

Connector Modeling in Vibration Prediction for Motorcycles

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ABSTRACT

Conventionally, when predicting motorcycle vibration using the data from drawings, connector characteristics for the components were uncertain. Therefore, using test pieces, we tested bolt fastening connector characteristics in order to investigate a potential analysis method. Further, a detailed FEM model was created, and the contact domain of connectors was calculated using the nonlinear-GAP static analysis of MSC/NASTRAN. Normal mode analysis was performed by converting the domain into a high rigidity spring element, and it was verified that the resonance frequencies for each mode were generally the same. By applying this method to an actual motorcycle and by also applying NASTRAN's RESVEC calculation to connectors of a component, the FEM model was improved.

1. Introduction

In motorcycles the frame is the key component that connects the engine, drive train, steering, suspension and other components. The frame must be strong enough to support the weight of these components and have appropriate stiffness in terms of vehicle performance. At the same time vibration characteristics acceptable to the rider must also be taken into account. These elements are interrelated, and a well-balanced structure is required. There are various types of engines and frames, and the element balance changes according to the vehicle performance and use. In addition, the nature of motorcycles is such that light weight and compact size are also vital issues, and the engine is often utilized as a part of the frame stiffness or rigidly connected to the frame due to limitations of the chain-drive drive train. Under these conditions, vibration analysis was performed in conjunction with other departments for strength, steering stability and other characteristics, and a finite element model was created to serve as a basis for analysis on the assumption of sharing with other applications.

2. Creating the Component Model

Recent advances in digital technology have promoted more detailed finite element models of the frame, muffler and other components, and major advances have also been made in terms of accuracy. Conventionally modeling required enormous amounts of manpower, but the recent adoption of 3D drawings has facilitated modeling and made it possible to create finite element models of components almost in parallel with drawing creation. The modeling technology of each component affects the completed vehicle vibration prediction accuracy, and it is extremely important that continued efforts be made to improve the accuracy of modeling technology through experimental modal analysis in undetermined elements such as welding and press fitting.

In addition, at an engine speed of up to approximately 10,000 rpm and taking into account second order vibration, a frequency range of up to 330 Hz is necessary for the completed vehicle vibration prediction of motorcycles. The component model accuracy was improved using experimental modal analysis and the point impedance at the connection points.

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2.1 Improving Modal Model Accuracy

It has been said that in order to obtain a correct modal model for the completed vehicle model, component modal models are necessary in the frequency band approximately 2.5 times that required for the completed vehicle model. This means that to predict completed vehicle vibration for motorcycles up to 330 Hz, the components must be expressed accurately up to 800 Hz, and extremely advanced measurement technology is needed to extract good modal data in this frequency range. Most causes of calculation and test MAC (Model Assurance Criterion) matrix deterioration are rooted in insufficient measurement, and if the initial MAC matrix is inaccurate, then correct modal model updating becomes difficult. The procedure for efficiently improving model accuracy consists of dividing component models into individual groups for which the mode shape or the modal frequency sensitivity can be calculated. Applying this procedure clarifies the model deficiencies, such as problems with modeling, welding, contact, plate thickness and so on. Fig. 1 shows an example motorcycle frame FEM model and the calculations and test MAC values obtained using an updated model.

2.2 Utilizing Local Stiffness

Correct connector characteristics depend on the vibration prediction accuracy of the component connection points, so it is extremely important to focus on the connection points. Thus, these were verified using the point impedance. The point impedance is the frequency response function (FRF) of the local vibratory response (generally, acceleration) according to the local force input at that same point. In MSC-NASTRAN, the local modes that produce rigid behavior can be calculated from V70.0. These modes supply the residual stiffness that is missing from normal mode analysis, and are thus called residual modes (RESVEC: Residual Vector)⁽¹⁾. An extremely accurate frequency response function can be easily obtained by taking into account the rigid body modes, flexible modes and residual modes. Fig. 2 shows example frequency response functions when taking into account RESVEC and when synthesized

from only the flexible modes up to 500 Hz. The point impedance can be divided into the four states of (1) rigid body behavior, (2) resonance behavior, (3) stiffness behavior and (4) local resonance behavior.

Connection points are usually domains that induce some weak point, so the stiffness behavior significantly affects the component connection. Stiffness behavior is generally local mode pre-resonance behavior at high frequencies. In order to correctly express the connector characteristics, it is important to obtain a modal model which includes the local mode that generates this stiffness behavior. If the local stiffness of the connector can be taken into account, then

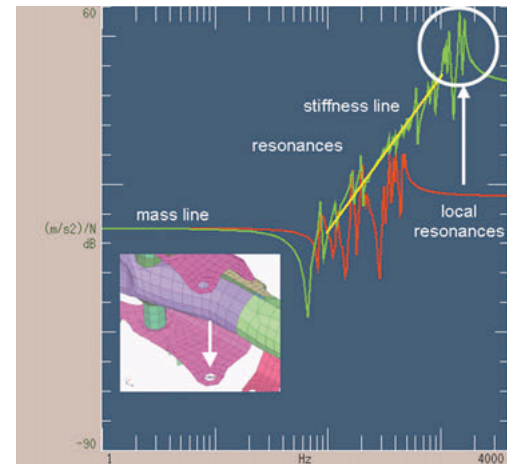


Fig. 2 Frequency bands for driving point FRF

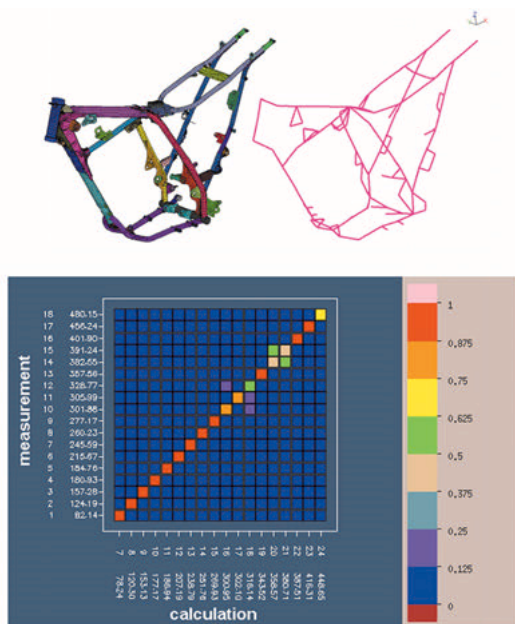


Fig. 1 Model and modal updating results of a motorcycle frame

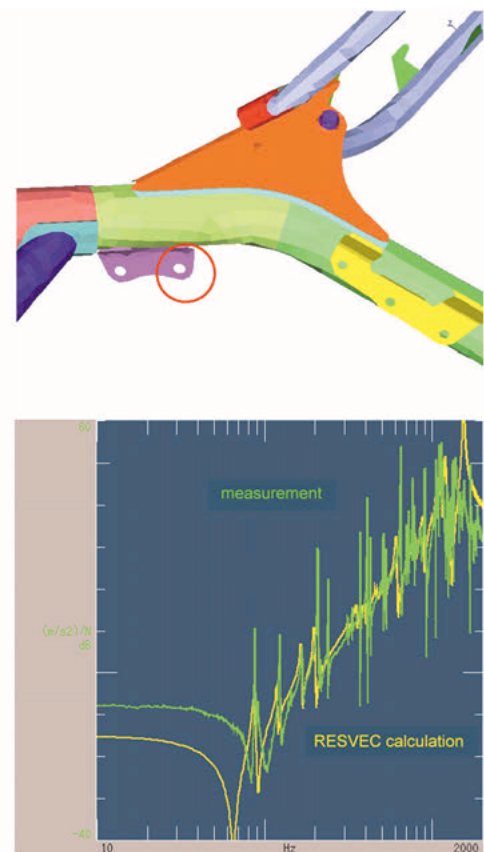


Fig. 3 Local model and driving point FRF updating procedure

there is no need to update the model expressing resonance behavior up to a sufficiently high frequency as previously mentioned. Therefore, the component modal models should be updated up to 1.5 times the frequency required by the completed vehicle model (approximately 500 Hz), and point impedance measurement should be utilized to correctly express the effects of local stiffness. The accuracy of local modeling (welding, press fitting, etc.), such as that shown in Fig. 3, can be improved by comparing the frequency response function calculated using RESVEC and the point impedance.

3. Component Connections

Elements of motorcycle component connectors include bolt fastenings, bearings, bushings and so on, and understanding the connector characteristics of these elements is an extremely important factor in vibration prediction and various other analyses. There are a number of methods for predicting system vibration such as the partial structure synthesis method, but connections between components are generally represented by a spring with six degrees of freedom as shown in Fig. 4.

However, these springs were obtained experimentally, and a database comprising a large number of tests was required⁽²⁾. In addition, when the structure changed significantly, these changes sometimes could not be supported or, occasionally, similar connectors were used, thus naturally lowering the prediction accuracy. To solve these problems, attempts were made to more analytically understand bolt fastening which is a typical connection method for motorcycles. Therefore, analysis was carried out using a test piece forming a simple connector and a simple model created by cutting out a portion of the frame in order to understand connector phenomena in detail. Finally, an example of an actual motorcycle engine-frame connection is presented.

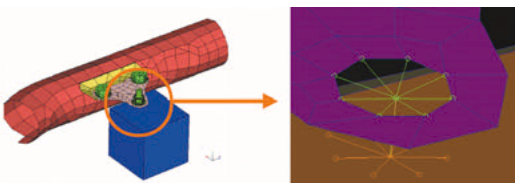


Fig. 4 Conventional connector modeling

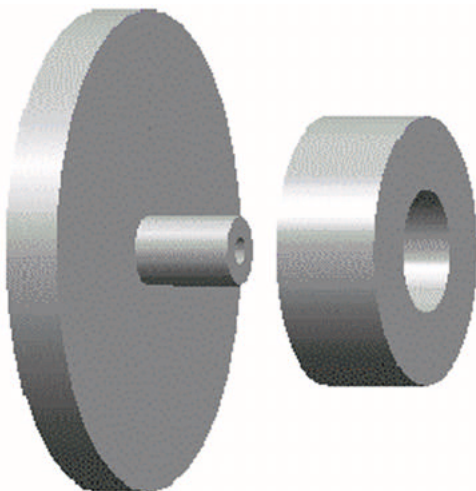


Fig. 5 Structure of test piece

3.1 Test Piece

The test piece hypothesized a motorcycle engine mount, and consisted of a two-piece component connected with an M10 bolt as shown in Fig. 5. The bolt axial force was controlled by affixing two strain gauges on the sides of the bolt. The surface roughness was at the same machining level as the frame and engine connectors. Tests varying the bolt axial force, attached plate thickness and other factors were conducted to investigate the relationship with the natural frequency, and the results showed that changes in the attached plate thickness had a significant effect.

Therefore, a detailed partial analysis model for the above test piece was created as shown in Fig. 6, and NASTRAN nonlinear-GAP analysis was performed to investigate deformation of the contact domain⁽³⁾. A 1/4-scale model was used to reduce the analysis time. For modeling conditions, peeling was thought not to occur between the bolt flange and the thin plate, so GAP elements were set only in the collar and thin plate contact portions, and the bolt flange and thin plate were connected by high stiffness springs. Fig. 7 shows the deformation drawing analyzed at an axial force of 1000 N. This figure shows that when axial force is applied, the thin plate portion peels from the collar. The same analysis was performed with varying plate thicknesses, and Fig. 8 shows the graph of the GAP element reaction force and the distance in the radial direction.

Portions with a positive reaction force are thought to be where the collar and thin plate are in firm contact, and the figure shows that the contact domain increases with the

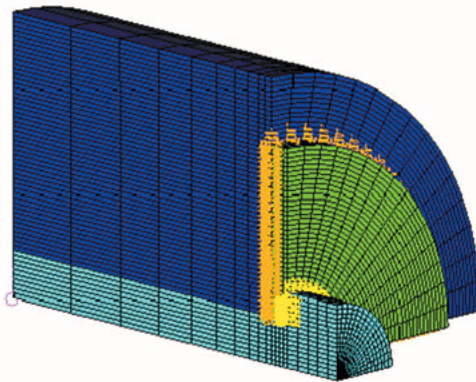


Fig. 6 Fine mesh model for GAP analysis

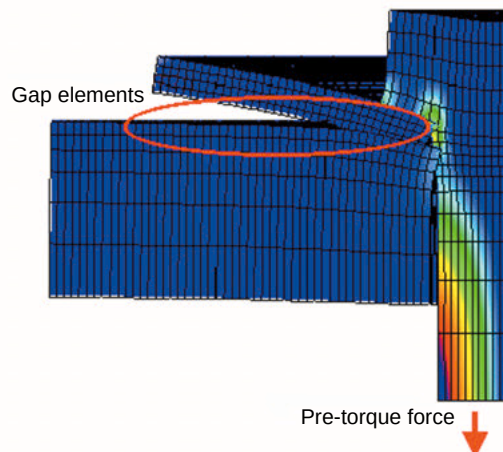


Fig. 7 Deformation result

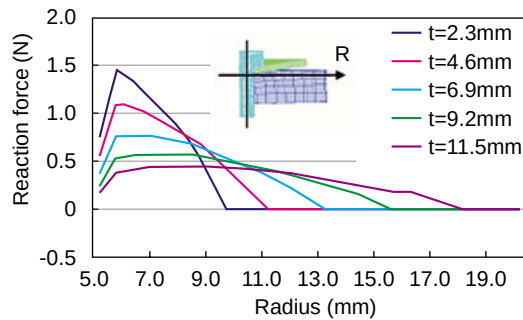


Fig. 8 Determination of contact area radius by reaction force of GAP elements

thickness of the attached thin plate. Once the contact domain was known, only this contact domain was connected with high stiffness springs ($1e4N$ to $1e5N/mm$). Modeling was then performed again, followed by normal mode analysis, and the results were compared with the measurement values for verification. Fig. 9 shows the calculation flow, and Figs. 10, 11 and 12 show the calculation and test results for each plate thickness and the resonance frequencies of an integrated body (referred to as "1 body" in the figures) which are provided as reference. The resonance frequencies matched closely regardless of the thin plate thickness, thus confirming the validity of the developed modeling method.

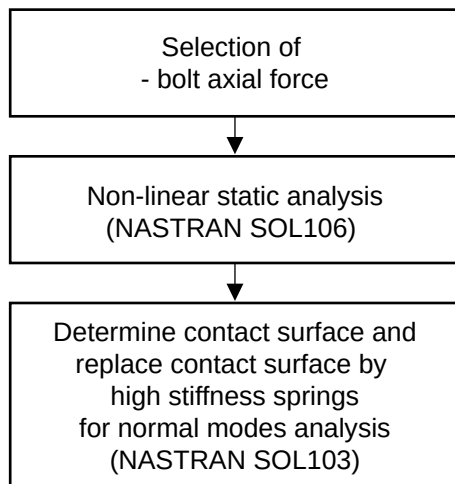


Fig. 9 Analysis flow



Fig. 13 Simplified model of engine frame connection and FEM model

3.2 Frame Cut-out Model

Next, analysis was performed on a connector often applied to motorcycle engines and frames. The frame portion of the analysis model consisted of a cut-out part of an actual frame that was restrained at both cut ends, and the engine portion was a cubed block model. Fig. 13 shows

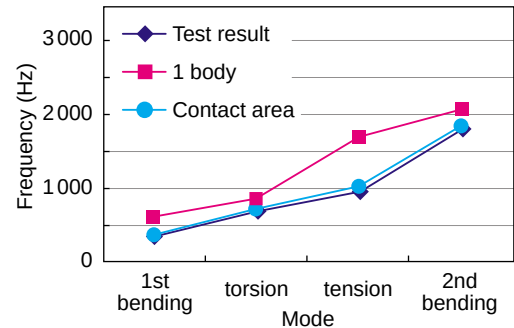


Fig. 10 Natural frequency of test-piece ($t = 2.3$ mm)

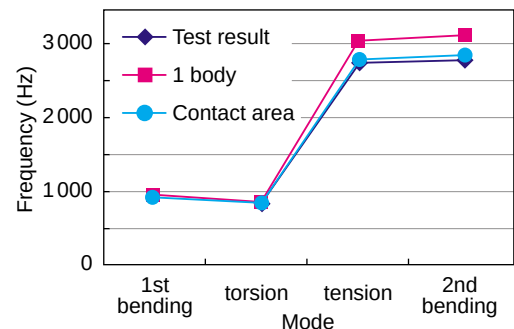


Fig. 11 Natural frequency of test-piece ($t = 4.6$ mm)

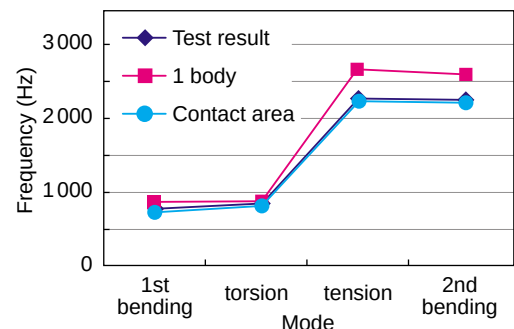


Fig. 12 Natural frequency of test-piece ($t = 11.5$ mm)

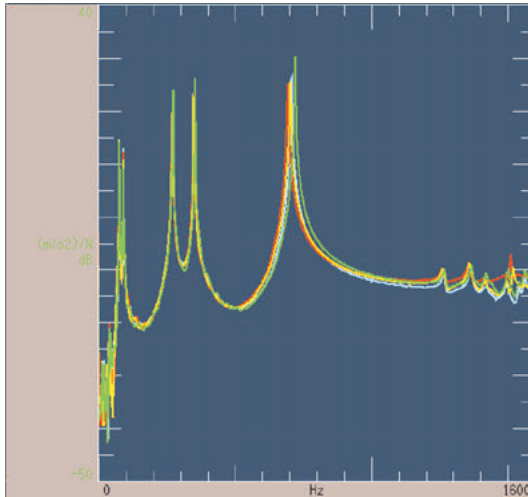


Fig. 14 Impact test result for reciprocity

the actual model used in the test and the FEM model. There are some faint mode resonance frequencies that can be confirmed by simplifying the model in this manner, and the excitation test results are shown in Fig. 14. This excitation test was performed to confirm reproducibility, and the bolt axial force was controlled according to the tightening torque. The modes near 800 Hz show significant variance due to repetition, and this is thought to be mainly due to the bolt tightening torque control. In contrast, the test piece described in the previous section did not experience significant variance up to 3.2 kHz.

In addition, there is an approximately 0.5 to 1.0 mm tolerance between the engine and frame mounts due to production related reasons. When creating a bolt fastening model, GAP elements are thought to exist in four domains as shown in Fig. 15. However, as shown in this same figure, depending on the mounting configuration it is also possible to reduce this number to two GAP elements without lowering the accuracy.

The bolt axial force can be expressed by adding the opposite force in the axial direction between the bolt and

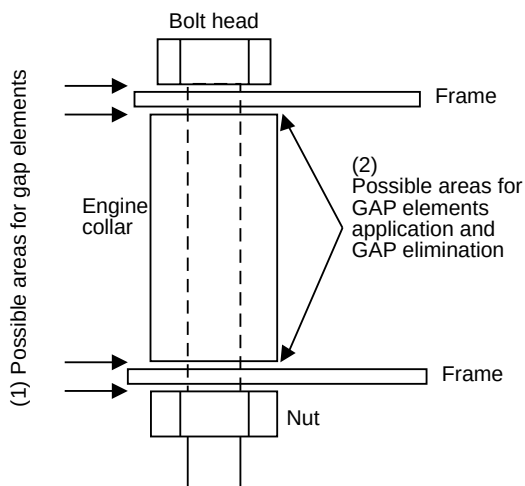


Fig. 15 Possible areas for using GAP-elements, Reduced number of GAP-areas

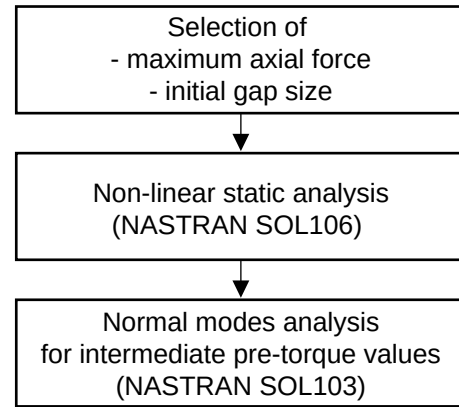


Fig. 16 Analysis flow

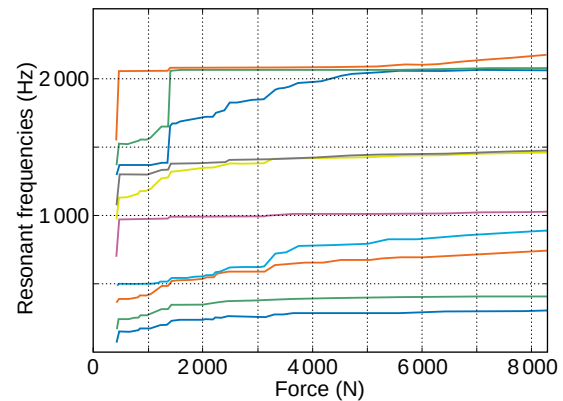


Fig. 17 Effect of bolt force on resonant frequencies

the nut. The effects of initial tolerance and axial force on the resonance frequency of each mode can be understood by specifying the maximum axial force, performing nonlinear static analysis using MSC-NASTRAN GAL elements, and then restarting normal mode analysis immediately thereafter. Fig. 16 shows this analysis flow, and Fig. 17 shows the analysis results. The horizontal axis is the bolt axial force, and the vertical axis is the resonance frequency. These results show that the gap closes gradually as the axial force is increased, and the resonance frequency of each mode increases together with the axial force. In addition, the rate of increase depends on the mode.

This same figure also shows that within the practical frequency range (0 to 330 Hz) for motorcycles, changes in the axial force around the normal bolt axial force of 8,000 N have little effect on the resonance frequency. Therefore, the GAP areas can also be replaced by high stiffness springs in those domains. However, if the frequency range is expanded to include higher frequencies, the effects of GAP analysis begin to appear. In this analysis it is extremely important to use highly detailed models near the bolt contact domains. Furthermore, though this method aids in the understanding of overall behavior, it is necessary to use an even finer mesh size in order to increase connector model accuracy. Applying this method for completed vehicles also achieves a balance with calculation resources.

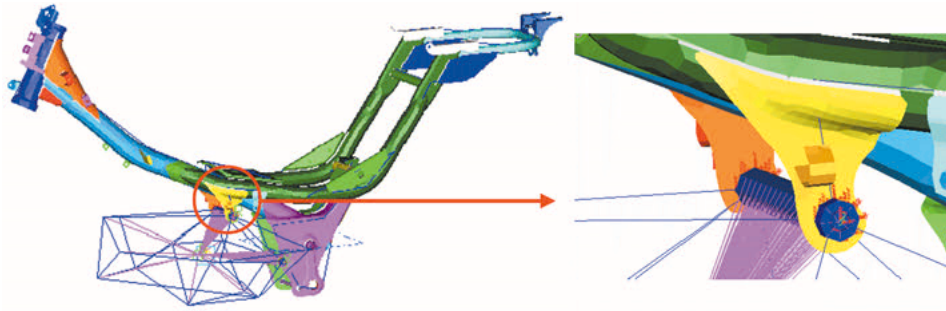


Fig. 18 Engine-frame FEM model (by new model)

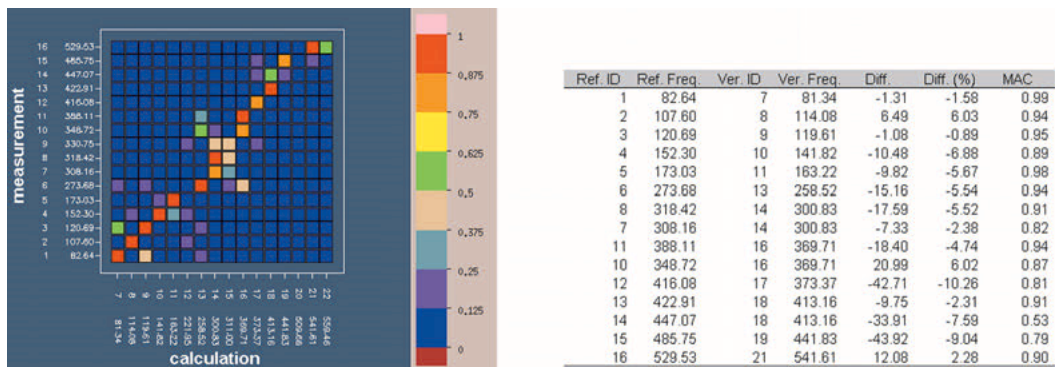


Fig. 19 MAC value of calculation and measurement results (by new model)

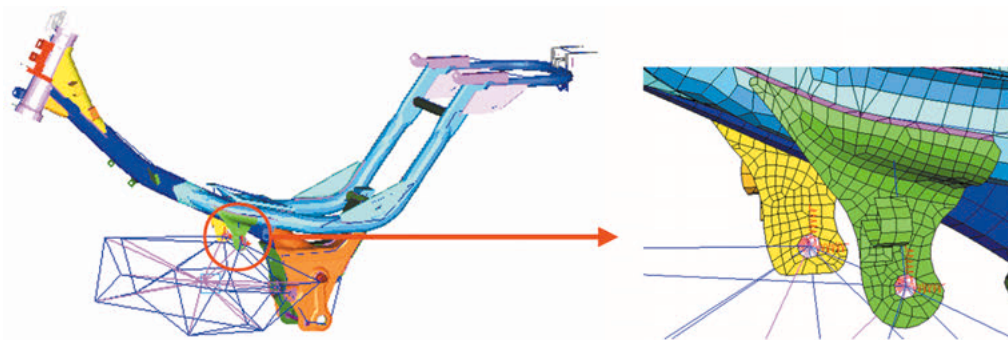


Fig. 20 Engine-frame FEM model (by conventional method)

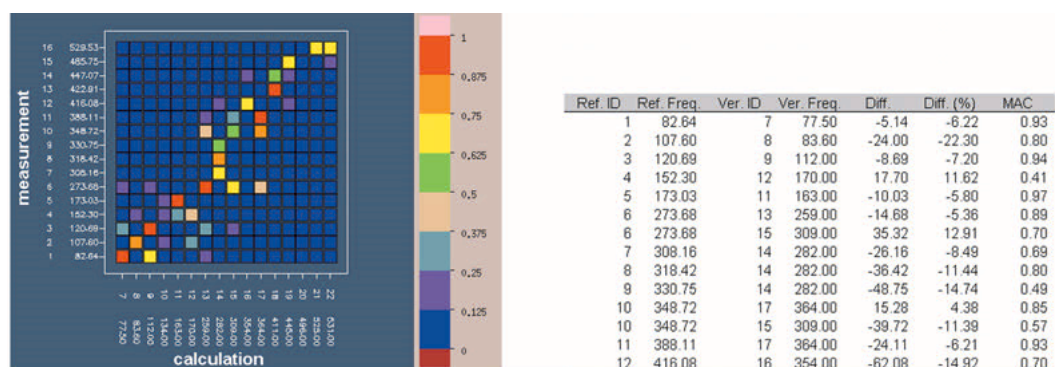


Fig. 21 MAC value of calculation and measurement results (by conventional method)

3.3 Engine-Frame Connection Model

Next, an actual engine-frame connection was investigated. Fig. 18 shows an FEM model of the engine-frame connection. In the practical frequency domain of 0 to 330 Hz, the engine is treated as a rigid body model, and the model was created using rigid body elements for all portions except for the collar portions of the connected engine. The engine collar portions and frame hanger portions were created in detail, then modeling was performed so that these portions contacted each other at the surfaces in the same manner as in the test piece, and they were connected by high stiffness springs. Fig. 19 shows the calculation and test MAC values for this engine-frame connection model. The MAC values are extremely high, and the frequency error is also within 5% for the most part, indicating the validity of this modeling method.

The model created using the conventional method and the corresponding calculation and test MAC values are shown in Figs. 20 and 21 for reference. However, the connector spring constant must be identified through tests. In addition, the rigid body connection points of the hangers which are non-connected as shown in Fig. 4 are vague, which causes the spring constant of the identified connector characteristics to change, so there are many undetermined elements. Furthermore, it was found that performing identification approximately five times such as in this study did not yield sufficient MAC values.

In the conventional method, connector characteristics were selected by experienced staff after thoroughly understanding the shape. However, this method provided highly accurate results simply by creating a detailed model of the connectors.

4. Conclusion

This paper presented the contents of connector characteristics that are important for the vibration prediction of motorcycles. The conclusions reached are as follows.

- (1) The connector characteristics between components that were formerly obtained through tests can be obtained analytically by utilizing GAP analysis.
- (2) The accuracy of local stiffness models for connectors can be improved by using RESVEC calculation. In addition, the number of flexible modes used can be reduced without causing a loss in accuracy.

References

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