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Special issue -89

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A review of rolling bearing engineering

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SKF (U.K.) Limited
Luton · Bedfordshire · LU3 3BL
Telephone: Luton (0582) 490049 (40 lines) · Telex 82205
Northern Sales
City Walls House · East Wood Ave · Shawlands · Glasgow
Telephone: 041-632 7171 Telex: 778119
Pennine Sales
Unit 5, Revie Road Industrial Estate · Elland Road
Leeds LS11 8JG
Telephone: 0532 704365 · Telex: 556441
Midland Sales
King Edward Court · High St · Sutton Coldfield B72 1XS
Telephone: 021-355 6931 · Telex: 338982

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Engineering for success

M. Sahlin
Group Chief Executive

The SKF Group would not exist today if it had not been for a young engineer who started the company based on a significant innovative idea. The importance given at an early stage to research and development by the young company was instrumental in providing the potential for growth and ultimately the platform for world leadership in the highly competitive field of rolling bearings.

Engineering prowess has always been of great importance at SKF and is no less important today. Even though the rolling bearing has been an industrial product for quite a long time, there is still a great development potential to be harnessed. The articles in this journal will serve to illustrate this statement.

The SKF tradition of research into bearing materials and bearing behaviour dates back to the very beginning of the company. One important outcome in the past was the method adopted as an ISO standard, which is used worldwide, to predict bearing life. The new bearing life theory developed at the SKF Engineering and Research Centre is an evolution of this original work. The theoretical insights gained have been translated into practical benefits involving materials, manufacturing technologies, product design and application engineering. All these aspects are integrated into our new General Catalogue.

SKF engineers and technical salesmen have always worked in close co-operation with bearing users, often at an early design stage. To ensure that the wealth of experience accumulated from an extremely wide range of application areas is effectively made available together with the theoretical knowledge, SKF founded its own College of Engineering last year. The College shares the same site as the SKF Engineering and Research Centre in the Netherlands and is dedicated to providing the best possible in-house training to SKF engineers from all parts of the globe.

It is our belief that this arrangement of having researchers and application engineers side by side will be of great value to our customers. The very latest technical findings will be implemented without delay and will provide evidence of the importance SKF attaches to innovation and its use.

The interaction between strategy and technological progress

I. Fernlund
Group Director
Research & Quality

The article shows the positive outcome of a concentration of research facilities at one location. Excellent people with first class equipment can move technology far beyond expectations.

SKF opened its central research facility in The Netherlands in 1972. A new bearing theory emerged from the Centre in 1984 and illustrates the development potential still available for rolling bearings. This is perhaps surprising as bearings are considered by many to be mature products.

Much work has been done since the theory was published to transform it into practical terms which can be readily applied by machine designers. This issue of the BBJ is devoted to demonstrating the level of simplicity and practicality to which the theory has been brought and also highlights some of the "foundation stones" without which this progress could not have been made.

SKF has a tradition of leading the world in research in rolling bearing technology. Originally this research was conducted in laboratories which were part of several of the major manufacturing plants in Europe and the USA. The emphasis differed from country to country. Because the parent company was Swedish it was inevitable that much of the early basic research was conducted in Gothenburg. In fact, only four years after the company was founded, the Board decided that a laboratory should be established for research into bearing materials and in early 1913 the laboratory building was opened.

It was soon realised that the best method of testing the suitability of bearing steels was to endurance test finished ball bearings. The results of these tests served not only to trigger improvements to steel quality, but also to bearing manufacture itself.

A consequence of this work was the development of a theory allowing bearing life to be predicted and quantifying bearing load carrying capacity.

The creation of a Research Centre and its influence on rolling bearing testing

After the enforced separation during World War II, co-operation between the various SKF laboratories grew and regular research conferences were held to discuss common problems. Organisation was restructured and the SKF Group was formed in 1965. Four years later, in 1969, it was decided to build a research centre to serve the entire SKF Group. Many countries were considered but The Netherlands was finally chosen, mainly because of its central location in respect of the SKF manufacturing units.

The research centre SKF-ERC (an abbreviation of SKF Engineering & Research Centre B.V.) was inaugurated in 1972 but it took some years before the centre was able to find its proper role in the organisation, namely to support the strategic objectives formulated by SKF Group Management. The centre has departments for mathematical modelling, material science, chemistry, physics, electronics, metrology and mechanical testing. An international team of specialists has been recruited both from inside the SKF organisation and externally.

After some years, the endurance testing equipment of five countries was transferred to a centralised facility at ERC. This step brought considerable improvement in both testing methodology as well as the evaluation of results. Standardised test procedures were developed and standardised methods of result analysis and evaluation introduced. Earlier differences in test conditions such as different test speeds and loads as well as different lubricants were eliminated.

The L_{10} life concept

A rolling bearing rotating under load may ultimately suffer from material fatigue. Typically, fatigue damage is characterised by a small piece of material breaking away from the raceway leaving a cavity. This cavity may then propagate into a crack and the bearing will fail.

If many apparently identical bearings are run under the same conditions until 10 % of the batch has failed from the material fatigue damage described above, then the batch is said to have attained its L_{10} life. Thus, the remaining 90 % of the bearings in the batch will survive for periods longer than the L_{10} life.

The L_{10} life is one of the basic concepts of rolling bearing engineering and is widely used by machine designers as a selection criterion for bearings. There is a relationship between the load carrying capacity of a bearing, quantified as the "Basic dynamic load rating C" and the L_{10} life. In its simplest form (for ball bearings) this relationship can be written as

$$L_{10} = \left(\frac{C}{P}\right)^3$$

where P is a load equivalent to that actually applied to the bearing. This relationship was originally formulated by two Swedish scientists, Prof. Gustaf Lundberg and Dr. Arvid Palmgren, based on the endurance tests carried out at the SKF laboratories. It was adopted by ISO in 1962 and became the internationally standardised method for calculations of the type described.

A rather more complex relationship was standardised in 1977 enabling lubrication conditions as well as superior material properties to be taken into account. Again SKF contributed with much valuable knowledge.

Discrepancy between test results and the ISO life equations

In the late 1970s very long lives were observed for bearings which were being endurance tested at ERC. These lives were far beyond those which could be expected from the standardised calculations based on the ISO equations, even using the 1977 modifications. This indicated that practice had outstripped theory and that it was now the theory that needed development. In 1980 ERC began thinking about possible modifications to the original Lundberg-Palmgren work [1]. The outcome was the presentation in 1984 by E. Ioannides and T. A. Harris at a joint ASME/ ASLE meeting in San Diego [2] of a new fatigue life model for rolling bearings.

The new theory introduces the concept of a fatigue limit and is able to account for the observed improvements in bearing life, as well as for differences in sub-surface stress fields through detailed investigation and analysis of these fields. The new life model also takes account of many phenomena which were not previously considered in the fatigue life predictions for rolling bearings. For instance, the influence of surface friction as well as local defects in the material — voids or surface damage — can now be considered.

The fatigue limit is defined in such a way that when a volume of the bearing material is subjected to a stress below a threshold value (the fatigue limit), the volume element will not fail by fatigue. There is very good correlation between test results and the new theory.

A specific example

To illustrate the preceding concepts in more practical terms a deep groove ball bearing 6309 (45 × 100 × 25 mm, see fig 1), commonly used for endurance testing, will be considered under various conditions. If radial loads of 18 600 N and 1340 N are applied, the corresponding L_{10} lives calculated in accordance with the ISO standard method are 6 days and 48 years, respectively, see fig 2. Modern SKF bearings, however, attain much longer lives. As a matter of fact, no fatigue has been recorded in bearings on test at ERC even after periods corresponding to 20 times the L_{10} life. In other words, it may be assumed that the fatigue limit is somewhere in excess of 18 600 N (see fig 3); the corresponding Hertzian maximum stress is 3300 N/mm^2.

This situation can be expressed in a different way; in the ISO equation, the fatigue limit is actually set to zero, whereas in reality it is 3300 N/mm^2. It should also be pointed out that this is not the theoretical limit. The theoretical limit is only reached at the so-called shakedown limit (approximately 4600 N/mm^2 for through-hardened bearing steel) when continuous plastic deformations occur and rapid destruction of the bearing ensues.

This example of the 6309 bearing also shows that there is still considerable development potential remaining. This is of course true of all rolling bearings.

The downsizing trend

Looking closely at various products, whether machines or electronic appliances, each successive generation is smaller than its predecessors but is still capable of doing the same tasks. This is now known as downsizing.

The ball or roller bearing is no exception. Due to the continuing improvements in the steels used for bearings as well as in the manufacturing processes used to produce the bearings, new generations of bearings are able to support increasingly heavy loads, or conversely, the same loads can be supported by smaller bearings than before. This trend was described in Ball Bearing Journals 220 and 221 "Roller bearing realiability — service experience and product development".

Future outlook

The new life theory and its ability to predict bearing life implies opportunities, but there are also threats.

The opportunities lie in a conscious and deliberate downsizing when selecting a bearing — in other words in the harnessing of the possible development potential. This strategy can be successfully applied only if the entire design and manufacturing processes are in order.

Outstanding performance in Quality Assurance and Quality Control are also prerequisites for success. The threats lie in the fact that there is more than one bearing manufacturer and there is always the risk that the competitors will be better at mastering the necessary quality techniques and attendant procedures.

SKF has taken up the challenge and is prepared to take all necessary steps to enhance SKF leadership of the rolling bearing industry.

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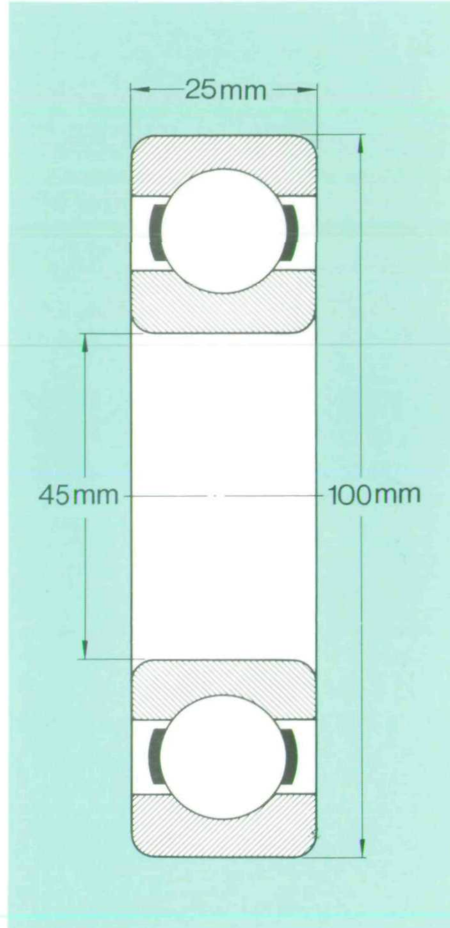


Fig 1 Principal dimensions of bearing 6309

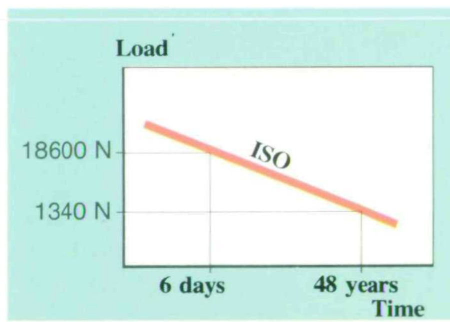
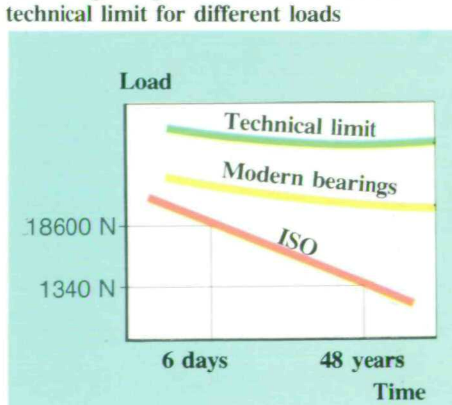


Fig 2 ISO L_{10} life predictions for bearing 6309 under different loads

Fig 3 Comparison of ISO L_{10} lives according to fig 2 with test results and technical limit for different loads



The new life theory and its practical consequences

J. Goodall Wuttkowski
SKF Head Office
Sweden

E. Ioannides
SKF-ERC
The Netherlands

Following a brief recapitulation of the differences between the classic life calculation method and the new life theory, some calculation examples are given. From these, the very considerable influence of contamination on bearing fatigue life can now be seen.

The Lundberg-Palmgren method [1, 2] for predicting rolling bearing life was established to satisfy the need for quantifying bearing performance and reflects the materials, manufacturing conditions and operating conditions of the time. It has been successfully used since then to make relative life predictions for ball and roller bearings of different types and sizes operating under different loads. The Lundberg-Palmgren life equation is, in fact, the basic rating life formula standardised by ISO in 1962 and incorporated in the present ISO 281/I-1977.

Improvements in production methods and more particularly in steel quality over the years have led to considerable extensions in actual bearing life as compared with calculated life. Empirical methods have been employed to take account of these improvements; basic dynamic load ratings have been increased at intervals and more recently "Life adjustment factors" have been introduced to take account of the desired reliability, material cleanliness and operating conditions.

However, it has still not been possible to predict the extremely long lives encountered in endurance testing under idealised conditions. Therefore, a mathematical model was developed by SKF and presented at the ASME/ASLE Joint Lubrication Conference in 1984 [3]. The new fatigue life model for rolling bearings is a generalised model for rolling bearings, of which the Lundberg-Palmgren theory represents a special case.

Derivation

The Lundberg-Palmgren theory assumes that the probability of a given volume element ΔV surviving N stress cycles and failing in the next dN is proportional to its size and is a function of its location $\vec{r}$ as well as of N itself. The probability actually increases slowly with N; this memory effect is an essential part of fatigue behaviour. Integrating over all volume elements to obtain the probability of survival of the volume V gives

$$\ln \frac{1}{S(N)} = \int_V G\,(N, \vec{r})\; dV \qquad (1)$$

where $G(N, \vec{r})\,\Delta V$ is the accumulated probability of survival over N cycles for the element ΔV. In the absence of knowledge of actual sub-surface stresses, Lundberg and Palmgren chose simply to give a functional form to the right-hand side of equation (1)

$$\ln \frac{1}{S(N)} \approx \frac{\tau_0{}^c N^e}{z_0{}^h}\, V \qquad (2)$$

where τ_0 is the maximum orthogonal shear stress developing within the volume at risk (the stress criterion) and z_0 is the depth below the raceway at which this stress occurs. V is a measure of the effective volume within which the shear stress differs sensibly from zero. The exponents c, e and h (assumed to be history-independent) were determined to give a best fit to the test data available at that time.

From the form of equation (2) it is clear that τ_0 can only represent Hertzian contact so that the life N at any given survival level S cannot reflect sensitivity to localised domains of high stress and no account could be taken of surface shear stresses produced by frictional sliding or lubricant viscosity. Finally, according to equation (2), bearing life cannot be infinite.

The Lundberg-Palmgren model is extended in the new life theory in two essential respects. Firstly, equation (2) is interpreted locally so that "τ_0" or τ becomes a local variable and the right-hand side of the equation becomes a

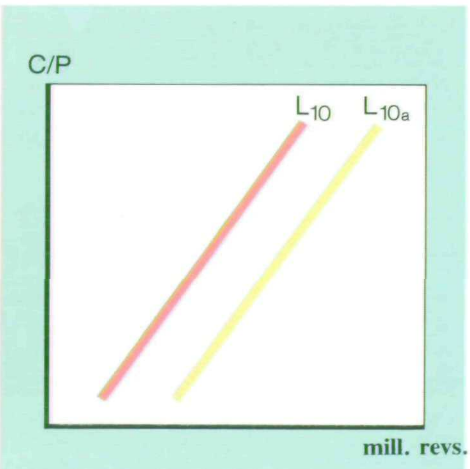


Fig 1 Load-life curves, basic rating life and adjusted rating life

"risk function". Secondly, a fatigue limit τ_u is introduced patterned on structural fatigue initiation, such that any volume element for which $\tau < \tau_u$ makes no contribution to the risk function. With these extensions, therefore, the probability of survival becomes

$$\ln \frac{1}{\Delta S} = A \frac{(\tau - \tau_u)^c N^e}{z_0{}^h} H(\tau - \tau_u)\, \Delta V \quad (3)$$

where $H(x)$ is the Heaviside step function. As a first approximation, equation (3) may be written in the integral form

$$\ln \frac{1}{S(N)} = \bar{A} N^e \int_{V_R} \frac{(\tau - \tau_u)^c}{z'^h} dV \quad (4)$$

where $\bar{A}$ is the volume-averaged value of A and the integration runs only over the region where the stress criterion exceeds the threshold value. The fatigue limit itself may also assume local values. The factor z_0, which was originally introduced to include the propagation interval between sub-surface damage initiation and its appearance on the surface, has been retained but should now be regarded as a local stress-weighted average depth z'. The Lundberg-Palmgren exponents remain the same.

The Lundberg-Palmgren formulation leads to a relationship between N and S depending on bearing geometry and load which can be reduced to the simple form

$$L_{100(1-S)} = \left(\frac{C}{P}\right)^p \quad (5)$$

L is the expected life (N in million revolutions) under a load P. C is the

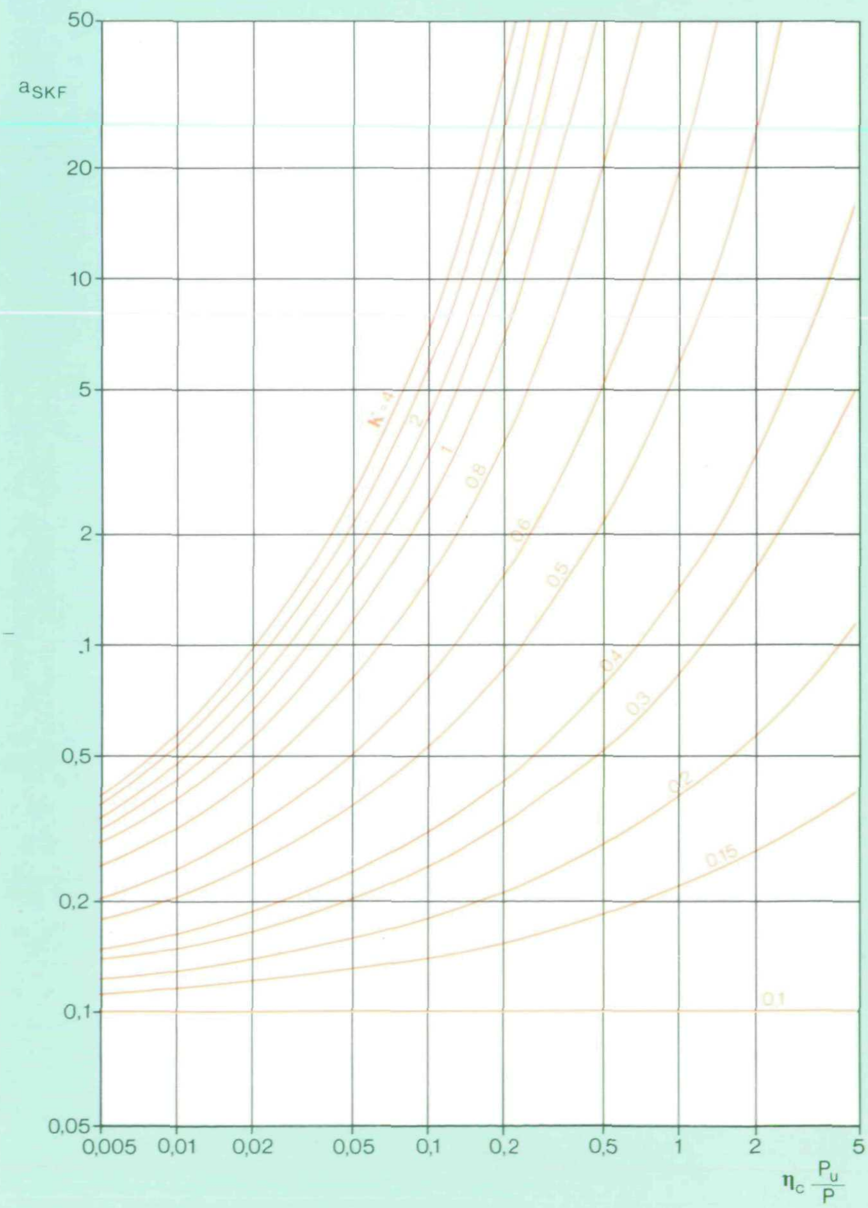


Fig 2 Diagram showing a_{SKF} factor for radial ball bearings

basic dynamic load rating and is the load under which 100 S % of a population of bearings will survive for at least 1 million revolutions. The ISO standardised life is based on this expression with $S = 0{,}9$ (L_{10} life represents a 10% failure rate) and using the Hertz analysis, τ is the maximum orthogonal shear stress and the load-life exponent p is 3 for ball bearings or 10/3 for roller bearings. The ISO life equation is therefore

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

and the adjusted rating life equation standardised subsequently by ISO reads

$$L_{na} = a_1\, a_2\, a_3 \left(\frac{C}{P}\right)^p \qquad (7)$$

The basic rating life L_{10} will decrease linearly with load on a logarithmic plot. The life adjustment factors a_1, a_2 and a_3 do not change the slope of this line; they only shift it to longer or shorter life (fig 1).

The introduction of the new life theory with its ability to introduce local stresses as explained above (equation (4)) has led to a generalisation of equation (6) as

$$L_{naa} = a_1\, a_{SKF} \left(\frac{C}{P}\right)^p \qquad (8)$$

In this equation the life adjustment factor a_{SKF} accounts for environmental conditions such as lubricant film (K) and contamination (η_c) as well as for the existence of the fatigue load limit of the material (P_u). Clearly the a_{SKF}

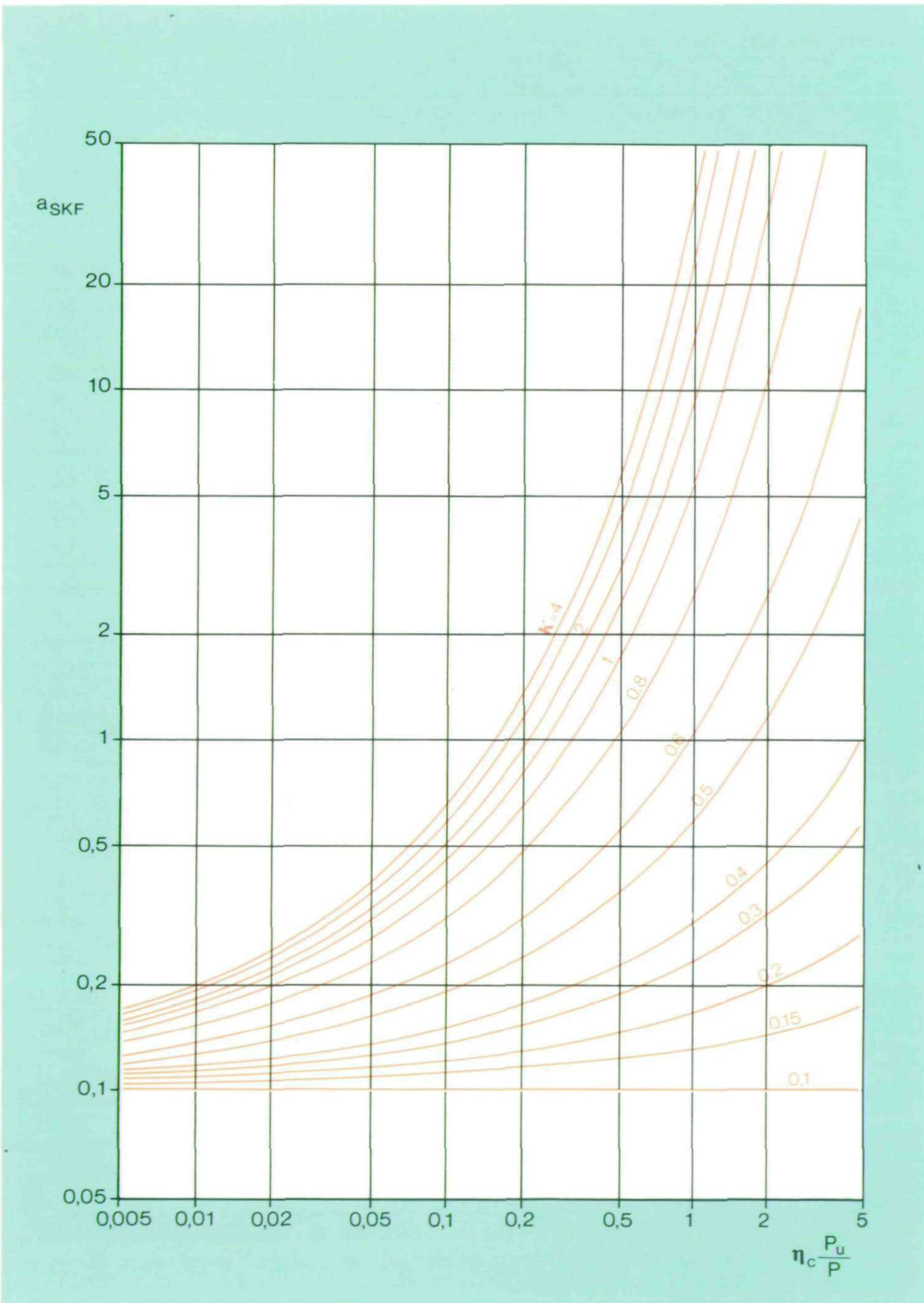


Fig 3 Diagram showing a_{SKF} factor for radial roller bearings

Table 1 Values of adjustment factor η_c for different degrees of contamination

Condition	η_c [1)]
Very clean Debris size of the order of the lubricant film thickness	1
Clean Conditions typical of bearings greased for life and sealed	0,8
Normal Conditions typical of bearings greased for life and shielded	0,5
Contaminated Conditions typical of bearings without integral seals; coarse lubricant filters and/or particle ingress from surroundings	0,5...0,1
Heavily contaminated[2)]	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.

[2)] 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_{naa}.

factor represents a complex relationship of the above-mentioned effects which has been simplified into a set of diagrams in the SKF General Catalogue 4000. Two of the diagrams for radial ball bearings (fig 2) and for radial roller bearings (fig 3) are shown here. In these diagrams the a_{SKF} values are given as functions of η_c (P_u/P) for different values of K. The General Catalogue also contains general recommendations for the contamination factor η_c whilst values of the fatigue load limit P_u are given for each individual bearing and will be found in the bearing tables.

Calculation examples

A deep groove ball bearing designated 6210 is operating under a purely radial load of 2500 N at a speed of 4500 r/min, lubricated in such a way that the viscosity ratio K is 3.

The basic dynamic load rating C for the bearing is 35 100 N and the P_u value 980 N. Using the original ISO life equation (equation (6)), the L_{10} life is 2768 million revolutions, or in operating hours, L_{10h} = 10 250 hours.

Using the SKF version of equation (7), viz.

$$L_{na} = a_1\, a_{23}\, L_{10} \qquad (9)$$

from fig 4 an a_{23} value of 2 is obtained for K = 3, so that, in operating hours, the L_{10ah} life becomes 2 × 10 250 hours.

Using the new life theory, $P_u/P \simeq$ 0,4. If utmost cleanliness is assumed then η_c = 1 (see Table 1) so that the value of a_{SKF} from the diagram (fig 2) becomes 50 (the maximum value of a_{SKF} considered meaningful for catalogue purposes). Rewriting equation (8) gives

$$L_{10aah} = a_{SKF}\, L_{10h} \qquad (10)$$

so that an L_{10aah} life of 50 × 10 250 operating hours is obtained which corresponds to some 58 years of continuous operation. Since utmost cleanliness is generally only obtained under controlled laboratory conditions, this result predicts the "infinite" lives found in endurance tests.

However, if cleanliness is not good, i.e. assuming an η_c value of 0,1, then from fig 2 the value of a_{SKF} is 1,8 and the adjusted rating life according to the new life theory L_{10aah} would correspond to just over 2 years of continuous operation. This example demonstrates rather clearly the considerable effect of contamination.

Another example is that of a spherical roller bearing designated 24164 CC/C3W33 in an industrial gearbox. The bearing rotates at 60 r/min under an equivalent dynamic load P = 750 000 N. The bearing is lubricated with the same oil as the gears and the viscosity ratio K is 0,6. The basic dynamic load rating C = 3 740 000 N.

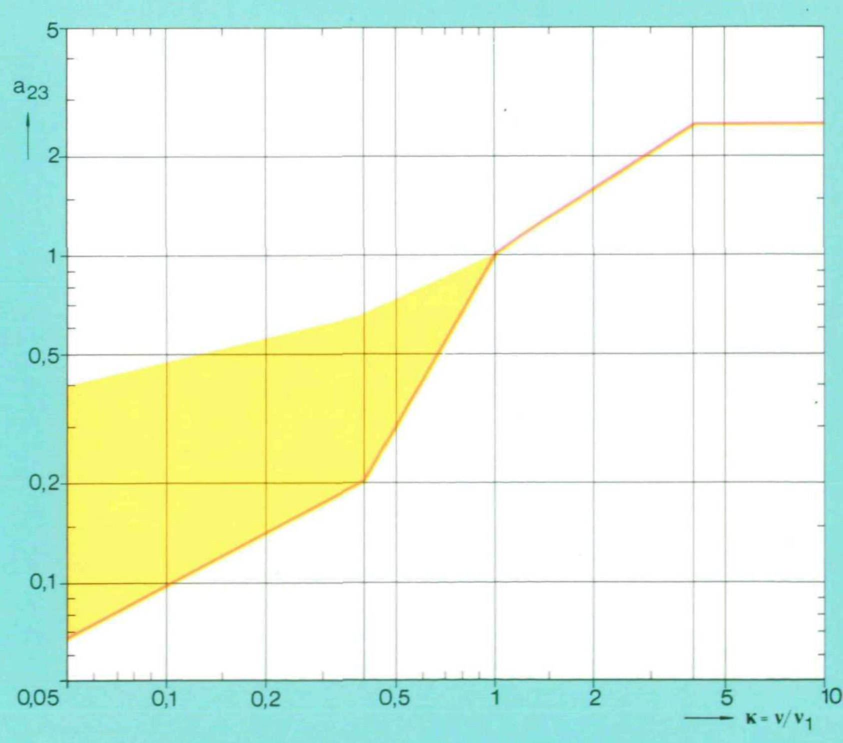


Fig 4 Diagram showing a_{23} adjustment factor as a function of the viscosity ratio K

In this case, the L_{10h} life is calculated as 58 800 hours. With a viscosity ratio K value of 0,6 the a_{23} value is 0,4 (fig 4) which gives an adjusted rating life according to equation (9) of 23 500 hours.

The P_u value is 510 000 N so that P_u/P = 0,68. If cleanliness is assumed not to be good (an open bearing in a gearbox) so that η_c = 0,1, then the a_{SKF} value for η_c (P_u/P) = 0,068 and K 0,6 from the diagram in fig 3 will be ~ 0,2. In fact, the average service life of these bearings in this application is only 15% of the L_{10} life, so that the calculated life of 11 800 hours is still longer than the service life and would indicate that the choice of η_c value was somewhat optimistic.

Discussion

The new life theory, even in its simplified "catalogue equation" form provides a tool for predicting bearing life more realistically than the ISO standard methods. The new method can be used to study the effects of contamination as well as lubrication on bearing life and shows the benefits of using sealed bearings or improving external seals. For clean applications, the much enhanced reliability is demonstrated and there are opportunities for downsizing bearings, or uprating machine performance.

As an example of the downsizing opportunities, fig 5 shows the lives for bearings 6309 and 6209 calculated using the old and new methods under a range of loads (C/P = 2–15) and for K = 4, as well as for different levels of contamination (η_c = 1– 0,1). The importance of controlling contamination is easily seen: the calculated L_{10aa} lives according to the new life theory can be either longer or shorter than the L_{10a} lives, depending on the level of contamination. It can also be seen that under a wide range of loads it is possible to substitute bearing 6209 for bearing 6309 and obtain the same calculated life, provided steps are taken to prevent contamination from entering the bearing.

It will be found, however, that in very many cases the new theory will give lives which are far shorter than the ISO L_{10} lives. This is because bearings in most applications have some degree of contamination and this effect is added to that of the lubricant film thickness as represented by K.

The new approach should be used in the first instance to examine the magnitude of the effect of contamination. The machine designer or bearing user can then decide whether it is desirable or feasible to pay more attention to sealing in order to clean up the immediate environment of the bearing or to introduce filtering into the lubrication system.

If sealed bearings are used it must be remembered that grease and seal life must be considered.

At present little information is available on contamination in everyday bearing operation although work is in progress to define typical η_c values for applications and much other research into the effects of contamination is being conducted.

With regard to the fatigue load limit it may be argued that it is an indication of the improvement in steel bearing design and manufacturing quality, as well as testing quality, that it is now possible to detect this limit.

Although Palmgren discussed the existence of such a limit for bearings in papers published in the early 1920s [4, 5] he could not observe it, either then or later. In fact it is only since the late 1970s and early 1980s that the "infinite" lives have been obtained, indicating that – as with other

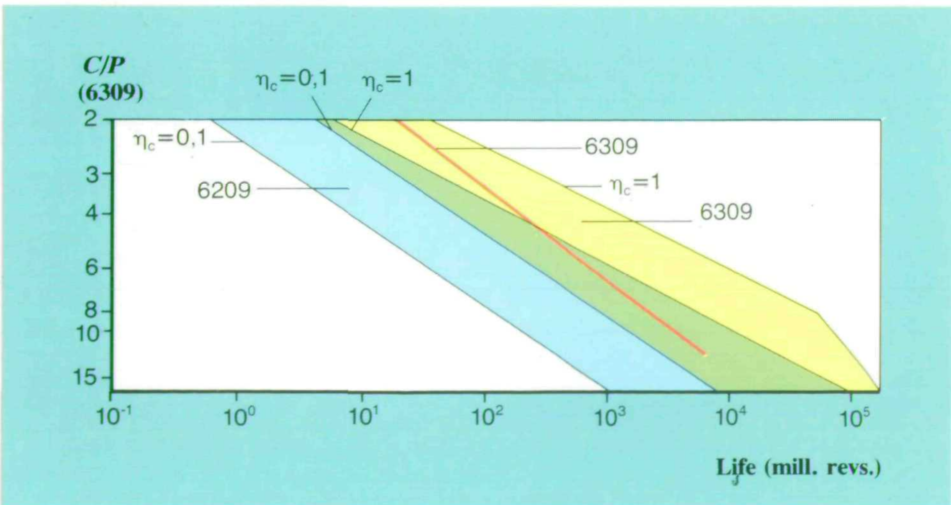


Fig 5 L_{10a} **life for bearing 6309 (red line) and** L_{10aa} **lives for bearings 6309 (yellow) and 6209 (blue) for** K **= 4 and levels of contamination ranging from** η_c **= 0,1 (contaminated) to 1 (very clean)**

mechanical and structural components – bearings will not experience fatigue if loads are light enough.

For SKF, the new life theory provides input and stimulus for new bearing designs in the long term and is already influencing development work and quality philosophies.

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Notation

a_1	life adjustment factor for reliability
a_2	life adjustment factor for materials
a_3	life adjustment factor for operating conditions
a_{23}	combined life adjustment factor for material and operating conditions
a_{SKF}	life adjustment factor based on new life theory
A	probability normalisation factor
c, e, h	Lundberg-Palmgren exponents
C	basic dynamic load rating, [N]
$G(N)$	elementary survival probability
$H(x)$	Heaviside step function
L_n	life (n % failure), [Mrev]
L_{10}	basic rating life (ISO 281/ I-1977), [Mrev]
L_{na}	adjusted rating life (ISO 281/ I-1977), [Mrev]
L_{naa}	adjusted rating life to new SKF life theory, [Mrev]
N	number of cycles
p	exponent of the life equation
P	equivalent dynamic bearing load, [N]
P_u	fatigue load limit for bearing, [N]
$\vec{r}$	position vector, [m]
$S(N)$	survival probability
V	volume, [m^3]
z_0	depth of maximum Hertzian shear stress, [m]
z'	stress weighted average depth, [m]
η_c	adjustment factor for contamination
K	viscosity ratio (υ/υ_1)
υ	kinematic viscosity of lubricant at operating temperature, [mm^2/s]
υ_1	requisite kinematic viscosity of lubricant at operating temperature, [mm^2/s]
τ	shear stress (orthogonal, fatigue limit, elastic limit etc.), [Pa]

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Performance testing: facilities and opportunities

R. Goss
SKF-ERC
The Netherlands

E. Ioannides
SKF-ERC
The Netherlands

SKF's performance test facilities at the SKF Engineering & Research Centre (ERC) in The Netherlands are unique in the bearing industry due to their size and sophistication. They are supplemented by extensive conformance measuring and advanced failure analysis facilities and continuously provide input for product improvement and development. These facilities also provided extensive supporting evidence for the New Bearing Life Theory and some of these results are discussed here.

The service performance of a rolling contact bearing depends not only on the materials, design and manufacture of the bearing itself, but also on the function of the lubricant and seals, the surrounding components and the service conditions. The latter

Mechanical Testing

The principal objective of the Mechanical Testing Department at ERC is to determine the functional performance of SKF products. Such product performance is investigated using static and dynamic rigs under conditions which simulate the important parameters of the operational environments. This simulation can vary from the complicated, full dynamic simulations to simpler block programme tests, or single parameter tests. The most sophisticated test rigs are the full dynamic simulators for automotive wheel hub units, DYANA (fig 1), and the railway axlebox bearing simulator for high speed trains, THISBE (fig 2).

DYANA is a road simulator which can test up to four hub units simultaneously. In DYANA, the six most important parameters affecting wheel bearing performance: drive shaft torque and speed, suspension bounce displacement, steer motion, and vertical and lateral tyre-road contact patch loads, are applied so as to simulate various types of route.

The test system utilises a technique called Remote Parameter Control (RPC*). This control technique

*RPC = Trademark, MTS Systems Corporation, Minneapolis, USA

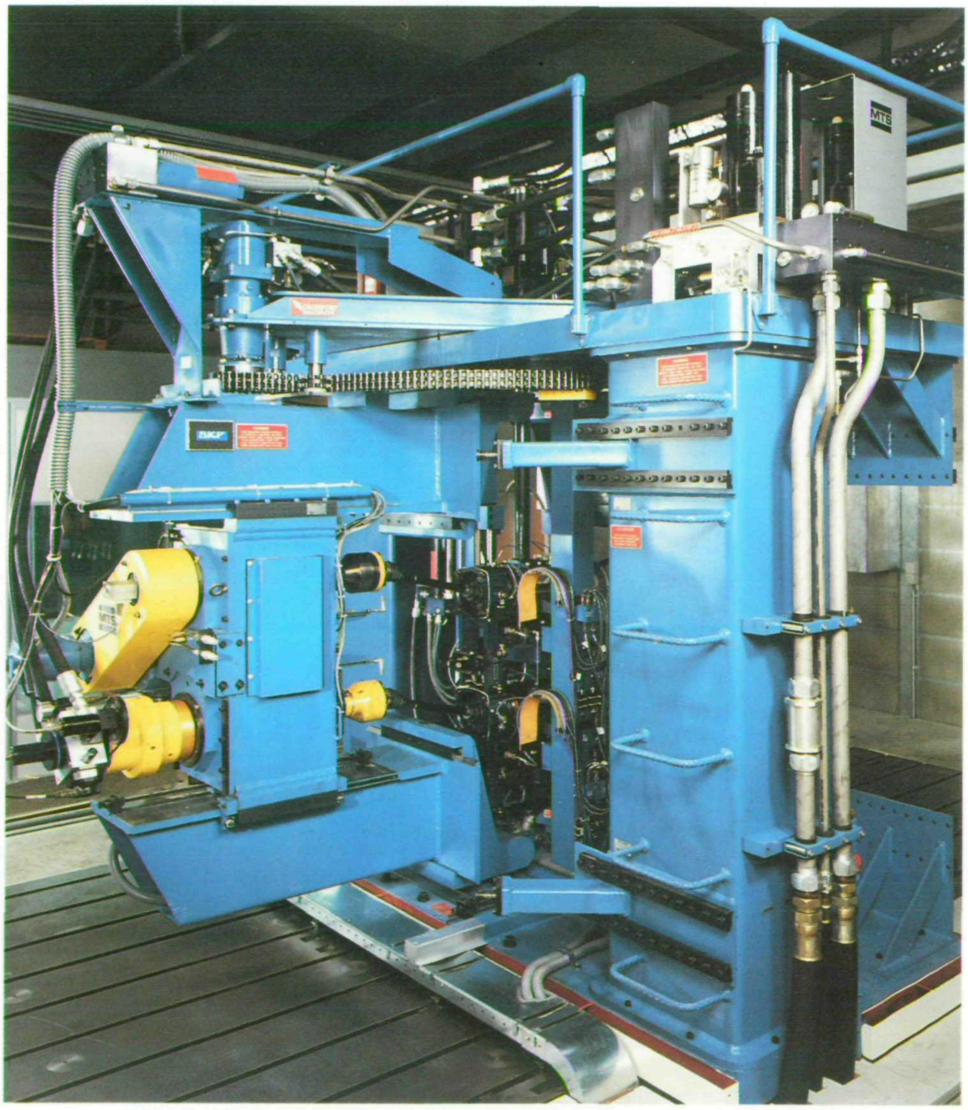


Fig 1 Dynamic testing and analysis system for automotive driven wheel bearing unit arrangements — DYANA

include loads, speeds, temperatures and contamination, that is, the parameters under which the bearings operate and which are not always precisely known. Testing, together with design and manufacturing, is an integral part of the development of rolling bearings. In general, testing can be classified into application-related testing and standardised testing.

Application-related testing, as the name implies, is closely connected to a specific application, or a class of applications, by reproducing in the test values of relevant parameters closely connected to the application or applications. In standardised testing, on the other hand, fixed and universal values of the relevant operating parameter or parameters are selected for monitoring the performance of the engineering components, and in our case of rolling bearings.

In the following sections, under the respective headings of 'Mechanical Testing' and 'Endurance Testing', facilities at ERC and some of the work carried out using these facilities will be described.

compensates for the fact that in many test systems it is not possible to measure a system response to a dynamic input at the point of application of that input. This means that there are either components of the test system, or the test specimen itself, between the point of application of the load, torque, speed etc. and the point at which the dynamic response is measured. In order to control the system response such that a specific dynamic time history is simulated, the system input must be adapted to compensate for the transfer function between the system inputs and outputs. In the case of a multi-parameter system such as DYANA, cross-coupling effects, the influence on the response of a particular channel to inputs other than the main input to that channel, must also be considered. These effects are compensated for by RPC which modifies the drive signals sent to the system by an iterative process until the desired response is obtained.

DYANA is used to perform acceptance tests on prototype hub units and to investigate the influence of design changes and dynamic conditions on wheel bearing performance. Durability tests are run to simulate Town, Belgian Pavé, Motorway and Country Road routes. The machine incorporates monitoring instrumentation which allows the performance of the hub units under test to be evaluated. The use of such a simulator allows the use of a wider range of measuring equipment than can be used in full vehicle testing; it allows repeatable tests to be made in order to compare different hub unit designs, and it allows acceleration of the test by being able to run day and night under computer supervision. Further details of this rig and its operation can be found in [1].

THISBE, the railway dynamic test rig, is capable of testing railway axlebox bearings under static, quasi-static, and dynamic conditions at speeds up to 400 km/h. The forces and moments sustained by a railway axlebox bearing are clearly complex. The dynamic forces and moments cannot be predicted accurately from suspension characteristics and wheel-rail data. To define these environmental conditions such that they can be realistically simulated in the test laboratory, the actual service

Fig 2 Test rig for high speed train bearings — THISBE

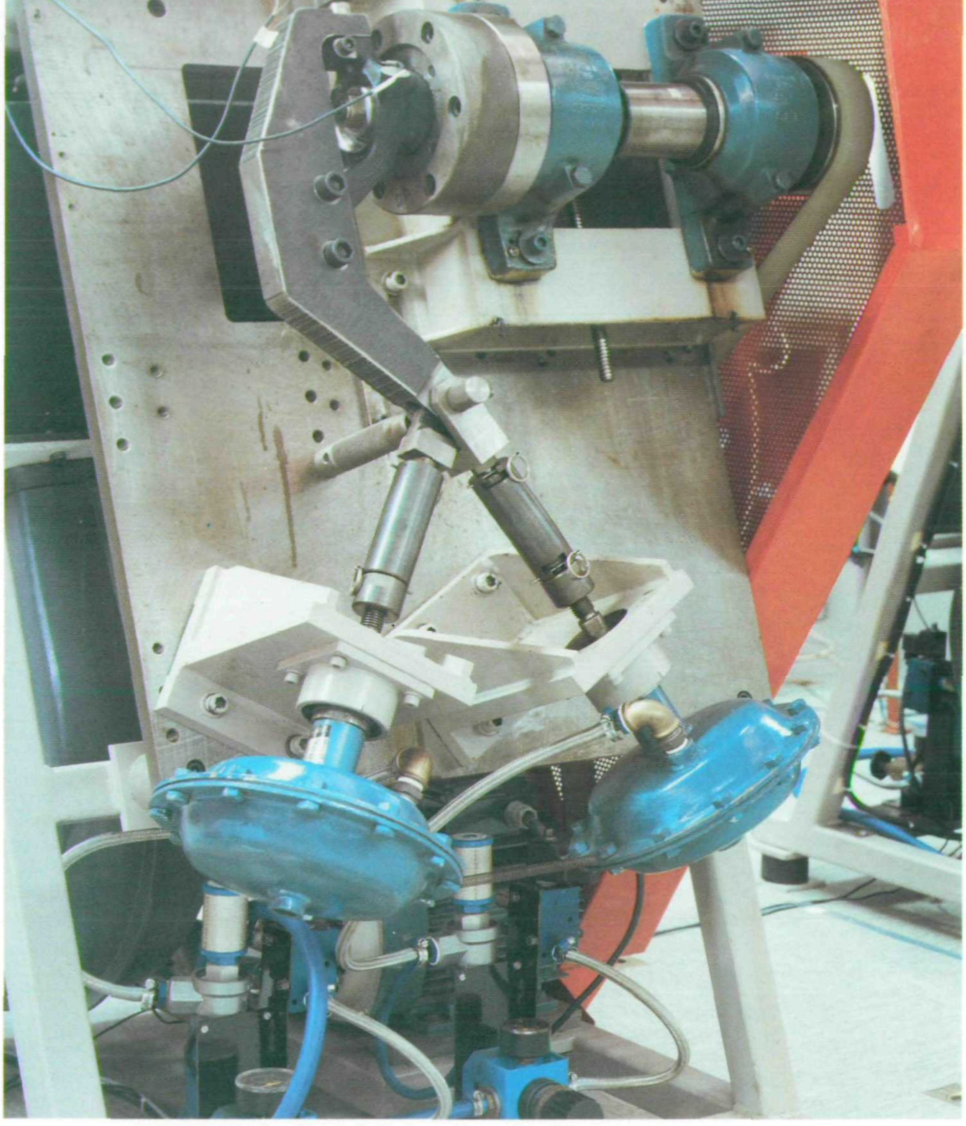


Fig 3 Hub unit rolling contact fatigue test machine

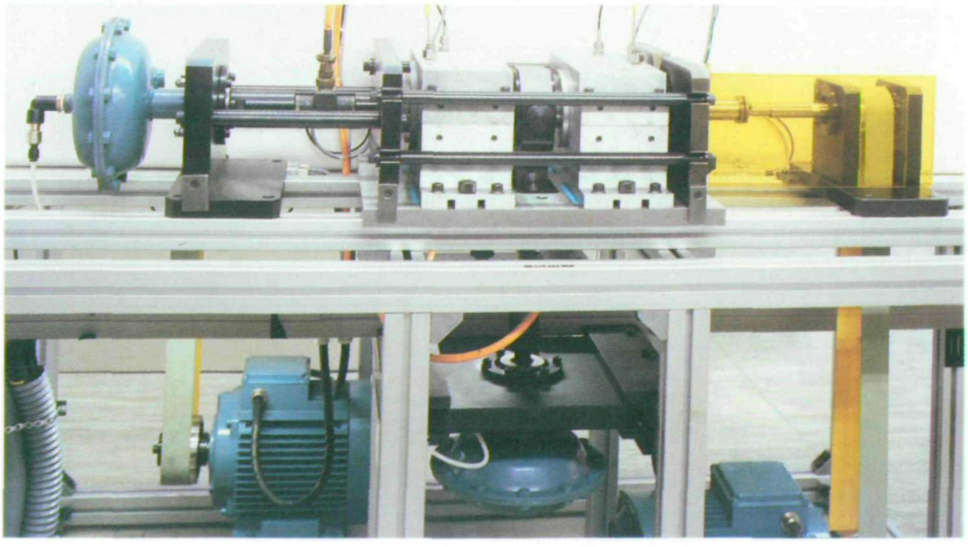


Fig 4 Contaminated lubrication environment test equipment — COLETTE

conditions must be measured using instrumented axleboxes. With such instrumented axleboxes the measurement of axlebox displacements and accelerations does not present particular difficulties. Displacement transducers can be located in the primary suspension, and a three-directional accelerometer mounted in the axlebox end cap. The connecting cables can be passed from the axlebox, through the vehicle, to the signal conditioning and recording equipment located in a convenient passenger coach. The parameters normally measured are:

— vertical axlebox displacement and acceleration,

— lateral axlebox displacement and acceleration,

— longitudinal axlebox displacement and acceleration,

— air velocity,

— bearing temperatures,

— axial bearing displacement,

— ambient temperature,

— train speed.

Such data from selected stretches of track are analysed using a Fourier analysis system, after suitable low-pass filtering and amplification. Finally, two techniques are employed for the preparation of the data as input to THISBE: amplitude histogram analysis and frequency analysis. During the test the rig is controlled by a computer random vibration control (RVC) system programmed with the desired combined displacement and acceleration spectrum (0 to 100 Hz). The RVC system sends a random Gaussian drive signal to the servo-

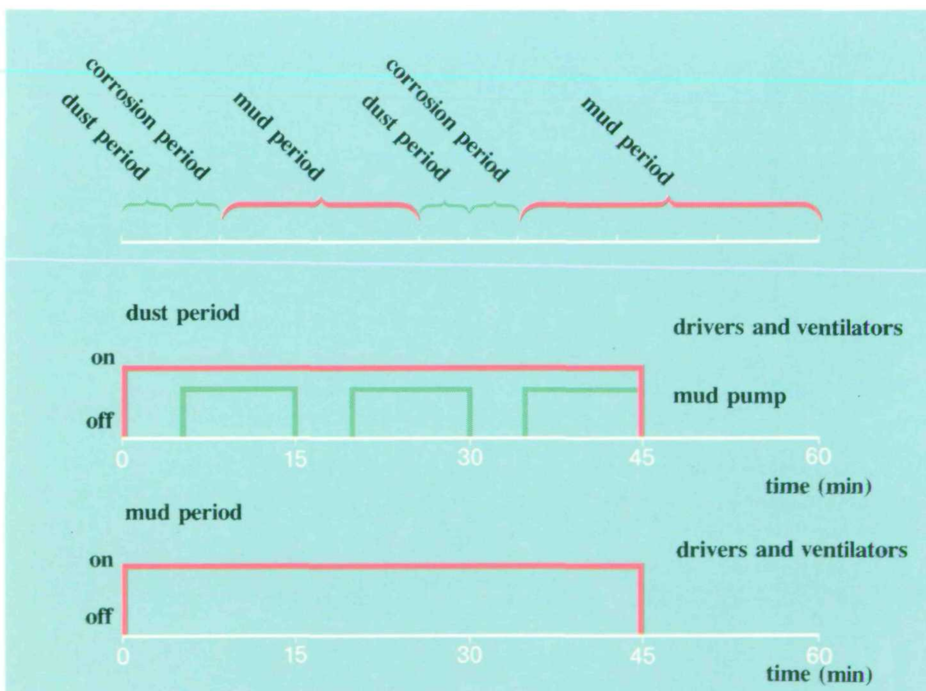


Fig 5 Service simulation test procedure

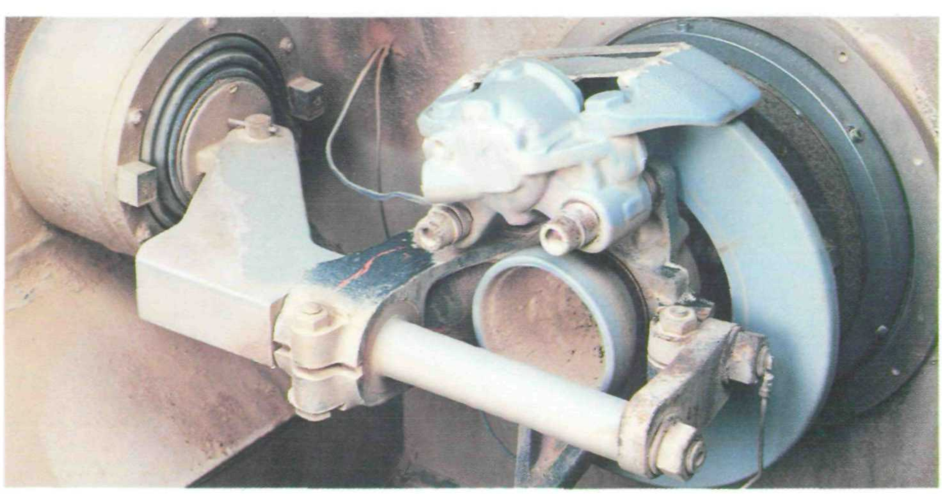


Fig 6 Wheel bearing service simulation rig and actual suspension system under test

controller of the vertical actuator. The vertical displacement and acceleration of the axlebox are measured using a displacement transducer and an accelerometer. The output signals are combined using cross-over filters, and Fast Fourier analysis is performed by the RVC system on this combined signal. The actual measured frequency spectrum is then compared with the desired frequency spectrum and the drive signal modified iteratively until the measured and desired spectra agree within specified limits. This closed loop process continues throughout the duration of the test, thereby compensating for any changes to the transfer function of the test machine caused by wear, temperature effects, or any other causes. The quasi-static forces and speeds are programmed using a micro-processor which controls the three independent servo-control systems for the force actuators and the hydraulic motor.

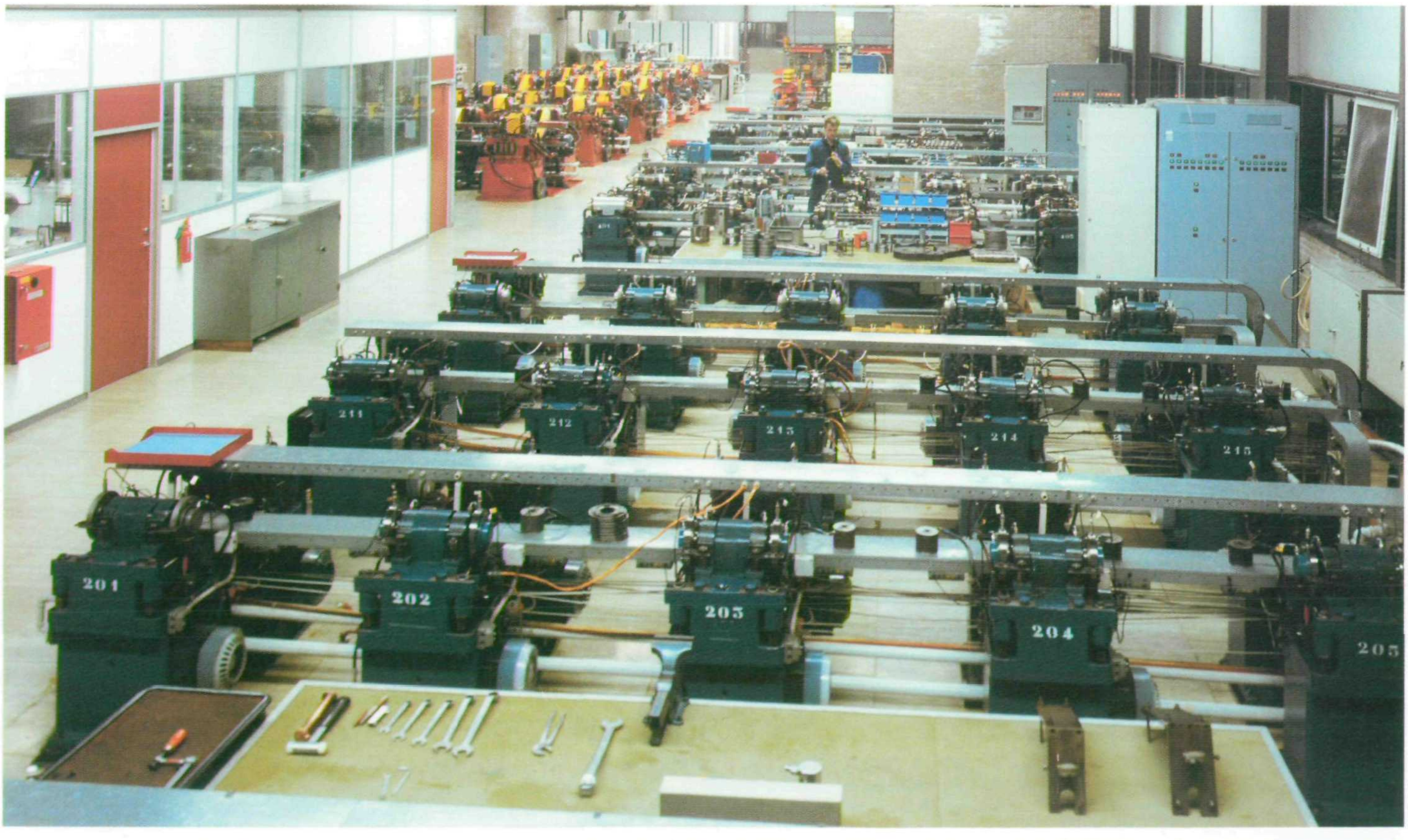
To ensure the correct load and thermal environment, THISBE is equipped with part of a bogie frame, including an original primary suspension (fig 2). A shaft driven by a servo-controlled hydraulic motor replaces the railway vehicle axle. Bearing forces are applied by servo-controlled hydraulic actuators. Forced air cooling is provided by a fan to simulate the effect of air passage. More details on the rig, its operation and results obtained can be found in [2].

Finally, it should be stated that the use of this sophisticated rig allows SKF to develop improved bearings for railway axleboxes without the need to carry out lengthy and expensive field tests. This not only improves bearing performance, but also shortens the time needed to introduce new products.

In addition to these two rigs, unique in the bearing industry, the Mechanical Testing Department in ERC includes a variety of other test rigs too numerous for them all to be described here. As an indication of their capabilities, some of the automotive and seal testing rigs will be briefly discussed below.

Hub unit rolling contact fatigue life is determined in a test system (fig 3) which applies high, but realistic, alternating cornering forces, equivalent to ±0,6g lateral vehicle acceleration to the unit under test. In order to obtain an accurate life prediction, 60 units are tested in 'sudden death' groups of four units. When any one unit in a group fails, testing of the other units is suspended. The individual 'group' lives obtained are evaluated using Weibull analysis to obtain an accurate assessment of the 90 % survival life

Fig 7 The Endurance Test Hall



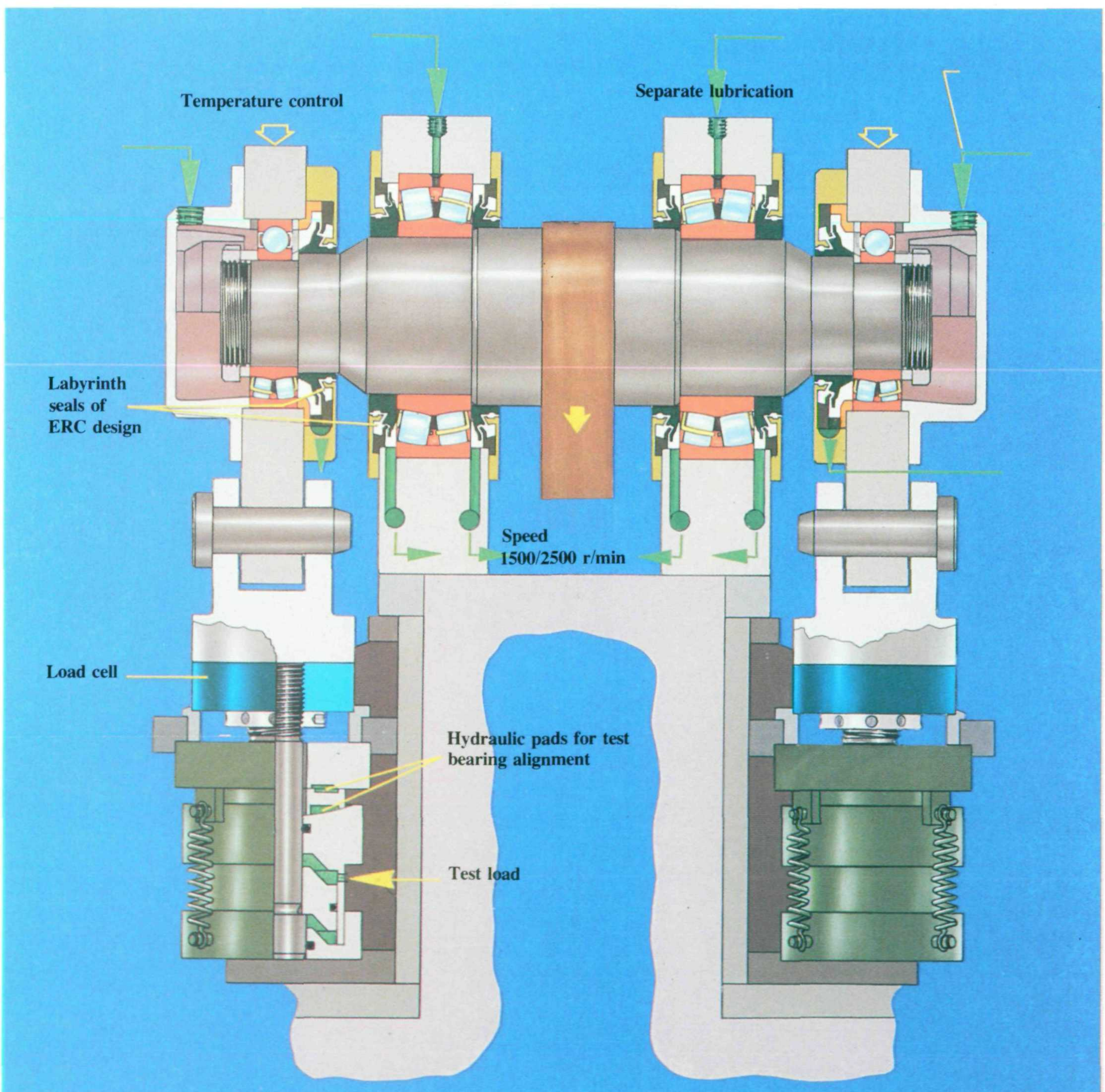
(the life at which 10 % of the units will have failed), and an evaluation of the confidence limits associated with this life assessment. This test strategy allows reliable comparative tests to be carried out to investigate different bearing designs under identical test conditions.

The same test rigs are also used to test the structural strength of flanged hub units. In these cases, however, the more severe cornering condition of 0,8g is constantly applied.

Gearbox bearings, which are subject to contamination by wear particles from the gears, are given special heat treatments to develop surfaces resistant to indentation by wear debris. The result of applying these treatments is a twenty times increase in service life under severe contamination conditions. In this application the service life is determined by an unacceptable increase in the noise generated by the bearing. A gearbox service simulation test rig (fig 4) has been developed to investigate the effects of such heat treatments on the performance of gearbox bearings. This test rig is capable of reproducing the effects of gear changes in a full simulation of gearbox service conditions.

To evaluate and improve the efficiency of wheel bearing seals, many individual seal parameters which affect the seal life are evaluated and

Fig 8 R3 endurance test machine configuration



optimised. Identical hub units, however, have performed differently in various cars, which is mainly a result of the individual suspension systems. The hub units are surrounded by different secondary protection systems such as extra plating, flingers, brake discs, extended boots etc. A direct evaluation with relevant suspension components is therefore required and a service simulation test procedure was developed to directly evaluate hub unit life in severe laboratory seal tests by reproducing service failure mode.

In concept, the wheel bearing seal test rig is an application-orientated test with the wheel bearing mounted in its typical suspension system, with extra plating, flingers, brake discs, cv-joints, extended boots etc. Alternating 0,3g cornering loads are generated pneumatically and transmitted to the test bearing by loading arms of equivalent length to the appropriate tyre radius. Dry and wet contaminants are alternately circulated around the hub unit while the bearing runs at 1000 r/min; equivalent to about 100 km/h. Regular stand-still periods allow the bearing to cool. The test procedure is shown in fig 5 and a photograph of the rig and an actual suspension system is shown in fig 6.

These extensive mechanical testing facilities allow ERC to:

— determine basic physical phenomena and important parameters which influence product performance,

— establish and verify calculation methods which describe the bearing performance,

— establish that the customers' requirements are fulfilled.

Endurance Testing

The Endurance Test facility at SKF is centralised at the Engineering & Research Centre in The Netherlands. It consists of 145 test machines concentrated into one test hall (fig 7). These machines, basically all of the same design, are available in three sizes: the R1 model, used for testing small ball bearings between 17 and 25 mm bore size; the R2 model, used for testing medium size ball and roller bearings between 25 and 50 mm bore size, and the R3 model, used for testing large ball bearings and medium size roller bearings. The R1 and R3 machines operate with pure radial load whilst the R2 machines operate with both radial and axial load.

Fig 8 shows the basic machine design. In all cases the test load is hydraulically applied and, where necessary, compensation is made for misalignment due to shaft bending by means of hydrostatic alignment bearings. Lubrication is usually by circulating oil: fully flooded thick film, or marginal thin film conditions. In all cases the cleanliness of the test lubricant is maintained by the use of high capacity multi-pass filtration down to 3 μm pore size. Further, the oil contamination level is monitored with an "Automatic Particle Counter", supported by analysis of impurities; regular chemical analysis is employed to warn of possible oil degradation. Test tooling is manufactured to very stringent specifications and all procedures used are designed to provide accurate, repeatable test results.

The test installation operates 24 hours/day throughout the year. This is made possible by the use of computer monitoring of such parameters as temperature, load, vibration and test administration.

The test programmes include pre-test measurements of bearing contact geometry, material and heat treatment

Fig 9 The Scanning Electron Microscope

parameters to provide background information for the bearings to be tested. Of prime importance is the post-mortem investigation of both failed and unfailed bearings. As materials and manufacturing methods have improved during the years, the modes of bearing failure have changed. To be able to continuously improve the product, it is essential to be able to identify both the reasons for failure and the ways in which it develops. The testing activity is therefore supported by advanced surface topography and form measuring equipment, as well as scanning electron microscopy (fig 9), all of these tools being used in the analysis of tested bearings.

Recently these facilities have been used extensively to provide experimental verification of SKF's new bearing life theory. Tests have been run on both standard production bearings and bearings specially manufactured from material deliberately produced with a high oxygen content, the latter being necessary to provide actual life estimates. Modern SKF MR material with its very low oxygen content has been shown to have "infinite" life, hence life estimates for this material are impossible to obtain within reasonable test times. The test programme was designed to take into account load, size and lubrication effects and included deep groove ball bearings 6205, 6206, 6309 and 6220 under high and medium load and also reduced and full film lubrication conditions. Line contact bearings were included in the programme by testing spherical roller bearing 22220. The complete programme has involved about thirty tests covering in total approximately fifteen hundred bearings.

The results show very good agreement with the predictions made by the new bearing life theory, i.e. the bearing life increases faster with the reduction of load than previously considered, e.g. fig 10.

Other work carried out in connection with the new bearing life theory is the study of the effects of contamination on life and the investigation of life reduction factors for contaminated

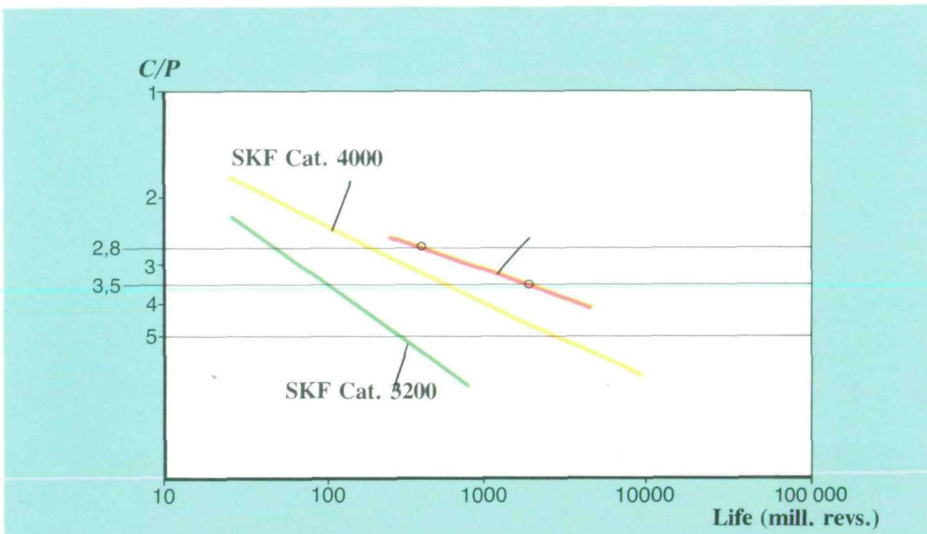


Fig 10 Test results and catalogue predictions for bearing 6309

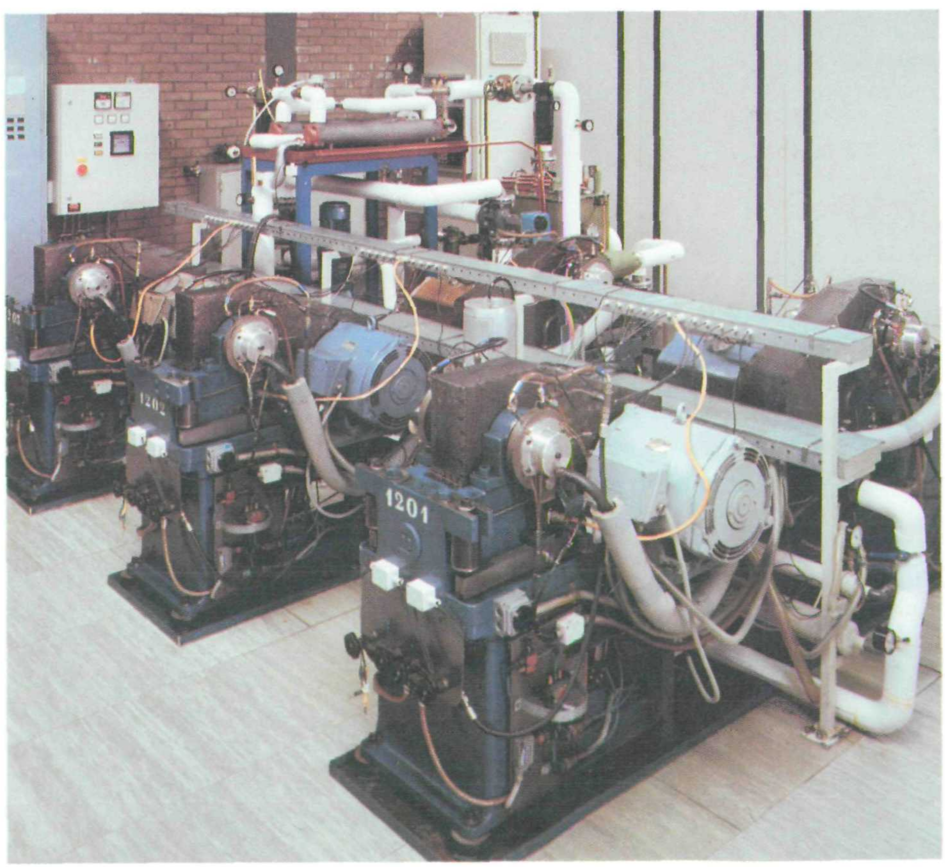
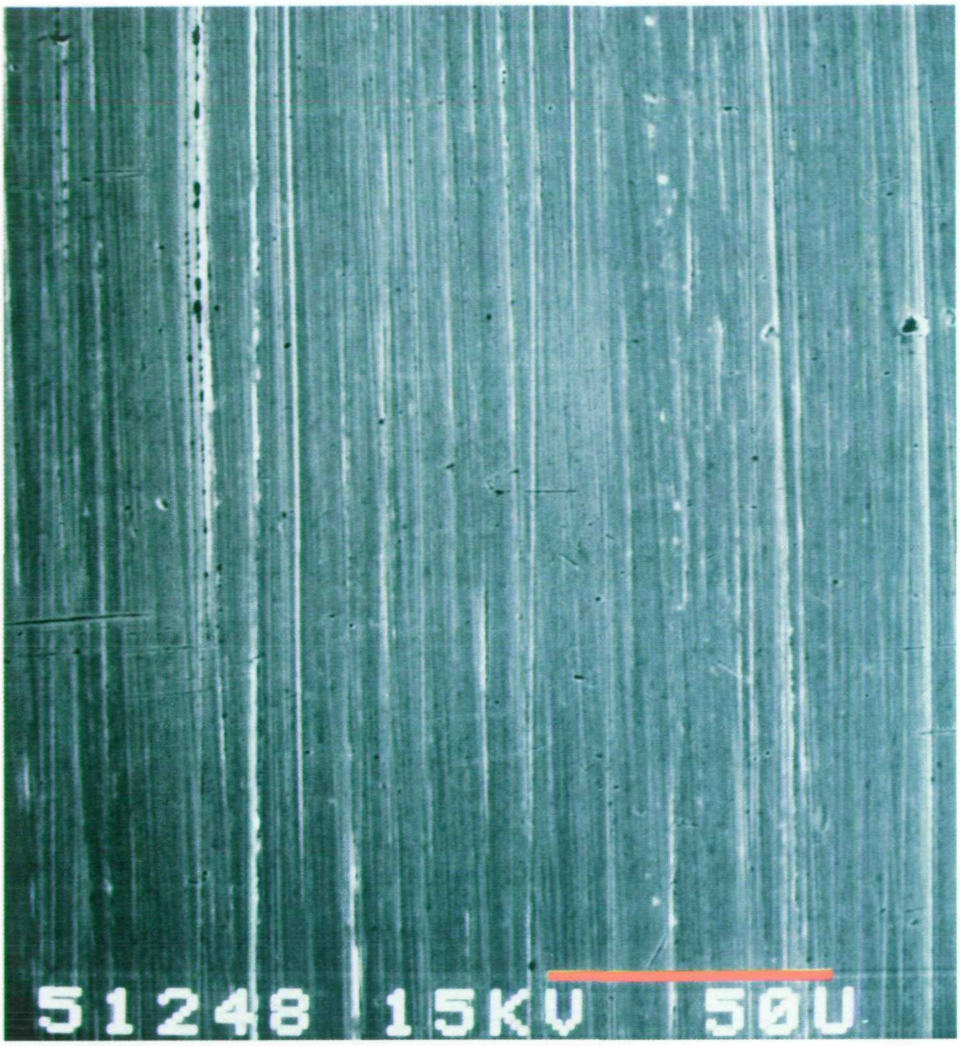


Fig 11 R2 test machines equipped for testing with contaminated lubricant



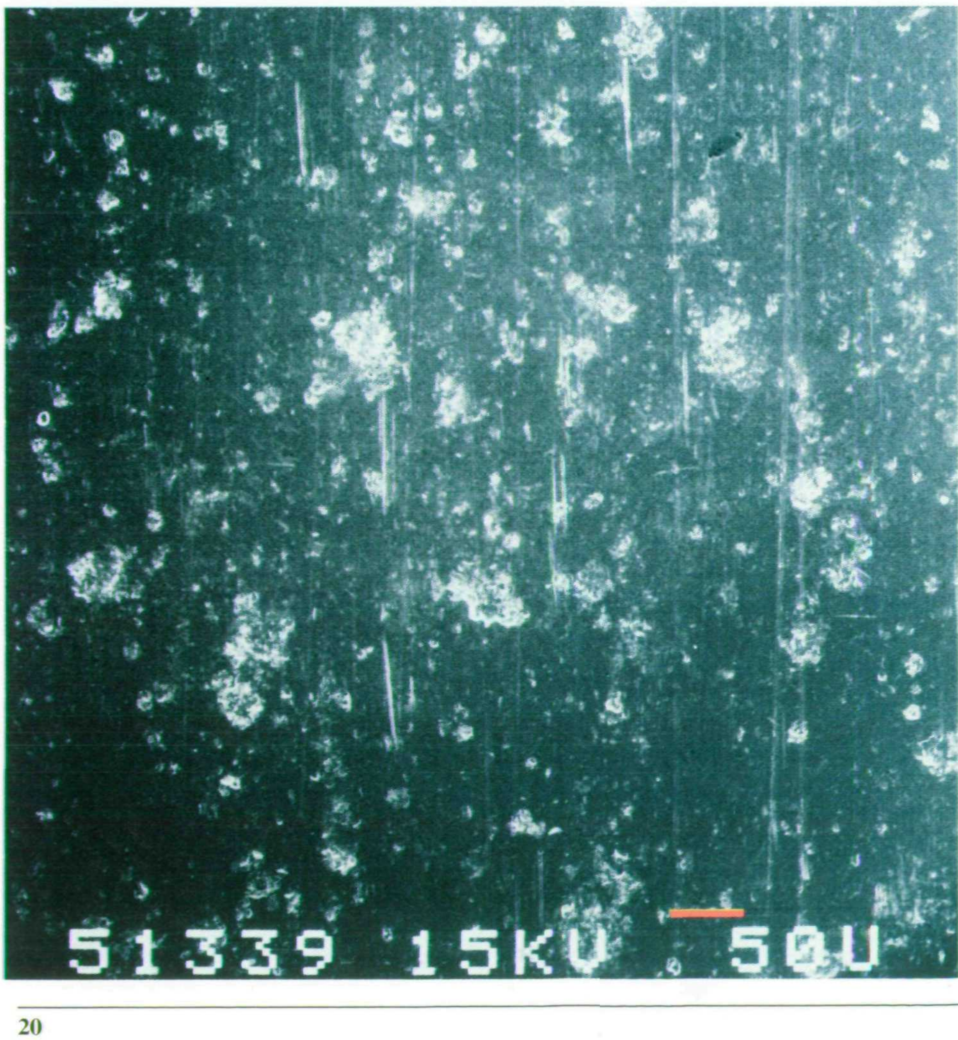


Fig 12 Scanning electron microscope pictures of new raceway surface (upper) and raceway surface after running in with contaminant (lower). The red line represents 50 μm

environments. Two methods are under evaluation:

In the first case, bearings are tested with Air Filter Fine Test Dust added to the lubricant (fig 11), i.e. the contamination is mostly silica and is continuous. Particle counts show that the size of a particle reduces during running, the particles being crushed in the bearing contact. To date, this type of testing has produced virtually no spalling failures, the bearings continuing to function up to 600 million revolutions, when the test was stopped. There is, however, a noticeable increase in bearing vibration and post-test investigations show wear on the contacting surfaces, indicated by weight loss and an increase in internal clearance. A possible reason for the absence of spalling-type failures is the continued action of the hard silica particles grinding away possible initiation sites before they can develop into spalls.

In the second case, bearings are first run in for half an hour in lubricant contaminated with graded contaminant taken from gearboxes (fig 12). The contaminant in this case is mainly of a metallic nature. The bearings are then carefully cleaned and life tested under clean, fully flooded lubrication conditions. With these tests, spalling failures did occur and were all identified as being related to overrolling indentations. This result was predicted in studies made at Imperial College in London and at SKF-ERC; see [4, 5, 6 and 7] which provided the background for the tests described above.

This method has been used to investigate the difference in bearing life which can be expected when using various materials and material treatments in manufacture. The results obtained are being used in the development of bearings which are resistant to the effects of contamination (fig 13).

The development of endurance test facilities and methods is a continuing activity directed towards never-ending improvements in the performance of SKF bearings.

Conclusions

We have briefly discussed some of the bearing performance test facilities of SKF at ERC in The Netherlands. These, together with the additional test facilities available at other SKF units, underline the SKF commitment to the development of advanced products and to never-ending improvement of the performance quality of its products. Performance can be defined in many ways, but basically good bearing service performance is characterised by the ability to rotate with minimum friction loss and noise, an acceptable service life and a safe failure mode. To provide such good service performance, an integrated approach to design, testing and manufacturing is required and the testing facilities at ERC provide the necessary test support with their wide range of rigs and equipment in both Mechanical and Endurance Testing. It is also this background that allows new concepts in bearing technology, like the new SKF life theory, to emerge and be verified.

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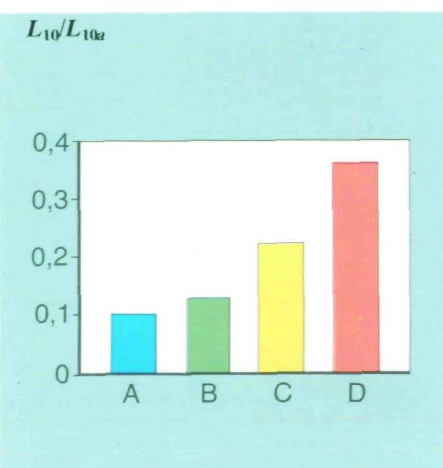
Fig 13 Life test results for deep groove ball bearings 6205 run in with contaminant; A to D represent different materials/treatments



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Dirty lubricants – reduced bearing life

Introduction

In the original work by Lundberg and Palmgren [1, 2], the theoretical calculations of probability of failure required material parameters to put into the model equations. These were taken from numerous experiments with both ball bearings (fig 1) and roller bearings. All these experiments were performed under the cleanest laboratory condition which could be managed, therefore it was assumed that there was no influence on bearing lives from dirt particles or surface roughness. However, the filtration level for the lubricants in those experiments where filters were used was no better than 10 μm absolute; thus, some of the failures were certainly caused by dirt particles and asperity interaction, giving surface initiated failure.

The ball bearing tests carried out until 1963 were all grease lubricated, therefore wear particles and dirt were not easily removed from the bearings. What were then called 'clean conditions' were used as a standard to give the nominal life of well-lubricated bearings. How well "well-lubricated" bearings were lubricated was not defined from the beginning, but from the 1960s when elastohydrodynamic theory became available [3, 4, 5], well-lubricated bearings had an oil film thickness to composite surface roughness ration of $\lambda = 2$ to $\lambda = 4$. The ball bearings included in Lundberg's and Palmgren's investigations had fine enough surfaces to obtain λ values of this order, but the old roller bearings had coarser surfaces: thus the major part of the old tests seems to have been carried out at λ values of around 0,5. Today we know that such a low λ value decreases the life of the bearing from its nominal value.

Over the years, the load-carrying capacity of rolling element bearings has increased. This has mainly been assumed to depend on the cleaner steels and the finer surfaces of the bearings, but at the same time the filtration of the lubricants used in the tests has become better and better. This means that some of the contribution to longer lives has come from the cleaner lubricants.

Today we know that all hard dirt particles larger than the oil film thickness decrease the bearing life. In rolling element bearings, the oil film thickness is normally of the order 0,1 to 3 μm, thus filtration should be kept at a similar level to obtain long bearing lives.

In all normal endurance tests, lubricants are virtually free from water. In outdoor applications and when the machines contain water or steam, the lubricant is very easily contaminated with water. It is not known why, but it is well-known that free water in a lubricating oil decreases the life of rolling element bearings by ten to more than a hundred times, depending on the water content.

E. Ioannides
SKF-ERC
The Netherlands

B. Jacobson
SKF-ERC
The Netherlands

It is obvious that the cleaner the oil, the better. Even dirt particles much smaller than the mean film thickness give wear if they are hard. If they are large enough to penetrate the oil film thickness, they give local stresses at the surface and thereby shorten the life of the bearing considerably. Such stresses can now be modelled theoretically and with the use of the new life theory predictions of the life reduction in the presence of contaminants can be provided.

Water can decrease bearing life. A concentration of water of 0,01 % is enough to decrease the bearing life to half its original value.

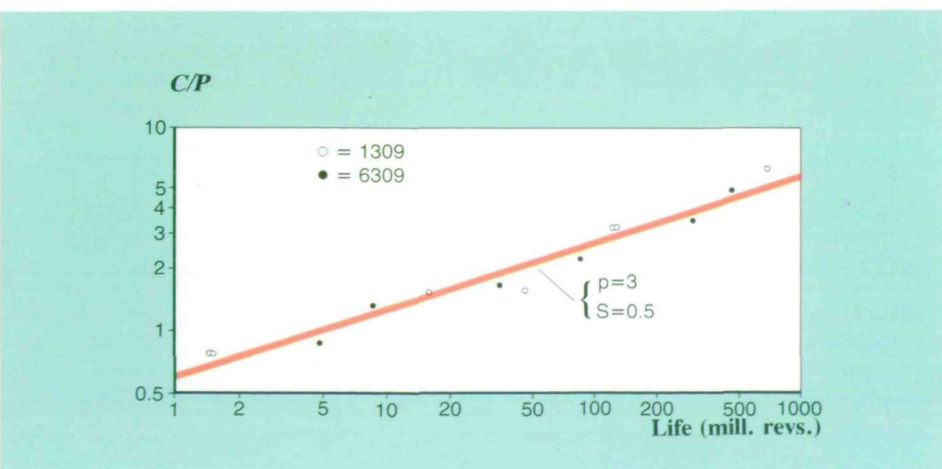


Fig 1 Experimentally determined relationship between *C/P* and *L* for ball bearings 1309 and 6309 from which the value of exponent p in the life equation was obtained; *S* = probability of survival [1]

Review of published papers

The earliest papers on contaminated lubricants for rolling element bearings dealt with water contamination. In 1968 and 1969 [6, 7], Schatzberg and Felsen published two papers on the effect of dissolved water on rolling element fatigue life. Under high stress conditions and in a laboratory environment, the presence of 100 ppm (0,01 %) water dissolved in the squalane lubricant decreased the fatigue life by 32 to 48 %.

Later in 1969, Yardley and Crump [8] showed some results from grease lubricated rolling element bearings in which they stated that dirt introduced into the bearing by the grease packing procedure, or through the grease channels on re-greasing, could rapidly destroy the bearing.

In 1971, Felsen, McQuaid and Marzani [9] studied the effect of sea water on flood-lubricated angular contact ball bearings. The fatigue life behavior of different lubricants was very similar. When the lubrication was good and $\lambda > 1,5$, the life of the bearings was about ten times higher than when $\lambda = 0,5$. For both lubrication regimes $\lambda = 1,5$ and $\lambda = 0,5$, the introduction of 0,1 to 1 % seawater decreased the L_{50} life by a factor of about five and the L_{10} life to half its value. So a consequence of the addition of seawater was a general increase in Weibull slope with increasing amounts of dissolved water. This was also accompanied by a change in failure mode, from ball failures to raceway failures, when the water content increased.

In two papers from 1975, Fitzsimmons together with Cave [10] and Clevenger [11] analysed the wear of taper roller bearings as a function of the amount of contaminant in the lubricant. They found that:

1. The wear was proportional to the amount of contaminant.
2. Taper roller bearings will continue to wear as long as the particle size of the contaminant is larger than the lubricant film thickness between the bearing surfaces.
3. For bearings to wear significantly, the contaminant particle hardness has to be greater than or equal to the hardness of the bearing material.

In their papers, they also refer to an earlier paper by Okamoto et al. [12] where they found life reductions of 80–90 % when bearings were contaminated with different amounts of particles of various hardnesses.

In a paper published in 1976 [13], Tallian describes an experimental investigation of rolling contact fatigue life for contaminated deep groove ball bearings.

The importance of cleanliness for bearing life is demonstrated by three groups of experiments, where the first group is ultrasonically washed, greased and sealed. The second group is kerosene washed, greased and open. The third group is sump lubricated with oil artifically contaminated with powdered hardening scale. The relative lives of the three groups were 100 %, 23 % and 2 % respectively. For the sump lubricated group a differential gear oil with 5 mg of powdered heat treatment scale up to 600 μm platelet diameter was used which gave a reduction in life of 1 : 50 or more.

On the other hand, Tallian shows that extreme cleanliness pays off, giving 15-30 times longer life than expected from normal calculations. Before this was published, Dalal et al. [14] carried out some preliminary tests under ultraclean conditions where the lubricating oil was filtered through a 3 μm Millipore filter. The five bearings in the test ran more than 40 times their theoretical L_{10} lives without failures. Under standard test conditions those bearings were known to have 4-5 times their theoretical L_{10} life. The significant increase in life observed demonstrated the beneficial effect of lubricant cleanliness on bearing life.

To investigate the influence of a few hard particles in the oil, Dalal and Senholzi [15] performed experiments with deep groove ball bearings and taper roller bearings, where they had made Vickers hardness indentations on the inner ring surface. The diagonal dimensions of indentations were 150–153 μm as compared to the Hertzian contact widths of 760 μm and 230 μm respectively for the ball bearings and the roller bearings. Seven of the fourteen ball bearings failed at the Vickers indentation, five failed elsewhere and two were suspended. The life of the indented ball bearings was about 7 times the L_{10} life as compared to more than 40 times the L_{10} life for the non-indented bearings in the earlier investigation.

In an interesting paper from 1979 [16], Loewenthal and Moyer describe laboratory experiments with deep groove ball bearings lubricated with an uncontaminated and an artificially contaminated lubricant.

In the tests with uncontaminated lubricant, the lubricant was circulated through a 49 μm absolute filter. This means that a lot of particles produced within the system were circulated through the bearings. Despite this the 10 % life for the test bearings was 672 hours, whereas the AFBMA standard 10 % life was 47 hours.

When test dirt was added to the oil at a rate of 125 mg per bearing per hour, the L_{10} lives were not changed very much for oils filtered through the 3 μm and 30 μm filters, but the L_{50} lives were decreased. As soon as dirt particles were introduced into the lubrication system, the Weibull slope increased considerably and the surfaces of the balls and raceways started to show progressive surface damage.

The same year, Perrotto, Riano and Murray [17] published an investigation on the effect of abrasive contamination on ball bearing performance. They did not study the life of the bearings, but how hard contaminants of different particle size affected the oil film separation of the bearing surfaces. The particles were of two materials and four sizes: silicon dioxide (SiO_2) particles, 50 μm in diameter and aluminium oxide ($A1_2 0_3$) particles with diameters of 3, 0,3 and 0,06 μm. The calculated oil film thickness for the tested deep groove ball bearings was 0,33 μm between the balls and the inner raceway, when they were running at 1700 r/min. This means that it should be possible to have the two finest grades of aluminium oxide particles in the oil film without causing damage to the surfaces.

When the tests started, the bearings were run in for 70 to 80 hours in clean oil until all electric contact through the oil films disappeared. Then new test oil with contaminants was injected into the circulating system and the bearing run with the dirty oil for one hour, after which the oil was drained out and a fresh charge of clean oil put into the system. The test was then continued to determine if the bearing could run in again. The purpose of this step was to obtain some qualitative measure of how badly the bearing surfaces had been damaged by the abrasive particles. Perotto et al. found that bearings exposed to particles larger than the elastohydrodynamic lubricant (EHL) film thickness (3 μm and upwards) showed no signs of running in again. When the dirty oil was replaced with clean oil, there was no change in the percentage of electric contact through the oil film. Even after more than two days of running with clean oil, there was no indication of run-in. Only when particles smaller than the EHL film (0,3 and 0,06 μm) were used did there seem to be a true recovery. The bearing contaminated with 0,3 μm particles returned to its well run-in state in less than 24 hours, and for the bearing contaminated with 0,06 μm particles, the recovery time was about 15 hours. One observation made by the authors was that the particles seemed to be crushed as soon as they came into the EHL contact, thus the damage from dirty oil stopped as soon as the particles were crushed down to a small enough size. Obviously, this size was much smaller than the oil film thickness, because even the 0,06 μm particles showed the same behaviour, despite the six times larger oil film thickness. Independent of particle concentration, all particles larger than the oil film thickness caused permanent damage to the bearing surfaces before they were crushed, and the 3 μm Al_2O_3 particles, or fragments of them, became embedded in the steel surfaces.

Under clean conditions, bearing life normally increases with steel hardness. In a paper published in 1981, Sugiura et al. [18] state that a heat treatment which decreases the crack sensitivity of a bearing steel can increase the rolling contact fatigue life significantly, especially for debris-contaminated lubricants.

In a continuation paper to their 1979 work, Loewenthal, Moyer and Needelman [19] studied the effect of ultraclean and centrifugal filtration on rolling element bearing life. This time they used the 3 μm absolute filter in the circulation system to remove any debris generated by the system itself. By doing so they increased the L_{10} life for the bearings by a factor of two as compared to the contaminated case with 3 μm filtration and the L_{10} life increased 64 % relative to the baseline tests with a laboratory-clean lubrication system. They also investigated the wear rate as a function of the filtration. The non-failed bearings in the baseline test with clean oil and 49 μm absolute filtration had 3,9 times the weight loss of the bearings from the ultraclean tests whilst the bearings from the centrifugal filter, 3, 30, 49 and 105 μm filter tests with contaminated oil had 6,6, 7,4, 12,5, 16,2 and 344,6 times the weight loss of the ultraclean test bearings.

As the wear rate is so high when the bearings are lubricated with contaminated oil, it is not only the filter rating which is important, but also how big a flow of oil per time unit the filter can accept. The filter must be able to catch at least as many particles per hour as the amount produced by bearing wear.

In a paper from 1982, Sayles and Macpherson [20] found that the early failures determining the L_{10} lives for contaminated bearings were mostly surface initiated. They experimented with standard 25 mm bore, extra light series, single row cylindrical roller bearings with a brass cage. The contamination of the lubricant was produced by a helicopter gearbox lubricated with the same oil as the test bearings. The oil with the gearbox debris was pumped through different filters to the bearing test rig. The cartridge-type filters were of absolute ratings 40, 25, 8, 3 and 1 μm; figs 2 and 3. Sayles and Macpherson found that oil film thickness had a minor influence on bearing life, but the filter rating had a much bigger influence. The finer filters always gave longer bearing lives, typically 6 times longer L_{10} life with a 3 μm filter than with a 40 μm filter. The most striking of their results is that surfaces damaged by dirt particles in the oil do not recover if the oil is cleaned after a short period. They made pre-runs with 40 μm filters for 30 minutes and then the rest of the test was run with 3 μm filtration. Despite the fine filtration during the main part of the test, the bearings failed just as early as if they had been lubricated with the dirty oil all the time. The damage done to the bearing surfaces by the dirt particles during the first half-hour was enough to cause early failure. These early failures were in the form of transverse cracks perpendicular to the bearing surface having a depth of 0,20 to 0,75 mm. This indicates that the mechanism of failure is associated with a change in surface contact properties rather than with the continued presence of a third body.

Another influence of wear is the redistribution of forces and stresses in

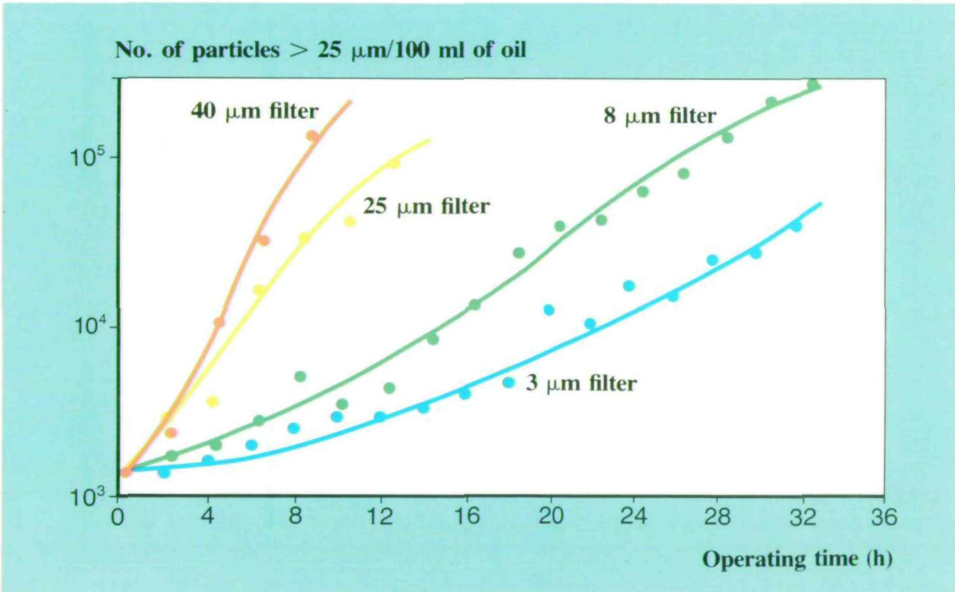


Fig 2 Rate of particle generation at various filtration levels [20]

Fig 3 Bearing life (L_{50}) as a function of filter size [20]

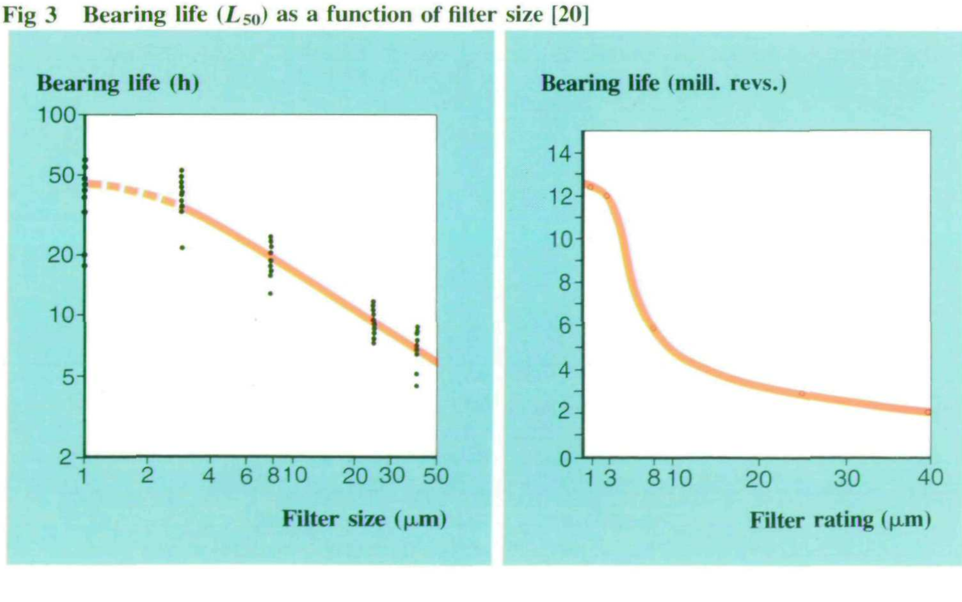
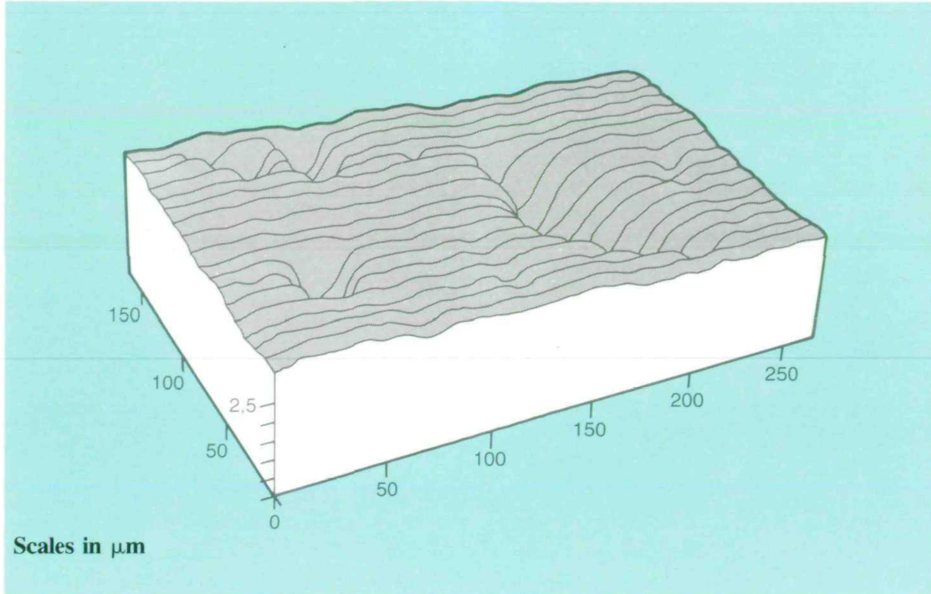


Fig 4 3-D isometric view of several debris dents formed under 40 µm filtration

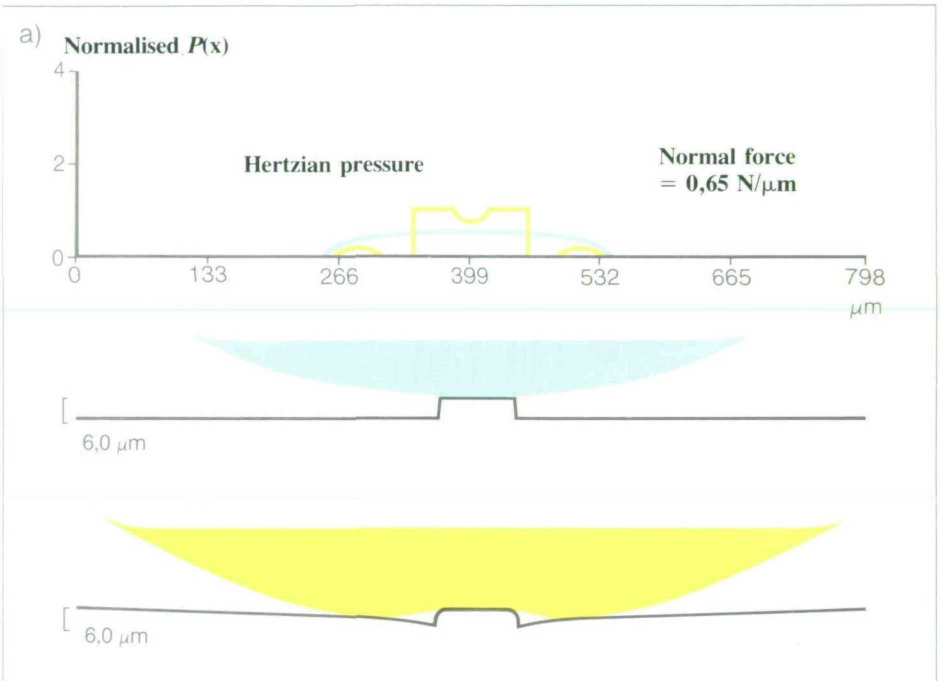


the bearing. In 1983, Lorösch [21] showed that experiments with worn bearings always gave pitting failure in the non-worn parts as the contact pressures there were higher. He also showed that under ideal lubricating and cleanliness conditions, the bearing life increased with decreasing load to a much greater extent than obtained by standard calculations.

To investigate the influence of dirt, Lorösch made small indentations in the rolling track with hard balls of different sizes. The steeper the edges of the indentations, the shorter the life of the bearing. When a 0,4 mm ball was pressed into the surface to a depth of 13 µm, the life of the bearing decreased with 98,5 % to 1,5 % of the life without indentation. His experiments showed that the sharp indentations never disappeared, but stayed and produced high stress concentrations for the rest of the bearing life.

When it comes to wear, the situation can be slightly different. Skorynin and Minchenya [22] showed in their paper from 1984 that the wear rate was very much dependent on the particle size for hard particles. When they added 10 ppm of electrocorundum particles, 5 µm in size, to the lubricating oil, the balls wore more than twice as much as the inner ring, outer ring and cage together because these particles were embedded in the plastic cage surface.

If one of the rolling element surfaces is much harder than the other, the result will be similar if the λ value is low. In their 1985 paper [23], Ishibashi, Hoyashita and Sonoda studied the influence of different hardnesses for the two surfaces working together. The surface roughness of the two roller surfaces was 3 µm R_{max} and the calculated oil film thickness was about 1 µm for the load range 1,5 GPa to 2,8 GPa. When a 520 HB roller was combined with a case hardened 800



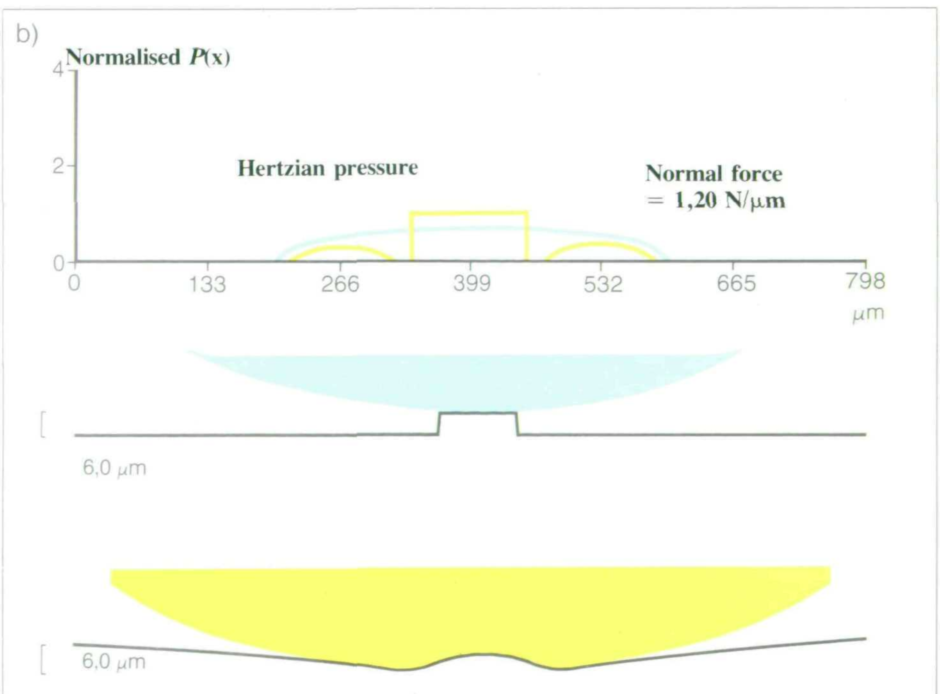


Fig 5a The numerical solution for a frictionless elastic/plastic contact of a smooth steel roller against a hypothetical stepped flat surface simulating the presence of debris. The top diagram shows the pressure distribution whilst the two lower ones show the original configuration and the resulting contact geometry. The pressure is normalised over hardness

Fig 5b As Fig 5a, but with an approximate doubling of normal load

HV roller at 1,5 GPa pressure they did not wear in, but metallic contact continued for 800 000 revolutions, after which the surfaces had pitting failures. In contrast to this, when 525 HB rollers with the same roughness of 3 μm R_{max} were used in equal hardness combinations, the surfaces wore in so that 90 % of the time there was no metallic contact and they could be rotated $6,5 \times 10^6$ revolutions at a contact pressure of 2 GPa without failure. For rollers with a mirror-like finish of 0,2 μm R_{max} and equal hardness, the metallic contact stopped after 10^6 revolutions and no pitting occurred within 10×10^6 revolutions, even at 2,4 GPa pressure. Their observation of metallic contact through the oil film for the first million revolutions is very interesting as the λ value for the mirror-like rollers was about 15. This means that it does not matter how fine the surfaces are ground, run-in can improve them. This also means that wear particles are always produced in new machinery and should not be allowed to approach the elastohydrodynamic contacts.

From the above review, it is obvious that the water and dirt content in the rolling element bearing lubricant should always be as low as possible. Even minute amounts of water of the order per mille and dirt particles a few ppm are enough to considerably decrease the life of bearings. If, on the other hand, it is not possible to reach those low contamination levels, life is always increased if the dirt level is lowered. In two papers [24, 25] dealing with sealed rollneck bearings for steel rolling mills, the sealing is not very successful. In the lubricating grease, the typical water content level reached is about 10 % and the hard particle content is 1 %. This is, however, so much better than the previous water content which reached 40 %, that the number of catastrophic failures has decreased from several per year to zero between the yearly inspections.

Theoretical results

In addition to the experimental work described above, theoretical calculations have been published in recent years in which an attempt has been made to predict the life reduction caused by contamination.

Previous experimental work [20] indicates that, in certain conditions, dents, formed by the overrolling of debris, act as sites of stress concentrations and as such are responsible for the initiation of surface fatigue associated with a life reduction of the bearings. This was particularly true in [20] where a gear machine produced the debris, predominantly in the form of ductile particles, which, in turn, produced smooth dents, fig 6. With the advent of the New Life Theory [26] it was possible to utilise stress information in the locality of the dents to predict the reduction of the fatigue life of the rolling bearings. In 1985, Webster, Ioannides and Sayles [27] published such calculations in which a number of dents from the tests described in [20] were selected and mapped three-dimensionally (3-D); fig 4. The information contained in these topography measurements was subsequently used in conjunction with a numerical contact model, coupled to a sub-surface finite element analysis, to evaluate the stresses in the bearing raceways in the presence of the dents. Then, as indicated above, these stress fields were used to calculate life reductions resulting from the presence of the dents.

In the event, the predicted life reduction was in reasonable agreement with that observed in the tests described in [20]. Since then, additional work has been published resulting from the collaboration between Imperial College and the SKF Engineering & Research Centre. First, Sayles and Ioannides [28] examined the effect of debris type and geometry in relation to the formation of dents.

The concept of elastic conformity around entrained debris was studied and it was shown that a critical debris aspect ratio may exist beyond which no damage on the contacting surfaces occurs. This is because the surfaces close elastically around the debris and further loading is taken up elastically by them, figs 5a, 5b. In a sequal to this work, Hamer, Sayles, and Ioannides [29] studied the debris and raceway deformation modelled as an extrusion process. The effects of the film thickness, the raceway and particle hardness and the particle size were introduced in the analysis and combinations of these parameters define a boundary between debris which can damage the contact surfaces and debris which cannot. An example of these regions for a particular configuration is given in fig 6. This work was experimentally verified by the same authors [31] when particles of different hardness were pressed between two anvils and the measured dents were compared to the predictions of [29].

Conclusions

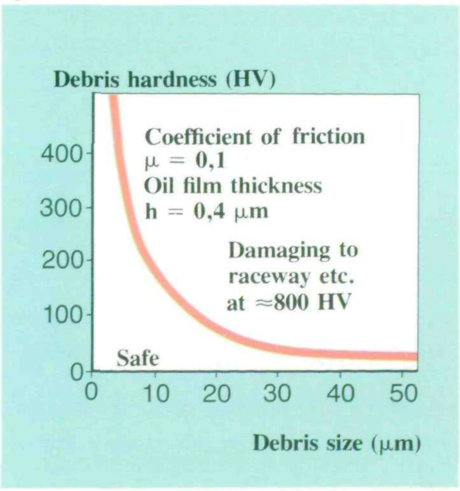
When considering oils contaminated with dirt particles, it is obvious that the cleaner the oil, the better. Even dirt particles much smaller than the mean film thickness produce wear if they are hard. If they are large enough to penetrate the oil film thickness, they give local stresses at the surface and thereby shorten the life of the bearing considerably. This was already demonstrated in 1972 [31] when Leibensperger and Brittain measured the stresses below asperities in Hertzian contacts using photoelasticity. Such stresses can now be modelled theoretically and with the use of the new life theory, predictions of the life reduction in the presence of contaminants can be provided.

This effort is now continuing together with studies of the denting process itself, so that complete answers can be provided to the selection of economical levels of filtration in real applications.

Finally, it is not well understood why water can decrease bearing life as much as it does but a concentration of water as small as 0,01 % is enough to decrease the bearing life to half its original value. The reduction of bearing life with water concentration is steepest when the water can be dissolved in oil, but even at high concentrations, more water gives shorter life. This means that it always pays to keep the dirt and water content in the lubricant as low as possible.

Reg: 74

Fig 6 Mapping of safe/damaging regimes of debris size and hardness combinations
Particle hardness:
mild steel ≈ 130 HV
pure aluminium ≈ 25 HV
annealed copper ≈ 40 HV
cold drawn brass ≈ 180 HV
plastics ≈ 1–10 HV



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Market-oriented product development as exemplified by SKF TQ-line taper roller bearings

F.-J. Markert
SKF Schweinfurt

All further development of rolling bearings must be based on a knowledge of customer demands and market trends. In improving taper roller bearings, SKF has carefully analysed the demands and designed the new product accordingly. The emphasis in the development programme has been placed on improving the contacts between the rollers and the guide flange, raceways and cage. The resulting design, known as TQ, can, without exaggeration, be said to constitute a new performance class.

The operational reliability of machines depends on the functional dependability of the rolling bearings incorporated. Bearing designs which meet user demands, together with high quality manufacturing methods, are prerequisites for the satisfactory operation of many sophisticated machines. If the steadily increasing operational demands placed on plant, gearboxes and other machinery are to be met in the future a systematic rolling bearing improvement programme is required. An example of this form of development is the SKF TQ-line of taper roller bearings which has considerably extended the usefulness of this type of bearing. The product meets customer demands, even those for very critical applications, with further potential in reserve.

The challenge of customer demands

The task of a development engineer is to ensure that the bearings not only meet the demands of today and tomorrow, but also those of the day after tomorrow. It is therefore necessary to understand user demands and market trends and to use these as a basis for new development, or modification work. An incorrect judgement of these factors could easily lead to developments which fail to meet the market's needs.

Before the main goals were defined by SKF for its development work on taper roller bearings, the market situation was analysed and customers were asked to specify demands and market trends in the various application areas. The most important requirements were as follows:

— high reliability,

— longer endurance and service lives,

— extended maintenance intervals,

— reduced friction,

— reduced operating temperatures,

— reduced operating noise (and vibration),

— reduced costs,

— increased insensitivity to adjustment errors,

— increased insensitivity to misalignment,

— elimination of heavy wear and seizure.

These demands are largely the same as those specified for other types of rolling bearings. It was surprising, however, to note the requirement for low vibration and noise, as this had previously only been specified for ball bearings and not for taper roller bearings. In addition to the technical properties, bearing prices are important to the customer, although it is the total cost of a bearing arrangement over its operating period, rather than the purchase price, which is decisive.

Main goals in development of TQ taper roller bearings

The above demands from customers were translated into the language of the bearing development engineer and the following goals were defined, fig 1:

— improvement of the roller/flange contact,

— improvement of the roller/raceway contact,

— improvement of the roller/cage contact,

— optimum exploitation of the bearing cross section, and

— suitability for collective adjustment.

As a result of the development work in the various areas mentioned SKF has introduced the TQ taper roller bearing. This is something more than just an improved taper roller bearing; in fact, it represents a new performance class. New research results and new manufacturing technology have paved the way for the new generation of taper roller bearings.

Improved roller/flange contact

For kinematic reasons, rolling is impossible in the contact area between the guide flange and the roller end. If the friction torque of a correctly designed plain bearing is plotted against the rotational speed the familiar Stribeck curve will be obtained, with its well-defined minimum which marks the start of hydrodynamic lubrication at a relatively low speed, see fig 2.

This characteristic curve will also be found for taper roller bearings, when the roller end/flange contact has an ideal form and is suitably lubricated. Fig 3 shows the components of the friction torque:

M_1, the load-dependent frictional moment at the roller/raceway contact,

M_0, the speed-dependent frictional moment at the roller/raceway contact produced by viscous friction in the EHD film, and

M_2, the speed-dependent frictional moment at the roller end/flange contact with its typical Stribeck form.

With the conventional design of roller end flange, it is impossible for a hydrodynamic lubricant film to form.

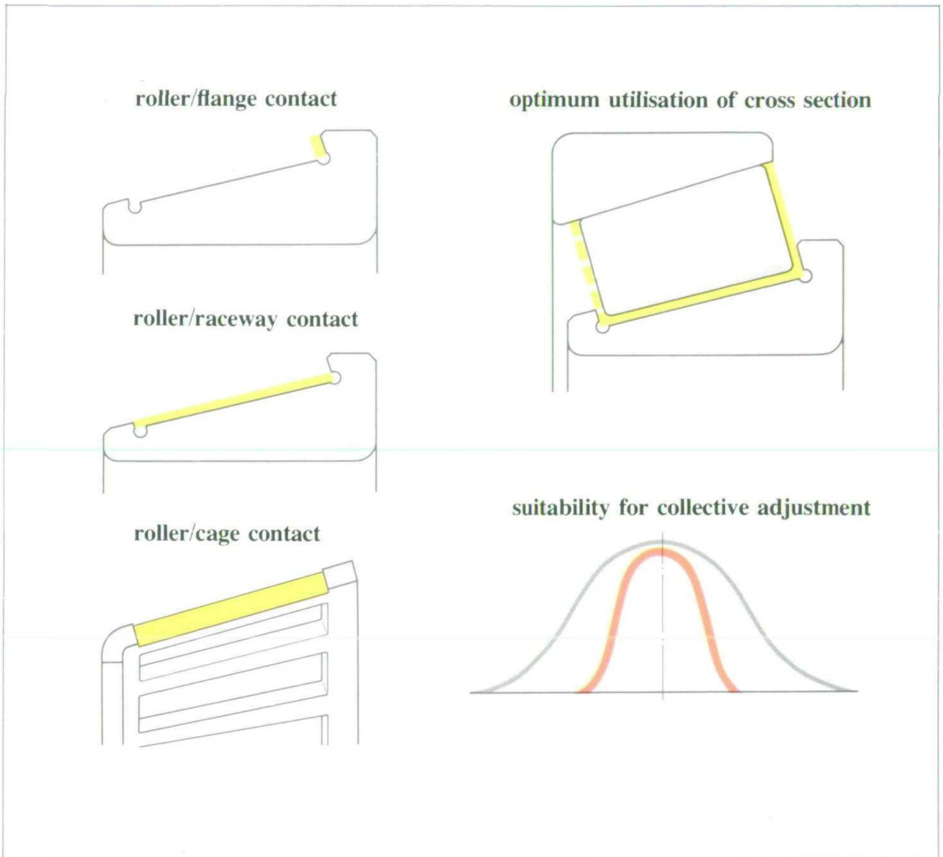


Fig 1 Main aspects of development work in design and manufacture of modern taper roller bearings

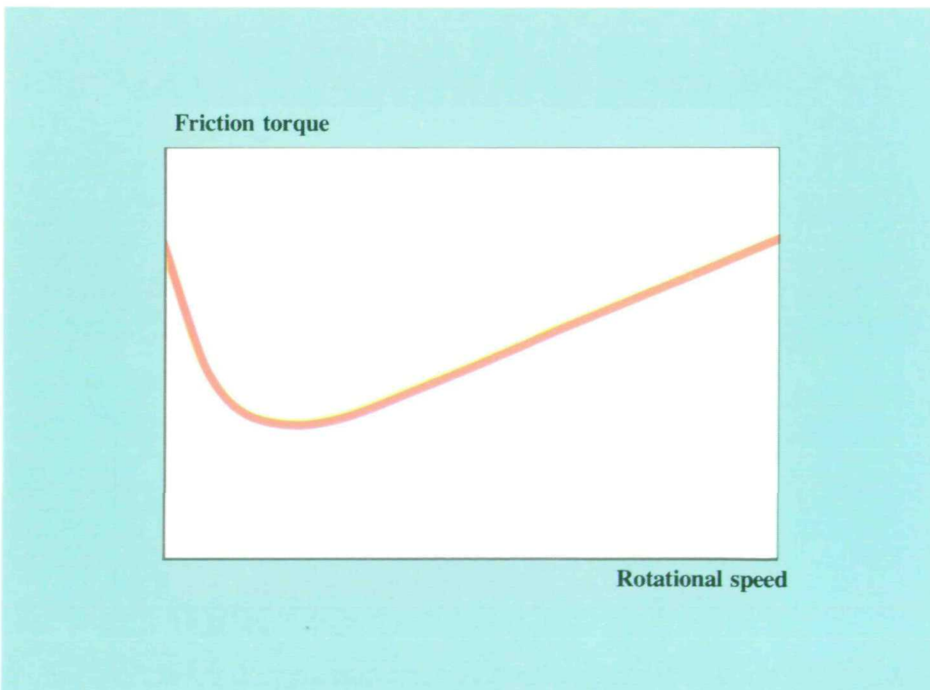


Fig 2 Characteristic friction torque curve for a plain bearing (the 'Stribeck' curve)

This accounts for the high proportion of mixed friction M_2, fig 4. Conventional bearings can reach the optimum lubrication condition, but only after a long and critical running-in process. During this phase, which takes the form of a mild wear process under moderate loads, the macro and micro profiles of the surfaces in contact eventually approach the optimum. The prerequisites for hydrodynamic or elastohydrodynamic lubrication conditions will then exist.

The running-in of conventional bearings, however, has considerable disadvantages. Reduced load, coupled with careful, controlled lubrication and monitoring are required for many hours of operation and, in spite of this, the running-in phase still involves a considerable risk. Each brief load peak, each contaminant particle and any lack of lubricant supply can drastically affect the "controlled" mild wear process—resulting in heavy wear, hot running and smearing, with the attendant risk of seizure.

Even when the critical running-in phase is completed without noticeable damage, the wear will have changed the adjustment of the bearings and this must be taken into account. The preload force will be reduced or the clearance increased. Additionally, wear debris will contaminate the lubricant and accelerate further wear of the bearing; any dents produced in the heavily loaded raceways will lead to premature fatigue.

Therefore, there are important reasons, from the bearing user point of view, to improve the roller/flange contact. This can be achieved in the following ways:

— by reducing the running-in period, the failure risk will be reduced;

— by reducing the geometrical changes during the running-in phase, the adjustment can be kept constant;

— by making the bearings capable of supporting the full load right from the outset and eliminating the time-consuming and critical running-in phase;

— by reducing the sensitivity to any errors of adjustment, mounting or service.

These customer demands are easily met by the TQ taper roller bearings from SKF. Fig 5 shows the friction torque/speed curve for the new SKF bearings compared to conventional bearings. The friction in the TQ bearings is appreciably reduced: it has been possible to develop bearings with a small proportion of mixed friction and a higher proportion of hydrodynamic lubrication in the roller/flange contact area from bearings having a higher proportion of mixed friction and boundary lubrication.

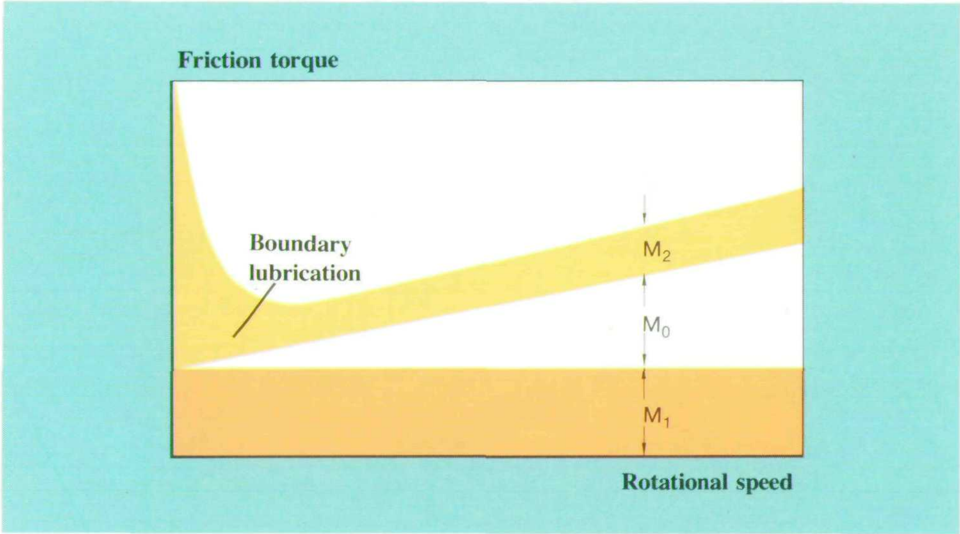


Fig 3 Friction torque components in an optimum taper roller bearing
M_1 = load-dependent frictional moment at the roller/raceway contact
M_0 = speed-dependent frictional moment at the roller/raceway contact
M_2 = speed-dependent frictional moment at the roller/flange contact

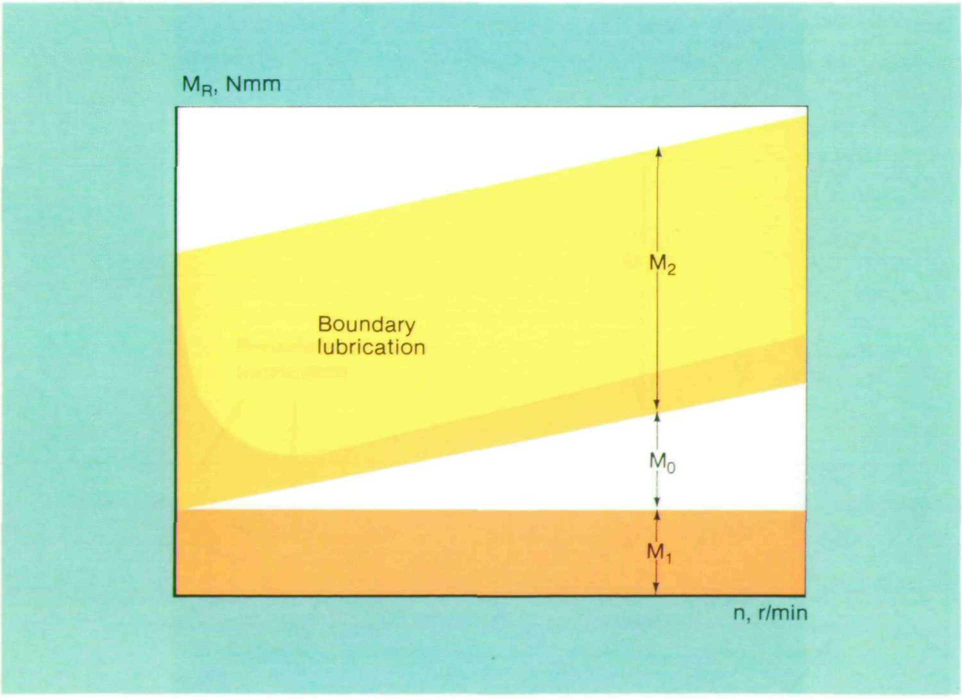


Fig 4 Friction torque components in a conventional standard taper roller bearing
M_1 = load-dependent frictional moment at the roller/raceway contact
M_0 = speed-dependent frictional moment at the roller/raceway contact
M_2 = mixed frictional moment at the roller/flange contact

Where special application requirements exist, the TQ bearings can also be supplied in two special designs: CL7A and CL7C. The special roller/flange contact of the CL7A design provides a controlled and small mixed friction proportion, which guarantees a friction torque only slightly dependent on speed, as well as very little scatter in the friction torque. This frictional behaviour is desirable when taper roller bearings are to be adjusted in line production, e.g. for cars.

The CL7C design has the pronounced minimum of the typical Stribeck curve and, therefore, has hydrodynamic lubrication right from the outset, with all the attendant advantages such as low friction, low operating temperature, low wear and long life. CL7C bearings are therefore used where operating conditions are difficult, e.g. where full load is applied at commencement, where stresses are high, for high-speed operation and where lubrication conditions are unfavourable.

The great advantage of bearings having hydrodynamic or controlled mixed friction can be clearly seen from the temperatures recorded during the running-in phase, see fig 6. Whilst conventional bearings show high temperature peaks, there is practically no peaking for the optimised bearings and they quickly reach their steady-state temperature. The use of TQ bearings therefore eliminates the risk of hot running and seizure which would otherwise be present during the first hours of operation. This contributes significantly to the bearings' operational reliability. The CL7C bearings can sustain twice the load of normal bearings before the torque peaks and smearing occurs in the rolling/flange contact area. Consequently, the new bearings will have an adequate safety margin in respect of smearing and seizure in cases where they have been incorrectly adjusted, e.g. using a greater preload force than that specified, even though conventional bearings might have been over-stressed. This means that the TQ

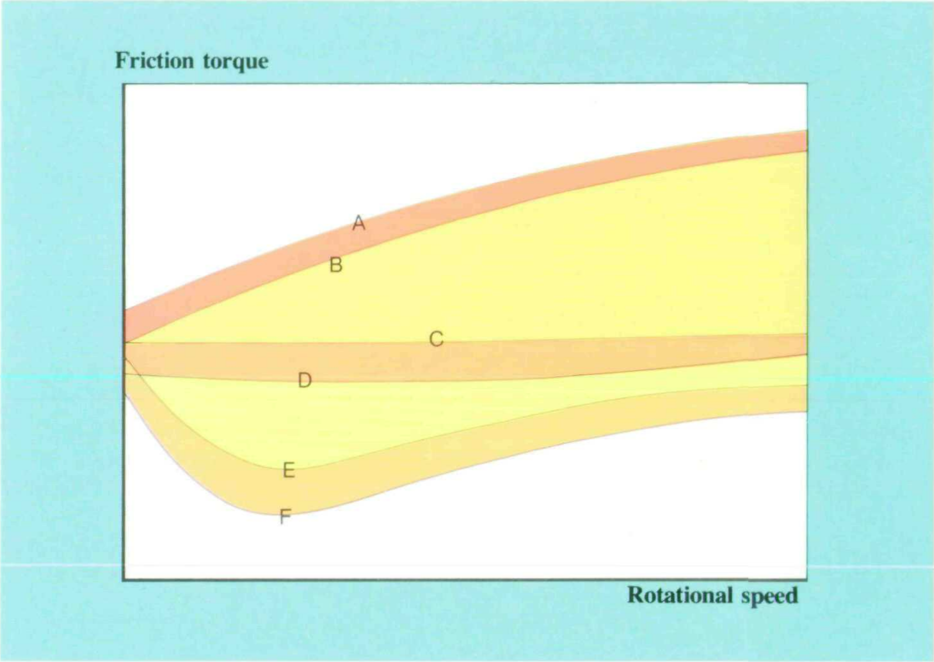


Fig 5 Friction torque of taper roller bearings as a function of speed
A–B conventional taper roller bearings with unfavourable roller/flange contact
A–C conventional taper roller bearings
C–F TQ-line taper roller bearings
C–D TQ-line taper roller bearings, CL7A design
E–F TQ-line taper roller bearings, CL7C design

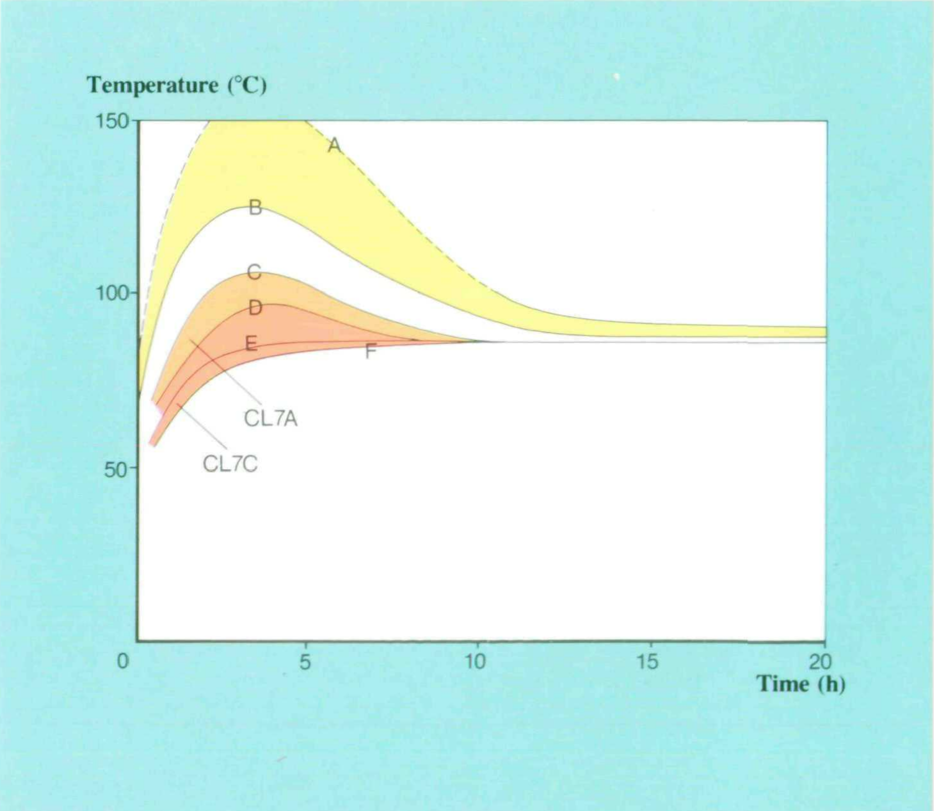


Fig 6 Operating temperature of various taper roller bearing designs during the running-in phase
A–B conventional taper roller bearings with boundary lubrication at the roller/flange contact
C–F TQ-line taper roller bearings
C–D TQ-line taper roller bearings with controlled mixed friction at the roller/flange contact, CL7A design
E–F TQ-line taper roller bearings with hydrodynamic lubrication at the roller/flange contact, CL7C design

bearings are less sensitive to errors of adjustment, mounting or service than conventional standard bearings.

The development work on the roller/flange contact has thus achieved the specified goals.

Improved roller/raceway contact

If the generatrix of raceway and roller is absolutely straight (which is the basic idea behind taper roller bearings) and if the roller length does not coincide with the raceway width (which is normally the case) stress peaks, known as edge stresses, will arise at the ends of the contact surfaces and considerably shorten the bearing fatigue life. This effect will be even more pronounced if the roller and raceway are misaligned with respect to each other by a few minutes of arc, because of elastic deformations of the shaft and housing.

With conventional bearings these edge stresses are reduced by crowning the rollers and/or the raceways. Unfortunately, in many cases, the demands for crowning are contradictory. For example, for hub bearings, a rather heavy crowning would be appropriate for the high stresses produced during cornering, whereas for driving straight ahead, slight crowning only would be more appropriate. Consequently the bearing designer has, up to now, had to live with a compromise.

Usually, bearings have been designed for a given representative average load. Extensive mathematical analyses and computer calculations have shown that the most favourable effect in respect of edge stress reduction is obtained when the sum of the deviations from the straight line for raceway and roller corresponds to a certain logarithmic function. These findings have been put into practice in the TQ bearings, which are also made with refined manufacturing methods.

Fig 7 shows the advantages of the TQ taper roller bearings compared with conventional bearings. Whereas rather uneven load distributions with stress peaks are obtained with the standard bearings under both light and heavy loads (they were designed for a representative moderate load level), the load distribution is relatively even for the TQ bearings under all load conditions, as well as under misalignment, and there are no stress peaks.

In addition to optimising the roller/raceway contact geometry, the microgeometry (surface roughness) has been refined. By this means it has become possible for the new bearings to attain adequate lives, even under very unfavourable lubrication conditions where conventional bearings would fail prematurely because of surface distress. TQ bearings can even be used successfully where the lubrication conditions prescribed by the catalogue, particularly the requisite lubricant viscosity, cannot be attained, e.g. in hydraulic motors where the hydraulic oil is used to lubricate the bearings.

In practice, this means that:

- life and reliability are increased;
- the sensitivity to misalignment and inadequate lubrication is reduced;
- the noise level is reduced.

Improved roller/cage contact

The cage of a correctly mounted and adjusted taper roller bearing will be less heavily loaded than would the cages of other bearing types because of the good guidance of the rollers against the flange resulting from the force component, produced under all load conditions, which acts against the flange. However, the cage must still perform the following important

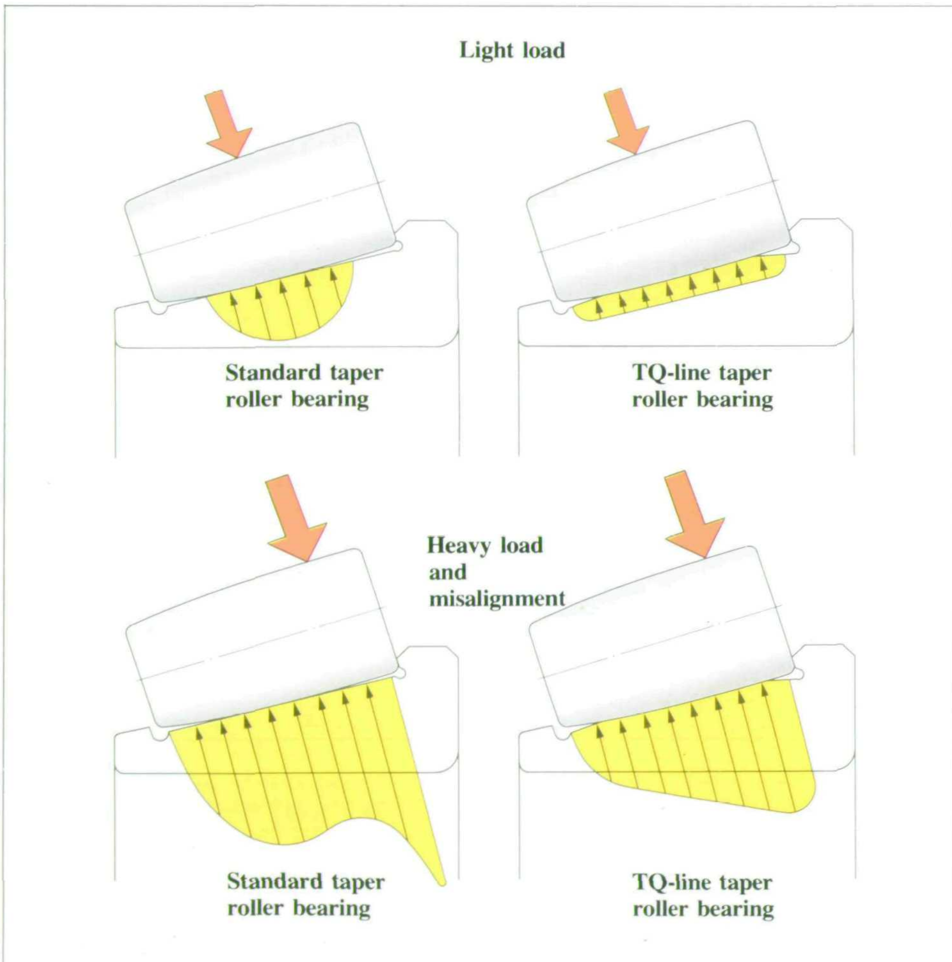
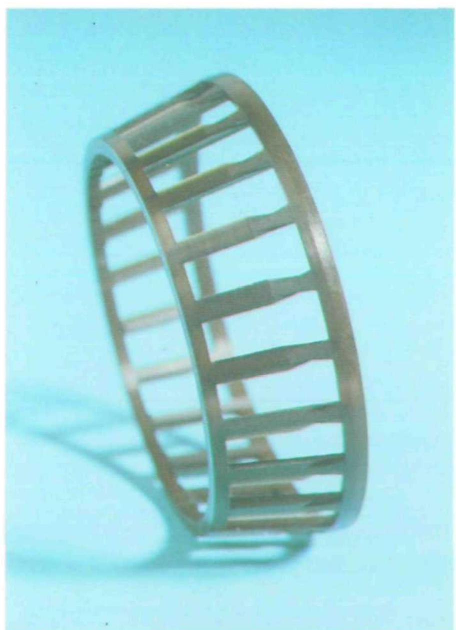


Fig 7 Stress distribution at roller/raceway contact

Fig 8 Injection moulded polyamide 6,6 cage



functions:

— prevent direct contact between adjacent rollers;

— maintain the spacing of the rollers, particularly in the unloaded zone;

— guide the rollers in the unloaded zone so that they enter the loaded zone correctly aligned;

— guarantee performance by preventing wear in the roller/cage contact.

To fulfil these requirements, the contact surfaces of the well-proven pressed steel cage were improved. The improvements mean that the forces between cage and roller, normally very small, can be transmitted without any problem. At the same time, the risk of foreign particles lodging in the contact area and scoring the rollers has been reduced.

In addition to the improved pressed steel cage, the well-proven alternative of an injection moulded polyamide 6,6 cage, see fig 8, is used in certain cases, e.g. for passenger car hub bearings. This cage further increases operational reliability. Lubrication conditions are improved because the lubricant is more easily drawn into the roller/cage contact areas and in extremely unfavourable conditions, where lubricant starvation occurs, the polyamide material will act as a 'solid lubricant' and prevent bearing seizure.

Optimum exploitation of bearing cross section

Bearing cross sections are generally prescribed by a standard, or in some cases by the user. Additionally, the diameter and angle of the raceway of the outer ring, or cup, are standardised for interchangeability reasons. Given these 'restrictions' and also the requisite wall thickness of the rings and the guide flange width (values which are based on experience) the given cross sections are well utilised in modern taper roller bearings.

However, nothing is perfect and improvements are always possible. The cross sections of the various bearing series were analysed and optimised for maximum load carrying capacity. In some cases, e.g. series 323, it was possible to increase the load carrying capacity of certain sizes quite considerably. The bearings therefore have either longer life or increased reliability.

Suitability for collective adjustment

A taper roller bearing must always be adjusted against a second bearing. Several methods are available to the user. In addition to the individual adjustment methods, collective adjustment (random statistical adjustment) has become popular. With collective adjustment, housings, shafts and bearings are randomly assembled and a given scatter in clearance or preload is accepted. However, the scatter of the various bearing, housing and shaft dimensions (lengths, widths and diameters) must be much smaller than is the case for standard executions. SKF can supply, on demand, bearings with appropriately reduced tolerances.

TQ-line taper roller bearings — the SKF answer to customer demands

Looking again at the market requirements for taper roller bearings now and in the future and comparing them with the new developments outlined above, it can be concluded that the new SKF taper roller bearings, the TQ-line bearings (fig 9), more than meet even the most difficult demands. The TQ bearings combine so many positive characteristics that it is no exaggeration to say that they constitute a new performance class.

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Fig 9 **TQ-line taper roller bearings**

The Quality Philosophy of SKF

Introduction

SKF has always enjoyed a reputation for the high quality of its products and services. This has been due to the technological leadership of the company and the superior quality of the materials used for its bearings coupled with an early recognition of many customers' requirements for technical service. Consequently, application engineering has always been an important activity within SKF bearing companies around the world.

However, the environment in which we operate is constantly changing; novel technologies are developed, new competitors emerge, markets and business cycles alter, consumer life styles and preferences change, national and international laws are introduced or modified etc. In order to stay healthy and to grow and prosper in this changing environment, it is necessary for SKF to continually develop and take advantage of the opportunities found in this constant process of change.

This article gives a brief review of the quality history of SKF and then focuses on the various means used during the last few decades to maintain and enhance the SKF quality image. The article is concluded with some thoughts about the future development of quality in SKF.

Review of the past

For more than half a century, the various companies established by SKF around the world were rather independent units, each operating mainly in its own country. Apart from financial and legal control exercised by the Swedish parent company, there was only limited central co-ordination, i.e. in the design and manufacturing technology areas. As a result, it was hardly possible during that period to speak about a common SKF product quality standard; each company had its own quality standard and satisfied the needs of its own market.

Except for the two world wars and the depression of the 1930s, this long period was characterised by a substantial growth rate in the world economy. It was largely a seller's market where the efforts were concentrated on producing enough bearings to satisfy the demand. Despite this, improvements were made in many respects, e.g. in terms of life of the bearings produced. Fig 1 demonstrates this clearly [1].

During the 1960s, the steel division of SKF began regular and systematic testing of bearing steel from its own mills and those of the main competitors. All the product parameters having a known influence on bearing performance, particularly

H. G. Herlitz
SKF Head Office
Sweden

SKF has always enjoyed a reputation for the high quality of its products and services. This article gives a brief review of the quality history of SKF and then focuses on the various means used during the last few decades to maintain and enhance the SKF quality image. The article is concluded with some thoughts about the future development of quality in SKF.

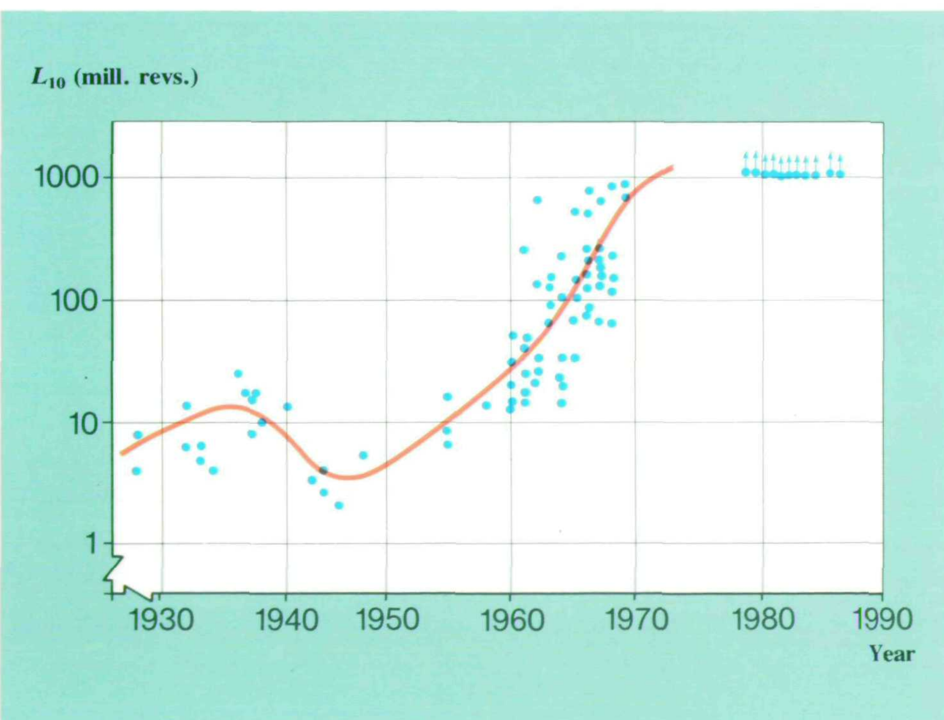


Fig 1 Results of endurance tests on SKF 6309 bearings

life, were tested and the results were used as a basis for further product development, primarily through process development.

The experience with this quality auditing system was very encouraging so that a similar product auditing system was introduced in the early 1970s for finished bearings. It was centrally co-ordinated with a key role played by the SKF Engineering & Research Centre in The Netherlands, but with all bearing companies of SKF involved. These product audits still constitute one of the corner stones of quality improvement and product development within SKF.

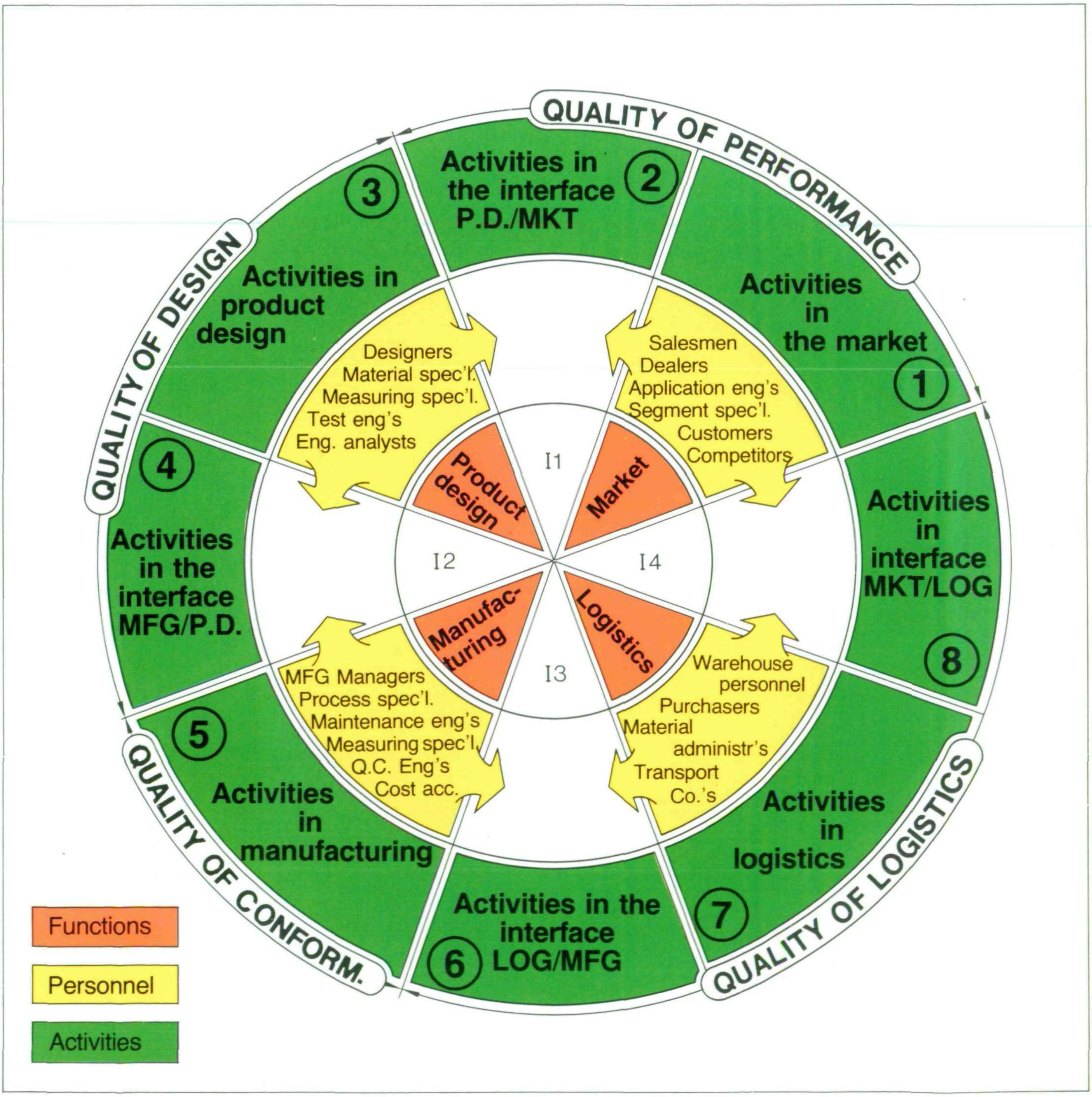
In the early 1970s, a major rationalisation of bearing manufacturing in SKF world-wide was started in the form of the so-called GFSS project (Global Forecasting and Supply System). The project, completed in the early 1980s, concentrated the production of each major bearing type to only one factory, which then supplied the market in many countries instead of only locally. In this way, longer production runs and hence lower manufacturing costs were reached. In addition, a higher product quality was obtained due to standardisation of methods and specifications. Extensive developments were made in the areas of manufacturing technology and quality control techniques, both having a considerable effect in terms of improved product quality.

During the course of the GFSS project, a Group-wide systematic approach to quality improvement was made. The organisation, systems, and techniques then introduced have been further refined and permeated to all SKF companies. It is no exaggeration to say that SKF today is in the forefront of modern quality techniques.

Quality Management

The top management of SKF

Fig 2 The SKF Quality Map



established a Group quality policy in the late 1970s. This policy is based on the Juran concept "Quality is fitness for intended use". The main elements are:

— To market only those products which will merit customer satisfaction by performing their functions reliably and effectively in accordance with customer needs and demands

— To maintain a leading position as regards quality of performance

— To deliver only those products which are in conformance with, or better than, the approved design (drawings and specifications).

Fig 2 shows the SKF Quality Map which illustrates the four main fields of activity where attention is focused to achieve quality.

— *Quality of design* measures how well the design satisfies the needs and demands of the customer

— *Quality of conformance* measures how well the manufactured product conforms to the design

— *Quality of logistics* measures how well the product is transferred from manufacturing to the customer

— *Quality of performance* measures how well the product satisfies the needs and demands of the customer.

The quality map also shows the various interactions required at the interfaces between different internal functions and the market or the customer. The development of this diagram has helped in the generation of new quality concepts and techniques, as will be shown later in the article.

Focus on quality is one of the top priorities of SKF. Group management monitors quality improvement in each company by means of three simple diagrams, showing the development over a period of time:

— Quality of conformance according to product quality audits

— System quality level according to quality system audits

— Quality maturity and employee involvement in quality work.

The close monitoring of quality of conformance during many years has resulted in a considerable improvement of conformance to manufacturing specifications and consequently a reduction of scrap and rework.

Regular quality system audits are performed in almost all manufacturing companies of SKF. The procedure used was developed entirely within SKF. It implies a detailed scrutiny and a very tough evaluation of the total quality system and has some unique features.

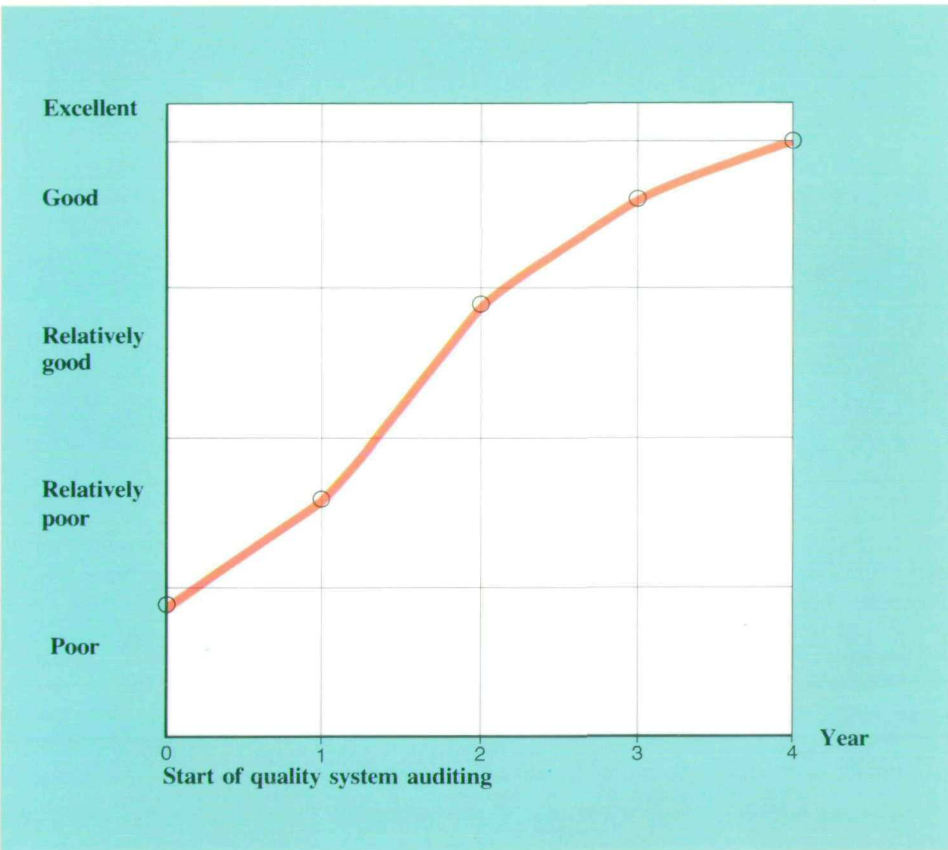


Fig 3 Typical evolution of quality system audit results

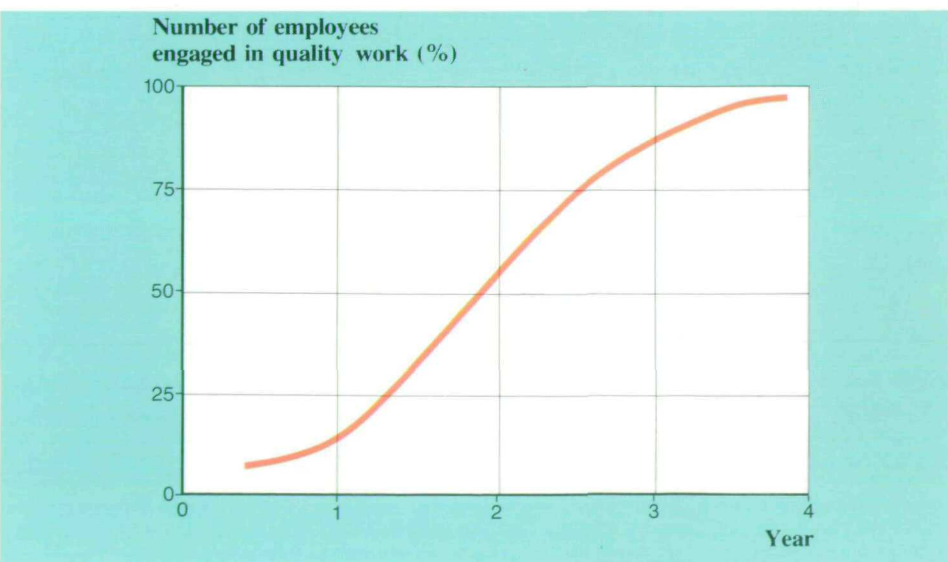


Fig 4 Quality Growth Diagram

With the aid of these audits, the performance of the quality management and assurance system is measured and monitored. Fig 3 shows a diagram of quality system audit results for an SKF company where improvement activities were needed and carried out.

The most important feature of these system audits is that they identify weak areas where corrective action is necessary. The management of the unit audited is then expected to take the necessary remedial action. In this way, the quality of performance of the product will gradually increase through better design, higher quality of conformance, and improved quality of logistics. In addition, failure costs will decrease, e.g. through less scrap and rework caused by non-conformance and fewer customer complaints.

The degree to which the total work force of a company is involved in quality work is illustrated in a so-called Quality Growth Diagram; see fig 4. A quantitative method for evaluation of a company's position in the diagram has recently been developed. The goal for all SKF companies is to reach the 100 per cent level in this diagram.

Quality Assurance

Present quality systems in SKF companies aim at safeguarding the required product quality level as early as at the design and manufacturing planning stages. By close contact with customers and through market research, the needs of the customers are defined and used as input to the design phase. Manufacturing experts are involved at an early stage in order to avoid designs that may cause non-conformance in manufacturing.

The quality assurance system is based on various international standards such as MIL-Q-9858A, AQAP-1, and ISO 9000-9004. Each of SKF's five business areas (Bearing Industries, Bearing Services, Speciality Bearings, Tools and Component Systems) has a quality manual containing the requirements to be met in order to achieve the quality policy level. Every company then has a set of procedures describing the various activities needed to fulfil the requirements of the manual and also detailed instructions for individual operations which may influence quality. The quality assurance department co-ordinates all quality work and is normally directly accountable to the Chief Executive. Each company has a Quality Board chaired by the Chief Executive and with representatives from most staff and line functions, including Quality Assurance. The Quality Board meets quarterly and is the top forum for important quality matters such as policy implementation, goal setting and resource allocation. A Quality Board also exists at Group level. It is chaired by the Group Chief Executive and deals with major quality matters of Group interest. Each meeting with this Board is devoted to one single business area.

Based on strategic business area quality goals, all manufacturing companies in SKF establish quality improvement plans for each year. Such a plan contains important quality goals, e.g. regarding quality of conformance, quality system audit results, training, quality-related investments, improvement activities and failure-cost reduction. It also contains a list of quality problems to be dealt with and the corresponding projects, including objectives, budgets and time plans. The appropriate Quality Board approves these annual plans and monitors the progress during the year. The overall guideline is always the same: *Never-ending improvement.*

Reduction of failure costs was mentioned above as one of the elements in annual quality improvement plans. A well devised quality-cost accounting and reporting system is necessary for goal setting and follow-up of failure costs. An improved system has recently been introduced in most manufacturing companies of SKF. Total quality costs are divided into two parts, i.e. the *Quality System Platform*, consisting mainly of what is generally called prevention and provision costs, and *failure costs,* consisting of internal costs (mainly scrap, losses, and rework), external costs (mainly customer complaint costs), and inspection costs related to non-capable and non-reliable processes. All SKF companies strive towards reduction of failure costs by maintaining a proper and balanced quality system platform and by persistent corrective actions.

To stimulate further quality improvement work, quality competitions are often organised, both Group-wide and at local level. Public recognition of prize winners provides motivation for all employees in the winning company or department.

A very important element in SKF quality improvement work is the introduction of modern quality techniques such as design review, capability studies, statistical process control, failure mode and effects analysis, statistical design of experiments, Taguchi methods, and quality function deployment, also called matrix product planning. Written procedures and instructions are developed and distributed, and training modules are produced or acquired and extensively used. Implementation goals are included in annual quality improvement plans. The result has been a dramatic improvement of product quality in SKF companies during the last few years. In spite of this, further improvements are possible and will be made.

Many SKF companies are experimenting with various forms of the quality circle concept, i.e. a bottom-up approach to quality improvement. In general, the purpose is to stimulate the individual employee to more involvement in quality work through the participative process offered by quality circles ("quality teams" and "quality improvement groups" are other terms used). Some companies have reported very good results but overall the experiences are not yet conclusive. Probably, the concept has to be adjusted to the specific cultural circumstances in each country and company to lead to success. It is also necessary that the company management actively supports the quality circle programme. Quality circles and similar activities constitute one way of moving a company upwards in the Quality Growth Diagram; consequently such activities are supported by SKF Group management.

A look into the future

Until recently, quality work in SKF has been largely focused on product quality and hence on manufacturing. This is only natural in a company whose main business is to make and sell a hardware-type product like rolling bearings. Most companies of this type, however, also provide some form of service in conjunction with the product sale. This is certainly the case with SKF. Examples: application design assistance, recommendations regarding bearing selection, advice regarding mounting of bearings, support in installing complex bearing systems, training in bearing maintenance, and emergency support in connection with breakdown of machines.

Because product quality has improved so much and because customer satisfaction depends to a considerable extent on good service as well, quality efforts in SKF are increasingly being devoted to the service area. Service quality is not a new concept but it has definitely drawn less attention than product quality, within SKF as well as outside.

Many activities are under way at present within SKF companies aiming at an improved quality of service to the customer. For obvious reasons, this work is concentrated on functions other than manufacturing, e.g. marketing and sales. Due to the fact that this area is not as well researched as the product quality field, much of the work is geared towards development and testing of new concepts and techniques tailored for the service sphere. Progress is more rapid than it was in the product quality field some years ago, because experience from this previous work can be utilised.

It is the intention to introduce quality assurance procedures in the service area during the next few years. Annual quality improvement plans are being established and will be followed up, and quality system auditing will be used to initiate improvement work.

The service quality efforts will not be limited to service towards customers but will also include service within the company. Every employee has a supplier and a customer—internal or external. All work in the company is done, ultimately, for a customer; consequently internal service quality will also have an impact on the customer-perceived quality. In addition, there are considerable potential cost savings associated with improved internal service quality.

The internal service quality work will cover all parts of the company. When this has been fully implemented, the position in the Quality Growth Diagram will be much improved. A good start has already been made in some functional areas, e.g. in data processing and in research and development.

With a reasonable extrapolation and a bit of imagination, a scenario of SKF 5–10 years from now might look like this:

> SKF is a company where the voice of the customer is the guideline. All employees are involved in quality-related work to achieve continual improvement and gain customer satisfaction. Prevention is the dominant part of quality assurance. Failure costs are negligible. Quality management is an essential part of the total management system. SKF has a clear edge over the competition because of the first rate quality of its products and services.

Is this a realistic prediction or wishful thinking? The momentum of present quality efforts in SKF and the dedication of SKF employees the world over to continuous improvement indicate that the scenario will come true.

Reg. 589

Reference

1. Åkesson J. and Lund T.: **SKF MR produced rolling bearing steels,** Ball Bearing Journal 231(1), pp 12-18.

SKF publications on bearing technology – more than just catalogues

Bearing users' demands differ and therefore the type of information required varies. To meet these various needs, SKF publishes product-oriented and application-oriented information, in addition to theoretical articles on bearing technology and related subjects. There are also Group-wide publications and publications produced for a specific "local" market. At first sight SKF product literature concerning bearing products appears to be an unstructured collection of various types of documents, but this is not the case, as the following will demonstrate.

Product-oriented information

Most product-oriented information published by SKF is in the form of catalogues, although special brochures and articles are produced to coincide with the introduction of new or re-designed products, or for exhibitions.

SKF product catalogues do not simply give lists of available bearings but provide all the relevant product data needed to enable the bearings to be correctly used. In addition to the tabulated information, namely dimensions, tolerances, loads, speed ratings, masses (weights) etc. recommendations are also given on how to select, calculate, mount, lubricate and maintain the products.

The basic document intended for the ball and roller bearing market is the SKF General Catalogue, which is a Group-wide publication. This means that the content is identical, irrespective of language edition. The SKF General Catalogue 3200 edition was published in 15 languages and more than 700 000 copies distributed. The new General Catalogue 4000, which has just been published, is an even more comprehensive document. The general technical section now covers some 170 pages and provides a good grounding in bearing technology.

The range of products shown covers "standard" bearings (i.e. conforming to International (ISO) or national standards) for all the major rolling bearing types and should meet the needs of most general engineering applications.

The General Catalogue is supplemented by several Product Catalogues dealing with certain product categories (e.g. thin section bearings) or directed at a certain market area (e.g. bearings for heavy engineering applications). Product Information leaflets (PIs) also supplement the General Catalogue and are produced when, for example, a certain series of bearings is redesigned, or when new bearings are introduced. The Product Catalogues and Product Information leaflets contain all the information relevant to the products being described, but for more general bearing technology, reference is made to the General Catalogue. They are also Group-wide publications.

Application-oriented information

The basic information dealing with bearing arrangements, the most suitable bearings to use for various applications, advice regarding fits, sealing, lubrication systems etc., is contained in the technical section of the General Catalogue and appropriate Product Catalogues and Product Information leaflets.

However, much of the more specialist application-oriented information produced by SKF takes the form of articles published in the Ball Bearing Journal. In addition to articles describing specific bearing applications, others of a more general nature relating to application engineering are included. Brochures are also produced, e.g. in connection with various exhibitions or special marketing drives. Material having more the character of a handbook is also published for certain application areas.

Theory, bearing technology

As explained elsewhere, SKF has spearheaded the understanding of rolling bearing technology and the Ball Bearing Journal, which has been published regularly since the early 1920s, has often been the vehicle used to publish the original work of SKF researchers. Because it is a Group-wide publication produced in six languages with a circulation at present of more than 65 000 copies, authors can be sure of being widely read.

In addition to the BBJ articles, SKF also publishes reprints of articles by SKF engineers and researchers which have appeared in other journals.

In conclusion, therefore, SKF publications represent a well-ordered collection of information which is not just in the form of catalogues. This is only one facet of the service provided by SKF to bearing users.

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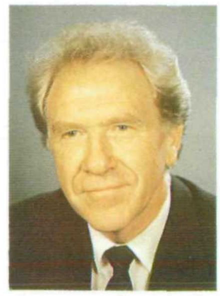
Bearing steel development

A. Kroon
SKF-ERC
The Netherlands



H. Nützel
SKF-ERC
The Netherlands

Introduction

The increasing demands on the life and performance of SKF rolling bearings require an improved quality level of the bearing steel and its treatment. The material for the rings and the rolling elements experiences extremely high stresses during cyclic over-rolling in service. To withstand these stresses the bearing steel must be very clean and practically free of material defects. The hardness and strength of the heat treated steel must be sufficiently high to guarantee the static load carrying capacity and resistance against rolling contact fatigue; but, it must also have a certain toughness in order to avoid fracture during mounting and service.

The ability of the ring and rolling element material to withstand rolling contact fatigue is primarily dependent on the cleanliness of the bearing steel. Macro inclusions, defined here as non-metallic inclusions with an individual length of at least 0,5 mm, have a very strong influence on the frequency of early failures and thus on the reliability of the bearings. By continuous improvements of the steelmaking processes around the world, but especially at SKF Steel (today OVAKO Steel), and by making special provisions and taking precautions during the steel manufacturing and processing, the content of macro inclusions has been drastically reduced in SKF bearing steels during the last 20 years (see fig 1). This development also had its influence on the other bearing steel producers. Today the maximum content of macro slag inclusions in bearing steels, used for SKF bearing manufacture, is specified at 10 mm/m^2, as determined by the step-down test. The generally applied testing methods for determining macro inclusions, step-down test and blue-fracture test, nowadays give constantly zero values. Therefore, more stringent test methods have to be developed to detect the low content of macro inclusions in the bearing steels.

The micro inclusions (mainly sulphides, titanium nitrides and oxides) are fairly homogeneously distributed in the steel and, as a rule, can be correlated with the sulphur, titanium and oxygen content. Oxides are the most dangerous inclusions with regard to rolling contact fatigue of bearings and therefore the reduction of oxygen in the steel will result in a significant improvement in bearing life. The diagram in fig 2, derived from extensive bearing tests, confirms this statement. The necessity of oxygen reduction in bearing steels by steel-refining processes was realised early at SKF and therefore efforts have been made towards continuous metallurgical process developments. As a result of these developments the oxygen content in today's bearing steel is now less than 10 ppm, compared to about 30 ppm in 1960 (see fig 3). The most significant success in steel-refining since the early 1970s has been achieved by the

The increasing demands on the life and performance of SKF rolling bearings require an increasing quality level of the bearing steel and its treatment. The extreme high stresses which the material for the rings and the rolling elements experiences during cyclic over-rolling in service can only be matched by a bearing steel which is very clean and practically free of material defects. The hardness and strength of the heat treated steel must be sufficiently high to guarantee the static load carrying capacity and resistance against rolling contact fatigue, but, it must also, however, have a certain toughness in order to avoid fracture during mounting and service.

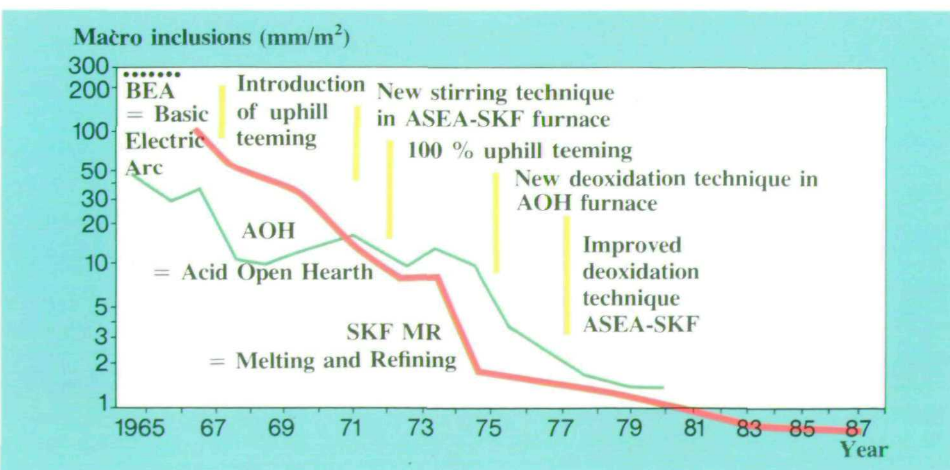


Fig 1 Reduction of macro inclusions in bearing steels between 1965 and 1988 at SKF Steel (now OVAKO Steel)

introduction of the new steelmaking technique, MR (Melting and Refining) process, pioneered at SKF Steel. In the MR process the steel scrap charge is in a first step melted in the SKF twin shell furnace and then in a second step refined in the ASEA-SKF ladle furnace. As a consequence of these developments the bearing steels currently produced are, with regard to oxygen and micro inclusion content, of a quality that 15 years ago could only be produced by special melts using processes such as ESR (Electro-Slag Remelting) or vacuum melting. The MR Bearing Quality steel (MR BQ) which is presently produced at OVAKO Steel is delivered to a micro inclusion specification, which has more stringent acceptance limits than those of the ASTM A295 specification (see Table 1).

The steel specifications for the manufacture of SKF bearings also have significantly lower limits for micro inclusions, especially oxide inclusions, than those given in ASTM A295. In cases where the DIN 50602 (SEP 1570) method is applied for rating of micro inclusions, the SKF material specifications allow only half of the K values as specified for bearing steels in DIN 17230. To reduce the content of the hard and sharp edged titanium carbon nitrides in steel, which negatively influence the rolling contact fatigue of bearings, the titanium content in bearing steels for SKF is specified to max 30 ppm. Similar

Table 1 MR BQ specification, comparing limits to those in current ASTM A295 specification

Specification	Macro-inclusions	Oxygen content	Titanium content	Micro-inclusions							
	mm/m^2	ppm	ppm	A		B		C		D	
				Th	He	Th	He	Th	He	Th	He
MR BQ	5	13	30	2,0	1,5	1,5	0,2	0	0	0,5	0
ASTM A295	—	—	—	2,5	1,5	2,0	1,0	0,5	0,5	1,0	1,0

Th = Thin He = Heavy

Fig 2 Relative life versus oxygen content

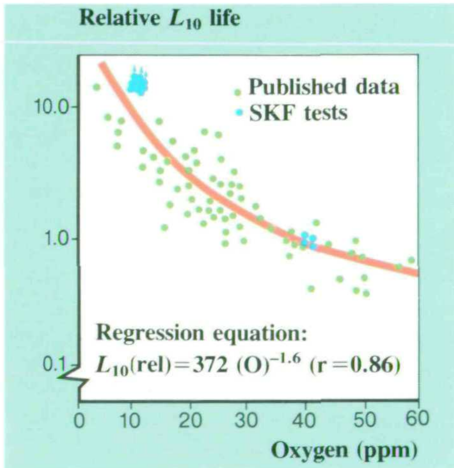
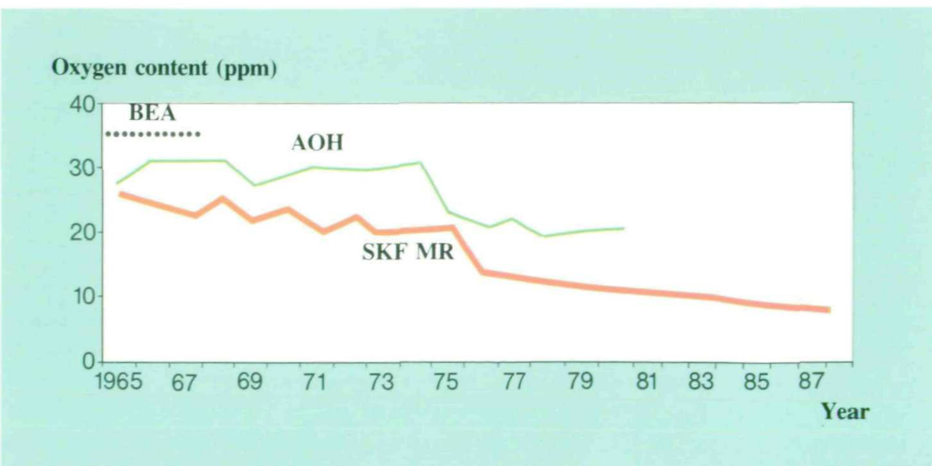
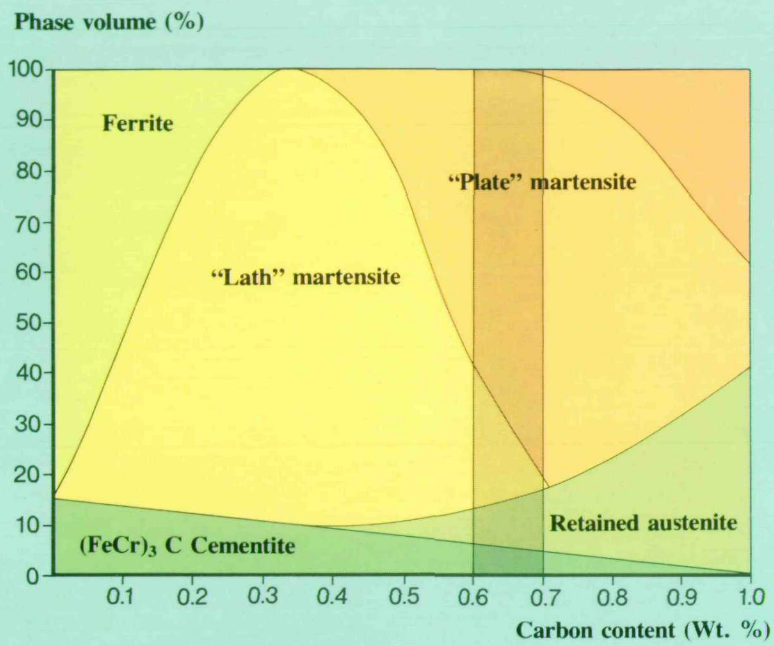


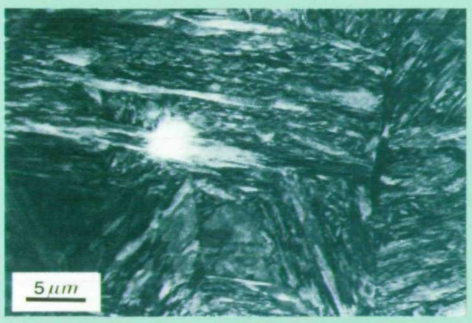
Fig 3 Reduction of oxygen content in bearing steels between 1965 and 1988 at SKF Steel (now OVAKO Steel); for abbreviations see fig 1



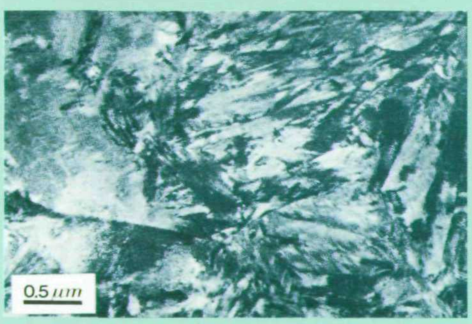
restrictions have been specified for the content of the trace elements arsenic, antimony, tin and lead. This stringent requirement on the cleanliness in bearing steel can only be met currently by a selected number of steelworks. To guarantee this high level at the steelworks, a system of continuous quality audits has been introduced.

Because of this high cleanliness in today's bearing steels, SKF bearings now achieve under ideal conditions indefinitely long lives, even if they run at stress levels which can be considered to be very high from an application point of view. It can, therefore, be stated that the steel used for the manufacture of SKF bearings has reached a cleanliness level where further improvements will be difficult to verify with conventional bearing tests.





1 Typical lath martensite (low carbon)



2 "Normal" martensite Typical martensite hardened and tempered ball bearing steel

3 Typical mid-ribbed martensite (high carbon, high temperature)

Fig 4 Relative proportion of microstructural phases related to carbon content of matrix during martensitic hardening of SKF 3 steel

Microstructure and mechanical properties

Rapid quenching of an austenitised SKF 3 type steel to room temperature results in the formation of martensite, a very complex metallurgical structure in which the carbon, formerly in solid solution in the austenite, remains in solution in the quenched phase. Unlike other phase transformations, which are normally accomplished by atomic diffusion, martensite forms by a sudden shear process in the austenite lattice. The 'highly' saturated carbon in solution in the ferrite leads to high internal microstresses and a distorted crystal structure, which is reflected in a high hardness and brittleness. This is the structure which we call "martensite".

The different martensite types, related to morphologies, depend mainly on the amount of carbon in solid solution in the austenite. The morphologies are very important because they have an effect on the strength and toughness of the hardened steel. In fig 4 the approximate proportion of these phases is shown for SKF 3 (SAE 52100) martensitic quenched with different carbon contents in solution.

With increasing carbon content in solution in SKF 3 (SAE 52100), by higher hardening temperatures, the shear strength of the austenite increases, which results in a lower *martensite start temperature*. Equally, the martensite finish temperature will be lowered, which leads to an increased amount of retained austenite. Distortion of the martensite will also increase, resulting in greater strength but decreased toughness.

Toughness in hardened SKF 3 (SAE 52100) rolling bearing steel is of vital importance in order to prevent premature fracture during mounting, or at an early stage of service. This requirement sets an upper limit of about 0,7 Wt.% carbon in solution in order to avoid the mid-ribbed plate martensitic structure which exhibits a typical inter-granular brittle-type fracture. In fig 5 the relationship between carbon in solution and plane strain fracture toughness is shown.

The bainitic microstructures produced in SKF 3 (SAE 52100) steel by quenching from the austenitisation temperature (840 °-860 °C) to the isothermal transformation temperature (210 °-250 °C) are examples of stable, low level or completely austenite-free structures, where both improvement in toughness and strength can be realised. Bainitic heat treated SKF 3 rolling bearing steel shows dynamic fracture toughness approximately 30 % higher than normal martensitic hardened SKF 3. The bainitic structure also shows a considerably higher threshold stress intensity needed for crack growth, see fig 6. The clear structural toughness superiority of the bainite compared to martensite of similar hardness is probably a function of both the advantageous compressive residual stresses which bainitic heat treatments produce in SKF 3 rolling bearing steels and the beneficial effect of the fine carbide dispersion in the lower bainitic microstructures. These properties of the bainitic heat treated SKF 3 rolling bearing steel are, of course, highly desirable for structural toughness reasons.

Fine grain size has a positive effect on the toughness properties of SKF 3 rolling bearing steels in the soft annealed or tough tempered conditions. However, the benefit of fine prior austenite grain size in hardened SKF 3 rolling bearing steel is not proven. The increased toughness observed on hardened specimens taken from cold rolled material compared to hot rolled, fig 7, could be simply associated with the finer prior austenite grain sizes. However, results from 'pure' thermal treatments to produce similar prior austenite grain sizes as those obtained from prior mechanical treatments (cold rolling) did not demonstrate the same increase in toughness, see fig 8. For this test all specimens were given the same final martensitic heat treatment: austenitised at 845 °C, quenched in oil at 50 °C and subsequently tempered at 160 °C for 90 minutes.

From these observations within the range considered, it would seem reasonable to conclude that prior austenite grain size alone cannot be used as the microstructural unit related to the toughness properties of hardened bearing steel.

Dimensional stability of bearing steels

Martensitic hardened and normal tempered bearing components made from through hardening bearing steels are dimensionally stable up to 100 °C for continuous running temperatures. If the temperature exceeds 100 °C dimensional changes will occur, which are temperature-time related. The dimensional changes are caused by transformation of the different phases present in the microstructure of the hardened and tempered bearing steels.

Directly after hardening (after quenching and before tempering) the microstructure of bearing steel comprises three phases: dimensionally stable undissolved carbides, martensite and retained austenite. With increasing temperatures, the following phase transformations take place:

1. Migration of the carbon from the martensitic lattice and formation of ϵ-carbide (room temp. to about 150 °C).
2. Transformation of retained austenite (100-250 °C).
3. Formation of Fe_3C cementite from ϵ-carbide (above 250 °C).

Formation of ϵ-carbide and cementite is associated with a dimensional shrinkage, and the transformation of retained austenite to bainite with dimensional growth. Therefore, a quenched and untempered bearing component at elevated temperatures will undergo three different dimensional change intervals: in the first interval a dimensional shrinkage, in the second interval a dimensional growth and in the third interval again a dimensional shrinkage. The time intervals depend strongly on the applied temperatures

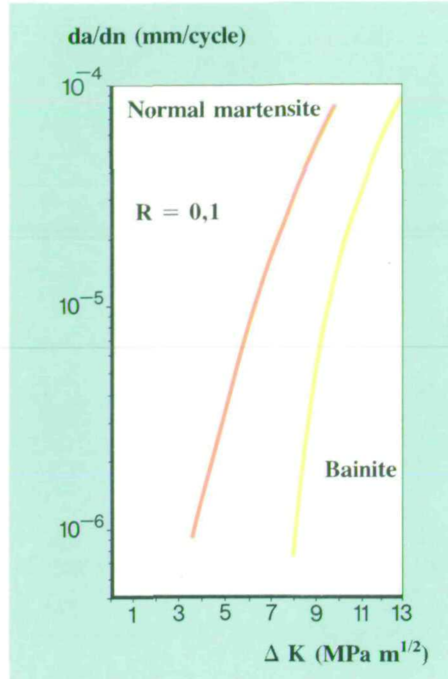


Fig 6 Average crack growth curves for normal martensite and bainite

Fig 5 Relationship of carbon content to fracture toughness for martensitic through hardened SKF 3 steel (860 °C for 20 mins, 160 °C for 90 mins)

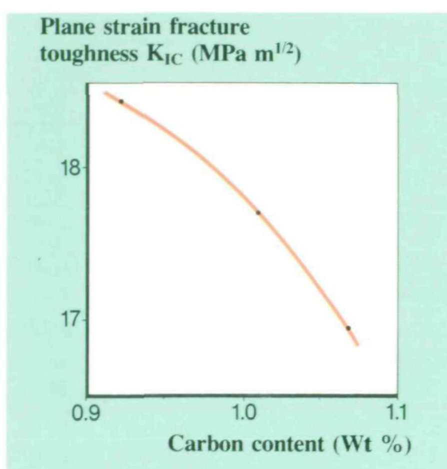
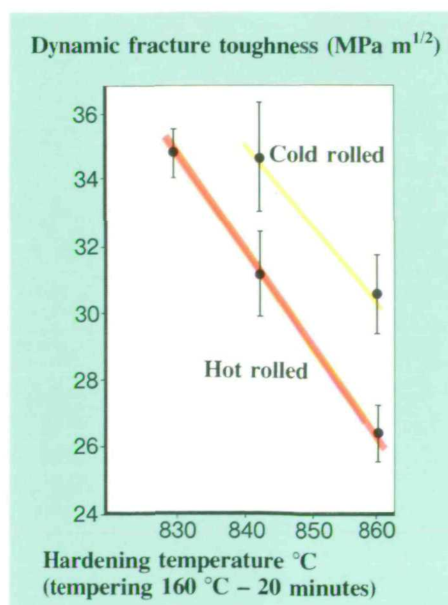


Fig 7 Influence of hardening temperature and prior working on toughness of martensitic hardened SKF 3 steel



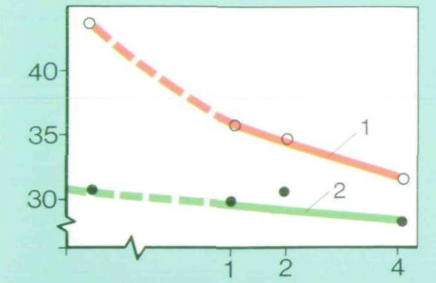


Fig 8 Comparison of mechanical and isothermal grain refinement techniques related to blunt notch toughness

and can overlap widely. The absolute values of the dimensional changes depend mainly on the heat treatment and are in the order of some 10 to about 120 μm/100 mm. An obligatory tempering of all bearing components at a minimum temperature of 150 °C, after quenching and before grinding, results in the precipitation of the carbides and thereby removes the first dimensional shrinkage stage from the final product. The tempering also relieves the high stresses from the quenching and reduces the brittleness of the steel in the as-quenched condition.

Dimensional changes of hardened and normally tempered bearings at service temperatures above 100 °C are therefore caused by the retained austenite transformation and by the formation of Fe_3C. The retained austenite content and the amount of formed Fe_3C increase with increasing austenitising temperatures and times. Consequently, the dimensional changes are greater for longer austenitising times and higher austenitising temperatures. The kinetics of the retained austenite transformations, as well as of the Fe_3C formation, have been studied and the equations to describe these kinetics are known. As the phase transformations are correlated to the dimensional changes, it is possible to calculate the dimensional changes for any temperature-time combination. Fig 9 shows how various temperature-time combinations and the expected dimensional changes are correlated. The dimensional changes for specimens heat treated at different austenitising temperatures are plotted against the Larson-Miller parameter $E = T\,(13,97 + \log t)/1000$, where T is in degrees Kelvin and t is the time in hours. In the lower part of the diagram temperature and time values can be read off for a given value of E, e.g. a value of E = 8 can be obtained at a temperature 300 °C (573 K) and the time 1 h, or 250 °C and 20 h or 200 °C (473 K) and 1000 h. Obviously, every other combination of the temperature-time field of interest can be selected from the diagram and related to a dimensional change, if the originally applied heat treatment is known.

For bearing components running at elevated temperatures, the dimensional changes which occur during service can also be compensated for by tempering treatments at high temperatures after quenching and before final grinding. The various stability classes, together with the allowable maximum size

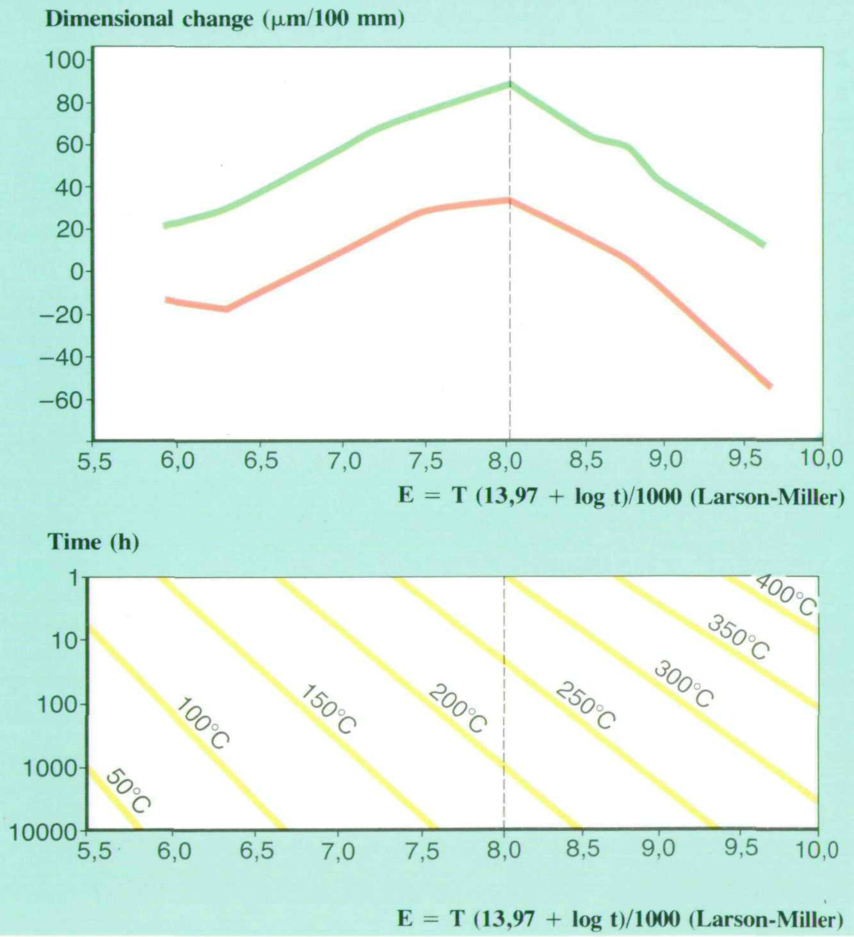


Fig 9 Diagram for determining dimensional changes and various temperature-time combinations

Table 2

Stability Class	Maximum dimensional change: Growth (μm/100 mm)	Shrinkage (μm/100 mm)	Service condition: Operating temp. °C	Time (h)
SN	+ 10	0	100	2500
S0	+ 15	0	150	2500
S1	+ 5	− 15	200	2500
S2	+ 5	− 15	250	2500

changes, are shown in Table 2. The treatments for achieving the various stability classes are, however, associated with a decrease in the original hardness of the quenched and normally tempered component. The greater the stability at higher temperatures, the lower the hardness of the component. For the S0 stability class the hardness of the component can be as low as 58 HRC.

Post quenching treatment for improved dimensional stability

To attain a given stability with a higher hardness, a sub-zero treatment between quenching and tempering is sometimes applied in order to reduce the retained austenite content. However, with such a sub-zero treatment the mechanical properties of the bearing steel, especially the toughness and fatigue values, are significantly reduced.

A variation in heat treatment has recently been applied in production which avoids these disadvantages. After austenitising, the components are quenched and immediately rinsed in water at 15-20 °C and then normally tempered. The investigations have shown that the introduction of the additional water rinsing has an effect on the retained austenite content, the transformation kinetics and the correlation between transformed austenite content and linear dimensional changes.

Fig 10 shows the dimensional changes after ageing at 220 °C for up to 10 hours for conventionally hardened samples, as well as for conventionally tempered plus water-rinsed samples. The dimensional change after 10 h of the additional water-rinsed samples is only 40 % of that of the non water-rinsed samples. This reduction in growth at elevated temperatures is due to two mechanisms. The retained austenite content of the conventional samples is reduced by the water rinsing, but this alone cannot explain the drastic reduction in dimensional growth from 100 μm/100 mm to 40 μm/100 mm. This reduction is achieved by a closer overlapping of the intervals for formation of carbide, and transformation of retained austenite, than is present after conventional quenching and tempering. Thus the growth due to retained austenite transformation is partly balanced by the shrinkage due to carbide formation. This process enables bearing components to be manufactured with improved dimensional stability without a reduction in hardness.

Response of material to cyclic rolling contact

The high stresses which are induced by cyclic over-rolling under high load in bearing raceways cause microstructural changes in the sub-surface regions of the bearing rings and rolling elements.

In a localised region under the raceway, corresponding approximately in depth to the highly-stressed sub-surface region, differences in etching response of the microstructure are observed, as shown in fig 11.

This phenomenon has been investigated in great detail at the SKF Engineering & Research Centre. Retained austenite measurements are a sensitive monitor of the onset of microstructural alterations. Long before any visible change occurs in the response of the sub-surface regions to etching, retained austenite, present as a consequence of normal martensitic heat treatment, starts to decay. An example of the changes as a function of depth below the raceway surface for an increasing number of revolutions in 6309 deep groove ball bearing inner rings, tested at a maximum contact stress of 3300 MPa, is shown in fig 12. This supports the observations of a narrow localised band of transformation in the sub-surface regions.

The results of systematic analysis of retained austenite decomposition at different contact stresses is shown in fig 13. The decomposition may be divided into three stages:

Stage 1: A rapid decrease in retained austenite during the early revolutions of testing (shake-down).

Stage 2: A period of retained austenite stability (steady-state).

Fig 10 Linear dimensional changes during ageing at 220 °C
Material: SKF 3
Hardening: 865 °C for 48 minutes — oil 50 °C for 10 minutes
Tempering: 160 °C for 120 minutes

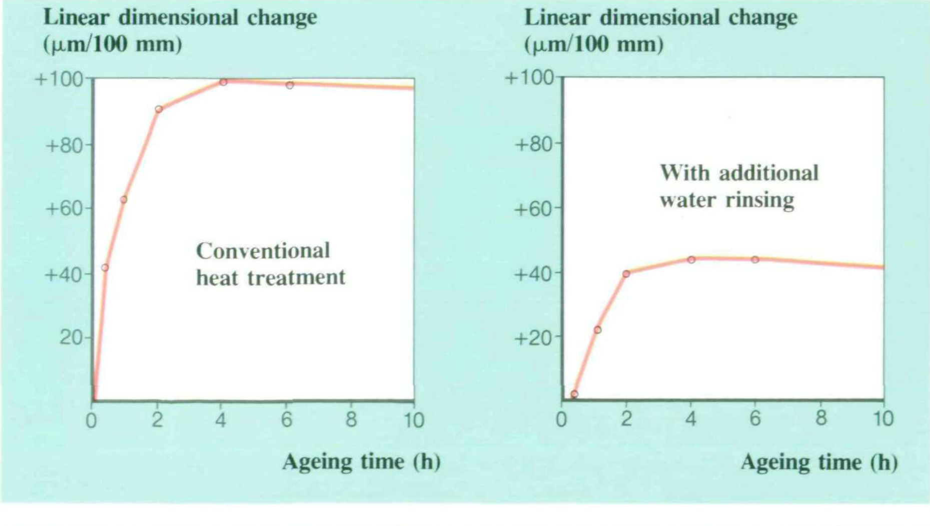


Fig 11 Optical micrograph of axial section through centre of ball raceway of a tested unfailed 6309 inner ring, showing zones of structural changes. Test conditions: maximum contact stress: 3800 MPa, inner ring rotational speed: 6000 r/min, operating temperature: 53 °C, full film oil lubrication. Test time was equivalent to 660 million revolutions

Stage 3: A further decomposition of retained austenite (instability).

The changes in retained austenite level as a result of the contact stresses are not the only measure of the changes in the sub-surface regions. Compressive residual stresses develop in a plane parallel to the bearing raceway in directions parallel to and perpendicular with the rolling direction; these are shown in fig 14. In a direction normal to the surface, tensile residual stresses exist in localised sub-surface regions of tested bearings, fig 15.

In addition to this, the existence of preferred orientation (texture) has been shown as a significant feature in Stage 3 of microstructural alterations. The preferred orientation develops gradually from accumulated micro deformation. This deformation activates particular glide systems in the crystalline material, by which the combination of glide and tilt of the atomic planes causes the texture.

Depending upon the bearing running conditions a so-called {100} <110> rolling texture can develop, which aligns the common (100) fracture plane parallel with the rolling contact surface.

The aforementioned tensile residual stresses in combination with the existence of a preferred orientation in a localised sub-surface region of tested bearings are a powerful driving force for crack growth in spalling fatigue failures.

In clean steels sub-surface cracks have been observed under extreme pressures extending over large distances without causing a spalling fatigue failure. At lower stresses of testing, in steels with a high content of oxide inclusions, fatigue spalls in early failures tend to be small and irregular. In later failures, particularly in cleaner steels, microstructural alterations allow spalls to propagate more extensively, and they are relatively 'flat'. Over 95 % of all failures in our experiments on clean steels show advanced texture development. Moreover, they tend to have a pronounced "V"-shape, with the sides of the "V" close to 45 ° to the rolling direction, consistent with cracks developing perpendicular to the rolling plane along the additional fracture plane orientations.

To account for these observations, it is suggested that crack initiation may start from several locations in the vicinity of stress-concentrating oxide or carbonitride inclusions, to give an irregular spall shape. In clean steels

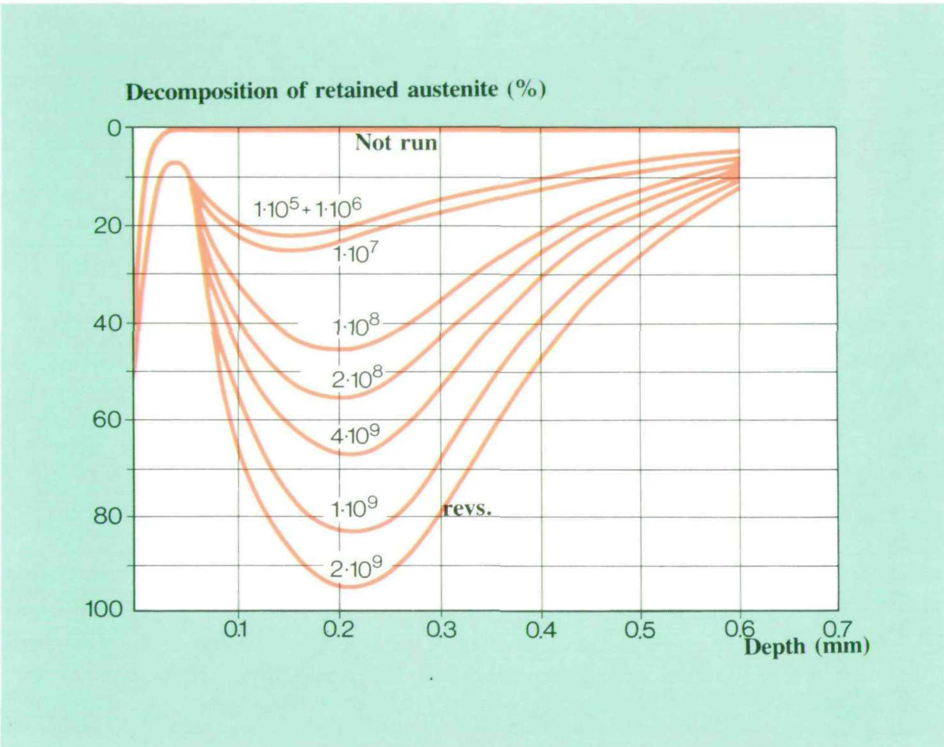


Fig 12 Decomposition of retained austenite versus depth beneath 6309 inner ring raceway tested for various numbers of revolutions at maximum contact stress of 3300 MPa

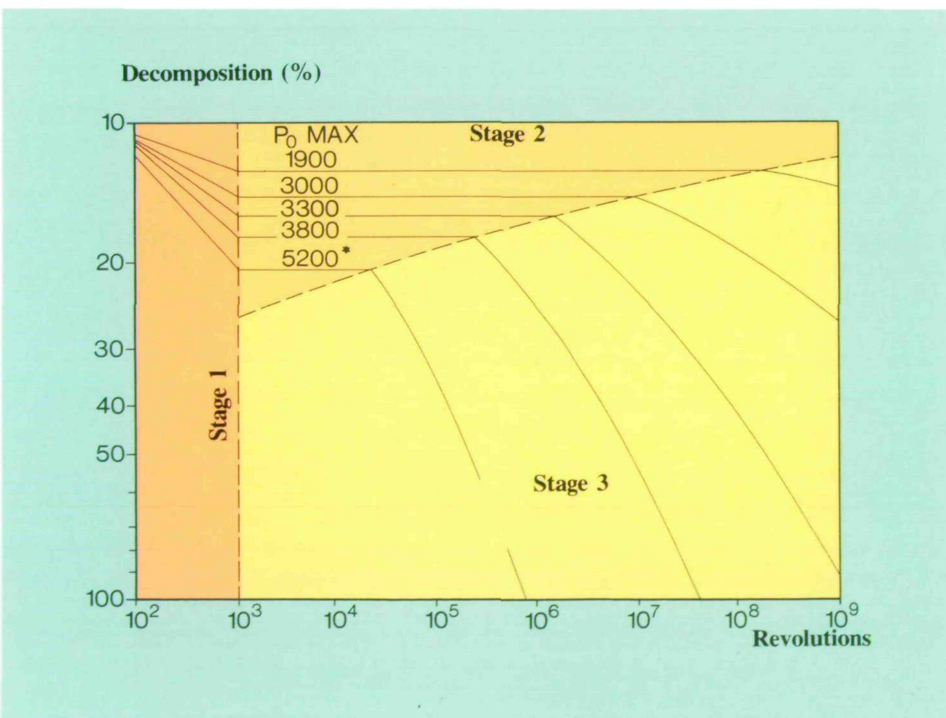


Fig 13 Retained austenite decomposition at 0,2 mm below the centre of the ball raceway in a 6309 inner ring as a function of the number of revolutions and the measured stresses P_0 (MPa). The solid lines represent the material response to contact loading during the shake-down, steady-state and instability stages. The diagrams represent the material response for 6309 inner rings with a raceway radius of 9,09 mm at a rotational speed of 6000 r/min, under full film oil lubrication and at a bearing operating temperature of 53 °C

containing fewer and smaller inclusions, accumulated microplastic deformation causes sharp textures which promote the formation of more regular-shaped spalls. Current test programmes indicate that the susceptibility to failure increases with Stage 3 development.

The local softening and microstructural changes in the sub-surface regions of rolling bearings imply a different resistance to fatigue failure from that in not run bearings. The new SKF life model can accommodate such variations by considering that the fatigue limit is a function of the number of cycles of testing, depending, for example, on the combination of stress, frequency of stressing and temperature.

The changes in the sub-surface regions of tested bearings are a consequence of the accumulation of microplastic deformation during cyclic stressing. The onset of the decay is dependent on a number of factors, including load, frequency and temperature. It is accompanied by local softening of the matrix. It is particularly significant that the duration of a "steady-state" stability decreases with increasing load. Conversely, at low contact stresses the transition to instability is only just commencing beyond 10^9 revolutions. This mechanism is consistent with the concept of a threshold stress for fatigue, which is in agreement with the analytical predictions from the new life theory.

Closing remarks

The continuous efforts of SKF to develop new processes for the melting and refining of high-cleanliness bearing steels and increase — by extensive metallurgical investigations — the understanding of the relationship between the microstructure of the heat treated bearing steel and its mechanical properties have significantly contributed to the high quality of the SKF bearing products of today. This article is an extract of the articles on material development published in BBJ 231 (1) and 231 (2).

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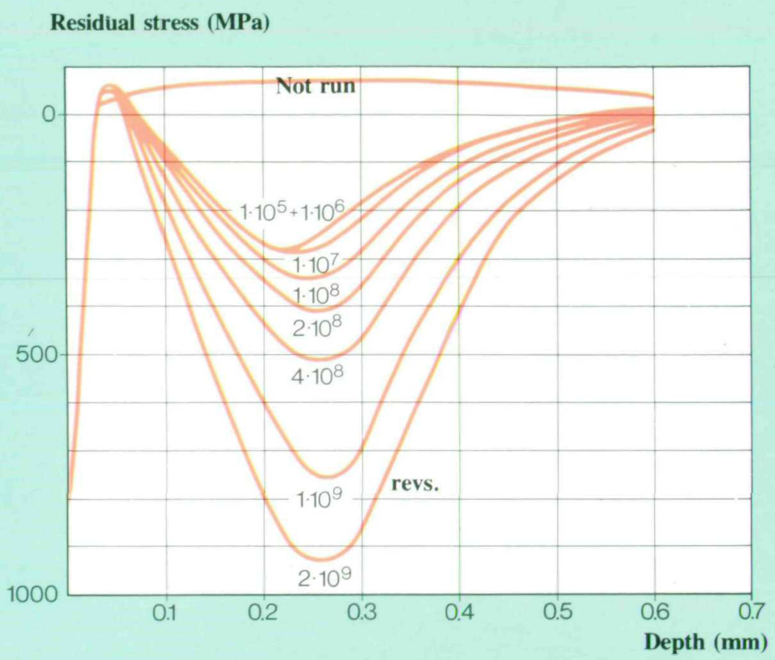

Fig 14 Residual stress as measured in the circumferential direction versus depth beneath the inner ring raceway of 6309 inner rings tested for various numbers of revolutions.
Test conditions: maximum contact stress: 3300 MPa, inner ring rotational speed: 6000 r/min, operating temperature: 53 °C, full film oil lubrication

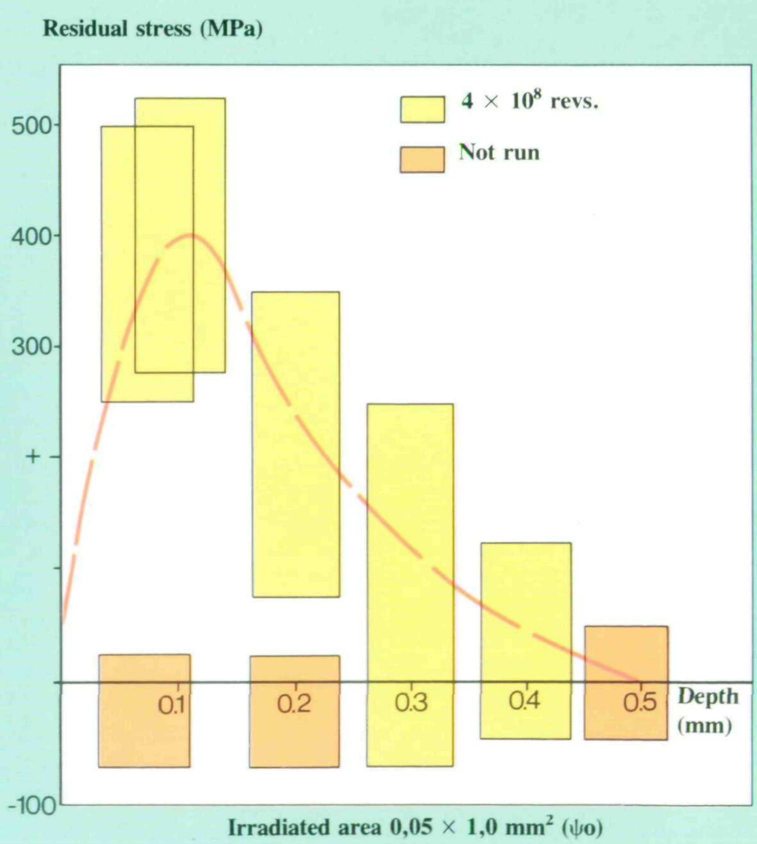

Fig 15 Residual stress as measured in the radial direction versus depth beneath the raceway of not run and overrolled 6309 inner rings. Test conditions: maximum contact stress: 3800 MPa, inner ring rotational speed: 6000 r/min, operating temperature: 53 °C. Depth determined on an electro-polished cross section through the inner ring


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