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THE BUGGY VEHICLE TRANSMISSION GEARBOX TECHNICAL STATE VIBRODIAGNOSTICS INVESTIGATION

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Abstract

Vibrations draw ahead in every mechanical object. The significant part of machines reliability is vibration level, which could be dangerous for machine technical state. Disturbance of balance state is the reason of vibration formation in machine parts. This vibration can exist and propagate even after their source expiration. Vibrations could be essential just after the crossing certain threshold marked by amplitude and the frequency of the phenomenon. They can be harmful for the object after crossing of this threshold of the vibration, or for his surroundings (e.g.: decrease of the durability of the material). In this paper were introduced of beach buggy chosen units and elements diagnostics research results with usage of vibroacoustics methods.

Keywords: *diagnostic inference, modal analysis , vehicle investigations*

1. Introduction

Vibroacoustics is one from these fields of the science which rise on the needs of diagnosing the current machine engines and devices technical condition. Using emitted vibrations, received during machine engine exploitation process as valuable information about dynamic properties of machine and other aspects we could find the relationships among them.

Every technical device in the given moment is in certain definite state. The most generally technical state of object, machine, vehicle can be describe as a set of the all parameters values that defining the given object in given moment of time t. The time sequences of this states could be consider as the time of the device existence. He leads inevitably the destructive influences of external extorting and internal factors to the machine condition change [1]. The use of technical diagnostics methods makes possible to qualification of current technical state of studied object, machine and vehicle.

The necessity of the technical state estimation is conditioned the possibility of making decisions connected with object exploitation and the procedure of next advance with object. The present development of automation and computer science in range of technical equipment and software creates new possibilities of realization of diagnosing systems and monitoring technical condition of more folded mechanical constructions. These new possibilities are connected with the new constructions of intelligent sensors, module software and the modules of transport and data exchange [2].

2. Vibroacoustics in mechanical engineering

The most valuable information about current machine engine technical state we could obtain during machine natural loads without disturbing this process. This kind of information obtainment is the basic domain of the technical diagnostics. The investigations of vibroacoustics processes in many cases are very complicated, in peculiarity when vibration processes step out in real physical arrangements. Up to now existing diagnostic procedures based on state symptoms slowly changes into diagnostics process based on machine engines models that describing their properties analysis.

Nowadays one of the most known ways to structural model creation of machine engines is modal analysis models utilization: experimental or operational modal analysis methods. The method choice depends on this what kind the input function character of the investigative object during the experiment has to have. Operational modal is the name for the technique to do modal analysis on operational data - cases where we do not excite the structure artificially but just allow the natural operating loads to excite the structure. Thanks to this during investigations we receive investigative data for real object working process in chosen measure points in relation to reference points [3]. Preparation process for diagnostics investigations in these methods contains measure and reference point's disposal and also frequency range define. The advantage of this method in use to identification of objects dynamic profiles are shore conditions and loads retain that is characteristic for these objects exploitation. Basis on measured signals on the output of object received in chosen measure and reference points for unknown natural loads of the arrangement, the estimation of modal parameters is proceed. Modal poles and natural frequencies are identified ant then the mode shapes are estimated.

This way of parameters estimation could procure some doubt – we have to take it into consideration during final analysis. The biggest problem of this method is that we do not know the value of exciting force on the arrangement. The exciting forces with random character doesn't have one point of reaction on investigated mechanical structure so received exploitation forces structural schedule is unable to identification. The more important is also fact that lots of machine engines used in industry cooperates with other technical objects, not necessarily with the same characters of dynamics loads. Best way to solve this problem will be separation from others machines the investigated machine engine. Unfortunately it refers with the machine engines working process stop so this action is unacceptable in this method usage. Disturbances triggered from next machine engines could cause the formation of additional poles on the created stabilization diagram. In modal analysis we had two ways of modal model parameters estimation: in time and frequency domain. Time domain estimation basis on information from vibrations and arrangement response in time domain signal. Estimation of modal model parameters in frequency domain basis on the input and output signal spectrum [3,4,5]. Nowadays during investigative process we often use the modal model estimation in frequency domain because there is possibility of limitation frequency range to this value in which we could recognize change of vibroacoustics signal during machine exploitation. The most valuable advantages of this method are:

- easiest possibilities of investigative data averaging which is used for noise reduction from signal,
- high precision of received results in case when exist an influence of vibration that lay behind of investigative range of vibrations,
- high precision of received results in case when is the high value of damping.

The frequency domain disadvantages are:

- the possibility of local minimum existence for signals with high noise level,
- the possibilities of troublesome mistakes connected with spectrum leak, existence of incorrect frequencies component in the signal and others.

Introduced below disadvantages and advantage of time domain modal model parameters estimation has similar sights to frequency domain, but this method is better in case, when we have to estimate:

- data with high level of noise,
- wide range of frequency during estimation.

It is also possible to use both of these methods in case, when we have measured vibrations in time domain booth from input and output source [6,7,8].

3. Transmission gearbox modal test

The investigations object was a buggy vehicle which was construct and built in the Bydgoszcz University of Technology and Life Sciences investigative laboratory. During construction phase the LMS Virtual.Lab rev 11 were used for construction optimization [9,10]. Figure no 1 present the buggy vehicle virtual model during tension and strain analysis. The final prototype of buggy vehicle was built in spring 2012 and it is shown on figure 2.

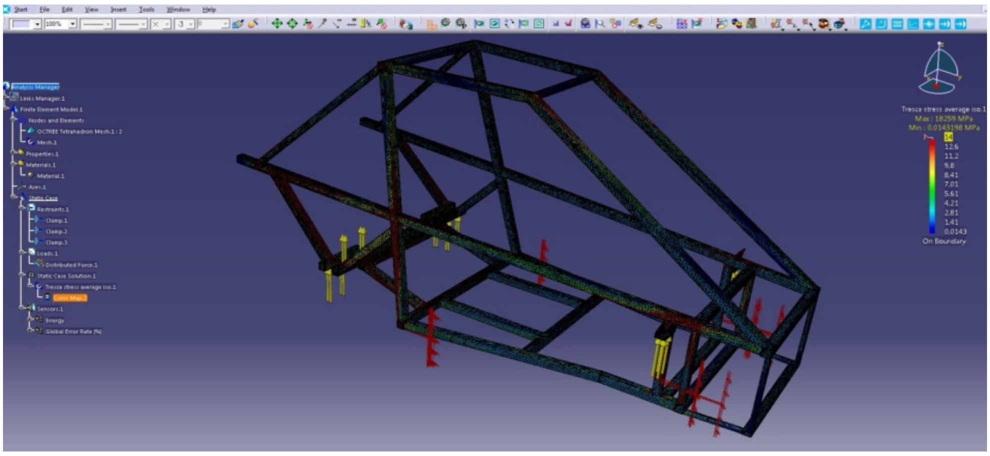


Fig. 1. Model of buggy vehicle created in LMS Virtual.Lab software during virtual tests

Fig. 2. Buggy vehicle during road tests [own source]

During exploitation of the buggy a few problems appear connected with vibration level. The transmission gearbox generated a high level of vibration and after few months the transmission mount points were damaged. To resolve these problems the modal analysis methods were used for vibration analysis. Conducted investigations of gearbox depended on delimitations of vibroacoustics measures for chosen gear sets and accomplishment the assessment of received results influence on transmission gearbox state by operational modal analysis methods. The LMS Test.Lab software with Modal Analysis Lite module was used for modal model poles estimation and analysis of mode shapes for multi degree arrangements with Time MDOF method. For mechanical object analysis several types of function could be used – time or frequency signals: autopower spectrum, crosspower spectrum, coherence and others.

Modal test could be divided into three phases. First phase of modal test is measurement set-up (system calibration, force and response transducers attachment). The second step of modal test is measurement of frequency response data – measured in time domain signal is transformed into the frequency domain functions. The last step of test is modal parameters estimation where measured frequency functions are used for modal model estimation. As a result we received the stabilization diagram with natural frequencies and damping factors, the modal participation factors and estimated mode shapes.

The dynamic signal LMS SCADAS III [11] recorder was used for gearbox signal acquisition. Measurements were realized with gearbox speed 930 min-1 on the various shifts. During measurement 90 seconds time periods of the signals were recorded with the frequency range 128 Hz. Figure no 3 presents transmission gearbox model with signal acquisition points.

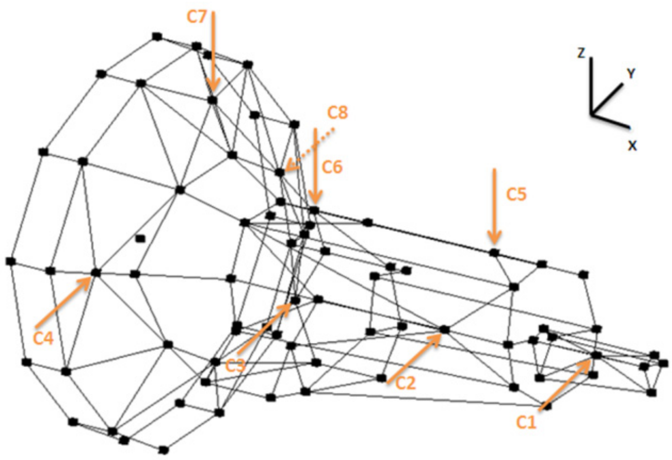


Fig. 3. Transmission gearbox model with signal acquisition points

Basis on modal analysis theory the Time MDOF module with non-linear Least Square Frequency Domain (LSFD) method and Balanced Realisation (BR) was used for modal parameters estimation [12]. LSFD is multiple degree of freedom method that applied for multiple inputs it generates global estimates for stabilisation diagram (system poles), modal participation factors and mode shapes. In first step of Time MDOF method we should define the frequency range within modal test will be done. The geometrical model creation in “Geometry” module will enable the arrangement natural frequencies visualisation. Figure no 4 present “Geometry” module during investigations.

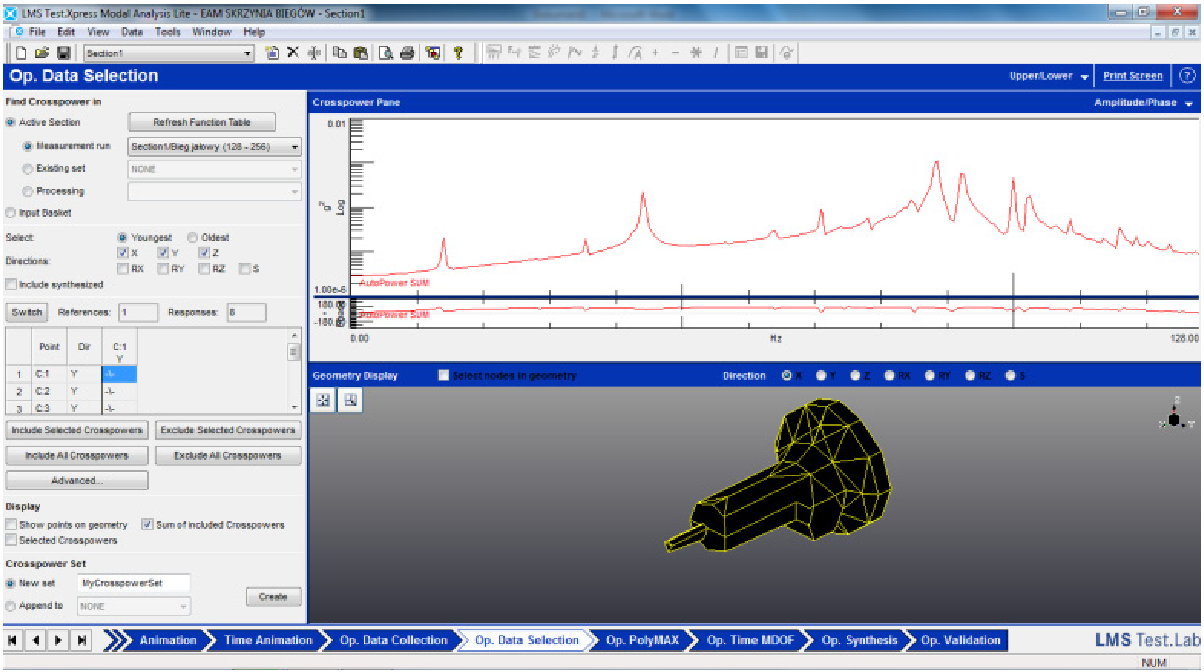


Fig.4. Data selection with "Geometry" module during investigations

In second step of Time MDOF the Balanced Realisation method was used. This is one of the "subspace" techniques which identify natural frequency, damping and mode shapes. A subset of the response functions can be selected as references. These are used in the computation of the cross power functions from the original time domain data. This method is useful in identifying the most dominant modes occurring under operational conditions [12]. Figure no 5 present's sample of Time MDOF stabilisation diagram for investigated transmission gearbox.

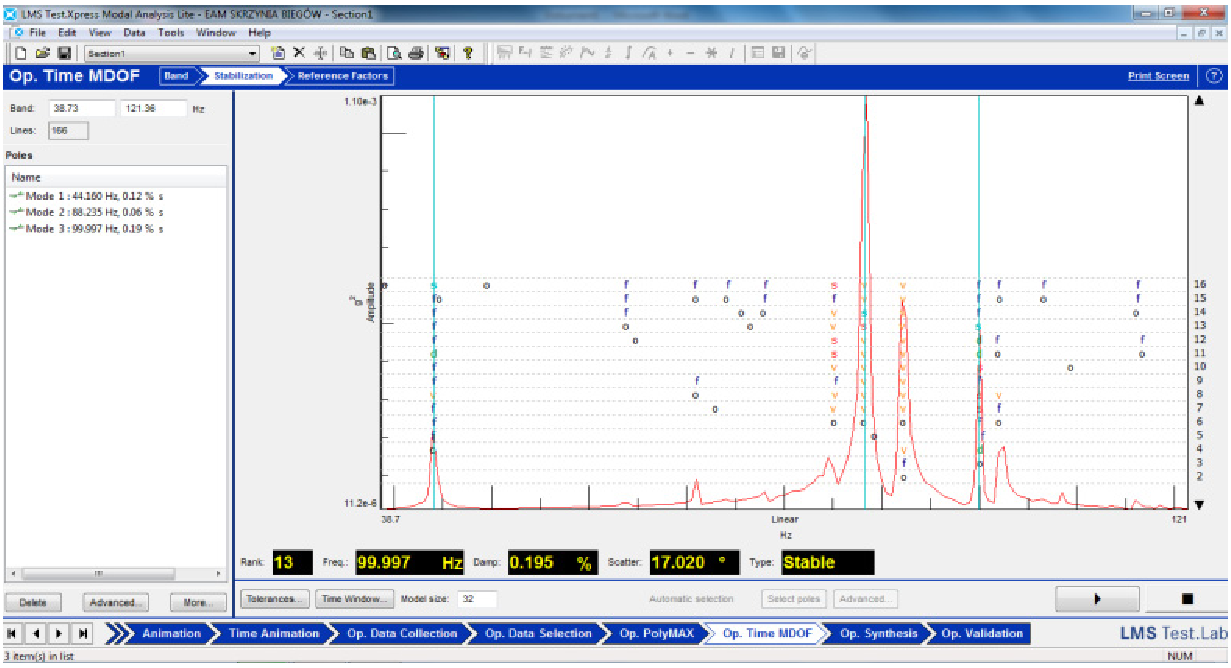


Fig. 5. Time MDOF stabilisation diagram

Analyzing individual simulated cases on the investigative transmission during investigation received set of stabilization diagrams, on which stable poles were mark. The stable pole marks parameters: frequency, modal damping and the mode shapes vector. The results of modal tests were introduced in table 1.

Table 1. Results of transmission gearbox modal tests

	Natural Frequency [Hz]	Modal Damping factor [%]	Modal model Order
Idle run	44,160	0,12	16
	88,235	0,06	14
	99,997	0,19	13
First gear	44,080	0,20	8
	70,976	0,13	14
	88,504	0,13	12
	99,986	0,03	16
Second gear	44,444	0,15	12
	88,890	0,22	10
Third gear	43,844	0,28	7
	70,433	0,25	12
	124,216	0,46	9
Fourth gear	42,627	0,17	8
	81,967	1,01	8
	87,734	3,04	10

The last step of Time MDOF modal parameters estimation were mode shapes estimation with LSFD method that sample results for idle run mode shapes were introduced in figure no 6. As modal tests results validation was used LMS Synthesis module with Auto-MAC criteria estimation - introduced in figure no 7, where we could calculate the error of estimation for all recognized mode shapes of object. As final results we obtain the dynamic state description of real technical object with estimation of predominant properties of natural frequency and mode shapes.

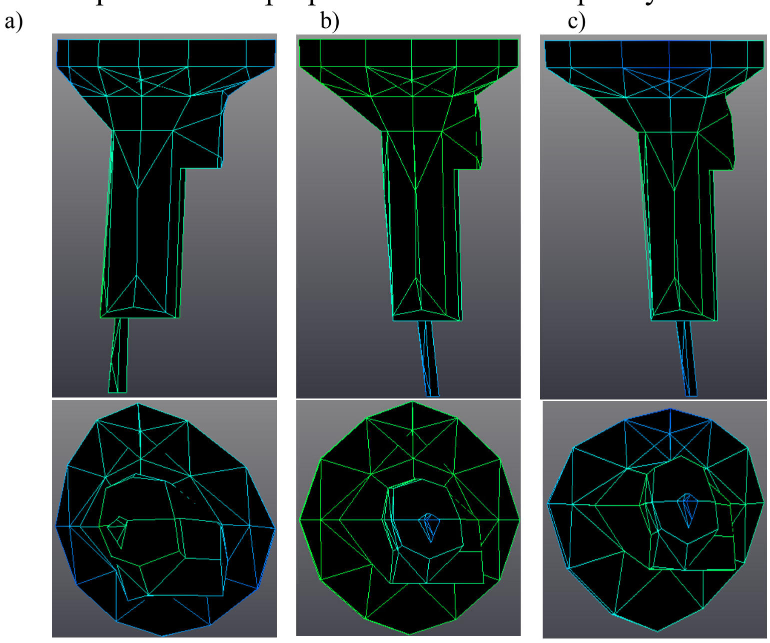
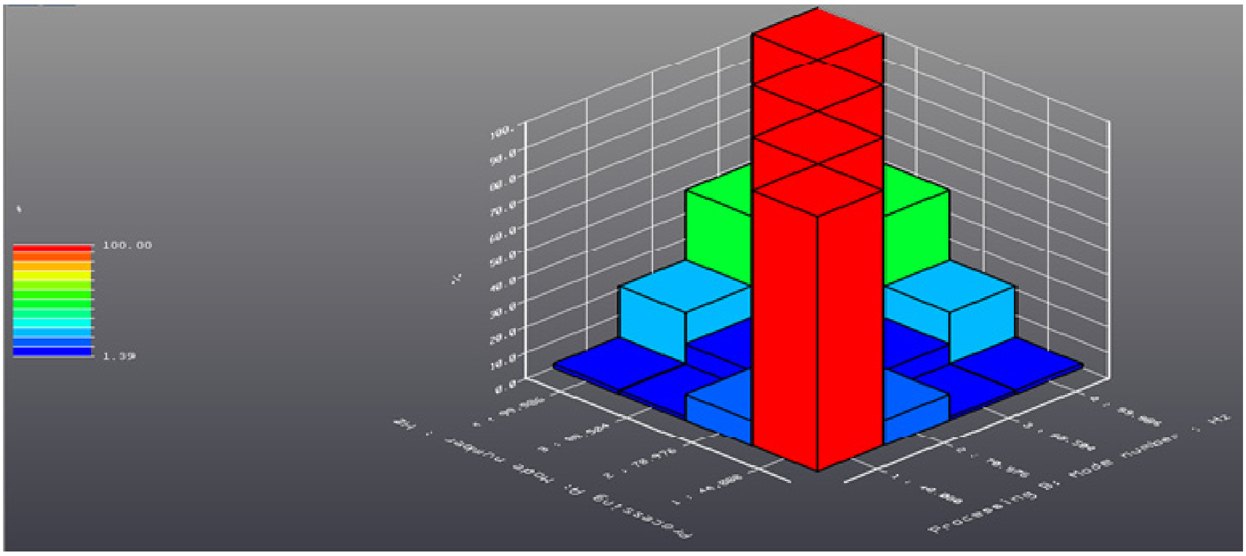


Fig. 6. Transmission gearbox sample mode shapes for frequencies: a) 44,16 Hz, b) 88,23Hz, c) 99,98 Hz



Auto Modal Assurance Criterion (%)

	Mode No.	Frequency	Mode 1 44.080	Mode 2 70.976	Mode 3 88.504	Mode 4 99.986
1	Mode 1	44.080	100.000	10.140	1.385	1.645
2	Mode 2	70.976	10.140	100.000	9.160	21.413
3	Mode 3	88.504	1.385	9.160	100.000	49.096
4	Mode 4	99.986	1.645	21.413	49.096	100.000

Fig. 7. Sample of LMS Synthesis module and Auto-MAC criteria estimation

The stabilization process ran with tolerance of individual modal parameters: 1% for frequency, 5% for modal damping and 2% for mode shapes vector. In such case as this, where the operational modal analysis is applied, there is only this mode shapes identification possibility which became sufficiently well extorted during the experiment identification.

4. Conclusion

The development and progress of the human civilization is guided by the desire of difficult questions solving often connected with the varied fields of the science. However very often takes place the situation when introduced solutions are more complicated that first solution. In modern technical constructions this problems are also similar. From that reasons new devices and new diagnostics methods are developing which will be able to provide valuable information about technical state of that products.

Presented in this paper conducted investigations and modern engineering application allows to quick process of transmission gearbox identification including their own vibration and gearbox body mode shapes visualisation. The advantage of this method is fact that the studied object can be investigated during normal process of exploitation, the investigations don't generate additional costs and we got results basis of real signals that are generated through the exploitation process of studied object.

Conducted investigations of gearbox depended on delimitations of vibroacoustics measures for chosen gear sets and accomplishment the assessment of received results influence on transmission gearbox state by operational modal analysis methods. As a result we received the stabilization diagrams with natural frequencies with damping factors, the modal participation factors and estimated mode shapes. Analysing results of investigations for idle run, the first identified natural frequency (44,160 Hz) describes the movement of shaft unbalances. The figures of this own vibrations are very well visible on the animation of modal model in the geometry model analysis as determined deformations of model. Second identified figure of natural frequency (88,235 Hz) is caused the differential mass schedule on the gearbox casing and also the unbalances shaft movement - that influences on whole gearbox stiffness. Third figure of natural frequency (99,997 Hz) results from the way of studied gearbox fasten and the kind of the supports on which the whole construction of gearbox leant.

Introduced in this paper results of investigations are only the part of realized investigative project and they do not describe wholes of the investigative question, only chosen aspects. This paper is a part of investigative project WND-POIG.01.03.01-00-212/09.

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