Available online at www.sciencedirect.com

ScienceDirect

Procedia Engineering 213 (2018) 227–238

Procedia Engineering

www.elsevier.com/locate/procedia

7th International Conference on Fatigue Design, Fatigue Design 2017, 29-30 November 2017, Senlis, France

Parametric calculations of fatigue life of critical part of trolleybus rear axle

Miloslav Kepka[a]*, Miloslav Kepka Jr.[a]

[a]*RTI - Regional Technological Institute, research center of the Faculty of Mechanical Engineering, University of West Bohemia,*
Univerzitni 2732/8, 30614 Pilsen, Czech Republic †

Abstract

The paper describes the application of probability approach when assessing fatigue life of structures. The case study includes an assessment of two components of the rear axle of the trolleybus. The predicted fatigue life distribution functions were compared with service data. The calculations helped to identify and prevent fatigue cracks and fractures.

Keywords: trolleybus; weld; rod pin; radius rod; fatigue failure; fatigue life distribution funcion.

1. Introduction

The methodology, which has been successfully applied for many years in the development of Skoda trolleybuses and buses, has been presented in the literature in the past [1] and recently [2]. This methodology is now being developed in connection with the production of battery buses [3]. In cooperation with RTI (research center of University of West Bohemia, Pilsen, Czech Republic) the following R&D areas of the former company's methodology are being improved: multibody simulation, FEM calculations, vehicle tests and their parts on electrohydraulic loading device, stress measurement during vehicle running on test and real roads, computational prediction of service fatigue

---

* Corresponding author. Tel.: +420-604-831-035; fax: +420-377-638-202
E-mail address: kepkam@rti.zcu.cz

1877-7058 © 2018 The Authors. Published by Elsevier Ltd.
Peer-review under responsibility of the scientific committee of the 7th International Conference on Fatigue Design.
10.1016/j.proeng.2018.02.024

life of important construction nodes and vehicle components, crash computation and experiments. Fatigue cracks (and fractures) occurred only sporadically. If they occurred, identifying their causes and eliminating their further occurrences were the lessons for designers and specialists dealing with the fatigue life of components. The presented case study is an example of a situation that has been resolved using commercial software (nCode) and parametric fatigue life calculations.

2. Case study description

The subject of the assessment was the area of the rear axle of trolleybuses, these trolleybuses were in operation in San Francisco. The fatigue cracks and fractures occurred in two parts (areas). The first area were welds, with which the short brackets were welded to the rear axle body. With the help of these brackets, the rear suspension beams and the radius rods were attached to the rear axles. The radius rods transmit the internal forces between the bodywork and the undercarriage frame and provide vehicle stability. Fatigue fractures also occurred in the neck of the rod pins at the ends of the radius rods. These pins provide the necessary degrees of freedom for the movements of the radius rods while driving on uneven ground. There are the critical places illustrated in sketches and photographs in Fig. 1, 2, 3 and in the following text.

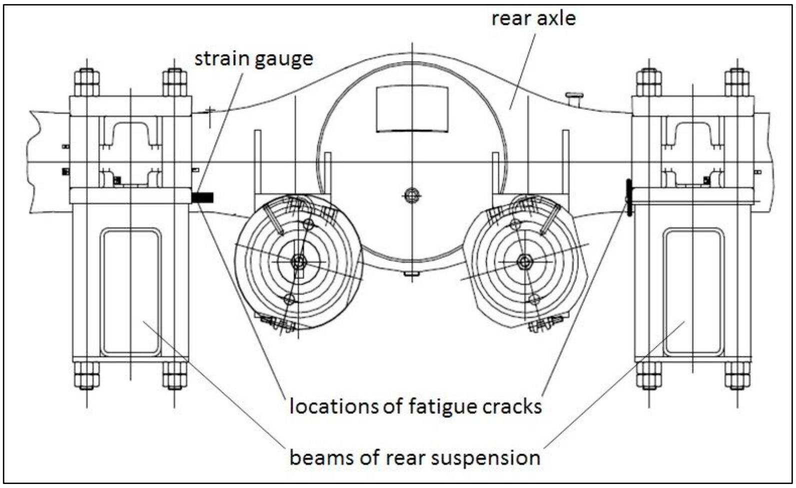


Fig. 1. Rear axle configuration.

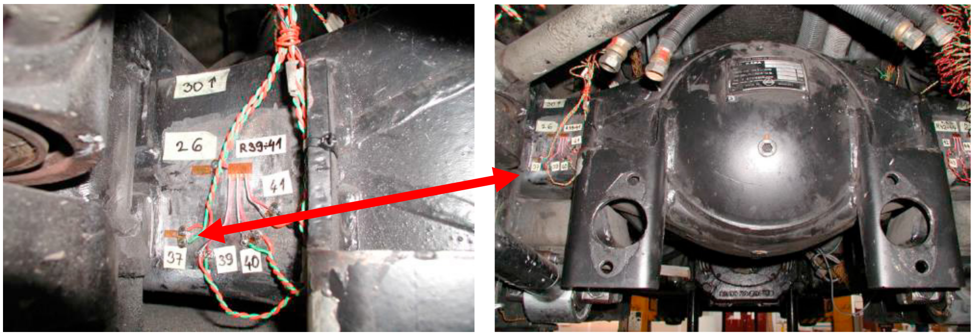


Fig. 2. Critical location of the rear axle and the strain gauge 37.

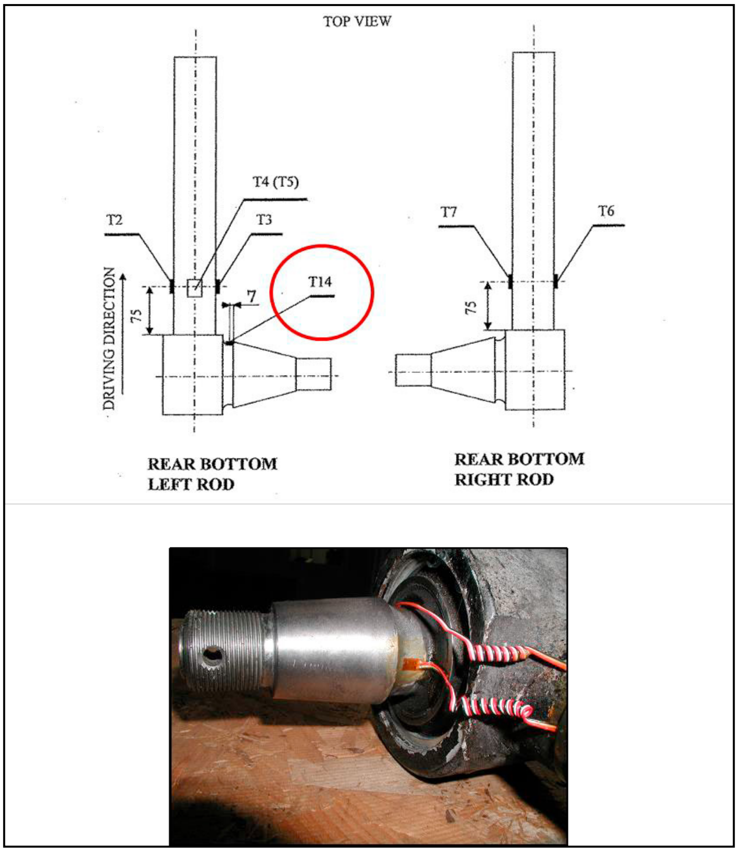


Fig. 3. Critical section of the rod pin and the strain gauge T14.

The causes of operational failures need to be identified and eliminated quickly; usually there is no time for complex supportive experiments. Fatigue cracks and fractures are always investigated at the site and in the metallographic laboratories. Sometimes the operational load measurement of the critical structural nodes or component can be performed. The fatigue characteristics of damaged structural parts or components are mostly to be estimated or deduced on existing knowledge from literature or standards. In this case, it is appropriate to take into account the scatter of the input data and the calculations of the service fatigue life to interpret by a probability approach.

A possible procedure for calculating the fatigue life distribution function (FLDF) depending on the scatter of the material properties and the random nature of the operational load was published by Kliman [4]. This method has been used several times in practice [5,6]. We have transformed the Kliman method into an engineering approach, which can be practiced with the support of commercial software, in our case with nCode software.

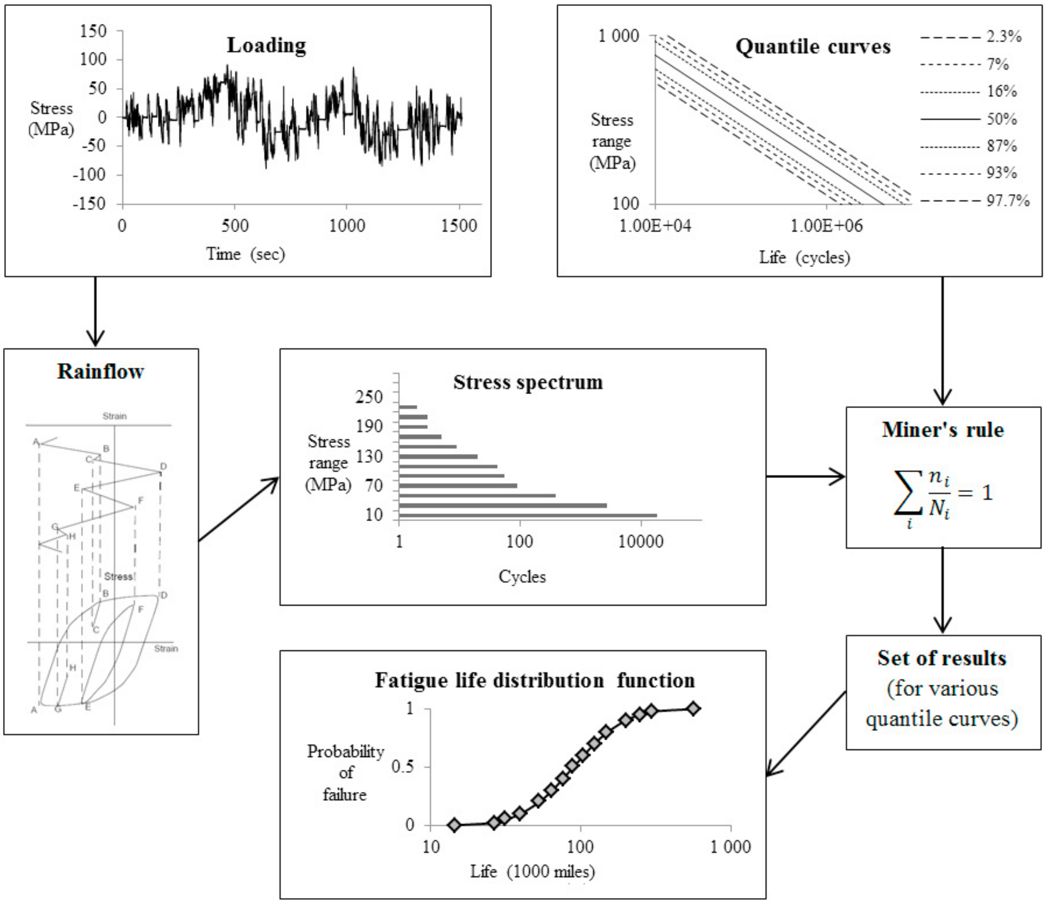


Fig.4. Calculation of FLDF (fatigue life distribution function).

A durability of a material or a component against high cycle fatigue damage is commonly characterized by an σ-N curve, which describes a relationship between stress range $\Delta\sigma$ and cycles to failure N_f. British Standard BS 7608 [7] is suitable for taking into account the scatter of material properties (in this case the fatigue characteristics of the assessed construction nodes). In this standard, the fatigue curve is expressed as follows:

$$\log(N_f) = logC_0 - s.d - m.\log(\Delta\sigma), \tag{1}$$

where
N_f is the number of cycles to fracture,
$\Delta\sigma$ stands for the stress range,
m is the inverse slope of the $\Delta\sigma$ versus log N_f,
s is the standard deviation of log N_f,
d is the number of standard deviations s below the mean fatigue life curve.

Standard deviation of log N_f can be calculated on basis of experimental data, the value of standard deviation can be also found in the literature (standards and guidelines). Fatigue curves for a various certainty of survival can be described by selecting standard deviation. For instance, at $d = 0$, equation (1) describes a mid-range fatigue curve (failure probability of 50%). Fatigue curves shifted by two standard deviations ($d = 2$) below the mean curve, provided that log-normal distribution applies, represent a failure probability of 2.3% (this means a probability of survival of 97.7%). In software nCode the certainty of survival is converted to a standard deviation using the following values. Linear interpolation is used for values not in the Table 1.

Table 1. Certainty of survival conversion to standard deviation [7].

Certainty of survival (%)	d - Number of standard deviations
99.9	-3
99.7	-2.5
97.7	-2
93	-1.5
84	-1
69	-0.5
50	0
31	0.5
16	1
7	1.5
2.3	2
0.6	2.5
0.1	3

Stress-time histories for critical locations of the investigated structural details were recorded with the trolleybus running a 16.2-miles urban route. The hilly test route with a profile typical of San Francisco had varying typical surfaces. The driving modes included braking, accelerating, turning and short stops. The measurement involved two limit load conditions: empty and fully-loaded vehicle. The measured stress-time histories were analyzed into stress spectra evaluated using the nCode software and the rain-flow counting technique. These stress spectra (histograms of cycle frequencies) are necessary input into the calculating the fatigue damage and predicting the fatigue life with the help of Miner´s rule. According to this hypothesis, the fatigue limit state is achieved when the following condition is fulfilled:

$$\sum_i \frac{n_i}{N_i} = 1, \tag{2}$$

where n_i is the number of cycles with stress range $\Delta\sigma_i$ and N_i is number of cycles to failure at the same stress range $\Delta\sigma_i$.

3. Results

3.1. Welded joint in the construction of the trolleybus rear axle

Both the rear axle housing and the welded bracket were made of St52 structural steels with a guaranteed strength of 520 MPa. The British Standard BS 7608 -1993 was used for estimation of the fatigue life curve [7]. It relates to the design and assessment of steel structures from the perspective of high-cycle fatigue. It applies to wrought structural steels with minimum yield strength up to 700 MPa. It was suitable for this case because this standard includes a class T fatigue curve. In the standard, the fatigue curve is expressed as follows:

$$\log(N_f) = 12.6606 - 0.2484d - 3\log(\Delta\sigma), \tag{3}$$

This fatigue curve was used for calculation of the fatigue life distribution function. When this curve is used, the fatigue life evaluation must be based on structural hot-spot stress. This procedure is compatible with this case because the actual fatigue cracks initiated in and emanated from the fillet weld toe, i.e. a hot-spot stress location. A schematic view of this situation is shown in Fig. 5. No. 37 strain gauge was placed perpendicularly to the fillet weld from whose toe fatigue cracks emanated in the damaged vehicles, as shown in Fig. 1 and Fig. 2. It was used for measuring the reference stress near and perpendicular to the fatigue crack propagation direction. There are other more comprehensive (multiaxial) approaches to evaluating similar scenarios [8], but the method chosen in this case was a justified and rational engineering technique.

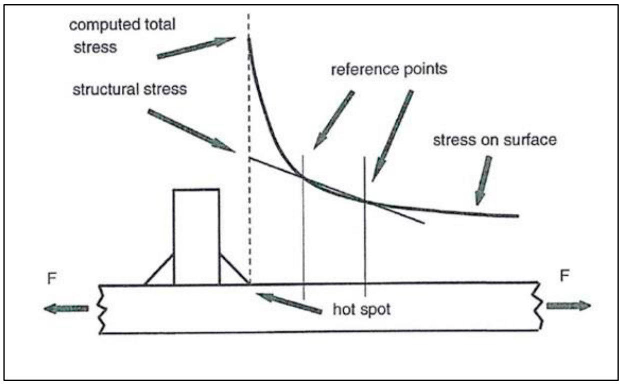
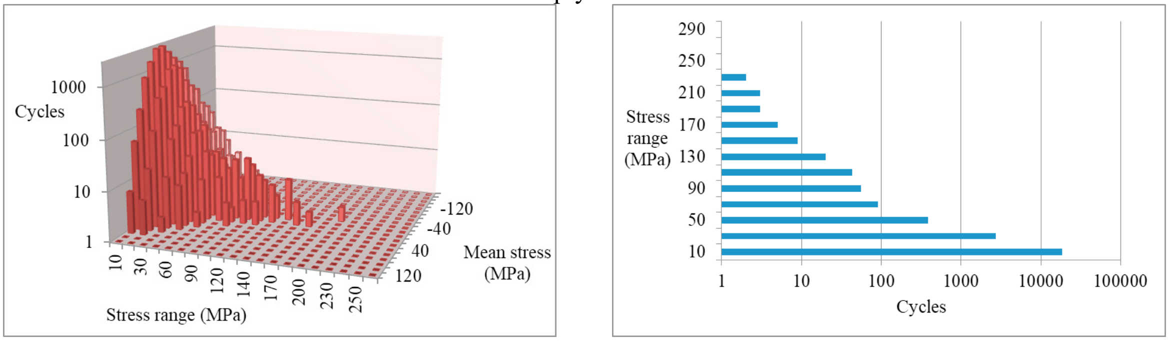


Fig. 5. Definition of structural hot-spot stress [9].

Theoretically, nominal, local and hot-spot stresses can be identified in welded structural details (Fig. 5). This case, unfortunately, does not involve any of these exactly. Nevertheless, an appropriate strategy for fatigue assessment had to be chosen. As the fatigue curve had been estimated for the hot-spot stress, the same kind of estimate has been used. The measured reference stresses were multiplied by 1.25 and the calculated values were then treated as the hot-spot stress. Fig. 6 shows one and two-parameter stress spectra evaluated using the nCode software and the rain-flow counting technique. These stress spectra (histograms of cycle frequencies) are necessary for calculating the fatigue damage and predicting the fatigue life.

Empty vehicle



Loaded vehicle

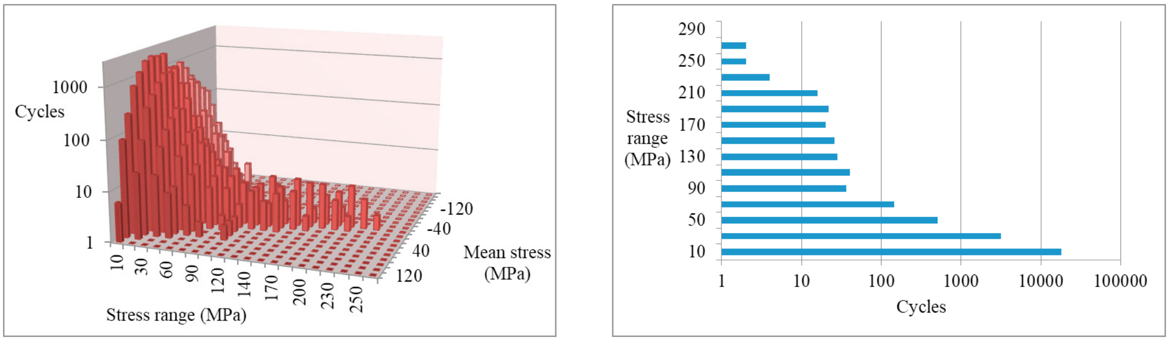


Fig. 6. Two-parameter and one-parameter stress spectra evaluated for critical welded joint.

The calculation considered the stress spectra analyzed for the estimated hot-spot stresses and estimated bi-linear fatigue life curve. The main feature of a stress cycle that affects the fatigue damage of welded structural details is the stress range. According to the recommendation in the mentioned standard, the influence of the mean stress of each cycle was not considered.

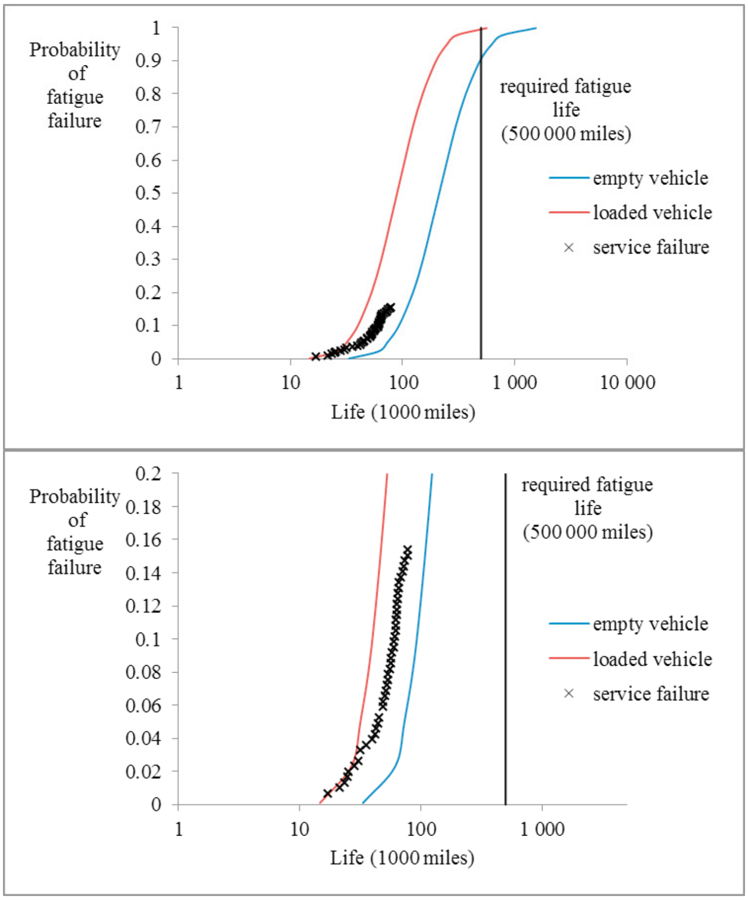


Fig. 7. Welded joint - predicted FLDF and service data.

Fig. 7 shows the calculated FLDF for both investigated service conditions: run with an empty vehicle and run with a loaded vehicle. Based on the figure, the fatigue life prediction was in a good agreement with the real-life findings. 273 vehicles had a total of 306 drive (rear) axles (240 single-body vehicles with a single drive axle and 33 articulated vehicles with two drive axles). Each axle contained two of these identical critical structural details (on the left and right side). Until the problem was solved, non-destructive inspections identified 47 fatigue cracks in 2 × 306 = 612 critical details under inspection. Fatigue cracks were found upon as low mileage as 17021 miles. The maximum mileage after which cracks were detected was 74782 miles.

nCode offers a useful tool for designers and computation engineers in the form of the back calculation to target life. Using this tool, so-called scale factor can be calculated for a particular fatigue curve. The scale factor is the value by which the known stress value must be multiplied to obtain the desired life which, in this case, is represented by 500 thousand miles.

When designing a structure, a “reasonable” level of reliability must be considered. In this case, the scale factor was determined on the basis of the design fatigue curve derived as shown in section 2 for the probability of survival of 97.7%.

The scale factor was calculated for both service conditions. If the desired total mileage of 500 thousand miles is to be achieved, the calculation shows that the scale factor must be 0.53 for a fully-loaded vehicle and 0.40 for an empty vehicle. Preliminary back calculations of fatigue life suggest that for reliable operation of the welded structural detail in question, the stress should be reduced to one half of the original value.

Since the quality of weld joint was found to be good, the result was mainly due to the high overload of the structure detail.

3.2. Radius rod pin

Radius rods ensure the appropriate geometry of axles with respect to the vehicle underframe. Together with other parts of the axle suspension system (shock absorbers and air spring bellows), they ensure noiseless transmission of force from the axle to the vehicle under any dynamic conditions during the ride (starting, braking, running over irregular surface and others). The radius rod is a weldment consisting of a tube with a head on each end. Metal-rubber bushings are press-fitted into holes in the heads. The configurations of the metal-rubber bushings may vary. The typical one involves overhung conical pins. Vehicles in service suffered fatigue failures of these pins. The fracture location was in the narrow region adjacent to the radius transition to the conical part.

Fig. 8 shows a photograph of a failed pin. After driving 70,000 miles a fatigue damage (initiation of fatigue cracks or complete fatigue fracture) occurred at approximately 50 of the total 612 pieces of installed and operated pins.

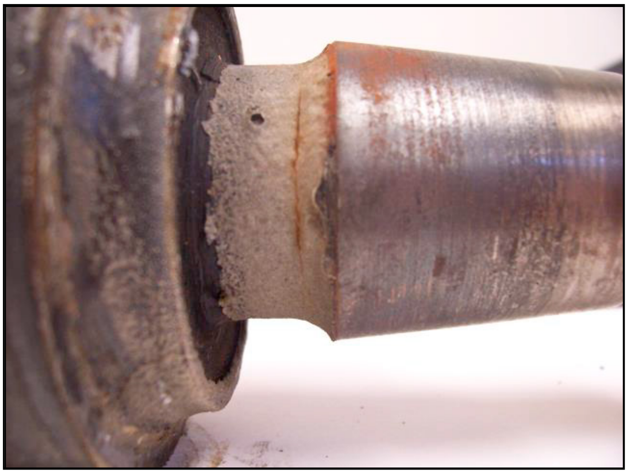


Fig. 8. Fatigue fracture of a radius rod pin.

Loading of both radius rods and radius rod pins were measured in real service condition (described in part 2). Fig. 3 shows installation of all strain gauges.

The critical strain gauge T14 was installed on the critical structural notch of the pin and therefore local deformations and stresses were measured directly. Fig. 9 shows measured stress-time histories. Fig. 10 shows stress spectra, which were evaluated with nCode software using the rain-flow method. The spectra are necessary for fatigue life calculations and predicting the FLDF.

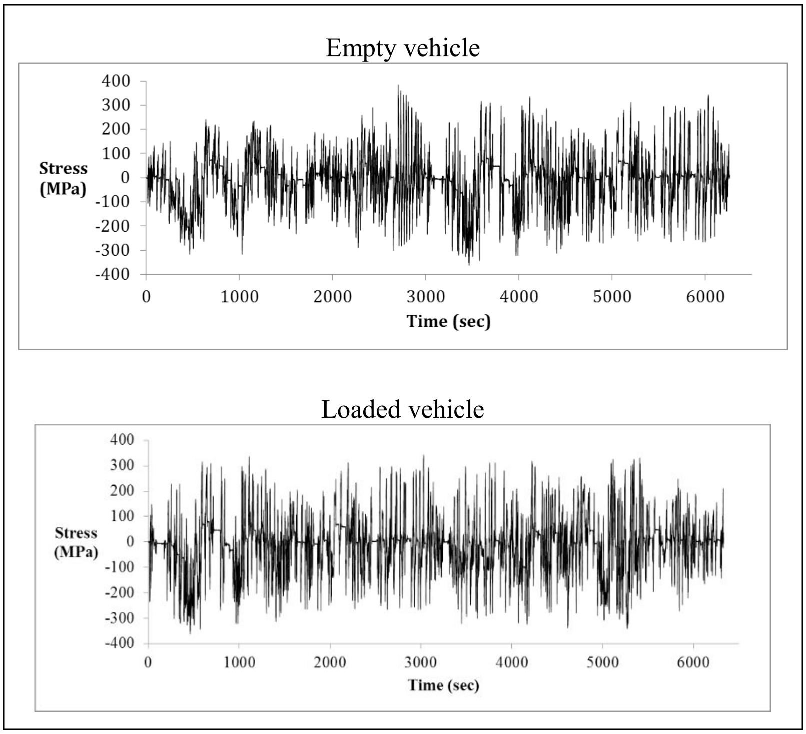


Fig. 9. Measured stress-time histories – rod pin - strain gauge T14.

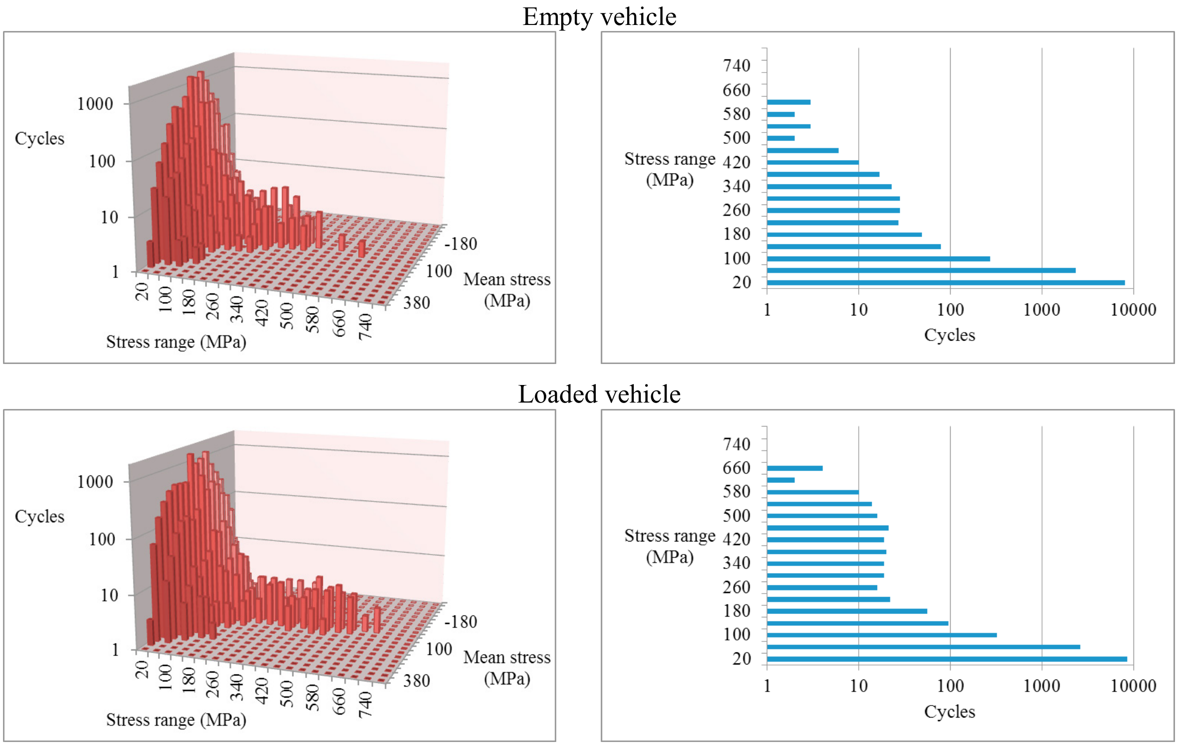


Fig. 10. Two-parameter and one-parameter stress spectra evaluated for critical cross section of rod pin.

The pins were made of various heat treated steels. Ultimate strength of these steels ranged between 700 – 900 MPa. Any relevant information about fatigue behavior was not found in literature despite of a thorough search and therefore the fatigue curve had to be estimated. The software nCode included material curves estimated according to Uniform Material Law [10]. The equation of a curve for ultimate tensile strength equal to 800 MPa is following:

$$\log(N_f) = 26.5863 - 0.1d - 7{,}47\log(\Delta\sigma). \tag{4}$$

Local stress approach was used for the calculation. The calculation considered measured local notch stresses and estimated material fatigue life curve. The main feature of a cycle that affects fatigue damage is the stress range. However, it is also influenced by the mean stress of each cycle. The Goodman mean stress correction was used from this reason.

The software nCode includes Material Modifications (Surface Finish and Surface Treatment). Two parametric combinations were used for the fatigue calculations. Fig. 11 indicates results for a good machined surface finish and non surface treatment. Fig. 12 indicates results for a good machined surface finish and a nitrided surface treatment. In addition both figures show also the real service FLDF. According to the results shown in Fig. 12 the service life of the rod would be probably sufficient. Conversely, evaluation in Fig. 11 shows that the component does not meet required service life without proper heat treatment. Deficiency in the quality of heat treatment was subsequently proved.

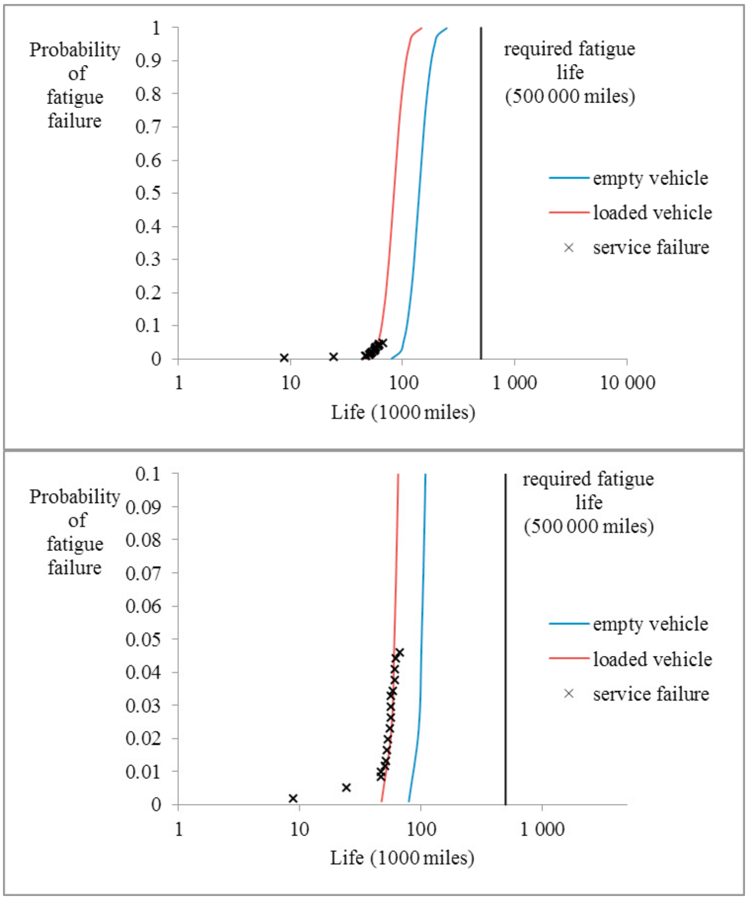


Fig. 11. Radius rod pin (good machined + non surface treatment) - predicted FLDF and service data.

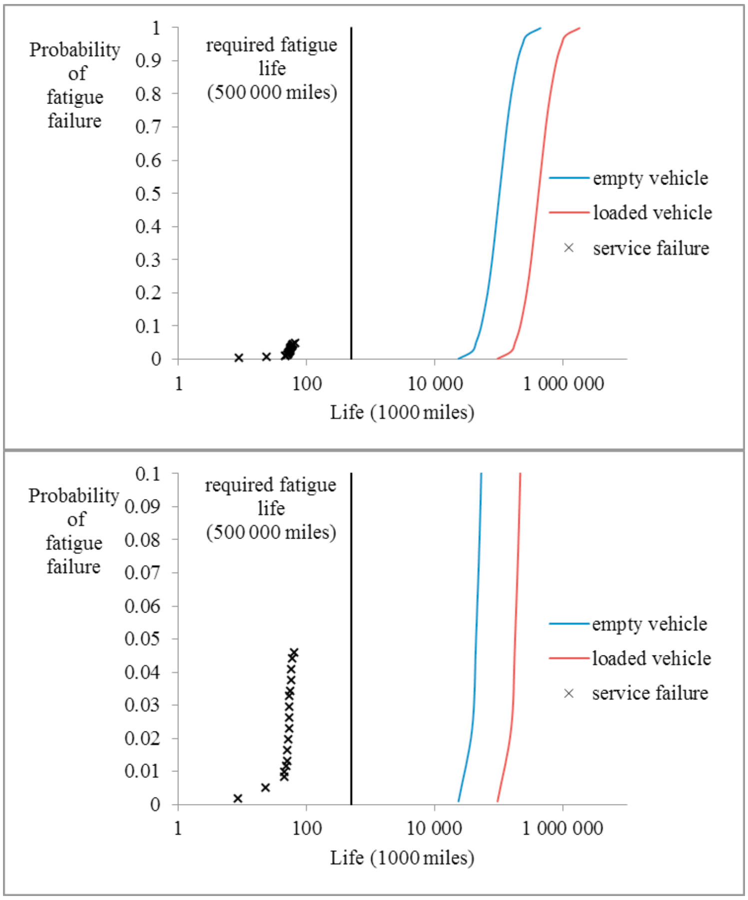


Fig. 12. Radius rod pin (good machined + nitrided) - predicted FLDF and service data.

4. Conclusions

The paper presents the potential of engineering work with nCode commercial software and fatigue life calculations in a probabilistic design concept. On the basis of the operational assessments of two critical parts of trolleybus rear axle where fatigue failures occurred the following conclusions were formulated.

a) The assessed weld joint was overloaded;
b) The assessed radius rod pin was not manufactured in required quality.

The occurrence of fatigue failures was completely eliminated. By a small change in the kinematic arrangement, the internal forces in the radius rods were reduced. In addition, the stress of the weld joint decreased significantly. Radius rod pins were replaced, the quality of surface finish and heat treatment has been carefully checked.

Acknowledgements

The paper has been prepared under project LO1502 Development of the Regional Technological Institute under the auspices of the National Sustainability Programme I of the Ministry of Education of the Czech Republic aimed to support research, experimental development and innovation.

References

[1] Kepka, M., Rehor, P., Methodology of Experimental Research into Operating Strength and Fatigue Life of Bus and Trolleybus Bodywork. International Journal of Vehicle Design, 1992, 13 (3): pp. 242-250.

[2] Kepka, M., Kepka,M. Jr., Barton, L., Remarks on Testing and Calculations of Fatigue Life of Bus And Trolleybus Structures, 16th International Conference on New Trends in Fatigue and Fracture (NT2F16), May 24-27, 2016, Dubrovnik, Croatia.

[3] Kepka, M., Spirk, S., Tests and computer simulations of electric buses. Proceedings M2D2015 - 6th International Conference on Mechanics and Materials in Design, 26-30 July 2015, Porta Delgada/Azores/Portugal.

[4] Kliman, V., Kepka, M., Václavík, J., Influence of scatter of cyclic properties of material on operational endurance of construction, Metallic Materials, 48 (2010), pp. 367–378, DOI: 10.4149/km 2010 6 367.

[5] Václavík, J., Marek, P., Chvojan, J., Kepka, M., Determination of the Fatigue Strength of a Railway Vehicle Node using the Probability Approach, Engineering Mechanics, 17 (2010), pp. 99-11, ISSN 1802-1484

[6] Kepka, M., Kepka Jr., M., Chvojan, J., Vaclavik, J., Structure service life assessment under combined loading using probability approach. Frattura ed Integrità Strutturale, 38 (2016), pp. 82-91

[7] BS 7608:1993. Fatigue design and assessment of steel structures.

[8] Margetin, M., Durka, R., Chmelko, V., Multiaxial fatigue criterion based on parameters from torsion and axial S-N curve, Frattura ed Integrita Strutturale, 37 (2016), pp. 146-152

[9] Hobbacher, A.F., Recommendations for Fatigue design of welded joints and components, Second Edition IIW225915 (ex XIII246013/ XV144013), Publisher: Springer ISBN: 978-3-319-23756 5

[10] Bäumel, A., Seeger, T., Materials data for cyclic loading, Supplement 1, Amsterdam; New York : Elsevier, 1990. ISBN: 0444886036