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Rolling Bearings in Bulk Conveyors

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7 & 8 September 1987
Sandton Holiday Inn
Sandton

The S.A. Institute of Materials Handling
The S.A. Institution of Mechanical Engineers

PAPER FOR INTERNATIONAL MATERIALS HANDLING CONFERENCE 1987

7 - 8 SEPTEMBER 1987

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ROLLING BEARINGS FOR BULK CONVEYORS

SYNOPSIS

Rolling bearings are an essential part of bulk conveyors supporting both the idlers and the pulleys.

For idlers the selection of bearing type and size depends apart from cost on service life as influenced by load, speed, lubrication, misalignment and contamination. Furthermore ease of mounting and bearing friction are important.

The bearing selection is also influenced by the dimensional quality of the roll and shaft and the seals used. As support bearings for conveyor pulleys, generally spherical roller bearings are used because of their insensitivity to misalignment and high radial and thrust load capacity.

Housing and seal design has to be adapted to the hostility of the environment. A special seal arrangement has been developed for the most severe operating conditions.

1) INTRODUCTION

Rolling bearings for conveyors are usually selected on basic dynamic load rating values, i.e. minimum required rolling contact fatigue life.

However, basic dynamic load ratings are applicable for clean, well lubricated and perfectly aligned bearings only.

In practice, quite large discrepancies are found between calculated fatigue life and actual service life.

In this paper different influence factors on bearing service life are discussed, and where possible, quantified.

2) ROLLING CONTACT FATIGUE LIFE AND FAILURE INTENSITY

The ultimate life of a well lubricated, correctly mounted and sealed rolling bearing is determined by the first sign of rolling contact fatigue on one of the rings or rolling elements. The basic rating life L_{10} is defined as the life that 90% of a sufficiently large group of apparently identical bearings under identical operating conditions can be expected to attain or exceed.

Therefore, there is a possibility that bearing failures occur before the calculated L_{10} life.

The relation between basic rating life, basic dynamic load rating and the bearing load can be expressed as

$$L_{10} = \left(\frac{C}{P}\right)^p \text{ where} \qquad (1)$$

L_{10} is basic rating life in millions of revolutions
C is basic dynamic load rating in N
P is equivalent dynamic bearing load in N
p is the exponent for life equation : $p = 3$ for ball bearings
$p = {}^{10}/_{3}$ for roller bearings

For bearings operating at a constant speed the basic rating life in hours can be calculated as follows :

$$L_{10h} = \frac{10^6}{60n}\left(\frac{C}{P}\right)^p \qquad (2)$$

Conveyor idler bearings are usually chosen to give a calculated life between 20 000 and 50 000 hrs (Ref 1).

For pulley support bearings basic rating life in the range between 40 000 and 50 000 hours are generally required. Although these lives seem comfortably long, the failure intensity, or the number of failed bearings per unit time can be quite important from installation maintenance and reliability point of view. This is especially so for long conveyor belts with a great number of idlers.

The failure intensity for bearings due to rolling contact fatigue can be calculated as follows (Ref 2)

$$Z(L) = \frac{0{,}105}{L_{10h}} \left(\frac{L}{L_{10h}}\right)^{\beta-1} \qquad (3)$$

where $Z(L)$ = failure intensity at Life L
β = dispersion coefficient

For rolling contact fatigue the dispersion coefficient β is approximately 1,11.

With sufficient accuracy the bearing failure intensity can then be given as

$$Z(L) = \frac{0{,}105}{L_{10h}} \qquad (4)$$

3) CONVEYOR IDLERS

3.1 Conveyor Idler Loads

The load on the idler set can be calculated as follows (Ref 3)

$$W_{tot} = \emptyset\, W_m + W_b \qquad (5)$$

where W_{tot} = total load on the idler set (N)
W_m = force due to material transported (N)
W_b = force due to belt mass (N)

$\emptyset$ = dynamic load factor

= $C_o V^2 + 1$

V = belt speed

C_o = factor depending on material transported

Measurements (Ref 3) gave following values for C_o :

Type of material	Fixed idlers	Suspended idlers
fine grained	0	0
individual chips	0,005	0
coarse chips on cushioning layer	0,009	0,005
coarse chips	0,014	0,009
lumps up to 100 kg	0,05	0,02

The load due to material mass can be calculated :

$$W_m = \frac{Q\ a}{0{,}36V} \qquad (6)$$

where Q = mass flow ($^t/_h$)

a = idler spacing (m)

The centre roll takes the largest part of the load W_{tot} on the idler set.

Depending on the troughing angle, volumetric loading, and conveyor incline, the centre roll of a three roll idler takes a minimum of 50% of the total load on the idler set, typical values are 60 - 70% (Ref 3) for a troughing angle of 35°. The radial load on the side rolls is typically 20 - 25% of W_{tot}.

For a five roll idler the centre rolls usually take 45% of the total load.

Although actual mass transport is often lower than the maximum capacity of the belt, the maximum values are used for calculating the fatigue life of the centre roll.

For perfectly aligned, straight horizontal belts the individual bearing load can then be calculated as :
(See fig 1)

$$F_b = \frac{W_t + W_i}{2} \qquad (7)$$

where W_i is the weight of the rotating part of the idler.
W_t is part of the total load on the idler set taken by the individual idler roll.

Roller unbalance loading can play a role at high roller rotational speed, thus high belt speed. For the normal belt speeds up to 3,5 m/s the influence of the unbalance load on fatigue life can be neglected.

Due to friction between belt and roller a thrust load can be generated. This thrust load is usually a few percent of the radial load only and thus does not have an effect on rolling contact fatigue life.

The life of individual idler bearings can then be calculated as

$$L_{10} = 10^6 \left(\frac{C}{F_b}\right)^p \quad \text{(revolutions)} \qquad (8)$$

Since the life of a conveyor roll can be considered ended when one of the two bearings fails, the life of a complete roll can be calculated as : (Ref 2)

$$L_{10r} = 2^{-1/\beta}\, L_{10} = 0{,}537\, L_{10} \qquad (9)$$

The failure intensity of idler rolls can thus be calculated :

$$Z(L) = \frac{0{,}105}{L_{10rh}} = \frac{0{,}105}{0{,}537\, L_{10h}} = \frac{0{,}195}{L_{10h}}$$

3.2 Influence of Misalignment on Idler Bearing Life

Equations 8 and 9 apply only under ideal conditions, i.e. when bearing misalignment is small. Due to shaft deflection under load and bearing seating tolerances, considerable misalignment values are possible. Bearing misalignment influences amongst other things internal load distribution and thus fatigue life. Using advanced computer programs, the influence of misalignment on nominal life can be calculated as influenced by, for instance, internal geometry, clearance and load.

In fig 2 the results of calculations for a 6205 deep groove ball bearing are given. It can be seen that for a 6205 with zero internal clearance after mounting, a misalignment of up to 2 mrad (7') does not have a detrimental effect on calculated bearing life. For a bearing with C3 clearance, giving an average clearance after mounting of approximately 12,5 μm, misalignments up to 4 mrad can be accepted.

For larger misalignment values, calculated life rapidly decreases and can become several orders of magnitude smaller than the basic rating life.

Taper roller bearings have line contact between rollers and raceways and misalignment does not only influence the load distribution amongst the rollers, but also the pressure distribution over the roller length (fig 3).

The roller and raceway profiles have a decisive effect on the occurrence of edge stresses. With a so-called "logarithmic" roller or raceway profile, the occurrence of edge stresses can be minimised, without loss of fatigue life under normal operating conditions.

SKF has developed the manufacturing method for "logarithmic" profile, and several bearing types are already available with this profile on either the rollers or raceways.

In fig 4 the effect of misalignment on calculated fatigue life of a K-LM 11949/KL-M 11910 VQ 051 (logarithmic profile on inner ring raceway) at a radial load of 1000 N and axial clearance of 0 and 25 µm respectively is given.

It can be seen that at zero clearance misalignment values up to approximately 2 mrad (7') and at 25 µm axial clearance values up to 4 mrad (14') do not have a detrimental effect on rolling contact fatigue life. For bearings with normal straight profile, however, misalignment values above 1 mrad are already detrimental.

Bearing misalignment due to shaft deflection can be calculated as follows : (fig 1)

$$\alpha = \frac{(l_r - 2x)a(W_t + W_i)}{4EI} = \frac{16(l_r - 2x)a(W_t + W_i)}{Ed^4} \text{ (rad)} \qquad (10)$$

For ball bearings with C3 clearance the total misalignment should be less than 4 mrad.

The German DIN standard 22112 (ref 4) recommends a maximum misalignment due to shaft deflection of 3 mrad, allowing 1 mrad for bearing seating errors.

3.3 Other Effects of Misalignment on Idler Bearing Performance

Misalignment does not only influence bearing load distribution and life, but it also influences the kinematic behaviour of rolling elements.

For DGBB this means that the balls do run on an elliptical path, causing ball speed variations. A stiff metal cage will, depending on the ball-to-cage pocket clearance, only allow a small degree of ball speed variation without the occurrence of excessive ball-to-cage forces. A flexible non-metallic cage, however, can tolerate considerable ball position errors due to ball speed variation without large ball-to-cage forces. This is the reason that the DIN 22112 prescribes the use of polyamide 6,6 non-metallic cages in deep groove ball bearings for conveyor idlers.

Large ball-to-cage forces also negatively affects bearing friction, increase the risk for cage breakage and thus bearing blockage, and cause wiping off of the lubricant from the ball surfaces and consequently lead more rapidly to lubricant starvation.

In taper roller bearings misalignment leads to uneven pressure distribution over the roller length and consequently influences the straight rolling behaviour of the rollers causing roller skew.

In combination with contaminated lubricant, this leads to cage pocket wear allowing the cage to drop and contact the outer ring raceway. Eventually the cage can smear and wedge between rollers and raceways and cause bearing blockage.

3.4 Conveyor Idler Bearing Lubrication

The grease used should have a sufficiently high base oil viscosity to separate the moving surfaces in the bearing at the operating conditions, yet do not cause unnecessary high running friction.

For a typical idler with 25 mm bore bearing running at 500 rpm the required base oil viscosity ν_1 at the operating temperature of say 40 - 50°C is 40 mm^2/s, i.e. most general purpose lithium soap greases satisfy this requirement.

The grease should contain rust inhibitors and should have sufficiently high oil bleeding to avoid lubricant starvation in the raceway contacts. The grease should be mechanically stable and should not slump in order to avoid mixing of contaminant in the sealing area with the grease in the bearing.

The free space around the bearing should be designed such that excessive grease from the bearing can collect there and act as a reservoir. Taper roller bearings tend to pump lubricant from the small side of the taper to the large side (fig 5) leading to lubricant starvation at the small taper end. A lubricant reservoir should therefore be created on that side.

Lubricant starvation will eventually lead to bearing failure. Relubrication at regular intervals will minimise the risk of starvation and dry running, but such an exercise is for conveyor idlers expensive, and, especially for the centre roll, complicated from a design point of view. Grease nipples tend to get damaged in installations and when the nipple is not wiped clean before the grease gun is applied, more harm than good is done because the contaminant is pressed right into the bearing. Therefore, in most idlers, no provision is made for relubrication. The relubrication interval (ref 5) is based on a probability of failure due to starvation of 1%, i.e. B_1 life.

Extensive tests have shown that life dispersion is usually smaller than for fatigue life and on average dispersion coefficient of 2,5 was found.

This means that, similar to rolling contact fatigue life L_{10}, a service life B_{10} due to grease failure can be defined. For a dispersion coefficient $\beta = 2,5$ it can be shown that

$$\frac{B_{10}}{B_1} \approx 3$$

For a complete idler the B_{10r} life will then be :

$$B_{10r} = 2^{-1/\beta} B_{10} = 0,75\, B_{10} \approx 2,25\, B_1 \qquad (11)$$

8/........

Under influence of contamination, however, the service life will be further reduced, and in general, the following expression can be used : (ref 9)

$$G_{10} = a_r L_{10} \qquad (12)$$

Where L_{10} is the nominal life due to rolling contact fatigue, G_{10} is the nominal service life and a_r is a correction factor depending on operating conditions, grease quality and contamination.

Example :

A 25 mm taper roller bearing running at 700 rpm requires a relubrication interval of 4 000 hours at 70°C and 8 000 hours when the operating temperature is below 50°C (ref 5).

For the latter case a nominal grease life $B_{10h} \approx 24\ 000$ hours can be calculated, which means for an average of 5 000 operating hours per year $B_{10h} \approx 5$ years.

For the complete roll $B_{10rh} \approx 3{,}6$ years.

The failure intensity of idler rolls according to equation (3) is now a function of running time :

$$Z(L) = \frac{0{,}105}{B_{10rh}} \left(\frac{L}{B_{10rh}}\right)^{\beta - 1} \qquad (13)$$

$$= \frac{0{,}105 . 2{,}5}{3{,}6} \left(\frac{L}{3{,}6}\right)^{1{,}5}$$

In fig 6 this failure intensity is plotted as a function of running time of the installation.

As a comparison the failure intensity due to a rolling contact fatigue life L_{10rh} of 5 and 10 years respectively are drawn.

It can be seen that it does not pay to design for a very long fatigue life if the service life due to grease failure is much shorter.

In fig 7 the failure distribution for rolling contact fatigue life and grease life are drawn for the individual bearings.

For these conditions $a_r = 0{,}25$.

Deep groove ball bearings have a much longer grease life than taper roller bearings (approximately 10 times) because they do not have a pumping action. Furthermore, the roller-end rib contact of a TRB acts as a plain bearing and is often the cause for a lubricant failure.

Theoretically, for a 25 mm ball bearing at 700 rpm the required relubrication interval at an operating temperature of 70°C would be 40 000 hours (ref 5).

However, such long lives cannot be considered as accurate, especially since contamination will reduce the effective grease life.

3.5 Conveyor Idler Friction

Idler friction contributes to the drive power requirements for the conveyor belt.

In ref 6 and 7 approximate values for required belt pull to transport the belt plus charge over conveyor idlers and pulleys are given.

$$F_{pull} = \text{constant} \ (W_m + 2W_b + W_{itot}) \ N \qquad (14)$$

where N is the number of idler sets.

For long conveyors and normal conditions a value of 0,021 for the constant is given in ref 7 so that per idler set :

$$F_{pull} = 0{,}021 \ (W_m + 2W_b + W_{itot}) \qquad (15)$$

From ref 6 the belt pull required to transport the charge can be estimated as :

$$F_{pull} = (0{,}03 \cdots 0{,}05) \ W_m \qquad (16)$$

As a first approximation the friction torque of the rolling bearings can be calculated with (ref 5)

$$M = u \ (W_t + W_i) \frac{d}{2} \ (\text{Nmm}) \qquad (17)$$

with u = 0,0015 for deep groove ball bearings and
u = 0,0018 for taper roller bearings

The circumferential force at the outer diameter of the idler F_{pull} is then : (see fig 8)

$$F_{pull} = \frac{2M}{D} = \mu\,(W_t + W_i)\,\frac{d}{D} \qquad (18)$$

For a Ø 127 mm idler with Ø 25 mm bore DGBB :

$$F_{pull} = 0{,}0003\,(W_t + W_i) \qquad (19)$$

Comparing equation (19) with equations (14) and (15) it can be seen that the belt pull required to overcome all friction sources is about 100 times larger than the bearing friction alone, i.e. idler seal friction, internal friction in belt and charge etc are much larger than the rolling bearing friction.

For ball bearings, the starting friction torque is less than twice the running friction torque. For taper roller bearings the difference can be, depending on the quality of the roller end/rib contact, larger.

3.6 Tolerances and Fits

Because the outer ring rotates, it is necessary to have a tight fit to avoid creep of the outer ring in its seating. Usually an M7 fit is chosen.

For optimum bearing life, the tolerances for seating ovality and squareness of the abutment should be at least one class better, i.e. IT6. For deep drawn bearing cups welded into the idler it is often not possible to achieve such narrow tolerances. The inner ring fit to the shaft should be loose to allow for idler expansion. Since shock loads can occur and to avoid rotation of the inner ring on the shaft when friction due to ingress of contaminant increases, a slight interference can be tolerated i.e. h6 or js6.

Ball bearings can be axially located simply by means of snap rings.

For taper roller bearings, the internal clearance depends on the axial position of the inner ring, and therefore relatively accurate location is required, for instance by means of a nut on the shaft.

11/......

3.7 Conveyor Idler Seals

An effective seal, preventing ingress of contaminant and moisture is essential to obtain optimum service life.

In the majority of cases calculated life is not reached due to ineffective seals. Side rolls do fail more often than centre rolls because contaminant can settle on the seal area and migrate into the bearing during expansion and contractions at starts and stops.

Contact seals may have a better sealing efficiency than labyrinth seals but the higher friction torque, especially the starting friction prevents a widespread use.

Multi-stage labyrinth seals are mostly used. The labyrinth should be grease filled.

A "pumping" effect due to wobbling, misaligned or unround labyrinth seals can accelerate ingress of dirt, leading to an increase of seal and bearing friction and eventual bearing failure due to seizure. A seized bearing can cause substantial belt damage.

3.8 Seize Resistant Bearings

SKF has developed a special deep groove ball bearing which does not seize even when relatively large amounts of contaminant have entered the bearing.

The special features are extensively described in ref 8 of which is mentioned here :

- A flexible non-metallic cage to minimise ball-to-cage forces under misaligned conditions
- large balls to minimise effect of dirt on rolling resistance
- "thick" outer ring to reduce distortion due to unround bearing seatings
- reduced width to prevent building up of contaminant and allows smaller end cap draw
- special raceway conformity and clearance to allow maximum misalignment

12/......

Extensive tests (ref 8, 10) have shown that despite a somewhat lower basic rating life of this bearing compared to a standard deep groove ball bearing, the actual service life is in most cases longer due to its lesser sensitivity to dirt.

4) PULLEY BEARINGS

To support pulleys, spherical roller bearings (SRB) are most commonly used due to their insensitivity to misalignment and high load capacity. Basic rating lives in the range between 40 000 and 50 000 hours are generally required.

For very low speed applications, an additional check should be made to see that the load ratio C/P is at least 3,5 to avoid excessive bearing loads. Thrust loads arise from misaligned pulleys and will affect SRB life. Standard cast iron plummer block housings can be used.

A variety of seals such as labyrinth, felt, double lip and V-rings are readily available. Tests have shown that the double lip seal gives the best protection against contamination by water and dirt. The condition of the seals should be regularly inspected, and when required, seals should be replaced.

When the bearings are regreased through a grease nipple, a grease escape should be built in through the appropriate selection of seals (such as V-rings) or provision of a grease escape hole in the housing. For very high load applications housings made of spheroidal graphite cast iron can be selected.

A special extra heavy housing series, which is one piece spheroidal graphite cast iron housing, is available with a heavy duty sealing arrangement. A V-ring is combined with a patented outward pumping labyrinth seal giving optimum bearing protection. The seals can easily be adapted for oil bath lubrication.

5) CONCLUSIONS

Because of the huge quantity of conveyor idlers in operation, bearing selection should not be done on basic rating life alone. Factors such as seal efficiency, grease life and effect of eventual bearing seizure on belt damage, drive power requirements and fire hazard should be considered.

The specially developed seize resistant ball bearing has proven itself, more than 10 million bearings are installed in conveyors in South Africa, and some 60 million all over the world.

6) ACKNOWLEDGEMENT

The authors would like to thank their colleagues at SKF in Luton and Schweinfurt for the information supplied and Ab SKF for the permission to publish this paper.

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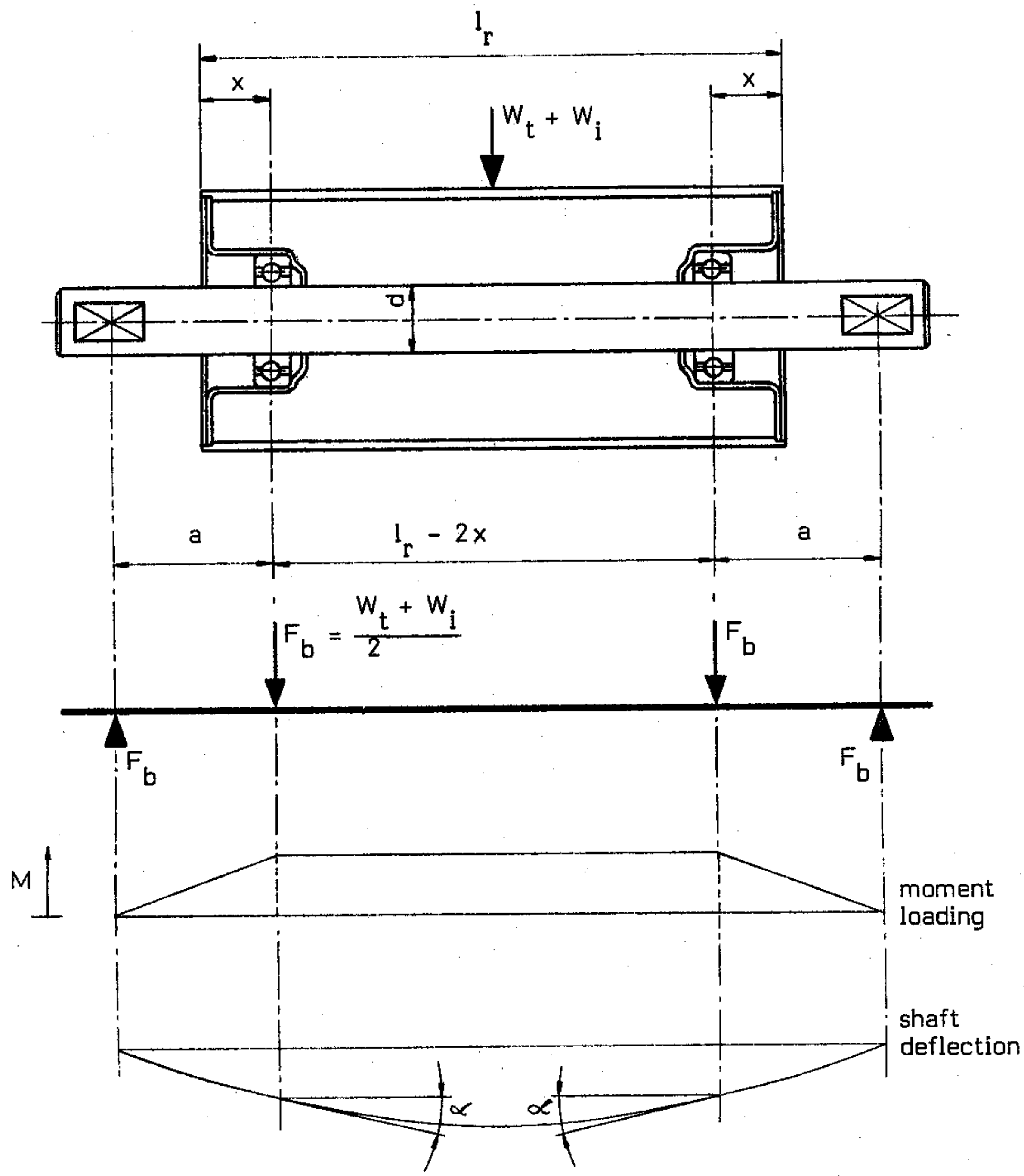

Fig 1 Loading and deflection of conveyor idler shaft

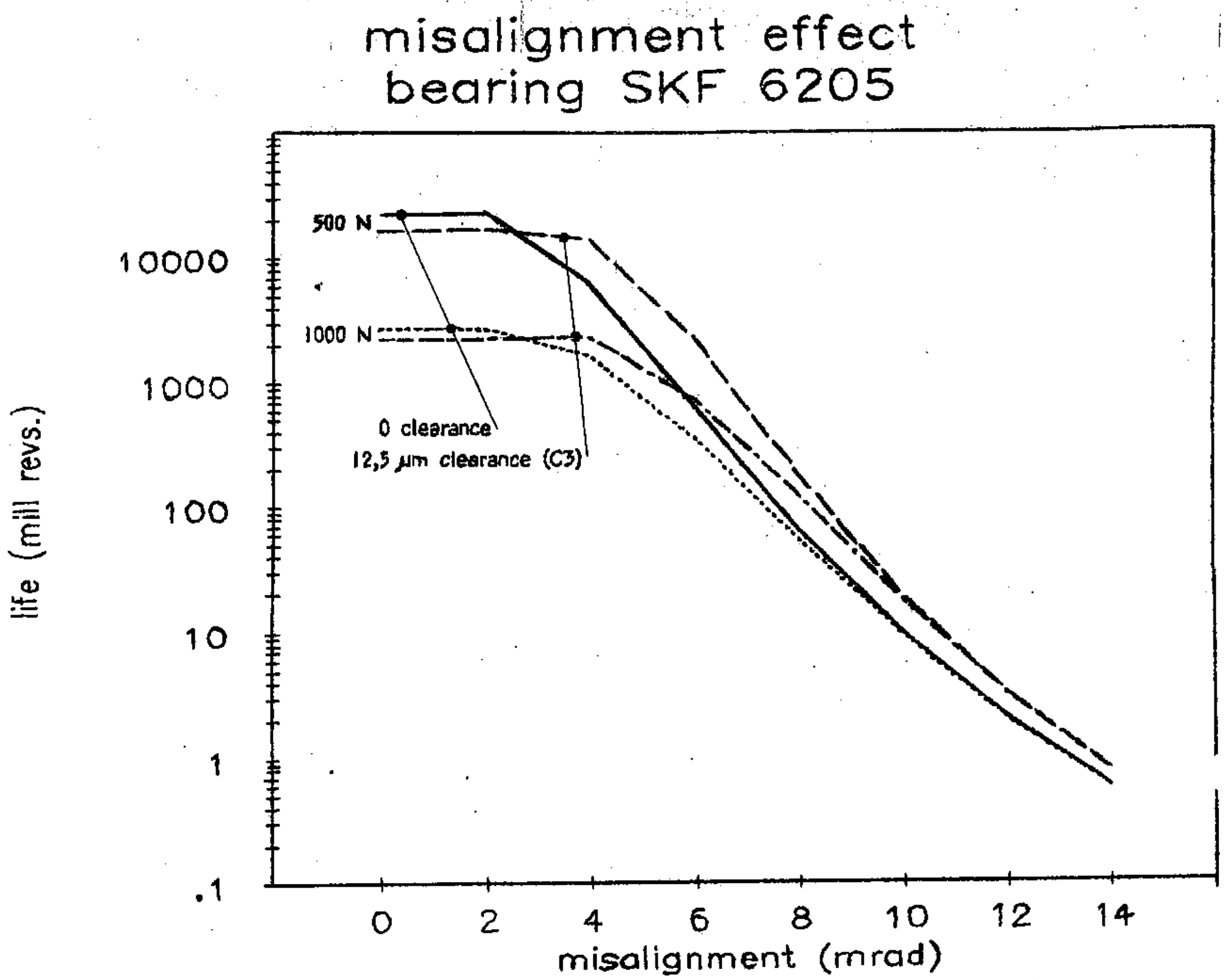

Fig 2 Influence of inner ring to outer ring misalignment on nominal life of a 6205 DGBB with respectively zero and 12,5 um radial clearance and radial loads of 500 N and 1000 N

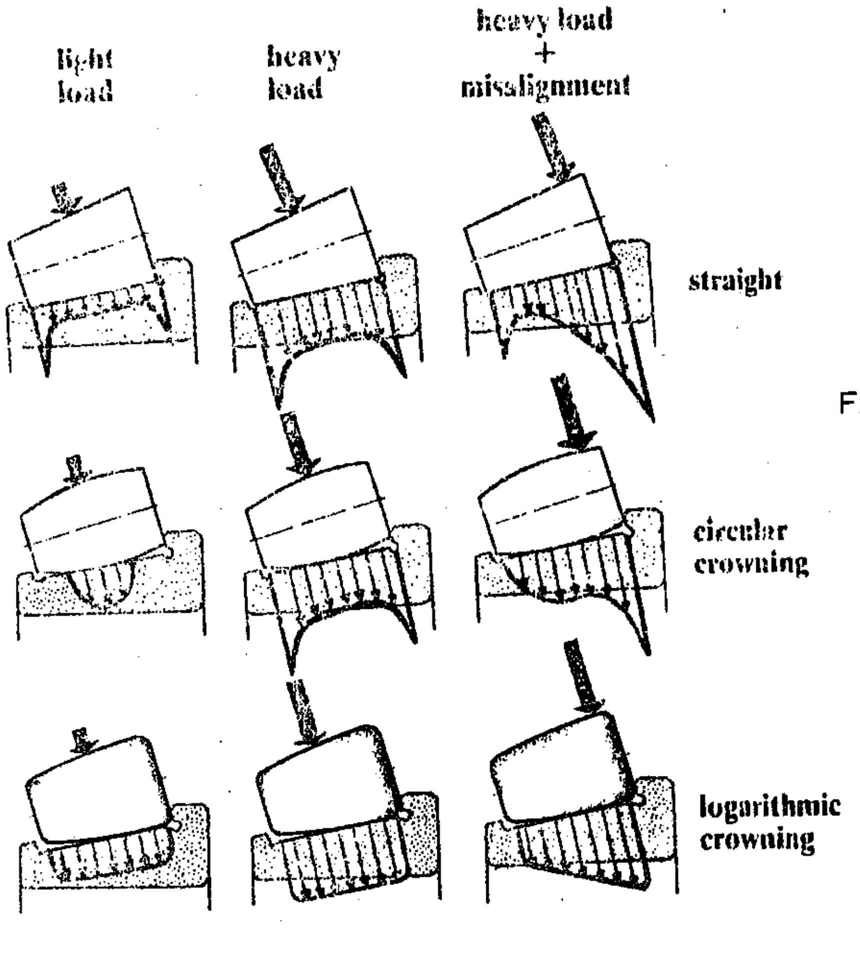

Fig 3 Influence of misalignment on pressure distribution over for a taper roller bearing

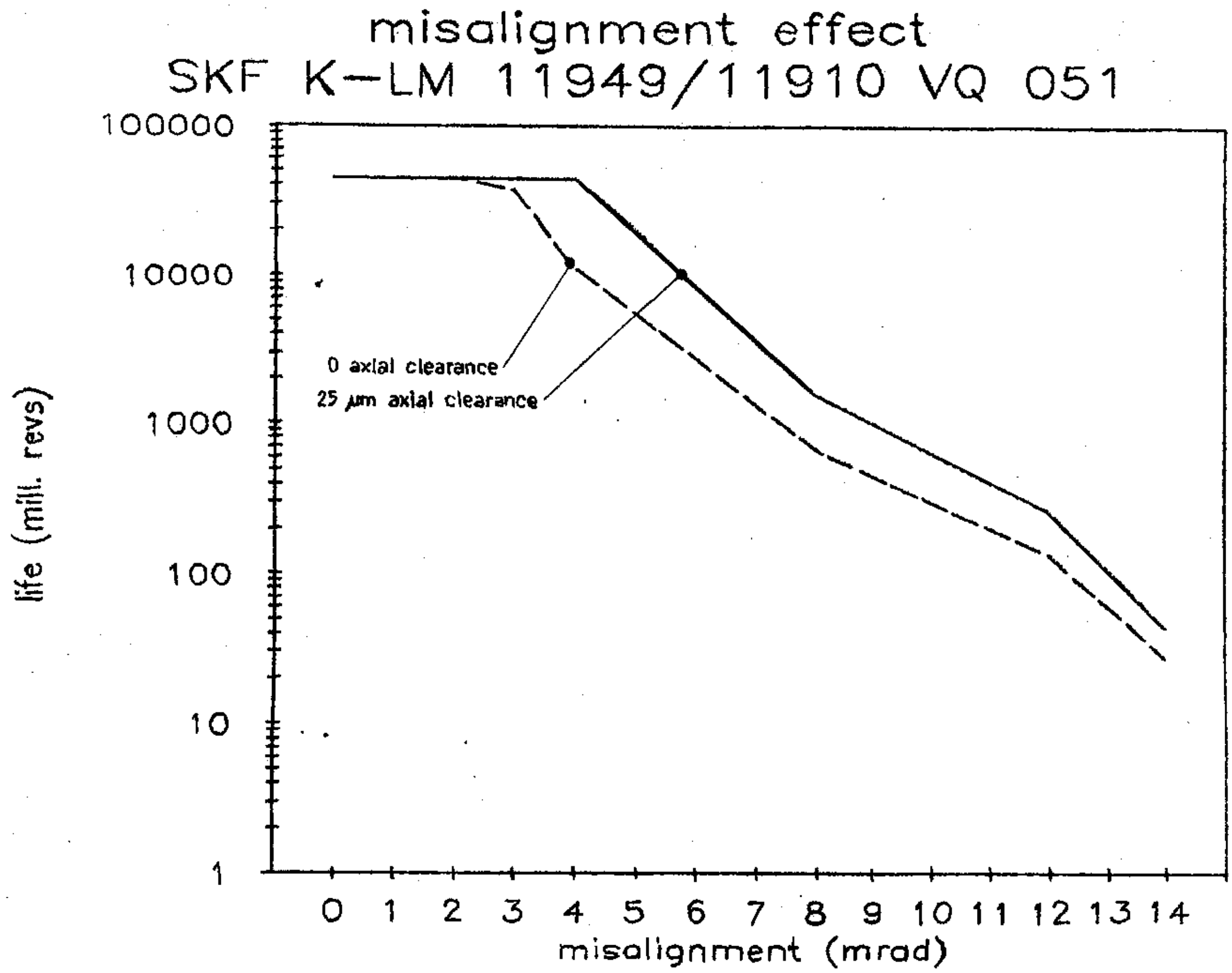

Fig 4 Influence of misalignment on nominal life of taper roller bearing K-LM 11949/K-LM 11910 VQ 051 with "logarithmic" profile on inner ring raceway. Radial bearing load = 1000 N

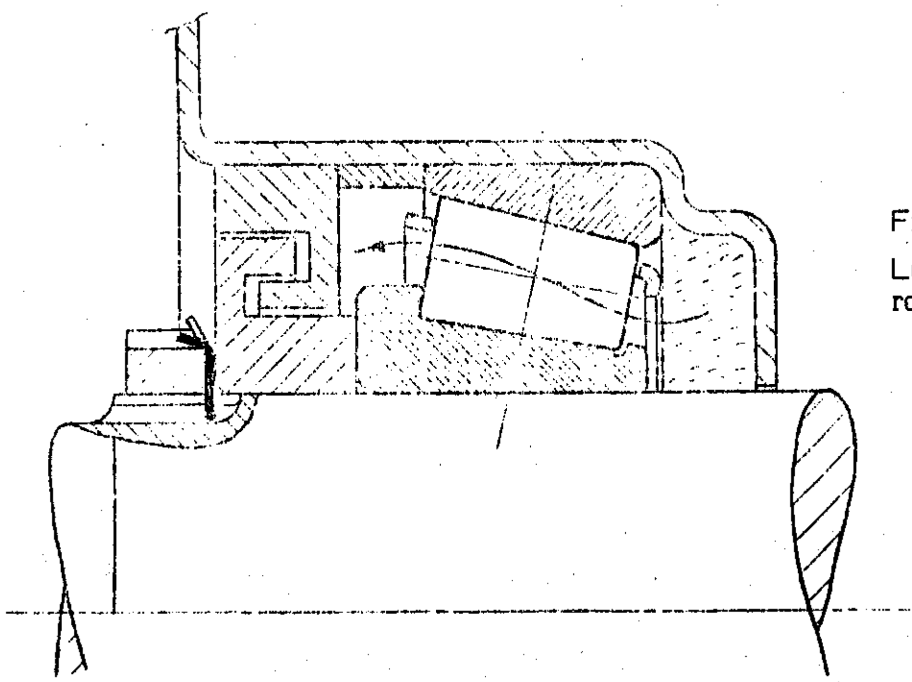

Fig 5

Lubricant flow in taper roller bearing

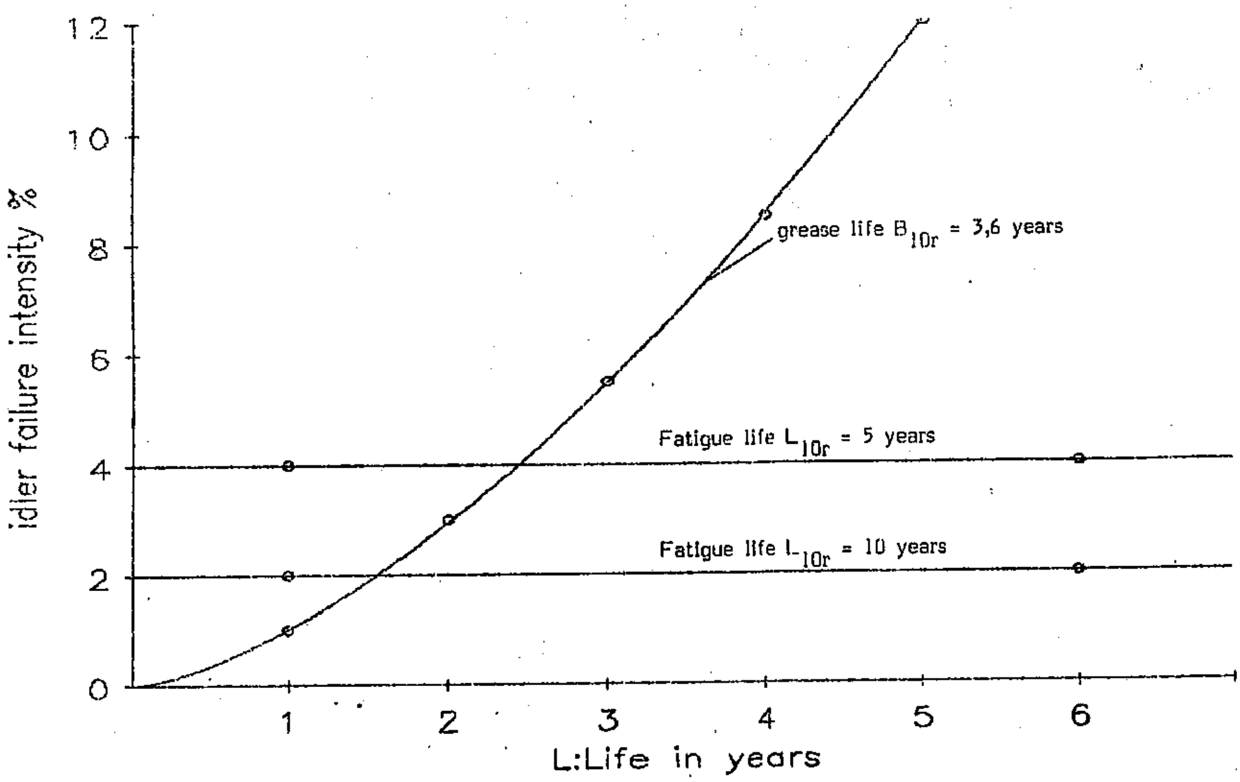

Fig 6 Failure intensity of idlers with 25 mm bore taper roller bearings due to lubricant starvation. Operating temperature below 50°C, idler speed 700 rpm. Average quality grease

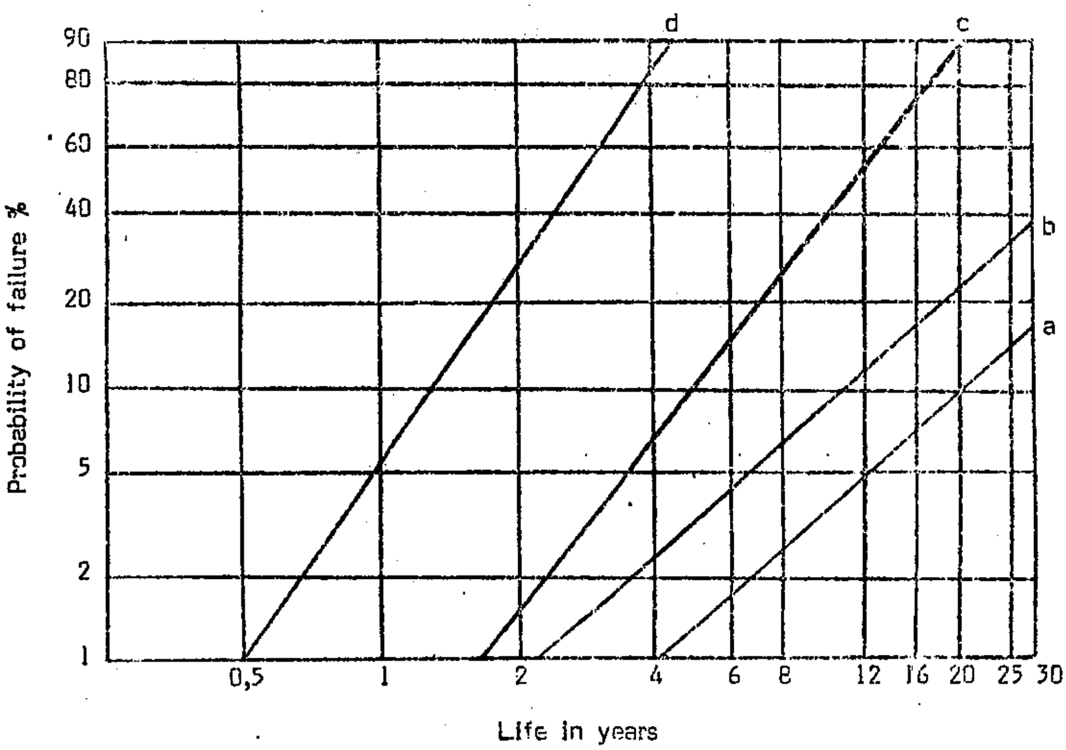

Fig 7 Failure distribution of conveyor idler bearings

line a : rolling contact fatigue of individual bearings
line b : rolling contact fatigue of one of both bearings in an idler roll
line c : service life due to grease failure under clean operating conditions) of individual
line d : service life due to ingress of contaminants and poor lubrication) bearings

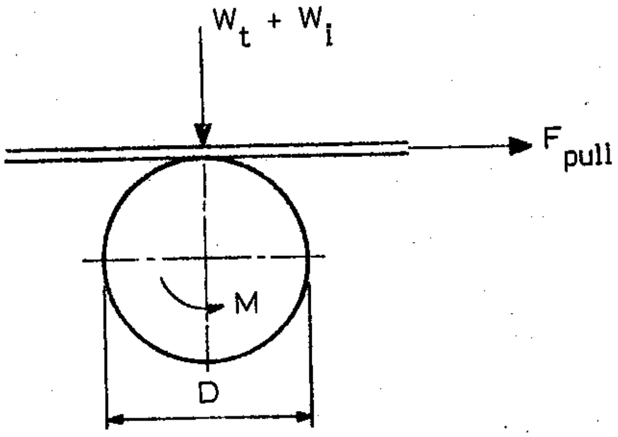

Fig 8 Belt pull to overcome bearing friction

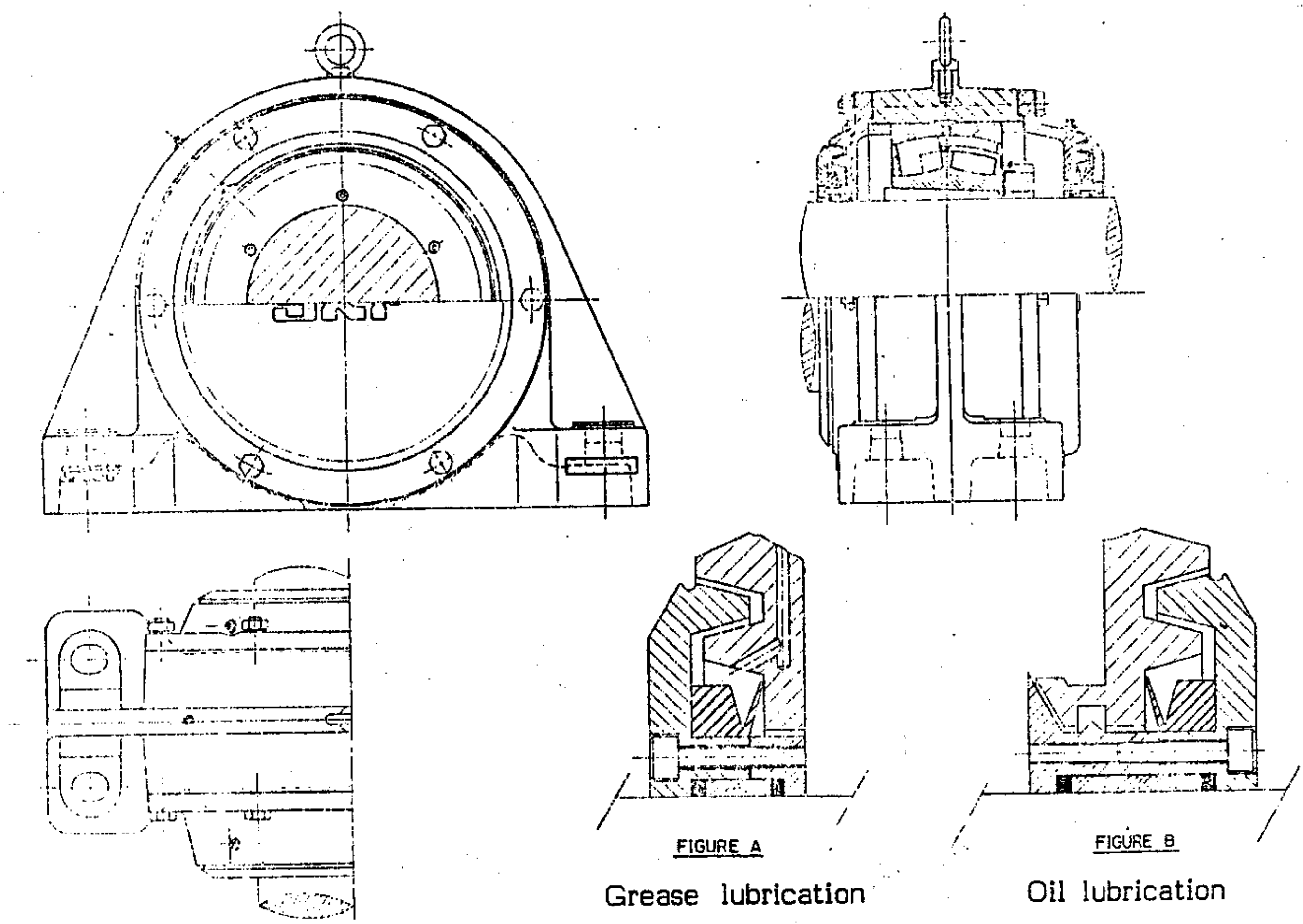

Fig 9 SBDD range of one-piece housings for extra heavy duty applications