Vibration Analysis Modeling Techniques for Improvement of Deck Vibration Resistance Performance

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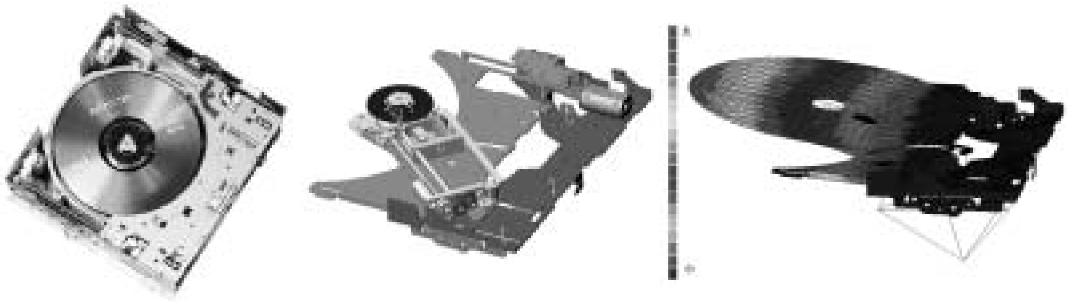

With the spread of 3-dimensional CAD (computer aided design) in recent years, product development utilizing simulation is becoming commonplace. In the field of stress (strength) analysis in particular, it has become a regular design tool, and at our company it has been incorporated into product design for parts such as brackets and decks, with the designers themselves verifying their designs via simulation as they work.

Our company specializes in in-vehicle article and its designs have to take full account of the vibration to which that article will be subjected by the vehicle. Traditionally this was dealt with by a process of trial-and-error using actual article, but in future vibration analysis techniques utilizing CAE (computer aided engineering) will be important for this purpose.

Especially for units such as CD decks which are composed of complex combinations of large numbers of mechanical parts, analysis that correlates with the actual article is not possible unless the combinations of the individual parts are accurately modeled including damping of vibrational forces.

This paper reports on sophisticated modeling techniques that are necessary to implement accurate vibration analysis.

1 Introduction

The spread of CAD (computer aided design) in recent years has brought a revolution in design methods utilizing CAE (computer aided engineering) that is still ongoing. Stress analysis in particular is establishing itself as the CAE process that is the first to be implemented in design.

At our company as elsewhere various types of CAE have become pervasive, especially for stress analysis. For vibration analysis however - a critical process for the company as an in-vehicle product manufacturer - the only CAE that has been brought into practical use thus far is a process for identifying resonance frequencies via analysis of the natural frequencies of parts in isolation. The rest of vibration analysis relies mostly on tests using actual article.

But as product mechanisms have become more complex it has become time-consuming to obtain results by actual-article tests alone, which makes it difficult to incorporate vibration resistant constructions into product design at an early stage. And even where CAE is utilized it is impossible to ascertain the vibration resistance performance of the product as a whole simply through analysis of the natural frequencies of the parts in isolation.

This means that vibration analysis must be conducted on products in the assembled state. However, for this it must be possible to define as adequate models the structural relations among the various assembled parts; otherwise the analysis itself will be liable to fail to converge, or else erroneous answers may be obtained. For total vibration analysis of a complex structure, techniques to provide adequate models of the joining, contacting and play, etc., among the parts are essential.

2 Problems in vibration analysis

The question of how the actual article is to be expressed in the analysis model is an important one for the application of vibration analysis to product development. Accordingly we present below an overview of the problems in turning article into models, and the remedies for such.

2.1 Selection of mesh elements to be used in the analysis

Selection of the analytic elements is important for implementing analysis. Elements are categorized into 1-dimensional (beams), 2-dimensional (shells) and 3-dimensional (solids). They are usually selected from among the elements that come as a standard component of the analysis software. However the software's primary tetrahedral elements, intended for application in 3-dimensional solid analysis, are not suited for practical use and therefore are generally not employed in analysis. In fact these elements are unsuitable for practical use not only in vibration analysis but also in stress analysis, for which similarly they ought not to be used. To make up for the absence of these elements, use is made of secondary tetrahedral elements having contact points at the mid-points between the apexes.

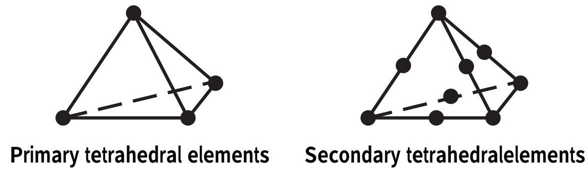

Fig.1 Elements of a tetrahedron

Fig. 2 shows the procedure for selection of elements and Fig. 3 the element configurations.

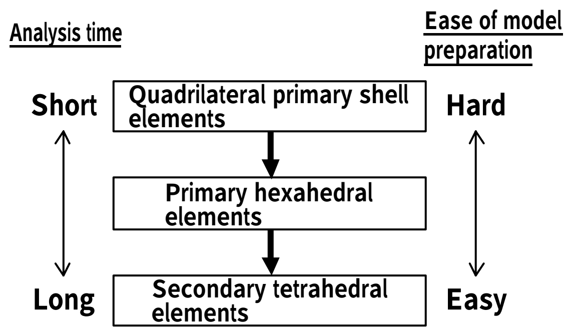

Fig.2 Procedure for element selection

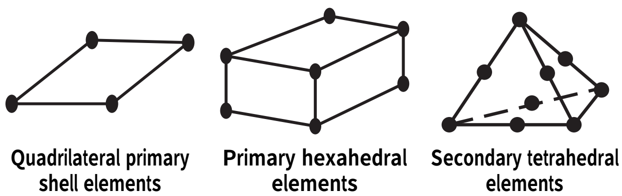

Fig.3 Element configurations

It is usual to use quadrilateral primary shell elements for thin structures such as plate metal parts, primary hexahedral elements for cubic-shaped solids, and secondary tetrahedral elements for other, complex solids. Since these element types all give more or less equivalent analysis accuracy as long as the element density is not overly low, the important thing is to choose the optimal elements for the shape modeled.

2.2 Constraints on analysis models

As mentioned in the Introduction, our company conducts identification of resonance frequencies in-house via analysis of natural frequencies at the isolated part level. Insofar as there are no joins with other mechanical components to be modeled, analysis of isolated parts is simple to perform. But such parts have portions that will join with other parts via adhesion or insertion fitting, etc.; these may superficially appear to be tightly fixed joins, but simply modeling them as completely stationary will often result in major discrepancy with the actual resonance frequencies. The reason for this is that strictly speaking the joining portions are able to move freely, albeit by a minute amount. Therefore even for analysis of parts in isolation it is necessary to take some special measures in the modeling that take account of the assembled state, such as adding to the analysis other parts that will influence the results, or assuring degrees of freedom by connecting the joining portions with spring elements or similar.

2.3 Modeling of assemblies

To obtain correct results in vibration analysis one has to model items in the assembled state. In the actual products however mechanical parts are not fixed together in an immobile condition by screws or welding, etc; rather they are configured to have play or similar movement freedom in order to assure contacting or sliding. But it is extremely difficult to represent contacting, play and so forth in a simple manner with linear analysis software, which is devoid of any concept of contacting.

Accordingly we opt to categorize into conditional sets the various configurations in which parts are held together, and their behaviors, and to set boundary conditions for each set when implementing modeling.

Specifically, for portions that combine with a low-rigidity part we substitute combinations of spring elements with 6 degrees of freedom (movement along and rotation about the X, Y and Z axes), while for parts with play we employ MPC (multi point constraints) to model the directions of force transmission between the parts, by substituting elements with the above-mentioned 6 degrees of freedom. The behavior and resonance frequencies are then derived by categorizing into conditional sets and expressing the changes in the fulcrums that occur when parts execute play motion. Where portions come into contact and have no behavior however they can not be modeled, and this factor must be taken into account when the analysis results are evaluated. Additionally the physical properties of the parts materials must be represented by inputting a structural damping coefficient for each material, so as to give an analysis approximating more closely to the actual article.

2.4 Analysis methods reproducing actual-article vibration test conditions

Evaluation of vibration resistance performance in product development has traditionally employed tests using vibration testing apparatus, and in order to apply analysis to such evaluation it was necessary to reproduce the conditions of such apparatus.

To solve this problem we employ the large mass method. Conducting response analysis using this method makes it possible to reproduce the conditions of the vibration tests and thus to have the analysis approximate even more closely to real situations.

The large mass method involves assigning to mass elements (representing connected parts) masses much greater than that of the object analyzed, and causing the mass elements to vibrate with particular acceleration rates and frequencies. When this is done the analysis model connected to the mass elements executes the same vibration as the mass elements because the latter have greater inertia. In this way vibration almost exactly like that in the vibration test can be reproduced.

2.5 Analysis time

Vibration analysis consists of identifying the resonance frequencies via natural frequency analysis, then conducting response analysis using the large mass method described in 2.4 above to check for problems in the vibration resistance performance under conditions simulating actual article tests. Response analysis involves performing calculations for each required resonance frequency and thus may take large amounts of time. This makes it necessary to simplify the analysis model by eliminating unnecessary portions so that elements are not multiplied to no purpose. The graph in Fig.4 plots the relation between the number of elements and the analysis time.

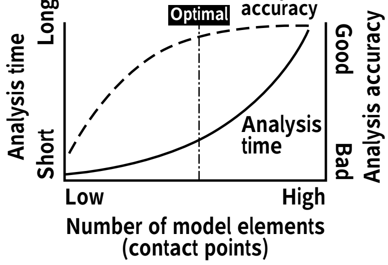

Fig.4 Graph plotting number of elements versus analysis time

3 Modeling techniques

The discussion of modeling techniques below describes procedures and methods for assembly modeling, taking as illustrative example the CD changer deck shown in Fig. 5.

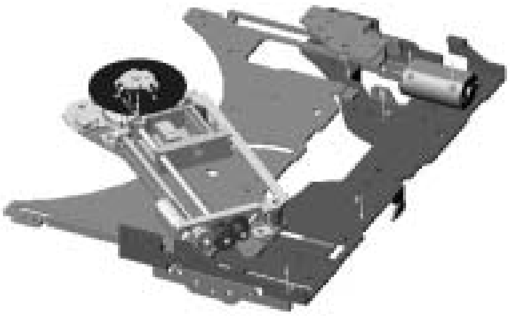

Fig.5 3-D CAD model

3.1 Consideration of parts for detailed modeling

Since modeling in unnecessary detail will only result in too much time being spent on the vibration analysis, it is necessary to consider and determine which parts should be modeled in detail. In the CD changer deck the lens portion of the player unit (PU), which is the data reading unit, is held in position by thin wires as shown in Fig. 6. It was therefore considered likely to be the part with greatest influence on the vibration resistance and was modeled as faithfully as possible. The procedure for the modeling of the PU is given below.

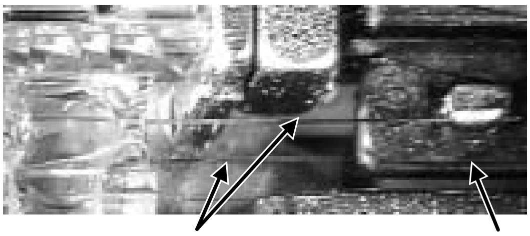

Fig.6 Suspension of actual article

①implification via 3-dimensional CAD model

Simplification of the PU was implemented using a 3-dimensional CAD model, simultaneously coordinating the gravity center positions. The simplified model is shown in Fig. 7.

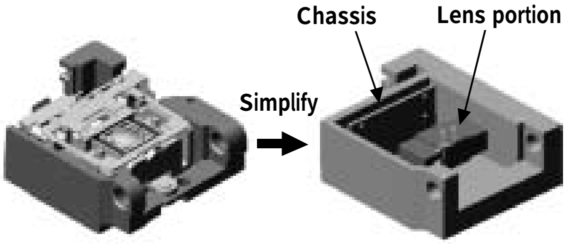

Fig.7 Simplified 3-D CAD model of PU

②Meshing of solid model

A 3-dimensional CAD solid model was fed into the analysis software, which generated a mesh model. Here the shape and the ease of mesh preparation permitted having the analysis software's auto generation functions build the model using secondary tetrahedral elements. The resulting mesh model is shown in Fig. 8.

Fig.8 Mesh model

③Modeling of suspension portion

As shown in Fig. 9, in the actual article the lens portion and chassis are connected by suspension wires 0.068 mm in diameter, while a gel damper is deployed in the vicinity of the chassis' fitting portion.

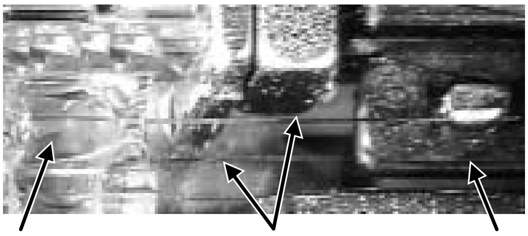

Fig.9 Suspension of actual article

As shown in Fig. 10, the suspension portion was modeled with beam elements in order to improve (maintain) the analysis accuracy and to keep down the number of elements. Since the gel damper produced damping, the portion where it is located was modeled separately from the others, and the damping that it produces was incorporated in the model via a structural damping coefficient that was used as one of the input parameters representing the material physical properties.

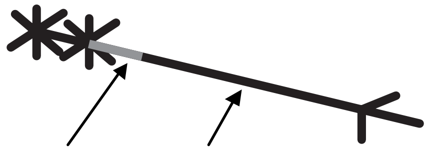

Fig.10 Suspension model

Fig. 11 shows the analysis model of the PU that was obtained by combining the PU mesh model in Fig. 8 with the suspension model in Fig. 10.

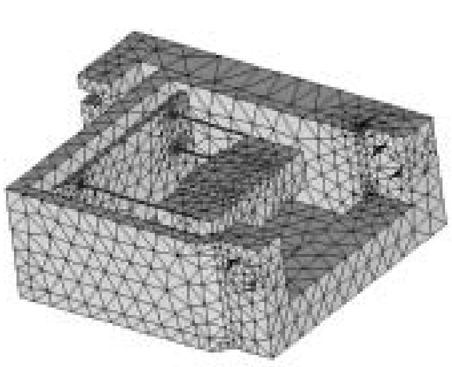

Fig.11 Mesh model of PU

3.2 Simplification of parts with little influence on analysis

As mentioned in 2.5, in order to keep the analysis time short it is imperative to reduce the number of elements in the analysis model. However, associated parts are liable to have large influences on the results of vibration analysis, and for this reason specific methods must be employed to insure that the parts eliminated in order to decrease the elements are not such as would influence the results. Several such methods are described below.

①Substitution of mass elements for parts

This is a method that substitutes mass elements for the parts targeted in simplification, so that the parts' weight will remain represented in the model and thus the graphic elements for them may be eliminated. The modeling procedure is as follows:

a. At the level of a group of several pertinent parts, the part weights and gravity center positions are identified using 3-dimensional CAD.

b. The parts to be eliminated are removed from the analysis model and replaced with mass elements that are added at the gravity center positions identified by 3-dimensional CAD and joined to the model via rigid beam elements, which are massless. Fig. 12 provides an example of a mass element model.

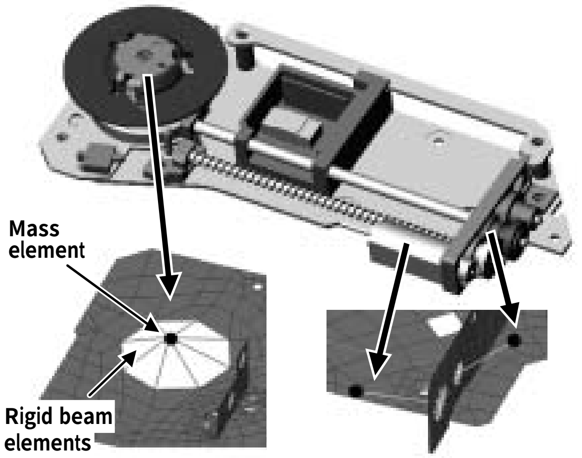

Fig.12 Mass element model

②Substitution of beam elements for axle parts

Because in many cases axles and similar parts themselves provide rigidity in a product, simplifying a model by substituting mass elements for them will with high probability lower the analysis accuracy. Accordingly this method models axles by substituting 1-dimensional beam elements for them in order to achieve reduction of the number of elements. Modeling axles as solids would give greater fidelity to the analysis model but would entail an increase in the number of elements. Fig. 13 shows an example of a beam element model.

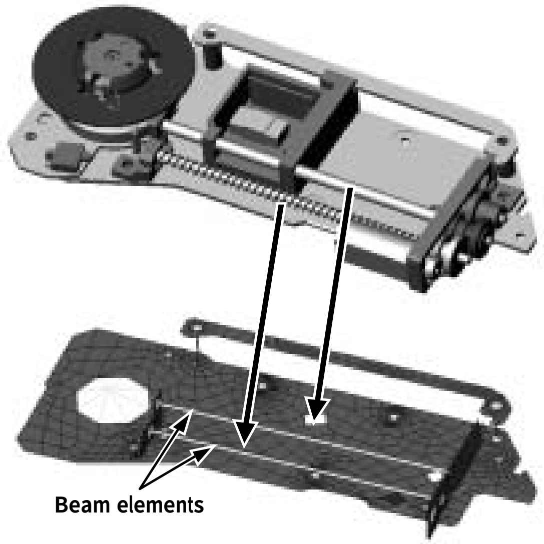

Fig.13 Axle beam element model

③Use of shell elements for plate metal parts

Many of the CD changer deck's parts consist of items made of pressed metal plate. This method uses shell elements to represent such parts, which is generally held to be efficient for vibration analysis of plate metal in that it yields satisfactory analysis accuracy with a reduced number of elements. The problem in analysis using shell elements is whether neutral planes in the plate thickness direction can be extracted from the solid model and used to implement modeling via surfaces. With the recent progress in 3-dimensional CAD however, even mid-range CAD is able to extract neutral planes with ease, either automatically or semi-automatically, so that the problems of modeling using neutral plane surfaces are steadily being resolved. The procedure for such modeling is given below.

a. The shell model is prepared using 3-dimensional CAD. Fig. 14 shows a neutral plane surface model.

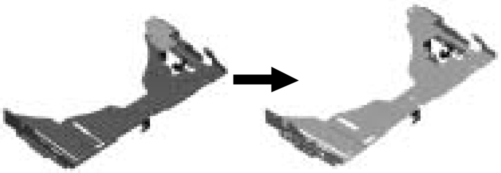

Fig.14 Neutral plane surface model

b. The surface model is fed into the analysis software, then the other parts that are screwed to the modeled part are joined with massless rigid beams at the screw hole positions, thus effecting joining of the parts.

c. The surface model is assigned plate thicknesses and material physical property conditions, then the analysis software's auto generation functions are used to construct meshes from quadrilateral primary shell elements. Fig. 15 shows the resulting shell element split model.

Fig.15 Shell element split model

④Methods for representing joins between parts

There exist various types of joins between the CD changer deck's parts, including screwed joins, sliding arrangements, sliding arrangements with play, and direct contacting. For each type the optional method of join representation must be selected. The various join representation methods utilized are presented below.

④-1.Method for screwed joins

Where parts are joined together by screws, the join is represented via rigid beam elements at the location of the screw holes in the manner shown in Fig. 16. This join representation method can be applied in both solid and shell models.

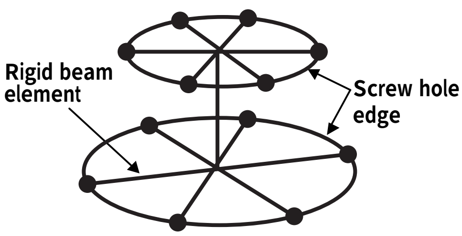

Fig.16 Joining of rigid beam elements

④-2. Method for sliding along axle

To represent a join that involves sliding along an axle such as shown in Fig. 17, MPC joining can be employed. This is done by applying constraints to 2 of the degrees of freedom (6 freedom degrees: movement along and rotation about the X, Y and Z axes), namely movement along and rotation about a particular axis, so that the same behavior as an axle will by necessity result. The modeling is then refined by adding constraints on the degrees of freedom or adding spring or other elements, etc., as required by the conditions of the model.

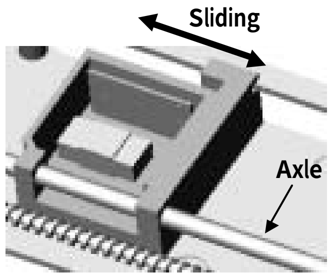

Fig.17 Sliding along axle

④-3.Method for sliding with play

Fig. 18 shows a sliding join between parts that includes play. When the join's sliding plate is deflected upward the fulcrum is on the left side, and when it is deflected downward the fulcrum is on the right side. In modeling, a join representation corresponding to this situation will be prepared by using MPC joining to combine degrees of freedom, in a manner similar to that above.

In the analysis model the upward motion behavior of the plate can be reproduced by constraining the X, Y and Z axis degrees of freedom at the point labeled "Left" in Fig. 18 and constraining the X and Y axis degrees of freedom at the point labeled "Right."

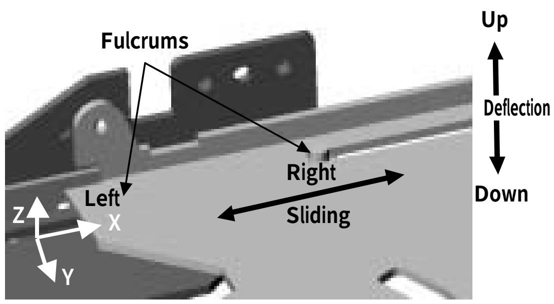

Fig.18 Sliding with play

Thus, to represent sliding joins with play it will be necessary to assign conditions according to each situation.

⑤Modeling via large mass method

The large mass method is used for modeling the situation in a vibration test. It involves assigning to the mass elements (described in ①above) drastically large masses, so that their inertia is far greater than that of the object analyzed. The mass elements are connected to the object analyzed in the model, and when they are made to vibrate at a particular frequency the object imitates their vibration owing to their greater inertia. This creates in the model a situation very closely resembling excitation in vibration testing apparatus.

In large mass element modeling, mass elements are created at desired locations, assigned sufficiently large masses, and joined to the analysis model by massless rigid beam elements as illustrated by Fig. 19. When analysis is conducted, response analysis loads are applied to the mass elements.

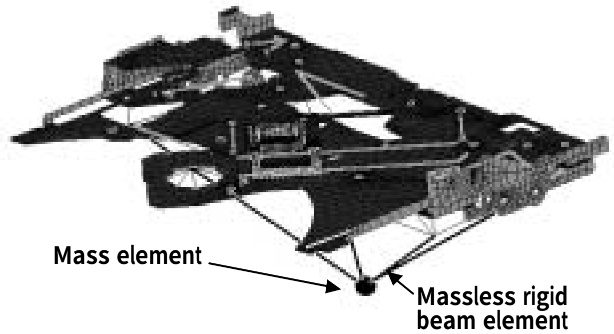

Fig.19 Model for analysis using large mass method

The modeling techniques in this section have been presented using the CD changer deck as an example, but these techniques can be applied to a wide range of items besides this deck by varying their combinations, and the boundary conditions that are commonly used for analysis, to suit the particular model.

4 Case study of application to CD changer deck

4.1. Object analyzed

The component of the CD changer deck that was subjected to vibration analysis was the PU, which is composed of 3 chassis. When a CD is played, one of the chassis slides so that the unit is in the playback state shown in Fig. 20.

4.2 Purpose of analysis

As can be seen from Fig. 21, with certain vehicles the deck's resistance to the vibration input G from the vehicle was low for vibration in the region of 30 Hz. Since this low resistance resulted in skipping during playback, the purpose of the analysis was to derive improvement measures for vibration resistance performance in the 30 Hz region.

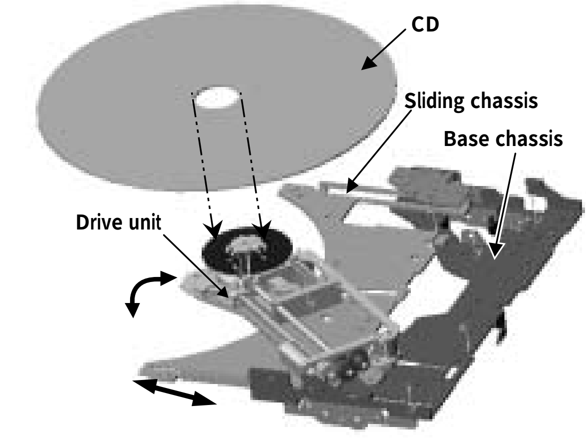

Fig.20 3-D CAD model

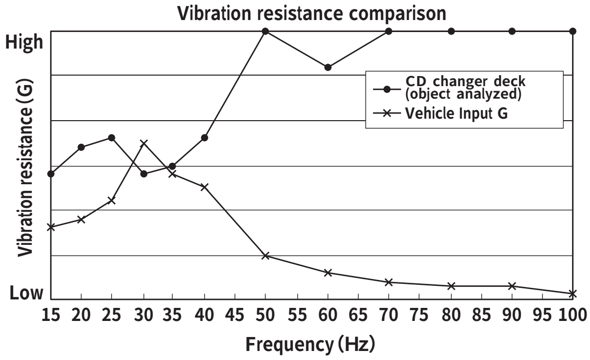

Fig.21 Vibration resistance performance graph

4.3 Comparison of analysis results with actual article

The reliability of the analysis results was verified by conducting analysis of the actual article for the purpose of comparison.

The analysis of the article took the form of response analysis using the large mass method described in 3-④. The skipping occurred when the PU became unable to read the CD data due to resonance of its chassis at a given vibration frequency. Accordingly the resonance frequencies were analyzed by examining the differences between given vibration amplitudes and the amplitudes of chassis vibration. Since the chassis amplitudes obtained were relative rather than absolute values it was assumed that the larger the amplitude, the more liable resonance was to occur.

Fig. 22 shows the displacement measurement positions, while Fig. 23 gives the analysis results in the form of a graph plotting the differences in amplitude at the excitation and measurement points.

It can be seen from Fig. 23 that the PU vibrated due to resonance of the drive unit. Hence it could be assumed that the PU was not the main cause of the skipping, and so the scope of the comparison was nar-

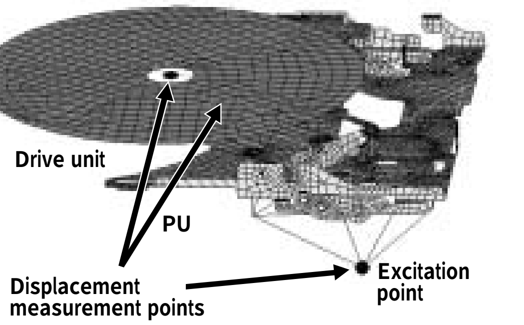

Fig.22 Displacement measurement positions

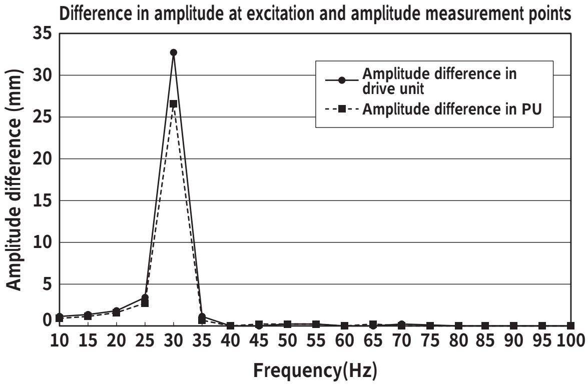

Fig.23 Displacement difference graph

rowed down to the behavior of the drive unit. Since the resonance frequency of the turntable coincided with the frequency at which the fall in the vibration resistance occurred, the analysis was deemed to be reliable.

4.4 Improvement measures and results

Once the reliability of the analysis results had been verified, we turned to deliberation of improvement measures based on the results.

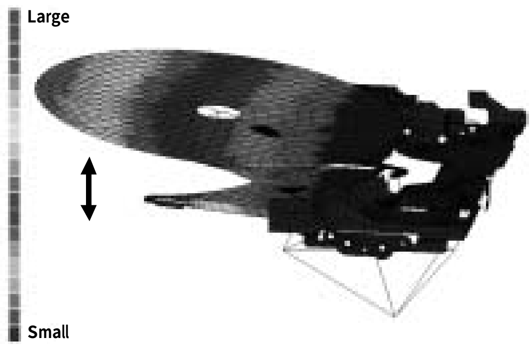

Fig.24 Behavior at 30 Hz

It can be seen from Fig. 24 that during resonance at 30 Hz the sliding chassis, drive unit and CD (labeled in Fig. 20) vibrated vertically, the fulcrum being the point of contacting between the sliding and base chassis. Such behavior can be verified as an animation run on the analysis software.

Accordingly the improvement measure decided on was to join the sliding chassis and base chassis, thereby eliminating the vibration. Analysis was then conducted using the improved model; the analysis results are shown in Fig. 25.

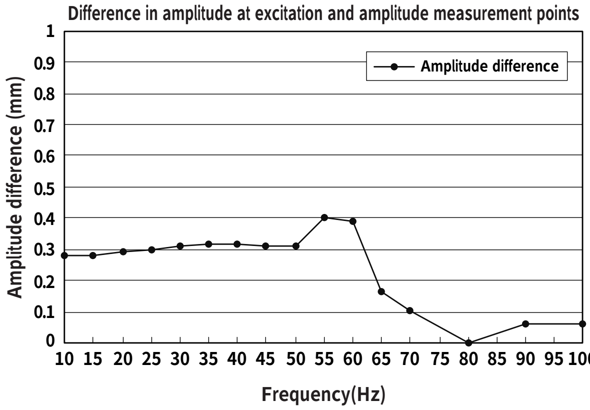

Fig.25 Displacement difference graph

4.5 Framing and verification of proposals for vibration resistant construction

To assure vibration resistance it was necessary to suppress the vertical motion behavior of the sliding and base chassis while securing their sliding performance. Figs. 26 and 27 show construction proposals for assuring vibration resistance via the sliding mechanism.

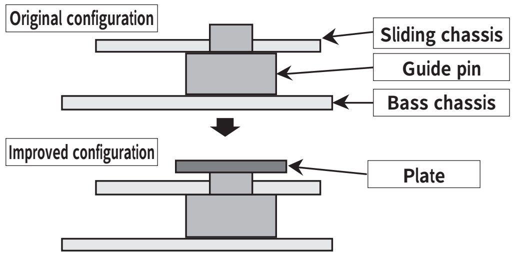

Fig.26 Vibration resistant construction proposal 1

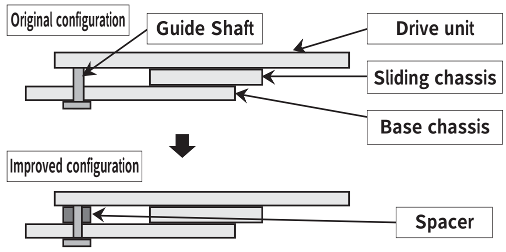

Fig.27 Vibration resistant construction proposal 2

Formerly the sliding chassis had been mounted from above and held down by a spring; in the improved construction use was made of a plate and spacer that suppress motion, thus controlling the vertical motion while preserving the sliding performance.

Testing was conducted to confirm the vibration resistance of the improved construction; the test results are given in Fig. 28.

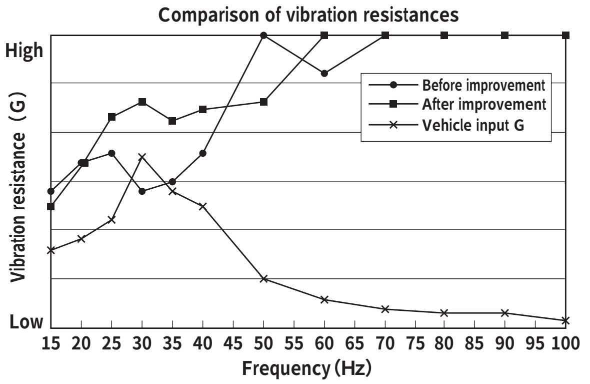

Fig.28 Vibration resistance performance graph for item with improved vibration resistance construction

4.6 Efficacy of employment of vibration analysis

Discovery of problems used to be tardy with the old development procedure because evaluation was carried out after design had been completed. And since improvement measures were derived via testing on actual article, it took time for satisfactory improvements to be completed. But now that vibration analysis is employed in the design stage it is possible to have problem-free construction from design onward, as Fig. 29 shows.

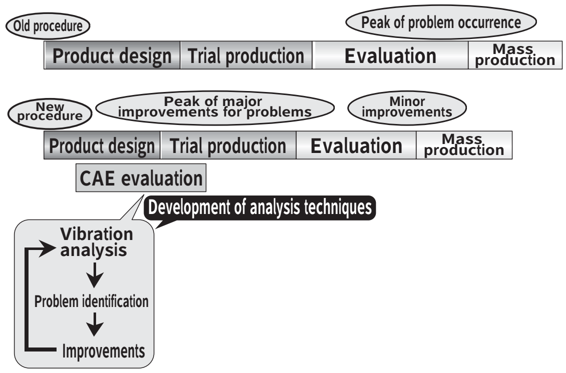

Fig.29 Efficacy of employment of vibration analysis

It also used to take some time to identify problem causes via actual-article tests. Vibration analysis however permits intuiting of problem causes and consequently should have the additional effect of allowing improvements to be found in shorter times.

Thus, employment of vibration analysis can be expected to have the effects of shortening the development period, improving product quality, and cutting costs.

5 Results

1. Modeling techniques have been established that make possible vibration analysis of products with contacting and play among multiple parts, which was previously problematic.
2. It is now possible to improve products' vibration resistance performance by implementing vibration resistance improvement measures based on the results of vibration analysis.

6 Conclusion

While the presentation of modeling techniques in this paper has centered on the CD changer deck, these are techniques that can be widely applied to various other products. Accordingly we will be actively utilizing vibration analysis for our product design in the future.

The future will also see us proceeding with the task of making vibration analysis into a design tool. Specifically we will be examining ways of turning the techniques into internal intellectual property (via IP database) so that vibration analysis can be utilized as a design tool similarly to other CAE processes. For achieving product quality improvement and cost cutting within short development periods, CAE will be an important, indeed vital tool.

Vibration analysis using the modeling techniques presented in this paper through the example of the CD deck vibration resistance improvement constitutes a highly important method that we believe will prove extremely effective for improving the vibration resistance performance of products of this type.

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Profiles of Writers

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