TESTING USED ROLLER BEARINGS FOR QUALITY AND SERVICE LIFE

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Roller bearings are important parts of machines and vehicles, likely to entrain serious breakdowns, damages, or even casualties when defective. In the planned preventive maintenance system (stiff cycle order), current practice of machine overhaul, bearings are exchanged without testing, after a given performance of running or a number of service hours. Because of the important standard deviation of bearing lives, and mainly, of the lack of information on the service time, on time of mounting the bearings, little used bearings that are perfectly flawless may happen to be sorted out rather than to be utilized up to the expected useful life at a needless increase of upkeep costs.

There is always a high number of bearings dismounted and sorted out untested. Remind of the two or three hundred bearings incorporated in a single high-power vehicle or agricultural machine. Certain bearings cost thousands of forints. Thus bearings dismounted in unknown state at bigger repair enterprises may be valued at estimated several million forints.

The idea arose that the scores of bearings dismounted certainly comprise several that can be reused. Bearings in perfect condition can be remounted or used in given, maybe subordinate applications with no concomitant life hazard (e.g. machines). A reliable, rapid bearing test method has to be developed for the fast classification of used, dismounted bearings. Such tests are likely to lead to conclusions that the time of bearing exchange could be postponed, the equipment overhaul periodicity increased.

Conventional testing of roller bearings involving bearing clearances, roller element and bearing groove surfaces, surface scaling, cracks and, maybe, bearing noise upon rolling is rather time and labour consuming. Besides of being labour intensive, these tests mostly damage or even destroy the tested bearings. Overall tests are unimaginable in the non-destructive way, without dismountig the bearing.

Experience and diagnostic methods now under development point to a relationship to exist between technical condition and noise i.e., vibration spectrum of the bearing [1—4]. Analysis of vibration patterns of sound and defective bearings permit to determine a critical vibration belonging to a given technical condition where the bearing still can further operate.

The objective following from the above imposed to solve the following problems:

— Development of a rapid, simple method of determining the condition of dismounted bearings relying on their vibration patterns.

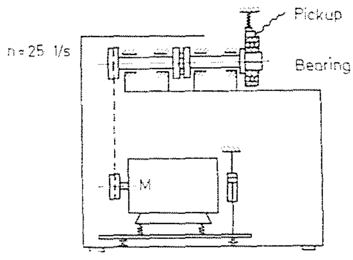
— Investigation by conventional methods of the bulk of bearings dismounted at a given repair workshop for eventual sound specimens fit to further operation. Examination of the possibility to distinguish them by simple vibration measurement.

— Comparison of service lives of new bearings to those of dismounted bearings rated by vibration measurement as satisfactory.

1. Bearing tester

To examine bearing characteristics by vibration measurement, the bearing has to revolve. Vibration of a roller bearing revolving in a bearing tester as a connected measurement system is superposed by those of the entire system. Therefore, a bearing tester is required to generate as little disturbing vibration as possible. Even the simplest equipment for actuating a bearing consists of a pivot bearings and drive such as that schematically shown in Fig. 1a. To minimize disturbing vibrations due to the tester, it comprises sliding bearings and continuous flat-belt drive. The equipment pivot edge can be supplied with an expansion sleeve to fix different bearings pushed on manually. During testing the outer ring is immobile, and the inner one is revolving. The outer ring of taper roller bearings is pressed by spring force to the rollers. The outer ring is contacted by the acceleration pickup. Fig. 1b is the photo of a such a bearing tester.

Bearing vibrations have been tested at 25 or 30 rps, generally in three bands (band I: 50 to 300 Hz; band II: 300 to 1800 Hz band III: 1800 to 10 000 Hz, 2.5 octaves wide) in conformity with service and foreign standard recommendations. Bearings are usually rated according to the mean square value of the vibration signal in each band (e.g. acceleration). Often complete spectra are recorded and compared [5]. In our tests complete vibration spectra from 50 to 15 000 Hz have been recorded, and effective values of vibration accelerations determined in each band. Thus, the electronic tester consisted of a pickup, a charge amplifier, a vibration analyzer, band filters and plotters. (Pickup KD 35a vibration amplifier RFT SM 233, wave analyzer FRA-2-a, X-Y plotter EMR-NE-233.)



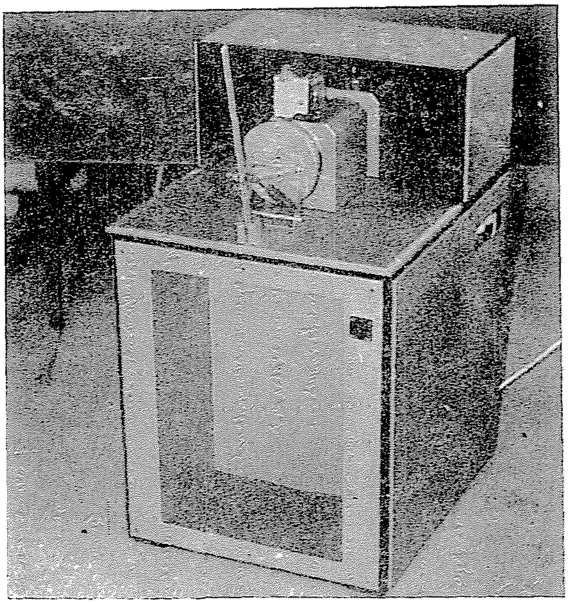


Fig. 1. a) Scheme of a bearing tester. b) Photo of a bearing tester

2. Comparative vibration analysis of bearings

50 used bearings each, dismounted from vehicles, were taken at random, and tested both in the conventional way, and by vibration measurement. Bearing clearances, impacts were measured, surfaces were macroscopically examined, roughness measured where possible without disjoining the bearing. For some bearings, rollers and grooves were tested for trueness to shape. Bearing vibration spectra were recorded by vibration measurement, and effective acceleration values determined.

The same tests were made on new bearings, taking 5 or 10 specimens from each type.

Vibration test results were invariably confirmed by the conventional test results. Flawless, sound bearings generated little vibration in any band.

Table 1

Effective vibration acceleration values (ms^{-2}) of bearings type 6212 ZN
(Band I: 50 to 300 Hz; band II: 300 to 1800 Hz; band III: 1800 to 10 000 Hz)

No.	Vibration value			No.	Vibration value		
	Band I	Band II	Band III		Band I	Band II	Band III
1	0.4	1.0	1.8	26	0	0.9	4.0
2	0	1.5	5.0	27	0	0.5	1.8
3	0.4	2.0	11.0	28	0.5	1.0	9
4	0	1.0	2.5	29	not evaluable		
5	0	1.0	5.0	30	injured, not evaluable		
6	9.0	13.0	11.0	31	0	3.0	6.0
7	0	2.0	8.0	32	0	0.5	8.0
8	0	1.5	7.0	33	0.5	1.5	7.0
9	0.4	0.8	2.2	34	0	0.9	9.0
10	0	2.0	7.0	35	0	0.9	1.8
11	0.2	1.0	4.0	36	0	1.0	8.0
12	0.1	1.0	4.0	37	1.5	1.5	3.5
13	0	0.8	2.0	38	17.0	5.0	8.0
14	0	1.8	0.9	39	0	1.5	2.5
15	1.0	2.0	3.0	40	1.0	1.8	14.0
16	0.2	0.8	4.5	41	0	0.8	1.8
17	0	0.8	6.0	42	2.5	5.0	10.0
18	0.6	2.0	15.0	43	0	1.0	3.5
19	0.1	2.5	8.0	44	0	0.8	2.5
20	0	1.8	4.0	45	1.0	1.0	2.0
21	0	1.0	3.5	46	0.2	1.0	3.0
22	0	2.0	2.5	47	0.5	1.0	5.0
23	0.1	1.0	6.0	48	0	1.0	5.0
24	0	0.7	2.5	49	0	1.0	4.0
25	4.0	3.0	8.0	50	0	0.8	2.5
1u	0	0.4	0.6	5u	0	0.2	0.3
2u	0	0.3	0.9				
3u	0	0.2	0.3				
4u	0.2	0.7	0.6				

u denotes new bearing

Effective vibration acceleration values in three bands generated by bearings type 6212 ZN are presented in Table 1 as an example.

Vibration spectra of some new bearings are seen in Fig. 2, and those of used bearings 1, 25 and 39 in Fig. 3. Rolling surfaces of the new bearings were perfectly sound. Bearing 1, however, with higher vibration values than those for the new bearings, exhibited signs of initial damaging. Bearings 25 and 38, exhibiting much more intensive vibrations than the new ones, were inflicted by damaged, cracked rolling surfaces. Fig. 4 shows inner rings and rollers of bearings 1 and 25. In spite of unfavourable light reflections, surface of bearing 1 is seen to be almost sound, evenly worn. The surface of bearing

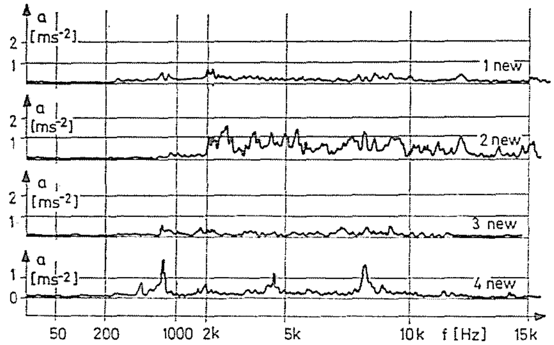


Fig. 2. Vibration spectra of new bearings

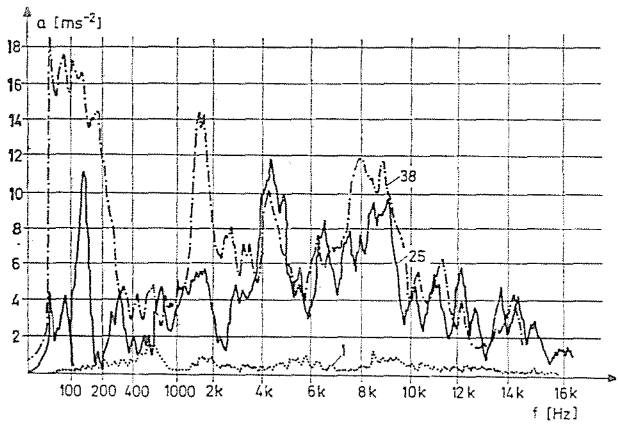


Fig. 3. Vibration spectra of used bearings

25 is rough with coarse scratches, seizures, rusty spots. Based on bearing geometries (e.g. number of rollers) and rps values, theoretical frequencies and those pertaining to one assumed flaw have been determined. Several vibration patterns display peaks corresponding to these frequencies. For heavily worn-out bearings, peaks are denser, higher for higher frequencies, flaws generating them may be assumed to be multiplied, that is, rollers have multiple defects, indentations or scales.

Similar tests were made on taper roller bearings types 32217 and 32308. Vibration patterns of three taper roller bearings, a new one, a used but macroscopically sound one, and a discarded one, are seen in Fig. 5. Also in this case, unambiguous correlation was found throughout between vibration intensity and the state of grooves and rollers. No such thing as e.g. a relatively favourable vibration spectrum for a rough, scaled groove was encountered. On the other hand, the groove surface might seem sound, at a bad vibration

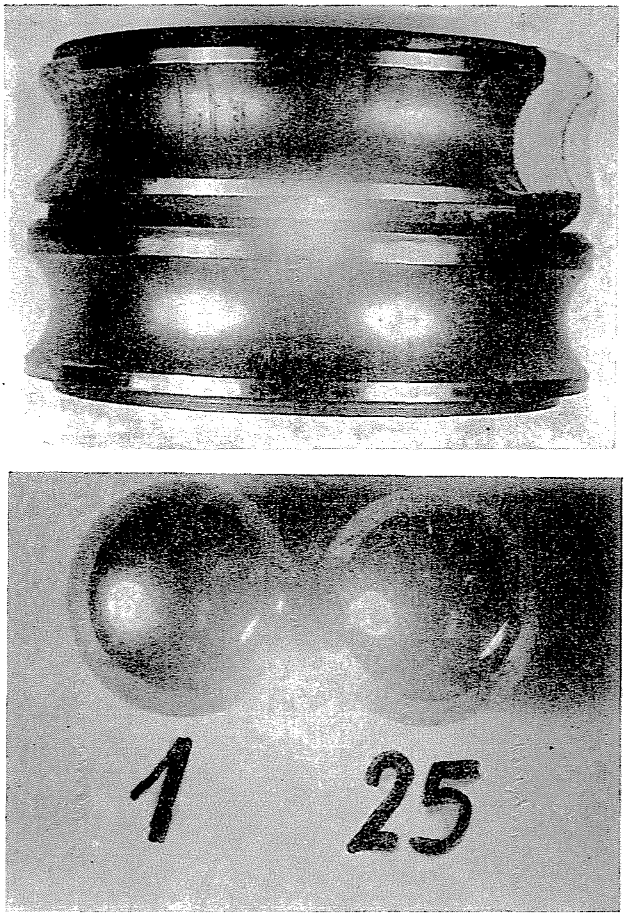


Fig. 4. Surfaces of the inner ring and of the roller element of used bearings

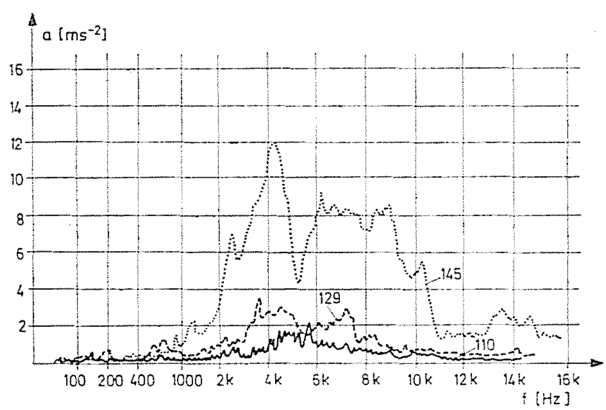


Fig. 5. Vibration spectra of new and used taper roller bearings

Table 2

Effective vibration acceleration values (ms^{-2}) of taper roller bearings type 32308
(Band I: 50 to 300 Hz; band II: 300 to 1800 Hz; band III: 1800 to 10 000 Hz)

No.	Vibration value			No.	Vibration value		
	Band I	Band II	Band III		Band I	Band II	Band III
101	0.1	0.8	3.4	126	0.8	2.4	13
102	0.1	1.1	5	127	—	—	—
103	4.5	6	11	128	—	—	—
104	—	—	—	129	0.1	1	2.5
105	8.5	3.6	11	130	not evaluable		
106	0	0.4	4	131	0.2	1	8
107	0.3	1.2	7	132	0.1	1	2.8
108	—	—	—	133	0.3	1.1	4.3
109	0.1	0.5	2.1	134	0.1	0.9	1.6
110	0.1	0.4	2.5	135	—	—	—
111	—	—	—	136	0	1	4
112	0.5	1	8	137	0	0.8	2.5
113	0.1	0.6	3.5	138	0	0.7	1.8
114	0.1	0.6	4	139	0.1	1.1	3.5
115	0.1	0.8	3.5	140	0.1	0.9	2.5
116	0.1	0.6	3	141	—	—	—
117	0.4	1	6	142	—	—	—
118	0.4	2	12	143	—	—	—
119	0.3	1.4	5	144	—	—	—
120	0.1	0.4	2	145	0.3	1	7
121	0	0.8	6	146	0.3	2	7
122	0.8	1	8	147	—	—	—
123	0	0.7	2.6	148	—	—	—
124	—	—	—	149	0.2	0.5	1.8
125	0.1	1	5	150	0	0.4	1.7
101u	0	0.2	1.8	104u	0	0.2	1.8
102u	0	0.4	2.4	105u	0.1	0.4	1.8
103u	0	0.2	1.8				

spectrum. In such cases, a detailed investigation invariably demonstrated some waviness or misshape. Table 2 shows vibration values for bearings type 32308, and Fig. 6 the densities of effective values of vibration accelerations in each band. (Unfortunately, several out of the 50 bearings could not be tested since their outer and inner rings were confused.) From among the bearings of the worst specra, the cracked outer ring of bearing No. 145 is seen in Fig. 7, and its failed roller, with marks of fatigue, in Fig. 8.

Along the service life of bearings, vibration acceleration values vary with the wear, similar to the overall wear diagram. At the start of abrupt bearing destruction, vibration intensity may grow to the tenfold, or sometimes to the hundredfold [6], [7]. Based on the comparison of a high number of conventional tests (surface examinations, gap and wear measurements) and

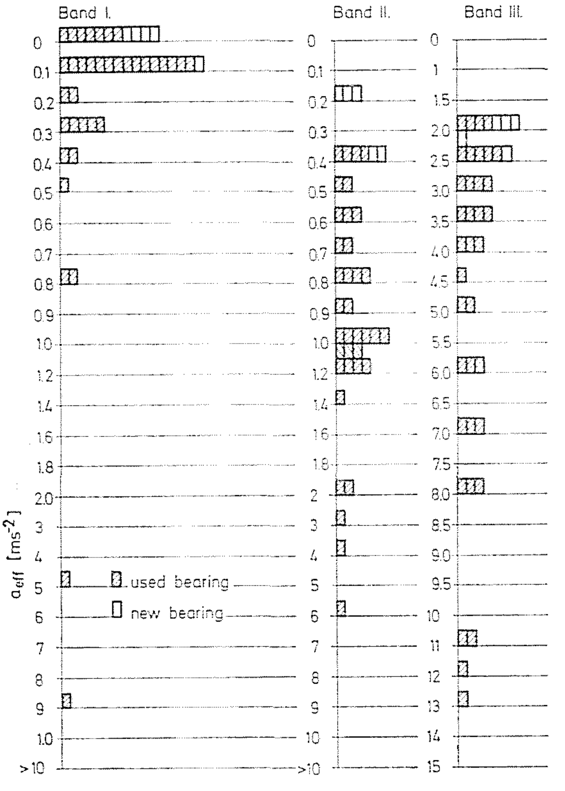


Fig. 6. Distribution of bearing vibrations between bands (type 32308)

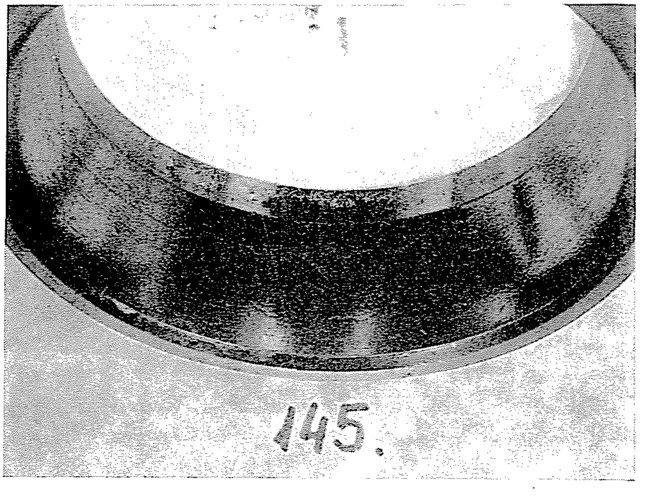


Fig. 7. Failed outer bearing groove of a used bearing type 32308

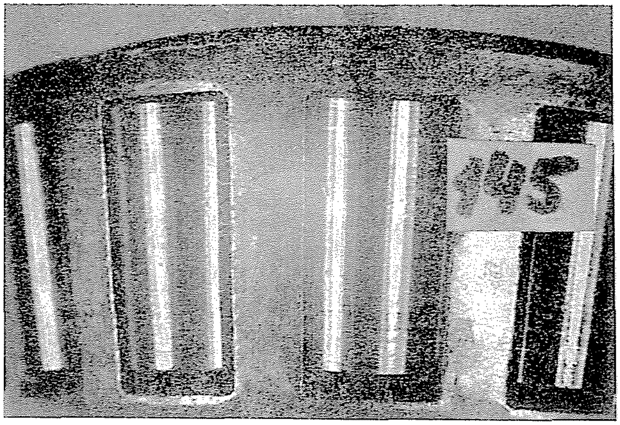


Fig. 8. Failed roller of a used bearing (type 32308)

generated vibrations, tolerable vibration values of still acceptable bearings were assessed by the time to the two- or threefold of new ones. Thus, bearings were rated as fit to reuse where vibration accelerations exhibited at most two or three times the effective values of new ones in each band. These bearings were assumed to work undisturbed until the next overhaul cycle. This limit is, however, likely to be modified according to operational tests.

This way, about 10% of the tested ball bearings type 6212 ZN, 40% of the more valuable taper roller bearings type 32217, and 38% of bearings type 32308 were rated as fit to reuse.

3. Investigation of the service lives of bearings

Comparative fatigue tests of new bearings and of those rated by vibration measurements as satisfactory were performed on ball and taper roller bearings. Tests made on the latter will be described below. Groups of ten taper roller bearings type 32308 each were composed, checking the quality and geometry of every bearing. Vibration spectra were recorded, and vibration acceleration values in the three bands were determined.

The group of used bearings was composed of bearings rated as satisfactory by vibration measurement. Composition of the group out of the fifty tested bearings caused no difficulties. The bearings were tested similarly as for the new ones.

Equipment for testing the service lives of bearings

Initially, in conformity with the original plans, a special machine was proposed to be constructed for testing the service lives of the bearings. To this aim, several testers were examined. Nevertheless it was decided to perform the tests on some appropriate testing machine available in this country. Namely, manufactured testers were assumed to have been developed relying on available experience, besides of the anticipated cooperation of experts in testing bearing service lives. In view of bearing dimensions and other circumstances, the tests were made on the bearing fatigue tester type RA-3k in the laboratory of the Hungarian Roller Bearing Works, with the contribution of the specialist team.

In the bearing fatigue tester, the proper load, either radial or axial, is produced by a hydraulic equipment. During the test, the bearings obtained forced lubrication. For every bearing, temperature and effective vibration acceleration values were measured. For an eccessive increase, the equipment automatically stopped.

To provide fatigue under identical conditions, bearings had to be forced on the pivot with equal overlapping. To provide for an adequate overlapping, bearings with variable borings had to be mounted on pivots of the boring size. To be pair-wise assembled and fatigue tested, bearings had to be selected not only from the aspect of inner boring but also of the radial clearance of the outer shell diameter (for ball bearings). Namely, deviating characteristics might lead to unfavourable run, seizure, uneven loading of the bearings, hence increase the testing error.

The fatigue apparatus operated at 46.6 rps. Fatigue test started with a short no-load run, then bearings were loaded by the equivalent of one third of the basic local capacity C. The ratio of radial to axial force was 2 to 3. Service life test results have been compiled in Table 3, 100% being the expected, cal-

Table 3

New bearings

No.	102u	101u	105u	103u	109u	104u	108u	110u	107u	106u
Service life %	45	91	144	264	404	808	1339	2244	2666	2740

Used bearings

No.	109	134	129	132	116	140	150	120	110	137
Service life %	37	124	224	296	296	536	970	1397	1615	2576

culated lifetime. The diagrams of service life vs. damage probabilities of new and used bearings are seen in Fig. 9. In calculations, assumption of Gaussian rather than Weibull's distribution in the evaluation was assumed to be a small error. Based on service life values, mean service life and standard deviation values were determined, plotted as straight lines for new and used bearings. Lines for new and used bearing groups are seen to be rather close. Service lives in either group were multiples of rated service life values. According to calculations, 95% of new bearings exposed to loading conform to their expected service life will be in working condition, while under the same conditions, only 17% of used bearings will be damaged, and 83% wil work. Provided bearings of a machine will be dismounted after the expected service life and selected by vibration measurement under the described conditions to be reused again for the expected service life, 30% of the used bearings would be damaged. Under the same conditions, 80% of new bearings would continue to be serviceable.

Bearing wear is seen in Fig. 10 to be proportional to the working time. Relative wear values of new bearings appear to be higher. No increased wear rate of used bearings compared to new ones was observed. According to the general wear diagram, new bearings are likely to wear faster in the run-in period, enhancing wear values.

Wear contributes to altering the surface properties and trueness to shape of rollers and grooves. Trueness to shape of a new and a used taper roller bearing determined in a Goulder Micron instrument are seen in Fig. 11. Unfortunately no such thing as shapes of one and the same roller when new and

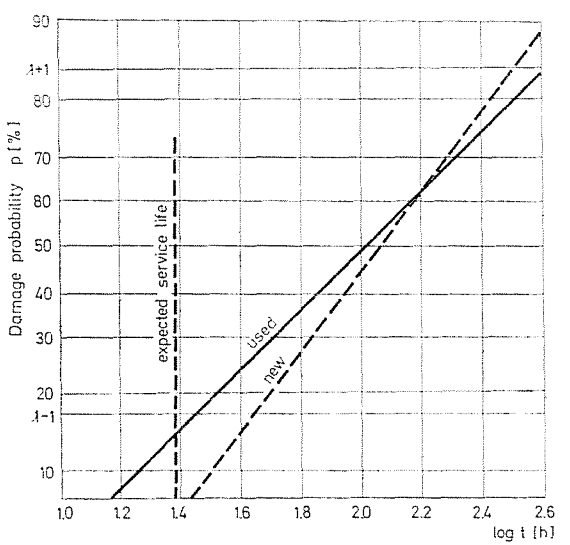


Fig. 9. **Diagram of service life damage probability of new and used bearings**

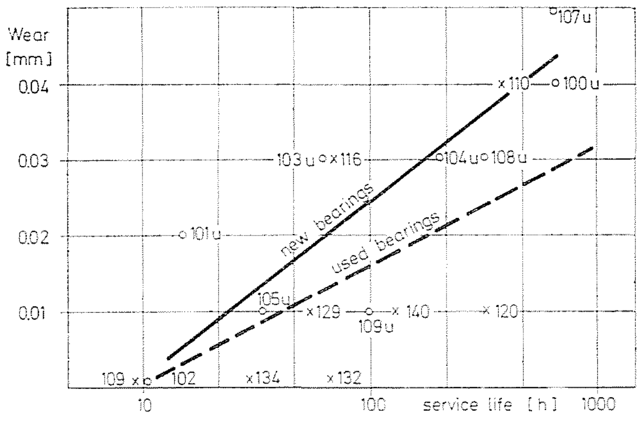


Fig. 10. Wear of taper roller bearings

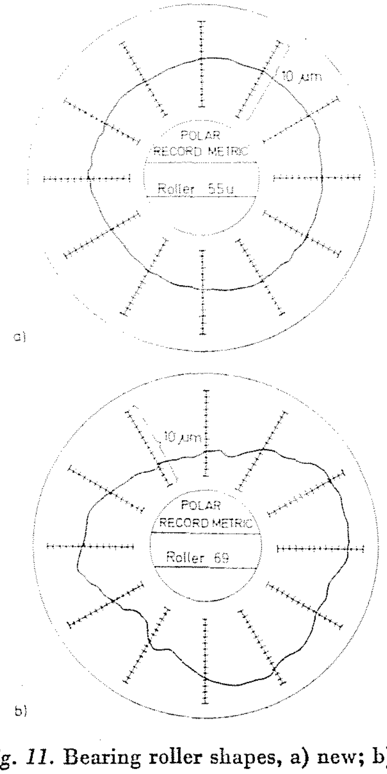


Fig. 11. Bearing roller shapes, a) new; b) used

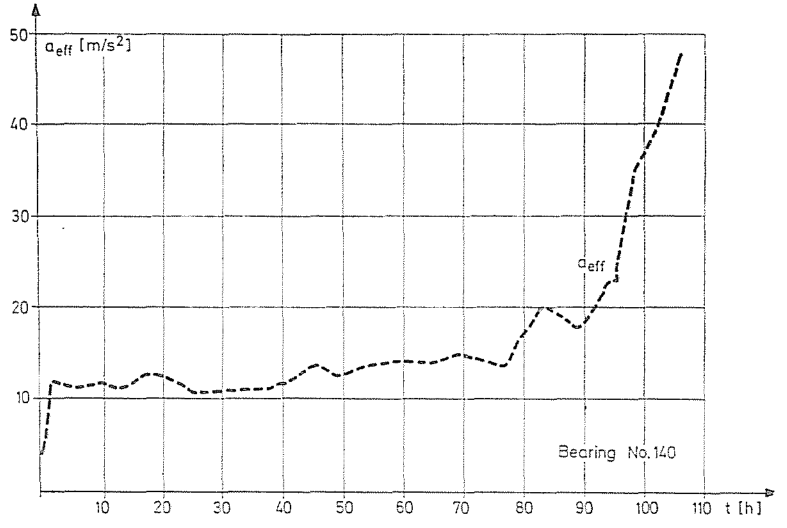


Fig. 12. Vibration intensity variation of a taper roller bearing in the service life test

after use can be demonstrated, since it would involve to dismount and reassemble the bearing, likely to induce further incertainties in the test.

As mentioned above, during fatigue tests, effective vibration acceleration values were measured, to be fed in a computer, to plot testing time vs. vibration acceleration diagrams. Unfortunately, the important fundamental mode of the machine, and the interferences caused the resulting diagrams often to differ from the anticipated ones. Fig. 12 represents the effective vibration acceleration value of bearing No. 140 vs. fatigue time.

Summary

Much of bearings dismounted from vehicles in course of preventive maintenance were found in tests to be fit to reuse. These bearings can be selected relatively fast and reliably by vibration measurement, rather than by the conventional testing method. Average service life of bearings rated as sound by vibration measurements is 75% of that of new, ones, although service lives of much of both new and used bearings exceeded several times the expected service lives. Reducing somewhat the reliability criteria, used bearings rated as sound by vibration measurement can be resubstituted to places not critical for life hazard. For several structural units, an economical approach to reliability would permit reuse of used bearings rated as sound by vibration measurement.

References

1. Schröder, B.: Lagerüberwachung mit SPM Stoßimpulsmethode. VGB. Kraftwerkstechnik. No. 8. 1977.
2. Mathiszk, S.: Früherkennung von Wälzlagerschaden. Die elektrische Ausrüstung No. 16. 1975.

3. ERIKSON, L.: Surveillance de l'état des roulements par la mesure de l'onde de choc. Mécanique Matériaux Electricité. No. 325. 1977.
4. BROWN, P.: Preventive maintenance of machinery bearings. Maintenance Engineering 20. No. 8. 1976.
5. POZNIAK, E. L.—ZUBERENKOV, B. I.: O raschete vibratsy obislovennihk nesoveshenstvom podripnikov kachenia. Mashinovedenie, No. 5, 1976.
6. HAURI, H.: Wälzlagerüberwachung. Technische Rundschau. Bern. No. 49. 1978.
7. BAUMGARTNER, K.: Die Beurteilung des Walzlagerzustandes mit der SPM-Methode in einem Kraftwerk. Der Maschinenschaden 51. No. 6. 1978.
8. MOLNÁR, L.—VARGA, L.: Design of Roller Bearing Supports. (In Hungarian). Műszaki Kiadó, Budapest, 1972.

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