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A practical approach to the exhaust silencing of a pneumatic rock drill

Morton Gary Barth

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A PRACTICAL APPROACH TO THE EXHAUST SILENCING OF A PNEUMATIC ROCK DRILL

BY

MORTON GARY BARTH, 1944-

A THESIS

Presented to the Faculty of the Graduate School of the UNIVERSITY OF MISSOURI-ROLLA

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Approved by

[Signatures]
ABSTRACT

One-third octave spectrum analyses of the exhaust noise produced by a medium sized pneumatic rock drill were obtained during free reciprocation to determine the most troublesome bands of noise. Noise reduction techniques were then used to design exhaust mufflers which would reduce the level of the exhaust noise to an acceptable level with a minimum increase in back pressure.

The prototype mufflers evaluated included an expansion chamber, a resonator, and various modified expansion chambers. One of the modified expansion chambers tested provided very good attenuation reducing the exhaust noise from 113 dBA to 87 dBA at the operator's ear position. It caused only a slight increase in the back pressure and no detrimental effect on drill performance. The resonator muffler did not provide acceptable attenuation of the exhaust noise.

Muffler development progressed under the assumption that icing would not be a problem. However, the icing characteristics of the final prototype muffler were studied and a possible method to prevent icing is suggested.
FOREWORD

Authorization was extended to several offices of the United States Bureau of Mines for Noise Research under the Coal Mines Health and Safety Program. One of these offices, the Rolla Metallurgy Research Center, was granted funding to develop a quiet pneumatic rock drill through modifications of internal components, drill case, and muffler during the 1972 fiscal year.

In keeping with the Rolla Metallurgy Research Center's program of close co-operation with the University of Missouri-Rolla, a graduate fellowship was awarded to the author, a student in the Department of Mechanical and Aerospace Engineering, for studies leading to a muffler which could be incorporated into the quiet pneumatic rock drill.
ACKNOWLEDGEMENT

The author wishes to thank his supervising professor, Dr. William S. Gatley, for suggesting this thesis topic and for guidance throughout the research program. The author also desires to acknowledge the support of the United States Bureau of Mines for a Fellowship under which this work was performed.

Particular appreciation is due the author's wife Dorothy Barth for her continual support and encouragement.
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I. INTRODUCTION

Pneumatic rock drills provide the most widely used method of percussion impact used to drill holes in mining operations. They have also been long recognized as one of the major sources of high intensity noise associated with the mining industry.

With the greater concern in recent years over noise as one of the several causes of deafness in the employees of the mining industry, the demand for quieter drilling operations has increased. With this increased demand for quieter drilling operations, a study into the practical exhaust silencing of a pneumatic rock drill will be met with enthusiasm by the mining industry.

In this research program, the exhaust noise of a medium sized pneumatic rock drill was measured and analyzed. A practical muffler was then developed which would reduce the exhaust noise to an acceptable level with a minimum increase in exhaust back pressure.
II. LITERATURE SURVEY

As a part of the United States Bureau of Mines Mineral Industry Health Program, a survey of forty metal, non-metal, and coal mining facilities across the nation was conducted by Derzay and Goodwin (1)* in October and November of 1969 to determine the noise levels that miners were subjected to in their working environments. The survey showed that a large percentage of the employees of the mining industry are working in an environment of excessive noise as compared with currently accepted guide lines for noise control.

A total of 62 different sources of noise creating 90 dBA or more, at a position corresponding to that of the operator's ear, were measured. Of these, 37 created noise levels greater than 100 dBA, 14 greater than 110 dBA, and 4 had values in excess of 120 dBA. All observations of noise levels above 115 dBA were from pneumatic rock drills.

Miller (2) determined that there were three predominant sources of noise produced by the rock drill: air exhaust noise, drill steel vibrations, and noise produced by mechanisms within the drill.

Walker (3) and Holdo (4) each state that the two main sources of noise are the air exhaust noise and the impact or drill steel noise. Holdo calculated, on an energy basis from measurements with an Atlas Copco BBD 45 drill, that 87.5% of the rock drill noise comes from the

*Numbers in parenthesis refer to listings in the Bibliography.
exhaust air and 12.5% from the impact of the piston in the drill against the drill steel, which causes vibrations in the drill steel and in various parts of the drill. He further concluded that, in relation to the useful energy produced by the rock drill, the noise energy was only 0.08%.

A detailed investigation into the sources of noise in pneumatic rock drills was conducted by Beiers (5) to determine the nature and approximate quantitative value of each important noise source in a particular test drill. He concluded that, at the operator's ear position, the order of importance of the individual noise sources contributing to the overall intensity were: exhaust process, mounting rattle between the drill and the pusher leg, movement of the rifle bar, rock penetration, valve movement, and the percussion impact between the piston and the drill steel shank.

Using a drill producing approximately 123 dB at the operator's ear, he was able to arrive at the following values for the three most offending noise sources: exhaust noise - 122.5 dB, mounting rattle - 113.5 dB, rifle bar noise - 109.5 dB.

Through various modifications of and additions to the rock drill, Beiers was able to reduce the overall sound pressure level from 117 dB to 98.5 dB at the operator's ear during an actual drilling operation in the field. This reduction in SPL was at the expense of a 45% reduction in drilling efficiency. The major cause for this reduced efficiency was the use of a nylon valve which oscillated less rapidly than its steel counterpart, causing a reduced penetration rate.
Knowing from previous research (2) that the exhaust noise was the most offending noise source, DeWoody, Chester, and Miller (6) worked to develop a muffler for quieting the exhaust of a rock drill through the use of electrical analogies. Their mufflers, which were reactive in nature, were designed from analogous electrical circuits and proved to be quite effective when tested under no-flow conditions. However, when tested with airflow, it was found that air exiting from the muffler, through a constricted exit opening, produced enough noise to nullify the benefits of the muffler.

Further tests by Chester, DeWoody, and Miller (7) concluded that the shape of the muffler is not critical, within limits, and that the muffler could be incorporated into the body of the drill. They also determined that increasing the number of exit openings while maintaining the same exit area has a markedly beneficial effect on the noise generated by the exhaust air as it exits from the muffler.

Chaffee (8) discusses the problem of reducing the exhaust noise in industrial pneumatic tools in rather broad terms which can be applied to pneumatic rock drills. He states that, due to the expansion of air through the tool itself, exhaust temperatures as low as -10 °F may be encountered. This being the case, care should be taken not to introduce materials into the exhaust flow that will cause icing. He further states that in reducing the air noise, care should be taken not to increase the back pressure, as this will effect the performance of the tool.
He concludes that expansion chambers seem to hold the most promise, as they give attenuation over a wide range of frequencies. Resonators hold little promise due to their rather narrow band attenuation.

In summary, investigators have distinguished the various noise sources in the pneumatic rock drill and have concluded that the unmuffled exhaust noise is the most offending source. Although mufflers have been investigated that are relatively effective under non-flow conditions, little has been published which illustrates how a practical muffler could be constructed to reduce the actual exhaust noise to acceptable levels.
III. EQUIPMENT AND PROCEDURE

In order that sound pressure levels quoted in this work could be compared to measurements taken by other researchers, sound measurement positions were selected according to the CAGI-PNEUROP (9) test code. Since the objective of this research program was to develop a muffler to suppress the exhaust noise, the measurement procedure was modified slightly in that rather than drilling vertically into granite while the noise measurements were being taken, the drill was suspended from ropes and springs and allowed to reciprocate freely (see Chapter VII for details).

A Chicago Pneumatic CP-59 Sinker Drill was made available by the United States Bureau of Mines, Rolla Metallurgy Research Center for studies. This is a medium sized machine (specifications listed on page 11) and is considered to be fairly representative of those commercially available. Tests confirmed (Appendix A) that the exhaust spectrum produced by the CP-59 drill was quite similar to that produced by two other medium sized drills obtained from other manufacturers. This being the case, a muffler designed for the test drill should be applicable to most other rock drill models with only slight modifications and should produce approximately the same reduction in the level of the exhaust noise.

The air compressor was not capable of delivering the required flow rate for sustained operation of the test drill at its recommended pressure, however it could deliver a total air pressure in excess of
100 P.S.I. at the drill. An air regulator and moisture trap were placed in the air line which enabled the supply air to be regulated to a static pressure of 73 P.S.I. at the drill, which is slightly below its recommended air pressure. In this manner, runs of up to two minutes with the freely reciprocating drill could be made with a constant air pressure and flow rate.

The exhaust noise produced during each run was recorded with the aid of a magnetic tape recorder (see Appendix B for list of equipment). A portion of that recorded tape, approximately 26 inches long, was then made into a loop and played continuously through a 1/3 octave analyzer and recorded on chart paper with the aid of a graphic level recorder. The result was a permanent graphic recording of the sound pressure level as a function of frequency*. The tape recorder speed selected was 15 inches per second which gave an excellent response for the tape recorder of ± 2 dB from 50 to 20,000 cycles per second (Hz). The tape loop was the maximum length that the tape recorder could accommodate. With a tape speed of 15 inches per second, this meant that each loop contained about 1 2/3 second of data.

*All 1/3 octave spectrograms in this thesis show the sound pressure level (SPL) in decibels (dB) defined by:

\[
\text{SPL} = 20 \log_{10} \frac{P_{\text{rms}}}{P_{\text{reference}}} \quad (\text{dB})
\]

where \( P_{\text{rms}} \) = root mean square acoustic pressure (dynes/cm²) and \( P_{\text{reference}} \) = reference acoustic pressure (.0002 dynes/cm²).
IV. PRINCIPLE OF DRILL OPERATION AND TEST DRILL SPECIFICATIONS

A. Operation of a Typical Pneumatic Rock Drill

All hand held pneumatic rock drills of the type investigated in this research appear quite similar in external appearance (Figure 1) and have basically the same method of operation. In all drills reciprocation is attained by means of a self acting or automatic valve which admits air first to one end of the cylinder and then to the other with the accompanying opening of suitable exhaust ports at the proper instant. In all but a few machines, the steel is rotated by the piston through the use of a rifle bar and ratchet.

Dickenson and Slager (10) state that valve action can be separated into three main classifications: flapper valves which rock back and forth on a flat seat, spool valves which slide back and forth on guides, and valves operated by the main current of the air itself. Some valves close by seating on flat surfaces, others by one cylinder sliding inside another.

In the operation of a typical pneumatic rock drill described by Beiers (5), compressed air is first admitted to the upper portion of the cylinder which forces the piston forward. At the end of the forward stroke, the piston strikes the shank end of the drill steel which is situated in the chuck bushing. The blow energy is transmitted through the length of the drill steel to the bit which chips away a small portion of the rock face and slightly deepens the hole.
Figure 1. Chicago Pneumatic CP-59 Sinker Drill

(Permission for use obtained from the Chicago Pneumatic Tool Company, ref. drawing no. 1019-C)
Just before the piston strikes the drill steel, the piston head uncovers an exhaust port in the cylinder wall which allows the compressed air in the upper cylinder to expand to the atmosphere. As the piston continues forward to the end of the stroke, it compresses the air which is trapped in the lower cylinder. The combination of the compression of air in the lower cylinder and the releasing of air in the upper cylinder causes the air valve to shift positions, redirecting the supply air to the lower cylinder. The piston motion is therefore reversed and it moves toward the backhead. The piston again uncovers the exhaust port and the same process occurs as in the forward stroke causing the air valve to again reverse its position. The supply air is now redirected to the upper cylinder and the cycle repeats.

Internal groves in the piston head engage with mating inclined splines on the rifle bar to form a system which rotates the drill steel. This allows a new rock surface to be presented to the bit for each blow and prevents the bit from becoming jammed in the hole. On the forward stroke, pawls in the rifle bar head slide over the teeth of the ratchet ring so that the piston does not rotate. On the backward stroke, the ratchet locks and forces the piston to rotate. This rotary motion is transmitted to the chuck bushing which in turn rotates the drill steel.
B. Test Drill Specifications

The drill specifications as listed by the manufacturer for the CP-59 Drill are:

- bore: 2 5/8 inches
- stroke: 2 5/8 inches
- weight: 60 lb
- recommended air pressure: 80 lb/in²
- recommended air consumption: 95 ft³/min
- reciprocation rate: 1860 blows per minute @ 80 P.S.I.
- piston weight: 4.46 lb
- force delivered by piston: 42 pounds per blow @ 80 P.S.I.
- rifle bar: 1:30 (e.g., one rotation for 30 inches of spline)
V. PARAMETERS SELECTED TO EVALUATE MUFFLER PERFORMANCE

In order that the mufflers developed for the suppression of exhaust noise during this research project could be utilized by the mining industry, four parameters were selected to evaluate the prototype muffler performance. These were the insertion loss, back pressure developed, adaptability to pneumatic drills, and icing characteristics.

A. Insertion Loss

Insertion loss is defined as the difference between the SPL at one point in space before and after the muffler is attached to the noise source. One-third octave analyses of the exhaust spectrum were utilized to identify the most troublesome bands of exhaust noise so that noise reduction techniques could be directed toward them. Additional 1/3 octave spectrograms then indicated if these techniques were successful. The overall reading on the "A" weighted network of a sound survey meter was obtained to give an indication of subjective response to the noise.

The Walsh-Healey Act (11) was used as a guideline for the establishment of a design objective for the attenuation produced by the exhaust muffler. It is realized that the operation of a pneumatic rock drill is intermittent with noise being produced during approximately six hours of an eight hour shift. This would mean that the overall sound pressure level produced during operation should be no more than 95 dBA.
Since the exhaust process is only one of the contributors to the overall noise level, many of which exceed 90 dBA, 90 dBA was chosen as the maximum allowable SPL to which the prototype muffler would reduce the exhaust noise.

Although a reduction of the exhaust noise below 109 dBA to 107 dBA would not be detectable in present drilling operations due to the high intensity noise produced by drill steel vibrations, research presently being conducted by the Rolla Metallurgy Research Center is making progress in reducing this source of noise. Drill steel noise, during a drilling operation, has already been reduced below 100 dBA. Continuing research will (hopefully) further reduce this noise source to a level such that the reduction of the exhaust noise to 90 dBA will be necessary to obtain a quiet drilling operation.

B. Back Pressure

Using the penetration rate while drilling vertically into granite as a measure of drill performance (see Chapter VI for additional information), it was determined that the back pressure produced by a prototype muffler should be no greater than 2.0 psi. An air gage placed at the inlet to the muffler allowed the back pressure to be measured without interfering with either the operation of the drill or the muffler attenuation characteristics.

C. Adaptability to Pneumatic Rock Drills

In order that the mufflers developed could be adaptable to new and existing pneumatic drills, it was decided that they must be small
enough to be incorporated into the case of new drills and practical enough to install on existing drills as an add-on feature. Since most drills of the medium classification studied in this research are about the same size, a length of from 8 to 10 inches toward the fronthead from the exhaust port was considered acceptable. It was felt that the cross sectional shape of the muffler need not be circular but could be as wide as the body of the drill, which is approximately 5 inches, and extend outward from the body of the drill by as much as 4 inches. The muffler could be extended from the exhaust inlet toward the backhead in order that air escaping from vents in the valve case of certain drills could be channeled through the muffler.

D. Icing Characteristics

Muffler development progressed under the assumption that icing due to water in the supply air would not be a problem and that in wet drills* the water could be injected into the drill steel in such a manner that it would not enter the cylinder when the drilling operation ceases. This would prevent icing within the muffler due to the discharge of the accumulated water through the exhaust port when the drilling operation is resumed. After prototype mufflers had been developed which were considered acceptable by the first three parameters, their icing characteristics were studied by the introduction of water into the exhaust air.

*Wet drills refer to those pneumatic rock drills in which water, rather than air, is used to remove dust and rock chips from the hole.
VI. DETERMINATION OF AFFECT OF BACK PRESSURE ON DRILL PERFORMANCE

Although Chaffee (8) had stated that in reducing the exhaust noise of pneumatic equipment, increases in back pressure should be avoided due to decreased tool performance, an investigation was conducted to determine what amounts of exhaust back pressure could be tolerated in a medium sized pneumatic rock drill without adversely affecting the drill performance.

The penetration rate into granite during a vertical drilling operation was selected as the measure of drill performance to be used to determine the effect of back pressure. The CP-59 test drill with a 2 ft length of drill steel and a Sandvik Cormant Detachable Bit (H thread, 1 3/4 inch diameter, X-design, with tungsten carbide inserts) was used for each of the test runs. The timing of the length of each run was aided by an electric clock with a sweep second hand. Measurements of the depth of a hole were made before and after each run with a meter stick. Back pressure was produced by placing various constrictions in the exhaust flow. Tests were conducted on separate days with improved techniques on each succeeding day. Each series of tests was completed in one day in order that the air supply pressure and flow rate would be the same for all runs.

During the first series of performance tests, runs were of one minute duration with the supply air pressure regulated to 73 psi. Feed pressure was applied by the operator pushing downward on the drill handle. It was attempted to maintain a constant feed pressure
for all runs. The results of the first series of performance tests are shown in Table I.

Table I
Average Penetration Rates for First Series of Performance Tests

<table>
<thead>
<tr>
<th>Back Pressure (lb/in²)</th>
<th>Average Penetration Rate (cm/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0</td>
<td>11.42</td>
</tr>
<tr>
<td>1.5</td>
<td>12.33</td>
</tr>
<tr>
<td>2.0</td>
<td>11.64</td>
</tr>
<tr>
<td>2.7</td>
<td>11.80</td>
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<tr>
<td>3.0</td>
<td>11.38</td>
</tr>
<tr>
<td>5.5</td>
<td>11.42</td>
</tr>
</tbody>
</table>

Data from this series of tests, which are plotted in Figure 2, seemed to indicate that the penetration rate increased with slight amounts of back pressure reaching a maximum at about 1.5 psi. Due to the relatively short duration of each run, this series of tests was not considered to be completely reliable. A second series of drilling runs was performed with the length of each run increased to two minutes. All other factors were unchanged except for a slightly lower supply air pressure. The results of this second series of performance tests are shown in Table II.
Figure 2. Effect of Exhaust Back Pressure on Penetration Rate
Table II
Average Penetration Rates for Second Series of Performance Tests

<table>
<thead>
<tr>
<th>Back Pressure (lb/in²)</th>
<th>Average Penetration Rate (cm/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0</td>
<td>9.71</td>
</tr>
<tr>
<td>1.5</td>
<td>10.10</td>
</tr>
<tr>
<td>2.0</td>
<td>9.82</td>
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<tr>
<td>2.7</td>
<td>9.81</td>
</tr>
<tr>
<td>5.5</td>
<td>9.30</td>
</tr>
</tbody>
</table>

Again the data (plotted in Figure 2) followed the same trend as during the first series of tests. Although the drilling rates were lower (due to decreased supply air pressure) the penetration was a maximum at about 1.5 psi back pressure and decreased with increasing amounts of back pressure.

In order to eliminate the effect of the operator on the test runs, a third series of drilling runs was designed. In this series, a "dead weight" of 75 pounds was attached to the handle of the test drill. With the dead weight being used to supply the feed pressure, the operator could not influence the results by applying a greater feed pressure on selected runs. The air pressure was approximately the same as used in the first series of tests and a run time of two minutes was utilized. The results of this series of performance tests are shown in Table III.
Table III

Average Penetration Rates for Third Series of Performance Tests

<table>
<thead>
<tr>
<th>Back Pressure (lb/in²)</th>
<th>Average Penetration Rate (cm/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0</td>
<td>10.75</td>
</tr>
<tr>
<td>0.6</td>
<td>11.65</td>
</tr>
<tr>
<td>1.1</td>
<td>11.65</td>
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<tr>
<td>1.9</td>
<td>11.12</td>
</tr>
<tr>
<td>5.6</td>
<td>9.85</td>
</tr>
</tbody>
</table>

During this series of tests, excessive bit wear, which affected the drilling rate, became noticeable in several of the runs in which back pressure was applied to the drill. However, even when these runs are averaged with those runs completed before the bit wear became noticeable, the average penetration rate (also plotted in Figure 2) shows the same trend as the data from the two earlier series.

Using these performance tests as a basis, it was concluded that a muffler for the exhaust air of the test drill could produce a back pressure of up to 2.0 psi without adversely affecting the drill performance. During muffler development, no attempt was made to induce a certain back pressure, rather it was desired to have as little back pressure as possible so that if moisture could not be removed from the supply air, icing would hopefully not become a major problem.
VII. DETERMINATION OF SOUND MEASUREMENT TECHNIQUE

Using the standards established by the CAGI-PNEUROP (9) test code, initial analysis of the noise produced by the test drill was made while drilling vertically into granite. However, it was realized that if significant reductions in the exhaust noise were to be recognized and analyzed the drill must reciprocate in a free running position.

A. Sound Measurements During Drilling Operation

The upper curve of Figure 3 shows the noise spectrum produced during a drilling operation in which the overall SPL measured at a point one meter from the front of the drill and at the same elevation as the geometric center of the drill was 118 dB (116.5 dBA). When a length of thick walled rubber hose of 3/4 inch inside diameter was used to vent the exhaust air away from the measuring location, the overall level was reduced to 112 dB (111 dBA). The noise spectrum for this configuration is also shown in Figure 3. This illustrates that, for the test drill, 75% of the total acoustic energy produced during drilling can be attributed to the exhaust process.

Since the noise produced by rock penetration and the vibrations of the drill steel was of such a high level, the effectiveness of mufflers attenuating the exhaust noise by 10 dB or more could not be measured. It was therefore decided that the drill would be mounted in a vertical position and allowed to reciprocate freely. In this manner, the only noise present would be that produced by the exhaust process and by the internal components of the drill.
Figure 3

One-third Octave Spectrogram for Noise Produced During a Drilling Operation With and Without Exhaust Hose Attached.
Fig. 3
B. Sound Measurements During Free Reciprocation

The drill was suspended from a rope as shown in Figure 4 and allowed to reciprocate freely. Various modifications were then made to the test drill to determine the optimum configuration which would allow the attenuation characteristics of prototype mufflers to be evaluated. The SPL was measured at two positions for each configuration; one corresponding to the same position, relative to the drill, as during the drilling operation (frontal position) and the other position corresponding to that of an operator's ear. One-third octave spectrograms for those sound measurements taken at the frontal measuring position are shown in Figure 5. Figure 6 shows the spectrograms obtained from sound measurements made at the operator's ear position.

With the drill operating in an unmodified configuration, the overall SPL measured at the frontal position was 117 dB (114 dBA), (curve A, Figure 5). The level at the operator's ear position was 112 dB (111.5 dBA), (curve A, Figure 6).

In order to determine if this operating technique would be satisfactory for measuring the effectiveness of mufflers, the exhaust hose was attached to the exhaust port as shown in Figure 7. The overall SPL was reduced to 108 dB (102 dBA) at the frontal measuring position and to 104 dB (100.5 dBA) at the operator's ear position. Curve B of Figures 5 and 6 show the 1/3 octave spectrograms obtained at these two positions.
Figure 4

CP-59 Test Drill Suspended From Rope During Free Reciprocation
Figure 5

One-third Octave Spectrogram for Noise Produced by Test Drill During Free Reciprocation at Frontal Measuring Position

A, Unmodified
B, Exhaust hose attached
C, Exhaust hose attached, rotation chuck plugged
D, Exhaust hose attached, rotation chuck plugged, air from valve case vented away from measuring location
Fig. 5
Figure 6

One-third Octave Spectrogram for Noise Produced by Test Drill
During Free Reciprocation at Operator's Ear Measuring Position

A, Unmodified
B, Exhaust hose attached
C, Exhaust hose attached, rotation chuck plugged
D, Exhaust hose attached, rotation chuck plugged, air from
   valve case vented away from measuring position
Figure 7

CP-59 Test Drill Suspended From Rope During Free Reciprocation with Exhaust Hose Attached
Although the bulk of the exhaust was carried by the exhaust hose, considerable air escaped through the rotation chuck. During the drilling operation, this air passes through the drill steel and is used to blow dust and rock chips from the hole. A plug was placed in the rotation chuck as shown in Figure 3 and the levels recorded as before.

Curve C of Figures 5 and 6 respectively show the noise spectrum obtained when measured at the frontal position, where the overall level was reduced to 103 dB (95.5 dBA), and at the operator's ear position, where the level was reduced to 96 dB (90 dBA). With the sound pressure level reduced to this level, the major source of air noise was now a hole in the front of the drill which vented air from the valve case.

The air coming from this vent was fed into the exhaust hose further reducing the overall SPL to 96 dB (89.5 dBA) at the frontal position and 92 dB (87.5 dBA) at the ear position. The spectrograms for these two positions are shown in curve D of Figures 5 and 6. Escaping air from another vent in the valve case was still the offending noise source. This vent was located under the side bolt and could not effectively be vented to an exhaust hose.

It seemed apparent that even if this source of air noise could be reduced, there would probably be other air leaks, noise produced by the vibrations of the internal components, and a lack of repeatability for measuring the effectiveness of mufflers if the drill were operated in this manner.
Figure 8

Rotation Chuck with Plug Installed
C. Isolation of Exhaust Noise From Body Radiated Noise and Air Leaks

A chamber was constructed in which the drill could be suspended and allowed to reciprocate freely. The exhaust air was vented to the outside of the enclosure with a short piece of one inch inside diameter rubber hose in order that spectrum analyses of exhaust noise could be obtained without the affects of body-radiated noises and air leaks from the drill.

The wall construction of the chamber was of 1/2 inch plywood on a frame of 2x2 lumber. On the inside of the enclosure, a 1/64 inch layer of lead sheet (1 lb/ft²), as shown in Figure 9, and a 6 inch layer of mineral wool, as shown in Figure 10, were attached. Figure 11 shows how the drill was suspended from 2x4's by rubber tubing to minimize vibrations transmitted to the walls. Joints in the plywood were sealed with window caulk and adhesive backed lead tape was used to seal the lead sheet.

The chamber was constructed such that no two surfaces were parallel. In this manner sounds inside the enclosure would not resonate and build up to such amplitudes that they would be transmitted through the walls. The mineral wool was used for sound energy absorption and the lead sheet to provide a high transmission loss through the walls. The lid was constructed in the same manner and a foam rubber weather strip seal was placed between it and the sides. The supply air hose entered the enclosure through a small hole in one side. The space between the hose and the hole was sealed with modeling clay to prevent air leakage.
Figure 9

Wall Construction of Enclosure Showing Lead Sheet
Figure 10

Wall Construction of Enclosure Showing Mineral Wool
Figure 11

Test Drill Suspended Within Enclosure
The outlet for the exhaust air was located 92 cm above a concrete floor and provided a very directional source of noise. The microphone position selected was one meter from the exhaust outlet and 50 cm above the center line of the exhaust air flow. Using this position, the microphone would be close enough to the sound source to allow small variations in the exhaust spectrum to be detected and would be out of the air stream such that air blowing on the microphone would not be a problem.

The spectrum shown in Figure 12 was obtained with the drill operating within the enclosure and the exhaust vented toward the microphone. The overall SPL was 116 dB (114.5 dBA). With the same microphone position, but using the exhaust hose to carry the exhaust noise away from the measuring location, the overall SPL was reduced to 77 dB. The spectrum for this configuration is shown in Figure 13. The high transmission loss through the enclosure allowed analyses of the exhaust noise and prototype mufflers to be performed without the effects of air leaks and body-radiated noise from the drill. The background noise (Figure 14) varied from 62 dB to a maximum of 70 dB and did not affect the exhaust analyses.

The exhaust spectrum obtained when the drill was operating within the enclosure was not exactly the same as that of the free running drill suspended from the rope but the shape of the spectrum was generally the same. There was a slight attenuation, due to the length of the hose, of from 3 dB to 4 dB at almost all frequencies except those in the range from 150 Hz to 250 Hz where the spectrum of the drill operating inside the enclosure was 3 to 5 dB higher.
Figure 12

One-third Octave Spectrogram for Exhaust Noise Produced by Test Drill Operating Within Enclosure
Fig. 12
Figure 13

One-third Octave Spectrogram for Noise Radiated from Enclosure with Test Drill Operating Inside and Exhaust Noise Carried Away from Sound Measuring Location
Figure 14

One-third Octave Spectrogram for Background Noise
Prototype mufflers could now be designed and evaluated using the exhaust noise from the enclosure. Knowing the variations between the actual spectrum and that obtained from the drill operated within the enclosure, corrections could be applied to determine the effectiveness of the muffler when installed directly on the drill.
VIII. DESIGN AND TESTING OF PROTOTYPE MUFFLERS

In the design of prototype mufflers, plane motion of the acoustic wave was assumed to exist within the muffler. With this assumption, the procedures given by reference 12 could be followed in the design of expansion chamber and resonator mufflers.

From a 1/3 octave spectrogram of the unmuffled exhaust, for the test drill operating inside the enclosure, the most troublesome bands of noise were distinguished and noise reduction techniques directed toward them. Additional spectrograms of the muffled exhaust were then made. A comparison of the two spectrograms indicated the effectiveness of the sound reduction technique. Finally, tuning of the muffler was utilized to determine the optimum muffler dimensions.

A. Construction of Prototype Mufflers

For ease of construction, prototype muffler bodies were fabricated from polyvinyl chloride (PVC) tubing having an inside diameter of 2 13/16 inches and an 11/32 inch wall thickness. The inlet end cap was made from a section of 3 1/2 inch diameter solid circular PVC with a centrally located inlet hole. This hole was threaded to accept a standard 3/4 inch pipe thread. A short section of standard 3/4 inch pipe was used to connect the muffler to the exhaust outlet of the enclosure, as shown in Figure 15.

*Tuning consists of varying muffler and/or tail pipe lengths to select the dimensions which give optimum sound attenuation.
Figure 15

Prototype Muffler Attached to Exhaust Outlet of Enclosure
Tail pipes for expansion chambers were made from lengths of PVC tubing having an inside diameter of 29/32 inches and an outside diameter of 1 5/16 inches. These were glued to the exit end plate which was constructed of 1/8 inch thick PVC. Three sections of threaded stock connected the inlet and exit end plates and allowed the position of the latter to be varied; this arrangement permitted the muffler to be tuned by varying the chamber length. Figure 16 shows the construction of a typical expansion chamber muffler.

The single chamber resonator mufflers consisted of an enclosed volume surrounding the exhaust pipe. The exhaust pipe was fabricated from the smaller-sized PVC tubing; communication between the enclosed volume and the exhaust pipe was achieved by holes drilled through the exhaust pipe. The outer shell for the enclosed volume was made from the larger-sized PVC tubing.

The exhaust pipe was glued directly to the inlet end plate. As in the expansion chamber mufflers, threaded stock allowed the position of the exit end plate to be varied for tuning of the resonators. Figure 17 shows the construction of a typical single chamber resonator muffler.

B. Expansion Chamber Muffler

Using the procedure given on pages 28 and 29 of reference 12, an expansion chamber was designed having a length of 8 inches. This muffler should have a transmission loss of 12 dB or more in the range of frequencies from 173 Hz to 480 Hz with a maximum attenuation of 15 dB at 388 Hz.
Figure 16

Expansion Chamber Muffler
Figure 17

Single Chamber Resonator Muffler
The length of the tail pipe was calculated using the procedure given by Harris (13) on page 21-32 who states that a tail pipe will improve the effectiveness of the muffler at frequencies in the vicinity of:

\[ f = \frac{1}{4} \left(\frac{C}{l_t}\right), \frac{3}{4} \left(\frac{C}{l_t}\right), \frac{5}{4} \left(\frac{C}{l_t}\right), \ldots \]

where:
- \( C \) = speed of sound in muffler
- \( l_t = l + .6r \) = effective length of tail pipe
- \( l \) = length of tail pipe
- \( r \) = inside radius of tail pipe

An inspection of the spectrum for the unmuffled exhaust (Figure 18) reveals that attenuation in the vicinity of 1000 Hz would be beneficial. Selection of this frequency as one near which attenuation is needed resulted in a design tail pipe length of 2.85 inches. Although attenuation at many of the lower frequencies would also be beneficial, selection of one of those frequencies would have resulted in a tail pipe length that could interfere with normal drilling operations.

Figure 18 shows the 1/3 octave spectrogram obtained for the design expansion chamber. This muffler reduced the overall SPL by 4 dB to 112 dB (109.5 dBA) and produced a back pressure of 0.9 psi. The attenuation was in two broad bands, one from 150 Hz to 700 Hz and the other from 900 Hz to 5000 Hz.

*See Appendix C for information concerning the calculation of the speed of sound.
Figure 18

One-third Octave Spectrogram for Expansion Chamber Muffler
Fig. 18

SOUND PRESSURE LEVEL, SPL (dB)

PREFERRED ONE THIRD OCTAVE CENTER FREQUENCIES (Hz)

- - - - - - - - - Unmuffled exhaust

- - - - - - - - - Design expansion chamber

- - - - - - - - - Tuned expansion chamber
The procedures given by references 12 and 13 are for the design of mufflers and tail pipes under non-flow conditions. Tuning of the muffler and tail pipe was performed to experimentally determine the optimum lengths under flow conditions.

Tuning of the muffler (Appendix D) resulted in the selection of an optimum chamber length of 8 1/2 inches. Tuning of the tail pipe (Appendix E) resulted in the selection of 3 inches as the optimum tail pipe length. The results of performance testing of the expansion chamber with the optimum dimensions as determined above is also shown in Figure 18. Although the overall SPL of 112 dB (109.5 dBA) was exactly the same as that of the design expansion chamber, this muffler provided additional attenuation at many of the low and medium frequencies thereby making its performance superior to the design muffler.

In order to reduce the sound pressure level to an acceptable level, it appeared necessary to use some type of sound energy absorbing material (Appendix F) within the muffler. In order that the material not be placed directly in the air stream, (thereby causing excessive back pressure and producing a possible icing problem), 3/8 inch mesh hardware cloth was used to form an air passage from the inlet to the outlet of the muffler as shown in Figure 19. The air passage was 1 1/8 inches in diameter and went straight through the center of the muffler. Steel wool filled the volume between the air passage and the outer shell of the muffler.
Figure 19

Air Passage Used in Expansion Chamber Muffler Formed From Hardware Cloth
Figure 20 shows the 1/3 octave spectrogram obtained for the muffler just described. The overall SPL was reduced to 101 dB (95 dBA) and the back pressure was reduced to 0.6 psi. This muffler was considered fairly effective and was chosen for further modification and testing.

C. Resonator Muffler

A resonator muffler was designed using the procedure given on pages 29 and 30 of reference 12. The design objective was to obtain at least 12 dB of attenuation in the range of frequencies from 150 Hz to 400 Hz. The minimum value of the attenuation parameter giving this attenuation is:

\[
\frac{\sqrt{C_o V}}{25} = 5
\]

where: \( C_o \) = conductivity  
\( V \) = resonator volume  
\( S \) = cross sectional area of exhaust pipe

The value of the resonant frequency of the muffler, \( f_r \), was calculated to be 250 Hz and the calculated value of \( \sqrt{C_o V} \) was 0.1265/inch. With these relationships, the values of \( C_o \) and \( V \) were found to be 0.657 inch and 41.1 in\(^3\) respectively. The required length of the resonator was calculated to be 8.50 inches.

An approximate formula, given on page 21-25 of reference 13, for calculating the conductivity of a group of openings is:
Figure 20

One-third Octave Spectrogram for Expansion Chamber Muffler with Steel Wool Liner
Fig. 20

SOUND PRESSURE LEVEL, SPL (dB)

PREFERRED ONE THIRD OCTAVE CENTER FREQUENCIES (Hz)

- - - - - , Unmuffled exhaust
- - - - - , Tuned expansion chamber
- - - - - , Tuned expansion chamber with steel wool liner
\[ C_0 = \frac{n S_c}{\lambda + 0.8 \sqrt{S_c}} \]

where:  
- \( n \) = number of orifices  
- \( S_c \) = individual area of orifice  
- \( \lambda \) = length of orifice

Choosing a size for the connecting holes and the use of this formula allowed the number of connecting holes to be calculated. Calculations revealed that either five 1/4 inch diameter connecting holes or sixteen 0.110 inch diameter connecting holes would satisfy the requirement for \( C_0 \).

Performance testing of the resonator muffler with five 1/4 inch diameter connecting holes revealed that the overall SPL was increased approximately 1 dB above that of the unmuffled exhaust to 117.5 dB (116.5 dBA) and that no measurable amount of back pressure was produced. The 1/3 octave spectrogram shown in Figure 21 reveals two bands of slight attenuation. One of these bands runs from 80 Hz to 800 Hz and the other from 2000 Hz to 5000 Hz. Between 1100 Hz and 2000 Hz the SPL was increased, being 7 dB higher than the unmuffled exhaust at 1500 Hz.

Theoretically, if \( C_0 \) remains constant, varying the size and number of the connecting holes will not change the resonator performance. A second resonator was tested with the same parameters as the first, however in this resonator sixteen 0.110 inch diameter connecting holes were used. Performance testing revealed that the spectrum
Figure 21

One-third Octave Spectrogram for Resonator Muffler
Fig. 21

PREFERRED ONE THIRD OCTAVE CENTER FREQUENCIES (Hz)

SOUND PRESSURE LEVEL, SPL (dB)

- - - - - - Unmuffled exhaust
- - - - - - Resonator muffler
- - - - - - - - Resonator muffler with steel wool liner
from this resonator was exactly the same as that of the first resonator tested.

The addition of steel wool to the enclosed volume of the resonator produced no change in the overall SPL and only a slight change in the 1/3 octave spectrogram which is also shown in Figure 21. As with the two other resonators tested, there was no measurable back pressure produced by this muffler.

Increasing the value of the conductivity (increasing the number or the size of the connecting holes) should improve the attenuation characteristics of the resonator. Additional 1/4 inch diameter holes were drilled in the exhaust tube of the resonator and these configurations were tested. The results of these tests are shown in Table IV and in the accompanying 1/3 octave spectrogram (Figure 22).

Table IV
Effect of Varying the Conductivity on Resonator Performance

<table>
<thead>
<tr>
<th>$C_0$ (inches)</th>
<th>Overall SPL (dB)</th>
<th>&quot;A&quot; Weighted SPL (dBA)</th>
</tr>
</thead>
<tbody>
<tr>
<td>.657 (5-1/4 inch holes)</td>
<td>117.5</td>
<td>116.5</td>
</tr>
<tr>
<td>2.790 (19-1/4 inch holes)</td>
<td>114.5</td>
<td>111</td>
</tr>
<tr>
<td>4.850 (33-1/4 inch holes)</td>
<td>113.5</td>
<td>109</td>
</tr>
</tbody>
</table>

Increasing the value of $C_0$ caused an increase in both the width of each of the bands of attenuation and the amount of attenuation in each of the bands. This increase in the number of connecting holes also caused an increase in the back pressure to 0.2 psi for $C_0=4.850$. 
Figure 22

One-third Octave Spectrograms for Resonator Mufflers with Increasing Values of Conductivity
Fig. 22

- Unmuffled exhaust
- $C_0 = 0.657$
- $C_0 = 2.790$
- $C_0 = 4.850$

Preferred one-third octave center frequencies (Hz)
Steel wool was added to the enclosed volume of the resonator with $C_0=4.850$ to (hopefully) reduce the overall SPL to an acceptable level. The addition of steel wool changed the spectrum, shown in Figure 23, to one wide band of attenuation from 200 Hz to 5000 Hz with a maximum attenuation of 18 dB at 1600 Hz. Since there was no additional attenuation above 5000 Hz or below 150 Hz, the overall SPL was only reduced by 1 dB to 112.5 dB (107 dBA). The back pressure was unchanged.

Increasing the volume of the resonator should also improve the attenuation characteristics of the muffler. Using the resonator with $C_0=4.850$, the length of the chamber was increased from 8 1/2 inches to 10 inches. Steel wool was again used to fill the enclosed volume for sound energy absorption. Performance testing yielded the spectrogram shown in Figure 24 for this configuration. Due to the increased length of the chamber, the overall level was reduced slightly to 112 dB (106 dBA) with no increase in back pressure.

Due to the failure of the resonator mufflers to satisfactorily reduce the overall level of the exhaust noise, they were not considered worthy of further modification and testing.

D. Modified Expansion Chambers

1. Exit Shape Modification

Chester, DeWoody, and Miller (7) had investigated the effect of varying the number of exit openings for expansion chambers while
Figure 23

One-third Octave Spectrogram for Resonator Muffler 8 1/2 Inches Long with $C_0 = 4.850$
Fig. 23

PREFERRED ONE THIRD OCTAVE CENTER FREQUENCIES (Hz)

- , Unmuffled exhaust
- , Resonator muffler
- , Resonator muffler with steel wool liner

SOUND PRESSURE LEVEL, SPL (dB)
Figure 24

One-third Octave Spectrogram for Resonator Muffler 10 Inches Long with $C_o = 4.850$ and Steel Wool Liner
maintaining a constant exit area and had concluded that beneficial effects could be realized, at the lower frequencies, when that number was increased. Using the 8 1/2 inch expansion chamber previously tested, the number of exit openings was increased while maintaining the same exit area as provided with the PVC tail pipe. Four configurations were tested as listed in Table V. Figure 25 shows one of the configurations tested, with eight 5/16 inch diameter exit openings.

Table V
Effect of Varying the Number of Exit Openings on Expansion Chamber Performance

<table>
<thead>
<tr>
<th>Number and Size of Openings</th>
<th>Overall SPL (dB)</th>
<th>Overall SPL (dBA)</th>
<th>Back Pressure (lb/in²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4-1/2 inch dia</td>
<td>113</td>
<td>112</td>
<td>1.0</td>
</tr>
<tr>
<td>8-5/16 inch dia</td>
<td>112-113</td>
<td>112</td>
<td>1.3</td>
</tr>
<tr>
<td>17-.221 inch dia</td>
<td>112</td>
<td>111-112</td>
<td>1.2</td>
</tr>
<tr>
<td>45-.136 inch dia</td>
<td>111</td>
<td>111</td>
<td>1.0</td>
</tr>
</tbody>
</table>

When compared to the SPL obtained with the tuned expansion chamber muffler, increasing the number of exit openings had little beneficial effect. What little added attenuation resulted was more than offset by the added possibility of icing due to exhaust air striking the end plate. This seemed to nullify the slight decrease in the overall SPL below that of the unmuffled exhaust and made the use of multiple openings impractical.
Figure 25

Expansion Chamber Muffler with Exit End Plate with Eight 5/16 Inch Diameter Openings
2. Air Passage Modification

When steel wool was added to the inside of the expansion chamber investigated earlier, the air passage was formed from hardware cloth. Due to its lack of rigidity, it was decided to fabricate this air passage from the small diameter PVC tubing and to drill numerous connecting holes between the air passage and the enclosed volume.

Initially, 120-1/4 inch diameter connecting holes were utilized, as shown in Figure 26, which gave 16.8% open area for the tube. This muffler reduced the overall level to 111.5 dB (106 dBA) and developed a back pressure of 0.2 psi. Figure 27 shows the 1/3 octave spectrogram obtained with the addition of steel wool to the enclosed volume which reduced the overall SPL to 105 dB (98 dBA) with a back pressure of 0.3 psi. This was not as satisfactory as the results obtained when the air passage was formed from the hardware cloth.

The size of the connecting holes was increased to 3/8 inch diameter, as shown in Figure 28, which gave 37.5% open area for the tube. With the 3/8 inch diameter connecting holes, the overall level was reduced to 112 dB (107 dBA) and the back pressure was increased to 0.7 psi which was not quite as favorable as the results obtained using 1/4 inch diameter connecting holes. Addition of steel wool to the enclosed volume reduced the overall level to 102 dB (95.5 dBA) and the back pressure to 0.6 psi. The spectrogram for this configuration, also shown in Figure 27, compares quite favorably with the earlier results obtained using a hardware cloth liner.
Figure 26
Expansion Chamber Muffler with Air Passage Tube Having 120-1/4 Inch Diameter Connecting Holes
Figure 27

Effect of Air Passage Tube Construction on Performance of Expansion Chamber Muffler with Steel Wool Liner
Fig. 27

- Unmuffled exhaust
- Hardware cloth liner
- PVC tube with 1/4 inch diameter holes
- PVC tube with 3/8 inch diameter holes
Figure 28

Expansion Chamber Muffler with Air Passage Tube Having 120-3/8 Inch Diameter Connecting Holes
The muffler constructed using an air passage fabricated from PVC tubing with 3/8 inch diameter connecting holes did not provide quite as great of an insertion loss as the muffler which used an air passage tube formed from hardware cloth. This muffler did provide a rigid air passage and protection to the steel wool liner from the turbulence of the exhaust air with no increase in back pressure. The advantages of the rigid air passage over the flexible hardware cloth seemed to outweigh its disadvantages. PVC tubing with 3/8 inch diameter connecting holes was selected for use as an air passage tube in future mufflers.

3. Extension of Muffler Toward Backhead

While various tests were being made in order to determine a good sound measurement technique, it was noted that air vented from the valve case of the test drill produced a sound pressure level of approximately 102 dB at the frontal measuring position. In order that this air could be vented into the exhaust muffler, an extension of 4 1/2 inches toward the backhead was added to the 8 1/2 inch expansion chamber. The exhaust air then entered at a position near the center of the muffler body and escaped through the tail pipe toward the fronthead as before. The air coming from the valve case entered the muffler extension and escaped at the same point as the exhaust air.

Initially a muffler was constructed in which the exhaust air entered at a right angle to the muffler body. Three configurations of this design were tested, each consisting of an air passage of
differing construction. In each of the three designs, the volume between the air passage and the outer shell of the muffler was filled with steel wool.

Curve B of Figure 29 shows the spectrum obtained when the air passage was fabricated from 3/8 inch mesh hardware cloth. This configuration was the most effective of the three in reducing the exhaust noise. The overall level was reduced to 93 dB (85 dBA) with a back pressure of 1.2 psi. Although this configuration was very effective in reducing the exhaust noise, the hardware cloth air passage offered insufficient protection to the steel wool liner. The air flow was able to tear small pieces of the steel wool loose and discharge them through the tail pipe.

An air passage was formed from the small sized PVC tubing as shown in Figure 30. This air passage utilized 176-1/4 inch diameter connecting holes with the same hole spacing as previously used. Curve C of Figure 29 shows the noise spectrum obtained for this configuration in which the overall SPL was reduced to 99 dB (92 dBA) and the back pressure to 0.8 psi.

In order to improve the attenuation characteristics of the muffler, the size of the connecting holes was increased to 3/8 inch diameter. The results were quite favorable and are shown in Curve D of Figure 29. With this configuration, the overall level was reduced to 96.5 dB (89 dBA) with a back pressure of 1.0 psi.
Figure 29

One-third Octave Spectrogram for Expansion Chamber Muffler with Extension Toward Backhead and Steel Wool Liner, Air Passage Tube Formed from Differing Materials

A, Unmuffled exhaust
B, Hardware cloth air passage tube
C, PVC tube with 1/4 inch diameter holes
D, PVC tube with 3/8 inch diameter holes
Fig. 29

PREFERRED ONE THIRD OCTAVE CENTER FREQUENCIES (Hz)

SOUND PRESSURE LEVEL, SPL (dB)

A, B, C, D
Figure 30

Expansion Chamber Muffler with an Extension Toward the Backhead, Air Passage Formed from PVC Tubing
Due to the advantage of a rigid air passage, and its acceptable attenuation characteristics, the muffler constructed with a PVC air passage tube having 176-3/8 inch diameter connecting holes was selected as the better of the three designs tested.

In order to more accurately model the muffler as it would be used on a rock drill, the air entrance was modified such that it was inclined at approximately 45° to the air passage tube. As determined above, 3/8 inch diameter connecting holes were utilized in the air passage. The volume between the air passage and the outer shell of the muffler was filled with steel wool as in the earlier tests.

The spectrum obtained for this configuration, shown in Curve C of Figure 31, was not as good as that obtained when the air entered at a right angle to the air passage. Although both were essentially the same above 2000 Hz, the right angle entrance provided an average of 3 to 4 dB greater attenuation below 1000 Hz. The overall level of 97 dB (90 dBA) was about the same as that obtained earlier, but the back pressure was reduced to 0.8 psi.

4. Cross Sectional Shape of Muffler Body

For ease of fabrication, initial prototype muffler bodies were of circular cross section. As mentioned in the discussion of the parameters used to evaluate prototype mufflers (page 14), the cross sectional shape of the muffler need not be circular but could be as wide as five inches and extend outward from the body of the drill by as much as four inches.
Figure 31
One-third Octave Spectrogram for Expansion Chamber Muffler with Extension Toward the Backhead and Steel Wool Liner, Air Inlet Inclined at 45° to Air Passage Tube

A, Unmuffled exhaust
B, Air inlet perpendicular to air passage tube
C, Air inlet inclined at 45° to air passage tube
Fig. 31

PREFERRED ONE THIRD OCTAVE CENTER FREQUENCIES (Hz)

SOUND PRESSURE LEVEL, SPL (dB)

A

B

C
In order to determine if this change in cross sectional area would be beneficial, an 8 1/2 inch expansion chamber muffler with an elliptical cross sectional shape was constructed as shown in Figure 32. The major diameter of the ellipse was 5 inches and the minor diameter 2 3/4 inches, which was approximately the inside diameter of the circular muffler. As with previous mufflers, a 3 inch tail pipe length was utilized. Performance testing agrees fairly well with the results obtained using a circular cross sectional shape (Figure 33). The overall SPL was reduced to 111 dB (106.5 dBA) and the back pressure produced was 0.7 psi.

Steel wool was added to the inside of the elliptical muffler. A centrally located air passage formed from the small diameter PVC tubing with 3/8 inch diameter connecting holes was utilized. The overall level was reduced to 102 dB (98 dBA) and the back pressure to 0.4 psi. When a circular cross sectional muffler, with the same type air passage and steel wool liner, was tested, the overall level was also reduced to 102 dB; however the "A" weighted SPL was 95.5 dBA. A comparison of the two spectrograms (Figure 34) shows the same general trend in attenuation with the exception of those frequencies above 5000 Hz where the elliptical muffler is better and between 400 Hz and 5000 Hz where the cylindrical muffler is better. The increased SPL between 400 and 5000 Hz accounts for the higher "A" weighted SPL of the elliptical muffler.

The number of connecting holes was increased to hopefully reduce this troublesome band of noise. Performance testing of this gave the same overall and "A" weighted SPL's as the initial elliptical muffler.
Figure 32

Elliptical Expansion Chamber Muffler
Figure 33

One-third Octave Spectrogram for Elliptical Shaped Expansion Chamber Muffler
Figure 34

One-third Octave Spectrograms for Elliptical and Cylindrical Shaped Expansion Chamber Mufflers with Steel Wool Liner
Unmuffled exhaust
Elliptical cross section
Circular cross section

Fig. 34
A third elliptical muffler 8 1/2 inches long was tested but the air passage for this muffler was formed from 3/8 inch mesh hardware cloth. This configuration reduced the overall SPL to 97 dB (91 dBA) with a back pressure of 0.4 psi. The spectrogram for this configuration (Figure 35) shows considerable attenuation at all frequencies above 250 Hz.

Although the elliptical muffler with the PVC air passage was not as effective as hoped, the elliptical muffler which utilized the hardware cloth air passage indicated the potential of the additional muffler volume for increased attenuation.

Combining the use of the elliptical cross section with the advantage of the extension toward the backhead resulted in the muffler configuration shown in Figure 36. The cross sectional shape of this muffler was elliptical with a major diameter of 5 inches and a minor diameter of 3 inches. The air passage was formed from the small diameter PVC tubing with 3/8 inch diameter connecting holes. The air inlet was inclined at 45° to the air passage and the volume between the air passage and the outer shell of the muffler was filled with steel wool.

Performance testing of this muffler was quite exceptional. The overall level was reduced to 92 dB (86 dBA) with a back pressure of 0.5 psi. The spectrogram for this muffler is shown in Figure 37.

This muffler provides the greatest attenuation of any muffler fabricated, with only a minimum increase in back pressure. It is
Figure 35

One-third Octave Spectrogram for Elliptical Shaped Expansion Chamber Muffler with Steel Wool Liner, Air Passage Formed from Hardware Cloth
Fig. 35

SOUND PRESSURE LEVEL, SPL (dB)

PREFERRED ONE THIRD OCTAVE CENTER FREQUENCIES (Hz)

- - - - , Unmuffled exhaust
- - - , Elliptical cross section
- - - - - - , Circular cross section
Figure 36

Elliptical Shaped Expansion Chamber Muffler

with Extension Toward Backhead
Figure 37

One-third Octave Spectrogram for Elliptical Expansion Chamber Muffler with Extension Toward Backhead
Fig. 37
small enough to fit onto the drill easily and has only one protrusion, the tail pipe, which could result in operator objection through interference with the process of changing drill steels. The length of the tail pipe was reduced to one inch, as shown in Figure 38, such that this would not be a problem.

This muffler did not have quite as good of an insertion loss as the previous muffler, with a three inch tail pipe, but still reduced the exhaust noise to an acceptable level. The overall SPL was now 94 dB (88 dBA) but the back pressure was unchanged (0.5 psi). The 1/3 octave spectrogram for this muffler is also shown in Figure 37.

If the greater insertion loss of the earlier muffler, with a 3 inch tail pipe, is considered essential, the tail pipe could be fabricated from a section of flexible tubing. This could be pulled out of the way when changing drill steels and would have the same attenuation characteristics as the rigid tail pipe.
Figure 38

Elliptical Expansion Chamber Muffler with Extension
Toward Backhead, One Inch Tail Pipe
IX. SOUND MEASUREMENTS OF PROTOTYPE MUFFLERS AT OPERATOR'S EAR POSITION

Although the sound measurements taken thus far are very helpful in determining the amount of attenuation that can be expected for a particular muffler, it is difficult to predict from them what sound pressure levels can be expected at the operator's ear position. It was desired to obtain these measurements without the effect of body radiated noise and air leaks from the test drill. This being the case, the drill was again allowed to reciprocate freely inside the enclosure designed for it. A section of 3/4 inch pipe one meter long was used to extend the exhaust exit from the side of the enclosure. Mufflers were then attached to the end of the pipe with the exhaust flow directed toward the floor as shown in Figure 39.

This arrangement very nearly approximated the position of the exhaust outlet of the muffler during vertical drilling. The concrete floor was a reflecting surface and approximated the top of a granite block or mine surface as would be encountered during an actual drilling operation. The length of the pipe allowed the muffler to be placed far enough from the enclosure such that sound reflected from it would not be a problem.

This length of pipe did cause a variation (on the order of ± 2 dB) at many of the frequencies in the noise spectrum. In order that sound measurements taken at the operator's ear position could be readily compared with those previously taken, corrections were applied to the noise spectrum to account for this attenuation.
Figure 39

Exhaust Exit Modification for Sound Measurements
at Operator's Ear Position
Sound measurements at the operator's ear position were made for those mufflers which showed the greatest potential for reducing the exhaust noise to acceptable levels. Those considered in this classification were:

1) expansion chamber muffler, 8 1/2 inches long.
2) expansion chamber muffler with extension toward the backhead, air inlet inclined at 45° to the air passage tube.
3) elliptical shaped expansion chamber muffler, 8 1/2 inches long.
4) elliptical shaped expansion chamber muffler with extension toward the backhead, air inlet inclined at 45° to the air passage tube.

Each of these mufflers utilized a steel wool liner and an air passage tube formed from PVC tubing with 3/8 inch diameter connecting holes. Muffler #4 had a tail pipe one inch long, the others had a tail pipe 3 inches long.

The overall level of the unmuffled exhaust at the operator's ear position was 115.5 dB (113 dBA) as compared to a level of 116 dB (114.5 dBA) when measured at the frontal position. The SPL's of the muffled exhaust, when measured at the operator's ear position, were on the average 1 to 2 dB lower on both the overall and "A" weighted networks than those obtained at the frontal position. Muffler #1 above (Figure 40) reduced the overall level to 99.5 dB (94.5 dBA) at the operator's ear position as compared to an overall level of 102 dB (95.5 dBA) when measured at the frontal position. The level was
Figure 40

One-third Octave Spectrogram for Expansion Chamber Muffler with Steel Wool Liner at Operator's Ear Position
reduced from 97 dB (90 dBA) at the frontal position to 96.5 dB (88 dBA) at the operator's ear position with muffler #2 (Figure 41) and from 102 dB (98.5 dBA) to 100 dB (95.5 dBA) with muffler #3 (Figure 42). Muffler #4 (Figure 43) was the most effective of those tested and reduced the level at the operator's ear to 93 dB (87 dBA) as compared to a level of 94 dB (88 dBA) when previously measured at the frontal position.

Each of the above listed figures also shows the noise spectrum for the unmuffled exhaust when measured at the operator's ear position and the spectrum for the muffled exhaust when measured at the frontal position.
Figure 41

One-third Octave Spectrogram for Expansion Chamber Muffler with Extension Toward Backhead and Steel Wool Liner at Operator's Ear Position
Fig. 41

Preferred one third octave center frequencies (Hz)

- Unmuffled exhaust
- Operator's ear position
- Frontal position

SOUND PRESSURE LEVEL, SPL (dB)
Figure 42

One-third Octave Spectrogram for Elliptical Shaped Expansion Chamber Muffler with Steel Wool Liner at Operator's Ear Position
Fig. 42

- Unmuffled exhaust
- Operator's ear position
- Frontal position

Preferred one third octave center frequencies (Hz)

Sound pressure level, SPL (dB)

50 100 200 500 1000 2000 5000 10,000
Figure 43

One-third Octave Spectrogram for Elliptical Shaped Expansion Chamber Muffler with Extension Toward Backhead and Steel Wool Liner at Operator's Ear Position
X. ATTENUATION CHARACTERISTICS OF PROTOTYPE MUFFLERS UNDER NON-FLOW CONDITIONS

In order to determine what effect the pulsating exhaust air flowing through a muffler had on its attenuation characteristics, an experiment was conducted in which these attenuation characteristics could be studied without airflow. "Pink noise"* was broadcast through an acoustic driver into the air inlet of four of the prototype mufflers developed. One-third octave spectrum analyses were then performed on the noise produced at the tail pipe of the muffler.

A comparison of the spectrogram for the muffled noise with that for the unmuffled pink noise broadcast from the driver yielded the attenuation provided by a particular muffler. The microphone position was identical for each measurement, relative to the exhaust outlet of the driver or the muffler.

The mufflers studied under non-flow conditions were the same mufflers which were used earlier to obtain sound measurements at the operator's ear position. The attenuation curves for each of these mufflers are shown in the accompanying figures. Each figure also shows the attenuation characteristics of the muffler with air flow. Those mufflers studied under non-flow conditions were:

1) expansion chamber muffler with a circular cross section, (Figure 44);

*"Pink noise" is defined as noise whose spectrum level decreases with increasing frequency to yield constant energy per octave of bandwidth.
2) expansion chamber muffler with a circular cross section and an extension toward the backhead, (Figure 45);
3) expansion chamber muffler with an elliptical cross section, (Figure 46);
4) expansion chamber muffler with an elliptical cross section and an extension toward the backhead, tail pipe one inch long, (Figure 47).

Each of the mufflers studied utilized an air passage formed from PVC tubing with 3/8 inch diameter connecting holes, a steel wool liner, and a tail pipe 3 inches long (except as noted). The mufflers with an extension toward the backhead had an air inlet inclined at 45° to the air passage tube.

It can be seen from the attenuation curves that only slight attenuation was provided by the mufflers at low frequencies. At those frequencies above 1000 Hz, attenuation in excess of 20 dB was not uncommon. The effect of flow was generally the same for each of the mufflers, and tended to increase attenuation at the lower frequencies and to decrease attenuation at the higher frequencies. This would tend to cause an increase in both the overall and the "A" weighted SPL's although the increase in the "A" weighted SPL (dB(A)) would in most cases be greater than the increase in the overall level (dB). For each of the mufflers studied, the trend in attenuation, with and without flow, was generally the same.
Figure 44

Attenuation Characteristics of Expansion Chamber Muffler with Circular Cross Section and Steel Wool Liner
Fig. 44

Preferred one-third octave center frequencies (Hz)

Attenuation (dB)

Without flow

With flow
Figure 45

Attenuation Characteristics of Expansion Chamber Muffler with Circular Cross Section, an Extension toward the Backhead, and a Steel Wool Liner
Fig. 45

Preferred One-Third Octave Center Frequencies (Hz)

- Without flow
- With flow
Figure 46

Attenuation Characteristics of Expansion Chamber Muffler with Elliptical Cross Section and Steel Wool Liner
Fig. 46
Figure 47

Attenuation Characteristics of Expansion Chamber Muffler with Elliptical Cross Section, an Extension toward the Backhead, and a Steel Wool Liner
Without flow

With flow

Fig. 47

PREFERRED ONE-THIRD OCTAVE CENTER FREQUENCIES (Hz)

ATTENUATION (dB)
XI. ICING CHARACTERISTICS OF PROTOTYPE MUFFLERS

No matter how effective a muffler developed for a pneumatic rock drill is in attenuating the exhaust noise, it must be able to operate on the drill without icing to such an extent that the additional back pressure produced, due to ice formation, either stops the drill completely or markedly decreases the penetration rate. The icing characteristics of two mufflers, which were considered to be representative of those developed during this research, were studied.

Both of the mufflers utilized an air passage formed from PVC tubing with 3/8 inch diameter connecting holes and a steel wool liner. The first of these was an expansion chamber muffler, 8 1/2 inches long, with a circular cross section. The second muffler was the elliptical expansion chamber which had an extension toward the backhead. The air inlet for this muffler was inclined at 45° to the air passage tube.

The drill was allowed to freely reciprocate inside the enclosure designed for it. Water was then injected into the exhaust air stream as close to the muffler inlet as possible with the use of a hypodermic needle and syringe. It was injected at a fast enough rate that the needle did not freeze but slow enough that most of the water froze to the inside of the air passage tube rather than being blown out the tail pipe.

When a sufficient amount of icing occurred within the muffler, such that a considerable increase in the back pressure resulted,
sound measurements were taken. These gave an indication as to what extent icing would effect the muffler attenuation characteristics. A visual inspection of the ice formation was made to determine where most of the ice build-up occurred and so that methods for minimizing this phenomenon could be devised.

During an actual mining operation, water vapor in the supply air, rather than water droplets, is the primary cause of icing. Even though water droplets were utilized to induce icing for these tests, it was felt that ice formation would occur in the same locations as if water vapor were present and only the rate of icing would be effected.

From a visual inspection following icing, ice build-up occurred in the same manner for both mufflers. Rather than filling the connecting holes, as may be expected, the ice formed on the wall of the air passage tube causing a gradual constriction of the flow area. This constriction of the flow area caused a corresponding increase in the muffler back pressure and an increase in the flow velocity. In the second muffler studied, with the air inlet inclined at 45° to the air passage tube, considerable ice build-up also occurred where the exhaust air impinged on the wall of the air passage tube directly opposite the air inlet.

One-third octave spectrum analyses were performed on the sound measurements taken after icing. These are shown in Figures 48 and 49 along with the spectrograms for the mufflers without ice build-up.
Figure 48

One-third Octave Spectrogram for Expansion Chamber Muffler, with Icing

A, Unmuffled exhaust
B, Muffled, no icing; 102 dB, 95.5 dBA
C, Muffled, icing; 102 dB, 97 dBA
Figure 49

One-third Octave Spectrogram for Elliptical Expansion Chamber Muffler with an Extension toward the Backhead, with Icing

A, Unmuffled exhaust
B, Muffled, no icing; 94 dB, 88 dBA
C, Muffled, icing; 94 dB, 90 dBA
Fig. 49

SOUND PRESSURE LEVEL, SPL (dB)

PREFERRED ONE THIRD OCTAVE CENTER FREQUENCIES (Hz)
The spectrograms show that, as icing occurs, the increased air turbulence at the muffler exit results in increased acoustical energy at the higher frequencies. The increased back pressure and decreased flow area also causes a smoothing out of the low frequency exhaust pulses and a corresponding decrease in the acoustical energy at the lower frequencies.

For both of these mufflers, the overall sound pressure level remained at the same level with icing as before icing. The "A" weighted SPL was higher with icing than in the same muffler without icing by approximately 2 dBA.

Most pneumatic rock drills utilized in the mining industry today use water rather than air to remove dust and rock chips from the hole. The mufflers developed during this research can be effectively utilized on these drills if the air passage tube is fabricated from two concentric cylinders of heat conducting materials. Connection from the inside of the air passage tube to the enclosed volume of the muffler, which houses the steel wool liner, could be achieved by tubes welded or soldered to both the inner and outer cylinders of the air passage tube. This would allow transmission of the acoustic waves from the exhaust air into the sound energy absorbing liner.

The water to flush the hole could then enter the base of the air passage tube, near the tail pipe, and be redirected from the top of the tube to the water inlet in the backhead of the drill. As the water passes through the air passage tube, it will act as a counter-flow heat exchanger maintaining the temperature of the tube wall above freezing.
If the temperature of the exhaust air maintains an average temperature of -15° F inside the muffler and tail pipe, 0.1 gal/min of water would be required to maintain the temperature of the air passage tube above freezing. This flow rate is not considered unrealistic and is available in most mining facilities.

If the supply air utilized in a mining operation is drawn from an ambient air with a relative humidity of 100%, over 40 cc of water will pass through the drill every minute. The amount of water injected into the prototype mufflers during these tests would simulate approximately 2 minutes of drilling and gave a relatively reliable indication of the actual icing problem that will be encountered in the mine.
XII. SUMMARY AND CONCLUSIONS

The purpose of this research program was to develop a practical exhaust muffler for a pneumatic rock drill. This muffler was required to provide a substantial reduction in the exhaust sound pressure level with a minimum decrease in the drill efficiency. It must be small enough to be incorporated into the case of a rock drill or used as an add-on feature with little or no operator interference.

Studies of the exhaust noise and attenuation characteristics of various prototype mufflers were performed without the effects of body-radiated noise and air leaks from the test drill. Also considered was the effect of exhaust back pressure on drill performance.

The results of performance testing of six mufflers, considered representative of those investigated, are shown in Table VI. Those mufflers listed in this table are:

1) Simple expansion chamber muffler, 8 1/2 inches long.
2) Single chamber resonator muffler, 8 1/2 inches long, $C_0 = 0.657$ inches.
3) Expansion chamber muffler, 8 1/2 inches long, steel wool liner.
4) Expansion chamber muffler with an extension toward the backhead, steel wool liner.
5) Elliptical expansion chamber muffler, 8 1/2 inches long, steel wool liner.
6) Elliptical expansion chamber muffler with an extension toward the backhead, steel wool liner.
Each of those mufflers with a steel wool liner also utilized an air passage fabricated from PVC tubing with 3/8 inch diameter connecting holes. The hole spacing was such that each tube had approximately a 40% open area. All of the expansion chamber mufflers had a 3 inch tail pipe except muffler #6, which had a tail pipe 1 inch long.

Table VI

Performance Tests of Six Prototype Mufflers

<table>
<thead>
<tr>
<th>Muffler Configuration</th>
<th>Frontal Position Overall Sound Pressure Level</th>
<th>Operator's Ear Position</th>
</tr>
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<tbody>
<tr>
<td></td>
<td>(dB)</td>
<td>(dBA)</td>
</tr>
<tr>
<td>1</td>
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<td>2</td>
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<td>116.5</td>
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<td>6</td>
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<td>88</td>
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</table>

From the results obtained, the prototype muffler shown in Figures 38 and 50 was developed. This muffler is shown attached to the test drill in Figures 51 and 52. Also from the results obtained, the following conclusions were drawn.

1. Slight amounts of exhaust back pressure developed by a muffler are not detrimental to the drill performance and may, in some cases, cause an increased penetration rate.
STEEL WOOL LINER

AIR PASSAGE TUBE (12 rows of 3/8 inch dia holes, TYP.)

EXHAUST AIR INLET

AIR INLET FROM VALVE CASE VENT

Figure 50. Prototype Muffler
Figure 51

Prototype Muffler Attached to Test Drill,
Front View
Figure 52

Prototype Muffler Attached to Test Drill,
Side View
2. In order that a muffler be small enough to be incorporated into a pneumatic rock drill and still reduce the exhaust noise to an acceptable level, it must combine reactive and dissipative elements.

3. Care should be taken in muffler development so as not to introduce materials or elements into the exhaust stream which will cause excessive back pressure and possible icing problems.

The prototype muffler developed during this research had an insertion loss of 22 dB and reduced the exhaust noise, at the operator's ear position, from 115 dBA to 87 dBA. The back pressure developed by this muffler is 0.5 psi, which causes approximately a 4% increase in the drilling efficiency for the test drill.

The prototype muffler combined a reactive element (an expansion chamber) with a dissipative element composed of a steel wool liner. The muffler can be utilized without modification in mining operations where the supply air is free of moisture. If the moisture cannot be removed from the supply air, the air passage can be modified, as suggested in Chapter XI, to prevent the accumulation of ice within the muffler.

Research presently being conducted by the Rolla Metallurgy Research Center, an office of the United States Bureau of Mines, is making progress in reducing the noise produced by drill steel vibrations and the internal mechanisms of the drill to produce a quiet pneumatic rock drill. The large reduction in the exhaust sound pressure level is necessary in order that the overall noise level
produced by the quiet pneumatic rock drill will meet those standards established by the Walsh-Healey Act (11).
BIBLIOGRAPHY


VITA

Morton Gary Barth was born in St. Louis, Missouri, on January 29, 1944. He received his primary and secondary education from the Mehlville Public School System, Mehlville, Missouri, which is a suburb of St. Louis. In the fall of 1962 he enrolled in the University of Missouri-Rolla and in May 1967 he received a Bachelor of Science Degree in Mechanical Engineering.

Following graduation he was employed by the Ralston Purina Company, Kansas City, Missouri, in production management. He was drafted into the United States Army in March of 1968 and received an honorable discharge in November of 1970 as a First Lieutenant.

In the Spring of 1971 he enrolled in the University of Missouri-Rolla, to pursue a Master of Science Degree in Mechanical Engineering and was appointed a graduate teaching assistant. In June 1971 he was appointed a Graduate Fellow with the United States Bureau of Mines, Rolla Metallurgy Research Center.

He married Miss Dorothy Johnson in November 1970.

He is a member of Pi Kappa Alpha, Pi Tau Sigma, and Tau Beta Pi Fraternities.
APPENDIX A

Exhaust Noise Produced by Three Medium Sized Pneumatic Rock Drills

In order to determine if mufflers developed during this research project could be effectively used on drills other than the Chicago Pneumatic CP-59 test drill, it was desirable to know if the exhaust noises produced by most medium sized pneumatic rock drills were similar or if each manufacturer's drill would require a distinct muffler design. The exhaust noise produced by the test drill and that produced by an Atlas Copco BBC 16 Drill and a Gardner Denver S 58 Drill were compared.

Each of the three drills was suspended from a rope and allowed to reciprocate freely while sound measurements were taken. For each test, the drill was suspended such that its geometric center was one meter above the floor. The same microphone position (one meter from the front of the drill and at the same height as the geometric center of the drill) was used for each test.

The overall SPL obtained for the Chicago Pneumatic CP-59 Drill was 117 dB (114 dBA), the Atlas Copco BBC 16 produced an overall SPL of 116 dB (113 dBA), and the Gardner Denver S 58 produced an overall level of 118 dB (117 dBA). The 1/3 octave spectrum analyses for the noise produced by each of these drills are shown in Figure A-1.

Although none of the spectrograms are exactly the same, each has the same general shape. The curves are similar enough to indicate that a muffler designed using the exhaust noise produced by the CP-59
test drill should produce approximately the same attenuation when installed on other medium-sized pneumatic rock drills.
Figure A-1

One-third Octave Spectrograms for Noise Produced by Three Medium Sized Pneumatic Rock Drills during Free Reciprocalation
SOUND PRESSURE LEVEL, SPL (dB)

PREFERRED ONE THIRD OCTAVE CENTER FREQUENCIES (Hz)

Fig. A-1
APPENDIX B

List of Equipment

General Radio Company Type 1560-P5 Microphone, Serial Number 4484

General Radio Company Type 1564 Sound and Vibration Analyzer, Serial Number 2397

General Radio Company Type 1521-B Graphic Level Recorder, Serial Number 3332

General Radio Company Type 1564-A Sound Level Calibrator, Serial Number 2606

General Radio Company Type 1382 Random Noise Generator, Serial Number 00689

B & K Type 2205 Sound Level Meter; Microphone- Type 4117, Serial Number 324005; Meter- Type 2205, Serial Number 339542

B & K Type 4230 Sound Level Calibrator, Serial Number 343399

Sony Stereo Tape recorder, Model TC 772, Serial Number 10075

Test Gauge, Duragauge, AISI 316 Tube, St. St. Movement, Serial Number 7ACD-343483-25, IC-1969-6

University Acoustic Driver, Model MA-25, Serial Number 34929
APPENDIX C

Measurement of Exhaust Air Temperature

Temperature measurements made with the aid of a chromel-alumel thermocouple located just inside the exhaust port of the test drill during a drilling operation revealed that the exhaust air temperature ranged from -10 °F to -30 °F. Calculations in the design of mufflers and tail pipes, which involved the speed of sound, were made assuming that the average temperature of the exhaust air inside the muffler and tail pipe would be -15 °F. With this assumption, the calculated value of the speed of sound is 1035 ft/sec.
APPENDIX D

Tuning the Length of the Expansion Chamber Muffler

In tuning of the expansion chamber, both the overall SPL and the 1/3 octave spectrograms for each configuration were used to determine the optimum length for attenuation of overall sound levels. The tail pipe used for each of the tests was the design tail pipe with a length of 2.85 inches. Table D-1 shows the overall SPL's for each configuration tested. The 1/3 octave spectrograms are shown in Figure D-1.

Table D-1
Results of Tuning Length of Expansion Chamber Muffler

<table>
<thead>
<tr>
<th>Muffler Length (in)</th>
<th>Overall SPL (dB)</th>
<th>&quot;A&quot; Weighted SPL (dBA)</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.50</td>
<td>112.5</td>
<td>110.0</td>
</tr>
<tr>
<td>8.00</td>
<td>112.0</td>
<td>109.5</td>
</tr>
<tr>
<td>8.50</td>
<td>112.0</td>
<td>109.5</td>
</tr>
<tr>
<td>9.00</td>
<td>112.5</td>
<td>110.0</td>
</tr>
</tbody>
</table>

Based upon the overall and "A" weighted SPL's and on the 1/3 octave spectrograms, the 8 1/2 inch length was selected as the optimum length. All of the spectrograms were very similar but the 8 1/2 inch length provided 1 to 2 dB greater attenuation in the range of frequencies from 160 Hz to 250 Hz and 1 to 2 dB greater attenuation from 700 Hz to 1200 Hz. The back pressure developed by each of these mufflers was approximately the same (0.9 psi).
Figure D-1

One-third Octave Spectrograms Obtained by Tuning the Length of the Expansion Chamber Muffler
Fig. D-1
APPENDIX E

Tuning the Length of the Tail Pipe for the
Expansion Chamber Muffler

The tail pipe length was calculated knowing that attenuation in
the vicinity of 1000 Hz was desired. Therefore, in tuning the length
of the tail pipe, the SPL in the 1/3 octave band centered at 1000 Hz
was utilized as a measure of tail pipe performance. The entire fre-
quency spectrum, including the overall and "A" weighted SPL's was
also considered to determine if any detrimental effects resulted from
a particular tail pipe length. In each of the tests, an expansion
chamber muffler 8 1/2 inches in length was utilized as the test
muffler. Table E-1 shows the results for each length considered.

Table E-1
Results of Tuning Tail Pipe Length for
Expansion Chamber Muffler

<table>
<thead>
<tr>
<th>Tail Pipe Length (in)</th>
<th>Overall SPL (dB)</th>
<th>&quot;A&quot; Weighted SPL (dBA)</th>
<th>SPL in 1/3 Octave Band Centered at 1000 Hz (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.00 (Figure E-1)</td>
<td>112.5</td>
<td>109</td>
<td>92</td>
</tr>
<tr>
<td>3.50 (Figure E-1)</td>
<td>112.5</td>
<td>109</td>
<td>92</td>
</tr>
<tr>
<td>3.00 (Figure E-1)</td>
<td>112.5</td>
<td>109.5</td>
<td>91</td>
</tr>
<tr>
<td>2.85 (Figure E-2)</td>
<td>112.5</td>
<td>109.5</td>
<td>91</td>
</tr>
<tr>
<td>2.50 (Figure E-2)</td>
<td>112.5</td>
<td>110</td>
<td>93</td>
</tr>
<tr>
<td>2.00 (Figure E-2)</td>
<td>112.5</td>
<td>110</td>
<td>94</td>
</tr>
</tbody>
</table>
From a comparison of the 1/3 octave spectrograms, it can be seen that as the tail pipe length is decreased the amount of attenuation at those frequencies below 300 Hz increases slightly while at the same time the SPL at 1500 Hz increases. This increase in the SPL at 1500 Hz accounts for the increasing value of the "A" weighted SPL. It was felt that the addition of some type of sound energy absorbing material to the muffler (verified in Appendix F) would attenuate the spectrum above 500 Hz such that the advantage of having additional attenuation at the lower frequencies would offset the increased SPL at 1500 Hz.

Since the 3 inch tail pipe provided exactly the same spectrum as the design tail pipe length of 2.85 inches, and each of these provided the most "well rounded" spectrum, the 3 inch tail pipe was selected as the optimum length.
Figure E-1

One-third Octave Spectrogram for Expansion Chamber Muffler with 4.00 Inch, 3.50 Inch, and 3.00 Inch Tail Pipes
Unmuffled exhaust
4.00 inches long
3.50 inches long
3.00 inches long

Fig. E-1
Figure E-2

One-third Octave Spectrogram for Expansion Chamber Muffler with 2.85 Inch, 2.50 Inch, and 2.00 Inch Tail Pipes
Fig. E-2

PREFERRED ONE THIRD OCTAVE CENTER FREQUENCIES (Hz)

SOUND PRESSURE LEVEL, SPL (dB)

- Unmuffled exhaust
- 2.85 inches long
- 2.50 inches long
- 2.00 inches long
APPENDIX F

Selection of Sound Energy Absorbing Material to be Used in Prototype Muffler Design

Three types of sound energy absorbing material were considered for possible use in the prototype mufflers; mineral wool, steel wool, and polyurethane foam. An expansion chamber muffler with a chamber length of 12 inches and a three inch tail pipe was used as the test muffler. An air passage of 3/8 inch mesh hardward cloth, 1 1/8 inches in diameter was located in the center of the expansion chamber and the sound energy absorbing material filled the volume between the air passage and the outer shell of the muffler. Substitution of the various materials gave the results shown in Table F-1 and in the accompanying spectrogram.

Table F-1
Results of Tests Using Various Sound Energy Absorbing Materials in an Expansion Chamber Muffler

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Overall SPL (dB)</th>
<th>&quot;A&quot; Weighted SPL (dBA)</th>
</tr>
</thead>
<tbody>
<tr>
<td>mineral wool (densely packed)</td>
<td>98.5</td>
<td>93</td>
</tr>
<tr>
<td>steel wool (medium packing density)</td>
<td>98.5</td>
<td>91</td>
</tr>
<tr>
<td>polyurethane foam (densely packed)</td>
<td>99</td>
<td>93</td>
</tr>
</tbody>
</table>

In order to select the best sound energy absorbing material, the expected length of service before replacement must also be considered along with the attenuation provided. Each of the configurations was
Figure F-1

One-third Octave Spectrogram for Expansion Chamber Muffler with Various Sound Energy Absorbing Materials as Liners

A, Unmuffled exhaust
B, Steel wool
C, Mineral wool
D, Polyurethane foam
Fig. F-1
inspected following its testing to determine if any detrimental effects resulted.

All three of the materials became oil soaked; however, the oil would seep out of the steel wool during periods when the drill was not in operation. Some of the oil seeped out of the mineral wool, but not as much as from the steel wool. Due to the tight cell construction of the polyurethane foam, very little of the oil that it absorbed seeped out.

Air turbulence within the muffler caused small pieces of the mineral wool to be torn loose and blown out the tail pipe. It was felt that the mineral wool would not have a useful life long enough to make its use as a sound energy absorbing material within the muffler practical. Neither the steel wool nor the polyurethane foam were damaged by the air flow.

Although the effect of moisture was not investigated, the probable effect on polyurethane foam would be the same as oil. Water would cause deterioration of the mineral wool and rusting of the steel wool. Rusting of the steel wool could be overcome with the use of stainless steel wool.

Although polyurethane foam and mineral wool provided greater attenuation at frequencies above 1000 Hz than the steel wool, the steel wool was more efficient in attenuating those frequencies below 1000 Hz. Based upon the attenuation provided and the practical usefulness of each, steel wool was selected as the best sound energy
absorbing material of the three and was utilized in prototype muffler design and testing.

It was found that a medium packing density (10-12 lb/ft$^3$) of steel wool provided almost as good attenuation characteristics as a very heavy packing density (16-20 lb/ft$^3$). A light packing density (3-5 lb/ft$^3$) did not provide acceptable attenuation. In all tests, No. 1 (medium grade) steel wool was used. Figure F-2 shows the 1/3 octave spectrograms obtained for various packing densities of steel wool in the test muffler.

No. 1 (medium grade) steel wool with a medium packing density was used in all instances where a steel wool liner was specified during prototype muffler design and testing.
Figure F-2

One-third Octave Spectrograms for Expansion Chamber Muffler with Steel Wool Liners of Various Packing Densities

A, Unmuffled exhaust; 116 dB (114.5 dBA)

B, 4 lb/ft$^3$; 101.5 dB (95 dBA)

C, 11 lb/ft$^3$; 98 dB (90.5 dBA)

D, 18 lb/ft$^3$; 97 dB (89 dBA)
Fig. F-2

PREFERRED ONE THIRD OCTAVE CENTER FREQUENCIES (Hz)

SOUND PRESSURE LEVEL, SPL (dB)