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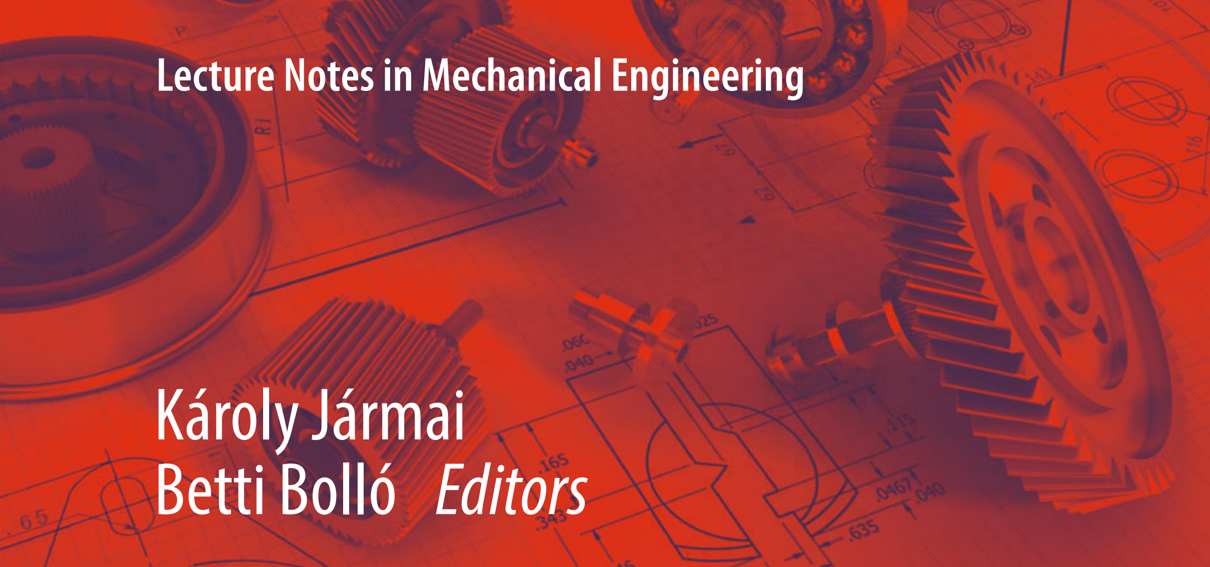

Károly Jármai
Betti Bolló *Editors*

Vehicle and Automotive Engineering

Proceedings of the JK2016, Miskolc, Hungary

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Lecture Notes in Mechanical Engineering

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Editors

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Editors
Károly Jármai
Miskolci Egyetem
University of Miskolc
Miskolc, Egyetemvaros
Hungary

Betti Bolló
Miskolci Egyetem
University of Miskolc
Miskolc, Egyetemvaros
Hungary

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Preface

The production of car and vehicle industry increased greatly in the past decades. People would like to reach the destination as quickly as possible. The quick transportation of persons and goods is more and more important. This is the case in Hungary, where the improvement of the car industry was great in the past decades. Great car producers settled here like Mercedes Benz, Audi, Suzuki, Opel and also small and medium enterprises connected to car element production have developed greatly.

Education has to follow this trend. Vehicle engineering training has a long tradition in Hungary. At the Budapest Technical University and Economics, at the István Széchenyi University in Győr they have a long-term experience in this kind of training. At the University of Miskolc, which is a successor of the Mining and Metallurgical Academy, the first technical higher educational institution on the Earth, founded in 1735, the mechanical engineering training started in 1949. The industrial demand forced the university to start vehicle engineering training also. It was accredited in 2015 and started this year.

The main requirements for cars and car elements are safety, manufacturability and economy. Safety against different loads such as permanent and variable actions is guaranteed by design constraints on stresses, deformations, stability, fatigue, eigenfrequency, while manufacturability is considered by fabrication constraints. The economy is achieved by minimization of the cost.

The main topics of the conference are as follows:

Design: Acoustic investigations, Car electronics, Autonomic vehicles, Fatigue, Industrial applications, Vehicle Powertrain, Modelling and simulation of vehicle informatics and electronic systems, Vehicle navigation, Visual systems of vehicles, Mechatronics, Numerical methods FEM and BEM applications, Vibration and damping, Stability calculations, Structural materials, Structural safety, Structural connections, Analysis and design of structural elements, Design guides, Fracture mechanics, Thin walled structures, Driver assist systems, Hybrid and electric cars.

Fabrication: Forming technologies, Surface protection, Production logistics, Manufacturing technologies, Welding technologies, Heat treatment, Innovative casting technologies, Industrial applications, Maintenance, Environmental protection, Lean technologies, Quality assurance, Gluing technologies, Production, Testing.

Economy: Life cycle assessment, Fabrication costs, Industrial applications, Cost engineering, Structural optimization.

Education: Vehicle engineering training, Dual training, Industrial practice, Training techniques, Training materials.

It is a great pleasure to organize this conference, to give participants an opportunity to show and discuss the new research results in a friendly atmosphere.

The organizers wish all participants successful days to collect new ideas and get new acquaintances.

Miskolc, Egyetemvaros, Hungary
October 2016

Károly Jármai
Betti Bolló

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- Foundation for the Development of the Education at the University of Miskolc,
- TÁMOP 4.2.1.B-10/2/KONV-2010-0001 entitled "Increasing the quality of higher education through the development of research—development and innovation program at the University of Miskolc supported by the European Union, co-financed by the European Social Fund."
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László Kota, assistant professor,
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October 2016

Károly Jármai
Betti Bolló

Contents

About the Editors

Dr. Károly Jármai is Professor at the Faculty of Mechanical Engineering at the University of Miskolc, where he graduated as a mechanical engineer and received his doctorate (dr.univ.) in 1979. He teaches design of steel structures, welded structures, composite structures and optimization in Hungarian and in the English language for foreign students. His research interest includes structural optimization, mathematical programming techniques and expert systems. Dr. Jármai wrote his C.Sc. (Ph.D.) dissertation at the Hungarian Academy of Science in 1988, became a European Engineer (Eur. Ing. FEANI, Paris) in 1990 and did his habilitation (dr.habil.) at Miskolc in 1995. Having successfully defended his doctor of technical science thesis (D.Sc.) in 1995, he subsequently received awards from the Engineering for Peace Foundation in 1997 and a scholarship as Széchenyi professor between the years 1997–2000. He is the co-author (with József Farkas) of four books in English *Analysis and Optimum Design of Metal Structures*, *Economic Design of Metal Structures, Design and optimization of metal structures, Optimum design of steel structure,* and three monographs in Hungarian, and has published over 610 professional papers, lecture notes, textbook chapters and conference papers. He has about 770 independent citations. He is a founding member of ISSMO (International Society for Structural and Multidisciplinary Optimization), a Hungarian delegate, vice chairman of commission XV and a subcommission chairman XV-F of IIW (International Institute of Welding). He has held several leading positions in GTE (Hungarian Scientific Society of Mechanical Engineers) and has been the president of this society at the University of Miskolc since 1991. He was a visiting researcher at Chalmers University of Technology in Sweden in 1991, visiting professor at Osaka University in 1996–1997, at the National University of Singapore in 1998 and at the University of Pretoria several times between 2000–2005.

Dr. Betti Bolló is Associate Professor at the Department of Fluid and Heat Engineering, University of Miskolc, Hungary. She received her M.Sc. degree from the University of Miskolc in Information Engineering (Systems of Power Engineering) in 2003. Her research interests include computational fluid dynamics and internal combustion engines. She wrote her dissertation (Ph.D.) at the Hungarian Academy of Science in 2013. The theme of her dissertation is a numerical investigation of flow past and heat transfer from a heated circular cylinder.

Part I
Design

Investigation of Rolling Element Bearings Using Time Domain Features

Dániel Tóth, Attila Szilágyi and György Takács

Abstract Rolling element bearings can be found widely in domestic and industrial applications. They are important components of most machinery and their working conditions influence the operation of the entire machinery directly. Bearing failures may cause machine breakdown and might even lead to catastrophic failure or even human injuries. In order to prevent unexpected events, bearing failures should be detected as early as possible. Different methods are used for the detection and diagnosis of bearing defects. These techniques can be classified as noise analysis, acoustic measurements, wear debris detection, temperature monitoring, vibration analysis etc. Vibration signals collected from bearings carry detailed information on machine health conditions. This paper deals with a bearing test procedure which based on vibration analysis.

1 Introduction

Vibration monitoring is one of the essential tool that allows to determine the mechanical health of different components in a machine. When the assessment of a ball bearing is performed by vibration analysis, several signal processing techniques can be considered. These techniques can be performed within either the time or the frequency ranges. Among these methods the time domain features are the most appropriate with random signals, where other signal analysis methods are not suitable. These methods facilitate fast data processing and computation. Numerous time domain statistical parameters have been used as trend parameters to detect the

D. Tóth (✉) · A. Szilágyi · G. Takács
University of Miskolc, Miskolc, Hungary
e-mail: toth.daniel@uni-miskolc.hu

A. Szilágyi
e-mail: szilagyi.attila@uni-miskolc.hu

G. Takács
e-mail: takacs.gyorgy@uni-miskolc.hu

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bearing failures. The most frequently applied stochastic features are the root-mean-square (RMS) value, peak value, skewness, impulse factor, shape factor, clearance factor, crest factor and kurtosis [1, 2].

2 Bearing Test Device

Rolling element bearing condition monitoring can be accomplished by using a test instrument. Such a device is located at University of Miskolc, Department of Machine Tools. The test device is illustrated in Fig. 1.

The equipment is suitable for performing the bearing fatigue and measurement investigations. The particular symbols have the following meanings:

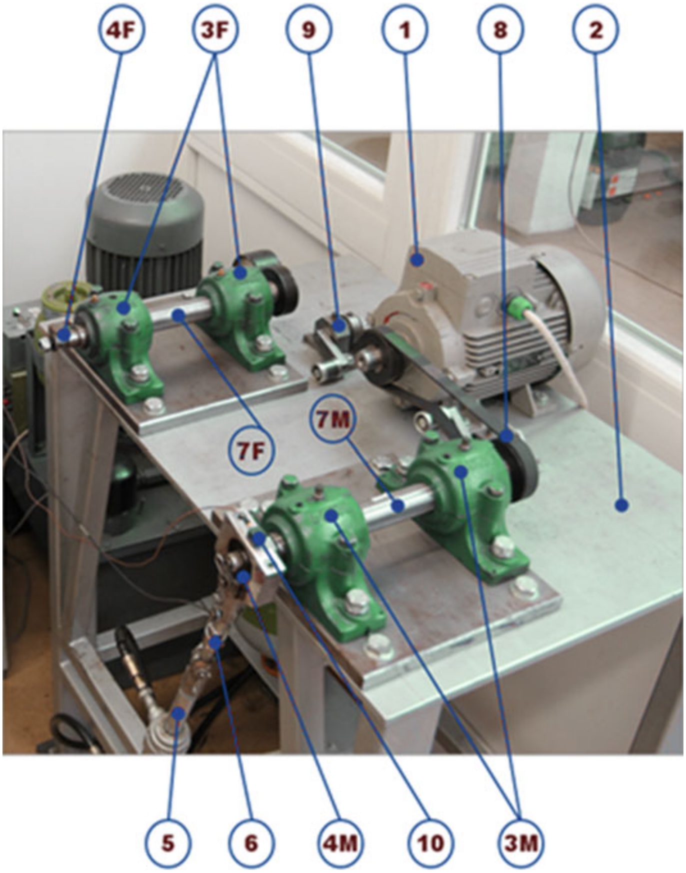

Fig. 1 Experimental test rig

- 1: three-phase motor,
- 2: rigid table,
- 3F: supporting bearings of fatigue side,
- 3M: special supporting plain bearings of measurement side,
- 4F: fatigued bearing position,
- 4M: measured bearing,
- 5: double-acting hydraulic cylinder,
- 6: load cell, the adjustment of hydraulic load,
- 7F: fatigue test shaft,
- 7M: measurement test shaft,
- 8: length ribbed belt,
- 9: belt tensioner,
- 10: piezoelectric vibration accelerometer.

During the measurements the "7M" shaft works at the given rotational speed (1500 min^{-1}), while the "6" hydraulic cylinder exerts artificial load (1 kN) for the "4M" bearing.

3 Description of Investigation

Fundamentally, two proceedings are used for the experimental analysis of rolling element bearings. One method is the fatigue test when the bearings operate until they get permanent damage, and we measure their vibration trends meanwhile. However, the process takes relatively long time, but it can be accelerated with the bearing overload and increased rotational speed. Another technique is the production of one or more artificial failure of the elements of bearings. In this case the vibration signal should be measured and compared to data of faultless bearings. According to the literature [3–5], generally this may use methods such as spark erosion, acid etching, scratching or mechanical indentation. In this research, we used a well reproducible method to create artificial faults. A Rockwell hardness tester applied to make defects to the inner ring of bearings. This method needs a bearing with plastic cage, because it should be disassemble and assemble non-destructively. Figure 2 shows the ball bearing type 6303 which used during experiments.

Fig. 2 The test bearing and the artificial defect on the inner ring

As it was written previously a Rockwell hardness tester is used to cause local defects. The type of this machine is HR—150A. It is suitable for examining the effects of three types of loads. The major load values are 60, 100 and 150 kg. The effects of each loading were examined more than 10 times.

Optical examination can be applied to measure failure size. Polarising microscope is widely used for higher resolution. Carl Zeiss Jenavert polarising microscope is applied to inspect the extent of the defect.

The average extent of the failure is 265 μm in diameter in case of 60 kg load, 411 μm in diameter in case of 100 kg load and 478 μm in diameter in case of 150 kg load. The following illustrations show the effects of different loads (Fig. 3).

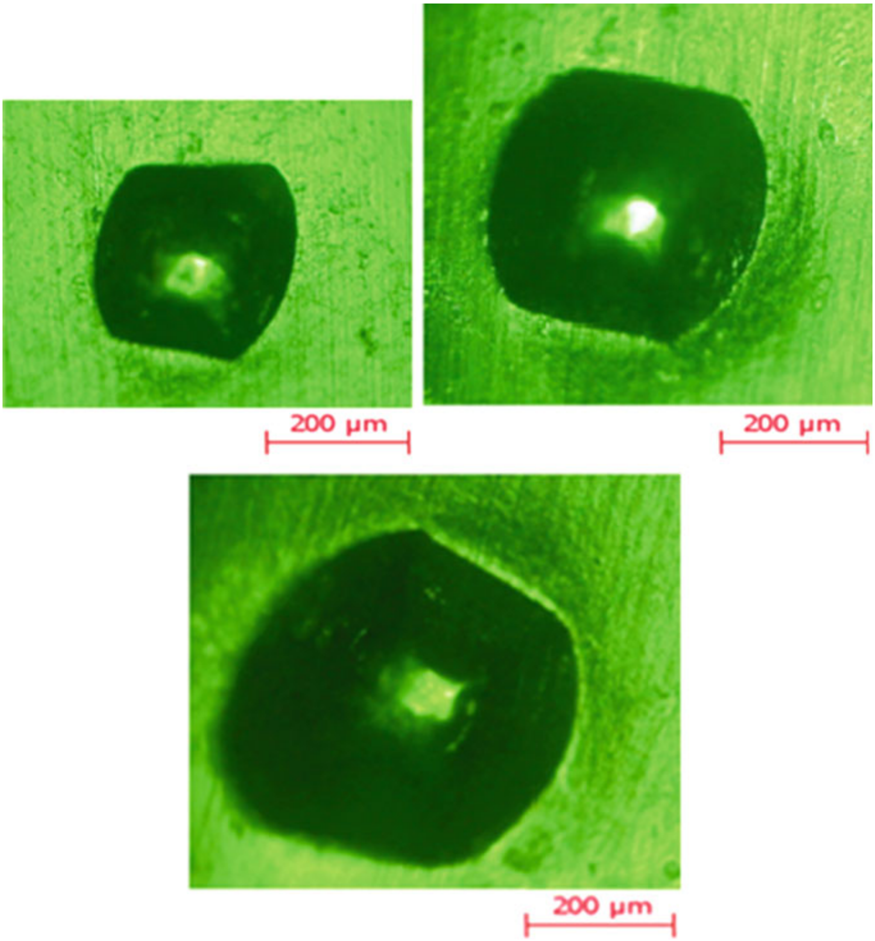

Fig. 3 Inner ring defects in case of 60 kg, 100 kg and 150 kg load (15 times magnification)

4 Analysis of Measurements

During the experiment, first of all the vibration patterns were measured from the examined bearing using piezoelectric vibration accelerometer (the type of it is Kistler 8632C50). After that the artificial defect was created and vibration patterns were measured again. It is followed by time-domain tests during which statistical features have been calculated. These stochastic indexes can be calculated by using the formulas below (Fig. 4).

The measurement cycles are performed at 9.6 kHz sampling frequency. Five vibration samples and 16,384-element samples were taken within each cycle. Statistical features were calculated based on sampled values. These parameters were computed by a program code, which runs in Maple mathematical software. Table 1 contains the statistical parameters in case of 60 kg load.

Table 2 includes the stochastic features in case of 100 kg load.

It is visible that the most of the parameters have doubled under this load. Table 3 contains the statistical parameters in instance of 150 kg load.

It is clearly visible that the statistical parameters of a defective bearing tend to be higher than the values of a normal bearing. The percentage increase is depicted in Fig. 5.

According to the graph it is clear that the Standard deviation, the Peak value and the RMS were the most sensitive to this artificial error. Nevertheless, it is obvious that the Kurtosis and the Skewness also have good correlation.

Fig. 4 Calculation of stochastic features [1]

Peak Value	$\frac{1}{2}\left(max(x_i) - min(x_i)\right)$
Root mean square	$\sqrt{\frac{\sum_{i=1}^{N}(x_i)^2}{N}}$
Crest Factor	$\frac{\text{Peak Value}}{\text{RMS}}$
Skewness	$\frac{\sum_{i=1}^{N}(X_i - \bar{X})^3}{(N-1)S^3}$
Kurtosis	$\frac{\sum_{i=1}^{N}(X_i - \bar{X})^4}{(N-1)S^4}$
Standard deviation	$\sqrt{\frac{\sum_{i=1}^{N}(x_i - x_m)^2}{N-1}}$

Table 1 Statistical parameters in new and damaged conditions (60 kg load)

	Peak value	RMS	Kurtosis	Skewness	Standard deviation		Peak value	RMS	Kurtosis	Skewness	Standard deviation
1	0.5688	0.0761	3.9468	1.0173	0.0461	1	0.6569	0.0792	4.3318	1.0370	0.0480
2	0.6150	0.0813	3.8500	1.1566	0.0485	2	0.6743	0.0831	4.1450	1.1802	0.0491
3	0.5632	0.0784	3.9200	0.9854	0.0463	3	0.6242	0.0790	4.4052	1.0209	0.0475
4	0.6369	0.0759	4.2139	1.0142	0.0449	4	0.6840	0.0784	4.7605	1.0403	0.0467
5	0.6807	0.0730	4.3995	1.0921	0.0442	5	0.7643	0.0802	4.8643	1.3295	0.0480
6	0.5376	0.0746	3.9534	1.0352	0.0523	6	0.6421	0.0755	4.5431	1.0723	0.0542
7	0.6712	0.0865	3.7533	0.8960	0.0508	7	0.7823	0.0921	4.0854	0.9302	0.0546
8	0.5114	0.0864	3.9900	1.0142	0.0441	8	0.6432	0.0887	4.3586	1.0274	0.0452
9	0.5255	0.0976	3.9631	1.0532	0.0553	9	0.6422	0.0992	4.2961	1.0689	0.0568
10	0.5931	0.0867	3.4861	0.8991	0.0521	10	0.7053	0.0888	3.8716	0.9582	0.0529
11	0.6274	0.0974	3.8255	1.0809	0.0580	11	0.7255	0.1032	4.2016	1.0904	0.0583

Table 2 Statistical features in new and damaged conditions (100 kg load)

	Peak value	RMS	Kurtosis	Skewness	Standard deviation		Peak value	RMS	Kurtosis	Skewness	Standard deviation
1	0.6225	0.0827	3.9236	1.1359	0.0477	1	1.1337	0.1482	5.2670	1.6243	0.0906
2	0.6508	0.0776	4.6827	1.1124	0.0471	2	1.1450	0.1425	6.1030	1.4740	0.0884
3	0.7425	0.0806	4.9360	1.1823	0.0495	3	1.3751	0.1461	6.7832	1.7498	0.0920
4	0.6674	0.0701	4.7007	1.3366	0.0435	4	1.1821	0.1260	6.5078	1.8236	0.0778
5	0.6150	0.0813	3.8500	1.2566	0.0485	5	1.0752	0.1487	4.7519	1.6767	0.0861
6	0.6569	0.0779	4.3318	1.0370	0.0463	6	1.1550	0.1385	5.2563	1.3732	0.0820
7	0.5618	0.0628	4.8172	1.2033	0.0385	7	1.0103	0.1120	6.2307	1.6778	0.0728
8	0.5970	0.0819	3.8334	0.9852	0.0488	8	1.0275	0.1456	4.9583	1.3146	0.0873
9	0.6475	0.0757	4.3553	1.0751	0.0459	9	1.1539	0.1358	5.5933	1.4559	0.0848
10	0.6527	0.0763	4.4994	1.0673	0.0455	10	1.1633	0.1360	5.6846	1.4334	0.0832
11	0.6705	0.0782	4.5762	1.2543	0.0484	11	1.1949	0.1366	5.8274	1.7112	0.0915

Table 3 Statistical indexes in new and damaged conditions (150 kg load)

	Peak value	RMS	Kurtosis	Skipness	Standard deviation		Peak value	RMS	Kurtosis	Skewness	Standard deviation
1	0.5824	0.0681	4.0148	0.9524	0.0406	1	1.5039	0.1683	6.0793	1.5524	0.1133
2	0.6386	0.0747	4.2957	1.0603	0.0453	2	1.4868	0.1816	6.5047	1.6997	0.1244
3	0.6777	0.0847	4.3517	1.1184	0.0554	3	1.5195	0.2044	6.5895	1.8040	0.1561
4	0.5527	0.0644	3.7721	1.0339	0.0428	4	1.4658	0.1561	6.0891	1.6297	0.1203
5	0.6292	0.0703	4.9252	1.3477	0.0431	5	1.5429	0.1570	7.3456	2.1513	0.1215
6	0.5875	0.0696	3.8739	0.9273	0.0415	6	1.4876	0.1402	6.1514	1.4710	0.1135
7	0.7160	0.0914	4.4818	1.4629	0.0564	7	1.5337	0.1945	6.5947	2.2913	0.1563
8	0.6309	0.0663	4.4436	1.2635	0.0411	8.	1.4771	0.1474	6.8939	2.0170	0.1122
9	0.6043	0.0839	4.4436	1.1636	0.0588	9	1.3790	0.1957	7.0717	1.8737	0.1592
10	0.5925	0.0825	4.1238	1.0366	0.0487	10	1.3913	0.1752	6.8102	1.6496	0.1334
11	0.6391	0.0855	4.4342	1.1429	0.0518	11	1.4432	0.1944	7.4114	1.8093	0.1425

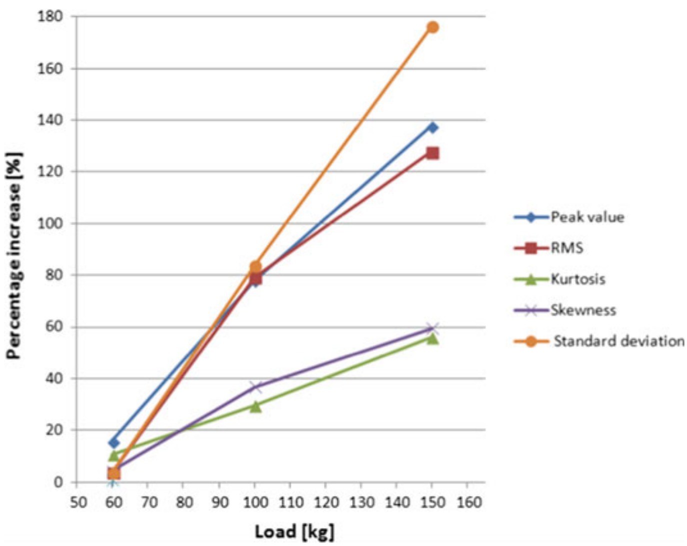

Fig. 5 The percentage increase of loads

5 Conclusion

Trustworthy and accurate measuring methods and devices are inevitable for rotary and bearing condition monitoring. The investigation of vibration signals is a significant technique for monitoring the condition of machine components. Stochastic parameters are widely used as features in failure diagnostics. Present paper shows that the time domain techniques can be effectively used in condition monitoring and fault diagnosis of ball bearings. These methods are reliable tools and they make possible fast data processing.

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Truck Floor Design for Minimum Mass and Cost Using Different Materials

Károly Jármai and József Farkas

Abstract In the chapter the floor structure of a truck produced by a company in Hungary has been investigated. The structure consists of steel members, or extruded Al-alloy longitudinal and cross members as well as a tread deck plate. Using an optimum design process, namely the Hillclimb optimizer, significant mass and cost savings may be achieved by decreasing the deck plate thickness and changing the profile, dimensions and number of cross members. Comparison is made using the combination of the steel and aluminium, or using only steel alone. Design constraints relate to fatigue stress range of welded joints, to local buckling of extruded or normal profiles and to fabrication size limitations. A special loading case is also considered when a wheel is staying on a curb and the floor is distorted.

1 Introduction

There are some trucks for beverage transport, where the truck structure has a steel chassis consisting of two longitudinal beams. The subframe is constructed from two longitudinal beams bolted on steel beams. They can be made from *Al*-alloys, or structural steel. The Al-alloy floor structure has three layers as follows (Fig. 1): cross members welded to subframe, the longitudinal members welded to cross members, tread deck plate distributing the pallet loads. The material of cross members is an *Al*-alloy AlMgSi0.7 according to German standard DIN 1725 [1] of $R_{p.0.2}$ = 215 MPa according to DIN 1748 [2] (international alloy type 6005A). The tread deck plate material is an *Al*-alloy AlMg2.5 (international alloy type 5052). These main structural parts are framed by side rails, which carry the loads from

K. Jármai (✉) · J. Farkas (Deceased)
University of Miskolc, Miskolc, Hungary
e-mail: jarmai@uni-miskolc.hu

J. Farkas
e-mail: altfar@uni-miskolc.hu

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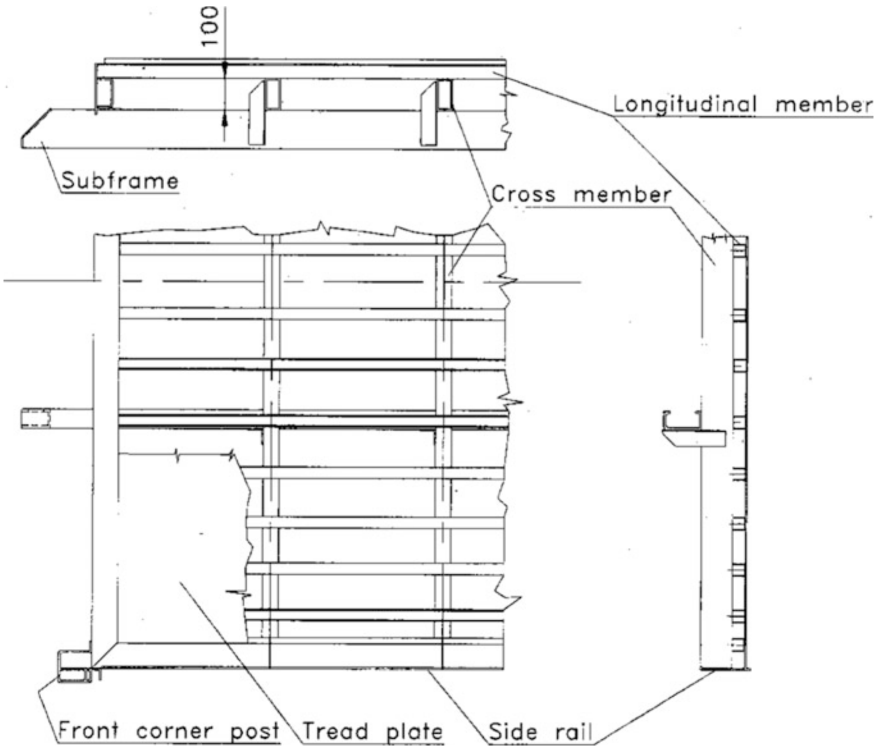


Fig. 1 Truck floor structure

roof, sidewalls and doors. We have made an optimization using aluminium, or normal steel in the floor structure. Due to the fact that the fatigue limit for the steel at Eurocode 3 up to 690 MPa and at IIW recommendation up to 960 MPa does not change, it does not worth to use higher strength steels, only normal structural steel.

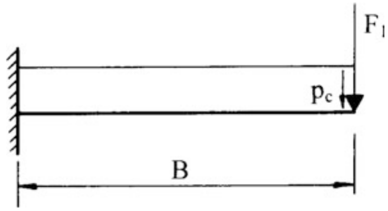
Our aim is to decrease the material cost of the floor structure by changing the profile, dimensions and number of cross members, the thickness of deck plate as well as the material grades.

2 Load Cases

2.1 Loads in the Horizontal Floor Position

Two load cases should be considered in the design of cross members as follows: (a) loads due to pallets, roof, door and side walls in the horizontal floor position; (b) the same loading as in (a) but a wheel is staying on a curb, thus, the floor is distorted.

Fig. 2 Loads on the cantilever part of cross members



Loads acting on an outside cross member are as follows:

a corner column		205 N
roof	2060/4	515 N
upper door	1420/2	710 N
front wall	1033/2	516 N
		F_1 = 1946 N

Load from pallets: mass of a pallet is F_p = 8500 N, intensity of the uniformly distributed load is $p = F_p n_p/(BL)$, where the number of pallets placed on the half floor n_p = 5, B and L are the dimensions of a half cantilever floor surface. The uniformly distributed normal load acting on a cross member is $p_c = pL/(n_c{-}1)$, n_c is the number of cross members.

The maximum bending moment in a cross member is (Fig. 2)

$$M_{\max} = \frac{p_c B^2}{2} + F_1 B = \frac{F_p n_p B}{2(n_c - 1)} + F_1 B \tag{1}$$

Calculating with F_p = 8500 N, n_p = 5, B = 720 mm, F_1 = 1946 N one obtains bending moments for different numbers of cross members. This number is limited by the dimension of pallets (800 mm) to $n_{c.min}$ = 10. Since the original number of cross members is 14, we calculate with n_c = 14, 12 and 10. For these values of n_c one obtains

$$M_{14} = 2.578, M_{12} = 2.792 \,\text{and}\, M_{10} = 3.1011\,\text{kNm}.$$

The corresponding shear forces are as follows:

$$Q = F_p n_p/(n_c - 1) + F_1;\; Q_{14} = 5215,\; Q_{12} = 5810 \text{ and } Q_{10} = 6668\,\text{N}.$$

2.2 Loads on the Distorted Floor

Measurements have been carried out on a truck loaded with pallets and with a wheel staying on a curb in a height of 91 mm. The measured deflections have

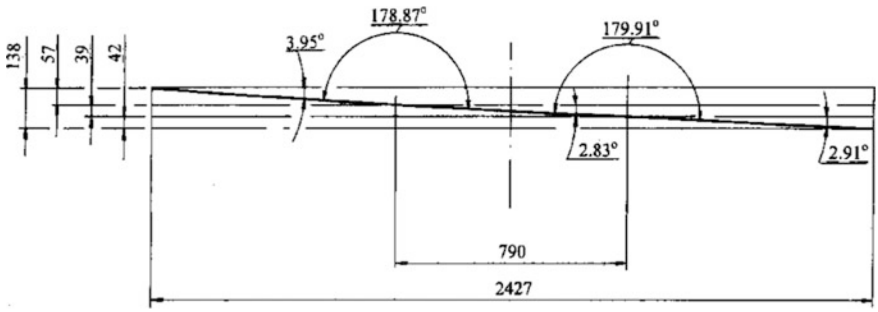


Fig. 3 Measured deflections of a distorted cross member, when a left truck wheel is staying on a curb

shown that the cross members near the wheel being lifted up are loaded by bending as it is seen on Fig. 3. This cross member can be modelled as a cantilever beam of its whole length L_c loaded by a force F corresponding to a deflection w. This deflection can be approximately calculated as $w = 138 - L_c\varphi$, where L_c = 2427 mm, $\varphi(rad) = 2.91°\pi/180° = 0.0508$, thus, w = 15 mm. Furthermore

$$F = \frac{3EI_x w}{L_c^3}; M_{c.\max} = FL_c \tag{2}$$

where E = 7 x 10^4 MPa is the elastic modulus of aluminium, E = 2.1 x 10^5 MPa for steel, I_x is the second moment of area.

3 Geometric Characteristics of Cross Members

The cross-section loaded by bending and shear consists of a cross member and a part of the deck plate (Fig. 4). We calculate an effective width of the deck plate 50t, t is the thickness. In the case of a rectangular hollow section (RHS) the geometric characteristics of this cross section are as follows [3]:

$$A = A_1 + A_2;\ A_1 = 2ht_w + 2bt_f;\ A_2 = 50t^2 \tag{3}$$

$$y_G = \frac{A_1}{A}\left(\frac{h+t}{2} + c\right); \quad y_c = h + c + \frac{t}{2} - y_G \tag{4}$$

$$I_x = \frac{h^3 t_w}{6} + \frac{bt_f h^2}{2} + A_1\left(y_c - \frac{h}{2}\right)^2 + A_2 y_G^2 \tag{5}$$

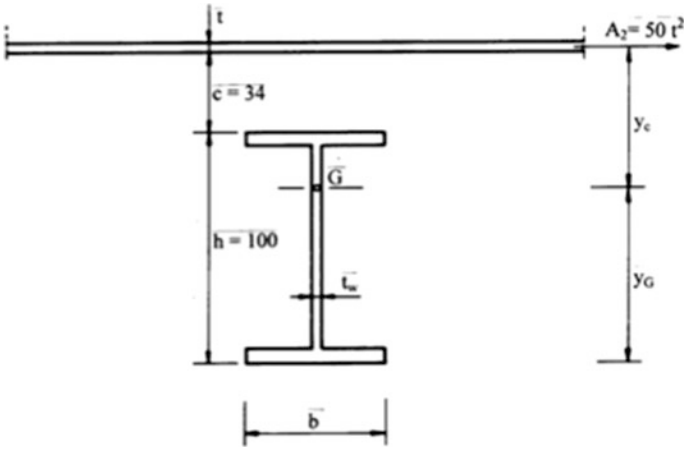


Fig. 4 Cross-sections of cross members

In the case of I-profile (Fig. 4) the characteristics are as follows:

$$A_1 = ht_w + 2bt_f \tag{6}$$

$$I_x = \frac{h^3 t_w}{12} + \frac{bt_f h^2}{2} + A_1\left(y_c - \frac{h}{2}\right)^2 + A_2 y_G^2 \tag{7}$$

In our previous calculations [4] we have made comparisons using the rectangular hollow section, I- and C-profiles. It was found that the best cross section is the I-beam. That is the reason why the I-profile has been chosen.

4 Design Constraints

4.1 Constraints on Fatigue Stress Range for the Horizontal Floor Position

$$\sigma_1 = \frac{M_{\max}}{I_x} y_{\max} \le \frac{\Delta\sigma_N}{\gamma_{Mf}}; \quad y_{\max} = \max(y_G, y_c) \tag{8}$$

$$\tau_1 = \frac{Q}{A_w} \le \frac{\Delta\tau_N}{\gamma_{Mf}}; \tag{9}$$

where $A_w = ht_w$ for I-profile.

Since the cross members are welded to longitudinal subframe beams, they should be designed considering the fatigue of welded joints. According to Hobbache [5] the fatigue stress range for number of cycles 2 x 10^6 in the case of transverse stiffener