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Resonance Detection and Acoustic Behaviour in Polymer Intake Manifolds

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RESONANCE DETECTION AND ACOUSTIC BEHAVIOUR IN POLYMER INTAKE MANIFOLDS

by

Amrith Ramanan

A Thesis
Submitted to the
Faculty of The Graduate College
in partial fulfillment of the
requirements for the
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A special thanks to my friend Saravanan for all his help and last but not least to Peter Thannhauser for his technical support (I can almost hear his standard ‘You Bet’ as a response to this!)

Amrith Ramanan
The Automotive Industry, over the last few years, has been placing major emphasis on the use of lightweight components in engines as well as other structural components. The material for these lightweight components are being chosen in such a way that they require minimal secondary operations to place the components in service. One area that has developed is the use of Polymer (Plastic) Intake Manifolds for Internal Combustion engines. Plastics meet all the criteria for quality low cost production with a minimum weight component. Plastic intake manifolds, however, have been found to be excessively noisy in operation. This noise can be attributed to the resonance of the plastic components on account of the pulsed flow through them. They also transmit a high frequency noise as the engine throttle is opened, which is also referred to as ‘hiss noise’. The research involved the development of a testing method for and the analysis of the resonance characteristics in plastic intake manifolds. The testing method involved the utilization of full field optical techniques such as time-averaged holography and real-time holography to identify the resonating regions and their characteristics. Analysis of the resonance phenomenon also required a study into the mechanical characteristics and air flow characteristics in plastic intake manifolds. The final result was the development of a useful testing method to analyze manifold resonance and identification of the range of resonant frequencies.
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CHAPTER 1

INTRODUCTION

Review of Relevant Literature

Intake Manifolds

Intake manifolds are separate sets of pipes attached to the cylinder head of automotive engines. It generally comprises a main trunk from which branch pipes are taken, one to each cylinder, and on which is mounted the fuel injection equipment. Their function is to deliver the air which passes the throttle body to the cylinders. The shape and size of inlet manifolds are such that they deliver high velocity air to the intake port of the cylinder with minimum losses. Most intake manifolds today have the individual runners sized to provide for optimum air flow at some particular frequency.

History of Materials used for Intake Manifolds

Until relatively recently, manifolds have been of either cast iron or aluminum, but now various types of plastics are used as alternatives. Cast iron as manifold material is cheap but is the most heavy. Aluminum is light, offers good thermal conductivity and therefore rapid warm up, and can be cast accurately. For these reasons, aluminum was found effective as a manifold material for quite a long time. However, aluminum
has certain disadvantages including its high thermal conductivity adversely affecting heat transfer to the injectors, which in turn can have a detrimental effect on hot starting. Moreover, production of cast aluminum manifolds of any complexity is difficult, requiring multiple cores and introducing problems of porosity.

Plastics, on the other hand, are very light but costly. However, they can be produced to much closer tolerances and therefore be utilized more economically than metals. Another important consideration with plastics is that the tooling costs associated with them are much lower.

Ford motor company was one of the first companies to turn to thermoplastic manifolds. Considering the advantages of using plastics as manifold material, they became hugely popular in the early 1990’s. Ford now produces thermoplastic manifolds at a rate of about 300000 per annum. The most common thermoplastic material that is now being used as intake manifold material is Dupont Zytel, 70G35, a heat-stabilized nylon 6.6 with 35 % short-fiber-glass filler [5]. By virtue of its low viscosity, it easily fills the mould completely and without moving the cores. Consequently, although this material is more costly than polyester, production cycle times are much shorter.

One of the primary reasons automotive manufacturers turned to plastic manifolds was the desire for light-weight and thereby improved fuel economy. For example, a thermoplastic manifold complete with brass inserts and other metal components, weighs only about 2.5 kg. The total weight saving in this case, relative to aluminum,
is about 60%, and the overall cost saving, varying with the fluctuating costs of raw materials, ranges from about 5% to 20% [8].

**Drawbacks of Plastic Intake Manifolds**

The Automotive Industry, over the last few years, has been placing major emphasis on the use of lightweight components in engines as well as other structural components. The material for these lightweight components are being chosen in such a way that they require minimal secondary operations to place the components in service. One area that has developed is the use of Polymer (Plastic) Intake Manifolds for Internal Combustion engines. Plastics meet all the criteria for quality low cost production with a minimum weight component.

Plastic intake manifolds, however, have been found to be excessively noisy in operation. This noise can be attributed to the resonance of the plastic components on account of the pulsed flow through them. In other words, when the frequency associated with the pulsed air flow through the manifold matches the resonant frequency of the plastic component it contributes to excessive noise. Excessive manifold noise is noticeable only during a certain specific range of engine operating RPMs, which shows that the frequencies associated with this RPM range causes the manifold structure to resonate, contributing to excessive noise.

Plastic manifolds also transmit a high frequency noise, referred to as hiss noise [10], as the engine throttle is opened. The noise appears to be louder at idle speed and can
be heard by a person standing outside and in front of the vehicle. A prior study conducted on hiss noise proposed a hypothesis that the noise was on account of the pulsed air flow through the manifold and that it was transmitted by the internal structure layout and radiated through the top shell of the manifold component [10]. There has been minimal research or experimental testing conducted to study excessive noise in plastic intake manifolds due to the pulsed flow through them.

Research Focus

Problem Statement

As already discussed in the previous section, plastic intake manifolds have been found to be excessively noisy during a specific range of engine operating RPM. When aluminum intake manifolds were being used, the noise issue was not of major concern due to the fact that aluminum had the ability to mask high and low frequency noise. This coupled with the fact that plastic manifolds came to be employed in service recently helps explain why very little research and testing has been conducted to analyze the noise phenomenon in plastic manifolds. This research involved the evaluation of testing methods for and the analysis of the resonance characteristics of plastic intake manifolds.
Experimental Test Method

The testing method involved mounting a plastic intake manifold on a special optical test table and subjecting it to the range of frequencies associated with normal engine operating RPMs, in the form of acoustic pressure waves (from speakers) and air pressure pulses fed directly into the manifold body with the help of pneumatic solenoid valves.

Full-field optical techniques such as time-averaged holography and real time holography were used to study the vibration characteristics of the manifold and to identify resonating regions.

Research Goal

The intended final result was the development of a useful and powerful test procedure to analyze manifold design and presentation of analysis techniques for determining the resonance phenomenon in plastic intake manifolds.
Holographic testing is a powerful tool in analyzing vibration characteristics of a solid body. Over the last several decades, holographic testing has proven to be an effective means of analyzing steady state vibration characteristics of solid bodies and has found application in various fields.

Some of the advantages of holographic testing are summarized as bullet points below:

1. Holographic testing has successfully been used in the last several decades to analyze vibration characteristics in solid objects.
2. Holographic pictures of vibrating objects clearly depict vibrating/resonating regions, as well as location of vibration nodes.
3. Different holographic techniques such as time-averaged holography, real-time holography, stroboscopic holography etc can be employed to study the vibration phenomena.
4. Surface displacement calculations are easy.
5. Facilitates non-contact analysis of objects. This is especially useful if the object has a complicated shape or if its vibration displacements can be altered by the weight/affect of contact type sensors (e.g. accelerometers).

6. Facilitates an easy and flexible experimental set-up.

7. Accurate and efficient analysis of vibration characteristics.

Holographic Principles

Holography is a laser recording technique that can capture and recreate three-dimensional images of an object [6]. Although some basic similarities exist between holography and photography, the method, equipment and results are totally different. In photography, the film records a two-dimensional image of the object on film. A hologram, by contrast, simultaneously records both the interference pattern created by the light waves reflected off the surface of the object and the reference beam light waves directed at the recording medium. The recorded pattern has stored in it both intensity and phase relationships of the light source allowing a three-dimensional image of the object to be recreated. Holograms are made using the beam of a coherent (exactly periodic), monochromatic (single wavelength) light produced by a laser.

To capture an interference pattern on a light sensitive recording medium, the typical procedure is to divide a single laser beam into two beams. The reference beam is directed toward the holographic film, while the object beam is directed at the object under study, as shown in Figure 1. As the object beam strikes the object surface, it is
modulated according to the physical characteristics of the object. The light that reflects off the object surface and toward the film now deviates in intensity and phase from the reference beam. The difference between the reference and object light waves is a strong function of the object and carries information about the observed surface on a microscopic level. The hologram is a recording of this complex scattered light distribution, which creates a precise image of the object being studied.

Figure 1: Holographic Recording Setup
Holographic Interferometry

The application of holography made use of during the course of this research is its ability to record two slightly different images and display the minute differences between them. This technique of recording two images, superimposed on each other, is called holographic interferometry. By using this non-contacting optical technique, three-dimensional images of diffusely reflecting objects can be produced which appear to be overlaid with interference fringes. The fringes are indicative of displacement that has occurred between the two exposures. In other words, light waves from the original holographic recording can be made to interfere with light waves from the slightly changed real object to generate a set of fringes. These fringes are functionally a topographic map of the deformation which occurred between the original recording and the second deformed recording [6]. The magnitude of the deformation can be quantitatively evaluated since the fringes are directly related to the wavelength of the light and the displacement that has occurred [3]. The fringe spacing and change in spacing reflect the deformation and the rate of change of deformation respectively [3].
Time Averaged Holography

Time averaged holography was used to identify resonance characteristics of the manifold throughout the experimental exercise. In time averaged holography, the holographic film is exposed to the vibrating object for an extended period of time so as to capture the deflection modes on the recording film. Time averaged holographic pictures of the resonating manifold were used for most of the analysis.

Real Time Interferometry

In this method, a single recording of the object in its reference state is made. Then the hologram is processed and replaced in the same position as in the recording. By looking through the hologram it is able to observe the interference between the reconstructed object wave and the wave from the real object in its original position. It is therefore possible to observe the deformation as it develops in real time by observing the changes in the interference pattern. This method, however, did not prove to be very effective on account of its disadvantage that the hologram must be replaced in its original position with very high accuracy.
TEST PROCEDURE

Experimental Setup

The Coherent Optics and Digital Image Processing Lab at the Western Michigan University was utilized for the experimental research test set-up. The set-up consisted of a polymer intake manifold mounted in the upright position on an optical table. The table was also used to arrange the flat surface mirrors, beam-splitters, beam-expanders and the He-Ne laser.

Figure 2: Experimental Test Setup
Figure 3: Experimental Test Setup

Equipment Used

1. SpectraPhysics Helium-Neon Laser (wavelength 632.8 nm)
2. Edmund scientific flat surface mirrors
3. Jodon VBA-200 beam-splitter
4. Newport Research beam expanders
5. Newport Research model 815 power meter
6. Newport Research model 845 digital shutter controller
7. Keystone Scientific red sensitive emulsion (PFG-01) holographic film  
   (10.2cm x 12.7 cm)
8. Keystone Scientific red sensitive emulsion (PFG-01) holographic plates  
   (10.2cm x 12.7 cm)
9. Thump 600 watt 10" low-frequency response speaker
10. Teknics radio receiver/amplifier
11. Peter Paul electronics pneumatic valves (24 DC, 20 Hz)

Test Specimen Specifications

Table 1
Intake Manifold Specifications

<table>
<thead>
<tr>
<th>Specimen</th>
<th>Polymer intake manifold</th>
</tr>
</thead>
<tbody>
<tr>
<td>Manufacturer</td>
<td>Mann-Hummel</td>
</tr>
<tr>
<td>Vehicle</td>
<td>Ford Escape 2000</td>
</tr>
<tr>
<td>Engine type</td>
<td>3.0L V-60 deg V6 engine</td>
</tr>
<tr>
<td>Manifold material</td>
<td>Heat-stabilized nylon 6.6 with 35% short-fiber-glass filler</td>
</tr>
<tr>
<td>Runner length</td>
<td>190mm X 10mm</td>
</tr>
<tr>
<td>Plenum volume</td>
<td>3500 cc</td>
</tr>
</tbody>
</table>
The test procedure involved two distinct excitation techniques

1. Acoustic excitation of the manifold using speakers.

2. Manifold excitation induced by pressure pulses generated using pneumatic valves.

Manifold Excitation using Speakers

Speakers located at the throttle body entrance and at various port locations were used to excite the manifold acoustically by pumping acoustic pressure waves through the
structure. The manifold as well as the speakers were mounted on an optical table as shown in figure 3. The location of the speakers is shown below in figure 5.

Figure 5: Location of Speakers

The speakers were driven through an amplifier and function generator, to sweep the range of frequencies associated with engine operating RPMs.

Time averaged holograms were then taken of the vibrating manifold. Once the holographic film was developed, the holograms would reveal the nature of vibration in the manifold in the form of interference fringes. The fringes, in turn, were used to determine whether the manifold structure was resonating (maximum displacement
amplitude) or merely shaking about its mounting brackets. This procedure was followed on a trial and error basis until the manifold was subjected to the entire frequency range associated with normal engine operating RPMs. Then studying the fringe patterns on the holograms, the resonant frequency range was identified.

**Pressure Pulses Generated with Pneumatic Valves**

An alternate method to excite the manifold was to generate pressure pulses within the manifold body with frequencies matching those associated with the operating engine RPMs. This method was chosen so that actual manifold operating conditions during engine running could be simulated. For this purpose, a pneumatic pressure line was connected to the manifold plenum and air was pumped in at a pressure of 10 psi, which is the average pressure measured in the manifold at an engine operating RPM of 2500 (Appendix A). All ports and other openings were then sealed with metal plates and an air supply inlet line was used to pressurize the manifold structure. An outlet line was connected near the throttle body entrance, which was in turn connected to a series of pneumatic solenoid valves. The function of the valves was to bleed out air from the manifold at predetermined frequencies. Turning on the air supply pressurized the manifold structure and the valves were then run to bleed out air from the manifold at the required frequency so as to generate pressure pulses within the manifold structure. A LabVIEW program was used to run the valves in sequence. The pneumatic valve setup is shown in Figure 6. The LabVIEW program is shown in Appendix B.
Figure 6: Experimental Setup with Pneumatic Valve Bank

Time averaged holographic pictures were then captured which showed the effect of the pressure pulses on the manifold structure in the form of interference fringes.
Challenges Encountered During Initial Testing

Challenges Encountered - Acoustic

The problem with trying to excite the manifold acoustically was associated with the amplitude of acoustic waves. The speakers used in the initial stage of the experiment were normal mid-range speakers that were incapable of generating acoustic waves of sufficient magnitude to cause resonance in the manifold structure. In other words, these speakers were not capable of moving enough air to cause any sort of resonance in the manifold structure. Some of the measures adopted to overcome this problem included sealing the ports with metal plates, moving the speaker alternatively between the throttle body entrance and the various ports, using more than one speaker at various locations etc. However, none of these measures proved to be a satisfactory solution.

The holographic pictures obtained during this phase are shown in Appendix C. These pictures clearly indicate that the manifold was only vibrating about its mounts with low amplitude. No resonant modes of vibration could be seen in any of the holographic pictures even with high amplitude of vibration.

The problem was overcome by using a speaker with high output, low frequency response and a large cone diameter capable of displacing enough air and drive acoustic waves through the manifold with sufficient force so as to cause resonance in the structure. The speaker used was a Thump 600 watt speaker with a frequency response ranging from 28 Hz – 5KHz.
The speaker was located at the throttle body entrance. The ports were then sealed with metal plates and acoustic waves from the speaker were tunneled into the throttle body entrance. Figure 7 shows the location of the low-frequency response, high output speaker near the throttle body entrance. Successful results were obtained by subjecting the manifold to the entire frequency range by this method. The manifold was found to resonate in the 100-110 Hz frequency range and holographic pictures obtained showed distinct fringe patterns indicative of resonance phenomenon in the manifold body.

Figure 7: Location of Low Frequency Response Speaker
The alternate method chosen to excite the manifold was to drive pressure pulses through the structure as this would help simulate actual manifold operating conditions. Initially, a vacuum pump was used to suck air out of the manifold, which would generate pressure pulses inside the manifold structure. The attempt proved to be futile, however, as the vacuum pump was incapable of generating pressure pulses of any significant amplitude so as to cause resonance. An improvement to this set up was to use a variac to control the frequency of the pump. This also did not yield any satisfactory results. Moreover, the pump was too noisy and tended to disturb the holographic process.

The next step chosen was to make use of solenoid pneumatic valves to drive pressure pulses through the structure. The problem associated with using valves, however, was the inability of available valves to operate at the required frequency range. Since it was evident from the results obtained by acoustically exciting the manifold that it was resonating in the 100 -110 Hz frequency range, the valves too needed to be capable of operating at that frequency range. The valves that were available, however, had a maximum frequency capability of 30Hz. It was therefore impossible to use a single valve to achieve the desired frequency range. The only way to overcome this problem was to use a number of these valves connected in such a way that the resulting frequency output would be much higher (in the 100-200 Hz range). For this purpose, ten of these valves were acquired and connected as shown in Figure 8. The valve
arrangement shows five pairs of valves connected in parallel, with the two valves in each pair connected in series. To achieve the required frequency range, the time offset required between the opening and closing cycle of one valve and the next was calculated to be 5 milliseconds.

A LabVIEW program was then created (Appendix B) which would act as a switch to drive these valves at the required frequency range. The LabVIEW program worked remarkably well and proved to be very versatile in that the frequency output of the valve bank could be efficiently controlled from 0 – 200 Hz. The holographic pictures obtained in this case too showed the manifold structure resonating in the 95 – 100 Hz frequency range on account of the pressure pulses within the structure. The complete setup with the valves is shown in Figure 6. Appendix C contains the holographic pictures taken during the initial ‘challenge’ phase of testing it for resonance.

Figure 8: Pneumatic Valve Bank Arrangement.
CHAPTER FOUR

TEST RESULTS, DATA ANALYSIS AND CONCLUSION

Test Results

The test results show the manifold to be resonating in the 100 Hz – 110 Hz frequency range during acoustic excitation and in the 95 Hz – 100 Hz frequency range when pneumatic valves were used for exciting the manifold.

Test Results - Acoustic Excitation

With the low-frequency response speaker being used to drive acoustic pressure waves through the manifold, the structure was found to resonate at around 110 Hz. Time averaged holograms obtained of the structure subjected to the frequency range of 100 - 110 Hz are shown in Figures 9 through 12.

The holographic pictures depict interference fringes indicative of the resonant mode of vibration in the manifold structure. The resonating region was found to be the central plenum. The fringe pattern indicates resonance through the length of the central plenum and extending into the runner closest to the throttle body entrance as shown in Figures 9 through 12. Additional pictures of the manifold in the resonant mode of vibration are shown in Appendix C.
Figure 9: Manifold Resonance 110 Hz

Figure 10: Manifold Resonance 110 Hz
Figure 11: Fringe Pattern on Central Plenum

Figure 12: Fringe Pattern on Runner Closest to Throttle Body
When the frequency was changed to a little above or below the 100 – 110 Hz range, the resulting holographic pictures showed no resonant vibration mode patterns at all. This is a clear indication that the manifold structure was indeed in a resonant mode of vibration in the 100 – 110 Hz frequency range. If this was not the case, the interference fringes in the manifold hologram would appear more or less the same at any frequency range.

Pictures taken of the vibrating manifold at frequencies above and below the resonant frequency range are shown in Figures 13 and 14.

Figure 13: Vibrating Manifold Holograph 120 Hz
Since the manifold structure was in a resonant mode of vibration in the 100-110 Hz frequency range, higher and lower modes of resonant vibration would be visible at twice/half the 100 – 110Hz frequency range. Subjecting the structure to higher and lower frequencies to observe the different modes of resonant vibration helped understand the structural response of the manifold at different frequencies.

For this purpose, the manifold was subjected to frequencies in the 220 Hz range and the 50 Hz range. The resulting interference fringes that were obtained from holograms taken are shown in Figures 15, 16 and 17.

Figure 14: Vibrating Manifold Holograph 80 Hz
Figure 15: Manifold Interference Fringes 220 Hz

Figure 16: Manifold Interference Fringes 225 Hz
Even though Figures 15 through 17 show the formation of interference fringes, they are not indicative of the resonance mode of vibration. The holographic pictures do not show the presence of any vibration nodes like the ones that were observed when the manifold was excited at the 100 – 110 Hz range. The explanation for this lies in the fact that at these higher and lower frequencies, the structure tends to respond to the excitation in a combination of more than one mode of vibration. In other words, these frequencies do not excite the manifold enough so as to isolate any one mode of resonant vibration. This indicates that the plastic manifold structure seems to provide better sound damping characteristics or responds structurally better at these higher
and lower frequencies. This is consistent with the observation that plastic manifolds are noisier during the start of acceleration when the throttle is initially opened than at higher speeds. It can, therefore be concluded that the manifold structure has weak damping characteristics only in the 100 – 110 Hz frequency range, thus contributing to excessive noise during engine operation in the corresponding RPM range.

Detailed data analysis has been carried out in the next section.

Test Results – Pneumatic Excitation

The test results obtained when pneumatic valves were used to induce pressure pulses in the manifold were quite consistent with the results obtained from acoustically exciting the manifold. Using pneumatic valves to generate pressure pulses helped create a more realistic simulation of what happens in a manifold during actual engine running. The results obtained showed the manifold to be resonating at 96 Hz, which is at very close proximity to the resonance observed with acoustic excitation, which was at 110 Hz. The discrepancy in resonant frequency range varying from the 100 – 110 Hz range to the 95 – 100 Hz range can be owed to the margin of error in reading frequency output from the pneumatic valve bank.

Figures 18 and 19 show time averaged holographic pictures of the resonating manifold.
Figure 18: Manifold Interference Fringes 96 Hz

Figure 19: Manifold Interference Fringes 98 Hz
Data Analysis

Engine RPM – Frequency Relationship

Acoustically exciting the manifold using the low frequency response speakers yielded results indicating that the manifold structure was resonating in the 100 – 110 Hz frequency range.

The frequencies associated with the range of engine RPMs was calculated using the formula given below

For a four-stroke cycle engine,

\[
\text{Frequency (Hz)} = \frac{\text{Engine RPM} \times \text{Number of cylinders}}{120}
\]  

(1)

In this case, the number of cylinders = 6

Figure 20 shows the frequencies associated with the range of normal engine operating RPMs.
From the figure, it is evident that the engine RPM range corresponding to the 100 – 110 Hz frequency range is 2000 – 2200 RPM. This RPM range corresponds to the start of acceleration, thereby indicating that the manifold structure tends to resonate at the short RPM range corresponding to initial throttle opening during initial acceleration.

This result is consistent with the observation that plastic manifolds are the most noisy during engine warm-up and during initial acceleration.

Furthermore, the fact that no resonance mode of vibration was observed at twice the 100 – 110 Hz frequency range proves that the manifold structure responds better or has better damping characteristics at higher frequencies. The RPM range in this case, corresponding to 200 – 220 Hz is 4000 – 4400 RPM.
Displacement Measurement - Non Resonant Frequency

The interference fringes that were detected at a frequency of 220 Hz were used to calculate the out of plane displacement values on the manifold surface when it was subjected to this non-resonant frequency range. The fringes in the image plane of the hologram are directly proportional to the displacement. The viewed fringes are the result of a total path length change of multiples of $\lambda/4$. The intensity of the fringes in the hologram is a result of the complex image waveform ($U_i$) between original and deformed states.

\[ U_i = A_0 e^{i\theta} \quad \text{(undeformed)} \]  
\[ U_i = A_0 e^{i(\theta + \delta)} \quad \text{(deformed)} \]

Where, $A_0 =$ amplitude

\[ \theta = \text{relative phase of the illumination source} \]

\[ \delta = \text{path length change} \]

The intensity ($I$) of the image, as resolved by the eye becomes,

\[ I = 2 I_o [ 1 + \cos(\delta) ] \]

Where, $I_o = A_0^2$

Bright fringes are present when

\[ [ 1 + \cos(\delta) ] = 2 \]
or when \( \delta = N \pi \)

\[ N = 0, 2, 4, 6, \ldots \]

Dark fringes correspond to

\[ [ 1 + \cos ( \delta ) ] = 0 \]

or when \( \delta = N \pi \)

\[ N = 1, 3, 5, 7, \ldots \]

\[ \delta = (2n - 1) \pi \]

\( n = \) numeric fringe count

The path length change occurs due to points on the surface of the object changing position. If the illumination source and the viewing positions are known

\[ \delta = \frac{2 \pi}{\lambda} \left[ (\sin \theta)u + (\sin \phi)v + (1 + \cos \theta)w \right] \]  \( \ldots \) (5)

where \( \lambda = \) wavelength of light used

\( u = \) displacement of surface along the horizontal axis

\( v = \) displacement of surface along the vertical axis

\( w = \) displacement of surface along the axis normal to the viewing plane

\( \sin \theta = \) orientation sensitivity to horizontal, in-plane displacement

\( \sin \phi = \) orientation sensitivity to vertical, in-plane displacement
\[(1 + \cos \theta) = \text{orientation sensitivity to normal, out-of-plane displacement}\]

Equating deltas,

\[
[(\sin \theta)u + (\sin \phi)v + (1 + \cos \theta)w] = \frac{N\lambda}{2}
\]  \hspace{1cm} (6)

If the beam is horizontal, then \(\sin \phi = 0\)

and if the dark fringes are counted, then

\[
[(\sin \theta)u + (1 + \cos \theta)w] = \frac{(2n - 1)\lambda}{2}
\]  \hspace{1cm} (7)

with small angles being used, the out-of-plane displacement \(w\) can be approximated from the above equation with

\[w = \frac{(2n - 1)\lambda}{4}\]  \hspace{1cm} (8)
Figure 21: Fringe Count on Vibrating Manifold – 220 Hz

Figure 21 shows the holographic picture of the manifold vibrating at 220 Hz. The fringes can be assigned numbers as shown above.
For the first fringe, \( n = 1 \)

Therefore, according to the derivation shown above,

Out of plane displacement for the first fringe, \( w = \frac{(2n - 1)\lambda}{4} \)

where \( n = 1 \)

\[ \lambda = \text{wavelength of He – Ne laser} = 632.8 \text{ nm (nanometers)} \]

therefore \( w = \frac{(2 \times 1 - 1) \times 632.8}{4} = 158.2 \text{ nm} = 158.2 \times 10^{-9} \text{ m} \)

Table 2 shows the displacement values for fringes numbered 1 to 6.

<table>
<thead>
<tr>
<th>Fringe count</th>
<th>Displacement (mm)</th>
<th>Length affected by displacement (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>( 1.58 \times 10^{-4} )</td>
<td>855 mm</td>
</tr>
<tr>
<td>2</td>
<td>( 4.75 \times 10^{-4} )</td>
<td>790 mm</td>
</tr>
<tr>
<td>3</td>
<td>( 7.91 \times 10^{-4} )</td>
<td>680 mm</td>
</tr>
<tr>
<td>4</td>
<td>( 11.07 \times 10^{-4} )</td>
<td>630 mm</td>
</tr>
<tr>
<td>5</td>
<td>( 14.24 \times 10^{-4} )</td>
<td>590 mm</td>
</tr>
<tr>
<td>6</td>
<td>( 17.40 \times 10^{-4} )</td>
<td>510 mm</td>
</tr>
</tbody>
</table>

Fringe number 6 is the last detectable fringe count and it lies in the region of the vibration mode pattern shown in figure 21. This is the area of maximum displacement. The length affected by displacement is the length of the area under each fringe which has been displaced by the corresponding displacement value given.
Displacement Measurement - Resonant Frequency

The reconstruction of a time averaged hologram made of a vibrating object provides a vibration measurement. The amplitude of vibration can be calculated from equations derived as shown in Appendix D.

The intensity of the hologram is some irradiance $I_o$ modulated by the Bessel function (see Appendix D for derivation)

$$I = I_o J_0^2 (\theta)$$

where $J_0^2 (\theta)$ produces the fringe pattern and

$$\theta = (2\pi/\lambda) C A(x, y)$$
The Bessel function is a function with decreasing amplitude with numerous zeros. The Bessel function produces fringes in the time average hologram. Because of the Bessel function, areas of zero displacement (nodes) are highly illuminated. The nodal pattern at resonant frequency is shown in Figure 23.

Displaced areas show decreasing intensity with the dark fringes indicating zeros of the Bessel function.

Figure 23: Nodal Shape on Resonating Manifold
The Bessel function equations are shown in Appendix D. The roots of the Bessel function are also shown in Appendix D.

The magnitude of vibration can be calculated through

$$\theta = \left(\frac{2\pi}{\lambda}\right) C A(x, y)$$

or

$$A(x, y) = \frac{(\theta \lambda)}{(2\pi C)}$$

where $\theta$ is the corresponding zeros of the Bessel function.

And $A(x, y)$ is the magnitude of displacement.

According to the derivation shown in Appendix D, fringe number one in Figure 24 corresponds to the first zero of the Bessel function, $J_1(0)$.

![Figure 24: Fringe Count on Resonating Manifold](image)
Magnitude of displacement,

\[ A(x, y) = \frac{(\theta \lambda)}{(2\pi C)} \]

where

- \( \lambda \) = wavelength of laser
- \( \theta \) = corresponding zeros of the Bessel function
- \( C \) = sensitivity vector for out of plane displacement

If \( \alpha \) is the angle between object illumination and viewing direction, then

\[ C = \frac{\lambda}{2 \sin \alpha} \]

In this case \( \alpha = 35 \) degrees

Therefore

\[ C = \frac{0.6328 \mu m}{2 \sin 35} = 0.552 \]

For the first dark fringe,

\[ A(x, y) = \frac{(2.405 \times 0.6328)}{(2 \times \pi \times 0.552)} = 0.4388 \]

The magnitudes of displacement for the fringes observed are calculated in Table 3.

The zeros of the Bessel function were used from the table in Appendix D.
Table 3

Magnitude of displacement across central plenum

<table>
<thead>
<tr>
<th>Fringe Count</th>
<th>J0 Root</th>
<th>Displacement Magnitude (µm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2.405</td>
<td>0.439</td>
</tr>
<tr>
<td>2</td>
<td>5.520</td>
<td>1.007</td>
</tr>
<tr>
<td>3</td>
<td>8.654</td>
<td>1.579</td>
</tr>
<tr>
<td>4</td>
<td>11.792</td>
<td>2.152</td>
</tr>
<tr>
<td>5</td>
<td>14.596</td>
<td>2.664</td>
</tr>
</tbody>
</table>

Figure 25 below shows the plot of displacement magnitude across the central plenum from a reference point near the throttle body entrance to a point near the center of the plenum. It is clear from Figure 25 that the amplitude of displacement, when the manifold structure is resonating, increases in an almost linear fashion from the throttle body side to the midpoint of the central plenum. The displacement amplitude from the midpoint of the central plenum to the end opposite the throttle body entrance will decrease in a similar manner. The center of the plenum will, therefore, have the maximum displacement amplitude.
Vibration Modes

From the vibration analysis of an object, it is possible to predict the response of the structure to any distribution of forces on its surface at any sinusoidal frequency. Holographic reconstructions showing vibration patterns on an object at various frequencies can be used to predict the response of the object at various frequencies and subjected to alternative excitations.

The deflections of a structure as a result of vibration can be expressed as a sum of a set of functions, each of which describes an independent response to the external forces applied to the structure. These responses are called the normal modes of the structure, and they represent the independent degrees of freedom of the structure. Each mode has a characteristic response as a function of sinusoidal frequency of
excitation. If the structure has very little damping, these responses will be sharply peaked at certain characteristic frequencies and the structure will ring at those frequencies if excited. This gives rise to sharp resonance at these characteristic frequencies.

The phenomenon of resonance in weakly damped structures gives rise to the idea that vibration modes may be isolated by exciting a structure at the natural resonances of each of its modes. However, all real structures respond to excitation by moving in a combination of all its modes. Therefore since the structure is capable of responding in more than one mode even at the natural frequency of any particular mode, it becomes difficult to isolate any one resonance mode.

In this case, however, one of the resonant vibration modes was isolated in the 95-110 Hz frequency range. The higher or lower modes of resonant vibration could not be isolated as the structure tended to respond better at the remaining range of frequencies. The vibration pattern observed in the holographic pictures of the manifold excited at the range of remaining frequencies (frequencies other than the 95-110 Hz range) seems to suggest that the structure was moving or responding in a combination more than one vibration mode thus preventing itself from resonating at one characteristic frequency.
Conclusion

The aim of the research conducted was to develop an efficient, functional and realistic test method to detect resonance in plastic intake manifolds and to perform a complete analysis of the resonance phenomenon in them. The research succeeded in developing a very versatile test procedure for analyzing manifold structural design for noise and vibration characteristics. Laser holographic test techniques that were used to analyze manifold design was shown to be an easy and accurate means to study resonance characteristics in plastic manifold structures.

Two alternate methods were developed to excite the manifold structure for resonance (1) acoustic excitation using speakers (2) using pneumatic valves to induce pressure pulses inside the manifold structure.

With both cases, time averaged holographic pictures of the vibrating manifold indicated resonance at a specific frequency range. Resonance fringes were observed in the 100 – 110 Hz frequency range with the acoustic mode of excitation and in the 95 – 100 Hz range when pneumatic valves were used. The holographic pictures clearly indicated the presence of a resonant vibration node. The resonant fringe pattern was seen to be traversing the length of the central plenum and extending into the runner closest to the throttle body entrance. The engine RPM range corresponding to the frequencies where resonance was observed was 2000 – 2200 RPM in the case
of acoustic excitation and 1800 - 2000 RPM when pneumatic valves were used to excite the manifold. These results were consistent with the observation that plastic manifolds are the most noisy during the start of acceleration when the throttle is initially opened.

Analysis of the vibration response of the manifold at the remaining range of higher and lower frequencies showed the absence of any resonant mode of vibration other than that in the 100 - 110 Hz range. This was indication that the manifold was responding structurally better or had better damping characteristics at these remaining range of frequencies. The absence of resonant modes of vibration at twice and half the 100 - 110 Hz range was due to the fact that the manifold structure responded to excitation by moving in a combination of more than one mode of vibration thus hindering the possibility of isolating any one resonant mode. It was, therefore, concluded that the plastic manifold structure, in this case, had weak damping capabilities in the 100 - 110 Hz range (acoustic) and 95 - 100 Hz range (pneumatic).

The biggest challenge during the course of the research study was simulating pressure pulses of magnitude close to those present in the manifold during actual engine running. Initial setbacks with acoustic methods of excitation included the inability to induce resonance in the structure due to the lack of amplitude of acoustic waves emanating from speakers used. The use of a low frequency - high output speaker with a large enough cone diameter solved the problem and was capable of inducing resonance in the manifold.
The alternate method of excitation used pneumatic valves to induce pressure pulses in the manifold after the structure had been pressurized via an air-supply line. This method was chosen to create a realistic simulation of actual pulsed flow through the manifold during actual engine running. Initial challenges faced with this method included the inability of individual valves to attain the desired frequency range and the absence of a switch to sequence valves if more than one had to be used. This problem was overcome by creating a LabVIEW program which would act as a switch to sequence the operating time of ten valves connected together to achieve the desired frequency range. The LabVIEW program created was capable of controlling frequency output of the valve bank from 0 – 200 Hz.

In conclusion, it can be said that the test procedure developed has great potential to evolve into a popular and acceptable method to analyze manifold design for noise and vibration characteristics. The analysis of the resonant modes of vibration, as shown in this study, will help optimize manifold design to eliminate excessive noise.
Recommendations for Future Research

There has been minimal research work conducted into the study of resonance phenomena in plastic intake manifolds. The widespread acceptance of plastic as manifold material, on account of its advantageous material properties, necessitates the need to address the noise phenomenon in them to enhance their performance as intake manifold material.

This research study provides an easy, efficient means to analyze manifold design for noise characteristics, thereby opening up avenues for further advanced and more concentrated research to address the noise issue in plastic manifolds.

Alternate testing techniques, other than the ones used in this study, include the use of stroboscopic holography using acousto-optical modulation for identification of vibrational modes of the structure. Stroboscopic holography involves the periodic attenuation of the laser to record specific phase angles of an object undergoing steady state vibration. This technique allows for real time identification and visualization of steady state vibrational modes.

Other aspects of the noise issue in plastic manifolds that could be researched include embedding reinforcing material along the regions that are poorly damped to study how the modified structure responds to pulsed flow. This is possible by analyzing the interference fringes in holographic pictures of the modified manifold structure.
Considering the gains that can be had by eliminating excessive noise in plastic intake manifolds, research studies of the noise issue in them certainly deserves considerable attention.
Appendix A

Data Relating to Pressure and Pulsed Flow Characteristics in Plastic Intake Manifolds
A1: Pressure measurement inside a plastic manifold at 2500 RPM.
A2: Pulsed flow through a manifold junction. The figure shows the formation and progression of compression and expansion waves.
Appendix B

LabVIEW Programs
Block Diagram

C1: LabVIEW program created for pneumatic valve operating time sequencing.
C2: Frequency control panel for LabVIEW program used to sequence valve timing.
C3: Initial LabVIEW program and control panel created that did not satisfy required valve time sequencing.
Appendix C

Holograms Challenge/Resonance Phase
Pictures of Resonating Manifold – Acoustic and Pneumatic Modes of Excitation
C1: Time average hologram of vibrating manifold taken during initial failure stage of acoustic testing. Fringe pattern was seen to sweep the entire manifold surface indicating that the structure was merely moving about its mounts.

C2: Time average hologram of manifold moving about mounting brackets
C3: Resonance vibration node in the manifold at 106 Hz excitation frequency when acoustic mode of excitation was used.

C4: Resonance mode at 100 Hz. Acoustic mode of excitation.
C5: Interference fringes formed on the central plenum and on the runner extending upward. Excitation frequency 225 Hz. Acoustic mode of excitation.

C6: Interference fringes observed on the structure at 50 Hz.
C7: Manifold resonance at 96 Hz. Excitation with pneumatic valves.

C8: Manifold structure resonating at 100 Hz. Resonance extending through central plenum and extending into runner closest to throttle body entrance. Excitation with pneumatic valves.
C9: Manifold resonance at 96 Hz. Distant view through holographic film. Excitation with pneumatic valves.
Appendix D

Derivation of Equations for Vibration Calculations
D1: Vibration Measurement

The reconstruction of a time averaged hologram made of a vibrating object provides a vibration measurement. The amplitude of vibration can be calculated from equations derived as shown.

For derivation purposes, it will be assumed that the vibration of an object can be described with a steady state sinusoidal vibration of the form

\[ W = A(x, y) \cos(\omega t + \alpha) \]

Where \( A(x, y) = \) amplitude
\( \omega = \) circular frequency
\( \alpha = \) phase angle

The phase angle (\( \alpha \)) and frequency (\( \omega \)) are not detectable from the time average hologram, however the amplitude \( A(x, y) \) is.

The derivation that follows indicate what is recorded in the hologram and how it is recorded.

The hologram records the complex waveforms of the object and reference waves

\[ U_h = U_r + U_o \]

Where \( U_r = \) reference wave \( U_o = \) object wave

The light paths are represented by the undeformed and deformed paths as follows

\[ U_i = A_0 e^{i\theta} \] (undeformed)
\[ U_i = A_0 e^{i(\theta + \delta)} \] (deformed)

Where \( \delta = \) phase change due to path length change

For a time average hologram, delta is a function of time and therefore equals
\[ \delta = \frac{2\pi}{\lambda} (\text{cW}) = \delta(t) \]

\( U_o \) is therefore a function of time

\[ I_h = U_h U_{h}^* = [U_r + U_o(t)] [U_r + U_o(t)]^* \]

\[ I_h = I_r + I_o + U_r^* U_o(t) + U_r U_o^*(t) \]

When the object wave is not a function of time and assuming a linear recording in the photographic emulsion

\[ T = 1 + (\beta t) I_h \]

where \( T \) = transmittance

\( I_h \) is now a function of time and

\[ E = f(I(t)) = \int_0^t I(t) \, dt \]

The transmittance is also now time dependant with the time intensity varying

\[ T = 1 + \beta \int_0^t I(t) \, dt \]

where \( \tau = \) total exposure time

The image viewed in the hologram when the reconstruction takes place is as shown below

\[ U_i = [1 + \beta(I_r + I_o)\tau] U_c + \beta U_r^* U_o \int_0^t U_o(t) \, dt + \beta U_r U_o^* U_o(t) \, dt \]

Examining the primary image terms only

\[ U_i^* = \beta I_r \int_0^t U_o(t) \, dt \]

where \( U_o = A_o e^{i\phi} e^{i\delta} \)

\[ \delta = \text{phase change due to path length change} \]

The path length change is represented by

\[ \delta = \frac{2\pi}{\lambda} (\text{cW}) = \delta(t) \]
\[ C = \text{total path length} \]

\[ U_i^* = \beta I_r \int_0^t U_o(t)dt \]

Taking the previously developed equation

\[ U_i = [1 + \beta(I_r + I_o)\tau] U_c + \beta U_r^* U_o^* U_o(t) dt + \beta U_r U_o^* U_o(t) dt \]

Then grouping the D.C constants into \( k \)

\[ U_i^* = k^* e^{i\phi} \int_0^\tau e^{i(2\pi/\lambda)c A(x,y) \cos(\omega t + \alpha)} dt \]

If the exposure time (\( \tau \)) >> period (\( P \)) then fractions of periods can be omitted and \( \tau = nP \). Then the primary image waveform becomes

\[ U_i^* = k^* e^{i\phi} \int_0^{nP} e^{i(2\pi/\lambda)c A(x,y) \cos(\omega t + \alpha)} dt \]

Now \( P \) and \( \omega \) are related by

\[ P = 2\pi/\omega \]

Therefore

\[ U_i^* = k^* e^{i\phi} nP J_o(2\pi/\lambda) A(x, y) \]

where \( J_o = \) Bessel function of zero order

\[ J_o(\eta) = (1/2\pi) \int_0^{2\pi} e^{i \eta \cos e} de \]

The image observed is always a function of the irradiance, which is the square of the image waveform

\[ I = U_i U_i^* \]

\[ I = K^* J_o \left[ (2\pi/\lambda) C A(x, y) \right] \]
The intensity of the hologram is therefore some irradiance $I_o$ modulated by the Bessel function and $\theta$

$$I = I_o \, J_0^2(\theta)$$

where $J_0^2(\theta)$ produces the fringe pattern and

$$\theta = (2\pi/\lambda) \, C \, A(x, y)$$

The Bessel function is a function with decreasing amplitude with numerous zeros. The Bessel function produces fringes in the time average hologram. Because of the Bessel function, areas of zero displacement (nodes) are highly illuminated. Displaced areas show decreasing intensity with the dark fringes indicating zeros of the Bessel function.

The magnitude of vibration can then be calculated through

$$\theta = (2\pi/\lambda) \, C \, A(x, y)$$

or

$$A(x, y) = \frac{(\theta \lambda)}{(2\pi C)}$$

where $\theta$ is the corresponding zeros of the Bessel function.

The Bessel function of the first kind of order zero is shown in the following equation and is identified as the $J_0$ function.

$$J_0(x) = \left[ 1 - \frac{x^2}{2^2} + \frac{x^4}{2^2 \cdot 4^2} - \frac{x^6}{2^2 \cdot 4^2 \cdot 6^2} + \cdots \right]$$
Appendix D2 presents the zeroth order ($J_0$) Bessel function with the roots of the function shown in Appendix D3.

D2: zeroth order Bessel function.
<table>
<thead>
<tr>
<th>Bessel Root #</th>
<th>$J_0$ Root</th>
<th>$J_1$ Root</th>
</tr>
</thead>
<tbody>
<tr>
<td>$J_0$-1</td>
<td>2.405</td>
<td></td>
</tr>
<tr>
<td>$J_1$-1</td>
<td></td>
<td>0.0</td>
</tr>
<tr>
<td>$J_0$-2</td>
<td>5.520</td>
<td></td>
</tr>
<tr>
<td>$J_1$-2</td>
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<td>$J_0$-3</td>
<td>8.654</td>
<td></td>
</tr>
<tr>
<td>$J_1$-3</td>
<td></td>
<td>7.02</td>
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<td>$J_0$-4</td>
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<td></td>
</tr>
<tr>
<td>$J_1$-4</td>
<td></td>
<td>10.089</td>
</tr>
</tbody>
</table>

D3: Roots of $J_0$ and $J_1$ Bessel functions
Appendix E

Holographic Pictures - Initial Test Setup Phase
Resonating Tube
Resonating Plate
E1: Time average hologram of resonating tube. Holographic trial runs were conducted to observe resonant fringe patterns in the aluminum tube. The bright area indicates the vibration node.

E2: Time average hologram of resonating aluminum tube.
E3: Time average hologram of resonating plate.

E4: Aluminum plate resonance fringes.
REFERENCES


[10] [http://www.dupontautomotive.com](http://www.dupontautomotive.com). *The Case of the Hissing Manifold*