

Advancement in Vehicle Development Using the Auto Transfer Path Analysis

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ABSTRACT

This paper presents the most recent advancement in the vehicle development process using the one-step or auto Transfer Path Analysis (TPA) in conjunction with the superelement, component mode synthesis, and automated multi-level substructuring techniques. The goal is to identify the possible ways of energy transfer from the various sources of excitation through numerous interfaces to given target locations. The full vehicle model, consists of superelements, has been validated with the detailed system model for all loadcases. The forces/loads can be from rotating components, powertrain, transfer case, chain drives, pumps, prop-shaft, differential, tire-wheel unbalance, road input, etc., and the receiver can be at driver/passenger ears, steering column/wheel, seats, etc. The traditional TPA involves two solver runs, and can be fairly complex to setup in order to ensure that the results from the two runs are consistent with subcases properly labeled as input to the TPA utility. However, auto TPA allows necessary data needed for the TPA analysis to be requested in a single frequency response analysis run. The TPA breaks down the total response to partial contributions from interface points under operation loads. Partial contributions to total response are then computed by multiplying transfer function with the force transmitted through each interface location. By comparing the results of two-step TPA with auto TPA, the effectiveness and efficiency of the auto TPA have been demonstrated for three load cases: a) powertrain excitation for cruising acceleration to engine torque (A/T) and noise to engine torque (P/T), b) propshaft imbalance, and c) rough road excitation, which is based on the power spectral density (PSD) function.

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INTRODUCTION

In the fiercely competitive world of today's automotive industry, Computer Aided Engineering (CAE) is playing a more and more important role in shortening the design cycle time, minimizing costs and improving the product quality. A recent advancement in technological changes has made it possible to have affordable and powerful workstations, desktops, and laptops with very large memory and disc space to run industrial finite element analysis with hundreds of millions of degrees-of-freedom (DOF) models. The CAE tools have become indispensable for the analysis, design, and optimization of automobiles for quality, reliability, durability, safety, performance, and fuel economy. Finite element method [1-2] and Boundary Element Method [3] are essential to support the virtual product development in order to bring the product to the market faster. The present author Mallikarjuna Bennur and his co-researchers published a series of papers on various higher-order theories in conjunction with the superparametric

and isoparametric elements [4, 5, 6, 7, 8, 9], and the application of superelement [10, 11] method with the hybrid technique based on Finite Element Method (FEM) and experimental test data [12, 13, 14].

The Two-step TPA[11, 15, 16, 17] either using CAE or laboratory experimental method is well established. This paper presents the one step TPA or auto TPA in conjunction with the superelement method, Component Mode Synthesis (CMS), and Automated Multi-Level Substructuring (AMLS) techniques [10, 15]. The principle of TPA is shown schematically in Figure-1. The FRF model description [15, 16] of the vibro-acoustical system which relates a loading or excitation vector $\{S(\omega)\}$ to the target response vector $\{t(\omega)\}$ by an FRF matrix $[H(\omega)]$ as

$$\{t(\omega)\} = [H(\omega)]\{s(\omega)\}$$



where s (or the source) can be a force or a volume velocity depending on whether you are considering a structural or airborne path, respectively. The target response t can be sound pressure P or displacement/velocity/acceleration X . Mobility is the velocity resulting from the application of a unit force and is measured basically the same way as an impact frequency response function, but acceleration is converted to velocity. It is a measure of dynamic stiffness which varies with frequency and can be used to calculate the transmissibility.

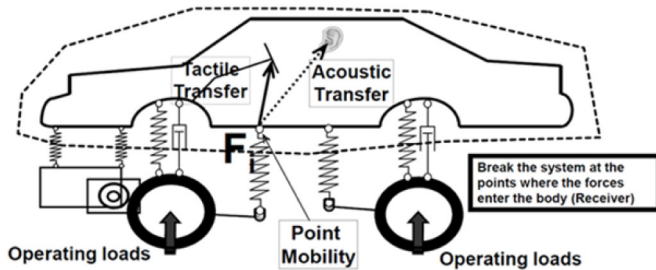


Figure-1. Principle of Transfer Path Analysis

Mobility optimization is developed to achieve the design goals in an efficient manner. This approach integrates FEA with mathematical optimization algorithms to form a complete structural design system. It enables engineers to gain insight into the areas of the structure that can be changed to have the greatest impact on meeting the mobility target. The various targets and sources are considered as two different subsystems and are connected to each other by means of a number of rigid or soft mount connections with bolts or mounts/bushings, forming the so called transfer paths. If the system is composed of N transfer paths, then the total target response can be written as the sum of the partial responses from the individual paths as

$$P_t(\omega) = \sum_{i=1}^N \frac{P_i(\omega)}{F_i(\omega)} \cdot R_i(\omega) \quad (2a)$$

$$X_t(\omega) = \sum_{i=1}^N \frac{X_i(\omega)}{F_i(\omega)} \cdot R_i(\omega) \quad (2b)$$

where $R_i(\omega)$ is the operational force at transfer path i , $X_i(\omega)$ is the structural displacement or velocity or acceleration for a unit force $F_i(\omega)$ at the interface path i , and $P_i(\omega)$ is the fluid sound pressure level at human ear in the vehicle cabin.

THEORETICAL FORMULATION

The following sections will briefly describe the various techniques/methods employed in the new auto-TPA.

One-Step or Auto Transfer Path Analysis

A simple representation of source-path-receiver concept is shown in Figure-2. There are multiple interface connection points among various components in the full vehicle system. For example, in the vehicle model presented here, there are six interface points (bolts) between the front cradle and body, two points (mounts) between the front cradle and engine, four points (bolts) between the transmission cross member and body, two points (mounts) between the transmission and body, four points (mounts) between the rear cradle and body, one point between the center bearing and body, three points (mounts) between the rear differential module and rear cradle, two points (mounts) between the body and front shock and two at rear shock to body, fourteen points in the front suspension and twelve points between the rear suspension and rear cradle, and four points between the exhaust and rear cradle, and two points from rear exhaust to body.

The frequency response function (FRF) or also called transfer function is the ratio of output to input. The drive-point transfer function is defined as the displacement or velocity or acceleration response at a point on the structure resulting from a unit force at the same point. Cross-point transfer function is a computed displacement or velocity or acceleration at one point resulting from a unit force at another point on the structure. Mobility, ratio of velocity to force, is a measure of dynamic stiffness which varies with frequency and can be used to calculate the transmissibility. Structural acoustic sensitivity is defined as the interior sound pressure due to unit force applied to the structure. This quantity is also referred to as the noise transfer function or P/F (pressure/force). The vibration or structure borne interior noise in the full vehicle system can be due to road inputs or powertrain or driveline disturbances or any excitation transmitted to the body through the front and rear suspensions, chassis frames, RDM, exhaust, etc. As described above, there are multiple interface connection points in the system, which contribute to the response point located inside the control volume. If a unit load is applied at one point degree of freedom of the control volume, it would excite the response point with certain amount; so called transfer function (H_{ci}). The reaction forces at the connection points (R_{ci}) would come from the analysis of the whole system. The contribution to the response from each connection degree of freedom would be transfer function times the reaction forces. Therefore the total contribution to the response would be summation of the contribution to response from each interface connection point and is given in equation (3).

$$P_t(\omega) = \sum P_{ci}(\omega) = \sum H_{ci}(\omega) * R_{ci}(\omega) \quad (3)$$

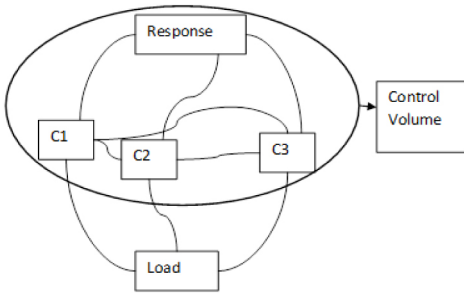


Figure-2. Representation of source-path-receiver concept

The reaction forces are calculated based on grid point force contribution at the connection point from each connecting element outside the control volume. The contribution from each element e_i to the connection point is

$$[F_{ei}] = [E_a(\omega)] * [d(\omega)] \quad (4)$$

where $E_a(\omega)$ is the complex element matrix for the element at connection point, $d(\omega)$ is the complex displacement vector for the degree of freedom corresponding to the element.

If the connection point is attached to rigid element, the contribution is also calculated based on rigid constraint equations similar to $[E_a(\omega)]$ and the corresponding Lagrange multiplier similar to $[d(\omega)]$. The reaction force at the connection point is then computed by adding the element contribution outside the control volume.

$$[R_{ci}] = -\sum_e [F_{ei}] \quad (5)$$

In the one step TPA process, the total system of equations is reduced to a few equations based on response DOFs, d_r , connection DOFs, d_c , and load/force DOFs, d_f .

$$\begin{bmatrix} G_{rr} & G_{rc} & G_{rf} \\ G_{cr} & G_{cc} & G_{cf} \\ G_{lr} & G_{lc} & G_{lf} \end{bmatrix} \begin{Bmatrix} d_r \\ d_c \\ d_f \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \\ f_l \end{Bmatrix} \quad (6)$$

The path contributions, $P_{ci}(\omega)$ from each connection points are calculated based on the above complex dynamic stiffness matrix (G). Then, transfer functions $H_{ci}(\omega)$ of the subsystem in the control volume are calculated by dividing the reaction forces, $R_{ci}(\omega)$ from the path contributions, $P_{ci}(\omega)$.

$$H_{ci}(\omega) = \frac{P_{ci}(\omega)}{R_{ci}(\omega)} \quad (7)$$

It is very important to select the connection points carefully. Be aware that the missing connection points in the reduced system of equation would lead to an incorrect contribution.

Sub-Structuring or Superelement

A Superelement analysis [10, 15, 18] is a form of substructuring wherein a model is divided into pieces with each piece represented by a reduced stiffness, mass, and damping matrix. Each superelement is processed individually and the solutions are then combined to solve the entire model. The final analysis (in which all of the individual superelement solutions are combined) involves much smaller matrices than would be required if the entire model were solved in a single solution. This technique has the advantage of reducing computer resource requirements, especially if changes are made to only one portion (superelement) of the model; in this case, only the affected superelement needs to be reanalyzed and the final analysis repeated. Superelement analysis presents procedural advantages as well, particularly when multiple engineering contractors are involved in the analysis. Each contractor can reduce his model to its boundary degrees-of-freedom, which is suitable for superelement analysis. The systems integrator then combines these reduced models into one model for a final analysis. Superelement analysis has the advantage that matrices can be passed from one organization to another without revealing proprietary modeling details or concern about whether the same grid point and element numbers are used by every participant.

Component Mode Synthesis

Component Mode Synthesis [10, 15, 18] is a form of superelement dynamic reduction wherein matrices are defined in terms of modal co-ordinates (corresponding to the superelement modes) and physical co-ordinates (corresponding to the grid points on the superelement boundaries). CMS is advantageous because there are fewer modal co-ordinates than physical co-ordinates - perhaps only one percent as many. In addition, CMS can utilize modal test data for sub-systems, thereby increasing the accuracy of the overall analysis. In the superelement creation, a care must be taken to select the superelement boundary grids, which depend on the boundary (constraint) connection points between the structures, the input loading points, and the output response points. When superelements are used in a static solution, a static reduction process is performed to replace the physical superelement model with reduced matrices that have the same properties. In dynamic analysis, a reduction must also be performed for all superelements. In static analysis, no approximation is introduced by using superelements. That is, the full static response of the superelement can be represented by the reduced matrices. In statics, the fixed-boundary solution added on to the boundary solution provides the exact solution to the problem. Unfortunately, the reduction methods in dynamics are not exact, as it is in the static solutions. In dynamic reduction, the equivalent (exact interior motion) requires calculating all of the modes of each superelement, which is normally prohibitive. Therefore, approximate reduction techniques are used.

In dynamic reduction, there is the assumption that the solution for each superelement can be represented by using superposition of a series of shape functions. If this approach is compared to the approach used in static analysis, the static part of the transformation is identical; however, instead of the fixed-boundary solution, there is a dynamic transformation matrix that must be multiplied by the motion of a set of generalized coordinates to obtain the solution for the interior DOFs of the superelement. There are two classes of reduction used for superelements in dynamics: static reduction and dynamic reduction. Dynamic reduction is the more accurate approach for most problems, but dynamic reduction also requires more computational effort and some user judgment.

The component mode synthesis (CMS) has been developed to a large extent as a natural consequence of the analysis procedure followed in practice when large and complex structures are analyzed. The general practical procedure is that different groups perform the analyses of different components of the structure under consideration. The advantage of considering separately the components of the structure is primarily one of time scheduling; i.e., the separate groups can work on the analysis and designs of the components at the same time. It is primarily for this reason that the component mode synthesis is very appealing in the analysis and design of large structural systems.

Automated Multi-Level Substructuring

Although a significant model size reduction can be obtained with the Craig-Bampton CMS method, it should be noted, however, that, there is one constraint mode needed for each interface degree of freedom and the cost of the CMS method could still be prohibitive for a sufficiently fine mesh with hundreds of millions of DOFs. In this respect, the Automated Multi-level Substructuring (AMLS) method has been proposed recently [19, 20, 21, 22, 23] which significantly reduces the computational effort as illustrated in the paper by Mallikarjuna Bennur[10] for real-life automotive engineering applications. In the AMLS, a finite element model of a structure is automatically divided into two substructures, each of which is then subdivided into its own substructures. This subdivision is repeated recursively until thousands of substructures have been defined in a tree topology shown in [Figure-3](#).

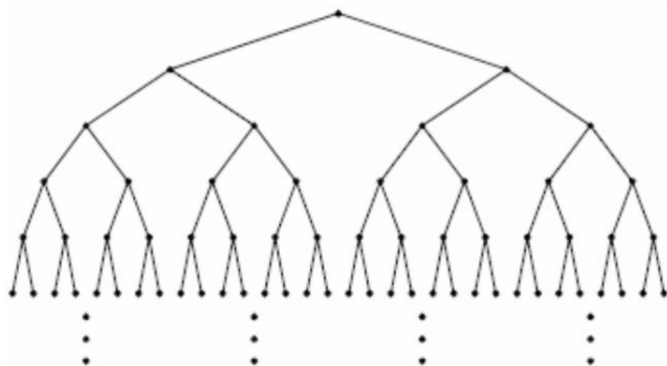


Figure-3. Automated Multi-Level Substructuring Tree

Residual Vectors

When performing dynamic analysis using the modal approach, it is very often desirable to employ residual vectors (modal augmentation) [10, 15, 18] to improve the quality of the solution. Qualitatively, the enforced motion shapes, or “modes,” may not necessarily be represented by the component modes of the structure, leading to a truncation of the enforced motion shapes during projection to the modal space. These shapes can be regained by the use of Modal Augmentation Vectors. Residual flexibility augments the mode shapes with shape functions found from static analysis with various types of loading functions. These auxiliary displacement shapes reduce errors due to modal truncation, particularly near the area of load application. These shapes are modified or discarded to make them linearly independent of the mode shapes. They are then made orthogonal to the mode shapes with a second Eigen solution in a reduced basis. This augmented system provides increased accuracy for modal dynamic analysis. Residual vectors are computed based on inertia relief, applied loads, adjoint load vectors, residual vector DOFs entries, and viscous damping.

NUMERICAL RESULTS AND DISCUSSION

In order to validate the finite element models, all the vehicle components were modeled with very refined mesh until the convergence was achieved. The elements warpage, aspect ratio, length, skewness, chordial deviation, Jacobian ratio, and interior angles were checked in order to have a fine quality finite element model. The full vehicle model has about 16 million DOFs. The salient features of the body[24] are shown in [Figure-4](#). The main contributing components and features (see detailed figures in the [appendix](#)) include the following:

A multi-link double-pivot MacPherson-strut front suspension is used with direct-acting stabilizer bar, driver-adjustable FE3 sport suspension with ‘Magnetic Ride Control’ real-time damping. It uses a double-pivot design for a more precise feeling of control, including more linear and communicative steering, and a smoother ride. It also improves impact isolation on bumps and rough surfaces. The double-pivot system incorporates a pair of ball joints and lower control links that replace a conventional wishbone at each wheel. The optimal balance of ride and handling is achieved via a suspension geometry that decouples road inputs by integrating a soft fore/aft “ride link” that provides greater isolation from impacts and a stiff, laterally positioned “handling link” for more direct steering feel and quicker lateral response. The control arms, the spindle and the body form an effective four-bar link for each side of the front suspension, creating virtual pivot points for reduced spindle length, significantly reducing steering wheel feedback due to wheel impacts and imbalance. There is also a direct-acting stabilizer bar.

Vehicle has a five-link independent rear suspension with lightweight, high-strength steel and efficient straight-link designs.