

Study of Modal Analysis from Order Tracking Data

Motoyasu SAKAGUCHI*

ABSTRACT

A Modal analysis technology using engine data taken under operational conditions is proposed. Taking the features of engine vibration into consideration, order tracking vibration data is used as an input to the subspace method and the identification of modal parameters is enhanced.

Paying attention to the mode shape used to validate FEM, and an accuracy validation by the proposed method was conducted. In the excitation test using pulses, pseudo tracking analysis was conducted. The results were compared with the modal analysis using the transfer function, and it was confirmed that a similar mode shape was obtained. By applying this method to actual engine vibration, the chain cover mode of diesel engines was identified, and it was confirmed that the orthogonality of the mode shape was enhanced compared to the mode shape obtained using conventional operational deflection shape and using average spectrum.

From the above, it is determined that the proposed method offers enhanced identification accuracy of the natural modes under actual engine operating conditions, as compared to the conventional method.

1. Introduction

Identifying the natural modes of actual structures is important for both identifying countermeasures to reduce vibration and increasing the accuracy of FEM models. Natural modes are usually identified through laboratory excitation tests. However, because of the effect of nonlinearity and differences in engine constraint conditions, the natural modes identified in excitation tests do not correspond to those in actual operation in some cases. For this reason, operational deflection shape (ODS) analysis, which simplifies the parameters by focusing on the amplitude of vibration at specific frequencies, has been used as a substitute procedure in the identification of natural modes in operation. This method is useful for studies of vibration countermeasures that visualize vibration⁽¹⁾. However, in analyses that focus on natural modes, such as comparisons between FEM and test results, and studies on the optimum measurement points for tests⁽²⁾, etc., the ODS is a fundamentally different phenomenon from natural modes, and the use of this method may cause reduced accuracy of the analysis.

On the other hand, an operational modal analysis

(OMA)^{(3),(4)} has been studied as a method of identifying natural modes from only the vibration response of actual machines in operation. The precondition for the OMA is that the excitation force acting on the structure should display no bias, such as white noise, in the frequency domain. However, a comb-shaped spectrum is generated due to rotational order harmonics dominant in the excitation force in an engine. Thus the application of the OMA is challenging, and accurate natural modes cannot be obtained.

Therefore, a method to identify natural modes in an operating engine accurately was proposed⁽⁵⁾. This method increased the accuracy of the modal identification by using order tracking data for the OMA input, and by taking the engine vibration characteristics into consideration. By focusing on the mode shape which is one of the natural mode characteristics and is frequently used in comparison with FEM results, the accuracy was verified again. From excitation tests simulating engine vibration, the mode shapes that were similar to the mode shapes obtained from conventional experimental modal analysis (EMA) using frequency response functions were obtained by the new method. Application of the new method to the actual

* Automobile R&D Center

engine vibration obtained better orthogonality of mode shapes than those obtained by the ODS. These results confirmed that the new method increased the identification accuracy of mode shapes.

2. Method Used to Increase the Identification Accuracy of Modal Parameters

2.1. Stochastic Subspace Identification Method

The method used in the OMA is known as the stochastic subspace identification method⁽³⁾ or the subspace algorithm⁽⁴⁾. The theoretical foundations of this method have been described in detail in the papers cited in the reference literature, and will therefore be dealt with here only in outline. The stochastic subspace identification method is a method of identifying natural modes by subjecting a Hankel matrix, which consists of the correlation functions of vibrational responses, to singular value decomposition, to determine the dominant matrix for the responses. Figure 1 shows an overview of the method. The correlation functions of the responses that are used for the input are obtained by inverse Fourier transformation of the cross spectrum. If the input excitation force satisfies the precondition of a low level of bias in the frequency domain, the stochastic subspace identification method can be used to obtain the natural frequency, mode shape, and modal damping ratio directly from the vibration data. However, because the method does not employ a frequency response function, scaling cannot be conducted using modal mass or other parameters. Because of this, the results obtained from this method can be used to verify FEM results, but cannot be used in substructure synthesis methods.

Since the stochastic subspace identification method does not need excitation tests as noted above, it has often been used in the analysis of structures for which such tests are inconvenient, such as large buildings. The method also enables natural modes to be determined under operating conditions in cases such as automotive suspensions⁽³⁾, in which modes are highly amplitude-dependent and it is difficult to identify modes in an excitation test that correspond to those occurring in actual operation.

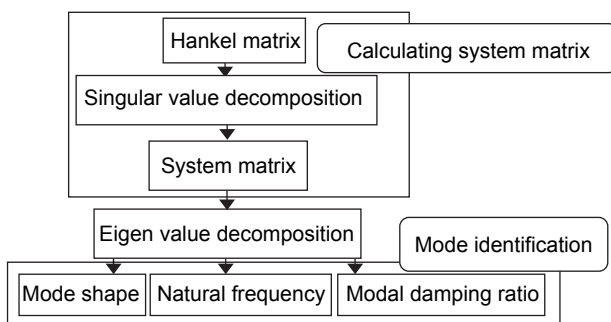


Fig. 1 Overview of OMA

However, when there is a specific bias in the excitation force, it is not possible to judge whether the result represents the true natural mode or the effect of the excitation force, decreasing the reliability of the analysis.

2.2. Issues Arising when the OMA is applied to Engines

As Fig. 2 shows, rotational order harmonics are dominant in engine vibration and generate a comb-shaped spectrum. The precondition that the excitation force should demonstrate a low level of bias in the frequency domain, as indicated in the preceding section, is therefore not satisfied, and the OMA cannot be used. To apply the OMA, it is necessary to reduce the bias of the excitation force by averaging the vibration spectrum over time or in relation to engine speed (Fig. 2). In many cases, commercially available OMA systems simply average the vibration spectrum, assuming white noise of the type generated by microtremors in buildings. When this method is applied to engine vibration, however, low response level vibration other than that of order harmonics is also included in the averaging. Therefore, the S/N ratio and other reliability of data decline and so does the accuracy of modal identification.

2.3. Proposal of a Method of Increasing Accuracy of Analysis in Engines

The order harmonic data obtained by tracking analysis should be used as the input data in the OMA to increase the identification accuracy of modal parameters when an engine is operating. The excitation generating the order harmonics originates from the inertial force of the interior of the engine, the excitation force produced by the operation of various mechanisms, including the meshing of the gears, polygon effect of the chain and the sprockets, and combustion pressure, among other sources, and therefore remains comparatively stable even when engine speed varies. By focusing on these matters, apply the order tracking analyses to vibration, and use the results as spectra. More accurate S/N ratios than those obtained by simple averaging of the vibration spectrum, as well as the response with a lower level of bias in the frequency of excitation force can be obtained by this method. It could therefore be assumed that the use of order tracking data would increase the accuracy of the OMA (Fig. 3).

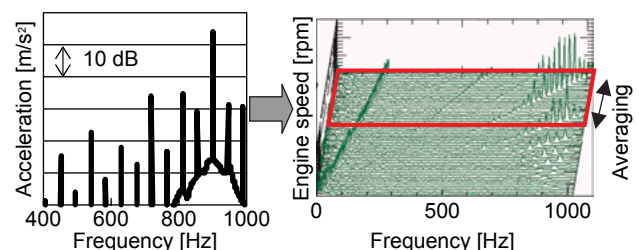


Fig. 2 Averaging engine vibration

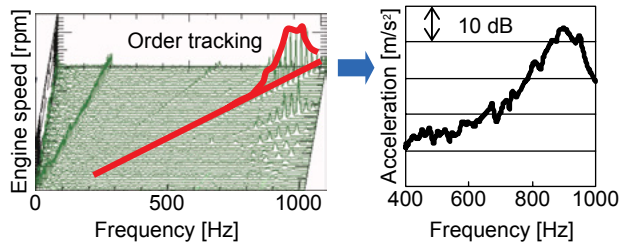


Fig. 3 Order tracking of engine vibration

3. Methods of Verifying Accuracy

3.1. Verification of Accuracy Using Excitation Tests

Prior to verifying the accuracy of the new method in operating engines, excitation tests were conducted to verify whether the OMA could be applied to order tracking data (Fig. 4). The targets for measurement were the cylinder head and cylinder block with the chain cover and head cover attached. An electromagnetic exciter was used to excite the oil pan deck of the cylinder block, and only the vibrational response of the head cover was measured. To enable a simulated order tracking analysis to be conducted in the excitation test, pulse waveforms such as saw-tooth waves were applied for excitation, and the interval between the pulses was continuously varied to simulate an increase in engine speed. Order tracking analysis was applied to this excitation, and the modes were identified by OMA. To provide a comparison, an excitation test was also conducted with random noise to compare the results of the OMA and EMA. The widely-used time domain poly-reference method was used for the EMA.

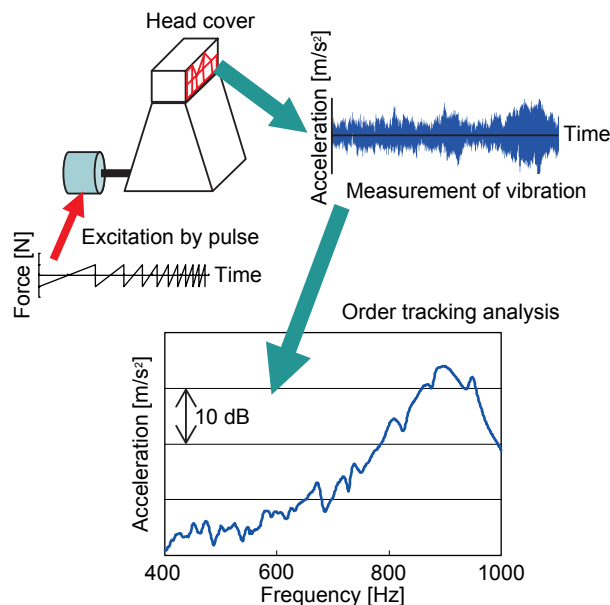


Fig. 4 Overview of pulse excitation test

3.2. Engine Operation Tests

Then the accuracy of the OMA was tested using the vibration generated by an engine operating on a test bench. Out-of-plane vibration on the surface of the chain cover in a 4-cylinder 2.2 L diesel engine was measured (Fig. 5). External drive was used to reduce noise generation by the combustion excitation force, and vibration was measured as engine speed was increased. An order tracking analysis focusing on the polygon order of the chain and sprockets was conducted as input to the OMA analysis. The conventional method of averaging the spectrum was also used, and the modes obtained from the OMA using the order tracking data were compared with the modes obtained using averaged data. The ODS, which has traditionally been used as a substitute to identify natural modes in operation, was also conducted by using the results of tracking analysis, and the results were compared to those by the OMA.

3.3. Comparisons of Mode Shape

A mode shape is an important parameter which is obtained from modal analysis and used in the visualization of vibration and comparison of results with FEM results. Therefore, the mode shape was focused on in testing the accuracy of the OMA. This section discusses a method of assessing the accuracy of mode shape identification using the Modal Assurance Criterion (MAC)⁽⁶⁾. When the mode vectors ϕ_r and ψ_s are given, the MAC can be found using Eq. (1), where the subscripts r and s express the order of the natural modes.

$$MAC(\phi_r, \psi_s) = \frac{|\phi_r^H \psi_s|^2}{\|\phi_r\|^2 \|\psi_s\|^2} \quad (1)$$

ϕ_r^H is the conjugate transpose of ϕ_r . The MAC expresses the degree of correlation between the two vectors ($0 \leq MAC \leq 1$). When $MAC = 1$, there is a complete correlation between the vectors; when $MAC = 0$, there is no correlation.

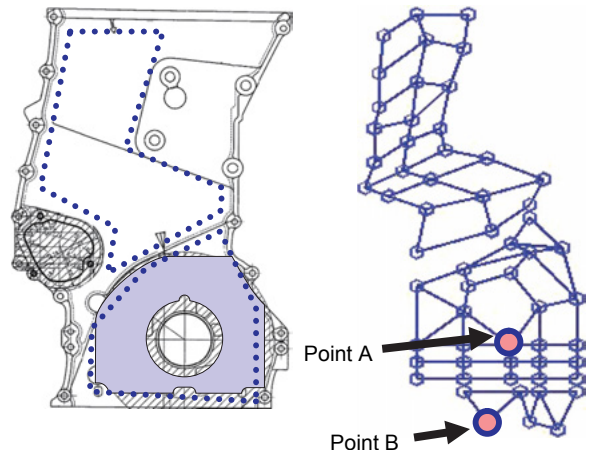


Fig. 5 Chain cover and measurement points

The MAC is usually calculated between multiple modes and expressed as a matrix. Correlation may be autocorrelation, i.e. correlation between the same modes, or cross correlation, i.e. correlation between different modes. In the case of autocorrelation, the lower the level of non-diagonal elements, the higher the orthogonality of the mode shapes, and the higher the degree of accuracy in identifying the mode. In the case of cross correlation, when the value of the diagonal elements is close to 1, there is a similarity between the mode shapes.

In order to objectively evaluate the accuracy of mode shape identification across multiple modes, in addition to formulating the matrix, the mean values of the MAC are evaluated. In the case of autocorrelation, by focusing on the orthogonality of the modes, the mean value of the non-diagonal elements expressed by Eq. (2) is evaluated. In the case of cross correlation, by focusing on the similarity of the modes, the mean value of the diagonal elements expressed by Eq. (3) is evaluated.

$$\text{Mean (Non - diag)} = \overline{\text{MAC}(\phi_r, \phi_s)} \quad (r \neq s) \quad (2)$$

$$\text{Mean (Diag)} = \overline{\text{MAC}(\phi_r, \phi_s)} \quad (r = s) \quad (3)$$

4. Results

4.1. Verification of Accuracy Using Excitation Tests

Figure 6 shows the results of a comparison of head cover mode shapes obtained by applying the OMA and EMA to an excitation test. Figure 6 (a) shows the MAC matrix for the OMA autocorrelation, 6 (b) shows the

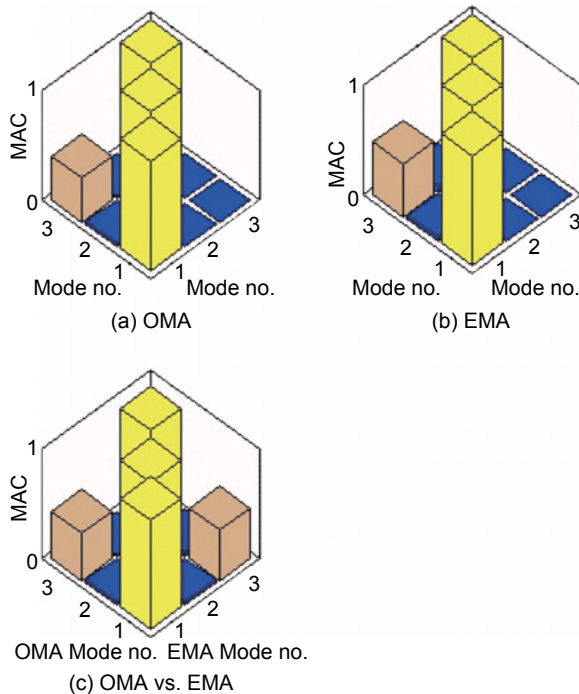


Fig. 6 MAC from excitation test

Table 1 Comparison of natural frequencies of head cover by OMA and EMA

Mode no.	Natural frequency [Hz]		Difference [%]
	OMA	EMA	
1	831	829	0.2
2	901	901	0.0
3	947	947	0.0

MAC matrix for EMA autocorrelation, and 6 (c) shows the MAC matrix for cross correlation between the OMA and EMA. In the case of the OMA and EMA autocorrelation, the MAC of 0.40 and above existed for the non-diagonal elements. As the OMA and EMA results display the same tendency, this can be assumed that the result of insufficient measurement points might be the reason rather than of the difference in accuracy between the OMA and EMA. The magnitude of the diagonal elements in the cross correlation shown in 6 (c) was 0.89 and above, and the mean value of the diagonal elements was high at 0.94. These results show that the mode shapes obtained using the OMA and EMA are similar, indicating that the OMA possesses the same level of accuracy in identifying mode shapes as the EMA.

Table 1 shows the natural frequencies determined by the OMA and EMA. The difference between the natural frequencies of the OMA and EMA was 0.2% or less. The results obtained from the both are almost the same, and thus the accuracy levels of the OMA and EMA in identifying natural frequencies are equivalent.

4.2. Engine Operation Tests

Figure 7 shows the results of a tracking analysis of 36th order vibration (the chain and the sprocket polygon order) of the chain cover. The measurement points were Point A and Point B shown in Fig. 5. Vibration peaks existed at 2200 rpm (1320 Hz), 2400 rpm (1440 Hz), and 2500 rpm (1500 Hz), and the analysis focused on these frequencies.

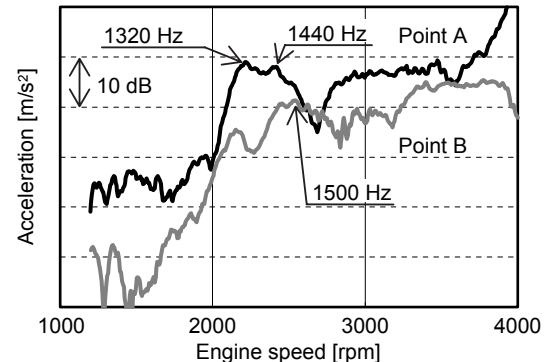


Fig. 7 36th order vibration of chain cover

Figure 8 shows the results of the ODS using tracking analysis results. Figure 9 shows the mode shapes obtained by applying the OMA to tracking data. The mode shapes shown in Fig. 9 are, respectively, the vibratory shapes of the upper section of the crankshaft through hole, the lower part of the crankshaft through hole, and the oil pan. All ODS results shown in Fig. 8 include the vibratory shapes of the lower part of the crankshaft through hole. This is assumed to be because modes cannot be separated in ODS, and are therefore displayed superimposed.

Figure 10 shows results for the autocorrelation MAC. 10 (a) shows the MAC matrix for the mode shapes obtained from the results of tracking analysis, 10 (b) shows the MAC matrix for the mode shapes obtained from averaging, and 10 (c) shows the MAC matrix for the ODS autocorrelation using the results of tracking analysis. Table 2 shows the mean value of the non-diagonal elements of MAC for autocorrelation. The maximum non-diagonal element for the OMA using tracking analysis was 0.11, the maximum for the OMA using averaging was 0.31, and the maximum for ODS was 0.66. The OMA using tracking analysis also displays the lower mean figure for non-diagonal elements in Table 2, with 0.06 compared to 0.20 for the OMA using averaging and 0.44 for the

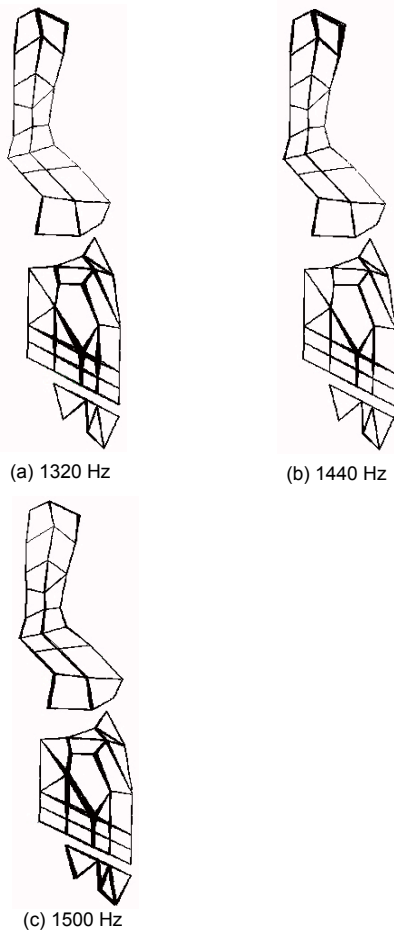


Fig. 8 ODS of chain cover

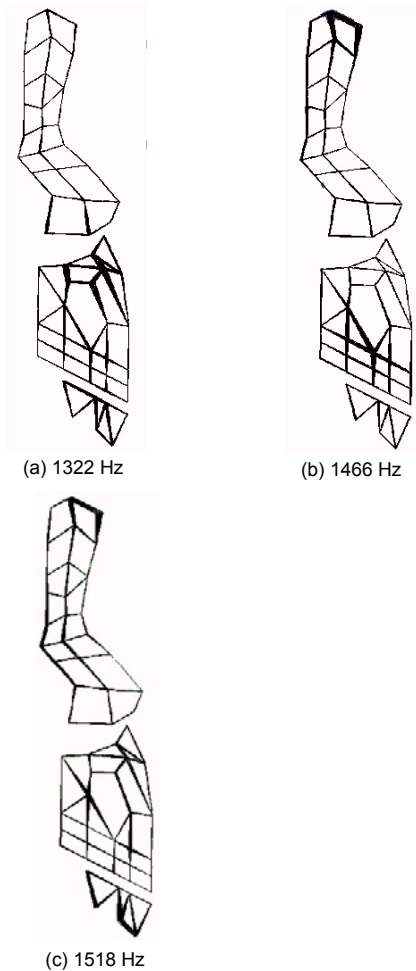


Fig. 9 Mode shape of chain cover

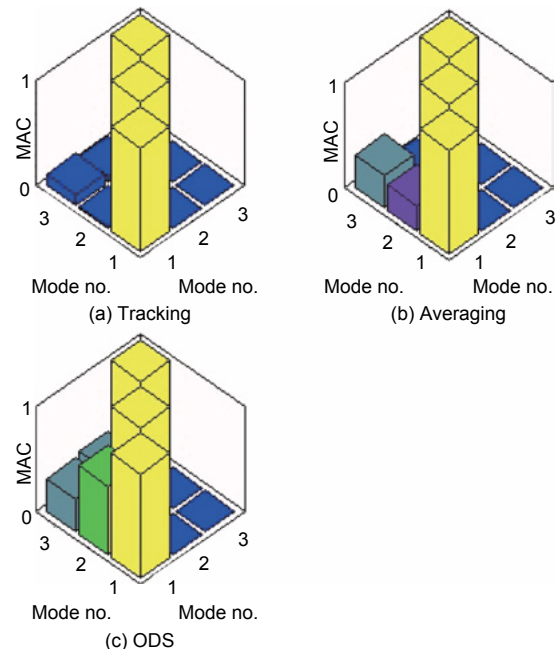


Fig. 10 MAC from operational condition

Table 2 Comparison of MAC mean from operational condition

Mean of non-diagonal MAC		
Tracking OMA	Averaging OMA	Tracking ODS
0.06	0.20	0.44

Table 3 Comparison of natural frequencies of chain cover by OMA from tracking and averaging

Mode no.	Natural frequency [Hz]		Difference [%]
	Tracking	Averaging	
1	1320	1322	0.2
2	1425	1466	2.9
3	1512	1518	0.4

ODS. The natural modes obtained by means of the OMA using tracking analysis demonstrate higher orthogonality and the mode shapes can be separated while the orthogonality of the mode shapes generated by the OMA using averaging and the ODS is lower and the shapes cannot be separated. The accuracy of the OMA using tracking analysis in identifying mode shapes is therefore higher than that of the conventionally used ODS or OMA using averaging.

Table 3 shows a comparison of natural frequencies obtained from the vibration data using tracking analysis and averaging. There is a 2.9% difference in the natural frequency identified in mode 2, but this would not represent an issue in practice. These results indicate that even the conventional OMA possesses a sufficient accuracy in identifying natural frequencies.

5. Conclusion

The order tracking vibration data has been used in the OMA to increase the accuracy of identifying natural modes in an operating engine. Studies focusing on mode shapes generated the following conclusions:

- (1) In a study using excitation tests with pulse waveforms, the OMA using the order tracking vibration data generated similar mode shapes to the EMA using frequency response functions, and demonstrates that the new method possesses identical accuracy with that of the EMA in identifying modal parameters.
- (2) In a study using the vibration of the chain cover in an operating engine, the orthogonality of natural modes obtained by means of the OMA using the order tracking data was higher than that of natural modes obtained by means of the conventional OMA that averages vibration and the ODS which is used as a substitute for the OMA, and demonstrated that the new method increases the accuracy of mode shape identification.

Only the objective natural modes under actual engine operating conditions can be identified by this method. In future, the application of this method to higher-frequency domains in which the mode density is higher can be expected to increase analysis efficiency.

References

- (1) Otsuka, M.: How to minimize diesel combustion noise by improving engine structure, 2005 JSAE Annual Congress Proceedings, No. 36-05, p. 7-10 (2005)
- (2) Senba, K., Takahashi, K., Sakaguchi, M., Yamagishi, J.: Building Block Approach Method Application to Powertrain Component Vibration Analysis, HONDA R&D Technical Review, Vol. 11, No. 2, p. 135-142
- (3) Hermans, L., Auweraer, H. V., Abdelghani, M.: Modal testing and analysis of a car under operational conditions, 16th IMAC, p. 1000-1006 (1998)
- (4) Hino, J., Komoto, T., Yoshimura, T.: Identification of Modal Parameters by Subspace Algorithm (Vibration Testing by Impulse Excitation), Japan Society of Mechanical Engineering Dynamics & Design Conference 2005, No. 603 (2005)
- (5) Sakaguchi, M.: Identification of modal parameters from order tracking data, 2006 JSAE Annual Congress Proceedings, No.36-06, p.1-4 (2006)
- (6) Yoshimura, T.: Modal Assurance Criterion, Modal Analysis Handbook, CORONA PUBLISHING CO., LTD., p.128 (2000)

■ Author ■



Motoyasu SAKAGUCHI