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FATIGUE SERVICE LIFE TEST OF THE TRIPPLE HYBRID HYDROGEN FUEL CELL BUS STRUCTURE

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ABSTRACT

This work describes the performance and evaluation of service life tests of a hydrogen cell bus structure. This is performed using short tests on the model road and long road tests with the help of an on-line monitoring system. Used methodology for the evaluation of the service life based on the probability approach, the description of the monitoring system and some obtained results are given in the contributions.

Keywords*:* fatigue, service life, bus, hydrogen cell, monitoring system, probability approach

INTRODUCTION

The TriHyBus project, coordinated by Ústav jaderného výzkumu Řež a.s. (the Czech Republic) in several past years, comprised research and development, implementation and a test operation of a 12-meter city bus (Fig. 1) with a hybrid electric propulsion using hydrogen fuel cells (Fig. 2). The bus was manufactured by ŠKODA ELECTRIC a.s. (Czech Republic) using the chassis of the Irisbus Citelis 12M (produced by Iveco Czech Republic, a.s.) and operates in the city of Neratovice (Fig. 12), where the first Czech Hydrogen filling station was built in the Veolia Transport bus park. The 48-kW Proton Motor Membrane fuel cell is used as the main power-source for its 120 kW electric traction motor. Additional 28 kWh traction accumulators and ultra capacitors are engaged while the bus accelerates or ascends, working alongside the fuel cell, allowing for energy recuperation while decelerating. The bus uses a hybrid design that increases efficiency of its propulsion system. The distribution and total amount of bus masses is slightly different compared with standard busses. That is the way the verification of the strength of the bus skeleton as well as the stability of the bus is under investigation.

Fig. 1 Front view of the TriHyBus

Fig. 2 PEM fuel cell in the back of the TriHyBus

COMPUTATIONAL SUPPORT OF THE TESTS

Body structures of busses are complicated mechanical systems subjected to the long-term operational stresses of a random nature. A combination of computational methods using MBS models, FEM and experimental procedures was used here for the fatigue life estimation of the TriHyBus structure.

In order to obtain a tool for dynamic analysis several types of multibody models of an empty (14 tons weight) and a fully loaded (18 tons weight) hydrogen bus were created (Polach and Hajžman, 2011). This model is formed by 21 rigid bodies coupled by 24 kinematic joints. The number of degrees of freedom in kinematic joints is 39 (Fig. 3). The TriHyBus multibody models have been created using the **alaska** simulation tool and on the basis of analytical derivation in the MATLAB system. The improved force-velocity characteristics of shock absorbers were derived by MBS model optimization on the base of comparison of time histories of the relative deflections of the air springs (Fig. 4) during the bus test on the model road (Fig. 6).

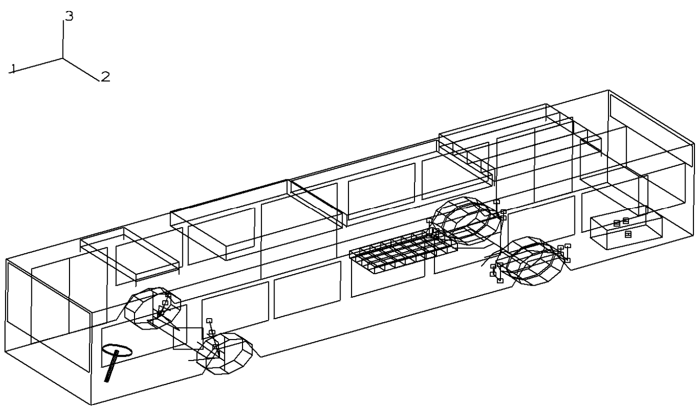

Fig. 3 Visualization of the TriHyBus multibody model in the **alaska 2.3** simulation tool

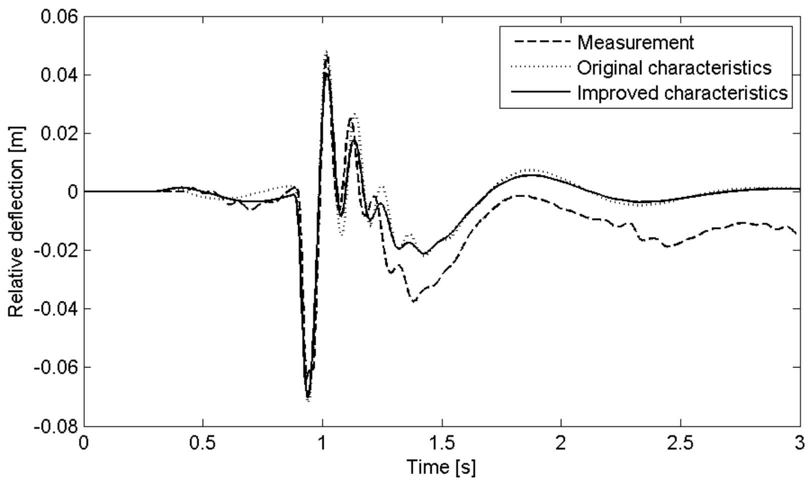

Fig. 4 Time histories of the relative deflections of the rear air springs determined at simulations with the multibody model and the same obtained at experimental measurement with real bus

Results of the MBS model were used for the calculation of the skeleton stresses using FEA. The FEA analysis was performed on several critical joints of the structure, coming out from recommendation of Regulation (EC) No 79/2009 of the European Parliament and of the Council of 14 January 2009 on type-approval of hydrogen-powered motor vehicles. The focus of FEA calculations was put especially on structure parts, carrying the hydrogen propulsion components. An example of the loaded FEA model of the holding frame of hydrogen containers is presented in Fig. 5.

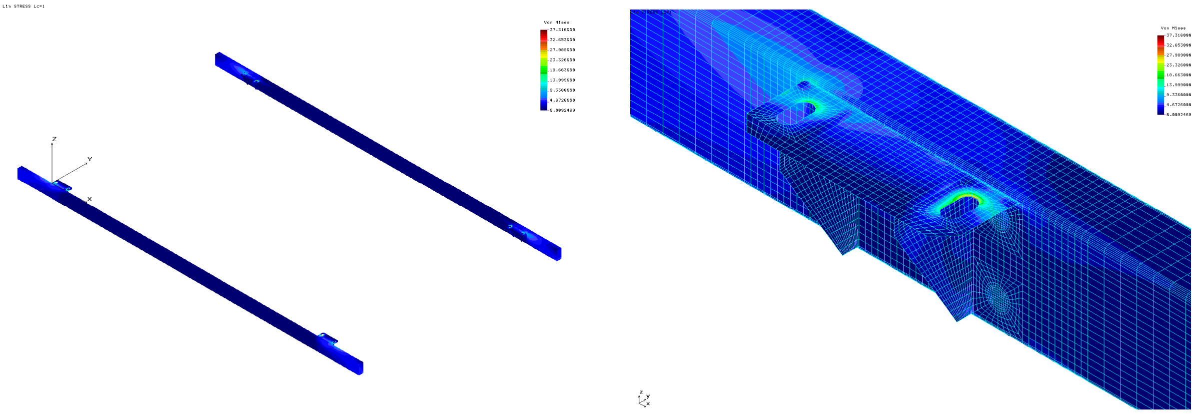

Fig. 5 FEA model of the frame for supporting the hydrogen containers

USED METHODOLOGY FOR SERVICE LIFE CALCULATION

The test on the model road (also called bump tests) is normally carried out in a standard way according to the ŠKODA RESEARCH road vehicles testing methodology. An artificial test track is created on a common bitumen road with a set of four portable standard bumps (Fig. 6), the shape of which is defined in the ČSN 30 0560 Czech Standard (Obstacle II: h = 60 mm, d = 500 mm – see Fig. 7).

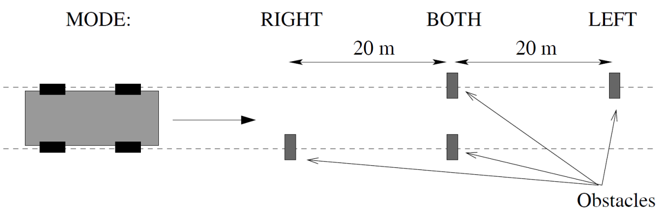

Fig. 6 Scheme of the model road test

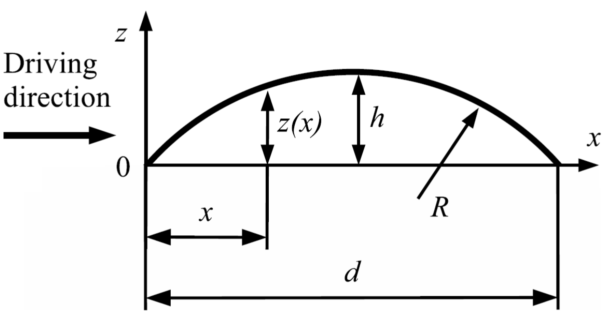

Fig. 7 The standardized artificial obstacle

Artificial obstacles are normally spaced out on the smooth road surface 20 meters apart. The first obstacle is run over only with right wheels, the second one with both and the third one only with left wheels at bus speed about 40 km·h^{-1}.

The maximum joint stress amplitude obtained during the bump test (Fig. 8) was compared with fatigue limit of the joint to check the condition of unlimited life. The service life calculation for joints, which did not fulfil the above mentioned condition, was performed using operation stress time histories. This was based both on the design spectra, derived from short test on the model road and real road spectra, obtained from bus standard operation with passengers at lines near Neratovice, Czech Republic.

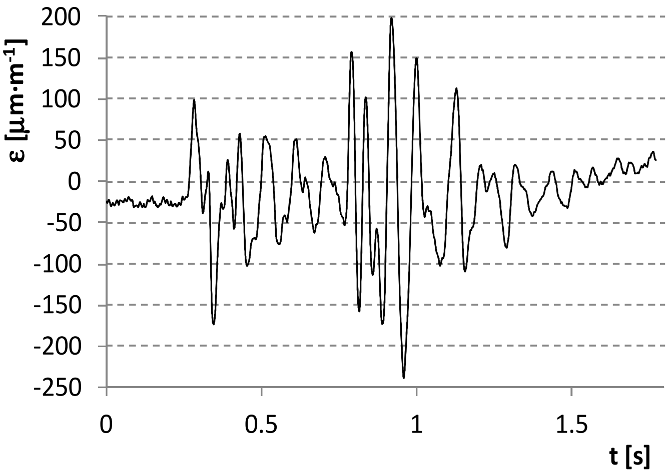

Fig. 8 Example of strain gauge strain history during bump test

The loading design spectra can be generated on the base of following equation (Kepka and Kotas, 1996):

$$h_i = N_{tot} \cdot \left(\frac{N_{\max}}{N_{tot}} \right)^{\left(\frac{\sigma_{a,i}}{\sigma_{a,m}} \right)^s} \tag{1}$$

$$h_i = \int_{\sigma_{a,i}}^{\sigma_{a,m}} H_i \cdot d\sigma_{a,i} \tag{2}$$

$\sigma_{a,i}$	stress amplitude,
$\sigma_{a,m}$	maximum stress amplitude in spectrum during bump test,
N_{max}	number of cycles with amplitude $\sigma_{a,m}$,
N_{tot}	total cycles number in spectrum,
s	parameter of spectrum shape,
h_i	cumulative rate of cycles with amplitude σ_{ai},
H_i	counting rate of cycles with amplitude σ_{ai}.

It is presumed, that the design spectrum represents a histogram of symmetrical loading cycles with mean value of σ_m = 0.

Distribution of cumulative counting rates can be plotted in semi-logarithmic coordinates as shapes with various parameters *s*. Spectra with linear distribution (*s*=1) were used, which are typical for a vehicle driven over unevenness of the road. Total number of cycles N_{tot} inside the design spectrum can be estimated for vehicles according to the following equation:

$$N_{tot} = \frac{NSU}{v} \cdot 3600 \cdot f \tag{3}$$

NSU	design vehicle service life in number of traveled miles,
v	average travel velocity [$km \cdot h^{-1}$],
f	dominated frequency of oscillating stress during bump test [Hz].

Occurrence of outstanding road unevenness occurred during bum test is supposed once in 200 km during the standard bus service on the base of experiences.

The fatigue damage, caused by the stress spectra was calculated using linear hypothesis of damage accumulation. Bi-linear fatigue *S-N* curves were used. The slope of the *S-N* curves and the knee point of the curve were used according to the measured joint specification.

PROBABILITY APPROACH FOR SERVICE LIFE CALCULATION

The assessment of the fatigue life based on deterministic approach needs to use mean values and safety coefficients. However, if these coefficients are not chosen properly, the interference between loaded stresses and fatigue resistance of the joint can cause the fatigue damage or the breaking of the material. That is the way the probabilistic approach is better to use to estimate the reliability of the structure.

For the probabilistic calculation all the input variables, especially the *S-N* curve and the histogram of stresses, have to be represented in probabilistic domain.

The model of *S-N* curve with log-normal distribution of number of cycles *N* to failure was used. The variability of *N* was determined using the standard deviation s(log *N*) for fractile *d* according given probability of failure. Following relation was given for *S-N* curve:

$$\log N_i = \log C_1 - m \cdot \log \sigma_{a,i} + d \cdot s(\log N) \tag{4}$$

C_1	shift of *S-N* curve according the used detail class,
m	slope of *S-N* curve,
s(log *N*)	residual standard deviation of log *N*,
d	fractile for left-sided tolerance limit and given probability.

The two-parametric histograms of stress amplitude rate were processed from measured stress time history using rain-flow method for the sequence of subsequent constant sections of the bus route. This histogram was recalculated to the one-dimensional one according relation (5). Using the sensitivity for mean stress *M* the influence of static mean stress σ_m was taken into account by increasing the stress amplitude $\sigma_{a,\,i}$ to $\sigma_{a,\,ef,\,i}$,

$$\sigma_{a,e,i} = \sigma_{a,i} + M \cdot \sigma_m \tag{5}$$

The estimation of distribution of cycles n_i inside each stress amplitude class $\sigma_{a,\,i}$ was performed using the set of the one – dimensional matrices.

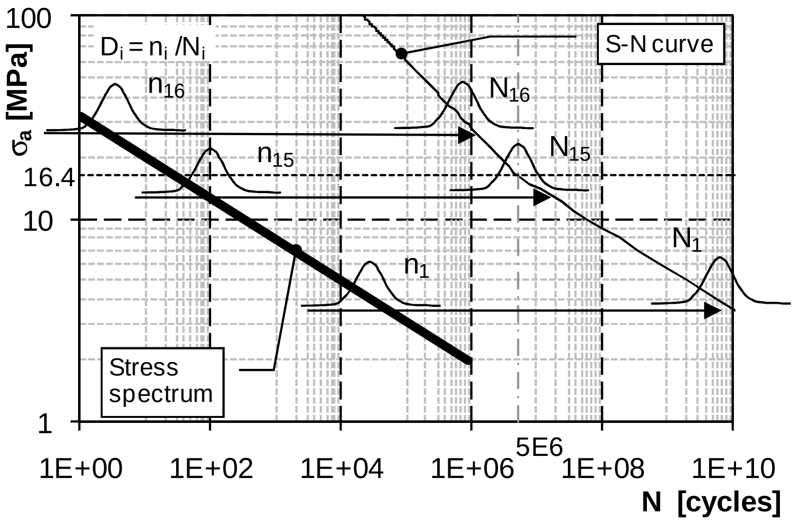

Fig. 9 The method for damage accumulation calculation

The calculation of the distribution of the fatigue life was performed using Monte-Carlo simulations (see Fig. 7).

DESCRIPTION OF THE TEST INSTRUMENTATION

A special measurement system for on-line monitoring of the bus performance has been developed and installed to the box above the bus window (Fig. 10).

Fig. 10 Used on-line monitoring system inside bus

The bus was provided with several strain gauges (see example in Fig. 11) at critical structure joints (obtained from FEA model and chosen also by experience) as well as with other sensors to be able to monitor the bus ride quality. The GPS position of the bus, the traction engine rpm and some signals from the CAN bus were monitored, as well.

The mobile computer was able on-line calculate separate rain-flow matrices for three performance cases: for each switch on and off of the bus, for each day and for given travelled distance. The splitting of the strain signal at the time domain to separate sections of constant driven distance is presented in Fig. 13.

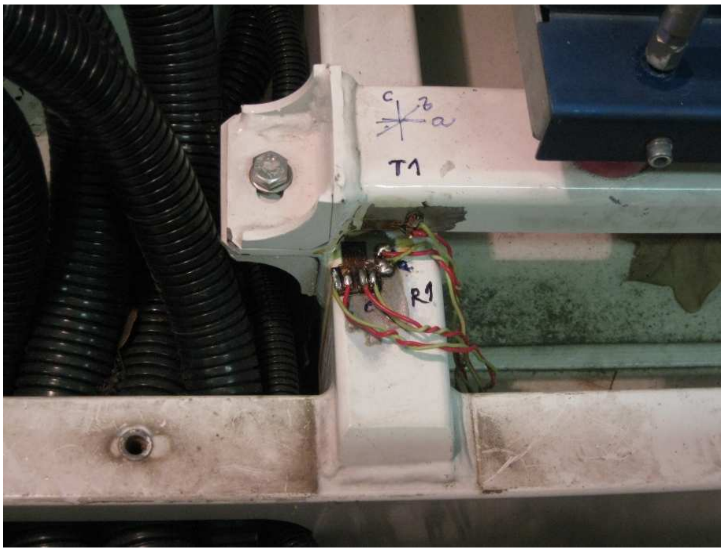

Fig. 11 An example of single strain gauge and strain gauge rosette, installed on bus roof structure

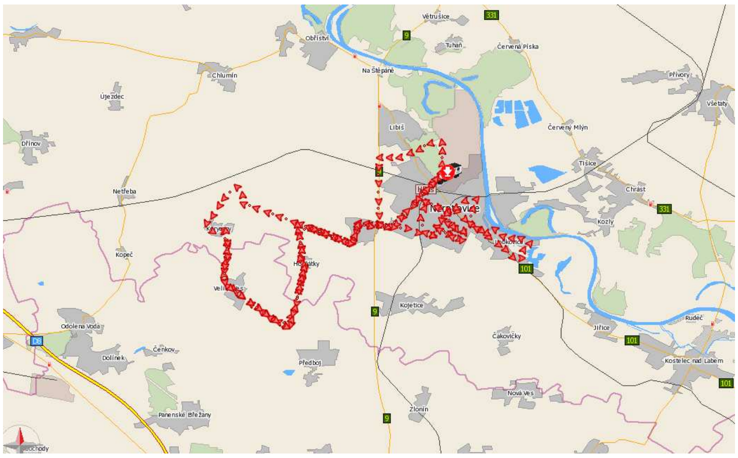

Fig. 12 The regular bus line near Neratovice, Czech Republic (several km north from Prague)

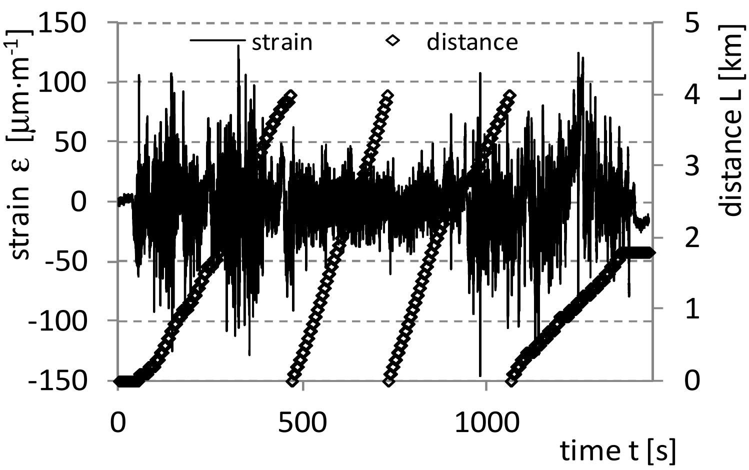

Fig. 13 Road test strain and distance history

RESULTS AND CONCLUSIONS

An example of evaluation of service life of critical structure joint at the holder of traction engine is shown in next figures. Both design and real spectra for this joint are shown in Fig. 14 together with mean *S-N* fatigue curve. The missing of high amplitudes and in general softer form of the real spectra (low travelled distance) can be seen in this figure.

The service life estimation was based on the damage calculation using *S-N* fatigue curves and both the design spectra and rain-flow matrix of road test stress amplitudes. The estimated distribution function of the probability of the failure of presented joint is given in Fig. 15. Here the mean life of the joint is estimated to be approximately 1 million km.

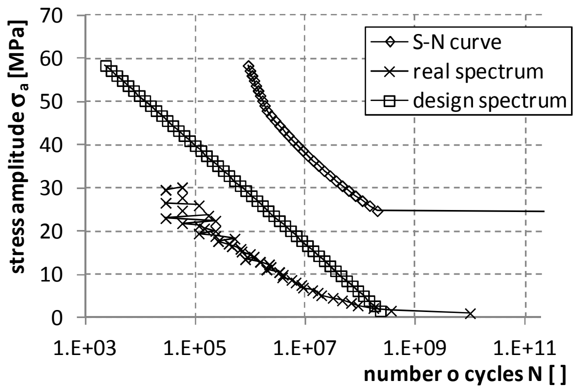

Fig. 14 Design and real spectra and fatigue curve

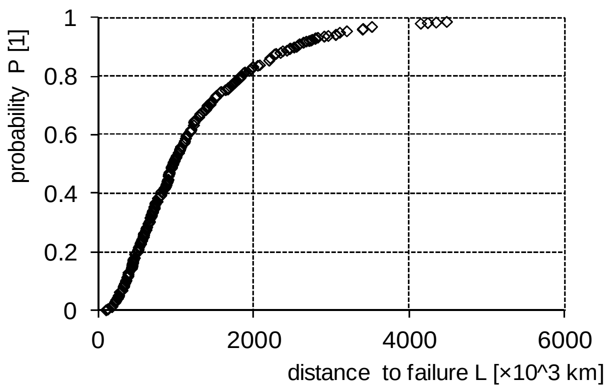

Fig. 15 Distribution function of fatigue life

The service life estimation of the bus is based on low performance hours for the present. It is expected, that long-time test will continue for several years.

ACKNOWLEDGMENTS

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