MEASURED VIBRATION MODES OF CONSTRAINED LAYER DAMPING USING TIME AVERAGED HOLOGRAPHIC INTERFEROMETRY

Paul R. Bernier
Slawomir T. Fryska
Christopher T. Griffen
Ann Marie Revello
General Motors Corporation
Chevrolet•Pontiac•Canada Group Engineering Center
Test Technology & Operations - Central Laboratory
30003 Van Dyke
Warren, Michigan 48090-9060
(313)575-3774

ABSTRACT

The dynamic response and vibration transmission characteristics of structures are determined by three inherent properties; mass, stiffness and damping. Of these, damping is least understood and most difficult to model, measure, and modify. Currently, the Complex Modulus test method is the most widely used to predict the relative effectiveness of a particular material. The one disadvantage of the method is that it cannot predict how the material will actually perform for a given application. In an effort to analyze the performance of constrained layer damping on a particular component, time averaged holographic interferometry was employed. Interferometry allowed imaging of the displacement amplitude field distribution of the component resonant modes. The test method also established a correlation between the interferometric modeshape results and animated modal analysis. The following paper discusses the interaction of these methodologies.

TEST TECHNOLOGY AND OPERATIONS OVERVIEW

This project which began mid-year 1988 was conducted by the Test Technology & Operations group at the GM - C•P•C Engineering Center as part of an effort to address noise, vibration and durability goals of design and development powertrain programs. Noise and vibration personnel utilize an extensive amount of sophisticated engineering tools to reach these targets. These tools provide the data acquisition and processing capability to understand system dynamic characteristics as they relate to steady state response, free vibration, onset and decay of transients, and mode instability/self-excited vibration. Damping plays a significant role in addressing component fatigue life, airborne and structural borne noise, as well as overall increased system impedance to provide greater vibration isolation. The most important responsibility of the group is to develop appropriate state of the art techniques to improve overall vehicle characteristics related to powertrain performance design criteria.

THE INTERACTION OF VIBRATION AND DAMPING

A vibrating structure at any point in the vibration cycle contains kinetic and potential (strain) energy associated with modal mass and stiffness values. Realistic behavior involves energy dissipation as well. The non-conservative nature of mechanical energy conversion by definition is "damping".

Unlike mass and stiffness, damping does not manifest itself as a single phenomena. The mechanisms may include interface friction, fluid viscosity, turbulence, acoustic radiation, eddy currents, magnetic hysteresis, and mechanical hysteresis (material damping).

The primary effects of increased panel damping are reduction of vibration amplitude at the system resonance, more rapid decay at onset of free vibration, decreased spatial conduction of vibration (increased system impedance), and increased isolation during steady state response.

Because damping incorporates several mechanisms to manage the transport of energy many methods of measurement are available including loss factor, damping capacity, reverberation time, decay rate, logarithmic decrement, and spatial decay rate. All of these interrelated methods quantify the damping estimate with the degree of correlation and accuracy dependent on the testing method employed, test specimen, experimental control tolerance (i.e. frequency, temperature, and vacuum), and the engineering interpretation of data.

If damping measurements are carried out on a component interacting with a larger structure, the parameter measured is the "effective damping" accounting for the total system effect. The more complex the modeshape, as well as effectively controlling several modes with a single damping design, presents a difficult optimization challenge because of the need for intelligent and compromising selection of attachment and coverage areas.

The loss factor associated with damping (the most commonly used damping parameter) of most metals and structural materials is usually quite low and relatively independent of amplitude, temperature, and frequency provided stress levels are under the fatigue limit, temperatures well below the melting point, and excitation frequencies are low.

In contrast to this linear and stationary behavior, viscoelastic compounds have elastic moduli and loss factors strongly related to frequency, temperature, and amplitude. These materials are characterized by three regions illustrated in Figure 1 found on the following page.

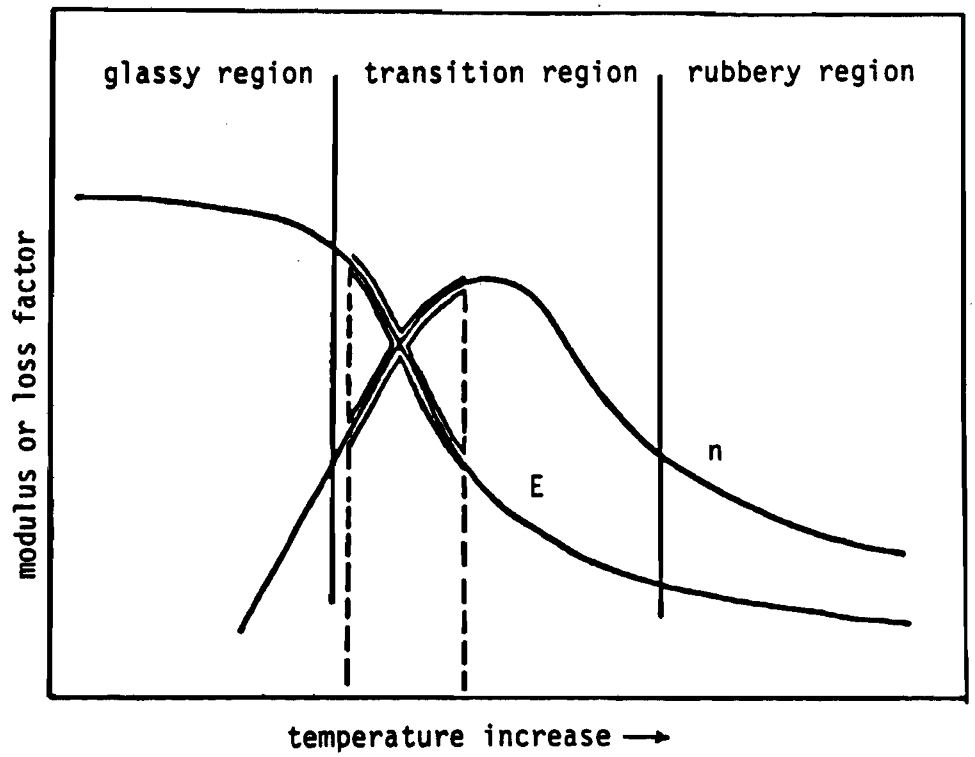
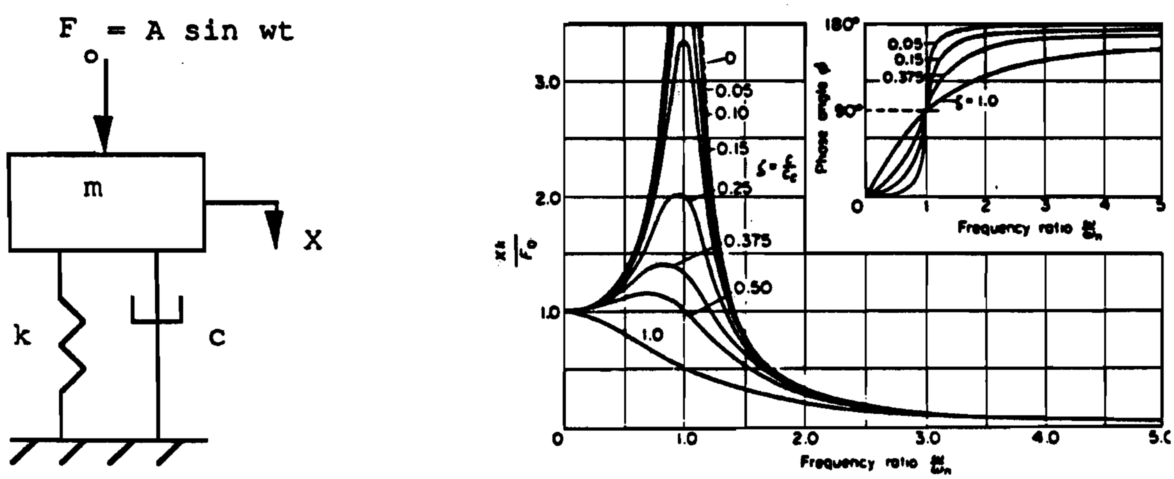


Figure 1: Loss factor and elastic moduli characteristics of damping materials.

The rubbery region offers little reaction force in generating any hysteretic loss to applied loads and no dissipation. In the glassy range the material behaves according to linear elastic theory with complete energy conservation. In the transition range maximum gains of damping occur with non-recoverable energy loss. This behavior is typical of most polymers and elastomer materials. The elastic modulus and loss factor can vary significantly depending on bond site inhibiting plasticizers and bond initiating fillers. The resulting change in dynamic properties of two nominally identical samples from different suppliers or different batches from the same source can result in dramatically different damping effectiveness.

Bending of a panel, which has a number of layers of damping materials, generally causes each layer to bend, extend, and deform in shear. With each type of deformation in each layer there is some storage of strain energy associated with it as well as energy dissipation. It is important to realize that when designing the matched performance of a damping material, the effectiveness of the material is dependent on the product choice, modeshape characteristics, bond integrity, and the forcing function excitation frequency in the operating environment. The material should undergo the same flexural strains as the panel surface when fixed directly to the structure. The method of bonding will affect the composite damping performance, since any deformation/displacement taking place within the adhesive layer will reduce the strain and dissipation energy in the damping sheet.

Modal analysis of discrete and continuous systems depends on solution of the characteristic equation of the eigenvalue problem. The necessary assumption for solution is no damping. By incorporating the approximation of mass and/or stiffness linear proportionality to damping or a lightly damped system (matrix cross terms are zero by Basile's theorem) the damped response solution can be obtained. The concept and convenience of damping expressed by vibration theory is explained by Figure 2.



EQUATIONS:

(1) $F_o \sin w t = mx + cx + kx$

(2) $e^{iwt} = \cos wt + i \sin wt$

(3) $$X = \frac{F/k}{\sqrt{[1 - (w/w_n)^2]^2 + [2\zeta (w/w_n)]^2}}$$

(4) $$\tan \phi = \frac{2\zeta (w/w_n)}{1 - (w/w_n)^2}$$

Legend: ζ = damping factor = c/c_c

ϕ = phase angle

Figure 2: Single degree of freedom oscillator theoretical model and governing equations.

The classical single degree of freedom oscillator with damping has an equation of motion under steady state harmonic forces described by equation 1. The response solution must take a form of equation 2. If terms involving the sine and cosine coefficients are equated after substitution of equation 2 into equation 1 the amplitude and phase relationship can be described by equations 3 and 4. The response is characterized by a ratio of the excitation force ratio to a combined stiffness involving the physical spring element and damping term. At relatively low excitation frequencies (relative to natural frequency) the displacement depends only on the force oscillatory amplitude and the spring constant. At high excitation frequencies the response is determined by the force amplitude discrete mass value and the excitation frequency squared. The system response is then bounded by the physical elements of the mass and spring. At or near resonance the loss factor plays a significant role. As the frequency of excitation approaches the natural frequency with no damping present the denominator approaches zero with theoretically infinite response. With damping or the loss factor present, the system "Q" application is not infinite with the degree of response inversely proportional to magnitude of damping. It is important that the loss factor is also varying with the excitation frequency. The overall damping design sensitivity is highly dependent on the ratio of the excitation frequency to the natural frequency.

An alternative way of expressing classical vibration modeling is by the use of complex stiffness notation. Most techniques for measuring complex stiffness use a material sample as a spring. The most widely used test method is the frequency response method or the Complex Modulus test method (American Standard Test Method E756-83). In this method, a variable frequency sinusoidal force is applied to the test sample and the amplitude of vibration is plotted as a function of frequency as shown in Figure 3 on the following page. The test method is versatile in that it enables damping measurements to be made over a range of frequencies as well as temperatures. The actual test method can differ among suppliers because of their different substrate bar size which produces different results for a particular damping material. Figure 4 on page 7 shows a schematic diagram of the Complex Modulus test apparatus. The test procedure is relatively simple. First, the damping material is bonded to the Oberst bar in a manner suitable for the material. The bar is then mounted into the test jig. The clamping force around the root of the bar simulates a fixed boundary condition. The transducers are positioned approximately 1mm away from the sample bar. Either a sinusoidal (sweep) or random (bandwidth) signal can be applied to the excitation transducer by means of a power amplifier/signal generator.

The frequency response of the bar is measured by the displacement, velocity, or acceleration transducer and recorded as a function of frequency and amplitude for a given temperature. The "effective damping" is obtained by applying

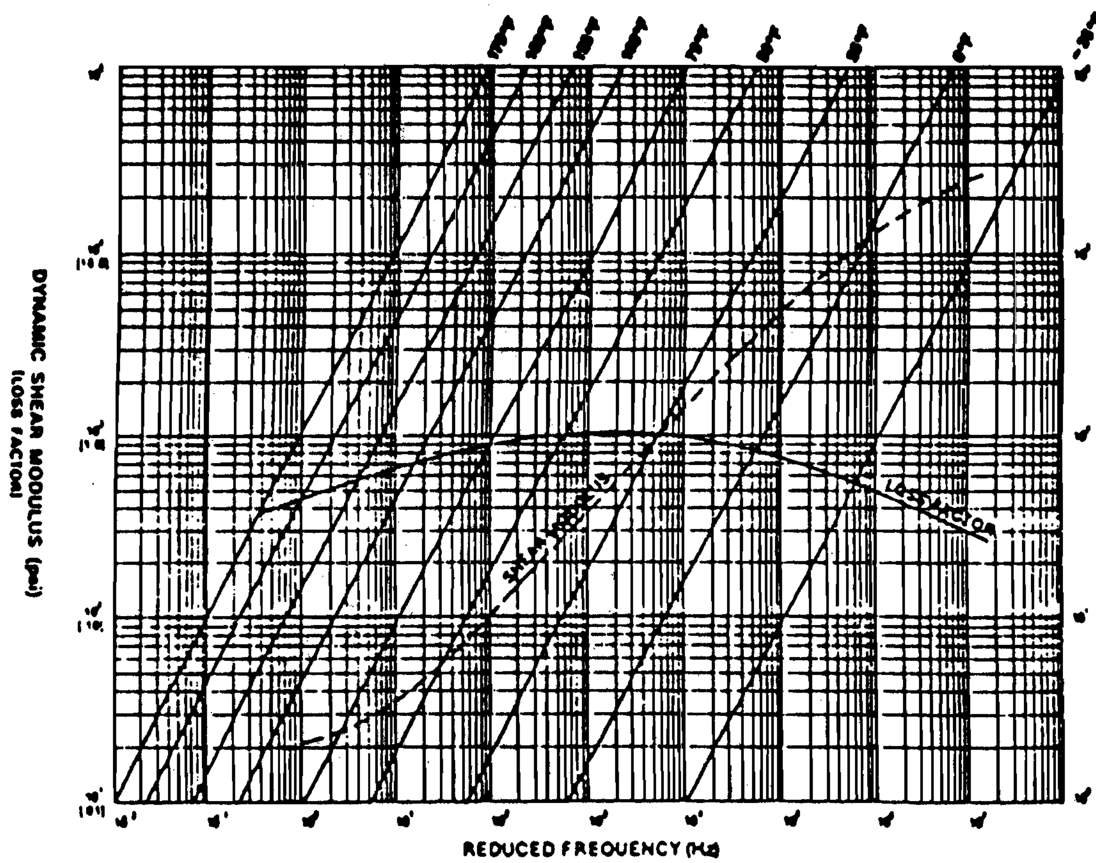


Figure 3: Example of data generated from Complex Modulus Test (Reduced Frequency Nomogram).

the half power bandwidth (3 dB down) at each resonant mode and taking the ratio of the frequency band defined by the half power points to the center resonant frequency to define the loss factor value. The test method assumes linearity, however high levels of excitation can generate non-linearities in the response leading to unreliable data. The amplitude of the force signal applied to the specimen is kept constant as a function of excitation frequency. The loss factor of the base metal is assumed to be zero (0.001) since it is at least a magnitude less than the composite bar. The material loss factor is then calculated from the composite measurement compensated by the Oberst bar damping value.

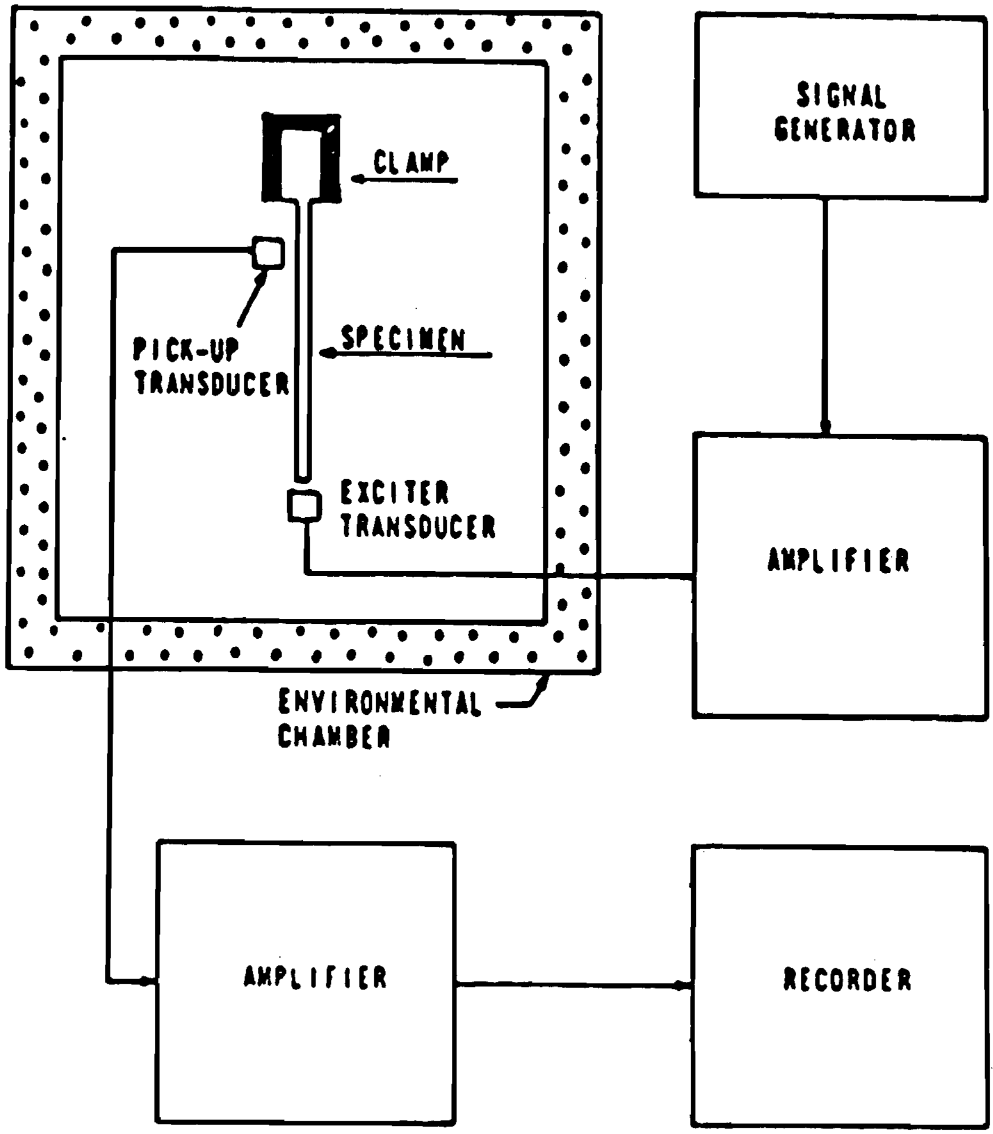


Figure 4: Schematic of Complex Modulus test apparatus.

DAMPING OPTIMIZATION AND TEST PROCEDURE

Base structure - The general flow and approach to damping optimization is shown in Figure 5 on the following page.

The base component chosen is predicated on prior testing of the overall system with results that suggest high sensitivity of the component to damping modifications and

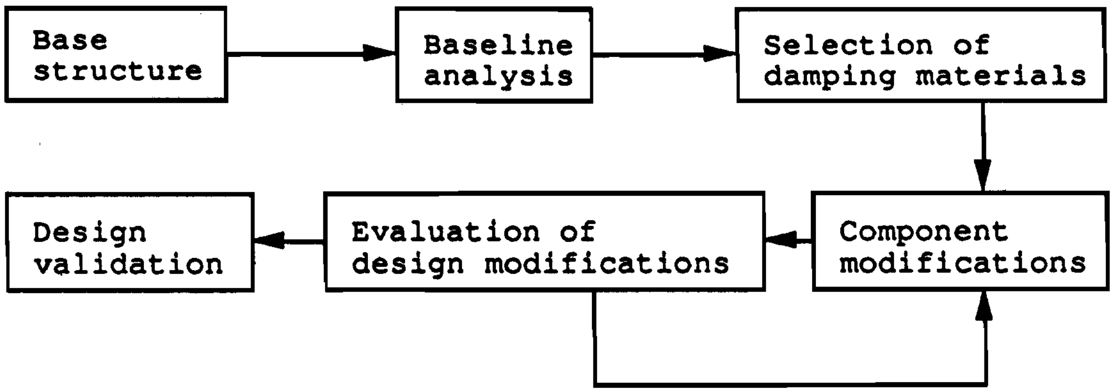


Figure 5: Flowchart of damping design optimization.

potential improvement of the dynamic response characteristics. For example, the transmission oil pan (Figure 6) indicated a high degree of noise contribution during various operating speeds based on sound intensity measurements and sound power rankings of the overall powertrain system.

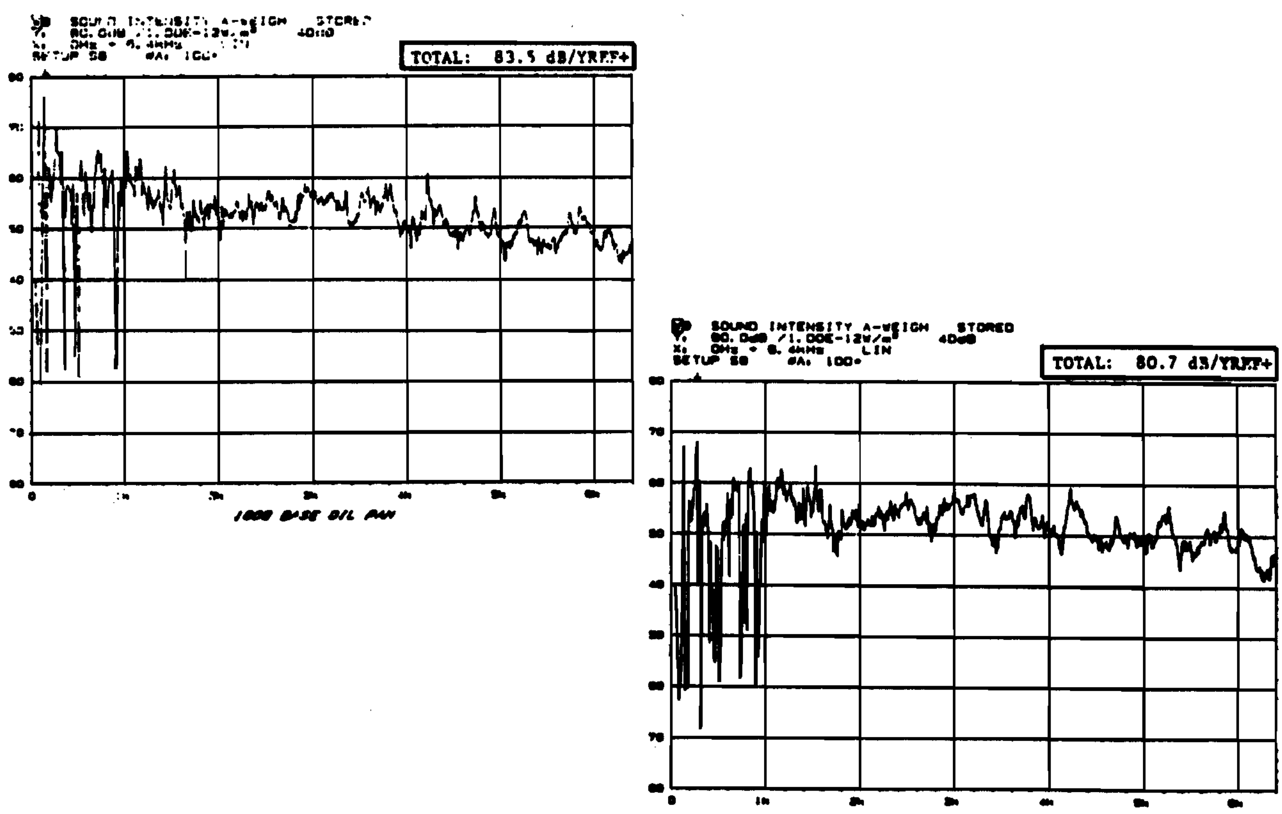


Figure 6: Data from semi-anechoic noise source testing.

Baseline Analysis - Upon completion of the powertrain baseline system testing at the GM - C•P•C semi-anechoic facility, experimental modal analysis and holographic interferometry methods were applied at the GM - C•P•C Optical Test laboratory (Figure 7).

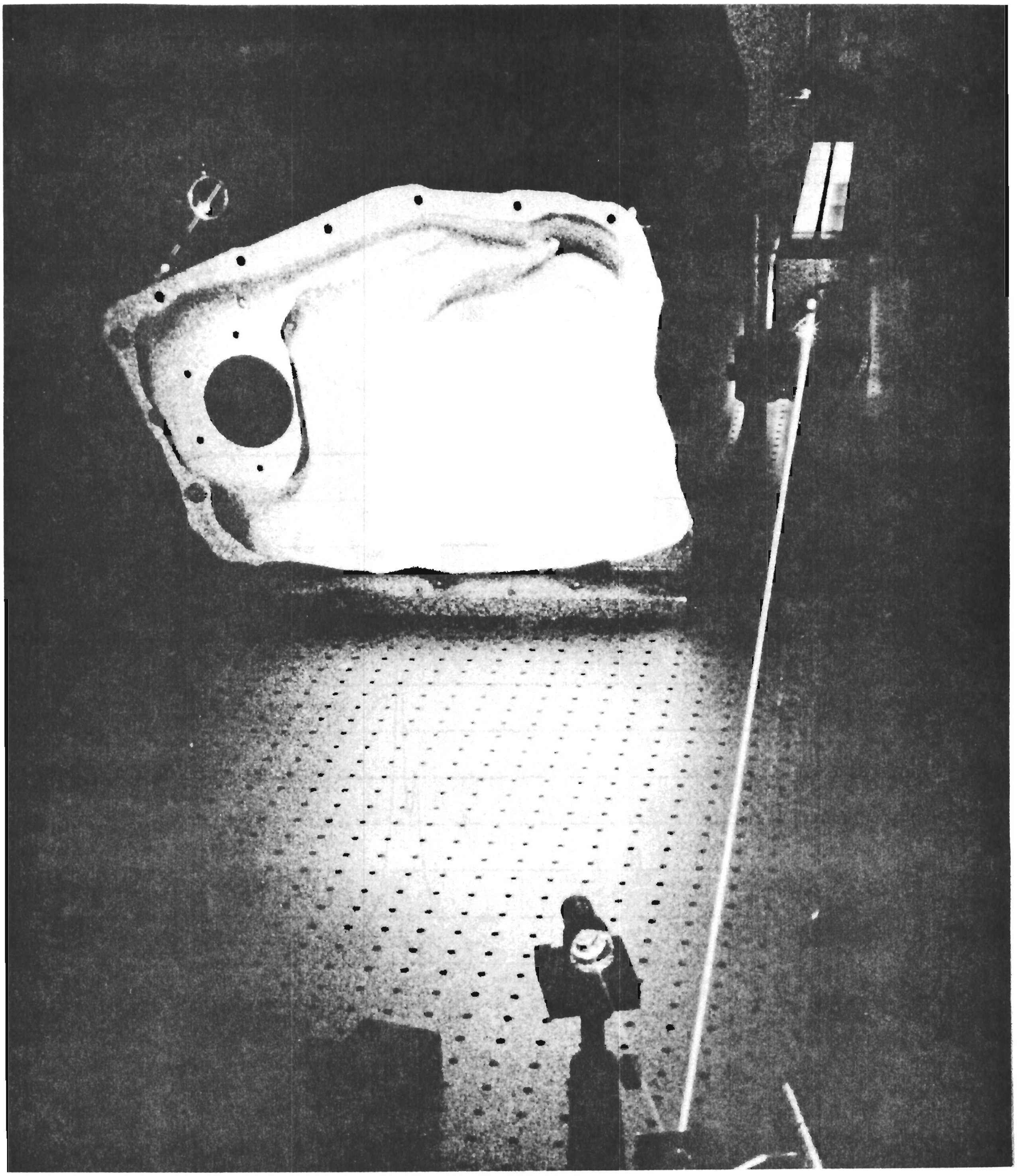


Figure 7: Holographic setup in the Optical Test laboratory.

No finite element analysis methods were used on this project application due to the hardware component availability. The modal analysis performed on the baseline pan identified resonant frequencies of the oil pan and corresponding modeshapes associated with excitation speeds identified by the signature analysis. The amount of baseline "effective damping" was also calculated using the half power bandwidth method on each resonant frequency.

The driving point frequency response function from the modal analysis provided the necessary resonant frequency information (Figure 8) to perform time averaged holographic interferometry and image the modeshape amplitudes already animated by modal analysis (Figures 9,10,11 on the following pages).

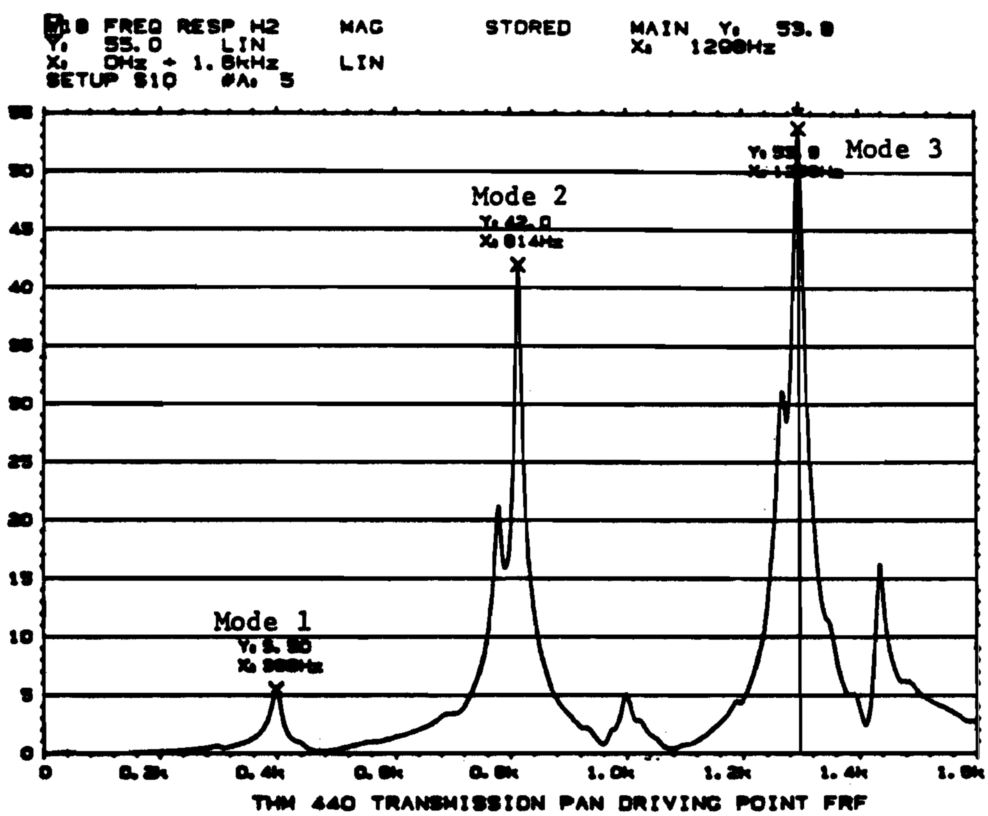


Figure 8: Driving point frequency response function of the baseline transmission oil pan.

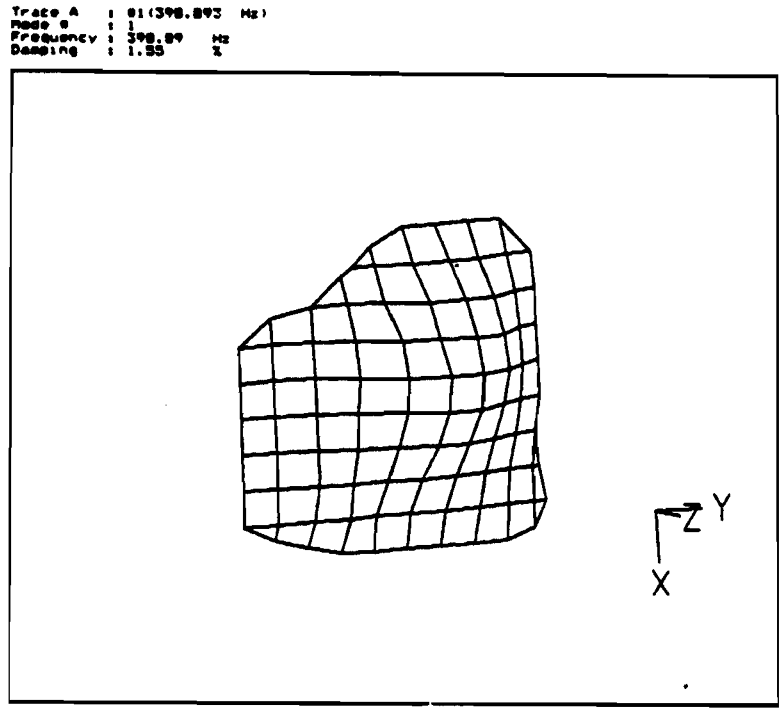


Figure 9 - Mode 1

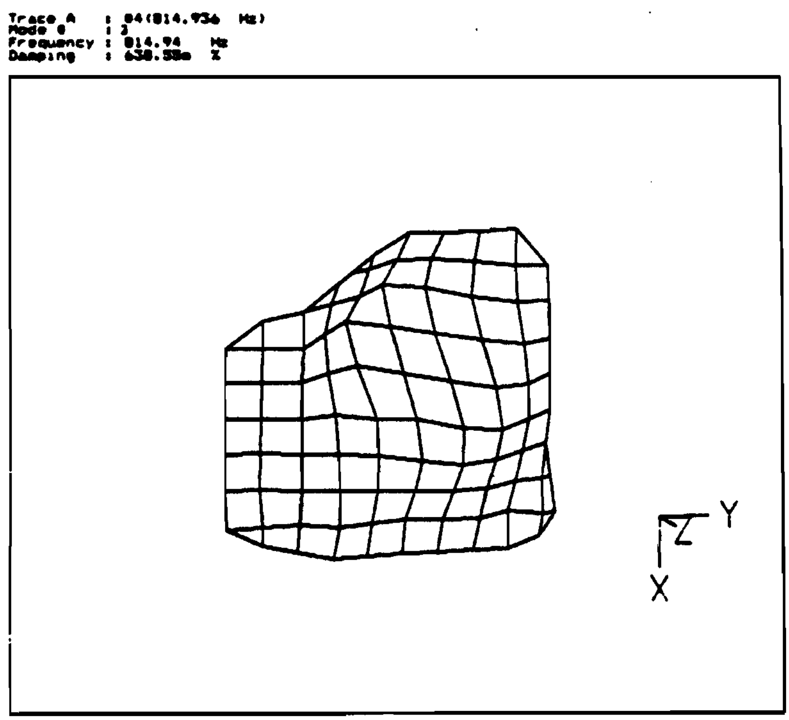


Figure 10 - Mode 2

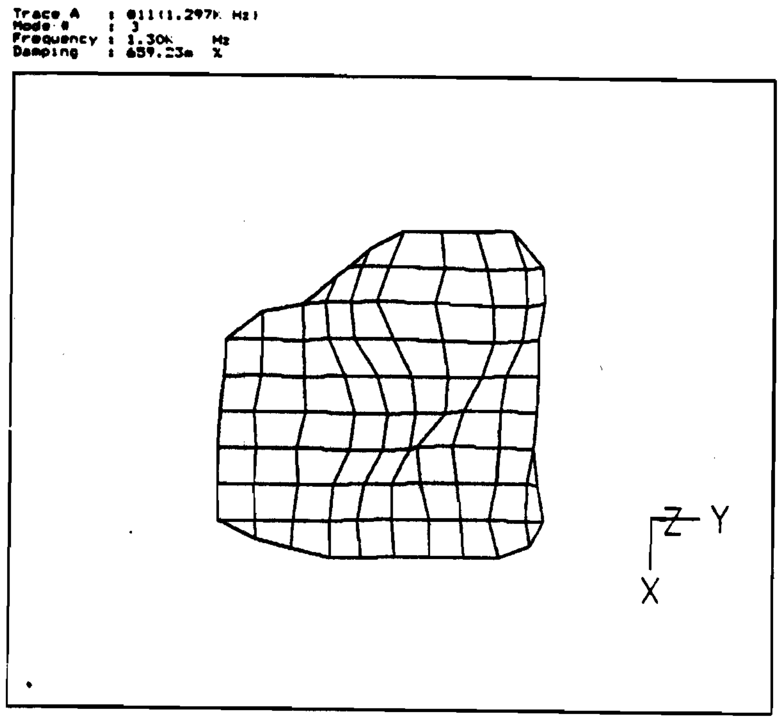


Figure 11 - Mode 3

Figures 9, 10, 11: Animated modal analysis modeshape images of baseline transmission oil pan.

An in depth review of interferometric techniques will not be discussed (reference 1), however a brief discussion is essential to appreciate the value of this supplemental method. The component to be imaged is placed on an isolation table and fixtured with an excitation device (an electromagnetic shaker placed normal to the pan surface) that is decoupled (isolated) from the optical elements on the table. The component is illuminated with a laser source (20mW 632.8nm HE-NE) with the reflected object light recorded on a high resolution photographic plate. Approximately 10% of the illumination beam is split to a second optical reference path that simultaneously exposes the plate. The plate is then developed by standard photographic techniques. Alternative recording media may be used such as thermoplastic cameras (used during this project). The choice is one of pure convenience. The recording process requires a second reconstruction or readout procedure to view and utilize the holographic image. This is achieved by illuminating the developed transparency plate with the original reference beam while viewing the plate. The hologram imaged will be the exact duplicate of the original component in three dimensions. The basic steps in forming the hologram can be used to record the time averaged (averaging of the maximum and minimum displacement over time) dynamic response of the object with the difference being the harmonic excitation of the component at it's resonant frequency during exposure. The reconstructed hologram will then contain both the original three dimensional image as well as displacement contours of the component response corresponding to bright and dark interference bands superimposed on the image. It is this information that is interpreted in conjunction with modal analysis. Time averaged interferometry images are shown for three modes in Figures 12,13 and 14.

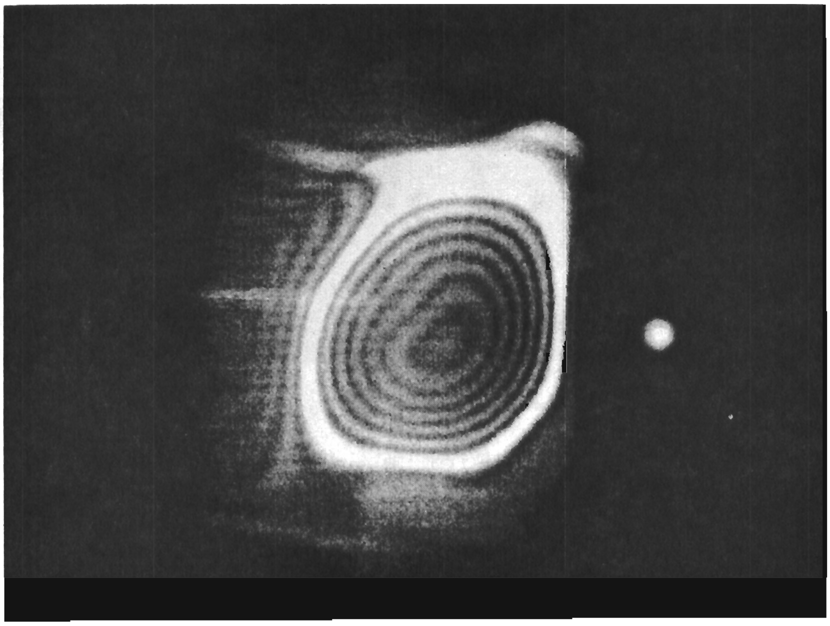


Figure 12 - Mode 1

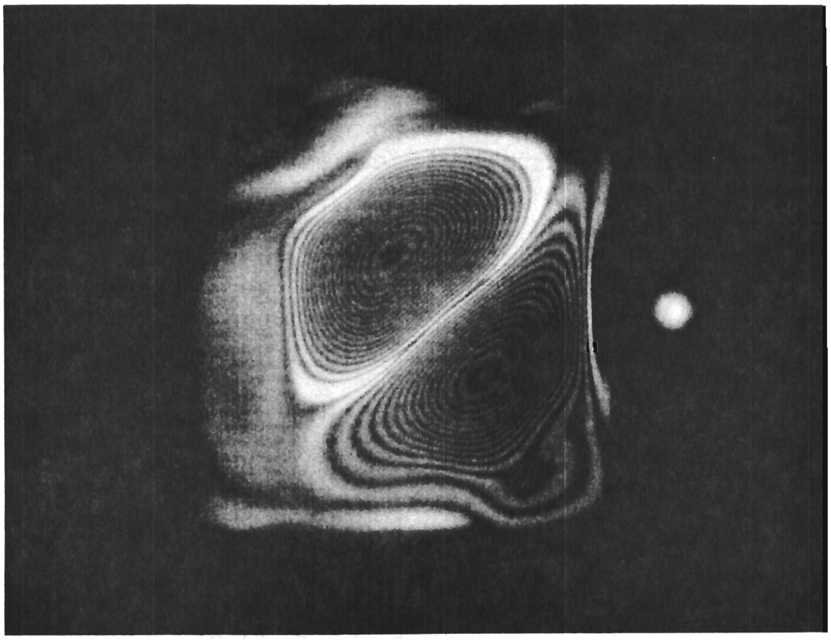


Figure 13 - Mode 2

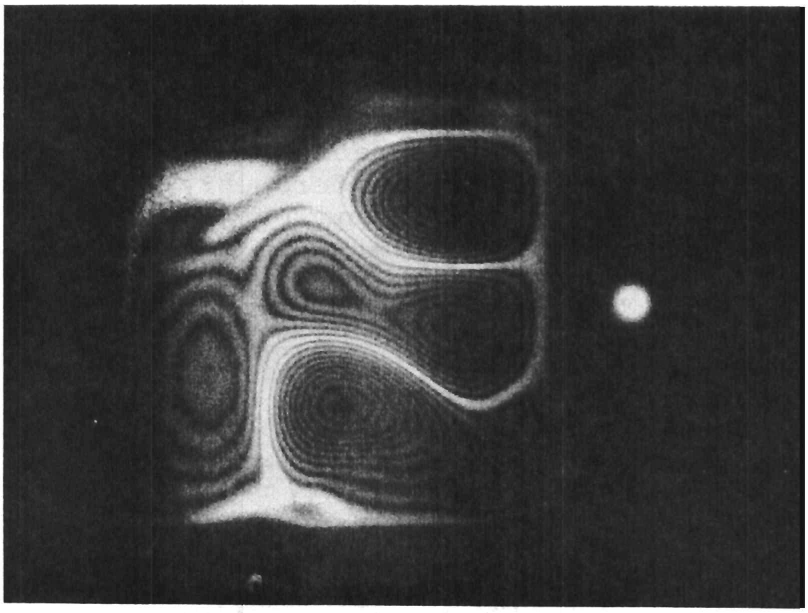


Figure 14 - Mode 3

Figures 12, 13, 14: Holographic interferometry modeshape images of baseline transmission oil pan.

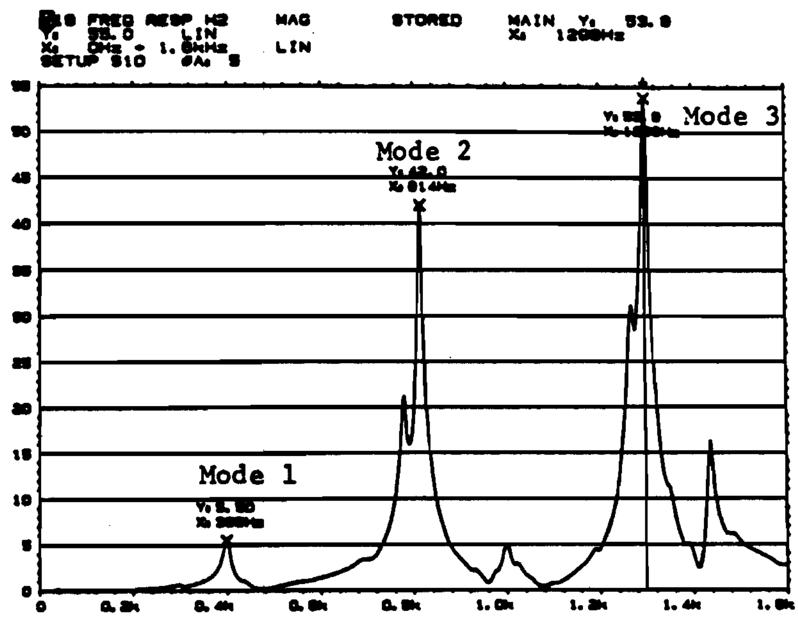
Selection of Damping Materials - After the resonant mode frequencies have identified along with environmental temperatures, appropriate damping materials and associated loss factors can be chosen from reduced frequency nomograms established by the damping supplier. A major caution at this stage of the development program is that the supplier test specimen construction and geometry, along with the particular specimen testing method, will generate nomogram data based on resonant modes and test specimen dynamic characteristics unrelated to the product design. This deficiency can be partially offset in the development cycle time by accurate placement of the damping layer, optimized boundary constraints, and secure bonding to the component.

Component Modifications - The animated modeshapes of the baseline transmission oil pan described the relative amplitude and phase as well as an estimate of nodal locations. The greatest shortcoming of this is the degree of resolution that is defined by a discrete measurement technique. The resolving capability is determined by the number of frequency response function measurements on the component surface. As the natural frequency/modeshape complexity increases so does the requirement for more data.

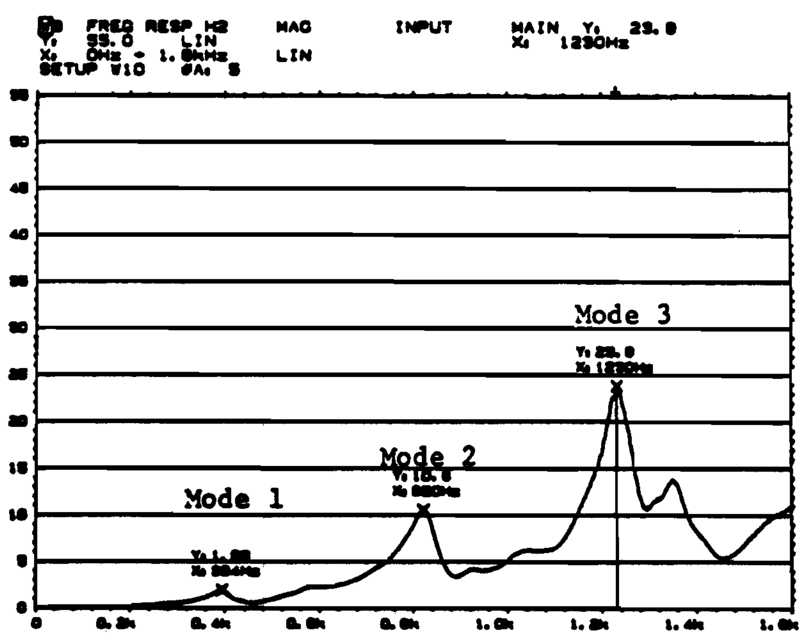
The interferometry method overcomes this limitation by the inherent full field imaging characteristics. The overall amplitude distribution, maximum displacement, and nodal boundaries are completely described. The first step in determining the component area to be treated is locating the nodal boundary for the modeshape of interest. For a combination of modeshapes the compromise involves their superposition as well as effectively controlling the maximum contribution mode. This can be done by careful analysis of sound intensity data. Once the application boundary is established the damping material and constraining layer can be applied.

Evaluation of the Design Modification - The modified component requires only a new driving point frequency response function to calculate the increase in damping and the changes in resonant frequencies associated with the addition of mass, damping, and residual stiffness from the constraining layer. Modal analysis is not required since the remaining information is obtained by re-imaging the new modeshapes with interferometry. Reductions in amplitude can be verified by decreased fringe density. Any disbonding between the damping material and structure will be evident by abrupt discontinuities in contour shape between successive fringes. A simple comparative fringe analysis can be performed with the assumption of a unity sensitivity vector (summation of the observation and illumination vector plane directions) and all displacements normal to the component surface. The fringe patterns are assigned and counted with the 0th order fringe defined as the brightest fringe located on the image. This is referred to as the first root of the 0th order Bessel function and is the optical analog of vibration nodes. Each successive dark to light fringe is assigned an increasing root number. The displacement at any point on the surface is then the root number x laser source wavelength/2. For a component location of the 9th root fringe at a wavelength of 632.8nm, the out of surface plane displacement is 2,847.8EXP(-9) meters.

Design Validation - Design/release depends solely on validation of the component. Assurance of meeting/exceeding design and development targets can include noise and vibration criteria, fatigue and durability goals, corrosion, and other requirements of the system that maybe influenced by the damping design. Regardless of the degree of design optimization the final measure of success relies on the integrated performance of the total system and the realistic gains achieved. Very rarely does this approach theoretical predictions and initial expectations. The final project results comparing the baseline and modified oil pan are summarized in Figures 15 thru 19, on the following pages, showing the difference in driving point frequency response functions, increase in damping, and decreased dynamic response of the resonant modes.



THM 440 Transmission Pan Baseline

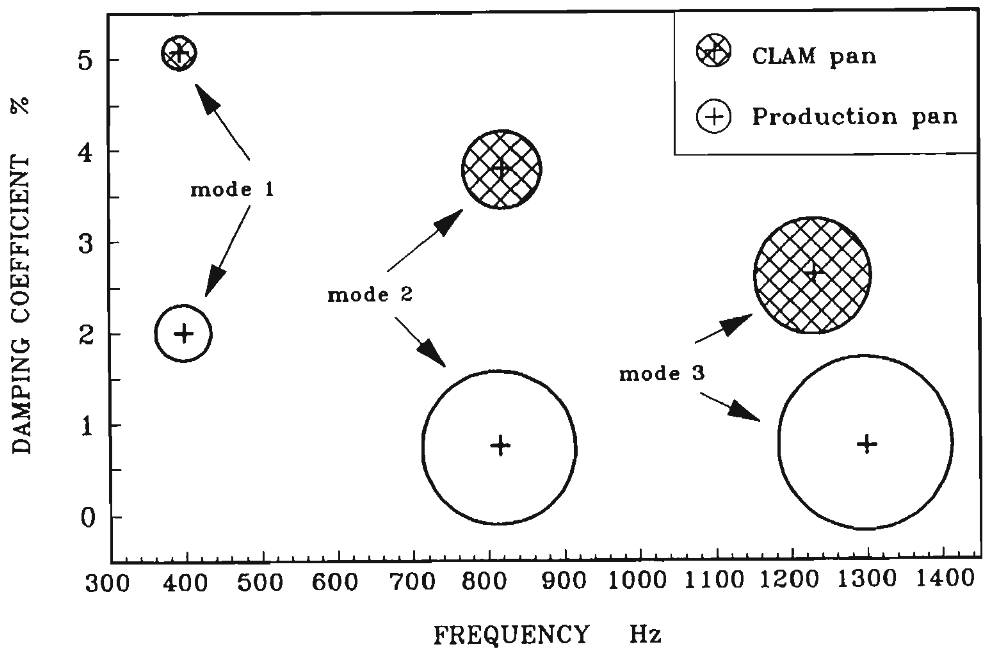


THM 440 Transmission Pan Constrained Layer

Figure 15: Comparative frequency response functions between baseline and modified transmission oil pan.

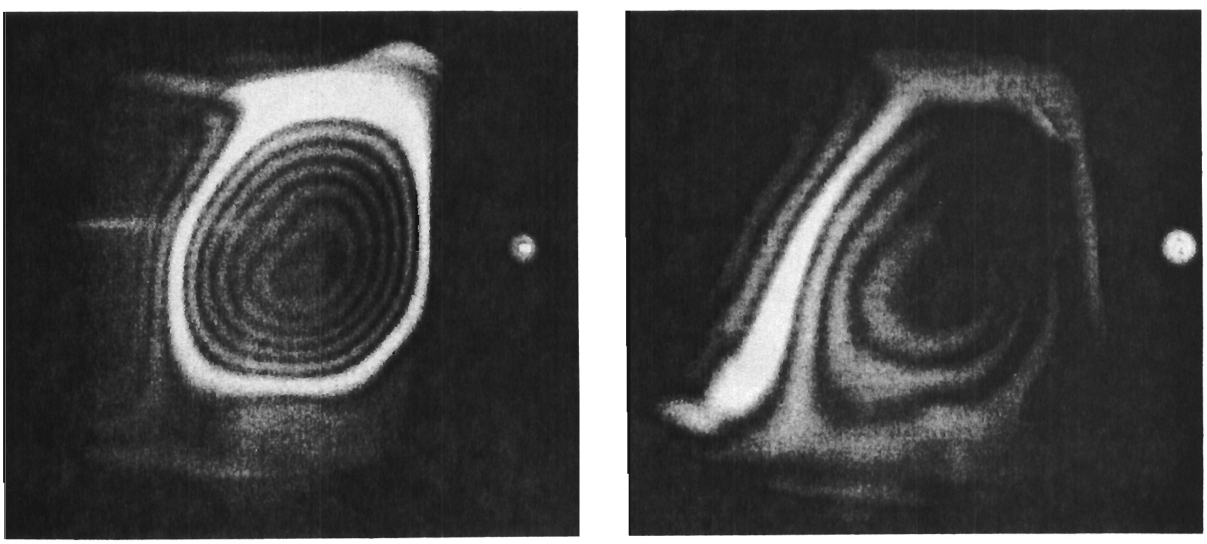
PRODUCTION vs. CLAM TRANSMISSION PAN

Comparison of resonance frequency, damping, and frequency response function amplitude



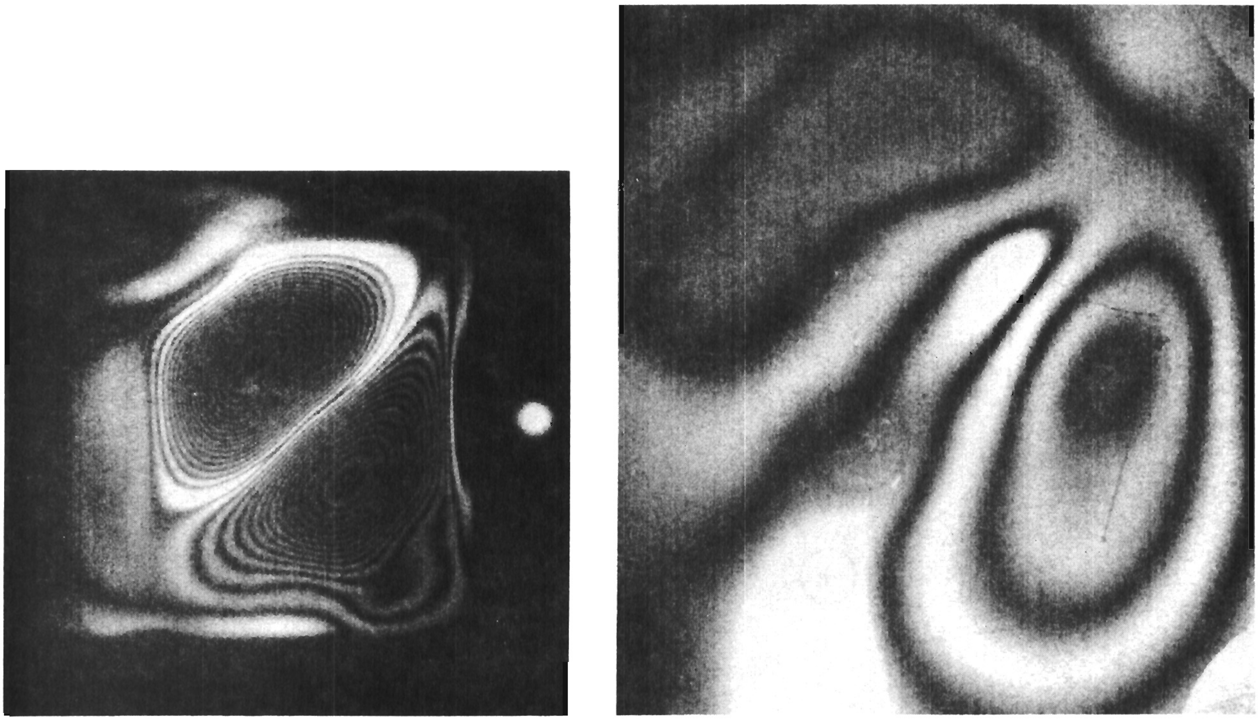
NOTE: circle diameter is proportional to the amplitude of the Frequency Response Function at a resonance frequency

Figure 16: Comparison of damping coefficient (loss factor) between baseline and modified transmission oil pan.



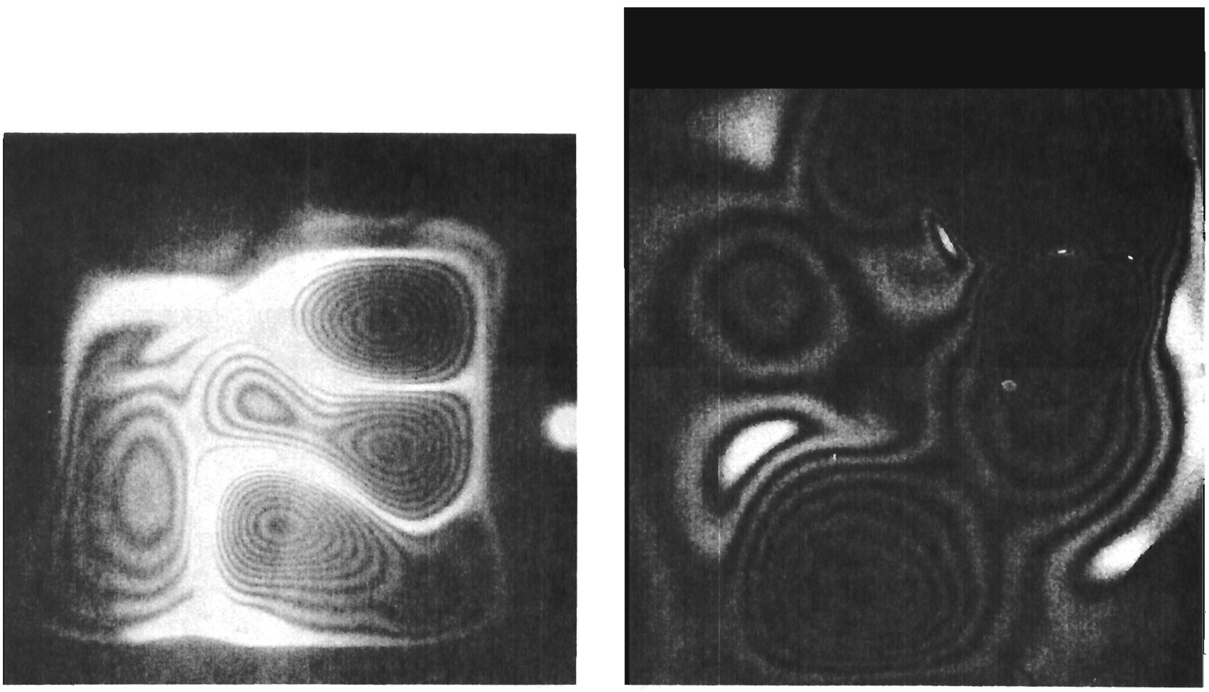
BASELINE MODIFIED

Figure 17 - Mode 1



BASELINE MODIFIED

Figure 18 - Mode 2



BASELINE MODIFIED

Figure 19 - Mode 3

Figure 17, 18, 19: Interferometric image comparison between baseline and modified transmission oil pan.

SUMMARY OF FIGURES

REFERENCES

(1) C.T. Griffen - Applications of Holographic Interferometry in Automotive Component Testing, SAE paper no.870883.

(2) A.M. Revello - Standardization of Vibration Damping Materials Performance Parameters, Chevrolet•Pontiac• GM of Canada Group Headquarters and GMI Engineering & Management Institute, May 13, 1988.

(3) University of Dayton - Vibration Damping Short Course.

(4) L. Meirovitch - Analytical Methods in Vibration.

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