

Establishment of a Method for Predicting Powertrain Vibration at Engine Mounting Points under Running Conditions

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ABSTRACT

It is well known that powertrain vibration characteristics are a critical factor influencing booming and other cabin noises. As vibration at the engine mounting points is one of the most influential of all the vibrations a precise analytical method for the prediction of vibration at the engine mounting points under running conditions was developed. This method uses simulation models that take into consideration the nonlinear characteristics of both main bearing's oil film and the engine mounts. This makes it possible to predict the vibration level at engine mounting points while preserving a practical level of accuracy. Application of this design method during the initial stages of basic powerplant structural investigations for powertrain design will enable the efficient evaluation of powertrain vibration characteristics.

1. Introduction

Consumers are looking for high quality with respect to automobile vibration and noise, and measures to reduce external noise are also required from an environmental viewpoint. As a result, there is an increased demand for quieter automobiles that feature both low vibration and low noise. This means that reducing powertrain vibration, which influences cabin noise, is becoming a requirement not just for luxury vehicles but for automobiles in all categories. Therefore, more accurate prediction technology must be established to respond quickly to market trends and to perform development within the limited automobile development period. Eigenvalue analysis using FE (Finite Element) models is currently the main tool for investigating powertrain structures. However, while eigenvalue analysis can predict the natural frequency of the overall powertrain, predicting powertrain vibration levels under running conditions is difficult. Therefore, this paper reports the establishment of a method for predicting vibration at engine mounting points under running conditions, and introduces cases where this method was applied to the evaluation of powertrain vibration characteristics.

2. Outline of Prediction Method

2.1 Analysis Procedure

Fig. 1 shows the analysis flow for this prediction method. Analysis was performed according to the procedure below.

- (1) Step 1: Calculation of powertrain vibration transfer characteristics
 - Creation of an FE model of the complete powertrain
- (2) Step 2: Calculation of cranktrain excitation force
 - Calculation of the excitation force through a FEM model of the cranktrain
- (3) Step 3: Calculation of engine mounting point vibration
 - Calculation of the vibration at engine mounting points under running conditions

2.2 Calculation of Powertrain Vibration Transfer Characteristics (Step 1)

2.2.1 Finite element model of the powertrain

The prediction accuracy of vibration at engine mounting points in the running condition depends greatly on the frequency response function (FRF) of each engine mounting

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bracket. Therefore, the accuracy of powertrain vibration transfer characteristics is an extremely important point. The FE model of the complete powertrain consists of all equipment, and is comprised of the cylinder block, cylinder head, gear box, transmission case, intake manifold, exhaust manifold, auxiliary devices, and engine mounting brackets. Fig. 2 shows the FE model of the complete powertrain.

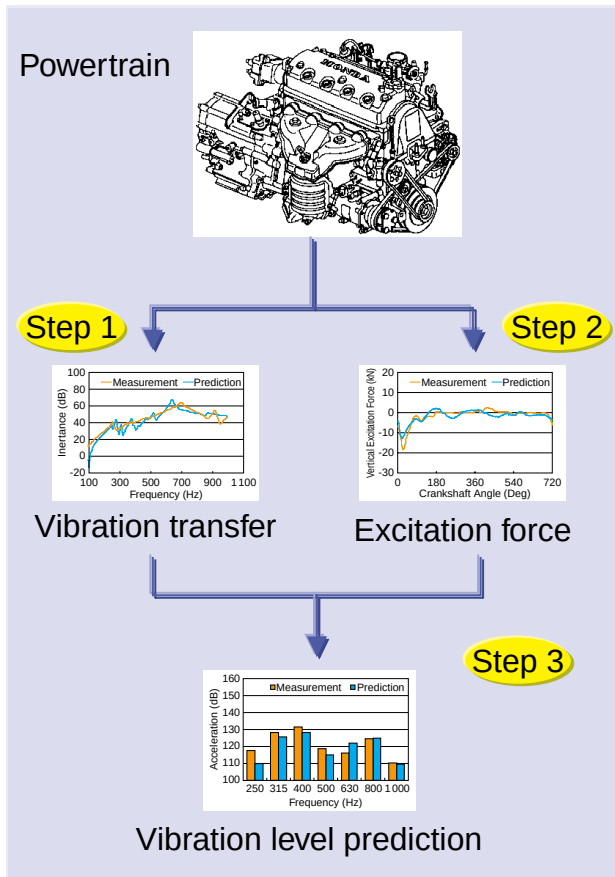


Fig. 1 Analysis procedure

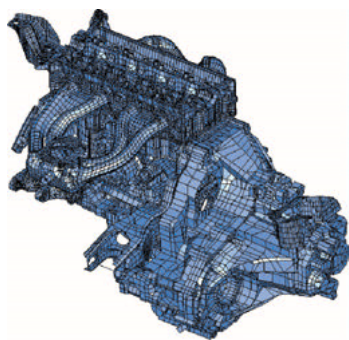


Fig. 2 FE model of powertrain

2.2.2 Calculation of Vibration Transfer Characteristics

As the correlation between the FRF of the initially created FE model of the complete powertrain and the measurement results was low, the FE model was updated.

When updating the FE model, the local resonance frequency and FRF of the engine mounting brackets were used as the target functions. As shown in Fig. 3, the calculated FRF exhibited excellent correlation with the measurement results, indicating that the FE model of the powertrain had more than sufficient accuracy for practical application.

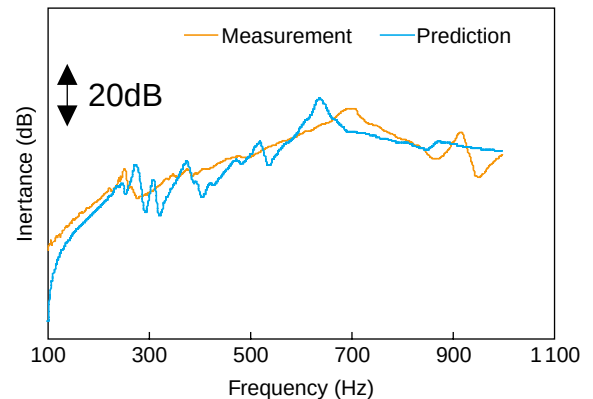


Fig. 3 Comparison of powertrain FRF

2.3 Calculation of Cranktrain Excitation Force (Step 2)

2.3.1 Finite element model of the cranktrain

The crankshaft behavior has a large effect on the cranktrain excitation force, so a highly accurate FE model is necessary. Fig. 4 shows the FE model of the cranktrain. This model is comprised of the crankshaft, pulleys, flywheel and clutch disc cover. An even more accurate FE model of the cranktrain was thought to be necessary, so the FE model of the cranktrain was updated with the natural frequency, FRF and modal assurance criterion (MAC) as target functions. Fig. 5 and Fig. 6 show a comparison of the calculated FRF and the MAC test results. The natural frequency shows an error of within ± 3 dB, and the MAC has an extremely high correlation of 0.7 or more, indicating that an excellent FE model was constructed.

2.3.2 Calculation of cranktrain excitation force

The gas force acting inside each combustion chamber is transmitted through the pistons and connecting rods to the crankshaft. The combined force of the piston side force generated between the pistons and cylinder liners and the inertia force acting on the pistons passes through the connecting rods as the excitation force to the powertrain, and acts on each crank pin in each firing order. The cranktrain excitation force is calculated using dynamic behavior analysis which takes into consideration the nonlinear oil film characteristics. Fig. 7 shows the calculated cranktrain excitation force on the 2nd main bearing. The conditions were an engine speed of 1600 rpm and wide open throttle (WOT). The calculated excitation force shows an excellent correlation with the measurement results.

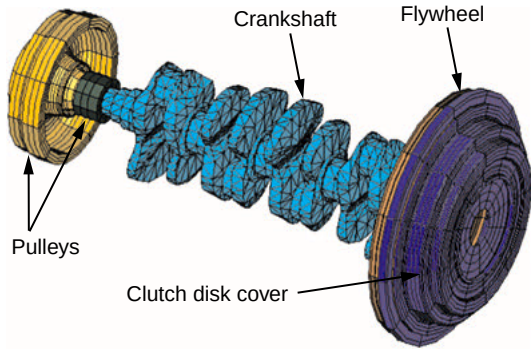


Fig. 4 FE model of cranktrain

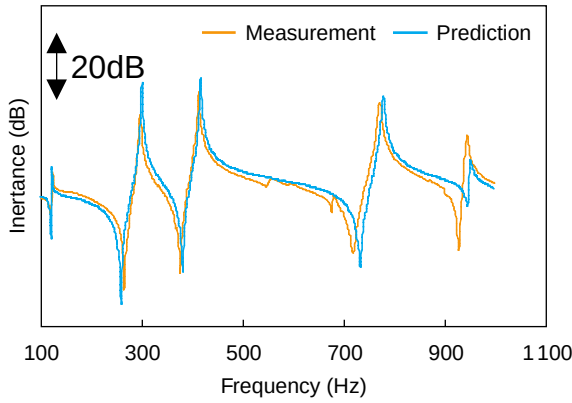


Fig. 5 FRF verification results for the cranktrain

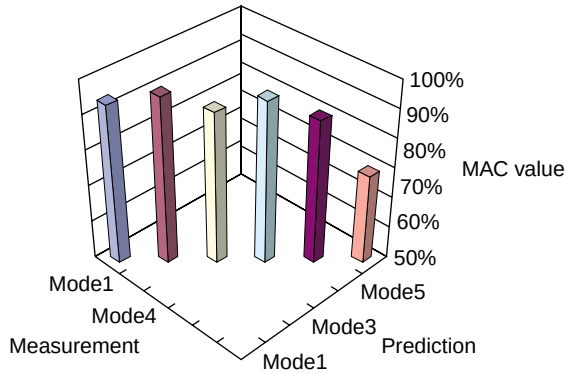


Fig. 6 MAC verification results for the cranktrain

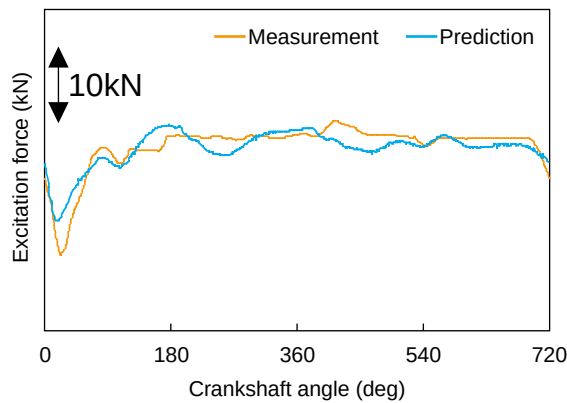


Fig. 7 Verification results for excitation force

2.4 Calculation of Engine Mounting Point Vibration (Step 3)

2.4.1 Engine mount model

Both rubber and sealed liquid type mounts are generally applied as engine mounts. However, the dynamic stiffness and attenuation, which depend on frequency and displacement, differ greatly for each mount type, so these become extremely important accuracy factors when modeling the powertrain under running conditions. Therefore, when modeling the engine mounts in this study, the accuracy was improved by changing from the conventional Kelvin-Voigt model to a mini oscillator model (MOM) that can express the nonlinear characteristics mentioned above. Fig. 8 shows both engine mount models. Parameter identification was performed to enable the strictest possible representation of measured mount characteristics, and the final engine mount characteristics used the same conditions as the set test bench conditions.

2.4.2 Calculation of engine mounting point vibration

The vibration level at the engine mounting points under running conditions was calculated in Step 1 and 2. Fig. 9 shows the vibration prediction results at engine mounting points under running conditions when the two engine mount model types mentioned above are applied. The conditions were an engine speed of 3200 rpm and wide open throttle (WOT). Changing from the Kelvin-Voigt model to MOM improved the vibration prediction accuracy at the mounting points by up to 15 dB for all frequencies up to 1 kHz. In addition, the error from the measurements was approximately ± 3 to 5 dB, which shows that the method presented in this paper of applying MOM to predict vibration at engine mounting points ensures more than sufficient accuracy for practical application.

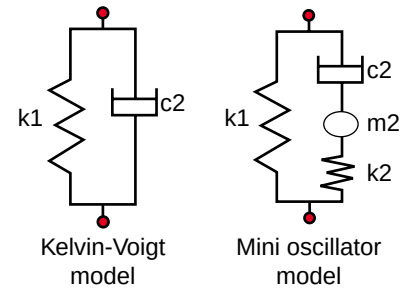


Fig. 8 Engine mount models

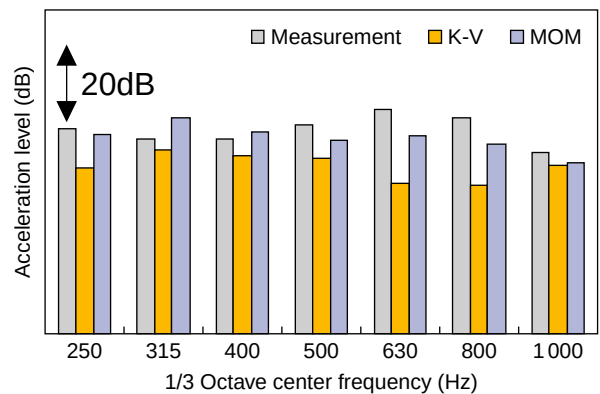


Fig. 9 Verification results for each mount model

3. Verification of Applicability to Actual Vehicles

The developed prediction method was applied to a transversally mounted in-line four cylinder engine in a front wheel drive layout and investigations were conducted to verify the feasibility of application to engine design.

3.1 Oil Pan Specification Changes

The prediction accuracy of the amount of the effects when the structure is changed was verified by changing the oil pan specifications. The test oil pan materials were steel and aluminum. Fig. 10 shows FE models of the aluminum and steel oil pans used for the investigations.



Fig. 10 FE model of the two oil pan specifications

Fig. 11 shows the prediction results for each aluminum oil pan specification, and Fig. 12 shows a comparison with the measurement results for each aluminum oil pan specification. In addition, Fig. 13 shows a comparison with measurements of the degree of the effects on vibration at engine mounting points by changing from steel to aluminum

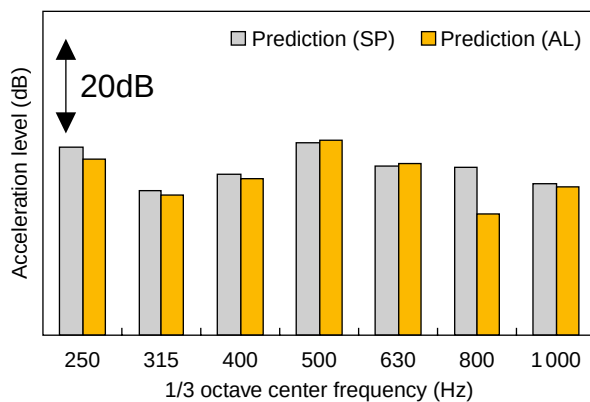


Fig. 11 Comparison of the oil pan specifications

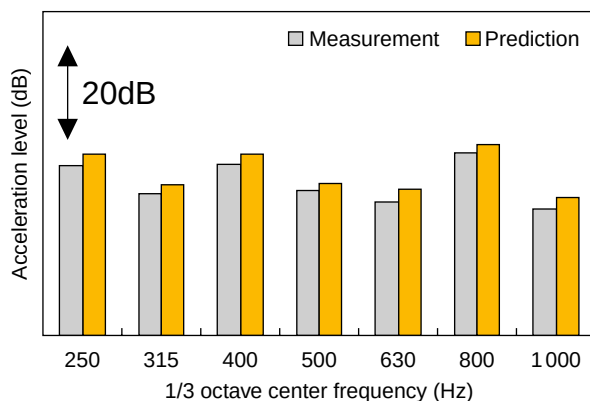


Fig. 12 Verification results for aluminum oil sump pan

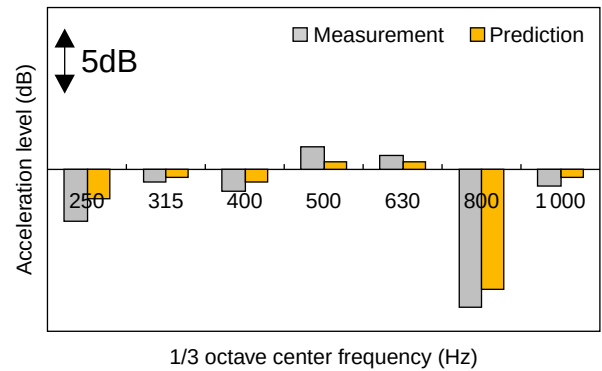


Fig. 13 Verification results for the degree of effects on vibration

oil pan specifications. The conditions were an engine speed of 4000 rpm and a load of 40 Nm. The developed method showed an error from the measurement results of approximately ± 3 dB, thus confirming prediction accuracy sufficient for practical application of both the vibration at the engine mounting points and the degree of the effects when the specifications are changed.

3.2 Cranktrain Specification Changes

Next, the prediction accuracy of the degree of the effects when the cranktrain specifications are changed was verified by changing the crankshaft specifications. The test crankshaft specifications were full-eight weight and semi-eight weight specifications. Fig. 14 shows the two crankshaft specifications used for the investigations.

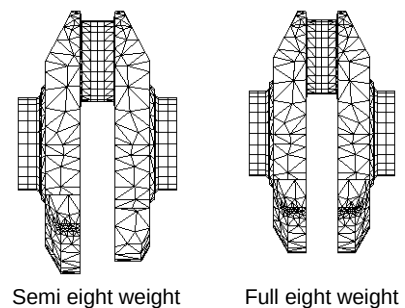


Fig. 14 Crankshaft specifications

Fig. 15 shows the results for full-eight weight specifications, and Fig. 16 shows the results for the degree of the effects on the vibration at the engine mounting points by changing from semi-eight weight to full-eight weight specifications. The conditions were an engine speed of 4000 rpm and a load of 40 Nm. The error from the measurement was approximately ± 3 dB, thus confirming that, like the changes to the oil pan specifications, the developed method is also accurate enough for practical application and which is capable of following changes to crankshaft specifications.

The results of investigating the feasibility of application in the above two examples confirmed that the developed method for predicting vibration at engine mounting points ensures a prediction accuracy appropriate for practical application. Therefore, the method was applied to actual engine design.

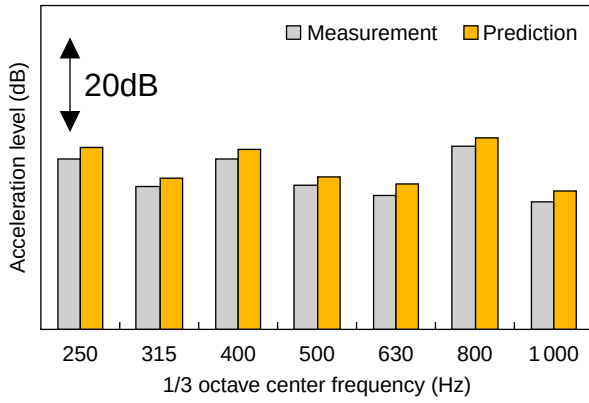


Fig. 15 Verification results for the two crankshaft specifications

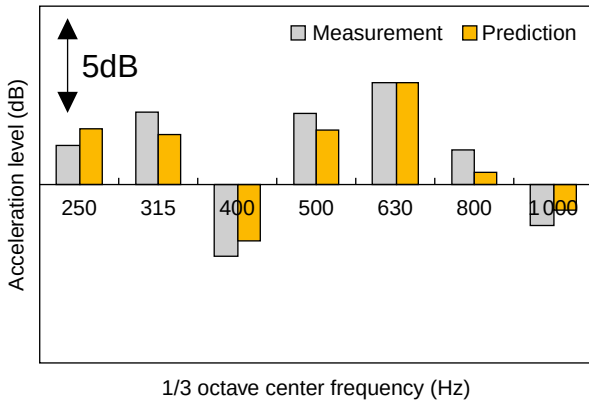


Fig. 16 Verification results for the degree of effects on vibration

open throttle (WOT). Based on these prediction results, SPEC-C was adopted as the most effective specification for reducing the total vibration within each frequency band of the vibration at the engine mounting points.

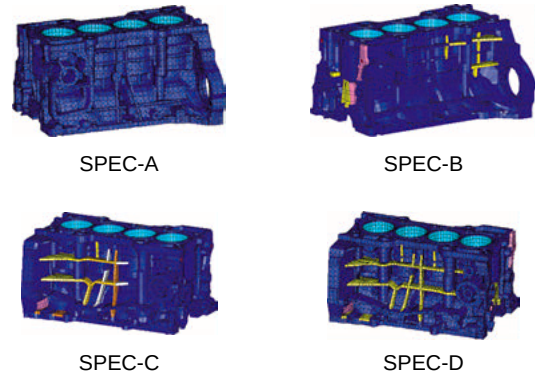


Fig. 17 Structural modification of the cylinder blocks

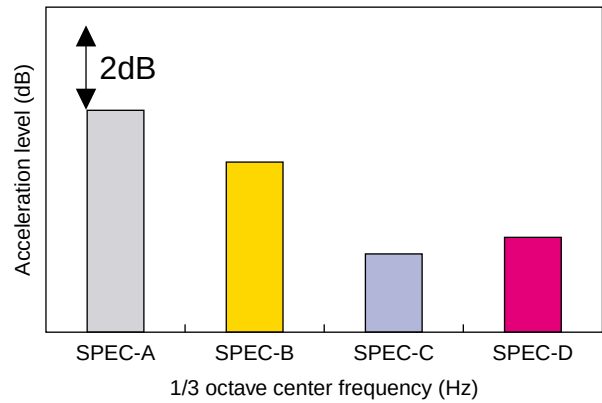


Fig. 18 Comparison of the total vibration level for each specification

4. Example of Application to an Actual Engine

An example of application of the developed method to investigations of the basic structure of an actual engine is introduced below using the example of structural modifications aimed at reducing the powertrain vibration of a transversally mounted in-line four cylinder engine in a front wheel drive layout.

4.1 Structural Modification of Cylinder Block

Ensuring cylinder block stiffness is important for reducing powertrain vibration. Of the various measures applied to ensure this stiffness, the vibration reducing effect depends largely on the position at which the ribs are attached, so the developed method was applied to investigate the rib positions. Fig. 17 shows the FE models of the cylinder block for each investigated specification. Investigations were carried out on a total of four types of specifications: SPEC-A, SPEC-B, SPEC-C and SPEC-D.

Fig. 18 shows the prediction results for each specification for the total within each frequency band (600 to 800 Hz) of the vibration at the engine mounting points. The conditions were an engine speed of 3000 rpm and wide

4.2 Gusset Stiffener Effects

In addition to the cylinder block stiffness, the stiffness of the powertrain couplings is also extremely important in reducing powertrain vibration. Therefore, the improvement of powertrain stiffness by gusset stiffeners was investigated by predicting the vibration at the engine mounting points. Fig. 19 shows the FE model of the gusset stiffener used in the investigation.



Fig. 19 FE model of a gusset stiffener

Fig. 20 shows the prediction results for the vibration at engine mounting points with gusset stiffeners added. The conditions were an engine speed of 3000 rpm and a load of 40 Nm. The effect of adding gusset stiffeners on reducing the vibration at the engine mounting points in the middle frequency range of engine acceleration noise was correctly predicted and subsequently investigations were promoted in design.

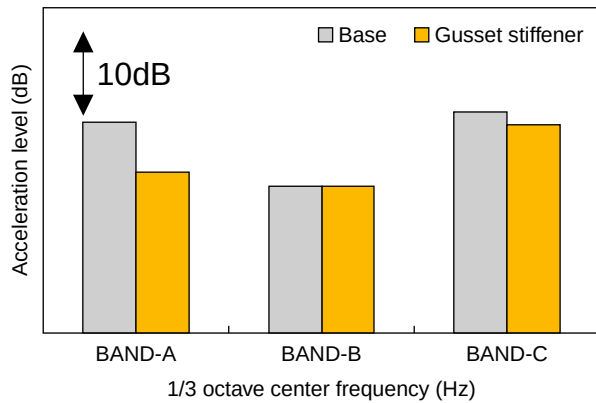


Fig. 20 Comparison of vibration level of a powerplant with and without a gusset stiffener

The above example shows that by applying the developed method to actual engine design, design guidelines can be obtained for efficiently reducing vibration and effective specification investigations can be carried out.

5. Conclusion

- (1) A method was established for predicting vibration at the engine mounting points under running conditions with accuracy appropriate for practical application. This was made possible by taking into account cranktrain behavior, oil film characteristics and nonlinear engine mount characteristics.
- (2) The developed prediction method was applied to a transversally mounted in-line four cylinder engine in a front wheel drive layout and investigated. The results confirmed that the developed method allows efficient evaluation of powertrain vibration characteristics, and is an effective method for investigating basic powertrain structures.

6. References

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