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An Investigation into the sources of vehicle tire noise

John Paoff

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An Investigation into the Sources of Vehicle Tire Noise

By

John M. Paoff

A Thesis Submitted in Partial Fulfillment of the Requirements for the

Master of Science In
Mechanical Engineering

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April, 2002
Permission Granted

Investigation of Tire Noise

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Date: 5/3/02 Signature of Author: ____________________________
Acknowledgments

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Thanks,

John
Abstract

Noise produced by a rolling tire on pavement has plagued automakers for years due to its complexity. Tire noise is dependent on many things such as tire material, tire construction, road surface texture, etc. In this investigation, an acoustical approach to modeling tire noise is presented. Based on the mechanics of a rotating tire, acoustical models can be developed. This, along with some acoustical analysis, leads to mathematical models that one can utilize in order to predict the noise that the tire will produce. These models will provide a good basis and starting point for reduction of tire noise and further modeling of tire noise.
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Chapter 1 - Introduction

In the last twenty years or so, tire noise, has been modeled through methods such as: vibration analysis, acoustical holography, and other techniques using various laboratory set-ups. Of all the methods out there isn’t any one theory that completely captures everything that is producing noise due to the complexity of the automobile tire.

Basic acoustical principles and measuring techniques allow one to determine a sound level of a source and to distinguish it from background or other simultaneously occurring noises. This is done by an addition and/or subtraction of sound pressure levels.

The goal of this work is to provide a basic understanding of acoustics and apply the principles to an everyday problem, tire noise. This objective was chosen because there is no complete mathematical or acoustical model for tire noise. This is not an attempt to create the perfect model, but through observations and basic acoustical principles, more pieces of the tire noise puzzle can be added.

Chapter 2 begins by going over the basics of sound; what it is, how it is measured, and some acoustical terminology. Chapter 3 goes into detail on how to manipulate, add, and subtract sound levels. Chapter 4 gives basic information on tire construction. Chapter 5 is about tires and elasticity. It covers how tires react under a load and its resulting deformations. Chapter 6 looks at work done in the past by M. Muthukrishnan [12] for SAE in 1990 titled “Effects of Material Properties on Tire Noise.” The author shows how noise levels go up with increasing speed and increasing weight. He also describes in detail how tire noise is affected by tire properties such as modulus of elasticity and Tan Delta. It shows how different combinations of tread and sidewall modulus will affect tire
noise levels. Chapter 7 is on the paper "Investigation into the Influence of Tire Construction on Coast-by Noise" by Don Brackin, Nishuhata, and Sauer Zapf [18]. The main points made in their investigation are that tire tread vibration is greatest near the contact patch of the tire. It is also concluded that shoulder tread vibration is the noise source that best correlates to measured tire noise. They also state how lowering the center contact pressure and or increasing the shoulder tread bending stiffness will decrease the tire tread vibration and reduce tire noise generated. Chapter 8 covers the relationship between road texture and tire noise. Written in a conference paper from Noise-Con 96 by Yasuo Oshino and Hideki Tachibana [7], they conclude that tire/road noise increases with an increase of road texture depth. Chapter 9 is on Plotkin, Montroll, and Fuller’s [4] study on noise sources that are consistent with air pumping. They describe how sound pressure due to air pumping is directly related to the second derivative of the volume of air displaced from tread voids. Chapter 10 goes into the methods of measuring tire noise that are commonly used. These methods include "The Coast By Method," "The Trailer Method," and "The Laboratory Drum Method." Chapters 11 – 14 are where the mathematical modeling through observation and experimentation is presented. Chapter 11 discusses tire patch frequency and as tire patch frequency increases so does the sound level. Chapter 12 is about tire bulge and its effect on tire noise. Chapter 13 takes you through the development of the rubbing theory, which is then backed up through experimentation. Chapter 14 concludes this investigation and gives future recommendations.
Chapter II - General Discussion of Sound

Sound can be described as a disturbance spreading through a physical medium, such as air. The ear perceives it as a pressure wave superimposed upon the ambient air pressure to the listener. The sound pressure is therefore the incremental variation about the ambient atmospheric pressure. To describe these pressure waves we call sound, mathematically, it is best to look at the attributes of a pure tone. A pure tone is a sinusoidal pressure wave of a specific frequency and amplitude, propagating at a velocity determined by the temperature and pressure of air.

A hypothetical sound generator, as described by Irwin and Graf, in *Industrial Noise and Vibration Control*, is shown below:

![Figure 1](image1)

**Figure 1 - [17]**

The source may be thought of as an elastic sphere, like a balloon, that expands and contracts sinusoidally at a frequency $f$. As the balloon expands, the surrounding air molecules are compressed. When the balloon contracts, the air molecules spread apart; the gas is rarefied. The sound wave generated will have a frequency equal to the number of times per second in which the balloon expands and contracts. The Amplitude of the maximum pressure is a function of the maximum expansion or excursion of the sphere.
The frequency, $f$, of an oscillating disturbance is equal to the number of times per second that the disturbance passes through both its positive and negative excursions. The number of cycles per second is termed **Hertz** (cycles/sec). The frequency of a simple pure tone sound wave is recognized as the pitch of the tone.

The period, $T$, of the sinusoidal wave is the time required for one complete cycle, and is related to the frequency, $f$, by:

$$T = \frac{1}{f} \quad [17] \text{pg. 3}$$

The wavelength, $\lambda$, is the distance between like points on two successive waves. The wavelength is related to the frequency and velocity of propagation by:

$$\lambda = \frac{c}{f} = cT \quad [17] \text{pg. 3}$$

in which the velocity of propagation, $c$, is in turn a function of the characteristics of the propagation-supporting medium. The velocity of propagation or speed of sound in air is given by the equation below:

$$c = \sqrt{\frac{\gamma \cdot P_a}{\rho}} \quad [17] \text{pg. 3}$$

Here $\gamma$ is **specific heat** at constant pressure over specific heat at constant volume.

$$\gamma = \frac{\text{specific heat (c.p.)}}{\text{specific heat (c.v.)}} \quad [17] \text{pg. 3}$$

$P_a = \text{Ambient or equilibrium pressure}$

$\rho = \text{Ambient or equilibrium density}$

For air $\gamma = 1.4$ (Assuming air is an ideal gas)
c = 49.03*(R)^{1/2} \text{ where } R \text{ is the temperature in degrees Rankin}

or \ c = 20.05*(K)^{1/2} \text{ where } K \text{ is the temperature in degrees Kelvin}

The \textit{sound power level} describes the acoustical power radiated by a given source with respect to the international reference of 10^-12 W. The \textit{sound pressure level} is proportional to the logarithm of the ratio of pressures squared. This is important in that the pressure squared is proportional to some sound power; thus, both the sound power level and sound pressure level are associated with power. Sound powers and sound pressures are commonly expressed in a logarithmic, rather than a linear scale called \textit{decibels}. The decibel is the logarithm, to the base 10, of the ratio of the quantity in question, to an arbitrarily chosen reference quantity. The argument of the logarithm is dimensionless.

\begin{equation}
\text{Level} = 10 \log \left( \frac{Z}{Z_O} \right) \tag{17} \text{ pg. 6}
\end{equation}

\(Z\) = quantity in question

\(Z_O\) = chosen reference quantity

The sound power level, \(L_w\), is defined as:

\begin{equation}
L_w = 10 \log \left( \frac{W}{W_re} \right) \tag{17} \text{ pg. 6}
\end{equation}

\(W\) = power in question

\(W_re = 10^{-12}\) watt

The sound pressure level, \(L_p\), expressed in decibels is:

\begin{equation}
L_p = 10 \log \left( \frac{P^2}{P_{re}^2} \right) \tag{17} \text{ pg. 8}
\end{equation}

\begin{equation}
L_p = 20 \log \left( \frac{P}{P_{re}} \right) \tag{17} \text{ pg. 8}
\end{equation}

\(P\) = Root-mean-square (RMS) sound pressure in question (Pa. or N/m^2)

\(P_{re} = \text{International Reference Pressure of } 20*10^{-6}\) Pa.
The human ear cannot respond to all frequencies in an unbiased manner. *Audible range* of a human being ranges from 20 Hz to 20,000 Hz, which in fact will vary with age, health, past exposure to noises, and so forth. The ear also acts like a *filter* and will favor certain frequencies over the others. The ear is most sensitive to sounds at the range of 1,000 to 5,000 Hz, and particularly at about 4,000 Hz. The perceived sound pressure level or loudness is frequency dependent.

Figure 2 - [19]
Chapter III - Sound Measure

The understanding of noise problems commonly demands that pressures and powers be manipulated by means of decibel additions and subtractions. Sound Pressure levels (in decibels) are averaged in the calculation of source directivity. For adding and subtracting and averaging decibels, quantitative analysis are used as well as charts and approximations. In this investigation, the appropriate quantitative analysis will be concentrated on.

Sound power levels are commonly added when determining the total sound power level of a source. Because noise can be random with respect to phase measurements, it is added on an energy basis. Assume the sound pressure levels \(L_{p1}, L_{p2}, L_{p3}, \ldots, L_{pi}\) are to be added. The sound pressure level by definition, is:

\[
L_{pi} = 10 \log \left( \frac{P}{P_{re}} \right)^2 \tag{17} \text{pg. 10}
\]

Where \(P\) is the root mean square (rms) sound pressure in question (Pa or N/m^2) and \(P_{re}\) is the International reference pressure of \(20 \times 10^6\) Pa or .0002 \(\mu\)Bar.

The next step is to determine the square of the pressure ratio;

The total sound pressure \(L_{pt}\), is simply

\[
\left( \frac{P}{P_{re}} \right)_{i}^{2} = \text{antilog} \left( \frac{L_{pi}}{10} \right)
\]

\[
L_{pt} = 10 \log \left( \sum_{i=1}^{n} \left( \frac{P}{P_{re}} \right)_{i}^{2} \right)
\]
Or in terms of sound pressure levels,

\[ L_{Pt} = 10 \log \left( \sum_{i=1}^{n} \text{anti} \log \left( \frac{L_{pi}}{10} \right) \right) \]

Simplifying further,

\[ L_{Pt} = 10 \log \left( \sum_{i=1}^{n} 10^{\frac{L_{pi}}{10}} \right) \quad [17] \text{pg. 10} \]

The expression for sound power levels may be expressed as

\[ L_{wi} = 10 \log \left( \sum_{i=1}^{n} 10^{\frac{L_{wi}}{10}} \right) \quad [17] \text{pg. 10} \]

Where \( L_{wi} = \text{total sound power} \)

\( L_{wi} = \text{i}^{th} \text{sound power level} \)

In many cases, it is desired to subtract background or ambient sound pressure level from a total measured level to obtain the sound pressure level produced from a single source. The procedure for subtracting decibels is similar to that of addition.

The total sound pressure in decibels is

\[ L_{Pt} = 10 \log \left( \frac{p}{p_{re}} \right) \quad [17] \text{pg. 13} \]

In terms of the mean-square pressure ratio,

\[ \left( \frac{p}{p_{re}} \right)_{t}^2 = \text{antilog} \left( \frac{L_{Pt}}{10} \right) = 10^{\frac{L_{Pt}}{10}} \]
The background or ambient noise may be represented by:

\[ L_p = \text{sound pressure level of the ambient or background noise}. \]

The sought-after sound pressure level of the ambient or background noise in decibels is:

\[ L_r = 10 \log_{10} \left( \frac{P}{P_0} \right) \]

When averaging decibels it follows directly from the summation that,

\[ L_p = 10 \log_{10} \left( \frac{1}{n} \sum_{i=1}^{n} L_i \right) \]

the average decibel level, \( L_p \), is determined by dividing the sum by the number of levels, that is,

\[ L_p = 10 \log_{10} \left( \frac{1}{n} \sum_{i=1}^{n} L_i \right) \]
Chapter IV - Tire Construction

Pneumatic tires serve three main purposes. They support the weight of a vehicle, absorb road surface irregularities, and provide traction on the road. Tires have a toroidal shape and are usually filled with compressed air. The carcass of the tire provides the structural support for the tire. The carcass is made up of many flexible filaments of high modulus cord, embedded in and bonded to a matrix of low modulus material, usually rubber. The chords of the tire are made of natural textile, synthetic polymer, glass fiber, or fine hard drawn steel.

The chords are anchored on the beads of the tire, which are high tensile steel wires that seat on the rim of the tire. The beads serve as a foundation for the carcass and provide it

Figure 3 Filament arrangements that are used in pneumatic tires. a) woven cord  b) weftless cord  c) Cord with light wefts [1] pg.360

Figure 4 Essentials of bead construction  (a) Low turn-up construction  (b) high turn-up construction  (c) Overlap construction  (d) Detail of typical bead [1] pg. 361
with adequate seating on the rim. The material in which the beads are incased in is pressed against the flange of the rim by inflation pressure.

Most of the tire’s vibrations and deformation occur in the sidewall. The tread of the tire is made of various types of rubber depending on the tire application. When the tire is inflated with air, the pressure causes tension in the chords within the carcass. Load from the weight of the automobile placed on the rim to the wheel, hangs primarily on the chords in the sidewalls through the beads. The chords or plies run at an angle from the centerline (circumferential center of treads) of the tire. The number of layers is determined by the tire type, the tire size, and the inflation pressure to be used. A typical tire will have from 2-20 plies with each layer running in opposite directions. This angle is called the **crown angle**. This angle plays a role in the ride and handling of the tire.

There are two basic types of tires, *Bias ply* and *Radial ply*. In a Bias ply tire, the chords extend diagonally across the carcass from bead to bead with a crown angle of about 40 degrees. When the tire is rolling, the diagonal plies flex and rub, this elongates the diamond shaped elements formed by the chords and the rubber filler. This flexing action produces a wiping motion between the tread and the road.
In a radial ply tire, there is one or more layers of chords extending radially from bead to bead, resulting in a crown angle of 90 degrees. Under the tread, at a low crown angle of about 20 degrees, are fitted several layered belts made of high elasticity material, usually steel. All together, there are two radial plies of rayon or polyester, two plies of steel cords, and two plies of synthetic material like nylon. A radial ply tire has a relatively uniform ground pressure distribution under the contact patch (no wiping motion). The ground pressure for a bias ply tire varies greatly from point to point as tread elements passing through the contact region go through a complex wiping motion.
Chapter V  Tires and Elasticity

In *Mechanics of Pneumatic Tires*, by S K. Clark [1] (section 3.8), the mechanism of load carrying of a toroidal or tire-like structure of an infinitely flexible membrane with a rigid tubular rim for the central zone or bore of the toroid is discussed, see figure below.

![Figure 7 Toroidal membrane on cylindrical rim](image)

It is assumed that the junction between the thin flexible membrane and the rigid tubular rim or base has zero bending rigidity. Inflation of the structure puts tensions in the membrane and it takes a shape as determined by equilibrium and compatibility conditions. The membrane tensions are resisted by reactions at the edge of the tubular rim. For the present purpose, these can be discussed in terms of two components; radially outward tension and tension in an axial direction (that is paralleled to the axis of rotational symmetry or rotation of surface generators) at each point around the edge of the rim. If a flat plate is pressed against the membrane while the structure is supported by the rim a reaction will develop between membrane and plate where the load will be equal to the product of the actual contact area and the inflation pressure.
As seen in the figure below, the curvature of the wall of the membrane increases in the region between the loading plate and the adjacent rim.

![Figure 8](image)

**Figure 8** Perspective sketch of toroidal shell contact [1] pg. 397

Hence, because of the increased curvature, the membrane stresses in this region are lower than elsewhere in the membrane walls. The deflection also causes the membrane to distort locally, increasing the angle between the direction of the wall and a line normal to the plate from the rim; this is true whatever the cross sectional shape, see figure below.

![Figure 9](image)

**Figure 9** Cross sections of Figure 7 showing deflections of sidewalls which reduce the tension component radially outward at the inner cylinder edge [1] pg. 397

This increase of angle reduces algebraically the cosine of the angle between the wall and the line of action of the applied load on the plate.
The net effect of the reduced tension and reduced component at the deflected region is to develop the required reaction. In effect, the rim hangs in the tensions of the undeflected walls as shown, see figure below.

![Diagram](image)

**Figure 10** Polar plot of radially outward component of wall tension of membrane toroid on inner cylinder [1] pg. 397

The radially outward components of the wall tensions are greater in the undeflected regions than in the deflected region.

The useful information we get from this is that the reduced tension in the deflected region causes the stiffness in the shoulder region to drop. Decreasing pressure on a tire will increase the amount the tire deflects, this deflection will increase the bulge of a tire. As a result, tire bulge increases with tire deflection, which means there will be less tire stiffness in the deflected region (the tire shoulder) which means there will be more vibration of the shoulder treads producing more noise. The elastic effects of the tire cause this to happen. The deflected sidewall will snap back into shape upon leaving the contact patch producing vibrations throughout the tire.
From Samuel K. Clark's, *Mechanics of Pneumatic Tires*, November 1971 [1], Figure 10, shows the rotation of a wheel transmitting torque, $M_t$. As a result of the torque transmitted through the wheel, two sets of forces act upon it. One is the reaction of the wheel axis $P_k$, and equal to it and in the opposite direction, the reaction of the road acting in the plane of contact. As a first approximation, it may be assumed that the reaction of the road $P_k$ is evenly distributed over the area of contact. The component of tangential stress from $P_k$ is denoted as $\tau_p$.

![Figure 11 Rotation of a Driving Wheel (the Distribution of Longitudinal Tangential Stress in the Contact Region of Driven, Driving and Braked Wheels). [1] pg. 490](image)

(1) total component of longitudinal tangential stress, $\tau_m$ Due to driving torque, (2) component of longitudinal tangential stress, $\tau_m$ Due to braking torque, (3) distribution of longitudinal tangential stress along the contact length of driving wheel; (4) ditto, for a free wheel; (5) ditto, for a braked wheel. [1] pg. 491
There is a certain amount of adhesion over the contact region. The tire possesses longitudinal tangential elasticity, allowing the torque of the tire will compress the tread elements in the zone immediately before the contact region (-) and at the same time will stretch the elements in the area just after the contact region (+). Again looking at Figure 11, an initially compressed element $\Delta x$, of the tread is released from longitudinal compression as it passes through the contact area, reverting to its normal state $\Delta x_1$, then it undergoes stretching and emerges from the contact region in a stretched state $\Delta x_2$. Since the elements, as they pass into the contact area, are in direct contact with the road, any change in their dimensions is prevented by the force with which element grips the road surface, and longitudinal tangential stress $\tau_k$ arise in the plane of contact. These stresses cause there to be an area of slippage at the rear of the contact patch due to the tension pulling the treads out of the contact patch.

Figure 12 Displacement of tread elements along contact length of tire: (a) free rolling, (b) driving, (c) braked. [1] pg. 465

Zone I Longitudinal tangential stress acting from tire to roadway in direction of motion.
Zone II Longitudinal tangential stress acting from tire to roadway opposite to direction of motion.
Chapter VI - Influence of Material Properties on Tire Noise

In the article “Effects of Material Properties on Tire Noise” by M. Muthukrishnan [12], SAE, 1990, the results of an experimental study to determine the effect of material properties on tire noise are discussed. The properties they use cover a wide range of moduli and tan delta for tread and sidewall. An explanation of tan delta is located on page 25. The tires were tested at different speeds, loads and inflation pressures. From this, they obtained overall noise levels and frequency content. The tests indicate a larger influence of tread modulus on tire noise, and it was observed that the interaction between tread and sidewall properties affect tire noise levels significantly.

The results of this report are presented in two parts: (1) the effects of load, pressure, and speed on tire noise, and (2) Material property effects on tire noise. The results of varying the load show that load changes do not affect noise level in any significant way. The testing was done at five different speeds, and five different microphone locations.

Figure 13  microphone locations around the tire [12] pg. 3
The changes in noise levels were at most + - 2 dBA. Next, pressure effects on tire noise, were shown by comparing noise levels at two different pressures for different loads and speeds. At the lighter load, an increase in 15 psi led to an increase in noise levels of about one to three dBA. At the larger load, there is a reversal and the increase in inflation pressure is accompanied by a decrease in the noise level by as much as two dBA.
The effect of speed on noise level is shown in the figure below. With increasing speed, the noise level generally increases. There is a large increase from 30 to 40 mphs at both loads. After 40 mph, the increase is not uniform and is even reduced at certain speed ranges.

![Tire Noise Speed Effects](image)

**Figure 16 Speed Effect on Tire Noise** [12] pg. 5

For the material property parameters on tire/road noise, a wide range of moduli and tan delta are used. Nine groups of tires were built, these groups can be seen in the Table below.
Experimental Design  
(Fractional Factorial)

<table>
<thead>
<tr>
<th>The Group ID</th>
<th>Modulus</th>
<th>Tan Delta</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Tread</td>
<td>Sidewall Package</td>
</tr>
<tr>
<td>A</td>
<td>Control</td>
<td>Control</td>
</tr>
<tr>
<td>B</td>
<td>Low</td>
<td>Low</td>
</tr>
<tr>
<td>C</td>
<td>High</td>
<td>Low</td>
</tr>
<tr>
<td>D</td>
<td>Low</td>
<td>High</td>
</tr>
<tr>
<td>E</td>
<td>High</td>
<td>High</td>
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<td>F</td>
<td>Control</td>
<td>Control</td>
</tr>
<tr>
<td>G</td>
<td>Control</td>
<td>Control</td>
</tr>
<tr>
<td>H</td>
<td>Control</td>
<td>Control</td>
</tr>
<tr>
<td>I</td>
<td>Control</td>
<td>Control</td>
</tr>
</tbody>
</table>

**Figure 17 Tire Groups** [12] pg. 2

There is one control group, A, four groups (B, C, D, and E) with varying moduli, and four more groups (F, G, H, and J) with varying tan delta and constant moduli for both the tread and sidewall.

Looking at tan delta, the elastic modulus is $E'$, and the $E''$ is the loss modulus. The two cases can be seen in Figure 18.

<table>
<thead>
<tr>
<th>Varying Elastic Modulus/Constant Tan Delta</th>
<th>Varying Tan Delta/ Constant Elastic Modulus</th>
</tr>
</thead>
</table>

**Figure 18 Tan Delta and Elastic Modulus** [12] pg. 2
In Figure 18, $E'$ is the **Elastic Modulus** or **Storage Modulus**, $E''$ is the **loss modulus**, $E^*$ is the **complex modulus**, $\delta$ is the **loss angle**, and 1,2 are the two conditions. All materials have a viscoelasticity, which is a combination of viscosity and elasticity in varying amounts. When viscoelasticity is measured dynamically, there is a phase shift between the force applied as stimulus (stress) and the strain (skew) which occurs in response.

Generally, the measurement results are represented as a complex elasticity modulus to insure accurate expression. This relationship is shown below.

If the relationship between $E^*$ and $\tan \delta$ is plotted, the result, is a graph like the one shown below.

\[
E^* = E' + iE'' \\
\tan \delta = \frac{E''}{E'}
\]

[12] pg. 6

---

**Figure 19**
From the results of the modulus testing it can be seen that tread modulus has a larger influence on overall noise levels than the sidewall modulus. An increase in tread modulus when averaged over all the varying conditions caused noise level increases from 2 to 7 dBA, and an increase in sidewall modulus caused a noise level increase from 1 to 3 dBA.

<table>
<thead>
<tr>
<th>Speed</th>
<th>Mic. No.</th>
<th>Tread Modulus</th>
<th>S.W. Modulus</th>
<th>Interactions</th>
</tr>
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<tr>
<td></td>
<td></td>
<td>Low</td>
<td>High</td>
<td>Low</td>
</tr>
<tr>
<td>60 MPH</td>
<td>1</td>
<td>82.7</td>
<td>89.5</td>
<td>86.1</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>82</td>
<td>87.5</td>
<td>84.7</td>
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<td>87.2</td>
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<td>84.5</td>
<td>81.3</td>
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<td></td>
<td>5</td>
<td>80.1</td>
<td>82.6</td>
<td>80.7</td>
</tr>
<tr>
<td>45 MPH</td>
<td>1</td>
<td>80.1</td>
<td>85.9</td>
<td>82.4</td>
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</tr>
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<td>73</td>
<td>76.4</td>
<td>74.1</td>
</tr>
</tbody>
</table>

avg. of B&D avg. of C&E avg. of B&C avg. of D&E avg. of B&E avg. of C&D

Figure 20 Modulus Effects on Tire Road Noise (dBA) [12] pg. 6
This experiment also shows that there is a cross coupling between tread and sidewall effects. For example, going from low to high tread modulus with low sidewall modulus increases noise level by five dBA. Where as doing the same with a high sidewall modulus produces an increase of 8.6 dBA. The influence of tread modulus on sidewall modulus effects are shown in the last two columns of Figure 17:

<table>
<thead>
<tr>
<th>Speed</th>
<th>Mic. No.</th>
<th>Low to High Tread Modulus</th>
<th>Low to High Sidewall Modulus</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Low S.W.</td>
<td>High S.W.</td>
</tr>
<tr>
<td>60 MPH</td>
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<td>5.0</td>
<td>8.6</td>
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<tr>
<td></td>
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<tr>
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<td>3</td>
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<td>6.2</td>
</tr>
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<td>7.6</td>
</tr>
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<td></td>
<td>2</td>
<td>2.9</td>
<td>7.1</td>
</tr>
<tr>
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<td>3.3</td>
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<td></td>
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</tr>
<tr>
<td></td>
<td>5</td>
<td>1.6</td>
<td>5.0</td>
</tr>
</tbody>
</table>

Figure 21 Interaction Effects Change in Noise Level (dBA) [12] pg. 7
The noise level spectrum was then integrated and plotted it as a function of frequency.

![Integrated Spectrum](image)

Figure 22 Integrated Spectrum [12] pg. 8

The integrated spectrum at a given frequency gives the total noise contribution up to that frequency starting from the frequency of interest. This is done for different moduli combinations for a speed of 60 mph.
It can be seen from this that in the frequency range from 750 to 1500 Hz, the contributions vary significantly depending on the particular tread/sidewall modulus combination. In this frequency range, many noise-generating mechanisms are involved. The major sources of noise are said to be (a) tread patterns, and (b) radial and tangential vibrations of the tread elements at the entry and exit of the contact patch. It is also described in this report that vibrations generated at the contact patch depend on stiffness (modulus) and the damping of the tread elements. It is then reasoned that high modulus tread blocks produce larger levels of vibration, resulting in more noise from 750 to 1500 Hz. It also is observed that a high modulus sidewall amplifies the vibrations of the high modulus tread elements more effectively. This explains the cross coupling effect observed in Figure 15 & 16.

Figure 29, shows the overall effects of tread and sidewall tan delta on tire noise for all testing conditions. It can be seen that the tan delta of either the treads or sidewall has negligible effect on overall tire noise levels.
Muthukrishnan concludes the following from his experiments:

1. Tread modulus has a much larger influence on exterior tire noise level than sidewall modulus.
2. Tan delta of either tread or sidewall has the least effect on tire noise level.
3. Significant interactions exist between tread and sidewall properties. Noise levels depend on the tread and sidewall conditions together.

The next few pages include other interesting graphs from "Effects of Material Properties on Tire Noise" by M. Muthukrishnan [12].
Figure 24, 25 Tread Tan Delta Effects on Tire Noise, Sidewall Tan Delta Effects on Tire Noise [12] pg. 9
### Pressure Effects on Tire Noise (dBA)

#### Roadwheel Testing

Load: 636 Lbs.

<table>
<thead>
<tr>
<th>Speed (MPH)</th>
<th>Microphone Location 1</th>
<th>Microphone Location 2</th>
<th>Microphone Location 3</th>
<th>Microphone Location 4</th>
<th>Microphone Location 5</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
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<td>35 psi</td>
<td>20 psi</td>
<td>35 psi</td>
<td>20 psi</td>
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<tr>
<td>30</td>
<td>75.4</td>
<td>76.6</td>
<td>73.5</td>
<td>74.5</td>
<td>73.8</td>
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<td>78.6</td>
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</tr>
<tr>
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</table>

#### Roadwheel Testing

Load: 1190 Lbs.

<table>
<thead>
<tr>
<th>Speed (MPH)</th>
<th>Microphone Location 1</th>
<th>Microphone Location 2</th>
<th>Microphone Location 3</th>
<th>Microphone Location 4</th>
<th>Microphone Location 5</th>
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</thead>
<tbody>
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<td></td>
<td>20 psi</td>
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<td>20 psi</td>
<td>35 psi</td>
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<td>81.4</td>
<td>80.8</td>
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<td>85.3</td>
<td>85.4</td>
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<td>83.2</td>
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</table>

Figure 26 Pressure Effects on Tire Noise (dBA) [12]

### Testing at 12 Hz.

<table>
<thead>
<tr>
<th>Item</th>
<th>Control</th>
<th>Low</th>
<th>High</th>
<th>Control</th>
<th>Low</th>
<th>High</th>
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<tr>
<td>Tread</td>
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<td>938</td>
<td>5048</td>
<td>0.22</td>
<td>0.11</td>
<td>0.265</td>
</tr>
<tr>
<td>Sidewall</td>
<td>100</td>
<td>530</td>
<td>2504</td>
<td>0.144</td>
<td>0.123</td>
<td>0.218</td>
</tr>
<tr>
<td>Rimstrip</td>
<td>2790</td>
<td>1444</td>
<td>8250</td>
<td>0.292</td>
<td>0.189</td>
<td>0.327</td>
</tr>
<tr>
<td>Bead Filler</td>
<td>5350</td>
<td>3901</td>
<td>23555</td>
<td>0.197</td>
<td>0.037</td>
<td>1.14</td>
</tr>
</tbody>
</table>

Figure 27 Material Property Values [12]
## Tire Noise (dBA) Levels for all Modulus and Tan Delta Combinations: Roadwheel Testing

Inflation Pressure: 35 psi

Load: 1190 Lbs.

<table>
<thead>
<tr>
<th>Speed (MPH)</th>
<th>Mic. No.</th>
<th>Control (A)</th>
<th>Modulus Combinations Low Tread</th>
<th>Modulus Combinations High Tread</th>
<th>Modulus Combinations Low Tread</th>
<th>Modulus Combinations High Tread</th>
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### Figure 28 Tire Noise, All Combinations [12]

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<th>Speed (MPH)</th>
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<th>Sidewall Tan delta</th>
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<td></td>
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<td>Extreme</td>
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<td></td>
<td></td>
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</table>

### Figure 29 Tan Delta Effects on tire Noise (dBA) [12]
Chapter VII - Tire Construction and Noise

In the article "Investigation into the Influence of Tire Construction on Coast-by Noise" by Doan, Brackin, Nishihata, and Sauer Zapf [18], the dominant source of tire noise is identified as shoulder tire tread vibration. It is generally known that a high peak value exists at around one kHz. In the coast-by noise spectrum and this frequency dominates the coast-by noise level. This phenomenon occurs in tires having treaded patterns and also for blank tires. Rib-style tread patterned tires are usually quieter and show the smallest difference in noise levels as compared to blank tires. Tire construction and materials will have to be taken into account to reduce noise levels. Noise is not only generated by tread patterns, but also by the vibration of the tire components.

![Figure 30 Excitation Level of Treads as They Pass Through the Contact Region, Typical Radial Tread Part time History [18]](image-url)
The vibrations were measured with accelerometers mounted at the center of the belt, near the belt edge, and the sidewall of the tire. The vibrations at the center and belt edges are referred to as 'tread part' vibrations. Looking at Figure 31 below, at 1 kHz it can be seen that the leading and trailing edges are noise sources. Results show that tire noise around 1 kHz is generated by tread part vibration at the leading edge, trailing edge, and shoulder treads, and that the acceleration of the tread part is greater than that of the sidewall especially around 1 kHz. The following graphs show that tire tread band vibration does in fact generate noise sources related to the sound produced by coast-by tests.

Figure 31 Graphs Showing Correlation Between Coast-By Noise and tread Vibration Levels [18]
The correlation coefficients that they reported for the shoulder tread vibration and the generated tire noise is relatively high, as compared to the center tread vibration and the generated tire noise.

<table>
<thead>
<tr>
<th>Table 1 - Correlation between vibration at the shoulder tread and coast-by noise/indoor drum tire noise</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coast-Drum Noise</td>
</tr>
<tr>
<td>40 km/h</td>
</tr>
<tr>
<td>Center Tread Vibration</td>
</tr>
<tr>
<td>0.442</td>
</tr>
<tr>
<td>Shoulder Tread Vibration</td>
</tr>
<tr>
<td>0.441</td>
</tr>
</tbody>
</table>

**Figure 32 Correlation between vibration at the shoulder tread and coast-by noise/indoor drum tire noise [18]**

They found high acceleration levels at the leading and trailing edges and relatively low levels in the contact region. They also say, “The contact pressure and tire stiffness may be used to describe this system of vibration, because the contact pressure can be thought of as a force which acts on the tire, and the amplitude of tire vibration can thought of as being dependent on tire stiffness. This means that tread vibration is equal to the input forces multiplied by some vibration transfer properties.” The tread vibration mechanism model could be explained in detail as follows: The input force can be seen in Figure 33 below.
Figure 33 Tire Forces at the Leading and Trailing Edges caused by Contact Pressure Variation and Road Surface Roughness [18]

The dynamic pressure increases at the leading edge (AB), stays nearly constant during contact (BC), and then decreases at the trailing edge (CD). This contact distribution can be assumed to be representative of the forces that excite the tire structure. The amplitude of the maximum dynamic contact pressure around B and C can be considered as the maximum amplitude of the excitation forces present. In addition, the road surface roughness tends to amplify the excitation forces and must be considered. The tire in Figure 33 is approximated by the spring mass damper and step input system of Figure 34.

Figure 34 A typical tire transfer function and system used to represent tire properties a) the transfer function measured as shown by the graph insert. b) Spring, Mass, and Damper System with Step Input Function. [18]
Where the maximum dynamic contact pressure, at the leading and trailing edges, represents the input forces. If this hypothesis is used, the displacement response step function \( x(t) \) in the time domain can be expressed as:

\[
x(t) = F \left( \frac{1}{K} \right) \left[ 1 - e^{-\zeta \omega_n t} \left( \cos \left( \omega_n \sqrt{1 - \zeta^2} t \right) + \frac{\zeta}{\sqrt{1 - \zeta^2}} \sin \left( \omega_n \sqrt{1 - \zeta^2} t \right) \right) \right]
\]

[18]

Where:

\( x(t) \) = displacement response

\( F \) = input force

\( K \) = stiffness

\( \zeta \) = damping ratio

\( \omega_n \) = natural radial frequency

Due to the difficulty of calculating the damping ratio and the mass effects, they further simplify the model by assuming the displacement response is only proportional to the input force multiplied by the reciprocal of the stiffness, as shown in the equation below:

\[
x(t) \propto F \frac{1}{K}
\]

[18]

In this equation, \( F \) represents the input force and \( 1/K \) is the transfer function. Further the input force, \( F \), can be approximated by using the maximum dynamic contact pressure and the road surface roughness, and can be represented in the relation below:
\[ x(t) \approx \alpha P \frac{1}{K} \]  

[18]

In which:

\( \alpha \) = road roughness coefficient

\( P \) = dynamic contact pressure

\( K \) = tire stiffness

The authors also say that the center contact pressure and the tread bending stiffness at the shoulder have the most influence on the shoulder tread vibration. Therefore, having a tire with a lower contact pressure around the center and an increased tread bending stiffness at the shoulder should reduce the shoulder tread vibration, thus reducing the noise it produces. The tire tread vibration on the shoulder was obtained using linear regression analysis.

**Figure 35** Calculated correlation values table and graph showing actual versus estimated tread vibration for a speed of 60 km/hr.
and results in the following equation:

\[ x(t) = a_0 + a_1 \cdot P_{ce} \cdot \frac{1}{K_{sh}} \]  

[18]

in which:

- \(a_0, a_1\) = single regression constants
- \(P_{ce}\) = static pressure around center
- \(K_{sh}\) = tread bending stiffness around the shoulder

Based on their research, they made the following conclusions:

1. The region of the tire in the vicinity of the contact patch dominates tire tread vibration.
2. The A-weighted spectra of tread part acceleration, coast-by noise, and indoor drum test noise display the same tendency (a high peak around 1 kHz.).
3. Shoulder tread vibration is the noise source that has the most relation to tire noise produced.
4. Lowering the center contact pressure and/or increasing the shoulder tread bending stiffness will decrease the tire tread vibration and reduce the noise generated.
Chapter VIII – Section 1: Road Texture and Tire/Road Noise

In the conference paper “Relationships between Road Texture and Tire/Road Noise” from Noise-Con 96, Yasuo Oshino and Hideki Tachibana [7], performed a study of noise radiated from a passenger car and a medium sized truck and different road surfaces (paving materials, chipping sized grain and surface texture). From this investigation, relationships between tire/road noise and road surface characteristics were developed. In this study, two kinds of measurements were made. The first one was performed on a test track paved with five kinds of constructions by using a passenger car and a medium sized truck. Therefore, it has been confirmed that the sound power spectrum of tire/road noise varies quite a bit due to differences in the road surface materials. They also stated that the open graded asphalt surface is the best among all tested to reduce the tire/road noise.

Tire/road noise was also measured at sixteen sites of public roads paved with dense asphalt concrete by using the same passenger car equipped with four types of tire and the same medium sized truck equipped with two types of tire. They found that the general tendency implied that tire/road noise increases with the increase of texture depth. This has been found for almost all tires but the relationship varies due to the type of tire.

These results suggest that there is a greater exciting force on the tread bands at the leading and trailing edges. So the deeper the texture depth, the more the tread bands get excited, and hence the greater the noise produced.
Chapter VIII – Section 1: Air Pumping as a Noise Source

Another source of tire noise is air pumping. Air pumping, occurs when air movement from tread voids give rise to monopole sound radiation. It is acoustically a local source, with sound radiated directly as the void compresses. This is validated as a sound source in the article “The Generation of Tire Noise and Carcass Vibration”, Plotkin, Montroll, Fuller, Internoise-1980 [4]. In their study, they found that there are concentrated noise sources consistent with air pumping. These were identified at the entrance to the contact patch. The sound pressure due to air pumping is directly related to the second derivative of the volume of air displaced from tread voids. The void profiles were directly measured for the test tires. This was accomplished by measuring the volume of water displaced from a bladder in the void as the tire was advanced through the contact patch. The measured profiles were then differentiated numerically. The calculated results showed good agreement with the measured sound pressure. The calculations strongly suggest that the concentrated sources observed were due to air pumping.
Chapter IX - Methods of Measuring Tire Noise

Tire/pavement interaction produces a non-uniform noise radiation. There are three areas in which noise is radiated; they are the leading edge, trailing edge, and sidewall regions, near the contact region (see Figure 31).

![Figure 36](image)

There are three prominent methods for measuring tire/road noise: The Coast by Method, The Laboratory Drum Method, and The Trailer method.

The Coast by Method is the most representative of actual field operating conditions and sound propagation to the road environment. In this method, the test vehicle coasts by a roadside microphone, which is placed 1.2m above the road level and 7.5 m from the centerline of the vehicle’s path. The engine is switched off. Using a time constant nicknamed fast and a frequency rating “A,” the maximum sound level during the coast-by is recorded. It is recommended that the frequency spectrum also be recorded at the maximum sound level, although this is not mandatory. Usually, five runs are made and averaged. As for all methods, the recommended speed is 70 km/hr. If lower or higher
speeds are required, it is recommended that they be chosen from 30, 50, 90, or 110 km/hr. The main reasons for choosing 70 km/hr are that this speed gives good signal to noise ratio and low influence of external variables (such as test vehicle design) as well as safe and practical driving conditions. In addition, 70 km/hr is a speed at which tire/road noise is likely to be a great nuisance to the environment in most types of traffic. This method can be used for type testing of tires and road surfaces, and for all testing where high precision and representative operation are essential. The Coast by Method does have its disadvantages, such as:

- a special test track or a road with suitable surface is required;
- a test vehicle equipped with 4-6 test tires is required; the test vehicle must be coasted along the test area;
- unless care is taken there may be an influence from vehicle type, brakes, transmission and suspension.
- There may be practical and safety problems for some vehicles when they are coasted.

In addition, it is necessary to minimize:

- climatic and meteorological influences;
- Sensitivity to disturbance from other traffic, if any, and other background noises.

In the **Trailer Method**, a test tire is mounted on a trailer, which is towed by a car or truck. The trailer may be of a single-wheel type or have extra supporting wheels. A
microphone is positioned close to the tire/road interface and the articulated vehicle is
driven along a test track or a road having a suitable surface. The microphone position is
0.2m outside the undeflected tire sidewall, .1m above the road level and 0.2m behind the
vertical axle plane. The 0.2m distances are changed to 0.4m for truck tires. In order to
increase the signal/noise ratio and reduce climatic influence, an enclosure around the test
tire and microphone is sometimes used. Special care concerning acoustical reflections
must then be observed. This method is suitable where relatively high precision is
required but some lack of realistic operation can be accepted. It is especially
recommended in environments with disturbing traffic, for instance on highly trafficked
roads where no other method is possible without closing the road. Long measuring times
can be used to reduce errors. The disadvantages of this method are:

- It requires a special towed trailer;
- Background noise from wind turbulence in the microphone can be a problem
  at low frequencies;
- The close measurement position gives some lack of realism due to acoustical
  reflections; and
- The near-field microphone location is unsuitable for road surfaces having a
  significant sound absorption.

In the **Laboratory Drum Method**, a test tire is mounted so that it can roll against a
drum surface. Special care must be taken concerning the acoustical environment. The
microphone is positioned as in the trailer method. A drum diameter of at least 1.5m is
required for an “outer drum” facility, when the tire is rolled against the outer part of the
drum shell. This method is suitable where high precision is important but lack of realistic
operation can be accepted. Surveys of noise emission from large numbers of tires under various operating conditions can be carried out in a short time. This method could also be useful for research and development work and for detecting small differences in noise emission from different tires. It is independent of weather conditions and requires little space and only one tire sample per test. Long measuring times can be used to reduce errors. The disadvantages of this method are:

- A special drum facility is required;
- The drum is not a good representative of road surface due to its curvature.

In all three methods, many factors influence the measured noise. These factors include tire type, road surface type, area of contact patch, tire inflation pressure, and vehicle speed. In general, with any combination of the above variables the sound level increases as the vehicle speed increases.

In the report *Tire/Pavement Interaction Noise Source Identification using Multi-Planar Nearfield Acoustical Holography* by Richard J. Ruhala and Courtney B. Burroughs (1999) [16], the authors used the trailer method and identified the major areas of maximum noise radiation to be the trailing edge, leading edge, and sidewall regions near the contact patch. Two tires were tested in this experiment, a monopitch tire and a production tire.

![Figure 37. Photograph of monopitch (left) and production tire threads. [14]](image)
The monopitch tire has 64 equally spaced transverse grooves cut in it, along with three circumfrential grooves. The tread passage frequency is equal to:

\[ f_t = \frac{NV}{C} \]

Where \( N \) is the number of transverse grooves, \( V \) is the vehicle speed, and \( C \) is the circumference of the tire. The production tire has tread blocks that vary in size and spacing around the circumference of the tire. The tire also has four circumfrential grooves that separate three rows of 55 tread blocks and two rows of 89 blocks. The sidewall has one ply polyester cord; the tread has three, one polyester cord and two steel cords. This test showed that there is speed dependence on sound pressure levels. Overall, both tires showed increased sound pressure levels with increased speed, but the sound pressure level from the monopitch tire did not always increase with speed. This, they say is probably due to the tread passage harmonics cycling through various resonances.
The noise levels of the leading edge, trailing edge, and sidewall change with speed. The noise level for the monopitch tire increased with speed above 40 km/hr at

![Graph](image)

**Figure 38** Sound pressure levels in dB. Ref. 20 μPa in the frequency range 2600 Hz. for (a) monopitch and (b) production tires. [14]
40*log(speed). For the production tire, the leading and trailing edges dominate the sidewall noise above 40 km/hr. The overall noise level increases at a rate of 40*log(speed) below 56 km/hr, and 20*log(speed) above 56 km/hr.

For further analysis, the monopitch and the production tires were tested at a speed of 58 km/hr on smooth asphalt pavement. For the monopitch tire, the areas of maximum radiation were localized to the sidewall near the leading edge, centerline of the leading edge.

Figure 39  Three views of active acoustic intensity from monopitch tire running on smooth asphalt at 56 km/hr. Frequency ranges are (a) 450 Hz. To 550 Hz. (tread passage frequency), (b) 900 to 1100 Hz., and (c) 1400 Hz. To 1600 Hz. Data are reconstructed [14]
edge, and centerline of the trailing edge. The spectrum was dominated by the harmonics of the tread passage events. At the tread passage frequency of 500 Hz, radiation from the sidewall dominated the sound power. This noise was mainly generated by the vibration of the sidewall due to the radial (normal) displacement of the tread blocks passing through the contact region. At one kHz the noise is generated by the same means, but has nearly equal radiation along the leading edge trailing edge, and the side of the contact patch. At 1.5 kHz, noise radiation is localized to the leading and trailing edges. The probable cause of this is vibration enhanced by air pumping and the second mode of the circumferential groove tube resonance.
For the production tire, less radiation is observed from the sidewall and more from the leading and trailing edges. Between 500 Hz and one kHz the frequencies increase with speed, showing that they are related to the tread passage events. The sound power is highest from 650 Hz to 950 Hz, which is probably amplified by the first mode of circumfrential groove resonance and air pumping. Again, the noise is likely generated

![Figure 40](image_url)

Figure 40 Three views of active acoustic intensity from production tire running on smooth asphalt at 56 km/hr. Frequency ranges are (a) 300 Hz. To 600 Hz. (tread passage frequency), (b) 650 Hz. to 950 Hz., and (c) 1300 Hz. To 1500 Hz. Data are reconstructed on planes touching surface of tire. Contour lines are in 2 dB. increments beginning at 78 dB. Solid contour lines are positive and dashed lines represent negative direction normal to the plane. [14]
from vibrations of the tread band due to radial (normal) displacement of the tread blocks passing through the contact region at the leading and trailing edges. The second mode of the circumferential groove resonance can be seen in the frequency region from (1300 Hz to 1500 Hz). At even higher frequencies, generation is localized to the contact patch near the leading edge and may be caused by treads being forced into the contact region.
Chapter X – Patch Frequency

In this section, a dynamic model of a tire is developed. The tire has a radius (‘R’) in meters, and angular speed (ω) in radians/sec. The outer circumference of the tire is made up of, N_p, number of patches. A patch consists of a tread pattern and a gap.

The total length of the patch is the *circumfrential length of the tread* (l) plus the *circumfrential length of the gap* (γ).

---

**Figure 41**

**Figure 42** Patch
The number of patches on a tire \((N_p)\) is the circumference of the tire divided by the patch length. (This is always a whole number)

\[
N_p = \frac{\text{tire circumference}}{\text{patch length}} = \frac{2\pi R}{l + \gamma}
\]

The frequency of patches passing through the contact region is the angular speed times the number of patches divided by the period, \(2\pi\):

\[
f_p = \frac{\omega N_p}{2\pi}
\]

Another form of this equation is:

\[
f_p = \frac{V N_p}{2\pi R}
\]

In which \(V\) is the vehicle speed and \(R\) is the tire radius.

Frequencies were then graphed for increasing speeds and a given \(N_p\). To find an appropriate range of angular velocities, car speeds, 0 to 180 km/hr, were chosen and converted to m/s. Next, a radius of .2159 m (17") was chosen; any wheel radius could have been chosen. The speed was then converted to angular velocity using the equation \(\omega = V/R\), which has units of radians/second. Next, random values of \(N_p\) were chosen starting from \(N_p=10\) to \(N_p=100\). These graphs show the linear relation between patch frequency and angular speed. (See Figure 43)
Figure 43 Frequency in Hz. vs. Angular Speed, $\omega$

The amplitude of noise produced from patches would be a function of the depth or length of the treads on the tire. The greater the tread length the greater the excitation when going through the contact region.
Chapter XI – Tire Bulge

When a tire is rolling, the tire carcass near the contact patch bulges out.

![Figure 44]

The extent of the bulge depends on the lateral stiffness, $K_L$, of the tire. It can be observed that the more bulge a tire has the louder the noise it radiates (flat tire). Since the amount of bulge is inversely proportional to the lateral stiffness of the tire, the amplitude of the noise is proportional to tire bulge and inversely proportional to the lateral stiffness.

$$A_{\text{NOISE}} \propto \text{Tire Bulge} \propto \frac{1}{K_L}$$

$$K_L \propto \frac{1}{A_{\text{NOISE}}}$$
It can also be said that the length of the contact region, \( C_p \), is inversely proportional to the lateral stiffness.

\[
C_p \propto \frac{1}{K_L}
\]

A simple analysis can be used to determine the frequency of the bulge with varying angular tire speeds.

\[ \omega = \text{the angular speed of the tire.} \]

\[ R = \text{the radius to the outside of the tire} \]

\[ C_p = \text{length of contact region} \]

\[ S_p = \text{Arc length of tire with in contact region} \]

\[ \theta = \text{Angle of Arc } S_p \]

\[ N_{sp} = \text{number of } S_p \text{'s per tire} \]

The arc length \( S_p \) equals \( R \times \theta \). Next, using the law of cosines:
\[ C_p^2 = 2R^2 - 2R^2 \cos(\theta) \]

Solving for \( \theta \),

\[ \theta = \cos^{-1}\left(\frac{-C_p^2 + 2R^2}{2R^2}\right) \]

Substituting into \( S_p \),

\[ S_p = R \cos^{-1}\left(\frac{-C_p^2 + 2R^2}{2R^2}\right) \]

The amplitude of noise is proportional to the velocity of the tire, which is equal to the tire’s angular velocity times the radius of the tire,

\[ A_{\text{NOISE}} \propto V = \omega \times R \]

In which \( \omega \) is the angular velocity of the tire, \( R \) is the outer radius of the tire, and \( V \) is the linear speed of the center of the tire
Chapter XII – The Rubbing Theory

A tire rubbing or sliding on a surface is another source of tire noise. This can be explained by what is called the Rubbing Theory. To explore the Rubbing Theory we consider the following model of a tire. The tire is rotating with some angular speed, \( \omega \), and has some weight, \( W \), acting downward at the center of the tire.

![Diagram of tire with labels](image)

**Figure 46**

Looking at the tire from the side, there is an area at the front of the tire where the tire is in compression and an area at the rear of the tire where the tire is in tension due to longitudinal stresses. Therefore, within the contact patch of the tire, there is an area of adhesion and an area of slippage. The length of the adhesion patch is denoted as \( L_c \) and...
the total length of the contact patch is denoted as $L_c$. The total area of the contact patch is denoted as $A$. From this, the slipping area can be found and is:

$$A_{sl} = A \left(1 - \frac{L_c}{L_t}\right)$$

The normal load on the sliding area is found by multiplying the weight by the ratio of the sliding area to the total contact patch area.

$$N_{sl} = W \left(\frac{A_{sl}}{A}\right)$$

For the sliding portion of the tire in the contact region, the noise produced is modeled using a skidding tire on pavement. For a tire on a frictionless surface, the velocity of a chunk of rubber in the contact region would be the angular speed of the tire times the outer radius of the tire. For a rolling tire, it can be said that a chunk of rubber in the slipping area, within the contact region, has a velocity relative to the road surface equal to the angular speed of the tire times the outer tire radius. A chunk of rubber in the adhesion area of the contact patch would have a speed of zero relative to the road. Looking at a chunk of rubber above the slide area, it is moving with velocity, $v$, relative to the road surface and has a normal load on it and an opposing friction force $F_f$. 

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The friction force is equal to the coefficient of road adhesion multiplied by the normal force on the sliding area.

\[ F_f = \mu_P * N_{SL} = \mu_P * W * \frac{A_{SL}}{A} \]

It is hypothesized that the sound pressure level \(L_p\) is a function of the normal force on the sliding area and the velocity of the chuck of rubber contained in the sliding area. The sound pressure level can further be hypothesized to be a function of the normal force, velocity, and the Rubbing Coefficient, \(\gamma_r\).

\[ L_p = f\left(N_{SL}, v\right) = f\left(N_{SL}, v, \gamma_r\right) \]
For example, the sound pressure level may equal the rubbing coefficient times the normal force on the sliding level times the velocity of the rubber block, and the velocity can be represented by the angular tire speed, $\omega$, times the tire radius, $R$.

$$L_p = \gamma_r * N_{SL} * v = \gamma_r * N_{SL} * \omega * R$$

The Rubbing Coefficient, $\gamma_r$, is to be determined experimentally.

In this experiment, there will be a rubber wheel rotated by a bicycle crank with an inertia wheel to help maintain a constant speed. The tire, while rotating, will be in contact with and rubbing against an asphalt road surface.
The asphalt surface was the parking lot at ZF Meritor in Maxton, NC and is similar in surface texture to that of most roads. In this experiment the rotation speed of the tire will be varied and a relationship between the sound pressure level $L_P$, and $\omega$, the wheel speed will be obtained. The same will be done with varying the normal force $N_{SL}$, by adding weight to the system to get the relationship between the sound pressure level $L_P$, and $\omega$. From these two relationships the rubbing coefficient, $\gamma_r$ will be determined.

For the measurement of the sound pressure level of the tire and asphalt rubbing, first the total sound pressure ($L_{Pb}$), the level of the crank, chain, the tire unit, and surrounding noise environment must be measured. Then the combined noise of the crank, chain, the tire unit, and surrounding noise environment along with the tire rubbing on the asphalt surface. ($L_P$) will be recorded. Next, the sound pressure level of the crank, chain, the tire unit, and surrounding noise environment will be subtracted from the overall sound
pressure level leaving the sound pressure of the tire rubbing on the asphalt. The total sound pressure level, L\textsubscript{pt} can be represented by:

\[ L_{pt} = 10 \times \log \left( \frac{P}{P_{re}} \right)^2 \text{ dB} \]

Here P is the root mean square (rms) sound pressure in question (Pa or N/m\textsuperscript{2}) and Pre is the International reference pressure of 20\times10\textsuperscript{6} Pa or .0002 \textmu Bar

This can also be put into terms of the mean-square pressure ratio as:

\[ \left( \frac{P}{P_{re}} \right) = \text{Antilog} \left( \frac{L_{pt}}{10} \right) = 10 \frac{L_{pt}}{10} \]

The background or ambient noise (L\textsubscript{pb}) is:

\[ L_{PB} = 10 \times \log \left( \frac{P}{P_{re}} \right)^2 \]

That is,
\[
\left( \frac{P}{P_{re}} \right)_B^2 = \text{Antilog} \left( \frac{L_{PB}}{10} \right) = 10 \frac{L_{pb}}{10}
\]

The sought after Sound Pressure Level (Lps) produced by the tire on the asphalt is given by:

\[
L_{PS} = 10 \ast \log \left[ \left( \frac{P}{P_{re}} \right)_t^2 - \left( \frac{P}{P_{re}} \right)_B^2 \right] \text{dB} \quad \text{or}
\]

\[
L_{PS} = 10 \ast \log \left( 10^{\frac{L_{Pt}}{10}} - 10^{\frac{L_{PB}}{10}} \right) \text{dB}
\]

In this experiment, the sound pressure levels were measured with a precision sound level meter made by Brue \& Kjaer. The sound level meter consists of a microphone, an amplifier and a detector with associated frequency and time weighting circuits, analogue DC output, and a digital display.

The sound level meter was placed 1 meter from the skidding tire. The bike crank system was turned without the tire being in contact with the asphalt, and a base sound level measurement was taken. The base sound level was 60 dB (A). The tire, a rubber wheel on a plastic rim, which has a radius of 2.5 inches, was then lowered so that it just made

Figure 49 Brue \& Kjaer Sound Level Meter
contact with the asphalt surface. The tire was then cranked to the desired 48 rpm and a 20-lb. weight was added. The sound pressure level was then recorded from the sound level meter. This was repeated 4 times and the sound pressure levels were recorded. The procedure was then repeated for rpm values of 60, 80, and 100. After reviewing the data and solving for \( \gamma_r \) (The Rubbing Coefficient), it can be seen that the coefficient decreases with increased speed. The Rubbing Coefficient vs. Wheel speed (rad/sec) was graphed and a trendline was fitted to the data. The trendline with the best fit was a logarithmic one, with the rubbing coefficient decreasing as follows:

\[
\gamma_r = -0.1538 \ln(\omega) + 0.4966
\]

This shows that in the original equation the relationship between the Rubbing Coefficient and the skidding wheel speed is a logarithmic one.

![Rubbing Coefficient vs. Tire speed](image)

Figure 50
This relationship between the Rubbing Coefficient and skidding wheel speed, the relationship for skidding wheel speed vs. sound level can be calculated. In the original equation:

\[ L_p = \gamma_r * N_s * \omega * r, \]

\( \gamma_r \) is replaced with \(-0.1538 \ln(\omega) + 0.4966\) and the following graph of tire noise vs. skidding wheel speed results (See Figure 51).

**Tire Speed vs. Tire Noise**

![Graph of tire speed vs. tire noise](image)

**Figure 51**
This trend in which the tire noise drops off after a certain speed can be seen in work done by M. Muthukrishnan in “Effects of Material Properties on Tire Noise” [12] in his graph of sound pressure level vs. speed (Figure 13). Knowing this, it is concluded that the rubbing coefficient varies logarithmically with skidding speed and that the sound pressure level peaks at a certain speed and then begins to taper off. The final equation for the sought after sound pressure level of the tire skidding on pavement is:

\[ L_{PS} = (-0.1538 \ln(\omega) + 0.4966) \times N_{SL} \times \omega \times r \]
Chapter XIII – Conclusions and Future Recommendations

In Chapter 2, sound is described as a disturbance spreading through a physical medium. One measurement of sound is called sound pressure level. Sound pressure levels can be added or subtracted to find a sound pressure level of a particular source. This method was used in finding the Rubbing Coefficient for the Rubbing Theory. Perceived sound pressure levels are frequency dependent.

There are two basic types of automotive tires, bias and radial ply. The difference in their construction affects their interaction with the road, which in turn could affect the noise produced by the tire. A tire on a rim, such as an automotive tire, deforms when weight is placed upon it. Deflections in the sidewall or shoulder region cause the stiffness in that region to decrease. The pressure in the tire also affects the amount the tire deflects or bulges (tire bulge). As the pressure decreases, the tire stiffness decreases in the shoulder region, which leads to increased vibration of shoulder treads, producing more noise. It is also noted, that due to longitudinal stresses in the tire there is an area of slippage at the rear of the contact region.

Tire noise is also affected by tread and sidewall material properties. When comparing the noise effects, the modulus of the tread has a greater influence on tire noise than the modulus of the sidewall, and the Tan Delta had little affect on tire noise.

The area of the tire near the contact region dominates tread vibration. In Coast by and Laboratory Drum testing by Doan, Brackin, Nishihata, and Sauerzapf [18], the noise produced by the tire peaks around one kHz. This testing also shows that shoulder tread vibration is the noise source that has the most relation to tire noise produced. It was also
shown that lowering the center contact pressure and/or increasing the shoulder tread bending stiffness will decrease the tire tread vibration and reduce the noise generated. The texture of the surface that a tire is rolling on affects the tire noise produced. The deeper the texture depth, the more the tread bands get excited, and the greater the noise produced.

Looking at a patch on a tire, which consists of a tread and a void, the void contains air. Due to the elasticity of the treads, these voids are compressed at the entrance of the contact patch. This causes the air to be forced or pumped out of the void. This air pumping creates a concentrated noise source, which directly relates to the second derivative of the volume of air displaced by the tread voids.

There are three common ways to measure tire noise. There is the Coast by method, the laboratory drum method, and the trailer method. The coast by method is very precise and best represents typical tire operation. The trailer method is also precise but lacks some realistic operation. The laboratory drum method is the best test for detecting small differences in noise emission from different tires but lacks in realistic operating conditions. Using the trailer method, Richard J. Ruhulla and Courtney B. Burrough (1999) [16], show that sound pressure levels in production tires increase linearly up until about 56 km/hr. They also show that at frequencies higher than 1500 Hz. noise generation is localized to the contact patch near the leading edge and may be caused by a slip-stick mechanism.

In Chapter 11 a dynamic model of a tire is developed. A tire contains a certain amount of patches (Np). The more treads you have, the more patches and the higher the patch frequency. It can be seen in Figure 43, that frequency, in Hz, increases linearly with
the angular speed, $\omega$, of the tire. Tires have a bulge near their contact region; the extent of the bulge depends on the lateral stiffness of the tire. The less the stiffness, the more the tire bulges, and the greater the amplitude of noise produced. The amplitude of noise is proportional to the velocity of the tire.

In Chapter 13, the Rubbing Theory is introduced. It proposes that the sound pressure level of rubber sliding on asphalt is a function of the Rubbing Coefficient * the normal force * the angular speed of the tire * the outer radius of the tire. The Rubbing Coefficient is determined experimentally to be a function of the angular tire speed. When the Rubbing Coefficient is placed back in the equation the sought after sound pressure level of a tire skidding on pavement is:

$$L_{PS} = \left(-0.1538 \ln(\omega) + 0.4966\right) \times N_{SL} \times \omega \times r$$

This equation could vary for different tire types and road textures but should have essentially the same shape.

Looking back at the methods described in Chapter 10, Sanberg, Ulf, and Jerzey Ejsmont [14] show that tire noise increases linearly with speed up until speeds of 56 Km/hr. For this linear region, it can be said; patch frequency $f_p$, also increases linearly, therefore, $f_p$ is proportional to tire noise. Beyond 56 Km/hr, things get complicated. Looking at the Rubbing Theory, tire noise varies logarithmically with wheel speed. One could say that around 56 Km/hr, the Rubbing Theory takes over and the relationship between wheel speed and tire noise goes non-linear. Also looking at Chapter 6, M. Muthukrishnan [12] shows the effect of speed on tire noise. The rubbing theory could explain why after certain speeds tire noise begins to drop, following the trend demonstrated by the rubbing theory and its domination at higher speeds.
In the future, this investigation into tire noise could be refined by using better equipment to refine the models developed and other tests to show the affects of patches and tire bulge on tire noise. For example, on the topic of patch frequency, a test could be run where identical tires except for the number of patches are tested at a given speed and the noise levels compared to the results in Chapter 11 and a correlation could be developed.

Another test that could be run to test out the effects of tire bulge on tire noise is the following: take a tire w/ a given $K_p$ (stiffness) and do coast by tests where the air pressure in the tire is adjusted to get different tire bulges. This would be a valuable test for it would give a relationship between tire bulge and tire pressure for a given tire and it would give you the relationship between tire bulge and tire noise or tire stiffness and tire noise.

On the Rubbing Theory, the experiment from Chapter 13 can be reproduced using better equipment. By this, I mean, that instead of using a lawn mower tire, one could use an actual car tire on pavement and record tire noise real time. After the noise-recording device is turned on, the tire is then brought from a stand still to skidding at its max speed (limited by motor power). This will give the relationship and relative rubbing coefficient for all speeds instead of just a select few. This can be repeated for a wide range of tires, weights, and skidding surfaces to form a more concrete relationship between tire noise and rubbing.

In conclusion, the basis of an acoustical model based on observation and general features of a tire (tire stiffness, the number of patches and the way the tire rubs the road) has been proposed. Referencing Wong [2], there is always a portion of the tire that is
slipping or rubbing against the road surface, and where there’s rubbing going on, there is the Rubbing Theory to help explain the relationship between complex tire dynamics and tire noise.
Bibliography


**Additional Acoustical Terms and Definitions:**

**Loudness** is the subjective impression of the intensity of sound. The unit is the sone. Loudness level is the sound pressure level in decibels of a pure tone of frequency 1000 Hz. that is assessed by normal observers as being equally as loud as the sound being measured. The unit is the Phon. The loudness, S, of a sound of 1000 Hz. is given by:

\[ \log_{10}(S) = 0.0301(P) - 1.204 \]

Where P is the loudness level in Phons

**Pitch** is the subjective estimate of a Tone as higher or lower on a scale. It is measured in Mels.

A **Decibel** is a unit expressing the magnitude of the ratio of two sound power or intensity levels. The number of decibels between two powers P1 and P2 is

\[ 10 \log \left( \frac{P_1}{P_2} \right) \]

For **sound pressure** the number of decibels is

\[ 20 \log \left( \frac{P_1}{P_2} \right) \]

A **Mel** is a unit of subjectively estimated Pitch. The Pitch of a 1000 Hz. tone at 40 dB. Above the threshold is taken to be 1000 Mels. The pitch of any sound judged to be double, then that pitch is taken as 2000 Mels.

A **Phon** is a unit of loudness level. The loudness level of a given Sound is the sound-pressure level (dB.) of a pure tone of frequency 1000 Hz. Which is assessed by normal
observers as being equally loud as the sound in question. Thus, Phons are efficiently expressed in dB. The value at the audible threshold is zero Phons, The threshold of feeling is about 140 Phons. A loudness level of 74 Phons corresponds to a sound pressure of one μbar of a 7000 Hz tone.

A **Sone** is a unit of subjective loudness. If S is the loudness in Sones and P the loudness level in Phons,

\[
\log_{10}(S) = 0.0301 \times P - 1.204
\]

**Sound Pressure** is the incremental variation about the ambient atmospheric pressure.

A **Pure Tone** is a sinusoidal pressure wave of a specific frequency and amplitude, propagating at a velocity determined by the temperature and pressure of air.