DIAGNOSTICS INVESTIGATIONS OF DMG-1A TRANSMISSION WITH OPERATIONAL MODAL ANALYSIS METHODS APPLICATION

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Summary

The modern engineering applications using virtual environment to simulating calculations conducting finds wider use in the process of projecting and structure dynamic analysis of machine engines and devices. Utilization of the modal analysis, aiming on the aid of transmission technical state description process brought to obtainment of three modal model parameters that describing the studied object. The analysis of results allowed to the qualification of characteristic modal parameters set for chosen technical state and introducing the natural frequency shapes for given modal model parameters of the studied mechanical construction. The LMS Test.Lab software with Modal Analysis Lite module was used for modal analysis. The results of investigations were a modal model parameters and mode shapes of investigated transmission.

Keywords: operational modal analysis, diagnostic inference

BADANIA DIAGNOSTYCZNE PRZEKŁADNI DMG-1A Z ZASTOSOWANIEM METOD EKSPLOATACYJNEJ ANALIZY MODALNEJ

Streszczenie

Metoda eksploatacyjnej analizy modalnej jest jedną z wielu technik wirtualnych wykorzystywanych w aspekcie diagnozowania maszyny i urządzeń. Istotną zaletą, szczególnie dla przemysłu jest fakt, że eksploatacyjna analiza modalna umożliwia identyfikację parametrów modelu modalnego jedynie w oparciu o pomiar odpowiedzi tego układu w trakcie eksploatacji. Podstawowym założeniem w tej metodzie jest to, że wymuszenie w trakcie pracy układu mechanicznego ma charakter losowy, a mierzony sygnał odpowiedzi układu na to wymuszenie jest niestacjonarny w czasie. W artykule przedstawiono metodę eksploatacyjnej analizy modalnej jako skuteczną formę diagnozowania maszyn. Przeanalizowano ponadto zachowanie się tej metody w przypadku wprowadzanych zmian do układu mechanicznego. Eksploatacyjną analizę modalną zastosowano do identyfikacji parametrów modalnych rzeczywistego układu mechanicznego jakim była przekładnia zębata DMG-1A.

Słowa kluczowe: eksploatacyjna analiza modalna, wnioskowanie diagnostyczne

1. Introduction

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 that introduced solutions are more difficult than on the beginning. 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 a valuable information about technical state of that products.

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 a valuable information about dynamic properties drawing ahead machine and other aspects or possible relationships among them. 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. Up to now an existing diagnostic procedures based on state symptoms slowly changes into diagnostics process based on machine engines models that describing their properties analysis.

Used in the diagnosing process models are identified basis on the real object investigative results. During diagnostics process we could create two kinds of diagnostic models: functional or structural. In diagnostics aspects of investigations, the structural models are more valuable, because they enable to show relations between current elements of construction and model specific properties. The diagnostics procedure for structural model requires the identification guidance of the model parameters and monitoring of these parameters changes on the basis of measurements guided during exploitation. Therefore this relationship between individual parameters and pointed construction elements are the basis for estimation of current dynamic state of investigated machine. Periodical monitoring of this relationship may manage to detection, locating and evaluation of the waste stage or damages of given mechanical unit.

Nowadays one of the most known way 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.

Preparation process for diagnostics investigations in these methods contain measure and reference points disposal and also frequency range define. The advantage of this method in use to identification of objects dynamic profiles is shore conditions and loads retain that are 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 refer with the machine engines working process stop so this action is unacceptable in this method usage. Disturbances triggered from next machine engines could causes the formation of additional poles on the created stabilization diagram.

2. Modal parameters estimation

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 in time domain and arrangement response. Estimation of modal model parameters in frequency domain basis on the input and output signal spectrum

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.

The LMS Test.Lab software with Modal Analysis Lite module was used for modal model poles estimation and analysis of mode shapes for multidegree arrangements with PolyMAX method.

PolyMAX method in frequency domain basis on the proper matrix formulation in frequency domain in shape:

$$\left[H\left(j\omega_f\right)\right]=\left[B\right]\left[A\right]^{-1}, \qquad [1]$$

where:

$\left[H\left(j\omega_f\right)\right]$ – FRF matrix with spectrum functions between all inputs m and all outputs l,

$\left[B\left(j\omega_f\right)\right]=\Sigma_{k=0}^{q}\left[\beta_k\right]s_f^k$ – polynominal matrix numerator,

$\left[A\left(j\omega_f\right)\right]=\Sigma_{k=0}^{q}\left[\alpha_k\right]s_f^k$ – polynominal matrix denominator.

Marking of mode shapes of mechanical construction is connected with utilization of Least

Squares Frequency Domain (LSFD) method in frequency domain.

3. DMG-1A transmission investigations

The investigations of transmission DMG-1A were conducted in the investigative laboratory in UTP Bydgoszcz. During investigations mass and springily properties of transmission were analysed. The main parts of transmission were:

- commutated engine,
- spur gear transmission,
- toothed pomp.

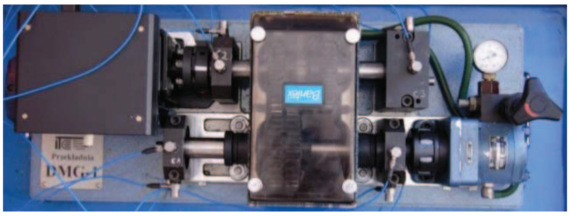

Figure 1. DMG-1A transmission [own picture]

DMG-1A transmission enable to provide measurement for chosen technical states. During investigations there were simulated four transmission cases:

- transmission in fit condition,
- transmission with broken tooth,
- transmission with skew axis,
- broken tooth and skew axis of transmission.

Figure no 2 presents the simulated technical state of transmission, when transmission is exploited with broken tooth.

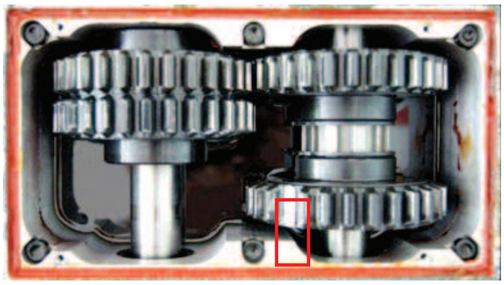

Figure 2. Transmission with the broken tooth [own picture]

Investigations were realized for three different rotation speed: (400, 600, 800 [min^{-1}]). Measure points were put on the bearing chassis of transmission. In each of measure points were realized measurements in three mutually perpendicular directions. Figure no 3 presents transmission DMG-1A geometrical model used for analysis with signal acquisition points.

The dynamic signal LMS SCADAS III recorder was used for signal acquisition. During investigations conducted 8 measuring sessions in which were recorded 13 signals of vibration acceleration, one of them was from reference point. The highest measure frequency range were definite till 400 [Hz]. In case of proper qualification of measure frequency range for DMG-1A transmission investigations the preliminary measurements were conducted with use of the impulse input function for chosen construction areas.

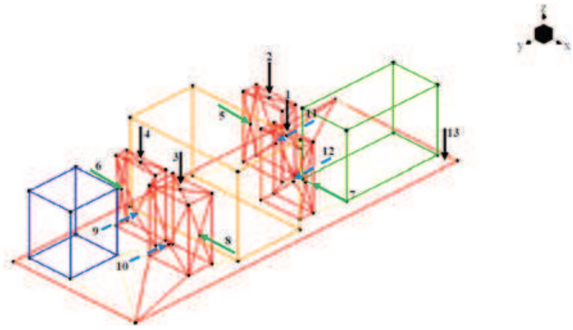

Figure 3. Transmission DMG-1A geometric model with mark of signal acquisition points [own source]

There were recorded per 300 seconds time response function with 512 [Hz] testing frequency. The essential aspect of the conducted investigative process was the proper modal model for the given construction technical condition obtainment. Observations of modal model individual parameters changes could enable to establish what influence of introduced changes call out in the construction through investigations in reference to transmission in fit condition. On figure no 4 were introduced respective stages of conducted investigations.

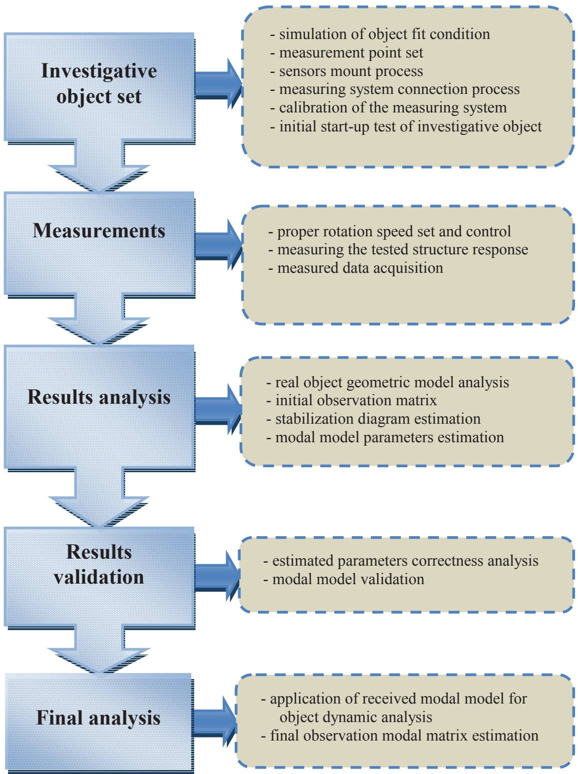

Figure 4. Respective stages of conducted investigations [own source]

It was decided during the experiment realization that measurement of time response courses will be realized both for the input function and for the arrangement answer on this input function – this possibility will enable the estimated poles proving process in both (time and frequency) domain. For the proper data usage in Modal Analysis Lite module the recorded time response signals have to be transformed to the spectral figure (Crosspower) using Op. Data Collection module. Figure no 5 presents sample of Op. Data Collection module window during investigations.

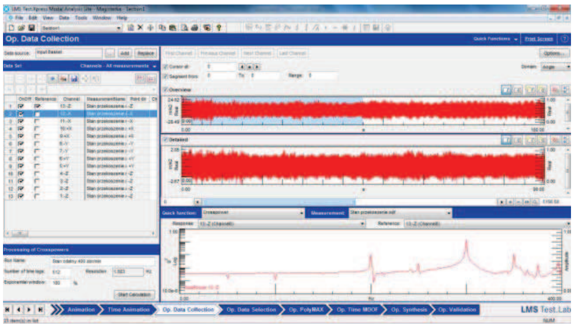

Figure5. Sample of Op. Data Collection module window during investigations [own source]

This data transformation enable to modal model estimation basis on stabilisation diagram analysis, where we mark order of modal model and damping factor for estimated natural frequency of object. Figure no 6 presents sample of stabilisation diagram with usage of PolyMax method during investigations for modal parameters estiation.

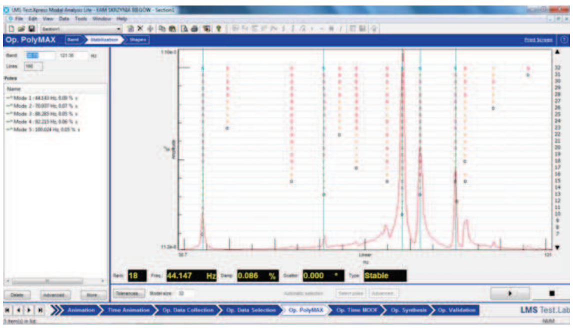

Figure6. Sample of stabilisation diagram with usage of PolyMax method used for results obtainment [own source]

Analysing 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 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.

Table no 1, 2 and 3 presents only the chosen part of results received during modal identification of transmission for transmission with broken tooth for five sensors (the first one was reference sensor) after stabilization diagrams analysis.

Table 1. Observation matrix for transmission with broken tooth for 400 min^{-1} rotation speed

Transmission with broken tooth			
Point	Rotation speed [min^{-1}]		
	400		
	Natral Frequency [Hz]	Damping factor [%]	Modal model Order
C1- reference point	49,916	0,57	9
	100,295	0,65	11
	199,925	0,05	9
	250,888	0,75	12
	299,966	0,01	12
	371,516	1,23	24
	384,296	2,15	8
C2	49,916	0,57	9
	100,295	0,65	11
	199,925	0,05	9
	250,888	0,75	12
	299,966	0,01	12
	371,018	1,02	20
	383,860	2,15	11
C3	49,916	0,57	9
	100,295	0,65	11
	199,925	0,05	9
	250,888	0,75	12
	299,966	0,01	12
	372,020	1,17	26
	382,921	1,72	19
C4	49,916	0,57	9
	100,295	0,65	11
	199,925	0,05	9
	250,888	0,75	12
	299,966	0,01	12
	372,029	1,21	23
	382,921	1,72	19
C5	49,916	0,57	9
	100,295	0,65	11
	199,925	0,05	9
	250,888	0,75	12
	299,966	0,01	12
	372,029	1,21	23
	384,296	2,15	8

Table 2. Observation matrix for transmission with broken tooth for 600 min^{-1} rotation speed

Transmission with broken tooth			
Point	Rotation speed [min^{-1}] 600		
	Natral Frequency [Hz]	Damping factor [%]	Modal model Order
C1- reference point	20,628	0,93	21
	50,070	0,75	19
	99,889	0,15	14
	199,898	0,01	14
	299,875	0,01	19
	370,101	2,96	10
C2	20,628	0,93	21
	50,070	0,75	19
	99,889	0,15	14
	199,898	0,01	14
	299,875	0,01	19
	370,628	2,90	7
C3	20,628	0,93	21
	50,070	0,75	19
	99,889	0,15	14
	199,898	0,01	14
	299,875	0,01	19
	370,628	2,96	10
C4	20,628	0,93	21
	50,070	0,75	19
	99,889	0,15	14
	199,898	0,01	14
	299,875	0,01	19
	370,628	2,96	10
C5	20,628	0,93	21
	50,070	0,75	19
	99,889	0,15	14
	199,898	0,01	14
	299,875	0,01	19
	370,628	2,96	10

On the basis of results introduced in tables we could conclude, that with growth of transmission rotation speed the number of estimated stable poles and recognised natural frequency decrease. The proper recognition of stable poles has influence directly on the respective mode shapes of construction. Under introduced figures presents most interesting mode shapes of transmission recognized during investigations, that shows more important construction properties. Comparing got results and individual mode shapes were noticed the change of all modal parameters. In case of modal model parameters manifested itself by natural frequency and damping factor value change. The transmission with broken tooth case is the best example for analysis of damping factor and suitable natural frequency value increase changes. The mode shapes change analysis is also valuable tool for construction analysis between all simulated transmission conditions that will marks the differences for all estimated natural frequencies. For example the figure no 7 present a sample of mode shapes visualization for transmission in fit condition, in which all estimated shapes has very similar character of movement. In this case of mode shapes, we could recognize only the movement of bearings casing.

Table 3. Observation matrix for transmission with broken tooth for 800 min^{-1} rotation speed

Transmission with broken tooth			
Point	Rotation speed [min^{-1}] 800		
	Natral Frequency [Hz]	Damping factor [%]	Modal model Order
C1- reference point	100,013	0,37	21
	199,948	0,01	8
	299,931	0,08	22
	372,545	2,23	11
C2	100,013	0,37	21
	199,948	0,01	8
	299,931	0,24	22
	371,775	2,07	14
C3	100,013	0,37	21
	199,948	0,01	8
	299,931	0,01	22
	371,775	2,07	14
C4	100,013	0,37	21
	199,948	0,01	8
	299,931	0,28	22
	371,775	2,07	14
C5	100,013	0,37	21
	199,948	0,01	8
	299,931	0,89	22
	371,775	2,07	14

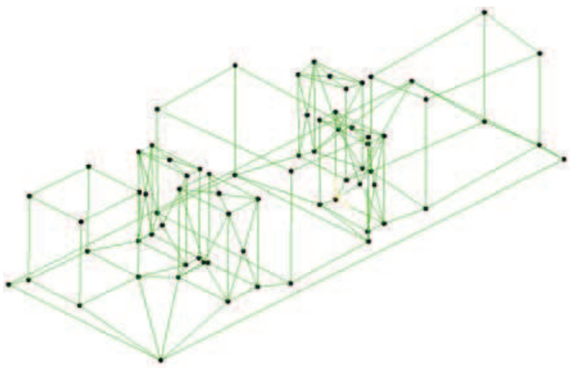

Figure 7. Mode shapes visualization for 49,917 [Hz] frequency for 400rpm [own source]

In case of transmission with broken tooth, the most important marked natural frequency was 384,296 [Hz] and their visualization. The extreme movement of mode shapes for this frequency present figure no 8.

Figure 8. Mode shapes visualization for 384,296 [Hz] frequency for 400rpm [own source]

Only one broken tooth in DMG-1A transmission caused significant change for all mechanical arrangement. The most visible was the specific deformation of bearing casing in direction of axis x, on which was situated shaft with broken toothed gear. The estimated mode shapes for frequency 384,296 [Hz] has bend phenomena.

In case of transmission with skew axis simulations the most important marked natural frequency was 49,716 [Hz] that has most complex phenomena. This frequency has shapes with twirl and bend phenomena on this transmission construction. Figure no 9 present estimated mode shapes visualization for transmission with skew axis simulations.

Figure 9. Mode shapes visualization for 49,716 [Hz] frequency for transmission with skew axis for 400rpm [own source]

In case of simulation of broken tooth and skew axis of transmission, two most important natural frequencies were recognized – each for one simulated damage of transmission. The first natural frequency 374,400 [Hz] in case of gear with broken tooth and the second frequency 379,022 [Hz] in case of skew axis of transmission.

The most important aspects of these diagnostics investigations is facts, that all marked natural frequencies are directly connected with simulated technical state and they are not marked for transmission in fit condition. This testifies about this, that in case of appearing damage in the transmission the completely new natural frequency appears in stabilization diagram, that is directly connected with specific kind of simulated damage. Also interesting phenomena is fact, that the mode shapes descend from transmission toothed gear damage is strictly connected with damping factor value increasing. In this case of mode shapes, the average value of damping factor has value in range from 2,07 [%] till 2,23 [%]. The average value of damping factor for other estimated mode shapes has value in range from 0,01 [%] till 0,93 [%].

4. Conclusions

The results of investigations were a modal model parameters and mode shapes estimations of complex mechanical object and structures basis on transmission DMG-1A investigations example. The most important natural frequencies and mode shapes were recognized. The sample results were introduced in tables and figures. As a final results we obtain the dynamic state description of real technical object with estimation of predominant properties of natural frequency and mode shapes.

The advantage of this method is fact that investigated object could be recognized during normal process of exploitation and modern engineering applications using virtual environment to simulating calculations lowering costs of investigations with higher precision of estimated results basis on real data measured during exploitation.

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