loads are higher than SKF and TIMKEN.
5. TIMKEN offers the shortest distance between the cone face and the dust guard, reduces the amount of movement and reduces the resultant wear on the cone back face. This design is an innovation in the new model TIMKEN AP-2™ bearing.
6. The TIMKEN AP-2™ design provides a polymer cage option with enhanced performance benefits in a high vibration rail application. In other words, unlike other manufacturers (SKF, NSK, NTN-SNR, and FAG), its design is suitable to bear high vibrations.
7. One of the more important SKF features is that its CTBU designs can be equipped with wheel impulse and sensors to detect operating parameters. Only this manufacturer offers this advantage.
8. NTN-SRN bearings are able to support high thrust loads despite their discreet profile. Furthermore, they are not as expensive as other manufacturers.

9. NTN-SRN provides a hermetic seal, but SKF improves it through the Labyrinth-Lip seal (LL) and Energy Efficient seal (LL E2).

10. SKF CTBU "Polimer Spacer" design avoids fretting corrosion and provoides lower wear rate of the inner ring side face/backing ring contact zone; i.e., offers a protection against fretting corrosion during more than 800,000 km of working.

11. SKF CTBU's sealed system can reduce the frictional moment by up to 75% compared to the garter seal arrangement.

12. A common feature for all these bearing manufacturers (SKF, TIMKEN, NTN-SRN, NSK, FAG) is the back-to-back arrangement, which provides a greater distance between load centres. This lets the bearings work properly in rolling stock axles for heavy moment load applications.

13. FAG can also supply polyamide cages individually as replacement parts. This replacement is very difficult to find in other manufacturers.

14. FAG bearings with "Polyamide Cages" offer several advantages; they have excellent emergency running characteristics, longer bearing rating life and lower running noise levels. These features are not found in NTN-SNR and SNK.