Measurement of shell noise in the exhaust tube of internal combustion engine.

Tejendra N. Bose
University of Windsor

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MEASUREMENT OF SHELL NOISE
IN THE EXHAUST TUBE OF
INTERNAL COMBUSTION ENGINE

A Thesis
Submitted to the Faculty of Graduate Studies through the
Department of Mechanical Engineering in Partial Fulfilment
of the Requirements for the degree of
Master of Applied Science at the
University of Windsor

By
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Windsor, Ontario
1974
ABSTRACT

In this investigation, a detailed study has been made of the shell noise, generated by the exhaust tube in the exhaust system of a four-stroke six-cylinder internal combustion engine. Preliminary investigations with exhaust gas pressure and exhaust noise were first made, which led to the measurement and analysis of the shell noise of the exhaust tube. This is a part of the noise reduction program in the automobile exhaust system. These results and facilities will enable the tube manufacturer to make comparative studies to determine the design criteria of the tube in regards to its size, shape, wall thickness, material and construction for the optimum noise reduction.

To provide a common basis of measurement, an absorption type muffler was used first to take various noise and pressure readings. These results were compared with the readings obtained by using a conventional reaction muffler.

The engine was installed inside a room which had hollow concrete blocks to give high transmission loss. The exhaust tube passed through the wall of the room and the shell noise was measured inside a specially built soft chamber. Thus the engine noise was completely isolated.
The following exhaust gas measurements were made:

1. Static pressure
2. Pressure pulse from a transducer
3. Temperature

Shell noise was measured by a capacitive type microphone. Attempts were made to do the measurements in the free-field conditions, but due to the presence of relatively high ambient noise levels, the chamber was fabricated and installed around the exhaust tube and the microphone was mounted inside the chamber.

Both, the exhaust gas pressure inside the tube and the shell noise were frequency analyzed under different running conditions of the engine, namely, under different loads and at different speeds.

The first phase of investigations was carried out on straight tubes of different materials, wall thickness and double shell construction.

These facilities would be used for further investigations with bent tubes, different mufflers or to simulate the actual condition of the automobile exhaust system.
ACKNOWLEDGEMENT

My heartfelt gratitude to Dr. Z. Reif for his help, encouragement and guidance in this project.

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National Research Council of Canada granted me the PIER fellowship for which I am grateful.

Standard Tube of Canada, Woodstock, Ontario initiated and financed this research, which was organized by the Industrial Research Institute of the University of Windsor, thereby giving me the privilege of working in this interesting project.
CONTENTS

ABSTRACT

ACKNOWLEDGEMENT

TABLE OF CONTENTS

LIST OF FIGURES

NOMENCLATURE

1. INTRODUCTION
   Subject of Investigation and Its Importance
   Scope

2. LITERATURE REVIEW

3. THEORY OF EXHAUST NOISE
   Firing Frequency
   Effect of Exhaust Gas Velocity
   Increased Sound Pressure

4. TEST SET-UP
   Engine Equipment
   Muffler
   Measuring Chamber

5. INSTRUMENTATION

6. PRELIMINARY MEASUREMENTS
   Transmission Loss of Engine Room
   Ambient Noise
   Behaviour of Test Chamber

Page

iii
v
vii
viii
xi
1
1
1
3
4
7
7
8
10
11
15
15
21
31
31
31
32
### LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Idealized Waveform of Exhaust Pulse</td>
<td>5</td>
</tr>
<tr>
<td>2. Arrangement of Manometer, Thermocouple and Pressure Transducer</td>
<td>12</td>
</tr>
<tr>
<td>3. Arrangement of Thermocouples</td>
<td>12</td>
</tr>
<tr>
<td>4. Arrangement of Manometer, Thermocouple and Pressure Transducer - Photograph</td>
<td>13</td>
</tr>
<tr>
<td>5. Exhaust Tubé, Showing the End Coupling and Thermocouples</td>
<td>13</td>
</tr>
<tr>
<td>6. Water Connection, Showing Cooling Arrangement For Engine</td>
<td>14</td>
</tr>
<tr>
<td>7. Absorption Type Muffler</td>
<td>16</td>
</tr>
<tr>
<td>8. Engine and Dynamometer Assembly</td>
<td>17</td>
</tr>
<tr>
<td>9. Chamber - Photograph</td>
<td>17</td>
</tr>
<tr>
<td>10. Chamber</td>
<td>19</td>
</tr>
<tr>
<td>11. Sectional View of Microphone Assembly</td>
<td>20</td>
</tr>
<tr>
<td>12. Fitting of Tube in the Chamber</td>
<td>20</td>
</tr>
<tr>
<td>13. Connection of Pressure Transducer, Pyrometer and Manometer</td>
<td>22</td>
</tr>
<tr>
<td>14. Internationally Standardized Weighting Curves</td>
<td>26</td>
</tr>
<tr>
<td>15. Frequency Response of Tape Recorder</td>
<td>26</td>
</tr>
<tr>
<td>17. Instrumentation - Photograph</td>
<td>29</td>
</tr>
<tr>
<td>18. Ambient Noise - 0° Angle of Incidence</td>
<td>33</td>
</tr>
<tr>
<td>19. Ambient Noise - 90° Angle of Incidence</td>
<td>34</td>
</tr>
<tr>
<td>20. Ambient Noise Inside the Chamber</td>
<td>35</td>
</tr>
<tr>
<td>21. Constant Voltage Signal - Output From Tape Recorder</td>
<td>37</td>
</tr>
<tr>
<td>22. Playback From Tape Recorder</td>
<td>38</td>
</tr>
<tr>
<td>23. S.P.L. vs. Log x</td>
<td>39</td>
</tr>
<tr>
<td>24. Experimental Set-Up</td>
<td>43</td>
</tr>
</tbody>
</table>
LIST OF FIGURES (Cont'd.)

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>25.</td>
<td>Frequency Analysis of Exhaust Noise at 1000 R.P.M.</td>
<td>47</td>
</tr>
<tr>
<td>26.</td>
<td>Frequency Analysis of Exhaust Noise at 1500 R.P.M.</td>
<td>48</td>
</tr>
<tr>
<td>27.</td>
<td>Frequency Analysis of Exhaust Noise at 2000 R.P.M.</td>
<td>49</td>
</tr>
<tr>
<td>28.</td>
<td>Frequency Analysis of Exhaust Noise at 1000 R.P.M. and No Load</td>
<td>51</td>
</tr>
<tr>
<td>29.</td>
<td>Frequency Analysis of Exhaust Noise at 1000 R.P.M. and 140 lbs. Load</td>
<td>52</td>
</tr>
<tr>
<td>30.</td>
<td>Frequency Analysis of Exhaust Noise at 1000 R.P.M. and 170 lbs. Load</td>
<td>53</td>
</tr>
<tr>
<td>31.</td>
<td>Frequency Analysis of Exhaust Noise at 1000 R.P.M. and 205 lbs. Load</td>
<td>54</td>
</tr>
<tr>
<td>32.</td>
<td>Relation Between Firing Frequency and Shell Noise - Tube No. 4</td>
<td>59</td>
</tr>
<tr>
<td>33.</td>
<td>Relation Between Firing Frequency and $\bar{F}$ - Tube No. 4 (No Muffler)</td>
<td>60</td>
</tr>
<tr>
<td>34.</td>
<td>Relation Between Firing Frequency and $\bar{F}$ - Tube No. 4 (With Muffler)</td>
<td>61</td>
</tr>
<tr>
<td>35.</td>
<td>Relation Between Frequency and $\bar{F}$ - Tube No. 4 (No Muffler)</td>
<td>63</td>
</tr>
<tr>
<td>36.</td>
<td>Relation Between B.H.P. and Shell Noise - Tube No. 4 (1000 R.P.M.)</td>
<td>64</td>
</tr>
<tr>
<td>37.</td>
<td>$\frac{\Delta T}{T_1-T_2}$ vs. Engine Speed</td>
<td>65</td>
</tr>
<tr>
<td>38.</td>
<td>Relation Between Firing Frequency and Shell Noise - Tube No. 6 (Full Load)</td>
<td>68</td>
</tr>
<tr>
<td>39.</td>
<td>Relation Between Firing Frequency and Shell Noise - Tube No. 9 (Full Load)</td>
<td>69</td>
</tr>
<tr>
<td>40.</td>
<td>Relation Between Firing Frequency and Shell Noise - Tube No. 20 (Full Load)</td>
<td>70</td>
</tr>
<tr>
<td>41.</td>
<td>Relation Between B.H.P. and Shell Noise - Tube No. 6 (1000 R.P.M.)</td>
<td>71</td>
</tr>
<tr>
<td>42.</td>
<td>Relation Between B.H.P. and Shell Noise - Tube No. 9 (1000 R.P.M.)</td>
<td>72</td>
</tr>
<tr>
<td>43.</td>
<td>Relation Between B.H.P. and Shell Noise - Tube No. 20 (1000 R.P.M.)</td>
<td>73</td>
</tr>
<tr>
<td>44.</td>
<td>Relation Between Frequency and $\bar{F}$ - Tube No. 6 (Without Muffler)</td>
<td>74</td>
</tr>
<tr>
<td>Figure</td>
<td>Title</td>
<td>Page</td>
</tr>
<tr>
<td>--------</td>
<td>----------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>45.</td>
<td>Relation Between Frequency and x</td>
<td>75</td>
</tr>
<tr>
<td></td>
<td>Tube No. 6 (With Muffler)</td>
<td></td>
</tr>
<tr>
<td>46.</td>
<td>Relation Between Frequency and x</td>
<td>76</td>
</tr>
<tr>
<td></td>
<td>Tube No. 9 (No Muffler - Full Load)</td>
<td></td>
</tr>
<tr>
<td>47.</td>
<td>Relation Between Frequency and x</td>
<td>77</td>
</tr>
<tr>
<td></td>
<td>Tube No. 9 (With Muffler)</td>
<td></td>
</tr>
<tr>
<td>48.</td>
<td>Relation Between Frequency and x</td>
<td>78</td>
</tr>
<tr>
<td></td>
<td>Tube No. 20 (No Muffler)</td>
<td></td>
</tr>
<tr>
<td>49.</td>
<td>Relation Between Frequency and x</td>
<td>79</td>
</tr>
<tr>
<td></td>
<td>Tube No. 20 (With Muffler)</td>
<td></td>
</tr>
<tr>
<td>50.</td>
<td>Relation Between Firing Frequency and x</td>
<td>80</td>
</tr>
<tr>
<td></td>
<td>Tube No. 6 (No Muffler)</td>
<td></td>
</tr>
<tr>
<td>51.</td>
<td>Relation Between Firing Frequency and x</td>
<td>81</td>
</tr>
<tr>
<td></td>
<td>Tube No. 6 (With Muffler)</td>
<td></td>
</tr>
<tr>
<td>52.</td>
<td>Relation Between Firing Frequency and x</td>
<td>82</td>
</tr>
<tr>
<td></td>
<td>Tube No. 9 (No Muffler)</td>
<td></td>
</tr>
<tr>
<td>53.</td>
<td>Relation Between Firing Frequency and x</td>
<td>83</td>
</tr>
<tr>
<td></td>
<td>Tube No. 9 (With Muffler)</td>
<td></td>
</tr>
<tr>
<td>54.</td>
<td>Relation Between Firing Frequency and x</td>
<td>84</td>
</tr>
<tr>
<td></td>
<td>Tube No. 20 (No Muffler)</td>
<td></td>
</tr>
<tr>
<td>55.</td>
<td>Relation Between Firing Frequency and x</td>
<td>85</td>
</tr>
<tr>
<td></td>
<td>Tube No. 20 (With Muffler)</td>
<td></td>
</tr>
<tr>
<td>56.</td>
<td>Types of Radial Displacement in Cylindrical Shell</td>
<td>86</td>
</tr>
<tr>
<td>57.</td>
<td>Element of a Cylindrical Shell</td>
<td>87</td>
</tr>
<tr>
<td>58.</td>
<td>Frequency of Pressure vs. Pressure Amplitude</td>
<td>96</td>
</tr>
<tr>
<td>59.</td>
<td>Engine Speed vs. Exhaust Pressure Frequency - Tube No. 4 (No Muffler - Full Load)</td>
<td>99</td>
</tr>
<tr>
<td>60.</td>
<td>Engine Speed vs. Exhaust Pressure Frequency - Tube No. 6 (No Muffler - Full Load)</td>
<td>100</td>
</tr>
<tr>
<td>61.</td>
<td>Engine Speed vs. Exhaust Pressure Frequency - Tube No. 9 (No Muffler - Full Load)</td>
<td>101</td>
</tr>
<tr>
<td>62.</td>
<td>Engine Speed vs. Exhaust Pressure Frequency - Tube No. 20 (No Muffler - Full Load)</td>
<td>102</td>
</tr>
</tbody>
</table>
NOMENCLATURE

c Velocity of sound in ft. per second

\( \text{dB} \) Decibel, unit of sound, re. 0.0002 dynes/cm²

\( \text{dB(A)} \) Sound pressure level in dB with 'A' weighting

E Modulus of elasticity

f Frequency of sound

F Amplitude of exhaust gas pressure in dB

\( F_n \) Natural frequency of tube in cps.

Hz Frequency in cycles per second

\( k_x \) Change of curvature in x - direction

\( k_\phi \) Change of curvature in \( \phi \) - direction

k\( _{x\phi} \) Twist

L Length of exhaust tube

M Mach number of exhaust flow

N Engine speed in rpm

r Distance between microphone and tape recorder

s Engine crankshaft speed in rps

SPL Sound pressure level

T Kinetic energy

\( T_1 \) Temperature of exhaust gas

\( T_2, T_3 \) Surface temperatures of tube

\( \Delta T \) Temperature drop
NOMENCLATURE (Cont'd.)

U  Strain energy
X  Amplitude of shell noise in dB
\varepsilon_{x}  Direct strain in x-direction
\varepsilon_{\phi}  Direct strain in \phi-direction
\sigma_{x}  Stress in x-direction
\sigma_{\phi}  Stress in \phi-direction
\tau_{x\phi}  Shear stress
\gamma_{x\phi}  Shear strain
\nu  Poisson's ratio
\rho  Density
1. INTRODUCTION

Subject of Investigation and Its Importance

The noise emitted by motor vehicles has become one of the most important sources of annoyance from noise. To control traffic noise, legislation has been established, which is based on a standard noise measuring procedure.

Currently, in the majority of cases, the highest contributor to the motor vehicle noise is exhaust noise and the reduction of exhaust noise has been extremely important.

Apart from efficient exhaust silencing, it has been observed that the shell noise produced by the exhaust tube contributes to the exhaust noise.

This investigation is primarily a design and development of the facilities to measure shell noise, with a view to recording and analyzing the noise. The results obtained for different tubes would be used for comparative studies to determine the design criteria of exhaust tubes.

Scope

This investigation deals with the measurement and analysis of the...
pressure pulses in the exhaust system of the engine. This has been correlated with the shell noise characteristics produced in the exhaust tube. Natural frequency of vibration of the exhaust tube has been studied and compared with the experimental results. Effect of engine speed and load on shell noise has also been determined.

This paper includes the test procedure and results of:

1. Pressure transducer response of the gas inside the exhaust tube.
2. Dependence of shell noise on load and r.p.m. of the engine.
3. Exhaust gas temperature and heat emission capacity of the exhaust tubes.

Frequency spectrum of the exhaust pressure and shell noise were obtained. Facilities were available for the measurement of peak and r.m.s. values of sound pressure level and automatic scanning of frequency spectrum with a 3.16 bandwidth filter. In addition to this, static pressure, exhaust gas temperature and temperatures at various points of the exhaust tube were measured.

The above results give the relation between exhaust gas pressure and shell noise characteristics.

From the frequency analysis of the shell noise it was possible to establish the firing frequency of the engine at which resonance of the tube occurs. At that firing frequency again, the relation between the shell noise - sound pressure level and load was investigated.
2. LITERATURE REVIEW

Only in the past decade an international awareness and realization of the importance of noise reduction has been felt very strongly, although the search for a satisfactory theory for exhaust silencer design has continued ever since the internal combustion engine came into use. Extensive experiments and researches have been carried out to determine the complex behaviour of the exhaust system. The study of exhaust noise on the basis of simple acoustic theory has been found to be unsatisfactory. The reason for this is, the exhaust pressure waves are not only finite but large, and no account is taken of the gas flow through the exhaust system. Therefore, it is not possible to predict the back pressure on the basis of acoustic theory, while the dissipation of acoustic energy is not being considered.

Many observers have studied these theories and a good amount of experimental work has been performed, mainly with a view to muffler design. Contributions by Davis, Haynes, Kell and Davies [10] in this field are considerable.

No known data is available on the shell noise characteristics of the exhaust tube.

[10] Numbers shown in square brackets indicate publications listed in reference.
3. THEORY OF EXHAUST NOISE

When the exhaust valve of a single cylinder engine opens, a high pressure pulse of exhaust gas with a steep wavefront travels down the exhaust pipe to the open end. In the absence of reflection from the end of the exhaust tube, a recording of the pressure at the exhaust port would be of the form as shown in Figure 1.

During its early stages in the exhaust tube, it is probable that the wavefront of the impulse travels with a velocity exceeding that of sound. This initial phase soon passes because the rate of loss of energy by the wavefront is large under these conditions, and after a short time, the pulse travels at the prevailing velocity of sound.

As the wavefront passes down the exhaust tube, the front steepens, because the particle velocity of the gas in the region of higher pressure is greater than that in the region of lower pressure. This steepening of the wavefront accounts for the "rasp" of a long unsilenced exhaust tube.

The slug of exhaust gas travels to the end of the exhaust tube and is expelled, but some part of the energy associated with the sound wavefront is reflected back with an 180° phase change. This reflected wave travels back and forth along the tube, gradually losing energy, until
Fig. 1. Idealized waveform of exhaust pulse measured at one point in the exhaust tube.
the next exhaust pulse from the engine arrives.

The energy lost when the main pulse or one of its reflected wavefronts reaches the end of the exhaust tube appears as sound, and thus exhaust noise is emitted from the pipe in pulses. The secondary bursts of noise formed when the re-reflected waves reach the end of the tube contribute little to the overall noise level compared with the energy issuing from the tube when the main pulse arrives.

The secondary pulses arrive at the open end of the tube in time periods \( \frac{1}{l} \) after each other, where \( l \) is the length of the exhaust tube and \( c \) is the velocity of sound. Since, however, they arrive alternately as positive and negative pulses, and phase reversal occurs at each reflection at the open end, the overall time period of the pressure variation is \( \frac{4l}{c} \) and the frequency is \( \frac{c}{4l} \). This is the first pass frequency for a tube open at one end and the dominant sound frequency in the exhaust noise is that firing frequency harmonic which most nearly coincides with the pass frequency or its multiples, \( \frac{3c}{4l} \), \( \frac{5c}{4l} \), etc.

Due to exhaust gas pressure inside the tube, there is radial displacement round the circumference at all cross-sections of the shell. For tubes with thin walls, there is vibration in the fundamental mode, which is flexural vibration of a beam, corresponding to \( n=1 \), where \( n \) is the number of degrees of freedom of vibration. The modes corresponding to higher values of \( n \) are characteristics of the vibrations of shells. This will be discussed later in greater detail.

At various firing frequencies of the engine, the tubes are excited
to produce noise by their vibration. The magnitude of the shell noise should mainly depend upon the vibration amplitude.

**Firing Frequency**

Firing frequency is the total number of ignitions in the cylinders of the engine per second. In the six cylinder engine, there are three ignitions in one turn of the crankshaft. At an engine speed of n r.p.m., the firing frequency is given by

\[
\frac{n \times \frac{3}{60}}{60} \text{ cycles per second.}
\]

**Effect of Exhaust Gas Velocity**

In an actual engine exhaust system, the alternating sound flow may be considered to be superimposed on a steady gas flow. A theoretical approach of determining the effect of flow of medium on the acoustic characteristics of an exhaust system has been made by J. D. Trimmer [5]. The conclusion of the theory is that the effect of velocity of gas on exhaust noise is a function of \(1 - M^2\), where \(M\) is the Mach number of the exhaust flow.

The exhaust tube impedance will vary with the flow velocity. According to the theory, the main effect of increased exhaust velocity is to lower the resonant frequencies of the gas column and to reduce the attenuation due to the tube at these frequencies for which the exhaust tube impedance reaches a maximum. Furthermore, the effect of an increase in temperature would be to move the nodal points farther apart. Thus, if an automobile engine runs under a full load at different
speeds, the higher the speed the higher the temperature and velocity of the exhaust gas. According to the effect of velocity, as mentioned before, the two factors tend to cancel each other, the higher temperature tending to spread the nodal points, the higher velocity tending to crowd them.

On the whole, these effects are probably relatively small, in as much as the exhaust column resonant frequency is reduced by only 9% at a Mach number of 0.3, which corresponds to an exhaust velocity of 600 ft. per second, when c is 2000 ft. per second.

In this investigation, different tubes are tested for comparative study under same test conditions. There will be only negligible differences in gas velocity at a given engine speed. Hence the effect of exhaust gas velocity can be neglected.

Increased Sound-Pressure

In the derivation of the acoustic theory, it is assumed that the sound pressures are very small in comparison with the static pressure of the medium. This assumption is made in order to permit the linearization of the equation of motion. However, some authors believe that, in connection with engine tests certain nonlinear effects were observed, particularly the buildup of sharp wavefronts in long exhaust tubes as evidenced by the explosive character of the sound from such tubes. Due to the high exhaust sound pressure inside the tubes, the nonlinear effects detected may indicate that the application of linearized theory may give errors. However, this investigation is primarily confined to
the measurement of shell noise and the above theory should not affect the shell noise measurement.
4. TEST SET-UP

The engine was installed inside a room, which also housed the ancillary equipments and instrumentation. The external walls of the room were concrete block construction and the intermediate wall between the engine room and storage was lined with glass-fibre to increase sound absorption. This effectively insulated the area so that the engine, intake and fan-noise were largely isolated from the measuring area which was outside.

Due to the presence of high ambient noise in the environment, a special soft chamber was built and the shell noise measurement was performed inside the chamber. The chamber was placed outside the engine room at a distance of 4' from the wall and was mounted on styrofoam bases to help vibration-damping. The chamber provided uniform measuring conditions throughout the year, which were independent of weather conditions such as rain, snow, wind, etc. The engine exhaust system passed through the engine room wall near the measuring chamber. One end of a flexible stainless steel tube was connected to the exhaust manifold of the engine. The other end of the tube protruded through the wall and had a conical seat to eliminate gas leakage due to misalignment with the measuring tube. The exhaust tube for which shell noise measurements were carried out, passed through both end walls of
the chamber, being bolted on the conical seat of the engine exhaust at one end. The other end of the tube was free, having provision to fit the muffler. The openings in both walls of the chamber, where the exhaust tube passed, were sealed by asbestos rope against interference of outside noise. The flexible tube, which is connected to the exhaust manifold, is fitted with a manometer, a thermocouple and a pressure transducer. The assembly is shown in Figure 2.

The temperature of the exhaust tube is measured at two points by two separate thermocouples as shown in Figure 3.

**Engine Equipment**

The test engine was a Chrysler Dodge-Dart slant-six cylinder, water cooled, four stroke 225 cubic inch engine. It was gasoline powered and had a compression ratio of 8.2:1. The engine was mounted on a welded frame with flexible rubber mounts to reduce vibration.

The problem of overheating of the engine was eliminated by providing a reservoir to maintain a constant flow of cold water to the radiator. The arrangement is shown in Figure 6.

The engine was connected to a dynamometer which is used to absorb and measure the power output. The Froude hydraulic dynamometer type D.P.X. was rated at maximum 150 b.h.p. and 4000/7500 r.p.m. Adequate supply of water to the dynamometer was maintained to keep the outlet temperature within allowable limits. Load was regulated by opening the sluice gates of the dynamometer. Measurements were made of engine speed, torque and engine water temperature.
Fig. 2. Arrangement of Manometer, Thermocouple and Pressure Transducer.

Fig. 3. Arrangement of Thermocouples.
Fig. 4: Arrangement of Manometer, Thermocouple and Pressure Transducer

Fig. 5: Exhaust Tube, Showing the End Coupling and Thermocouples
Fig. 6. Water Connection, Showing Cooling Arrangement For Engine
Muffler

A special absorption type muffler was designed and fabricated to study the effect of the reflection of sound in the commercial reaction muffler. A sketch of the muffler is shown in Figure 7.

The outer case of the muffler was 4' X 2' X 1' rectangle, made of thin sheet metal. Inside it a tube was made of flyscreen, about 3" diameter and was bent at several points along the length in the shape of a sausage. The two ends of the tube terminated into the 3" diameter holes in the opposite walls (2' X 1') of the box. One end was fitted with a flange to adapt the free end of the exhaust tube.

Measuring Chamber

The shell noise measurements were carried out inside the chamber to eliminate the effect of any mechanical noise from the engine and ambient noise. The chamber was specially designed to create, as well as possible, a free-field condition. A transmission loss of 30dB was estimated while designing the chamber. Figure 10 shows the sectional view of the chamber. It was made of four unequal walls to reduce the possibility of formation of standing waves. The walls were made of frames with 3/4" thick plywood boards in the outside. Two layers of gypsum boards were used, which were separated from each other by means of wooden spacers, to provide an air gap in the middle which was filled with fibreglass. The air space between the boards was maintained at 4" to obtain greater transmission loss. One gypsum board was fixed to the inner side of the plywood, whereas the other board formed the inside
Fig. 7. Absorption Type Muffler
Fig. 8. Engine and Dynamometer Assembly

Fig. 9. Chamber
wall of the chamber. From inside, all the walls were covered with 1" thick rigid fibreglass insulation. 4" X 1" rigid fibreglass strips were fixed on all the inside walls in different patterns to improve the noise absorption qualities.

One side of the chamber, same construction as the other walls, was made removable, with lining all around, to seal the joints and seams.

Each end of the chamber had a 3" diameter hole in which a flange was fitted for the exhaust tube to go through.

On a side wall, a flanged bushing was provided and a copper tube was made slide-fit to the inside bore of the bushing. A sectional view is shown in Figure 10. The end of the copper tube inside the chamber was carrying the microphone. The cable from the microphone and preamplifier assembly was brought out through the other end of the tube and the end was sealed by rubber plugs.

It was possible to change the distance of the microphone from the exhaust tube by moving the copper tube in and out. Figure 11 shows the microphone and preamplifier assembly inside the chamber. The overall dimensions and the design of the chamber is shown in Figure 10.

Figure 12 shows how the exhaust tube is assembled in the chamber.
Fig. 10. Ch
Fig. 10. Chamber
Fig. 11. Sectional View of Microphone Assembly Inside the Chamber.

Fig. 12. Fitting of Tube in the Chamber.
5. INSTRUMENTATION

The pressure transducer, pyrometer and manometer were connected to the exhaust tube inside the room at a distance of about 4' from the exhaust manifold of the engine. The arrangement is shown in Figure 13.

**Pressure Transducer**

Kistler quartz crystal pressure transducer number 6011L1 was used which was provided with water cooling facilities and was suitable for the measurement of wide range of pressure of fast pressure variation in a relatively high temperature environment.

The specifications of the pressure transducer are as follows:

- **Test Pressure**: 7,500 p.s.i.
- **Full Scale Range**: 5,000 p.s.i.
- **Linearity**: 0 - 50 p.s.i. FS ±1% FS
- **Range Sensitivity**: 0 - 30 p.s.i., Max. deviation 10%
- **Natural Frequency**: 120,000 c.p.s.
- **Capacity**: 5 pf ± 2
- **Operative Temperature Range**: -450°F to +500°F

The exhaust gas pressure in the tube was converted to electrical charge signals by the pressure transducer, featuring high linearity and
Fig. 13. Connection of Pressure Transducer, Pyrometer and Manometer
repeatability over a wide range of pressures and uniform charge sensitivity at very high temperatures. This type of pressure transducer permits measurements of high-frequency pressure variations and fast rise-time components of firing pressures. Quartz crystal pressure transducer used with the charge amplifier, is capable of near-static response.

The basic sensitivity of this instrument is unit charge per unit pressure and is expressed as /picocoulombs per psi (pCb/psi).

Considering the low-frequency response to be measured, in moderately high temperature, the transducer was mounted in an adapter which provided water cooling. The transducer was mounted on an accurately machined surface on the exhaust tube, with its diaphragm flush with the wall of the exhaust tube. Sealing sleeve and ring were used to ensure trouble-free, leakproof operation.

Charge Amplifier

Kistler model 504E Dual Mode Amplifier is an all-solid-state unit for use with piezoelectric transducer. When the dial is set and locked at the transducers sensitivity in pCb per psi, output signals have integral values of psi. Increasing applied pressure produces a negative-going charge signal, which is inverted to a positive-going output signal in the charge amplifier. The amplifier is required to convert the high-impedance charge signal from the transducer to a low-impedance voltage signal for recording on the level recorder. The alternating effects of cable and transducer capacitances are eliminated from
the measuring system.

The transducer was connected to the charge amplifier, which again was connected to the oscilloscope and the direct input of measuring amplifier. The pressure variation was recorded in the level recorder.

Manometer

- The manometer was used to record the static pressure of the exhaust gas. Since the pressure to be recorded was very low, it was necessary to use water.

Thermocouple

A pyrometer was used to record the exhaust gas temperature inside the tube. The range of the instrument was up to 1500°F and was used with a chromel-alumel thermocouple.

Microphone

A capacitive type 1" diameter free field microphone was used - B & K type 4145. This type of microphone can measure noise levels as low as 22 dB. For measuring the shell noise, the microphone inside the chamber was pointed towards the exhaust tube to get a better frequency response. The microphone, together with the other sound measuring equipments, was calibrated to read the sound pressure level (S.P.L.) in dB re. 2 X 10^-5 N/m^2 directly.

Measuring Amplifier

The B & K 2606 measuring amplifier is basically a wide range
voltmeter with interchangeable scales for direct reading of voltage, sound level and acceleration for various transducer sensitivities with the range setting indicated on the scale. Used with the condenser microphone, the 2606 becomes a precision sound level meter to IEC recommendation [13]. The meter indicates true RMS for signals of high crest factor. It also contains the internationally standardized sound level meter frequency weighting networks A, B, C, and D.

Figure 14 shows the characteristics of the different weighting networks.

**Heterodyne Slave Filter**

The R & K type 2022 heterodyne slave filter is a narrow band filter having selective bandwidths 5, 10, 31.6 and 100 Hz. The filter has been used as a frequency analyzer, in combination with the Beat Frequency Oscillator.

**Beat Frequency Oscillator**

The R & K Beat Frequency Oscillator type 1022 is a precision signal generator, specially useful for acoustical measurements. This instrument works with two high frequency oscillators. This instrument is provided with a regular stage so that constant sound pressure or vibration levels may be maintained. The 1022 may be swept continuously through its frequency range by means of an external motor to cover 20-20,000 Hz.
Fig. 14. Internationally Standardised Weighting Curves.

Fig. 15. Frequency Response of Tape Recorder.
**Level Recorder**

The B & K Level Recorder type 2305 is suitable for recording of signal levels in the frequency range 2 Hz to 200,000 Hz. In this particular case, it was used for recording the sound pressure level (S.P.L.) on a time scale as well as for recording the frequency spectrum of a signal. Driven with the Beat Frequency Oscillator the system was automatically synchronized with the frequency calibrated paper QP1102.

**Frequency Analysis**

Figure 14 shows the combination of instruments as used for obtaining a frequency spectrum.

For the frequency analysis of shell noise, the microphone and pre-amplifier assembly is connected to the preamp. input of the measuring amplifier 2606 and for exhaust pressure, the pressure transducer, via the dual mode amplifier, is connected to the direct input of 2606.

The input and output of 2606 are connected to the corresponding input and output of the slave filter 2020.

The Beat frequency oscillator is used to provide 120 kHz and 100 to 200 kHz control frequencies for the slave filter. The slave filter is driven from 1022 by connecting the 120 kHz and 100-120 kHz sockets on the rear of the oscillator to the corresponding sockets on the rear of the slave filter.

The mechanical drive of the level recorder 2505 is used for recording on preprinted frequency calibrated paper. The scanning
**Fig. 36.** Instrumentation For Frequency Analysis and S.P.L. Measurement
Fig. 17. Instrumentation
system of the oscillator is mechanically connected to the drive shaft of the recorder by a special flexible shaft UB 0041.
C. PRELIMINARY MEASUREMENTS

The preliminary tests involved a detailed examination of the interference of mechanical noise from the engine, effect of ambient noise and the behaviour of the chamber as regards to reverberation inside it. The effect of engine speed, engine load, and exhaust tube length on shell noise as well as exhaust gas pressure was studied. The data, thus collected, were used to establish a standard test procedure for the subsequent comparative study of a wide range of tubes.

Transmission Loss of Engine Room

Transmission loss of engine room was measured by means of a speaker, played inside the room at different frequencies. The sound pressure level was first measured by placing the microphone inside the room and then outside the wall. The results were compared and it was observed that TL has been in the region of 30 dB at lower frequencies and about 40 dB at 1000 Hz and higher frequencies.

Ambient Noise

Ambient noise was measured by a microphone, connected to the pre-amplifier input of the measuring amplifier 2606 and plotted on the level recorder. First, the microphone was placed near the exhaust tube, of
which shell noise was to be measured, with 0° angle of incidence, i.e. pointed towards the traffic. Figure 16 shows the ambient noise level in dB on time scale linear response. The same reading was taken with the microphone with 90° angle of incidence and the plot is shown in Figure 19.

**Behaviour of Test Chamber**

The microphone and pre-amplifier assembly was fitted to the copper tube of the test chamber and the microphone was pushed inside the chamber near the exhaust tube. The removable end of the chamber was shut off and all openings were sealed. The ambient noise was measured with the same setting of instrumentation. The recording is shown in Figure 20. By comparing Figures 15 and 20 it is observed that, on the linear scale there is an attenuation of about 30 dB.

The following tests were performed to determine the noise reflection characteristics inside the chamber, particularly to check the formation of standing waves:

A tape recorder was used to record constant electrical signals at various frequencies. The Lohberg tape recorder used for this purpose had a frequency response of 40 - 4,500 Hz ± 2 dB. The frequency response characteristic of the tape recorder is shown in Figure 17.

Other technical specifications of the tape recorder were:

- **Sensitivity**: 0.096 mv/μbar
- **Impedance**: 200 ohms
1. Ambient Noise
90° Angle of Incidence
Distortion below .5% from the amplifier and below 3% from the tape

Noise level 58 dB below signal level at 3% distortion at tape.

A constant voltage signal, equivalent to 90 dB was supplied from the high frequency oscillator to the input of the tape recorder. The frequency of the signal was changed from 40 to 400 Hz in steps of 20 and from 400 to 800 Hz in steps of 50, without changing the amplitude. The recording was then played back through the level recorder as shown in Figure 21 to check for correct reproduction.

The tape recorder was then placed inside the chamber. The microphone was set in front of the speaker of the tape recorder. The chamber was closed and sealed for all possible leaks. The recording was then played back and the response from the microphone was plotted on the level recorder. For each signal the distance between the microphone and speaker was varied and set at 2", 4", 6" and 8". The recording of the microphone output is shown in Figure 22 for frequencies 160, 180 and 200 Hz and is typical for all frequencies. This plot clearly indicates that the noise level changed with the distance of the microphone, showing close resemblance to a free-field condition inside the chamber.

Figure 23 shows the relation between S.P.L. (dB) and log \( r \), where \( r \) is the distance between the microphone and the tape recorder speaker. The readings cover a range from 80 to 260 Hz. At frequencies lower than 80 Hz the amplitude is too low and there is considerable interference of ambient noise. However, the general trend indicating close free field condition was discernible.
<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>60</th>
<th>80</th>
<th>100</th>
<th>120</th>
<th>140</th>
<th>160</th>
<th>180</th>
<th>200</th>
<th>220</th>
<th>240</th>
<th>260</th>
<th>280</th>
<th>300</th>
<th>320</th>
<th>340</th>
<th>360</th>
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<th>420</th>
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<th>580</th>
<th>600</th>
<th>620</th>
<th>640</th>
<th>660</th>
<th>680</th>
<th>700</th>
<th>720</th>
<th>740</th>
<th>760</th>
<th>780</th>
</tr>
</thead>
</table>

**Fig. 21.** Constant Voltage Signal - Output from Tape Recorder
7. **CALIBRATION**

All the instruments were separately calibrated according to manufacturer's recommendations.

The pressure transducer and charge amplifier were calibrated by means of the Kistler calibrator model 541A. The transducer sensitivity of the charge amplifier was set for 10 psi per volt. After the level recorder was calibrated, the exhaust gas pressure, that is, the pressure transducer output as recorded on the level recorder, was converted into absolute values in terms of psi and dB re. 0.0002 dyn/cm².

The absolute value = the value in dB on the recording
- amplification in the measuring amplifier
  + the number of dB indicated by input attenuator, re 10 mv.

From this, it is possible to calculate the amplitude in terms of mv by using the relation:

\[ \text{dB} = 20 \cdot \log_{10} \frac{\text{measured \, mv}}{10} \]

psi can be calculated from the relation 10 psi = 1 volt.

Equivalent sound pressure level (S.P.L.) is given by:
\[
\text{S.P.L.} = 20 \log_{10} \frac{\text{psi} \times 6.8947 \times 10^4}{.0002} \text{ re. } 0.0002 \text{ dynes/cm}^2
\]

For all shell noise measurements, the system was calibrated by means of the B & K sound level calibrator type 423A, which produces a sound pressure level of 93.6 dB for free field microphone re. 2 \times 10^{-5} N/m^2 ± 0.3 dB at the microphone diaphragm. The calibration frequency is 1000 Hz.

By using the calibrator and selecting suitable amplification in the measuring amplifier and proper attenuation in the level recorder, the log paper in the recorder was calibrated. This calibration was used for all subsequent frequency analysis.
III. PRELIMINARY MEASUREMENT OF NOISE

Measurement of Temperature

Temperature was measured at points $T_1$, $T_2$, and $T_3$ as shown in Figure 24, where

$$T_1$$

is the temperature of exhaust gas

and $T_2$ and $T_3$ are surface temperatures at two points of the tube of gauge length 3'.

While taking the temperature readings, the door of the chamber was kept open in order to eliminate the effect of accumulation of heat inside the chamber.

At points $T_2$ and $T_3$, thermocouples were attached to the tube surface with ring clips and were insulated with patches of asbestos.

Temperature drop in the gauge length of 3' is given by

$$\Delta T = T_2 - T_3$$

The section of the permanent exhaust tube between the measuring point $T_1$ and the coupling with the test-tube was insulated partly by being contained in the wall and surrounded by mineral wool and the remainder being wrapped in a fibre blanket. Thus, the drop in temperature of the exhaust gas between $T_1$ and $T_2$ was small. Assuming the exhaust gas temperature inside the tube at the location of $T_2$ to be the same as $T_1$, the temperature difference across the wall of the tube, at entry to the
gauge length of the test-tube, is given by \( T_l - T_A \).

\( T_A \) is the ambient temperature.

Thus, the ratio of \( \frac{\Delta T}{T_l - T_A} \) is a satisfactory basis for comparison of the heat emission capacity of the tested tubes.

**Effect of Muffler**

The theory of the effect of muffler will be discussed here with a view to establishing the actual test procedure. The subject of radiation from an unflanged circular pipe has been dealt with in great detail by Levine, Harold and Schwinger [12]. According to Don D. Davis [8], if the engine is considered to be fitted alternately with an open exhaust pipe and then the same pipe being terminated in a reaction muffler, the reflected waves in the exhaust tube in both cases are very strong. Furthermore, the same sound source is feeding the exhaust tube in both cases. It is obvious that the incident waves will have about the same strength. Thus it is possible, in approximation, to calculate the attenuation as the difference between the sound pressure levels of the incident wave entering the muffler in the exhaust tube and the incident wave leaving the muffler. This approximation should be valid in the frequency range for which the open pipe reflection coefficient is near unity. It has been observed by various authors, that at sufficiently low frequencies, the exhaust tube is terminated in a zero impedance (total reflection) with a phase shift of 180° between the incident and reflected waves. It has already been established that the exhaust tube length has a very definite effect on the sound characteristics of
a complete engine exhaust system. Since the open exhaust tube itself reflects a large part of the sound, it is entirely possible that under certain conditions a muffler could permit more sound to escape than does the open exhaust tube, with a resultant negative attenuation. A negative attenuation value, under the present definition of attenuation, does not imply that sound energy has been created inside the muffler. It means simply that the percentage of the sound energy which reaches the atmosphere is greater with the muffler installed than it is without the muffler.

The following measurements were taken on the same exhaust tube, with and without the muffler:

- Overall Shell Noise (S.P.L.)
- Frequency Analysis of Shell Noise
- Frequency Analysis of Exhaust Pressure.

These observations showed the effect of the muffler. Furthermore, the same muffler used with different tubes, will generally produce different reflection characteristics inside the tubes and the shell noise will be affected.

**Exhaust Gas Temperature**

The velocity of sound at atmospheric temperature is about 1140 feet per second. The higher temperature in the engine exhaust gas will result in a higher sonic velocity. From the data available about the engine, the sonic velocity inside the exhaust tube is estimated to be about 1,600 feet per second. It is believed that the primary effect of
a change in the exhaust gas temperature is the corresponding change in the velocity of sound. By knowing the exhaust gas temperature, the approximate velocity of sound may be determined by using the relation which has been found for air $C = 49\sqrt{T}$ feet per second where $T$ is the absolute temperature on the Fahrenheit scale.

The temperature of exhaust gas at different engine speeds and loads was recorded. For subsequent measurements of shell noise with different tubes, the engine was run with the same loads and at the same speeds and the corresponding gas temperature was maintained.

**Effect of Engine Speed on Exhaust Noise**

Relation between exhaust noise (not shell noise) and engine speed was studied by placing the microphone at a distance of about 6" from the exhaust end and pointing towards the exhaust tube. The tube used for this study was 12' long and 2" diameter. The engine noise at different engine speeds was frequency analysed, measuring the dB level of the amplitude. Figures 25, 26 and 27 show the results obtained at engine speeds of 1000, 1500 and 2000 r.p.m. The frequency spectrums show that the engine exhaust noise is made up of low frequency components. At all different speeds studied, the exhaust noise consists predominantly of the firing frequency and the presence of higher harmonics is noticeable.

The frequency analysis has been carried out at 3.16 bandwidth.

It may be observed from these figures that the exhaust noise level increases with the increase of engine speed.
Fig. 25. Frequency Analysis of Exhaust Noise at 1000 R.P.M.
Effect of Engine Load

A similar study as above was carried out to find the effect of engine load on exhaust noise. The engine speed was kept constant at 1000 r.p.m. and the dynamometer was loaded with 140, 170 and 205 pounds. Figures 29, 29, 30 and 31 show the frequency spectra of the exhaust noise at 3.16 bandwidth with no load on the dynamometer, 140, 170 and 205 pounds respectively.

From these figures it is observed that the frequency of the main harmonics is not changed, due to the fact that the engine speed is the same in all cases. The amplitude of S.P.L. (dB) has changed considerably, increasing with higher loads. It is also clear from these plots that the higher harmonics are predominant with the increase of load.
Fig. 28. Frequency Analysis of Exhaust Noise at 1000 R.P.M. & No Load
Fig. 30. Frequency Analysis of Exhaust Noise
1000 R.P.M. & 170 Lbs. Load
Fig. 31. Frequency Analysis of Exhaust Noise
1000 R.P.M. & 205 Lbs. Load
MEASUREMENT OF SHELL NOISE

After having made the preliminary investigations regarding the effect of tube length, engine speed and torque, actual shell noise measurement was carried out on a tube of the following specifications:

- overall length = 12'
- outside diameter = 2"
- wall thickness = .058"
- material = 1010 hot rolled

The set-up of the instrumentation was the same as explained before and shown in Figure 16. The pressure transducer system was connected to the direct input of the measuring amplifier 2600 and the microphone assembly for shell noise was connected to the pre-amplifier input. Before starting the actual measurement, the system was again calibrated as explained before. All openings of the chamber were sealed properly to eliminate the possibility of interference of engine noise or ambient noise. All measurements were plotted in the level recorder. Before any readings were taken, it was made sure that the engine speed was constant, the engine reached a steady state of running and the exhaust gas temperature was not changing. For both, exhaust gas and shell noise, dB levels were plotted. The following measurements were taken:
Sound Pressure Level of the Shell Noise: This is the microphone response, was recorded on time-base. For doing this, the output from the measuring amplifier was connected to the level recorder and the external filter disconnected. First, the reading was taken "linear" to determine the overall sound pressure level. Then the reading was taken with the frequency weighting network "A".

These measurements were carried out with full load on the engine and at different speeds, from 500 r.p.m. to 1200 r.p.m. in steps of 100.

Frequency Analysis of the Shell Noise: These measurements were also taken with full load on the engine and same speeds as in the previous case. The frequency analysis was carried out at 3.16 bandwidth.

Frequency Analysis of the Exhaust Pressure: This is the pressure transducer output. The frequency analysis was done with the same running conditions of the engine and the same set-up of the instrumentation as for shell noise.

Exhaust Gas Temperature: Exhaust gas temperature was noted at each engine speed and load.

The first set of the above measurements was taken without any muffler at the exhaust end of the tube. Then a commercial muffler was fitted to the exhaust tube and the above process is repeated to get a second complete set of readings.

From an initial examination of the above results, it follows that,
shell noise steadily increases with engine speed and reaches the maximum value at 1000 r.p.m. and after that gradually drops again at still higher speeds. Whereas, the exhaust gas pressure, as is not quite unexpected, steadily increases with engine speed. The occurrence of maximum shell noise at 1000 r.p.m. is obviously due to the natural frequency of the tube in this particular experimental set-up.

After the natural frequency of the exhaust tube was determined, further investigation was carried out at 1000 r.p.m. The first set of readings was taken without a muffler. The engine speed was kept constant and the dynamometer load was changed from 0 to 34, 68 and 102 pounds. First the sound pressure level of shell noise was plotted for both "linear" and "A" weighting under the different loads. Then the frequency analysis of shell noise and exhaust pressure were carried out under the same conditions. The above investigations at 1000 r.p.m. and different speeds were repeated with the muffler on.

For all subsequent measurements of shell noise and exhaust pressure, the commercial reaction type muffler was used.
10. RESULTS AND DISCUSSION

Figure 32 shows the relation between the S.P.L. in dB of shell noise and the r.p.m. of the engine or its firing frequency. In all the cases plotted, S.P.L. reaches the peak at the engine speed of around 1000 r.p.m. The overall effect of noise without the muffler is about 11 dB higher in linear than with a muffler. This difference is about 7 dB with "A" weighting network. The maximum S.P.L. of shell noise with muffler and with "A" weighting is only 61 dB.

From the plots of the frequency spectrum on the level recorder of both shell noise and exhaust pressure, the frequencies of noise and pressure are separated for the same firing frequency of the engine. The ratio \( \frac{x}{F} \) is calculated for the corresponding frequencies where \( x \) is the amplitude in dB of shell noise, and \( F \) is the amplitude in dB of exhaust pressure. The ratio \( \frac{x}{F} \) which is a dimensionless quantity is plotted against the firing frequency, as shown in Figure 33, in the case without a muffler and in Figure 34 with the muffler. For this tube it is clear that without a muffler, maximum \( \frac{x}{F} \) occurs at a firing frequency of 50. This corresponds with the experimental results of S.P.L. measurement as shown in Figure 35. As shown in the graph, Figure 33, this is also true for a second waveform. With a muffler, however, the pattern is slightly different, which could be attributed to the
Figure 33. Relation between Firing Frequency & X
X = Shell Noise
F = Exhaust Pressure
reflection of noise inside the tube.

Next, the $x$ value is plotted against the frequency of sound itself, as shown in Figure 35. It is interesting to note that one component shows its peak at around 85 Hz and repeats itself at around 35 Hz, whereas another waveform shows its peak at 35 Hz. This gives a clear picture of the behaviour of the tube with exhaust gas pressure.

It has been observed that the maximum shell noise occurs at the engine speed of 1000 r.p.m., i.e. the firing frequency of 50 Hz. Figure 36 shows the plot of S.P.L. dB against B.H.P. It is quite obvious that the shell noise increases with load.

Figure 37 shows the relation between $\frac{\Delta T}{T_1-T_A}$ and the engine speed in r.p.m. with full load and no muffler. From the graph it is observed that, heat lost by the exhaust tube depends predominantly on the thickness of the wall and to a lesser extent on the material and construction of the tube. However, the effect of engine speed is comparatively small.

On pressure amplitude and noise amplitude estimated overall instrumentation error is $\pm 3\%$. Predominantly because of engine speed fluctuation, the error in frequency is estimated at $\pm 5\%$. 
Fig. 35. Relation between Frequency & \( \frac{X}{\bar{l}} \)

- \( X \) = Shell Noise
- \( F \) = Exhaust Pressure
FIGURE 37

$\frac{\Delta T}{T_i - T_a}$ vs. ENGINE SPEED

FULL LOAD - NO MUFFLER

PIPE # 1
PIPE # 5
PIPE # 9
PIPE # 20

ENGINE SPEED RPM.
11. COMPARATIVE STUDY

With a view to making a comparative study of the behaviour of exhaust tubes of different construction, the above analysis of shell noise was repeated with the three following tubes:

Tube #6: length (overall) = 12'
0. D. = 2"
thickness = .056
material = 1010 alum.

Tube #9: length (overall) = 12'
0. D. = 2"
thickness = .056 outer laminated
.036 inner
material = 1010 CR both

Tube #20: overall length = 12'
0. D. = 2"
Triple wall laminated,
.036 thick - 1010 CR outer
.030 asbestos paper sandwiched layer
.044 thick - 1010 CR inner

Figures 30 to 35 show the plots of S.P.L. dB against firing
frequency, relation between S.P.I., and B.H.P. at 1000 r.p.m., \( x \) against \( \frac{F}{F} \)

firing frequency and \( x \) vs. frequency of noise for the different tubes.
Fig. 38. Relation between Firing Frequency and Shell Noise
FIG. 39. Relation between Firing Frequency and Shell Noise
Fig. 40. Relationship between Firing Frequency and Shell Noise
Fig. 41. Relation between B.H.I. & Shell Height
Fig. 42. Relation between B.H.P. & Shell Noise
Fig. 43. Relation between B.H.P. & Shell Noise
Fig. 44. Relation between Frequency & $\frac{X}{F}$

$X =$ Shell Noise
$F =$ Exhaust Pressure
Fig. 46. Relation between Frequency & $\frac{X}{P}$

$X$ = Shunt Noise
$F$ = Exhaust Pressure
Fig. 47. Relation between Frequency & \( \frac{X}{F} \)

\( X = \) Shell Noise
\( F = \) Exhaust Pressure
Fig. 48. Relation between Frequency & \( \frac{X}{F} \)

- \( X \) = Shell Noise
- \( F \) = Exhaust Pressure
Fig. 51. Relation between Firing Frequency & $X$

$X$ = Shell Noise
$F$ = Exhaust Pressure
Fig. 53. Relation between Firing Frequency & $X$

$X$ = Shell Noise

$P$ = Exhaust Pressure
Fig. 54. Relation between Firing Frequency & x.

x = Exhaust Pressure

Firing Frequency Hz

x-Shell No. 20 (to muffler)

x-Shell No. 20 (to muffler)
Fig. 55. Relation between Firing Frequency & $X$

$X$ = Shell Noise
$F$ = Exhaust Pressure.
Fig. 56. Types of Radial Displacement in a Cylindrical Shell

Fig. 57. Element of a Cylindrical Shell
Fig. 56. Types of Radial Displacement in a Cylindrical Shell

Fig. 57. Element of a Cylindrical Shell
12. ANALYTICAL VERIFICATION OF SHELL NOISE

In a cylindrical shell, the different types of radial displacement at a particular cross-section are shown in Figure 56 for \( n = 1, 2 \) and 3, where \( n \) is the number of degrees of freedom of vibration. First, the modes corresponding to \( n = 2 \) and higher values are considered to determine the natural frequencies.

An element of a cylindrical shell of mean radius \( a \), uniform thickness \( h \) and length \( d \) is shown in Figure 57. The co-ordinate directions are \( x \), \( y \), and \( z \), measured positively outward from the middle surface. The components of displacement of a point on the middle surface are \( u \), \( v \), and \( w \) in the \( x \), \( y \), and \( z \) directions respectively.

Let \( \varepsilon_x \), \( \varepsilon_y \), and \( \gamma_{xy} \) are the direct strains in the directions \( x \) and \( y \) directions and shear strain respectively at a distance \( z \) from the middle surface.

In the middle surface let \( \varepsilon_x \), \( \varepsilon_y \), are the strains in directions \( x \) and \( y \);

\( k_x \), \( k_y \) are the changes of curvature in directions \( x \) and \( y \);

\( \gamma_{xz} \), \( \gamma_{yz} \) are the shear strain and twist.
Then the strains at a distance \( z \) from the middle surface are given by

\[
\begin{align*}
\varepsilon_x' &= \varepsilon_x + zk_x \\
\varepsilon_y' &= \varepsilon_y + zk_y \\
\gamma_{xy}' &= \gamma_{xy} + 2zk_x \phi
\end{align*}
\]

where

\[
\begin{align*}
\varepsilon_x &= \frac{\partial u}{\partial x} \\
\varepsilon_y &= \frac{1}{a} \frac{\partial v}{\partial \phi} + \frac{\gamma}{a} \\
\gamma_{xy} &= \frac{\gamma v}{\partial x} + \frac{1}{a} \frac{\partial u}{\partial \phi} \\
k_x &= \frac{2}{a} \\
k_y &= \frac{2}{a} \frac{\partial \phi}{\partial x} \\
k_{xy} &= \frac{1}{a} \left( \frac{\partial \phi}{\partial x} \frac{\partial \phi}{\partial y} - \frac{\partial \phi}{\partial y} \frac{\partial \phi}{\partial x} \right)
\end{align*}
\]

The strain energy

\[
U = 0 \int_{-\frac{h}{2}}^{\frac{h}{2}} \int_{-\frac{h}{2}}^{\frac{h}{2}} \left( \varepsilon_x' \varepsilon_x' + \sigma_y' \varepsilon_y' + \tau_{xy}' \varepsilon_y' \gamma_{xy}' \right) \, a \, d\phi \, dx \, dz
\]

The stress-strain relations are

\[
\begin{align*}
\sigma_x &= \frac{E}{1-\nu^2} \left( \varepsilon_x' + \nu \varepsilon_y' \right) \\
\tau_y &= \frac{E}{1-\nu^2} \left( \varepsilon_y' + \nu \varepsilon_x' \right)
\end{align*}
\]
\[
T_x \phi = \frac{E}{2(1-\nu)} Y_x \phi
\]

where \( E \) = modulus of elasticity

\( \nu = \) poisson's ratio

From strain energy expression, after integrating with respect to \( z \),

\[
U = \int_0^{2\pi} \int_0^1 \frac{Eh}{2(1-\nu^2)} \left[ \left( \frac{\partial u}{\partial x} \right)^2 + \frac{1}{a^2} \left( \frac{\partial v}{\partial x} \right)^2 + \frac{2\nu}{a^2} \frac{\partial u}{\partial x} \frac{\partial v}{\partial x} \right] + \frac{1}{12} \left( \frac{\partial w}{\partial x} \right)^2 + \frac{1}{4} \left( \frac{\partial w}{\partial x} \right)^2 - \frac{3\nu}{3\phi} \right]^2 \right] \, d\phi \, dx
\]

The kinetic energy, after integrating with respect to \( z \), is

\[
T = \int_0^{2\pi} \int_0^1 \frac{1}{2} \rho h \left[ \left( \frac{\partial u}{\partial t} \right)^2 + \left( \frac{\partial v}{\partial t} \right)^2 + \left( \frac{\partial w}{\partial t} \right)^2 \right] \, d\phi \, dx
\]

Assuming

\[
u = U(x) \cos \phi \sin \omega t
\]

\[
v = V(x) \sin \phi \sin \omega t
\]

\[
w = W(x) \cos \phi \sin \omega t
\]

For a simply supported shell of length \( l \),

\[
W(x) = C \sin \frac{m\pi x}{l}
\]

where \( m \) is an integer and it is the number of half waves in the axial direction.

Also, \( V(x) = B \sin \frac{m\pi x}{l} \)

and \( U(x) = \Lambda \cos \frac{m\pi x}{l} \)
Substituting into \( \psi \), and integrating,

\[
\psi_{\max} = \frac{n\hbar v}{4\pi(1-v^2)} \left[ \frac{\lambda^4}{4} \frac{A^2}{A^2} + (nB + c)^2 - 2n\lambda A(\phi + nB) \right. \\
+ \frac{\hbar^2}{12\alpha^2} \left\{ \frac{\lambda^4}{4} \frac{C^2}{C^2} + (nB + n^2 c)^2 \right. \\
- 2n\lambda^2 c(n^2 c + nB) + 2(1-v)(\lambda c + \lambda c)^2 \right\}
\]

and \( \psi_{\max} = \frac{1}{4} \pi \hbar \omega \lambda^2 (A^2 + B^2 + C^2) \)

where \( \lambda = \frac{\pi \omega}{v} \)

Applying the Rayleigh-Ritz method in the form

\[
\frac{\partial}{\partial A} \left( \frac{\lambda^2}{2} \frac{A^2}{A^2} \right) = \frac{\partial}{\partial B} \left( \frac{\lambda^2}{2} \frac{B^2}{B^2} \right) = \frac{\partial}{\partial C} \left( \frac{\lambda^2}{2} \frac{C^2}{C^2} \right) = 0
\]

\[
\left\{ \frac{\lambda^2}{2} (1-v)n^2 - \Delta \right\} A - \frac{1}{2} (1-v) \lambda n B - \frac{1}{2} (1-v) \lambda n C = 0
\]

\[
- \frac{1}{2} (1-v) \lambda n A + \left[ \frac{1}{2} (1-v) \lambda^2 + n^2 - \Delta + \beta \left( n^2 + 2(1-v) \lambda^2 \right) \right] B
\]

\[
+ \left[ n + \beta \left( n^3 + (2-v) \lambda^2 n \right) \right] C = 0
\]

\[
- \lambda n A + \left[ n + \beta \left( n^3 + (2-v) \lambda^2 n \right) \right] B + \left[ 1 - \Delta + \beta \left( \lambda^2 + n^2 \right)^2 \right] C = 0
\]

where \( \Delta = \rho a^2 (1-v^2) \frac{\omega^2}{E} \)

and \( \beta = \frac{\hbar^2}{12\alpha^2} \)

where \( \lambda = \text{axial wavelength factor} \).

Eliminating \( A, B \) and \( C \)

\[
\Delta^3 - k_2 \lambda^2 + k_1 \Delta - k_0 = 0
\]
\[ K_0 = \frac{1}{2} (1-v)^2 (1+v) \lambda^4 + \frac{1}{2} (1-v) \beta \left[ (\lambda^2+n^2)^4 - 8 \lambda^2 n^4 - 2n^6 + n^4 \right] \]
\[ K_1 = \frac{1}{2} (1-v) (\lambda^2 + n^2)^2 + \frac{1}{2} (3-v-2v^2) \lambda^2 + \frac{1}{2} (1-v) n^2 + \frac{1}{2} (3-v) \beta (\lambda^2+n^2)^3 \]
\[ \Delta_2 = 1 + \frac{1}{2} (3-v) (\lambda^2 + n^2) \]

\[ \Delta = \frac{K_0}{K_1} + \frac{K_2}{K_1} \left( \frac{K_0}{K_1} \right)^2 \]

Now the above theory is applied to calculate the first few natural frequencies of the tube #4.

For tube #4:

\[ E = 30 \times 10^6 \text{ lb/in}^2, \quad v = 0.3, \quad \rho R = 0.283 \text{ lb/in}^3 \]
\[ a = 0.971, \quad 1 = 66'', \quad h = 0.058 \]

\[ \Lambda = \frac{285}{386.4} \times \frac{0.971^2 (1-0.3^2) w^2}{30 \times 10^6} \]
\[ = \frac{285}{386.4} \times \frac{0.971^2}{30 \times 10^6} \times 0.91 w^2 \]
\[ = 0.21 \times 10^{-10} w^2 \]

\[ \omega = \frac{\Lambda^{1/2}}{\sqrt{0.21 \times 10^{-10}}} \]
\[ = \frac{\Lambda^{1/2}}{\sqrt{0.21}} \times 10^5 \]
\[ = 2.18 \times 10^5 \Delta^{1/2} \]

\[ f = \frac{\omega}{2\pi} = 34730 \Delta^{1/2} \text{ cycles/sec.} \]
\[
\lambda = \frac{m \cdot n \cdot a}{h}
\]

Although the frequency increases with the number of axial half waves, \( m \), other parameters being constant it does not necessarily increase with \( n \). Thus \( m = 1 \) and various values of \( n \) are considered.

Therefore \( \lambda = 0.04622 \)

\begin{align*}
\text{n = 2} & \quad k_0 = 0.015149 & \quad \lambda^2 + n^2 = 4.00214 \\
& \quad k_1 = 7.035 & \\
& \quad k_2 = 6.405 & \\
\text{n = 3} & \quad k_0 = 0.54487 & \quad \lambda^2 + n^2 = 9.00214 \\
& \quad k_1 = 31.52 & \\
& \quad k_2 = 12.153 & \\
\text{n = 4} & \quad k_0 = 6.05122 & \quad \lambda^2 + n^2 = 16.00214 \\
& \quad k_1 = 96.887 & \\
& \quad k_2 = 22.603 & \\
\text{n = 5} & \quad k_0 = 57.613 & \quad \lambda^2 + n^2 = 25.00214 \\
& \quad k_1 = 233.87 & \\
& \quad k_2 = 34.753 &
\end{align*}
Values of the coefficients $k$ and the natural frequencies for various values of $n$ are tabulated below:

<table>
<thead>
<tr>
<th>$n$</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>$K_0$</td>
<td>0.015149</td>
<td>0.54487</td>
<td>6.05122</td>
<td>37.813</td>
<td>166.74</td>
</tr>
<tr>
<td>$K_1$</td>
<td>7.035</td>
<td>31.52</td>
<td>96.887</td>
<td>233.87</td>
<td>485.157</td>
</tr>
<tr>
<td>$K_2$</td>
<td>6.403</td>
<td>12.153</td>
<td>22.603</td>
<td>34.753</td>
<td>49.603</td>
</tr>
<tr>
<td>$K_0$/$K_1$</td>
<td>0.002153</td>
<td>0.017286</td>
<td>0.062456</td>
<td>0.158208</td>
<td>0.343683</td>
</tr>
<tr>
<td>$K_2$/$K_1$</td>
<td>0.000004</td>
<td>0.000191</td>
<td>0.00091</td>
<td>0.003719</td>
<td>0.012080</td>
</tr>
<tr>
<td>$\Delta$</td>
<td>0.002157</td>
<td>0.017401</td>
<td>0.063566</td>
<td>0.161927</td>
<td>0.355763</td>
</tr>
<tr>
<td>$\Delta^{1/2}$</td>
<td>0.0464</td>
<td>0.1319</td>
<td>0.2517</td>
<td>0.4024</td>
<td>0.5965</td>
</tr>
<tr>
<td>$f$(Hz)</td>
<td>1611</td>
<td>4580</td>
<td>8741</td>
<td>13975</td>
<td>20716</td>
</tr>
</tbody>
</table>

According to these calculations, it can be inferred that for $n \leq 6$, the lowest natural frequency is 1611 Hz in the mode $n = 2$. This value is much higher than the frequency of the exciting force, that is, the exhaust gas pressure. Hence, the gas pressure cannot significantly resonate the tube at this frequency.

Now the natural frequency of the tube is calculated by taking $n = 1$, which gives the flexural vibration like a beam.

In this case, the tubes are uniform and simply supported. The
natural frequencies of the various tubes are calculated by using the formula:

\[ \omega_n = \left( \frac{m^4}{I} \right) \frac{E I}{\rho A} \]

where \( I \) = moment of inertia

For tube #4:

\[ \omega_n^2 = \left( \frac{m^4}{I} \right) \frac{E I}{\rho A} \]

\[ = \frac{m^4}{I} \times \frac{50 \times 10^6 \times \pi (D^4 - d^4)}{283 \times 64 \times \pi (D^2 - d^2)} \times 386.4 \times 4 \]

where \( D \) = outer diameter of tube

and \( d \) = inner diameter of tube.

Taking \( D = 2'' \) and \( d = 1.884'' \),

Therefore, \( \omega_n = \frac{m^2 \times 114.64}{\sqrt{D^2 + d^2}} \) radius per second

Therefore \( \frac{\omega_n}{\pi} = 51.04 \) cycles per second, where \( n = 1 \).

This gives the fundamental mode of natural frequency of the tube.

The following table shows the frequency of shell noise peak and the calculated value of fundamental natural frequency of the tubes.

<table>
<thead>
<tr>
<th>Tube</th>
<th>Frequency of Shell Noise in cycles per second</th>
<th>Calculated Value of Natural Frequency of Tube ( \frac{\omega_n}{\pi} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>54</td>
<td>51.64</td>
</tr>
<tr>
<td>6</td>
<td>50</td>
<td>51.69</td>
</tr>
<tr>
<td>9</td>
<td>50</td>
<td>51.28</td>
</tr>
<tr>
<td>20</td>
<td>52</td>
<td>50.31</td>
</tr>
</tbody>
</table>

The natural frequency of the tubes \( F_n \) is shown in Figures 32, 35, 38, 39, 40 and 44 thru 50.
13. RESONANCE OF GAS COLUMN INSIDE THE TUBE

From the readings of the exhaust gas pressure at different engine speeds without load, the graph is plotted as shown in Figure 58 to give the relationship between the exhaust pressure amplitude and the frequency. It is observed that there are several peaks at different frequencies.

The mean gas temperature at idling speed is 650°F. The velocity of sound can be approximately calculated from:

\[ C = 49.03 \times \sqrt{459.7 + T_F} \]

where \( T_F \) = gas temperature in °F

\[ C = 1640 \text{ ft. per second} \]

\[ \frac{C}{4L} = 21 \]

\[ \frac{3C}{4L} = 63 \]

\[ \frac{5C}{4L} = 105 \]

\[ \frac{7C}{4L} = 147 \]

These values are plotted in Figure 58, which shows the pass frequencies of the exhaust tube, at which no attenuation occurs.
Fig. 58. Frequency of Pressure vs. Pressure Amplitude
14. CHARACTERISTICS OF EXHAUST GAS PRESSURE

Figures 59, 60, 61 and 62 show the plots of exhaust gas pressure frequency against the speed of the engine in revolutions per second. Frequency of the exhaust gas increases with engine speed and the relationship is linear. For all the tubes, two different lines are obtained. The gradient of one is twice the gradient of the other at values of approximately 3 and 6 respectively. This gives the following relationship:

\[
\frac{\text{Frequency of gas pressure}}{\text{Engine speed in rev. per sec.}} = 3 \text{ and } 6
\]

In terms of firing frequency: \( \frac{\text{Frequency of gas pressure}}{\text{Firing frequency of engine}} = n \cdot 3 \), where \( n \) is the ratio between firing frequency and engine speed.

Since this is a six cylinder engine, there are 3 firing strokes in each turn of the crankshaft, that is, in each revolution of the engine.

Hence, the frequency of firing of the engine in cycles per second is given by:

\[
\text{engine speed per second} \times 3\text{ i.e. } n = 3
\]

Hence, the frequency of predominant exhaust gas pressure components \( f_{1/2} \) is given by

\[
f = 3n_s \cdot 2n_s
\]

where \( s \text{ rps} \) is the engine crankshaft speed.
Therefore, from the previous relation,

\[
\frac{\text{Frequency of gas pressure}}{\text{Firing frequency of engine}} = 1 \text{ and } 2
\]

That means, the fundamental frequency of gas pressure is the same as the engine firing frequency, with the presence of higher harmonics. Only the fundamental and first harmonic were clearly distinguished. The extension of this analysis to higher orders was not possible because, if present, owing to their relatively low amplitude, they could not be distinguished from background noise.
10. RECOMMENDATIONS

From the results obtained with the different tubes it is quite clear that there are many variables which govern the shell noise characteristics of the exhaust tube. Most important factors are the natural frequency of vibration of the exhaust tube, its size and construction. Engine speed, load and type of muffler are also important factors to influence the magnitude of shell noise.

It is possible to make a very comprehensive study of the shell noise by simulating the exhaust system of an automobile in its entirety. It has been observed that the influence of muffler on shell noise is considerable. To achieve maximum reduction of shell noise it may be necessary to use a muffler suitable for a particular engine and its exhaust system.

Further reduction of shell noise, if required, could be achieved by designing suitable resonators to reduce the pressure components which produce maximum shell noise.

The facilities created through this research project would be ideal to continue with the above investigations experimentally. Experimental verification of shell noise of any exhaust system could be carried out.
16. CONCLUSION

Figure 23 shows the plot of S.P.L. (dB) against log r inside the soft chamber, where r is the distance of the microphone from the source of sound. The results do not show any major deviation from the inverse law of sound pressure $p \propto \frac{1}{r}$. Also, there is no evidence of significant standing wave formation inside the chamber.

The frequency spectrum of the exhaust pressure is the same as that of shell noise in respect of frequency, but amplitude of the peaks are different.

The exhaust pressure increases with engine load and r.p.m. The fundamental frequency of exhaust gas pressure is the same as the engine firing frequency, with the presence of higher harmonics.

The maximum shell noise appears to result from the vibration resonance of the tube in the fundamental mode of transverse bending vibration. Shell noise increases with the increase of load on the engine. Shell noise of tubes is much lower in magnitude than the exhaust noise. With the use of muffler, the shell noise reduces considerably.

The S.P.L. changes for different tubes, depending on their wall thickness and construction, but the difference is small.
BIBLIOGRAPHY


VITA AUCTORIS

I was born in Calcutta, India on March 1, 1934. I completed "Matriculation" from Tirthapati High School, Calcutta in 1949 and in 1953 I received my B.Sc. degree from the University of Calcutta. I got my Bachelor degree in mechanical engineering in 1957 from the Benaras University, Benaras, India.

I worked several years in precision machine tool design in West Germany and India. In June, 1968 I immigrated to Canada and started working in the design and development of transfer type machine tools. I joined the University of Windsor, Ontario, Canada part time for graduate studies in 1970. Later, during 1972-73 I was awarded the PIER fellowship by the National Research Council of Canada and studied full time for M.A.Sc.

In July, 1973, after completing my research at the University of Windsor, I joined Cummins Engine Company to work as Engineer in the applied mechanics laboratory.