A PRELIMINARY INVESTIGATION OF IMPACT NOISE

A Thesis

Presented in Partial Fulfillment of the Requirements
for the Degree Master of Science

by

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Nomenclature

\( A_p \)  
surface area of the hammer  
\( \text{ft.}^2 \)

\( A \)  
axial location  
\( \text{ft.} \)

\( a \)  
offset of slider-crank-mechanism  
\( \text{ft.} \)

\( b \)  
length of drive crank  
\( \text{ft.} \)

\( c \)  
length of connecting rod  
\( \text{ft.} \)

\( j \)  
\( \sqrt{-1} \)

\( k \)  
wave number w/c  
\( 1/\text{ft.} \)

\( w \)  
rotational speed  
\( \text{Rad./sec.} \)

\( \phi \)  
angular position of connecting rod  
\( \text{Rad.} \)

\( \dot{\phi} \)  
angular velocity of connecting rod  
\( \text{Rad./sec.} \)

\( Q_m \)  
instantaneous flow rate  
\( \text{ft.}^3/\text{sec.} \)

\( r \)  
radial distance  
\( \text{ft.} \)

\( R \)  
radius of hammer  
\( \text{ft.} \)

\( P_{\text{ref}} \)  
reference pressure  
\( \text{2.9x10}^{-9} \text{psi} \)

\( P_{\text{real}} \)  
real component of acoustic pressure  
\( \text{psi} \)

\( P_{\text{img}} \)  
imaginary component of acoustic pressure  
\( \text{psi} \)

\( P_{\text{rms}} \)  
root-mean-square pressure  
\( \text{psi} \)

\( P_0 \)  
density of air  
\( \text{lb.}_m/\text{ft.}^3 \)

\( T \)  
time duration of impact  
\( \text{sec.} \)

\( t_i \)  
time of contact  
\( \text{sec.} \)

\( \Theta \)  
angular position of drive crank  
\( \text{rad.} \)

\( \Theta_i \)  
drive crank angle at contact  
\( \text{rad.} \)

\( \Theta_c \)  
drive crank angle at top-dead-center  
\( \text{rad.} \)

\( v \)  
instantaneous relative velocity of hammer  
\( \text{ft./sec.} \)

\( x \)  
position of hammer  
\( \text{ft.} \)

\( v \)
List Of Figures

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CHAPTER 1

Introduction

New health and safety laws, like the Walsh-Healy act, have recently been enacted. Noise pollution is such a health hazard covered by the Walsh-Healy Act. With these new laws the engineer is faced with the responsibility that his designs must be made safe and free from all health hazards. The engineer must immediately find ways to isolate man from adverse noise. This will only be a temporary solution to the problem. It will then be the engineer's task to eliminate or reduce the noise pollution so that man can live and work in safety and comfort.

The objective of this thesis is to study some of the parameters associated with impact noise. Impact noise produced from two different sources was studied. The classical source is the vibration of parts. Machine parts vibrate after impact and in turn they vibrate the surrounding air. The vibrating air results in pressure fluctuation which are the sensations stimulating the ear. The other source associated with impact noise is the air being squeezed out from the impacting bodies. For this thesis the impacting bodies were two flat plates. Air squeezed out from between two plates results in pressure fluctuations and sound, although the conditions to cause it are not understood. An example of this phenomenon occurs when someone claps his hands. The hands do not vibrate excessively yet there is a loud sound produced.
Very little work on this subject has been done. This is evident by the lack of published literature. Apparently, little has been published related to the phenomena of impact noise and what the source characteristics are. The only work located by the author was a Japanese paper. 1 This paper considers the sound that is produced by the collision of two steel balls.

This thesis embodied three goals to provide a preliminary study of impact noise.

The first goal was to learn the comparative importance of the escaping air at impact as a source that generates noise. The second goal was to learn the importance of the vibrating machine parts as an impact source of noise. Finally, since this is a preliminary work, areas of future research were to be identified and recorded for future reference.

These goals were accomplished by two studies. First, a mathematical model was developed to predict the acoustic pressure of two impacting flat bodies. This model predicts the acoustic pressure only for the noise source where the air is squeezed out from between the two bodies. Secondly an experimental test machine was designed and built so that the noise produced by two impacting bodies could be measured. A description of these studies is embodied in the next two chapters.

1Numbers in brackets refer to references in the List Of References.
CHAPTER 2

Theoretical Analysis

2.1 Introduction

In this chapter a theoretical model to predict the noise level produced from the air flow resulting from impact of two circular plates will be developed. The axially moving plates shall be called the hammer and the stationary plate the anvil. A spherical source equation will be used to predict the acoustic pressure and this pressure will be converted to the sound pressure level. This model is only for the noise produced from the air flow that results from the air being squeezed from between the two plates. It does not consider the noise from vibrating parts. An equation for the instantaneous velocity of the hammer must first be written.

2.2 Hammer Velocity

Since the theoretical results were to be compared to the experimental results and it was anticipated that a slider-crank mechanism would be used in the experimental work, it was necessary here to determine an equation for theoretical hammer velocity. Figure (1) is a schematic diagram of a slider-crank mechanism with appropriate linkages and geometry labeled.
Figure 1  Schematic diagram of a slider-crank mechanism

The position equation in complex notation is

\[ be^{j\theta} + ce^{j\phi} = x + aj. \]  \hspace{1cm} (1)

Differentiating once gives the velocity equation

\[ b\omega j e^{j\theta} + c\dot{\phi} j e^{j\phi} = \dot{x}. \]  \hspace{1cm} (2)

To make equation (2) most useful \( \phi \) and \( \dot{\phi} \) need to be evaluated in terms of mechanism dimensions and position and velocity of the driving crank. Taking the imaginary part of equation (1)

\[ bj \sin \theta + cj \sin \phi = aj. \]  \hspace{1cm} (3)

Solving for \( \phi \) yields

\[ \phi = \sin^{-1} \left( \frac{a - b \sin \theta}{c} \right). \]  \hspace{1cm} (4)

The velocity \( \dot{\phi} \) is obtained in a similar way. Taking the imaginary part of equation (2)
\[ b \omega \dot{\phi} \cos \Theta + c \dot{\phi} \dot{\phi} \cos \phi = 0. \] (5)

Solving for \( \dot{\phi} \) yields
\[ \dot{\phi} = -\frac{b \omega \cos \Theta}{c(\cos \phi)}. \] (6)

The expression for hammer velocity will later be substituted directly into the equation for acoustic pressure with \( \phi \) and \( \dot{\phi} \) available as in equations (4) and (6).

2.3 Acoustic Pressure

The model used to predict the acoustic pressure is based on spherical noise source theory. It is assumed that the impact location is a point source and that the sound radiates from that point uniformly in three dimensional space. The equation for acoustical pressure in the near field \([3][4]\) is
\[ p = \frac{\rho_0 \omega Q_m}{4\pi r^2} \left[ e^{i(\omega t - kr)} \right] + \frac{\rho_0 \omega Q_m}{4\pi kr^3} \left[ e^{i(\omega t - kr)} \right], \] (7)

where \( Q_m \) is the instantaneous volumetric flow rate across a sphere of radius \( r \), \( \rho_0 \) is the density, and \( \omega \) is the frequency. The wave number, \( k \), is the frequency \( \omega \) divided by the speed of sound. The last term of equation (7) accounts for the near field effects. These are due to the fluid flow pressure. The near field effects are noticeable whenever
\[ \frac{k}{r} < 1. \] (8)
To find $Q_m$ it is now assumed that the air flow is incompressible and viscous. This is a valid procedure to find source strength of an acoustic wave. From the continuity equation and ideas developed in squeeze-film bearings theories [3], one can write

$$Q_m = A_p \nu,$$  \hspace{1cm} (9)

where $\nu$ is the instantaneous relative velocity of the impacting bodies and $A_p$ is the area of the plate. For the case where the impacting bodies are flat circular plates, it follows that

$$Q_m = \pi R^2 \nu,$$  \hspace{1cm} (10)

where $R$ is the radius of the circular plate.

Substituting this into equation (7) yields

$$p = \frac{\rho_0 \omega R^2 \nu}{4r} \left[ e^{i(\omega t - kr)} + \frac{\rho_0 \omega R^2 \nu}{4kr^2} e^{i(\omega t - kr)} \right].$$  \hspace{1cm} (11)

Using the hammer velocity in equation (2) one can write

$$p = \frac{\rho_0 \omega R^2}{4r} \left[ bw_j e^{i\Theta} + c\phi_j e^{i\phi} \right] e^{i(\omega t - kr)}$$

$$+ \frac{\rho_0 \omega R^2}{4kr^2} \left[ bw_j e^{i\Theta} + c\phi_j e^{i\phi} \right] e^{i(\omega t - kr)}.$$  \hspace{1cm} (12)
Expanding and collecting terms yield

\[ p = \frac{-\rho_0 \omega^2 b R^2}{4r} [e^{i(2\omega t - Kr)}] - \frac{\rho_0 \omega \phi c R^2}{4r} [e^{i(\omega t + \phi - Kr)}] \]

\[ + \frac{\rho_0 \omega^2 b R^2}{4kr^2} [e^{i(2\omega t - Kr + \frac{\pi}{2})}] + \frac{\rho_0 \omega \phi c R^2}{4kr^2} [e^{i(\omega t + \phi - Kr + \frac{\pi}{2})}] \]  

(13)

This equation may now be used to solve for the acoustic pressure directly. Each term can be written in its real and imaginary components:

\[ P_{\text{real}} = \frac{-\rho_0 \omega^2 b R^2}{4r} \cos(2\omega t - Kr) - \frac{\rho_0 \omega \phi c R^2}{4r} \cos(\omega t + \phi - Kr) \]

\[ + \frac{\rho_0 \omega^2 b R^2}{4kr^2} \cos(2\omega t - Kr + \frac{\pi}{2}) + \frac{\rho_0 \omega \phi c R^2}{4kr^2} \cos(\omega t + \phi - Kr + \frac{\pi}{2}), \]  

(14)

And

\[ P_{\text{img}} = \frac{-\rho_0 \omega^2 b R^2}{4r} \sin(2\omega t - Kr) - \frac{\rho_0 \omega \phi c R^2}{4r} \sin(\omega t + \phi - Kr) \]

\[ + \frac{\rho_0 \omega^2 b R^2}{4kr^2} \sin(2\omega t - Kr + \frac{\pi}{2}) + \frac{\rho_0 \omega \phi c R^2}{4kr^2} \sin(\omega t + \phi - Kr + \frac{\pi}{2}). \]  

(15)

It should be noted that to make the units consistent every term in the above two equations must be divided by a gravitation constant. The magnitude of the acoustic pressure is then found by

\[ p = \sqrt{P_{\text{real}}^2 + P_{\text{img}}^2}. \]  

(16)

Since all experimental data will be presented in terms of the sound pressure level (SPL), the equation for the SPL is presented
here for completeness [8]

\[ SPL = 10 \log_{10} \left( \frac{P_{\text{rms}}}{P_{\text{ref}}} \right)^2 \]  

(17)

where \( P_{\text{rms}} \) is the root mean square of the pressure and \( P_{\text{ref}} \) is the reference pressure equal to \( 2.9 \times 10^{-9} \) psi.
CHAPTER 3
Experimental Program

3.1 Introduction

The experimental program was divided into three different studies. The first study was performed to determine the relationship between sound pressure level and the radial location from the source. The results of this study were compared to the theoretical model developed earlier. The second study determined the variation of sound pressure level as a function of both axial location and radial location. This study was completely experimental since the theoretical model has no provisions for the affects of axial location. The last study to be made was called the impending impact noise study. Impending impact noise was defined as the noise produced when the anvil was withdrawn just enough to prevent any solid contact between the hammer and anvil. This noise was produced solely by the air being squeezed from between the two approaching bodies. The tests performed will try to determine if noise is produced without actual impact of the parts. Before the results are presented a description of the testing apparatus and testing procedures will be given.

3.2 Testing Machine

The test machine was designed to allow easy adjustment of the parameters to be studied. These included:
1. running speed
2. hammer velocity at impact
3. hammer material
4. position of measuring instruments

Figure 2 shows an elevation drawing of the test machine. A slider crank mechanism was used to produce a reciprocating motion of the hammer. The running speed could be adjusted in two ways. First, the motor had a variable-speed control which allows for speed variation from 877 RPM to 2630 RPM. Further speed reduction was accomplished through different V-belt pulley combinations.

The hammer was mounted at the end of a sliding shaft so that the impact point would be well ahead of the bearings. This construction reduced the amount of bearing noise detected by the sound measuring equipment. Provision was made to enable easy removal of the hammer so that different hammer materials could be tried.

The stationary anvil was mounted on the slide of an engine lathe using a support structure made of structural channel. This anvil support was made heavy and rigid to reduce vibration. Vibration of the structure would cause undesirable noise. Mounting of the anvil was accomplished by three axially-placed 1/2 - 20 UNF studs for parallel alignment with the hammer.

The hammer velocity at impact could be adjusted by moving the anvil axially on the lathe bed. Adjustment in this direction was accomplished by an adjusting screw located in back of the slide. Once in place, holdown clamps were tightened.
FIGURE 2
ELEVATION OF TEST MACHINE
MANTOCHOW  JUNE 19, 1972
NOT TO SCALE
To contain the noise produced by the hammer drive, a box was built around it so that only the hammer protruded. The box was made from 3/4 inch plywood and was lined with 2 inch fiberglass on the sides with a 4 inch thickness on the top.

The hammer material which gave the most satisfactory results was rubber. A one inch thick, 50 durometer, piece of rubber was glued to a steel plate. This was then mounted on the end of the sliding shaft. This hammer was used for all of the tests reported in this thesis.

Figures 3, 4, and 5 are photographs of the actual testing machine.
Figure 3 on page 14 Close Up Of Hammer And Anvil
Figure 4 on page 16 Drive Mechanism
Figure 5 on page 18 Acoustic Box And Axial Support
3.3 Instrumentation

Figure 6 shows a schematic of the test instrumentation.

![Schematic Diagram]

**Figure 6**
Schematic of the Test Instrumentation
A portable sound level meter was used to measure the acoustic pressure. The position of the portable sound level meter was determined by dimension R, the radial distance from the center of the anvil, and dimension A, the axial distance measured from the contact surface of the anvil. It should be noted that the axial position can be either positive or negative as shown in Figure 6. The output signal of the portable sound level meter was recorded on one of two instruments. The first instrument was an oscilloscope with a polaroid camera attachment. This arrangement provided good results, but considerable film was wasted because of a poor triggering technique. The other instrument to be used was an oscillograph strip chart recorder. Continuous data could be taken and recorded on the paper strip. This eliminated the triggering problem experienced with the oscilloscope. Samples of data recorded by both instruments can be found in the appendix.

The position of the hammer relative to the anvil was measured by a Fotonic Sensor. This is a fiber optics measuring instrument. A light probe was located in the center of the anvil. Light reflected back from the hammer was used to measure the distance from the anvil to the hammer. Figure 7 is a typical trace recorded from the Fotonic Sensor while the testing machine was operating.
The time $T$ is the total time the hammer remained in contact with the anvil. The total time duration of contact was used to determine the impact velocity. A sample calculation for impact velocity can be found in the appendix. The clearance between the hammer and anvil for impending impact was determined by the method shown in Figure 7.

3.4 Testing Procedure
3.4.1 Testing Program For Comparison Of The Theoretical Model To Experimental Data To Study The Effects Of Radial Location On The Sound Pressure Level.

Specific running speeds, impact velocity and axial location were selected. The running speed was measured with a hand tachometer. The impact velocity was determined as outlined in section 3.3. The axial location was selected to give the maximum pressure pulse. This was determined by moving the sound measuring equipment axially until a maximum pressure pulse was observed.
Once this position was determined it was held constant for all radial measurements. Data was then taken at 5 radial locations the largest of which provided an almost unmeasurable pulse.

3.4.2 Testing Program To Obtain Experimental Data To Show The Relationship Of Sound Pressure Level To Both Axial Position And Radial Location.

A specific running speed and impact velocity was selected as described in section 3.4.1. Measurements of sound pressure level were made at three radial locations, 3.25 inches, 4.0 inches, and 8.0 inches, and four axial locations.

3.4.3 Testing Program To Study Impending Impact Noise.

A specific running speed was selected. The test machine was run with the anvil far enough away so as not to cause any impact. The anvil was then positioned by the adjusting screw until the desired amount of minimum clearance was obtained. Measurements of sound pressure level were then taken at one specific radial location and one specific axial position.

3.4.4 Determination Of The Natural Frequency Of The Anvil Support.

An accelerometer was securely taped to the back of the anvil support. It was mounted in an axial position on about the center line of the anvil. The output of the accelerometer was fed into an oscilloscope. The natural frequency of the support was determined by striking the anvil once with a hammer and observing the frequency of the resulting vibration. This output was then visually compared to the output for the acoustic pressure from section 4.1.
CHAPTER 4

Results

4.1 Theoretical and Experimental Results of Sound Pressure Level for Different Radial Locations at One Specific Axial Position.

Figure 8 is a plot of the theoretical sound pressure level and the experimental data points as a function of radius. The speed of the drive crank was 386.5 RPM and the impact velocity was 6.4 IPS. An axial location was selected 5/8 inch toward the crank from the impact surface because it appeared that the maximum pressure pulse for any given radial location occurred there. The plot in Figure 8 is a plot of the sound pressure level as a function of the radial distance. Sample data may be found in the appendix.

4.2 Experimental Results of Sound Pressure Level As A Function Of Both Axial Position And Radial Location.

Figure 9 is a plot of the experimental sound pressure level as a function of both the axial position and the radial location. The drive speed at which this data was taken was 399 RPM with a hammer impact velocity of 6.9 ips. Sample data of this study may be found in the appendix. It should be remembered that the theoretical model does not take into consideration any effects of the axial location.

4.3 Experimental Results For The Sound Pressure Level For Impending Impact Noise.
---: THEORETICAL MODEL PREDICTION
○: EXPERIMENTAL DATA POINTS

IMPACT VELOCITY = 6.4 in/SEC.
AXIAL POSITION = +5/8 IN.

M. ANTOCHOW 8/15/72

FIGURE 8 THEORETICAL AND EXPERIMENTAL RESULTS
OF SOUND PRESSURE LEVEL PLOTTED AS A FUNCTION
OF RADIAL LOCATION FOR A SPECIFIC AXIAL
POSITION
FIGURE 9 EXPERIMENTAL RESULTS OF SOUND PRESSURE LEVEL AS A FUNCTION OF AXIAL POSITION FOR SEVERAL DIFFERENT RADIAL LOCATIONS
Sample data of the studies on impending impact noise is presented in Figure 11, in the appendix.

Inconsistencies in the testing machine's performance prevented reliable results. The clearance between hammer and anvil could not be held constant at top-dead-center.

Peak experimental sound pressure levels experienced for top-dead-center clearances in the range from 0.0 inches to 0.020 inches were 113db to 118db. The peak experimental pressure occurred 0.01 sec before top-dead-center. From Figure 10 the theoretical magnitude of acoustic pressure, for the time at which the experimental pressure peak was observed, is $1.13 \frac{\text{lb}}{\text{ft}^2}$. This corresponds to a SPL of 126 db.

A qualitative result from this part of the experimental work was that the peak acoustic pressure occurred sometime before the hammer reached top-dead-center.

4.4 Results From The Anvil Support Natural Frequency Test

It was observed from this test that the choppiness of the acoustic pressure, which occurred after impact was about the same frequency as the natural frequency of the anvil support.

4.5 Discussion of Results

The results presented in section 4.1 shows the relationship between sound pressure level and the radial location. The experimental data follows the trends predicted by the theoretical model very well in the near field. It should be remembered that the theoretical model predicts only the noise produced from the air being squeezed from between the plates.
Figure 10: Theoretical magnitude of acoustic pressure and hammer velocity plotted as a function of time for positive hammer velocities only.

Driveshaft velocity = 390.2 RPM
Radial position = 3 1/4 in.
Axial location = +5/8 in.
M. Antochow August 21, 1972
Section 4.2 shows the relationship of the sound pressure level to both the axial position and radial location. Although it is not clear what the exact mathematical relationship is between these parameters, it is clear from the data presented that the sound pressure level is dependant on the axial location.

The studies done on impending impact noise indicate the importance of this phenomenon. It can be seen by observing the sample data that a pressure pulse occurs sometime before the mechanism reaches top-dead-center. When converted to sound pressure level this peak represents a noise of the magnitude of 113 db to 118 db. It is interesting to note that the acoustic pressure pulses for impending impact are smooth signals that climb to a peak and then drop off. The acoustic pressure signals after impact are choppy. This can be seen if one compares Figures 10 and 12a.
CHAPTER 5

Recommendations For Future Work And Conclusions

5.1 Recommendations For Future Work

There are several modifications that should be made to the laboratory machine before any further testing is done using this machine. A more positive method of fastening the hammer to the sliding shaft should be found. The plate, to which the hammer is bolted, is fastened to the shaft with two flathead screws. Even with lockwashers behind these screws, they still come loose during operation of the machine.

The hammer guide now used is cylindrical and rotates about its own axis as it changes direction of travel. At the end of travel this causes an extra degree of freedom so that the position of the hammer is not uniquely determined. This caused the inconsistency in the impending impact study. A square hammer guide should solve the problem.

To make a more consistent running machine, bearings of a different type should be used. The present bearings have too much slop which causes problems when trying to hold a few thousandths of an inch clearance.

It was the designer's intent to make the anvil support massive so that it would not vibrate at its own natural frequency. However, the support does vibrate upon impact.
 Modifications should be made here especially if work in determining the relationship of axial location and sound pressure level is to continue. More steel used as bracing would make the structure more rigid. Another possible solution to stop the anvil support from vibrating would be to damp out the vibration. Damping material could be placed under the support.

The instrumentation also needs revising. A method to measure directly the hammer velocity at impact should be found. A smaller microphone with a positive method of positioning would better approximate point measurement of sound pressure level.

Of course, the mathematical model can be improved greatly by including the compressibility of the air and the axial location effects.

5.2 Conclusions

Several conclusions can be made from the work presented here. First, noise is produced just before contact occurs. This is a major source of impact noise and is produced from the air being squeezed from between the approaching bodies. For the mechanism used in this thesis, escaping air was a more important source than structural vibration. Although the escaping air source generated more noise in this study, the structural vibration source was still highly important. It seems likely that both will have to be treated in order to greatly reduce noise arising from impact machines. The third conclusion that can be made is that the theoretical model used in this thesis does predict with reasonable accuracy the acoustic
pressure and the sound pressure level in the near field. The final conclusion is that there is considerable dependence of the sound pressure level on the axial position of the measuring instrument.
LIST OF REFERENCES


3. Faulkner, L., M.E. 794 class notes, The Ohio State University.


Appendix A  Sample Data

Figure 11 on page 35 Sample Data For Impending Impact

axial location = +5/8 in.
radial location = 3 1/4 in.
acoustic pressure vertical scale: 1 division = $3.51 \times 10^{-4} \text{ psi}$
acoustic pressure horizontal scale: 1 division = .01 sec.
Fotonic Sensor vertical scale: not to scale
Fotonic Sensor horizontal scale: 1 division = .01 sec.
axial location = +5/8 in.
impact velocity = 6.4 in./sec.

a) upper left  radial location = 3 1/2 in.
acoustic pressure vertical scale: 1 division = 2.25x10^{-5} psi
acoustic pressure horizontal scale: 1 division = 2 msec.
Fotonic Sensor vertical scale: 1 division = .001 in.
Fotonic Sensor horizontal scale: 1 division = 2 msec.

b) upper right  radial location = 8 3/8 in.
acoustic pressure vertical scale: 1 division = 6.5x10^{-6} psi
acoustic pressure horizontal scale: 1 division = 2 msec.
Fotonic Sensor vertical scale: 1 division = .001 in.
Fotonic Sensor horizontal scale: 1 division = 2 msec.

c) lower center  radial location = 18 1/4 in.
acoustic pressure vertical scale: 1 division = 2.5x10^{-6} psi
acoustic pressure horizontal scale: 1 division = 2 msec.
Fotonic Sensor vertical scale: 1 division = .001 in.
Fotonic Sensor horizontal scale: 1 division = 2 msec.
Figure 13 on page 39 Sample Data For Axial Position Studies
impact velocity = 6.9 in./sec.

a) upper left: axial position = -1/4 in., radial location = 3 1/4 in.
acoustic pressure vertical scale: 1 division = 6.5x10^{-5} psi
acoustic pressure horizontal scale: 1 division = 2 msec.
Fotonic Sensor vertical scale: 1 division = .001 in.
Fotonic Sensor horizontal scale: 1 division = 2 msec.

b) upper right: axial position = 0 in., radial location = 3 1/4 in.
acoustic pressure vertical scale: 1 division = 1.3x10^{-4} psi
acoustic pressure horizontal scale: 1 division = 2 msec.
Fotonic Sensor vertical scale: 1 division = .001 in.
Fotonic Sensor horizontal scale: 1 division = 2 msec.

c) lower left: axial position = +1/4 in., radial location = 3 1/4 in.
acoustic pressure vertical scale: 1 division = 1.3x10^{-4} psi
acoustic pressure horizontal scale: 1 division = 2 msec.
Fotonic Sensor vertical scale: 1 division = .001 in.
Fotonic Sensor horizontal scale: 1 division = 2 msec.

d) lower right: axial position = +1 in., radial location = 3 1/4 in.
acoustic pressure vertical scale: 1 division = 1.3x10^{-4} psi
acoustic pressure horizontal scale: 1 division = 2 msec.
Fotonic Sensor vertical scale: 1 division = .001 in.
Fotonic Sensor horizontal scale: 1 division = 2 msec.

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Sample Calculations

In order to calculate the acoustic pressure the time of impact must be determined. This is done by first calculating the drive crank angle at contact and then calculating the time at which the crank is in this position.

It is assumed that contact occurs at one half of the total time duration of impact before the mechanism reaches top-dead-center. A discussion of total time duration of impact can be found in chapter 3.3. This corresponds to a drive crank angle of

\[ \theta_i = \theta_c - \frac{\omega T}{2}, \]  

where \( \theta_c \) is the drive crank angle at which the mechanism is at top-dead-center. By studying the geometry of Figure 14 one can write

\[ \theta_c = \sin^{-1}\left(\frac{a}{b+c}\right). \]

\[ \text{Figure 14} \]
Mechanism at Top-Dead-Center
The time $t_i$ of impact is

$$t_i = \frac{\Theta_i}{\omega}.$$  

(3)

The given information for this problem is:

- $a = 0.026$ ft,
- $b = 0.146$ ft,
- $c = 1.33$ ft,
- $R = 0.25$ ft,
- $r = 0.292$ ft,
- $T = 0.004$ sec,
- $w = 40.47$ Rad/sec.

Solving for $\Theta_c$ yields

$$\Theta_c = \sin^{-1}\left(\frac{0.026}{0.146 + 1.33}\right) = 0.0176 \text{ RAD}.$$  

(4)

Substituting the given information into equation (1) yields

$$\Theta_i = 0.0176 - \frac{40.47(0.004)}{2} = -0.0633 \text{ RAD}.$$  

(5)

Solving for the time of impact gives

$$t_i = \frac{-0.0633}{40.47} = -0.00156 \text{ SEC}.$$  

(6)

Connecting rod angle and rotational speed can be calculated from equations (4) and (6) from section 2.2 and are rewritten here

$$\phi = \sin^{-1}\left(\frac{a-b\sin\Theta}{c}\right),$$  

(7)

and

$$\dot{\phi} = \frac{-bw\cos\Theta}{c(\cos\phi)}.$$  

(8)
Substituting the given information into equation (7) and (8) yields

\[ \dot{\phi} = \sin^{-1} \left[ \frac{0.026 - 0.146 \sin(-0.063)}{1.33} \right] = 0.026 \text{ RAD.} \]  

(9)

and

\[ \dot{\phi} = \frac{(-0.146)(40.47)[ \cos(-0.063)]}{1.33 \cos(0.026)} = -4.436 \text{ RAD SEC.} \]  

(10)

Now using the imaginary part of equation (2) from section 2.2 rewritten here

\[ \chi = -b \omega \sin \theta - c \dot{\phi} \sin \phi, \]

(11)

one can calculate impact velocity.

Substituting into equation (11) yields

\[ \dot{\chi} = -0.146(40.47)[ \sin(-0.063)] - 1.33(-4.436)[ \sin(0.026)] = 0.53 \text{ FT SEC.} \]  

(12)

All the information required for the calculation of acoustic pressure is now in hand.

Equations (14) and (15) of section 2.3 are used to calculate the real and imaginary components of acoustic pressure and are rewritten here

\[ p_{\text{real}} = \frac{\rho \omega^2 b R^2}{4r} \cos(2\omega t - kr) - \frac{\rho \omega^2 c R^2}{4r} \cos(\omega t + \phi - kr) \]

\[ + \frac{\rho \omega^2 b R^2}{4kr^2} \cos(2\omega t - kr + \frac{\pi}{2}) + \frac{\rho \omega^2 c R^2}{4kr^2} \cos(\omega t + \phi - kr + \frac{\pi}{2}), \]

(13)
\[ P_{\text{mag}} = \frac{-\rho_0 \omega^2 b R^2}{4r} \sin(2\omega t - k r) - \frac{\rho_0 \omega c \phi R^2}{4r} \sin(\omega t + \phi - k r) \]
\[ + \frac{\rho_0 \omega^2 b R^2}{4kr^2} \sin(2\omega t - kr + \frac{\pi}{2}) + \frac{\rho_0 \omega c \phi R^2}{4kr^2} \sin(\omega t + \phi - kr + \frac{\pi}{2}). \]  

(14)

Substituting the given information into equations (13) and

(14) yields

\[ P_{\text{real}} = \frac{-0.76(40.47)^2(0.26)(25)^2}{4(2.92)(32.2)} \cos[2(40.47)(0.016) - 0.036(292)] \]
\[ - \frac{0.76(40.47)^2(1.33)(4.436)(25)^2}{4(2.92)(32.2)} \cos[2(40.47)(0.016) + 0.026 - 0.036(292)] \]
\[ + \frac{0.76(40.47)^2(0.26)(25)^2}{4(2.92)^2(32.2)(0.036)} \cos[2(40.47)(0.016) - 0.036(292) + \frac{3\pi}{2}] \]
\[ + \frac{0.76(40.47)^2(1.33)(4.436)(25)^2}{4(2.92)^2(32.2)(0.036)} \cos[2(40.47)(0.016) + 0.026 - 0.036(292) + \frac{3\pi}{2}]. \]  

(15)

and

\[ P_{\text{imag}} = \frac{-0.76(40.47)^2(0.26)(25)^2}{4(2.92)(32.2)} \sin[2(40.47)(0.016) - 0.036(292)] \]
\[ - \frac{0.76(40.47)^2(1.33)(4.436)(25)^2}{4(2.92)(32.2)} \sin[2(40.47)(0.016) + 0.026 - 0.036(292)] \]
\[ + \frac{0.76(40.47)^2(0.26)(25)^2}{4(2.92)^2(32.2)(0.036)} \sin[2(40.47)(0.016) - 0.036(292) + \frac{3\pi}{2}] \]
\[ + \frac{0.76(40.47)^2(1.33)(4.436)(25)^2}{4(2.92)^2(32.2)(0.036)} \sin[2(40.47)(0.016) + 0.026 - 0.036(292) + \frac{3\pi}{2}]. \]  

(16)

Now using equation (16) from section 2.3 rewritten here

\[ P = \sqrt{P_{\text{real}}^2 + P_{\text{imag}}^2}, \]

(17)

the magnitude of the acoustic pressure can be solved. Taking the result of equations (15) and (16) and substituting them into equation (17) yields,

\[ P = \sqrt{(259)^2 + (-0.016)^2} = 259 \ \text{lb}_f \text{ft}^2. \]  

(18)
Finally using equation (17) from section 2.3 rewritten here

\[ SPL = 10 \log_{10} \left( \frac{P_{\text{rms}}}{P_{\text{ref}}} \right)^2 \]  \hspace{1cm} (19)

and substituting all the known information yields

\[ SPL = 10 \log_{10} \left[ \frac{0.259}{144(\sqrt{2})(2.9 \times 10^{-9})} \right]^2 = 113 \text{ dB} \]  \hspace{1cm} (20)