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Final Report

EXHAUST SILENCING OF THE T-95 TANK ENGINE

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ABSTRACT

This report describes an investigation into the problem of designing an effective muffler for the T-95 tank engine. The muffler is designed to be mounted in the engine cooling air duct and arranged so that the exhaust gas mixes with the cooling air to provide an overall reduction in the temperature of the discharged gases.

A simplified method of sound analysis is described for comparison of the effectiveness of various muffler designs.

Air-flow measurements are made to determine the allowable restriction to be placed in the air duct.

Various types of mufflers are tested both on an actual engine and on a white-noise generator.

The design of acoustic filters, mufflers, and associated piping are treated in detail. The design of a specific muffler for the AOS-895-3 engine is developed with pictures of its partial construction. Test results of this muffler on the AOS-895-3 engine indicate about a 3-db higher noise level in the firing rate than the Maxim silencers. This was accomplished with mufflers of approximately one-half the Maxim muffler volume at a back pressure of 2.25 in. Hg, 2400-rpm full load. With a back pressure of 2.0 in. Hg, the firing-rate level increases about an additional 2 db. If an increase in back pressure to perhaps 2.5 to 3.0 in. Hg could be tolerated, it may be possible to reduce the exhaust level to a point where it is masked by the fan noise and thus inaudible to the ear.

The muffler type chosen as most suitable for this purpose is a mechanical pulse absorber streamlined to provide negligible restriction to cooling-air flow.

The temperature of mixed gases and cooling air can be limited to about 300°F (2400-rpm, 800-ft-lb, 120°F fan discharge air) with the use of exhaust wings on the muffler.

OBJECTIVE

The object of this investigation is to develop a silencing and exhaust system, suitable for enclosure in the waste cooling-air duct of the T-95 Medium Tank and which shall most nearly satisfy the following requirements:

1. Reduction of engine noise to a level similar to that of combat vehicles now in the field.

2. Durability to be at least 500 hr of operation.

3. Back pressure of the engine exhaust due to the muffler not to exceed 2 in. Hg.

4. No restriction to cooling-air flow greater than 1 in. H_2O.

5. Reduction of exhaust temperature as much as possible by mixing the exhaust gas and the waste cooling air to reduce infrared detection.

6. Permit routine maintenance of engine and component parts with minimum interference.

7. Sufficiently flexible design to allow installation in a variety of vehicles with a range of horsepowers from 100 to 1000, merely by using multiple units or by redimensioning the unit without changing the general configuration or internal construction, if such flexibility be attainable.

TERMINOLOGY

Sound Level: Sound level, measured in decibels, is the sound-pressure-level reading obtained with a sound-level meter. Characteristics of the sound-level meter are determined by the American Standards Association. The reference pressure used is .0002 microbar (.0002 dyne/cm^2), which is about the threshold of hearing at 1000 cps. This reference pressure serves as the 0-db level.

Decibel (db): Using a reference pressure of .0002 microbar (.0002 dyne/cm^2), the sound-pressure level (L_p) in decibels is

$$L_p = 20 \log_{10} \frac{p}{.0002} \text{ db} \quad .$$

The decibel is thus a function of the log of the sound-pressure ratio. Example: Reduction of a sound level from 100 db to 94 db is the reduction of the sound pressure to 1/2 value, i.e., 20 to 10 microbars.

Sound Spectrum: The sound spectrum is the continuous range of audio frequency components, and as a function of time is a description of its resolution into components which are sinusoidal functions of time and can be different phase and amplitudes.

Octave Band: An octave band is a range whose extremes are doubled in frequency: 100-200 cps, 200-400 cps, 60-120 cps. One-third octave bands are similar to octave-band analysis using 1/3 octave per band for more detailed analysis.

White Noise: White noise is noise of random frequencies (differing from a tone), the amplitudes of which are also random but whose average power is constant per unit band width.

Masking: Masking is the effect of the presence of one sound which raises the threshold of audibility of a second sound.

Pass Frequencies: Pass frequencies are those which can pass through a filter with little or no attenuation.

I. INTRODUCTION

The list of requirements stated under the heading of Objective is in effect a list of problems involved in providing satisfactory silencing for this particular engine. Several of these conflict in the attainment of an optimum solution. For example, the minimum restriction to cooling-air flow and the provision for ease of maintenance demand a minimum-size silencing device, whereas low engine back pressure and low sound level are improved, usually, by large-volume silencing devices. Therefore it is obvious that a compromise solution must be accepted to meet all the requirements in the best way.

To develop suitable silencing devices, we should examine the fundamental conditions existing in an exhaust system. First, consider the exhaust-gas flow. At the beginning of the lift of the exhaust valve, a pressure differential exists across the valve, forcing the gas into the exhaust manifold. This pressure difference exceeds the critical pressure ratio for sonic flow with the result that there exists sonic and supersonic velocity flow for a short period. After this period, flow becomes a function of the cylinder pressure, the manifold pressure, and the valve opening. Whatever the flow condition, flow ceases with the closing of the exhaust valve. At this instant, the gas is a forward moving pulse which is retarded by the partial vacuum it creates in the closed pipe chamber behind it. This may be thought of as a mass-spring system, the mass being the gas pulse which is traveling along the tube, the spring being the rarefied gas behind the pulse. This cycle is repeated each time an exhaust valve opens and it results in a succession of very-high-velocity pulses traveling through the manifold.

The exhaust sound sources can now be considered in relation to these pulses. The initial expansion through the valve can create sound energy as a shock wave when the pressure ratio is greater than critical when the flow is sonic and as turbulent noise when the flow is subsonic. The pulse itself is capable of generating very high sound levels as it enters the atmosphere from the exhaust system. A product of this flow pattern is a fundamental frequency equal to the pulse rate.

$$\text{Fundamental frequency (cps)} = \frac{xn}{120}$$

where

x = engine rpm
n = exhaust discharges into a manifold for each camshaft revolution.

This is true if the engine firing order for that manifold has equally spaced timing. Two manifolds exhausting into a common point would double this frequency (n = no. of pulses into both manifolds).

Another factor which is quite important concerning the pulses is a resonant condition which can occur in an exhaust tube. The phenomenon is analogous to the act of rapidly drawing a cork out of a tube. The cork, as it is withdrawn, causes a rarefaction of gases in the tube and a tone is produced when an oscillation results in the readjustment of the partial vacuum.

The pulse in our case acts as the cork; the exhaust pipe with the valves closed act as the tube. The resonant sound has a wave length equal to four times the tube length. Other secondary resonances can occur at (2n - 1) times the fundamental resonance. Depending then on the pulse rate, engine rpm, and the tube length, these resonances can produce either very high pressures at the valve at the instant it opens or a vacuum, depending on the portion of the oscillation which is involved. Pictures taken of these pressures are shown in Fig. 24 on page 36.

A search of available literature on muffler design revealed very little information. However, two basic approaches to this problem of silencing were apparent:

1. Acoustic filters.
2. Mechanical pulse absorbers.

The most useful acoustic filters are of two types. One is the Helmholtz resonator filter which consists of a cavity connected to the exhaust pipe by a narrow slit. High sound absorption is realized at one frequency, the resonant frequency (see Appendix). The other is called an expansion chamber. This filter provides high sound absorption over a wide band of frequencies with a few narrow transmission bands. The amount of attenuation depends on the change in cross-sectional area and the pass frequencies are determined by the length of the chamber (see Appendix).

A mechanical pulse absorber depends for its operation on catching the high-velocity pulse of exhaust gas in a chamber of appropriate volume and bleeding the gas off slowly through a perforated wall. A by-pass channel is provided for the low-velocity gas flow. The design of the basic stage pulse absorber is described in detail in Section IV of this report.

II. GENERAL DESCRIPTION OF TEST

A. TEST-CELL SETUP

An AOS-895-3 engine, coupled to a 600-hp Midwest eddy-current dynamometer, was used for the tests under the same conditions as in the previous investigations (University of Michigan Engineering Research Institute Projects 2109 and 2599) conducted for the Detroit Arsenal on Exhaust Gas Ejectors.* Torque, rpm, temperatures and pressures were recorded as in the previous investigations.

The cooling-air flow of the AOS-895-3 engine was modified so as to simulate the conditions existing in the AOI-1195 engine in the T-95 tank. A sketch of the setup appears as Fig. 1.

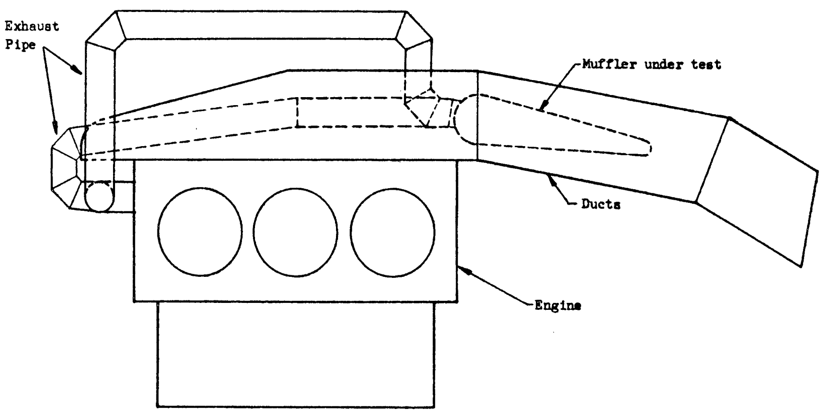

Fig. 1. Engine modifications.

The exhaust manifolds, which normally exhaust at the flywheel end of this engine, were rerouted to approximate the AOI-1195 engine manifolding. One exhaust pipe was located in the cooling-air duct. The other pipe was located outside the duct but entered the duct through the top. This top entrance avoided expensive modification of the engine shroud and associated equipment, yet presented the same restriction to cooling-air flow as the AOI-1195 engine setup.

A strain-gage-type pressure pickup was installed cn the exhaust manifold of one exhaust bank. The electrical output of this pickup was amplified and displayed on an oscilloscope. Pictures were taken of these displays for a permanent record. These data were used in the study of exhaust flow under varying conditions.

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*Contract Nos. DA-20-089-ORD-36259 (May, 1955), DA-20-018-ORD-14681 (Sept., 1957).

B. DESCRIPTION OF TEST

Normal sound levels were encountered in the vicinity of a high-powered engine such as is used in a military tank, range from 90 to 120 db. To obtain a well-integrated and representative signal from the engine, the microphone was placed 40 ft from the end of the engine cooling-air duct. Eight microphone positions on an arc of 40-ft radius (see Fig. 2), were used to establish the overall sound spectrum of the engine. This procedure is similar to that followed in the 2599 report.

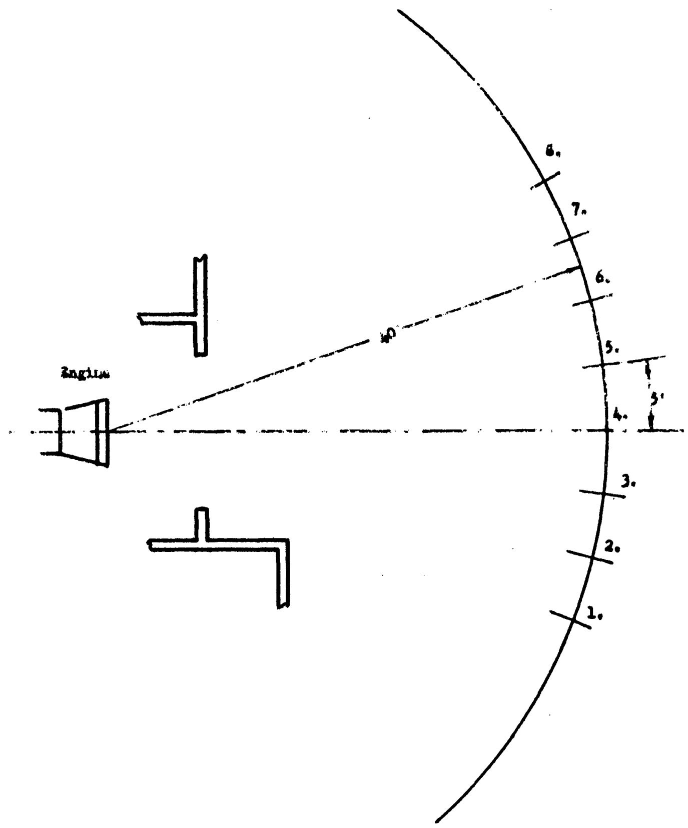


Fig. 2. Microphone locations.

The spectrum at each position was analyzed with a Bruel and Kjaer 1/3 Octave Filter and Level Recorder. The levels in each band were averaged for the eight positions to obtain the overall sound spectrum. Since the levels in any band did not vary more than 2 db from the average band level, it was felt that only one microphone position was sufficient for comparison of muffler configurations for this engine installation. Consequently, only microphone position 4 was used in all other sound spectrum measurements for this report. The procedure used to obtain these data is as follows. First, a reference level must be established by calibration. The General Radio Transistor Oscillator produces a 400-cps pure tone which is fed into the calibrator and is adjusted to give an 85-db level output at this point. This tone is fed into the microphone (see Fig. 3) and recorded on the tape. This establishes the relationship of the level of an input signal into the microphone to that recorded on the tape. The en-

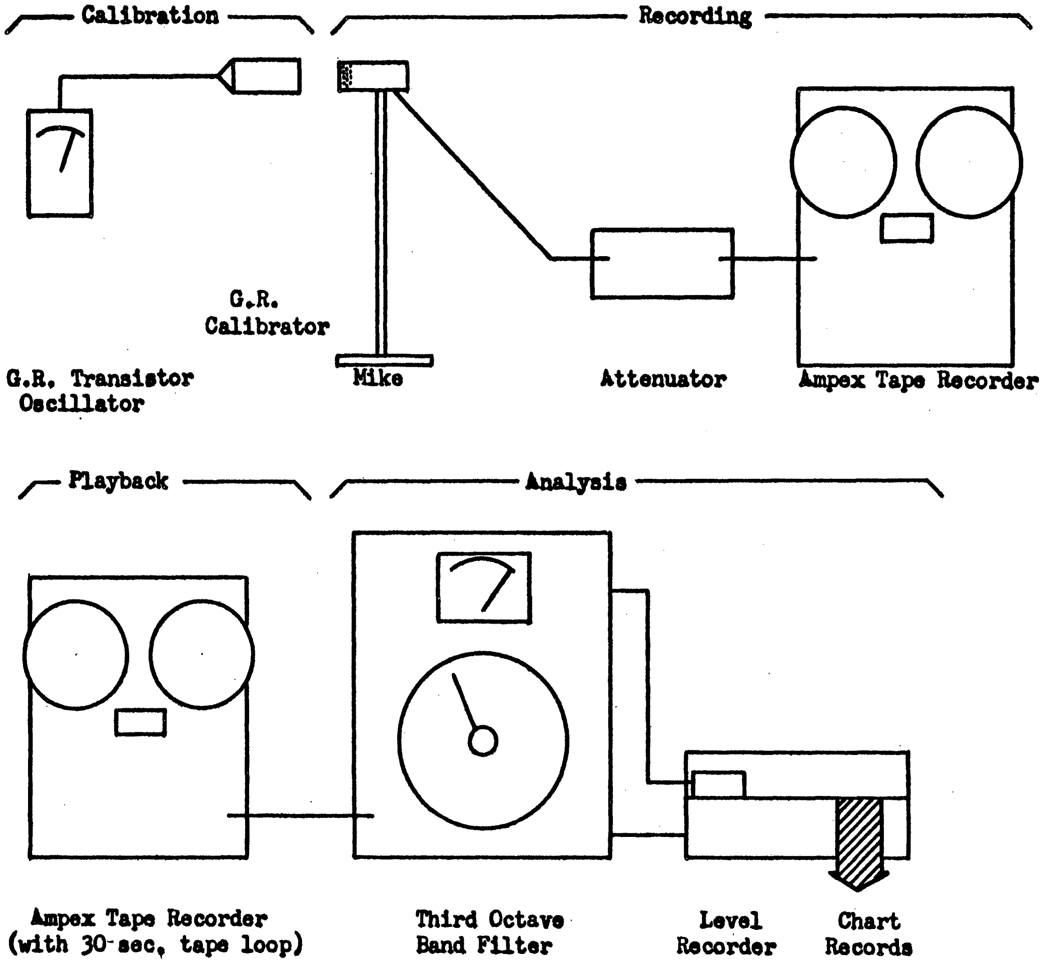


Fig. 3. Simplified block diagram of acoustic instrumentation.

gine is then started and brought up to a stabilized load condition of usually 2400-rpm 350-ft-lb engine torque. A recording is made from position 4 of Fig. 2 of about one-minute duration. The attenuator in the recording setup is used to reduce the high noise level of the test for recording purposes by acting as a volume control. It differs from a volume control in that the amount of sound-level decrease can be precisely set and thus does not disturb the calibration. The level recorded plus the attenuation is then directly comparable to the calibrated level previously recorded on the tape.

The second step then consists of playing the recorded sounds back for analysis. Using the 400-cps calibration tone recorded on the tape, this signal establishes a calibrated level on the graph records produced on the level recorder. A 30-sec tape loop is then made from the one-minute sound run so a continuous source of the sound can be had for analysis. The output of this tape loop is then fed into the Third Octave Band Filter which divides the spectrum into its frequency components. It is important to note that the filter samples portions of the total frequency spectrum in one-third octave bands from about 40 to 10,000 cps. The sound level of each band containing the frequencies of the total spectrum which fall within that band result in one plotted point on the sound spectrum curves (Fig. 7). The band is shifted in 1/3-octave increments by a mechanical drive system on the filter to cover the complete spectrum and each increment becomes a point on the curve.

C. LABORATORY SETUP

The acoustic properties of some available commercial mufflers were measured in the Acoustics Laboratory, using a white-noise source. Electrical noise from a neon lamp was amplified to drive a loudspeaker. The speaker was fitted with a cone to perform the transition from the 16-in. speaker diameter to the 3-1/2-in. muffler inlet diameter. A noise level in excess of 90 db was achieved at the inlet to the muffler under test. Sound coming from the muffler outlet was picked up by a microphone and fed to the 1/3-octave analysis system. The resultant sound spectrum showed the acoustic attenuation properties of the muffler under test.

III. TEST PROCEDURE AND RESULTS

A. AIR-FLOW DETERMINATION

In the initial designing of a muffler, the first requirement was a determination of the size and shape of a muffler which would create a restriction to cooling-air flow not to exceed one inch of water. It was desirable to predetermine the effect of muffler size in the design stage and then, after construction, actually to measure the restriction to flow by test.

To avoid a costly and time-consuming test setup involving flow nozzles and associated equipment, the effects of restriction to flow were developed from CAE Report No. 419.[1]

With the modified cooling duct installed on the engine, a Pitot tube traverse was made across the duct and the quantity of air flow determined. Manometer pickups were installed under the fan shroud, similarly to those in the CAE report. Then, with controlled air temperature and engine speed, it was possible to make direct manometer comparisons with the results shown in the CAE report, and to determine the fan static pressure differential with the cooling duct installed.

The cooling duct was then progressively blocked with steel strips. A curve was plotted as shown in Fig. 4 wherein Static Pressure Differential was plotted against Percent Restriction (% of duct area blocked off). This was used to determine the muffler cross-sectional area with a restriction similar to a number of straight-edged orifices.

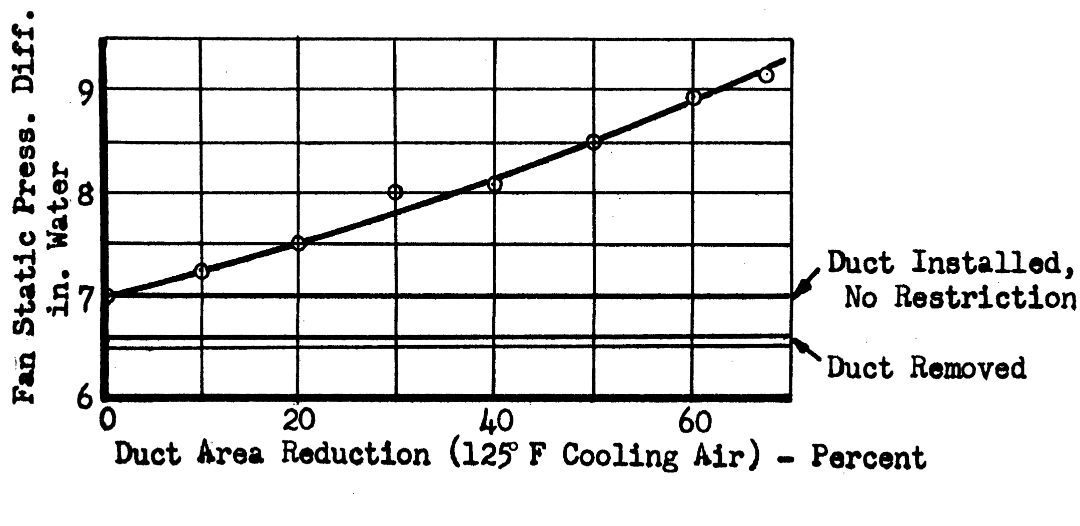


Fig. 4. Fan static differential pressure vs. duct restriction.

Mufflers, having considerable length as compared to the steel strips, indicated less flow resistance in all cases than would be expected from the curve. This is attributed to a Venturi effect caused by the muffler in the duct acting as a throat. If ideally designed, so that the throat area is not sufficiently small to cause a critical pressure, the pressure head built up by the muffler will be due only to the friction on the walls and to air turbulence. If the throat is constricted so that critical pressure is exceeded, the flow will then be greatly affected by the area of the throat.

Certain effects not yet mentioned are (1) ejector effect of exhaust gas, (2) increase in volume due to heat exchange between muffler and cooling air, (3) effect of nonuniform and turbulent flow.

The ejector action helps to offset the detrimental heating and turbulent effects, with varying results, depending on the muffler design (see Fig. 4). The flat muffler, without exhaust passing into it, improved air flow probably because it acted as a directing vane, decreasing turbulence.

To provide a more nearly perfect Venturi, it is important to isolate the muffler section from the remainder of the duct. Vertical metal sheets would accomplish this as shown in Fig. 5a. This will isolate the duct portions which have a difference in static pressure and velocity.

A second item which should increase fan performance is a radial directing fin above the fan. This would help distribute and turn the cooling air in the duct. The flow measurements show a major portion of the air to be at the top surface of the duct, due to the sharp bend at the fan. A muffler centered in the duct has most of the air passing over its top surface and very little beneath. A smoothly rounded vane, installed as shown, should decrease fan load (Fig. 5b). This is indicated by the increase in flow caused by the installation of the flat muffler (see Fig. 6).

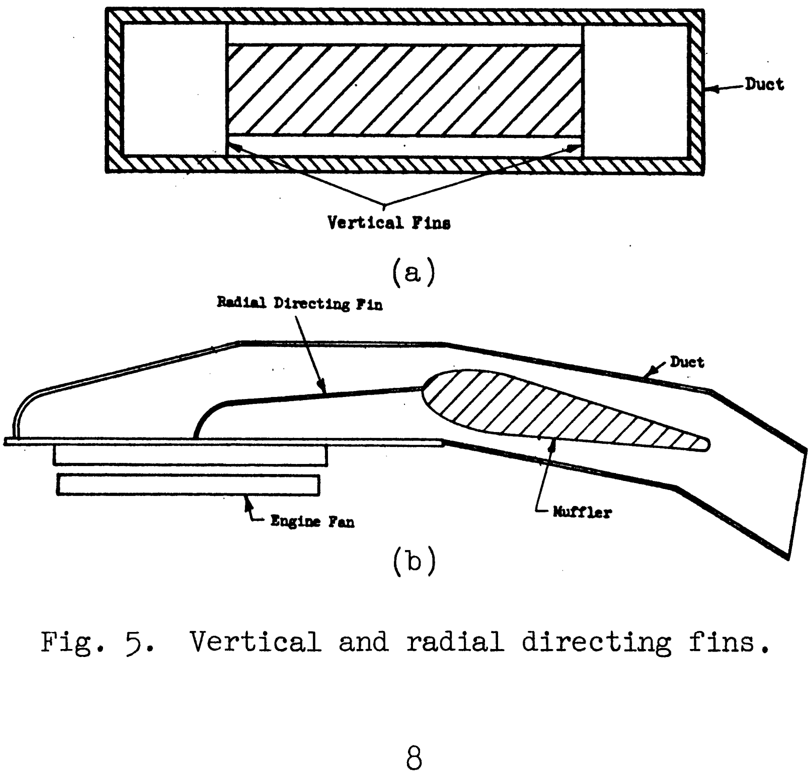


Fig. 5. Vertical and radial directing fins.

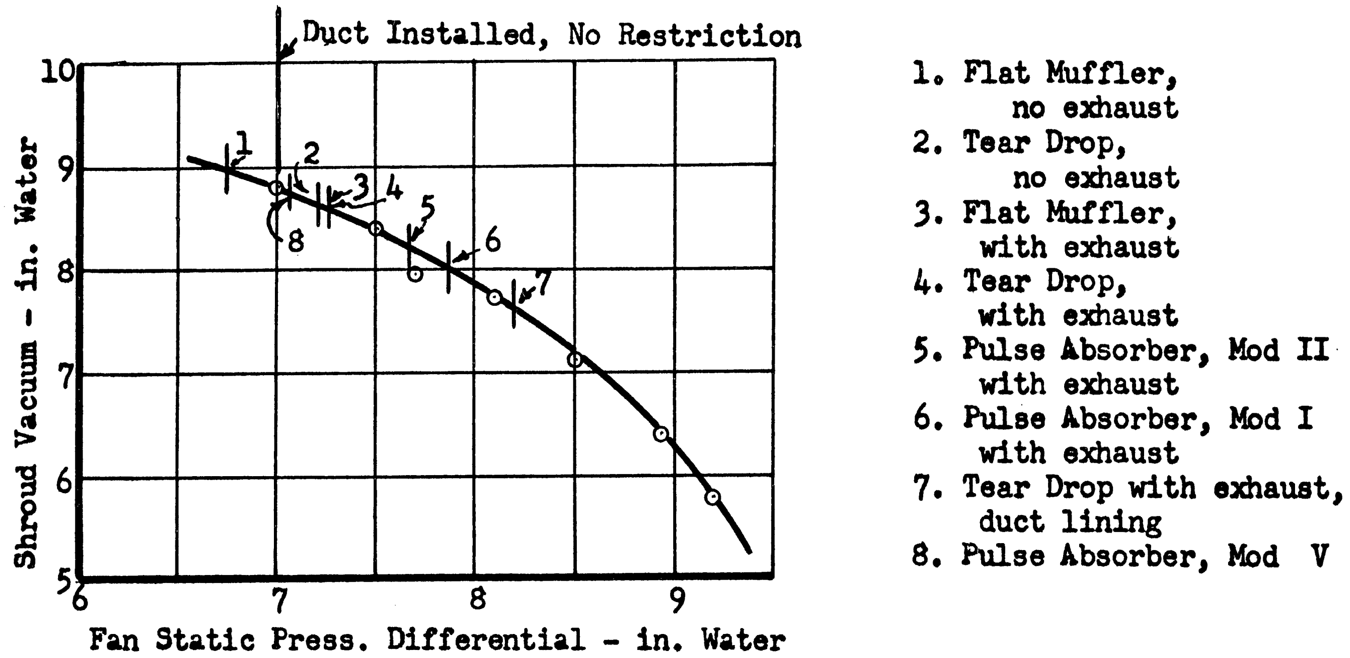


Fig. 6. Effects of mufflers on static pressure differential.

Acceptable air-flow results were obtained in the various tests without the use of smoothly shaped mufflers, vertical fins or directing vanes. Only acoustical material (see Section III-B-2) in the duct caused the pressure rise to exceed one inch of water. It seems certain, then, that the use of the improvements suggested will result in a pressure rise well below the one-inch limit.

B. SOUND DETERMINATION

Several muffler types were suggested by the contractor. After the air-flow restriction was evaluated, it was decided to group the muffler types according to possible methods of silencing rather than according to external shape. It was felt that the shape offered considerable flexibility once good silencing control was effected.

The following muffler types were listed in the Contract:

1. Straight pipes with duct sound absorption.
2. Mixing tubes with duct sound absorption.
3. Flat muffler.
4. Tear-drop muffler.
5. Tube-type muffler.
6. Any other type or types which results may dictate.

Sketches of the first five types are shown in Fig. 7. Testing of Item 1 was conducted separately and the results are listed below (see Sections III-B-1 and III-B-2).

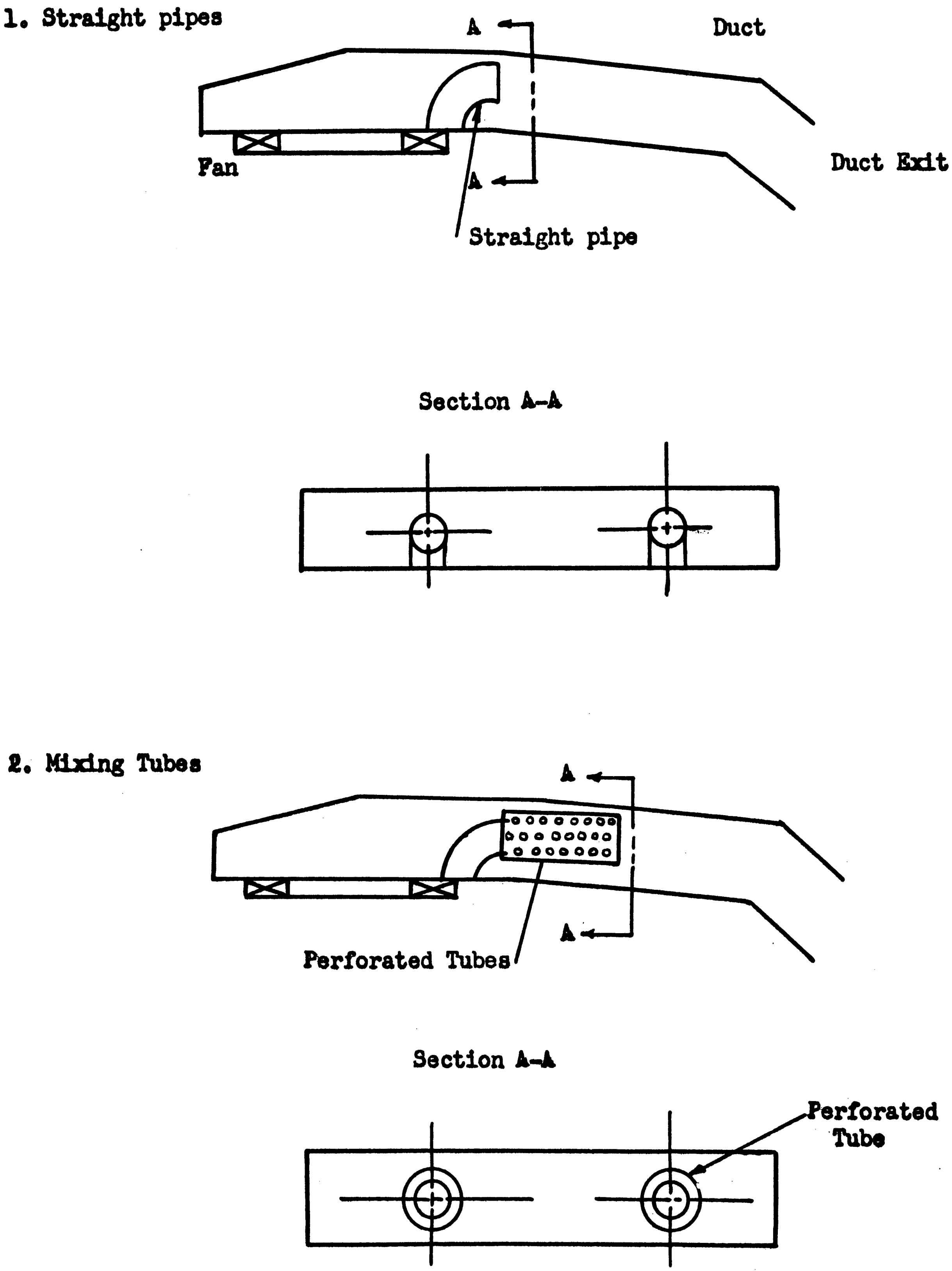
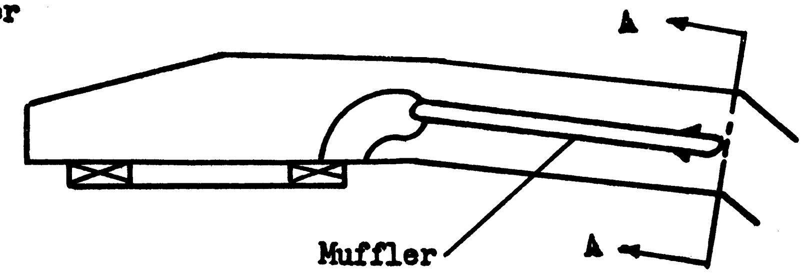
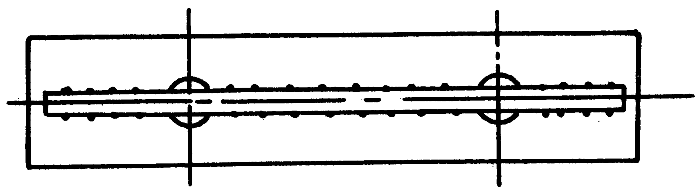


Fig. 7. Sketches of proposed mufflers.

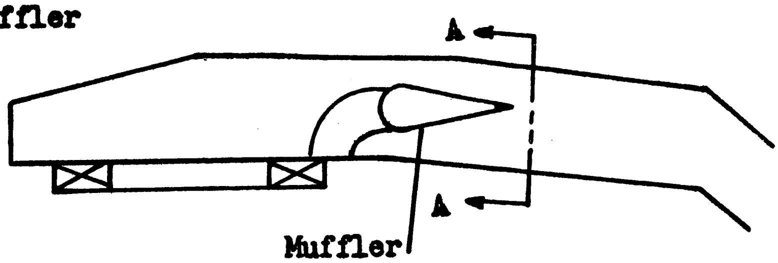
3. Flat Muffler



Section A-A



4. Tear-drop Muffler



Section A-A

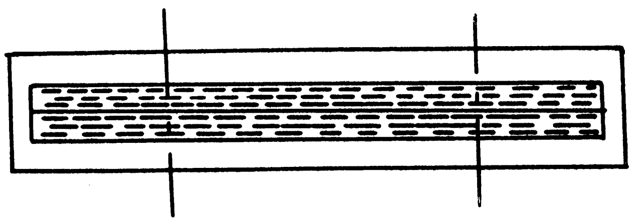
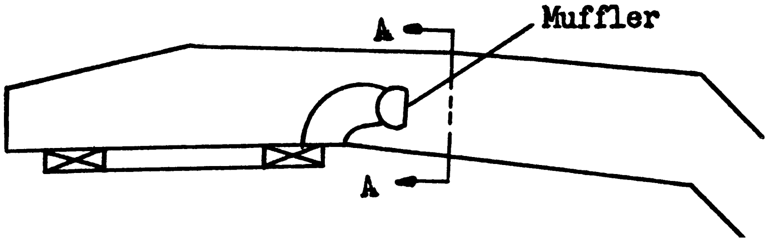


Fig. 7. Continued.

5. Tube Muffler



Section A-A

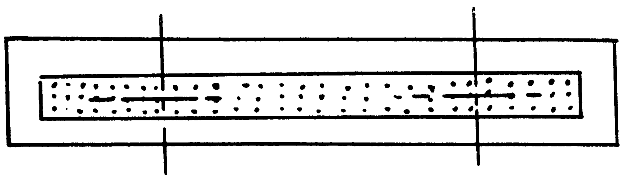


Fig. 7. Concluded.

Item 2, mixing tubes, was felt to have merit by allowing more engine accessibility. This problem was attacked in three ways: 1) acoustically, with duct treatment as in Section III-B-2; 2) with additional muffler absorption material as in Section III-B-3; and 3) as a pulse absorber, using methods described in Section III-B-7.

Item 3, the flat muffler, is discussed in Section III-B-4 below.

Item 4 was combined with Item 5, the two varying only in the external shape. The internal construction of this type was based on the expansion-chamber principle. Results for this type are explained in Section III-B-6 below.

Item 6, as a result of our tests, became a combination of the "tear-drop" type suggested, as far as outside shape was concerned, and the mechanical pulse absorber construction inside (see Section III-B-8).

1. Sound Levels

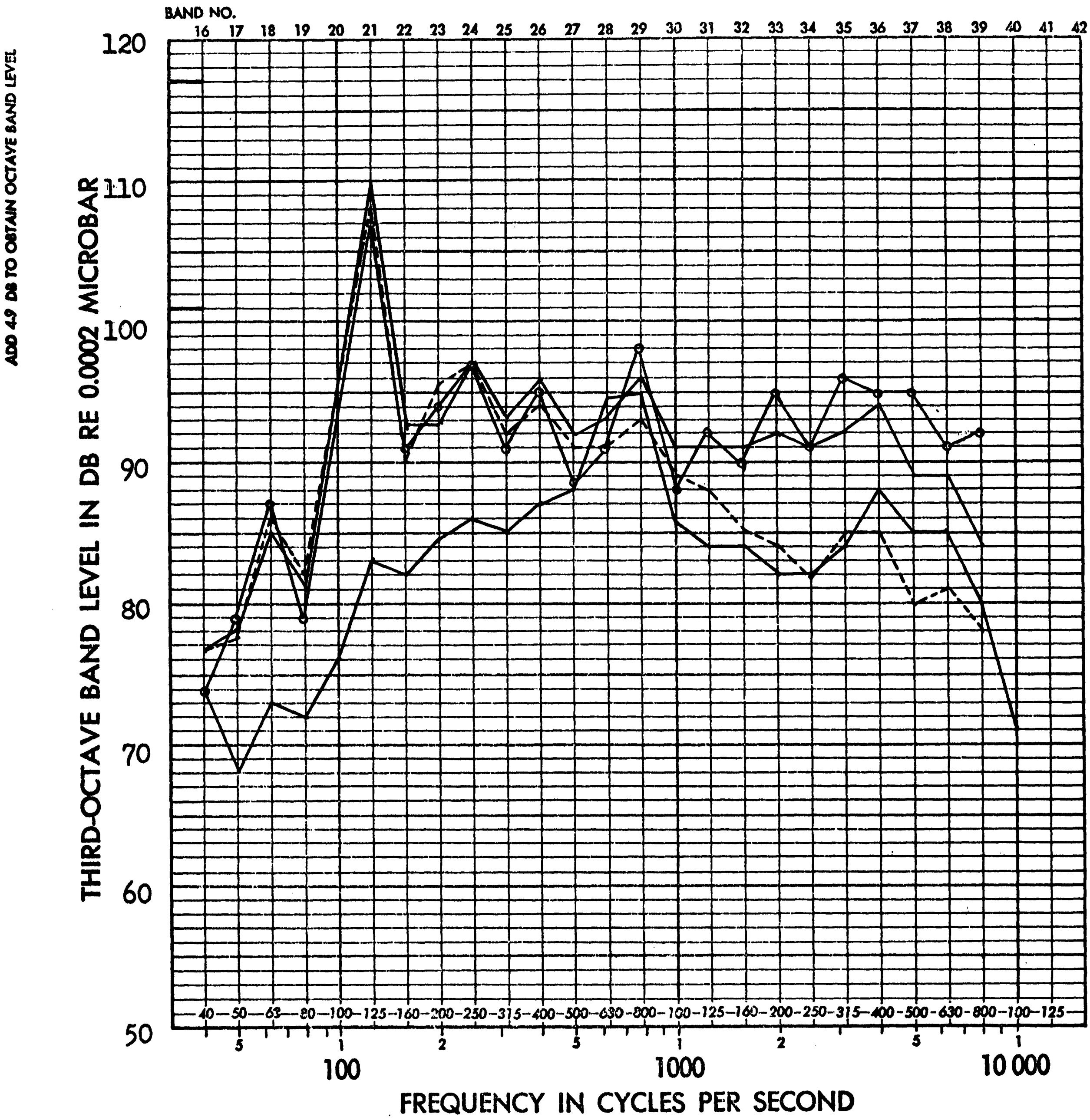
a. Sound Limits.—The task of silencing the T-95 tank engine was first attacked by attempting to establish sound-level "limits" between which to work. The levels determined were not regarded as absolute limits but were used merely as an index of what could or could not be done with a muffler under our test conditions.

It was felt at first that the worst possible condition would be that involving no muffler at all. The exhaust pipes were allowed to fire straight into the duct at the position where a muffler was to be attached. Although it was later discovered that certain muffler conditions could be found which exceeded this "high limit" sound level, it was felt that the ultimate muffler design should certainly perform better than no muffler at all. In an attempt to achieve the quietest operation possible with the test-cell arrangement, the engine was fitted with two Maxim silencers as in the usual tank situation. The output pipes of the mufflers were connected to a 10-in. exhaust duct which emptied out of the side of the building. This effectively removed the exhaust sounds from the overall engine spectrum, resulting in a "low limit." These "high" and "low" sound-level curves are shown in Fig. 8. For purposes of comparison these two curves are shown as solid lines on all graphs which follow.

b. Interpretation of Results.—These limit curves illustrate a very important point about the methods which must be used in this investigation and in the interpretation of the subsequent results which are included in this report. The sound spectrum of this engine is not dominated by the exhaust noise but by the combined effects of the exhaust noise, fan noise (which includes movement of air plus blade whine), and engine noise. A complete elimination of the exhaust noise will still leave a high noise level, and as shown here in the high and low limits a reduction of about 16 db is possible. The average overall level remains at 101 db. The sound reductions accomplished in the various designs must be investigated by examining the sound spectrum and not the overall sound level. For this reason overall sound-level measurements were not satisfactory for the investigation and the 1/3-octave-band analysis was used.

Since the final decision as to the effectiveness of a muffler will depend on a person or persons actually listening to the results and arriving at a conclusion, it would be well to mention a phenomenon known as masking. Despite any elaborate sound analysis, the human ear will be the ultimate judge, and in this regard the masking effects are very important. Masking, as stated in the terminology, is the effect one sound has on raising the threshold of audibility of a second sound. In this case we have the fan noise acting as a mask on the exhaust noise. In some of the muffler tests, a decrease of the 120-cps exhaust tone to a level of 103 db gave a very satisfactory reduction of noise to the listener. It is unnecessary to have any further reduction in sound level if the level is below the threshold of audibility. This is especially true in the case of limited muffler volume where a reduction of sound level is obtained mainly through higher muffler restriction.

Also, along this line, a muffler which may sound acceptable under one set of conditions may or may not be acceptable if the masking level is reduced, i.e., installation with a quieter fan or in a tank which has higher absorption for the fan noise.



-o- Damping on duct (outside)

--- Effect of internal duct treatment Fiberglas lining

—— "high" and "low" levels (see Section III-B-1)

Corresponding average overall levels are shown on the extreme left.

Fig. 8. Duct treatment spectra.

2. Effect of Duct Treatment

The effect of acoustic treatment of the air duct on the exhaust noise was actually tested with the tear-drop muffler (Section III-B-5) installed. The basic duct was constructed of 16-gage sheet steel, reinforced with 1 x 1 x 1/8 angle iron. A 1/2-in.-thick blanket of Fiberglas with a density of 2 lb/cu ft was laid over the outside of the duct and held in place by 26-gage galvanized sheet steel. Fiberglas was chosen as the damping material because of its resistance to high temperatures. The layer of steel on the outside insured surface contact between the Fiberglas and the duct. This arrangement provides maximum damping effect of the surfaces.[2] However, the effects of acoustical and mechanical damping treatment on the duct was negligible, as can be seen in Fig. 8. This indicates that the mock-up of the duct used here was inherently damped sufficiently for the application and additional damping had little effect.

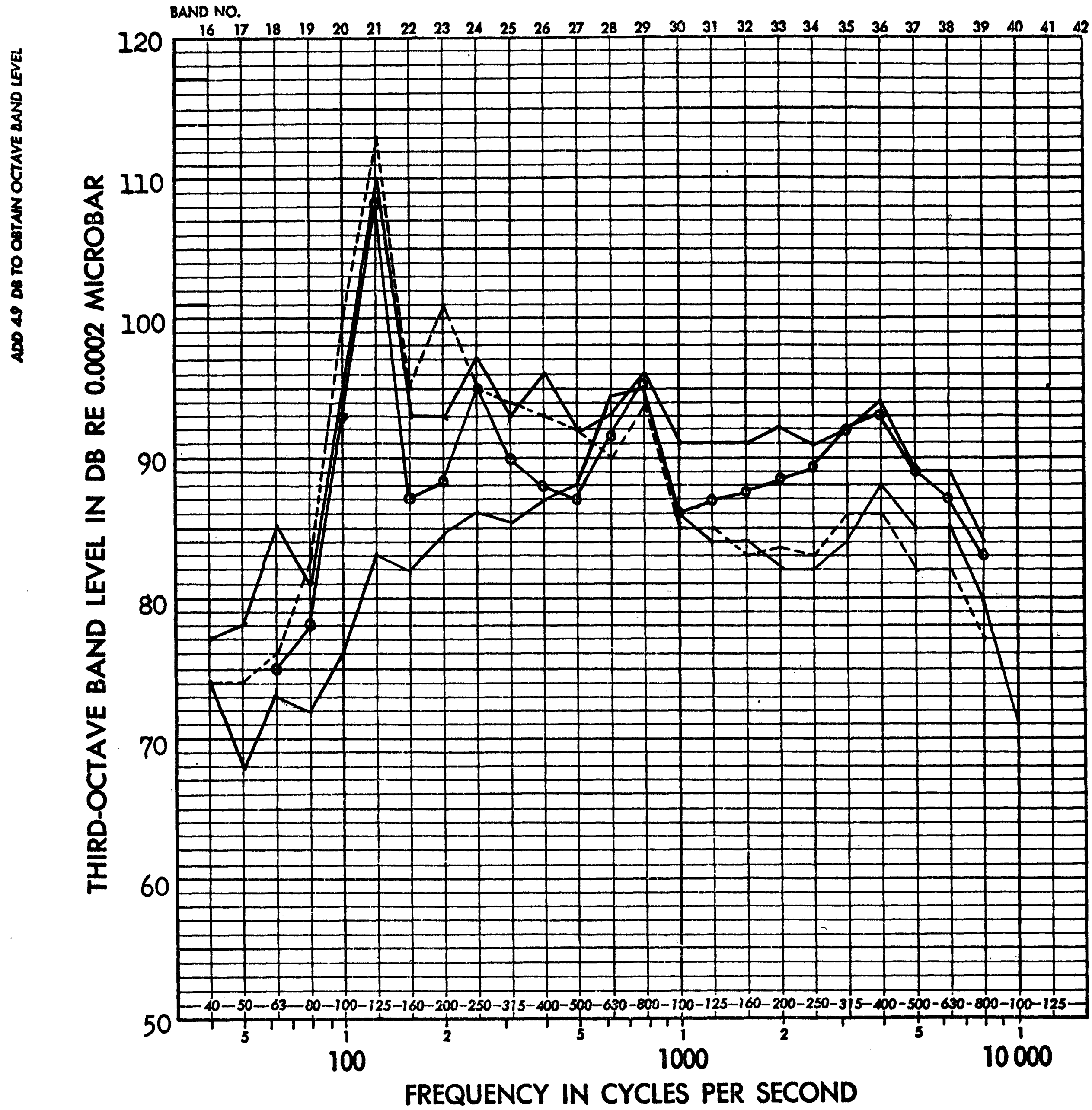
Acoustic absorbing material placed on the inside of the duct lowered the sound level in the high-frequency bands (Fig. 8). However, the major components of exhaust noise are in the frequency bands below 400 cps. Wave lengths and energies of these components make them very difficult to treat by this method of sound absorption. In addition to offering very poor quieting at low frequencies, the Fiberglas lining, because of its rough surface, increased the static pressure differential on the engine cooling fan by 1 in. H_2O.

3. Absorptive Muffler

As a further test of the acoustic absorptive treatment as applied to the problem of quieting the engine exhaust, two "Hollywood" type straight-through mufflers were installed. Each of these consisted of a perforated steel tube surrounded by a 1-1/2-in.-thick covering of Owens-Corning TW-F Fiberglas. This assembly was covered with 26-gage galvanized steel for mechanical protection.

In operation these mufflers gave very poor performance. After the test, the mufflers were opened and the Fiberglas was found fused together in many places. Although the decrease in performance due to deterioration of the muffler as it was run was not as great as expected, the overall performance was poor compared to no muffler at all. Figure 9 shows the spectrum of the Fiberglas mufflers with the "high" and "low" curves for comparison. The drop in level above 1000 cps is attributed to the Fiberglas duct lining which was in place for the test.

The manifold back pressure during this test was 2.25 in. Hg at 350-ft-lb engine torque. This pressure reading seemed rather high considering the fact that the mufflers were the straight-through type. An investigation of pressures along the manifold showed that the reading obtained in this case was entirely due to bends in the pipe between the manifold and the mufflers. Further tests showed that the back pressure, due to the pipes leading to the mufflers, was



-o- Flat muffler

--- Absorptive muffler (Fiberglas)

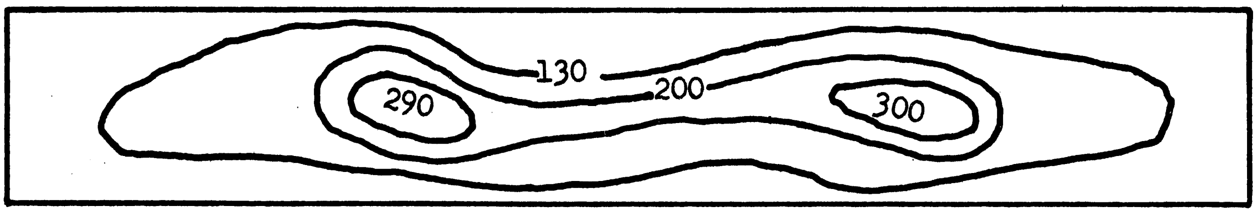
Corresponding average overall levels are shown on the extreme left.

Fig. 9. Absorptive and flat muffler spectra.

approximately 2.70 in. Hg at 2400-rpm full load. For this reason, it was felt that the back pressure read at the inlet to the muffler was a more valid figure for comparison of muffler types. Later measurements were taken at the muffler inlet.

4. Flat Muffler

The flat muffler supplied by the contractor was found to have the rather high back pressure of 6.0 in. Hg, measured at the manifold. It was found later that the particular muffler supplied to us had not been modified to reduce this back pressure as had the one currently in use in their field tests. As is apparent from Fig. 9, the sound spectrum is little different from that of no muffler. However, the mechanical layout of the exhaust ports provides excellent mixing of exhaust gases with cooling air (see Fig. 10).



Air temperature (°F) at duct outlet, 2400 rpm, 350 ft-lb.
Flat muffler installed.

Fig. 10. Temperature distribution with flat muffler.

This flat muffler uses much the same principle of sound absorption as the "Hollywood" mufflers but does not have sufficient area or depth of absorbing material.

5. Tear-Drop Muffler

A modification of the tear-drop muffler requested in D4 of the proposal was built and tested. From air-flow measurements, it was determined that a muffler installed in the duct could block off 37% of the area without loading the cooling fan excessively. Using the expansion-chamber principle, it was desired to achieve the maximum change in cross-sectional area; consequently, the design shown in Fig. 11 resulted. The exit tubes were designed to have the same total cross-sectional area as the inlet pipes and were set into the muffler to provide a reverse flow path.

Figure 12 shows the sound spectrum obtained with the tear-drop muffler. The combined manifolds firing-rate peak (125 cps) was greatly enhanced although the remainder of the spectrum was reduced from that of no muffler. Overall sound level was the same as that obtained with no muffler. These results indicated

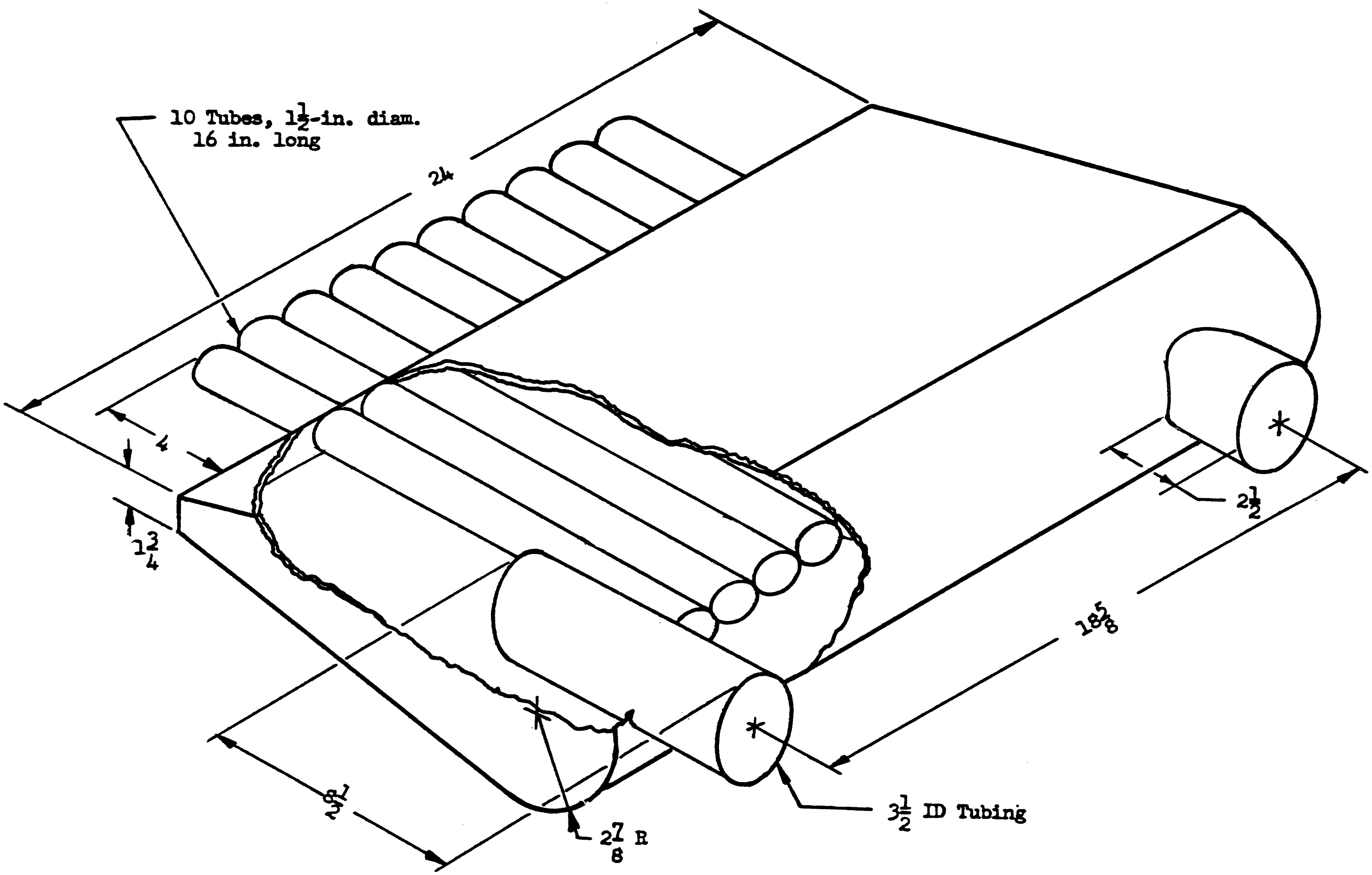
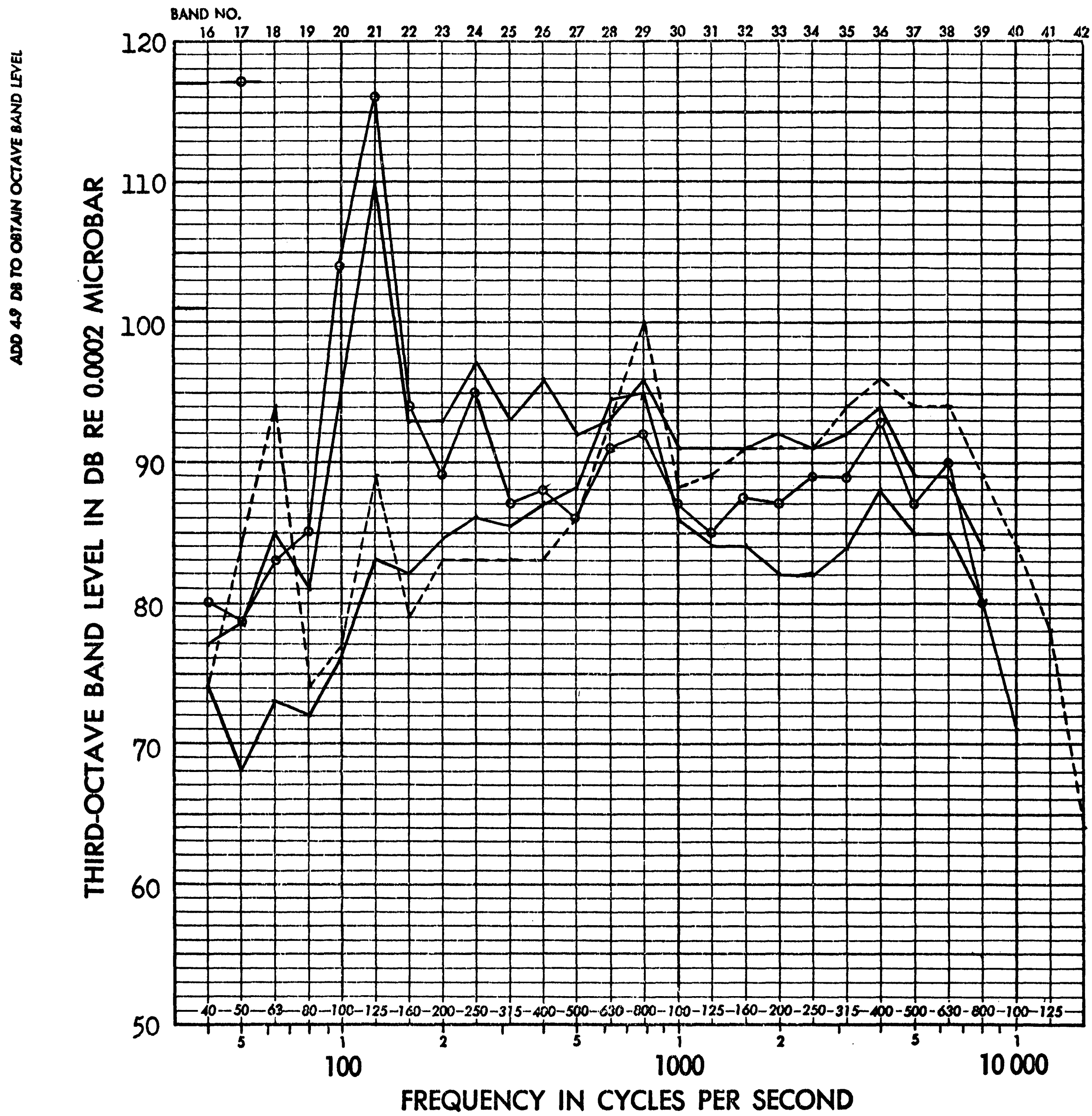


Fig. 11. Tear-drop muffler construction.



--- Maxim silencers (2)
-o- Tear drop

Corresponding average overall levels are shown on the extreme left.

Fig. 12. Tear-drop and Maxim muffler spectra.

that this tear-drop muffler was very little, if any, improvement over no muffler at all except for slightly better exhaust gas distribution in the duct. The tear-drop shape gave very little air-flow resistance and the volume could have been increased without excessive air restrictions. However, it was felt that the available volume could be better utilized with another design.

Mixing-tube-type mufflers have essentially the same construction as the tear drop tested and will give similar results. It should be pointed out that the tear drop or Venturi shape offers the lowest resistance to air flow for a given muffler volume. This consideration led to the modified shape of the pulse absorber described later in Section III-B-8.

6. Maxim Silencer

The sound spectrum with two Maxim silencers in place outside the duct is shown in Fig. 12 for comparison. In this case, the muffler outlet pipes were pointed toward the door of the test cell in the normal manner. The firing rate peak now occurs in the 63-cycle band. The reduction in amplitude is largely due to the increased muffler volume and position in the test cell (see Section III-B-10). A calculation showed that the volume available for the muffler inside the duct was equivalent to only that of one Maxim silencer and could not be used as efficiently.

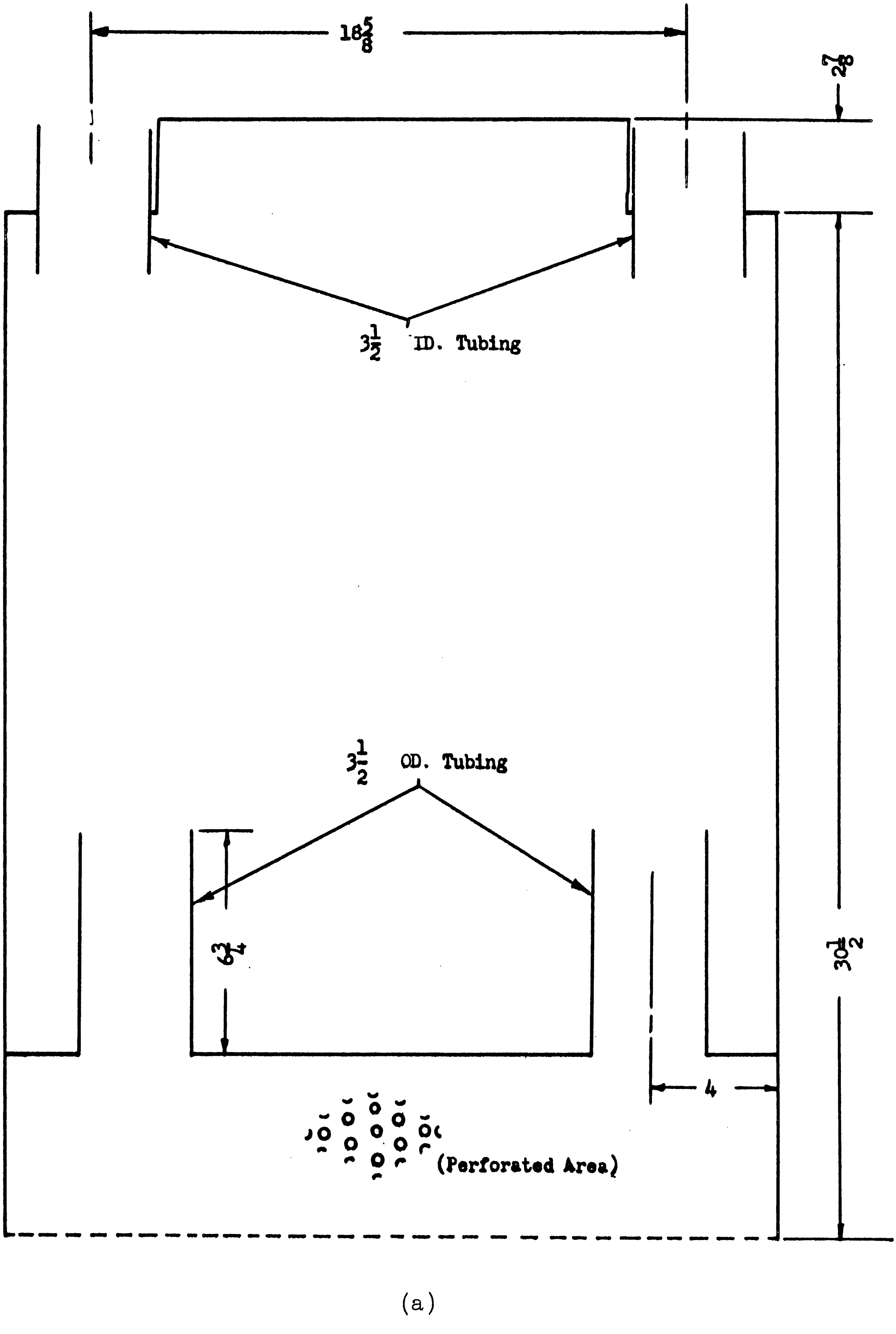
In both tests involving Maxim silencers, an interesting observation was made regarding the high-frequency end of the spectrum. Whenever the Maxim silencers were installed, although the overall sound level was low, the levels in the high-frequency bands were high. With a muffler installed in the air duct, a general lowering of the high-frequency levels resulted. This effect leads one to believe that, if the firing peak can be sufficiently reduced by a muffler installed in the air duct, a quieter engine-exhaust system can be realized than with the conventional muffler installation.

7. Acoustic Expansion Chambers

Using 37% of the cross-sectional area and as much length as possible, the two approaches to muffler design mentioned in Section I were tested. Two identical metal boxes were built so that each muffler would have the same total volume. The leading edge of the box was rounded to smooth the flow of cooling air while the trailing edge was tapered and perforated over its entire surface for exhaust-gas mixing. Two tubes projecting into the box through a rear wall were installed so as to form a single expansion chamber (see Fig. 13a). The resulting sound spectrum (Fig. 14), at the standard engine load, was only very slightly lower than the high limit at the 120-cps firing rate. Also, as the engine was run through its speed range, resonances could be heard.

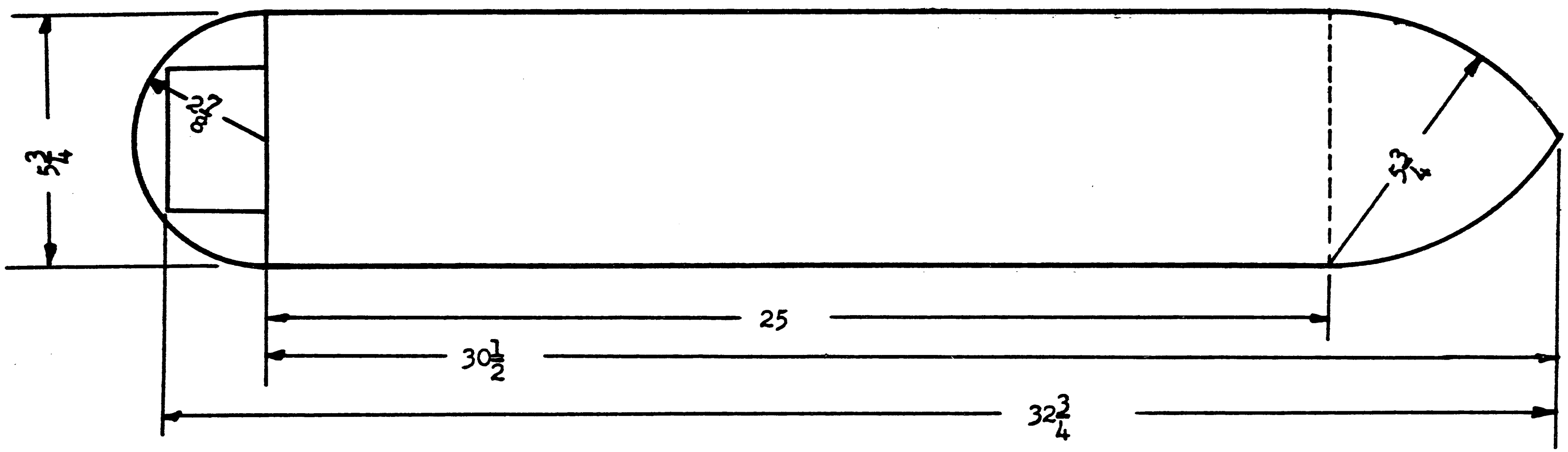
In an attempt to reduce the number of resonant peaks and increase the attenuation, a second wall was placed in the chamber through the center to form

a. Single acoustic expansion chamber (top view).



(a)

Fig. 13. Acoustic-expansion-chamber designs.

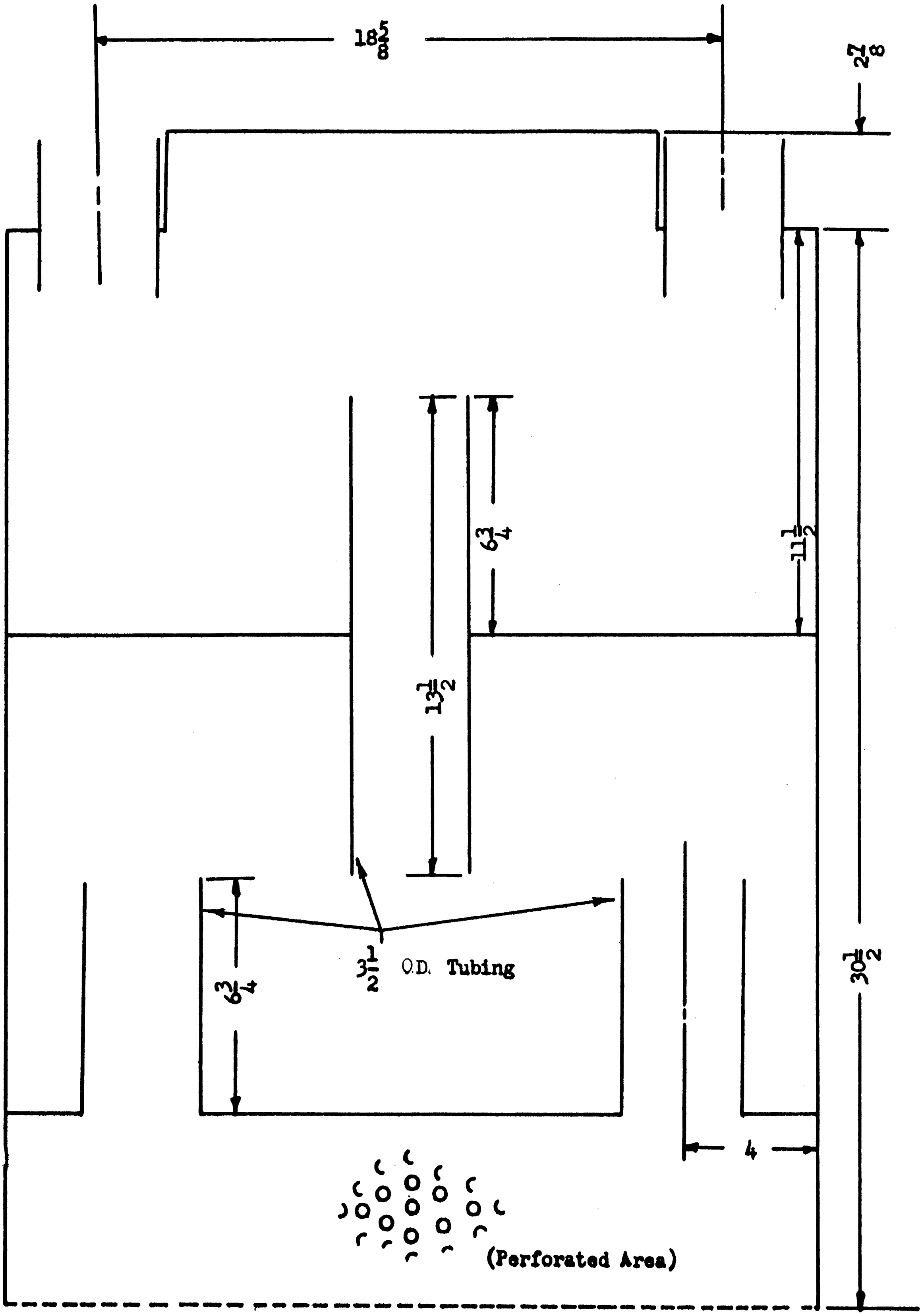


b. Expansion-chamber and pulse-absorber mufflers (side view).

(b)

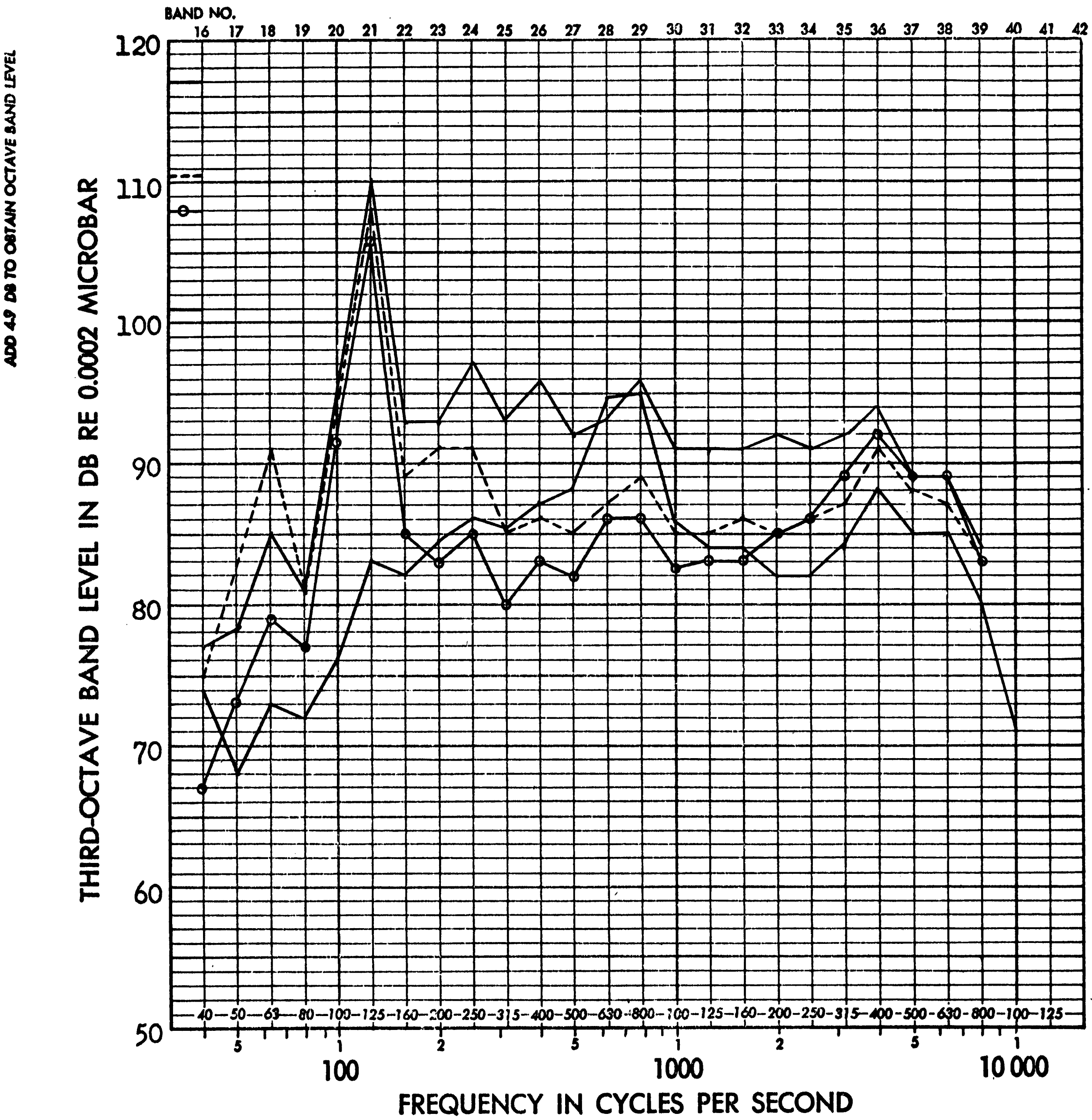
Fig. 13. Continued.

c. Double acoustic expansion chamber (top view).



(c)

Fig. 13. Concluded.



--- Single expansion
-o- Double expansion chamber

Corresponding average overall levels are shown on the extreme left.

Fig. 14. Acoustic-expansion-chamber spectra.

a double expansion chamber (see Fig. 13c). A slight improvement was observed in the sound spectrum (Fig. 14), but objectionable resonances were still present. The full-load muffler back pressure was 2.50 in. Hg at 2400-rpm, 800-ft-lb engine torque.

8. Mechanical Pulse Absorber

The second metal box was fitted with baffles to form a two-stage pulse absorber. This type of construction is discussed in Section IV of this report. The construction details and the gas flow are shown in Fig. 15a; the model with the top cover removed is shown in Fig. 15b.

The resultant sound spectrum is shown with the single expansion chamber in Fig. 16. The firing rate was reduced 3 db and the noise level did seem lower to the listener. Also, the resonances heard in the other muffler types were no longer noticeable.

Back pressure at full load, 2400 rpm, was measured at 2.25 in. Hg.

Since this type of muffler provided a noticeable change in sound level, removed resonances, and provided for flexibility in design, it was decided to concentrate on improving the pulse absorber for this application.

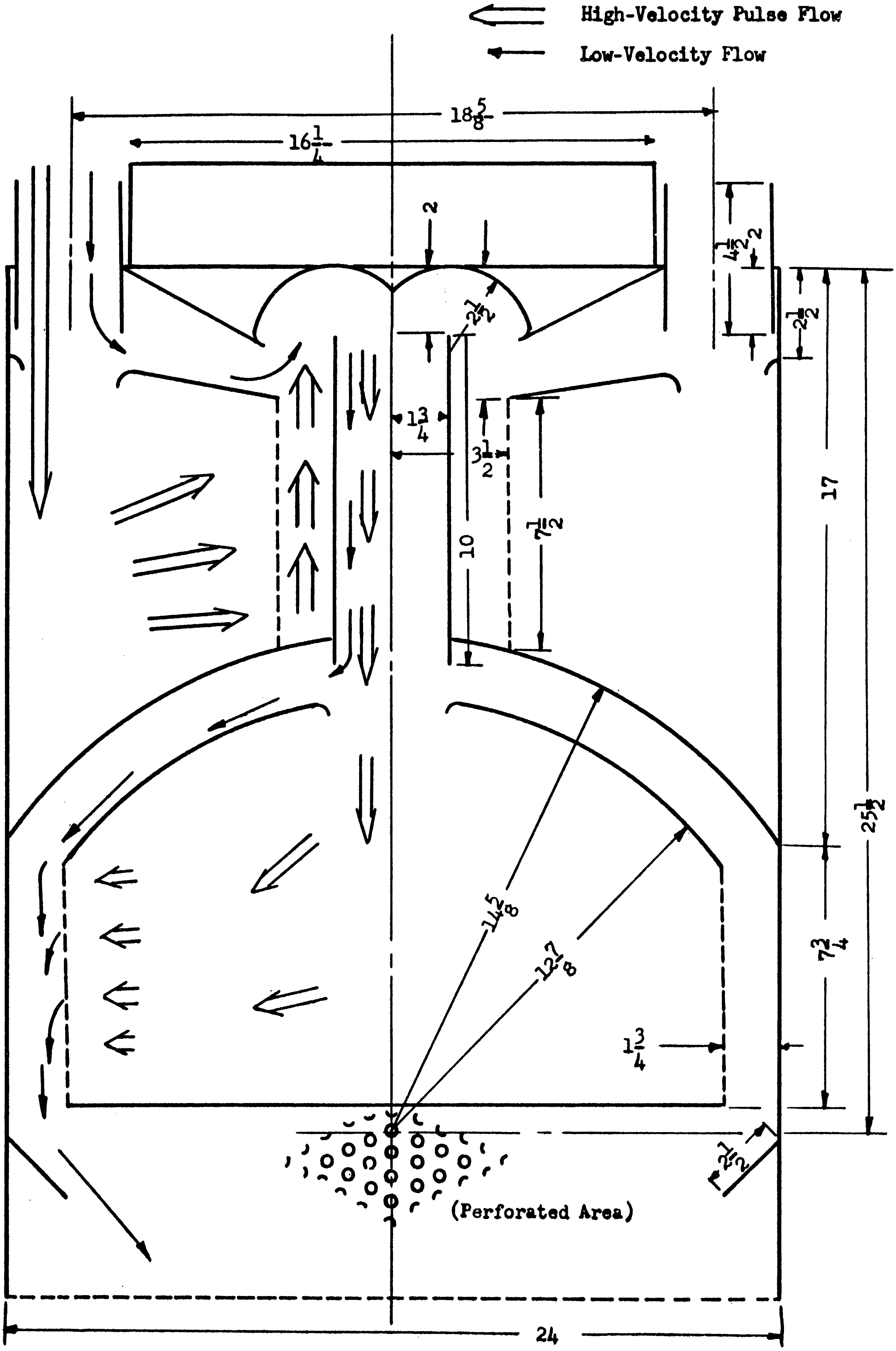
9. Improvements in the Double Pulse Absorber

The first modification of the pulse absorber was to increase the volume of the first chamber and remove the reverse flow path (Fig. 17). This resulted in a 2-db drop in the firing-rate peak (Fig. 18).

A temperature traverse of the mouth of the air duct showed a region of high temperatures near the center (Fig. 19). To improve the mixing, the perforated end of the muffler was sealed up and side wing outlets were installed on the muffler (Modification II, Fig. 20). This allowed the second absorbing chamber to be increased in size. The resulting exhaust mixing was considerably better. A new temperature traverse revealed no temperature higher than 275°F under 350-ft-lb load at the mouth of the cooling duct (Fig. 21). Full load (800 ft-lb, 2400 rpm) resulted in temperatures of about 300°F.

Modification III separated the muffler into two mufflers, with a plate welded vertically through the center. This made the second chamber one-half the original volume, and decreased the low-velocity flow path in each section. The pulse rate in this case becomes one-half the pulse rate of the engine, as each section serves only one bank of cylinders. The sound spectrum is shown in Fig. 22. The 120-cps peak has been reduced 8 db from that of no muffler. This reduction was sufficient to cause the muffler level to be masked completely by the fan noises, and as a result the engine tones could no longer be heard.

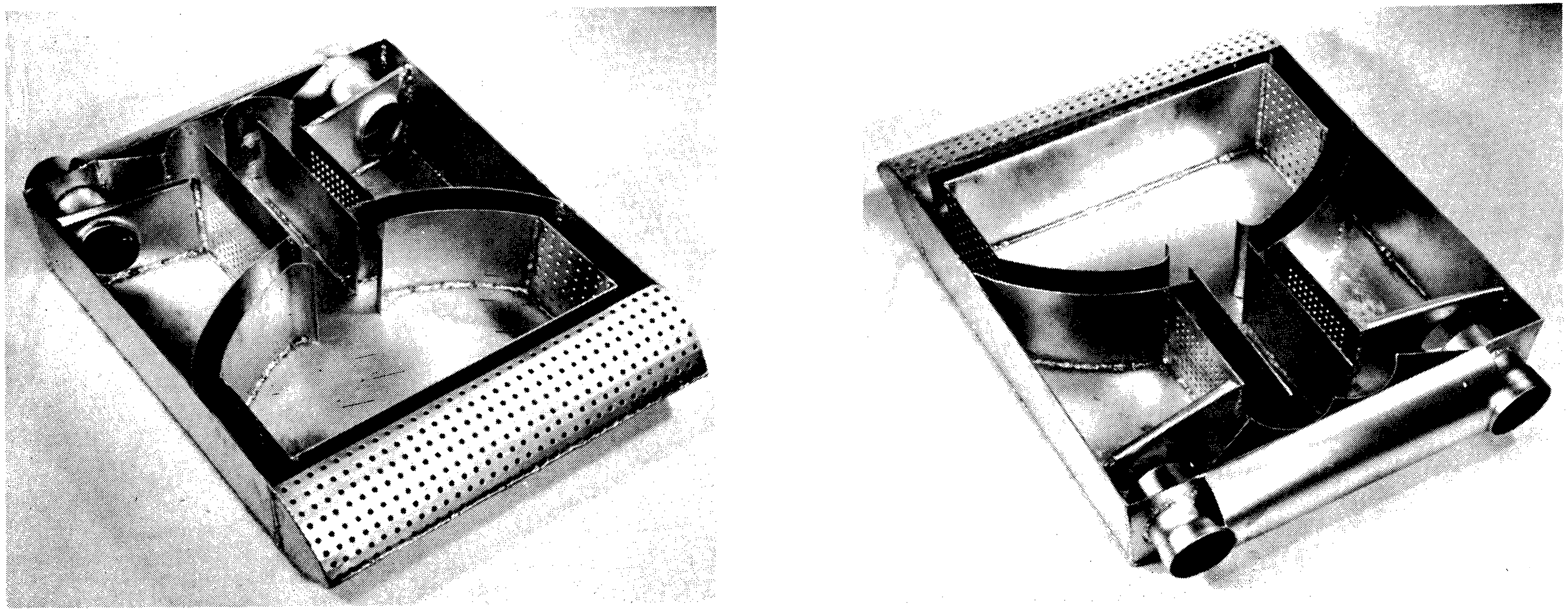
a. Mechanical pulse absorber with gas flow (top view).



(a)

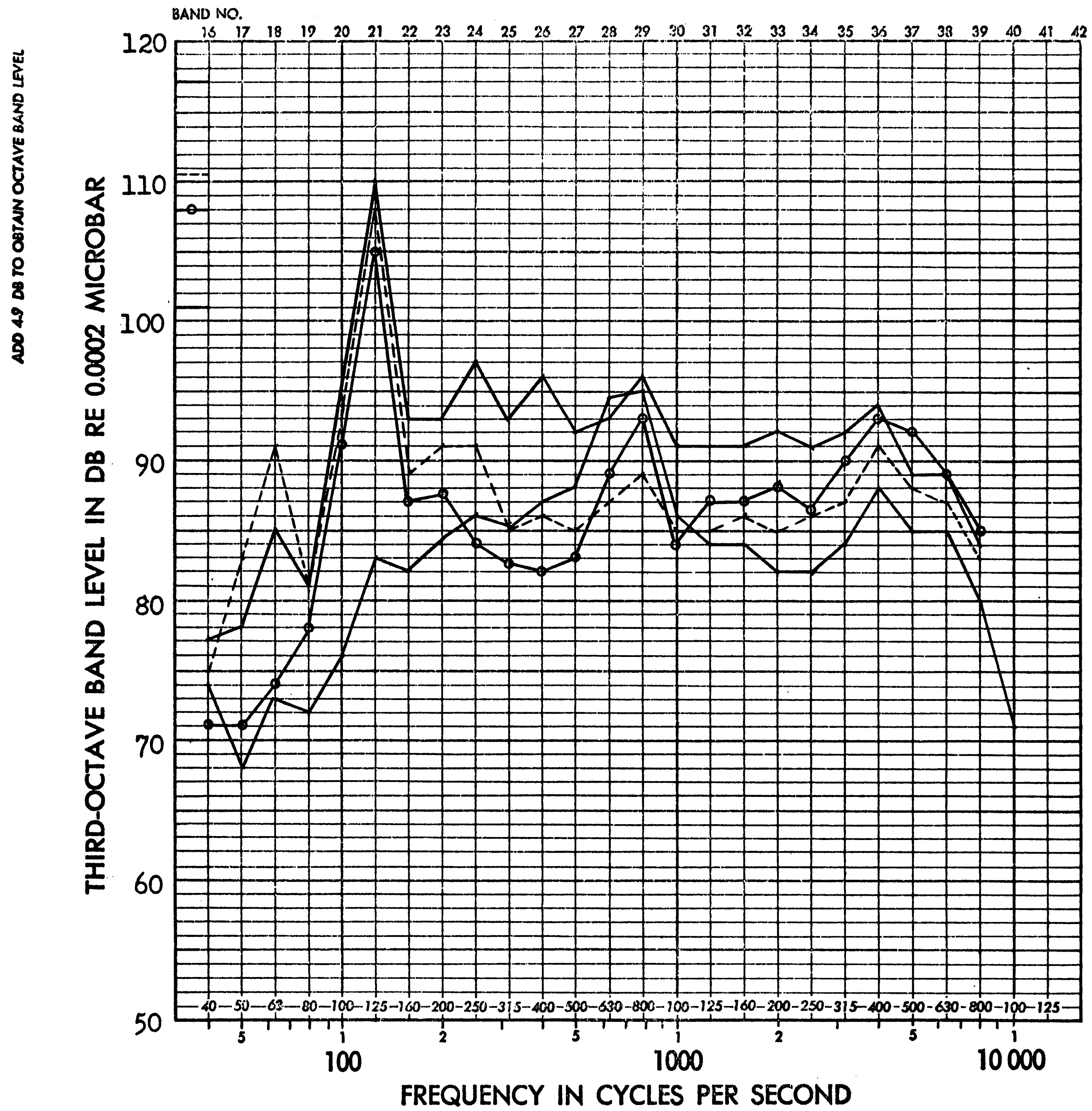
Fig. 15. Mechanical-pulse-absorber design with exhaust flow.

b. Pulse absorber, top removed.



(b)

Fig. 15. Concluded.



-o- Double pulse absorber

--- Single expansion chamber

Corresponding average overall levels are shown on the extreme left.

Fig. 16. Pulse-absorber and expansion-chamber spectra.

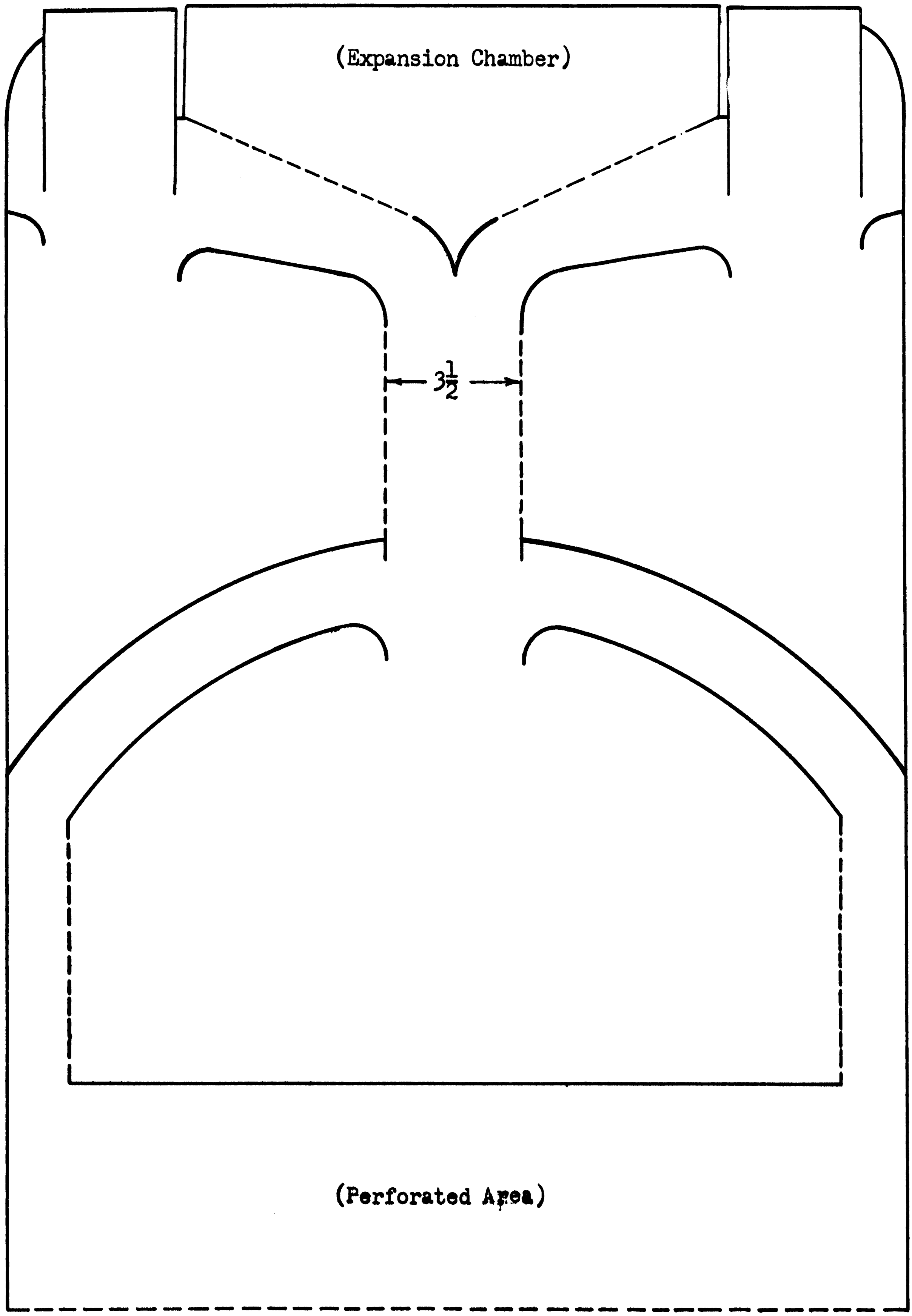
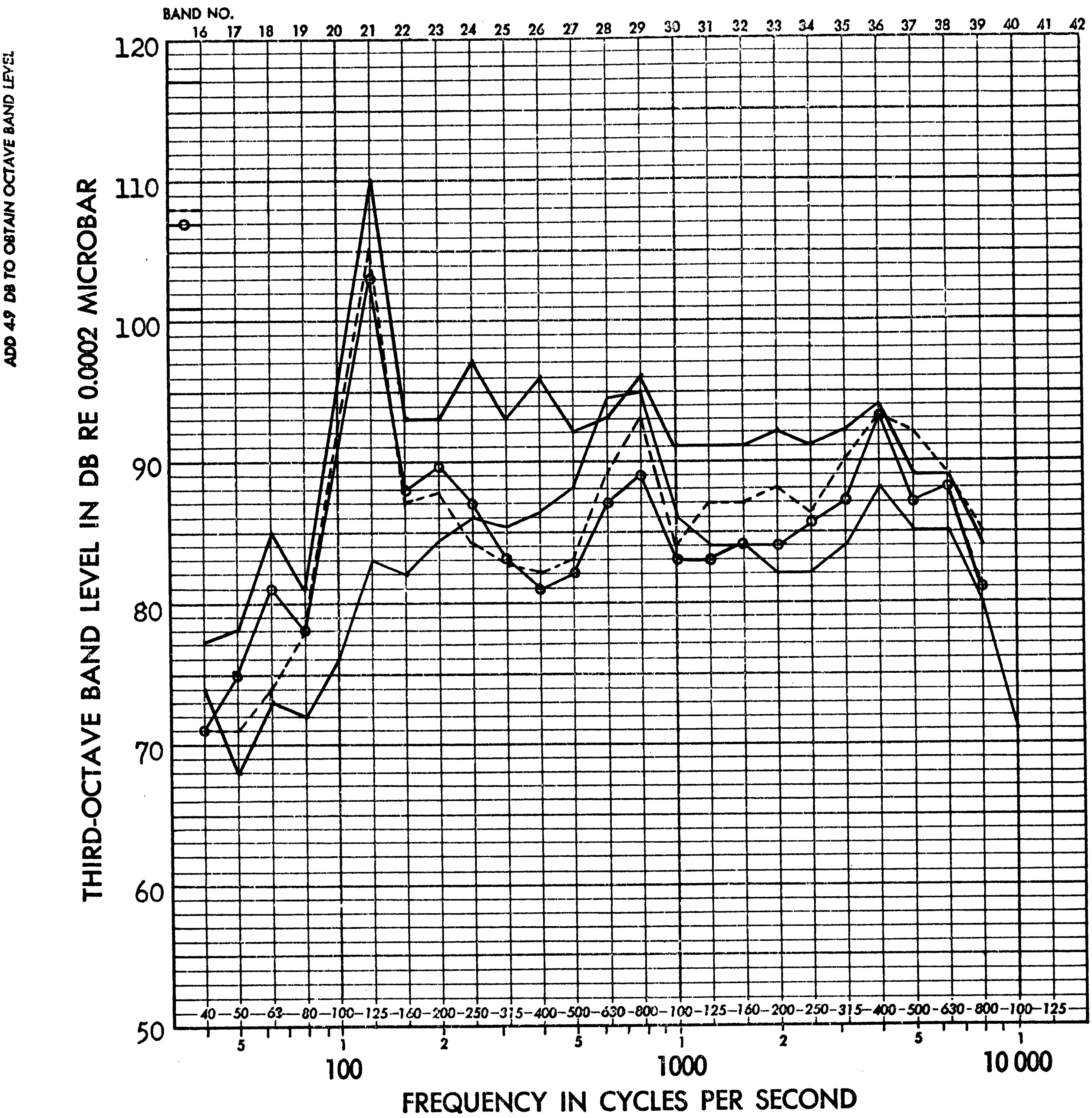


Fig. 17. Mechanical-pulse-absorber design, Modification I (top view). Dimensions are similar to those in Fig. 15.

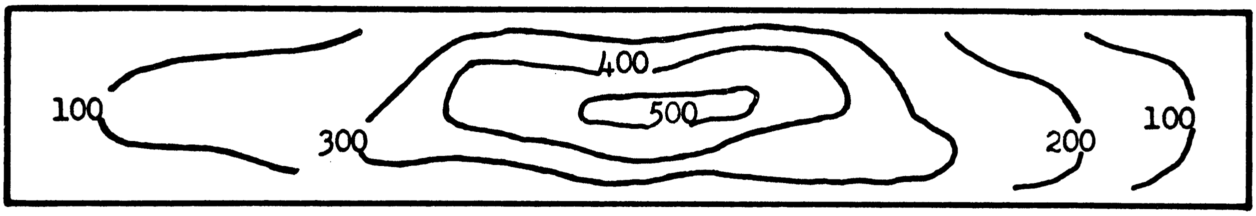


--- Double pulse absorber

-o- Double pulse absorber, modification I

Corresponding average overall levels are shown on the extreme left.

Fig. 18. Pulse absorber, Modification I, sound spectrum.



Air temperature (°F) at duct outlet, 2400 rpm, 350 ft-lb.
Pulse absorber, modification I installed.

Fig. 19. Temperature distribution with pulse absorber, Modification I.

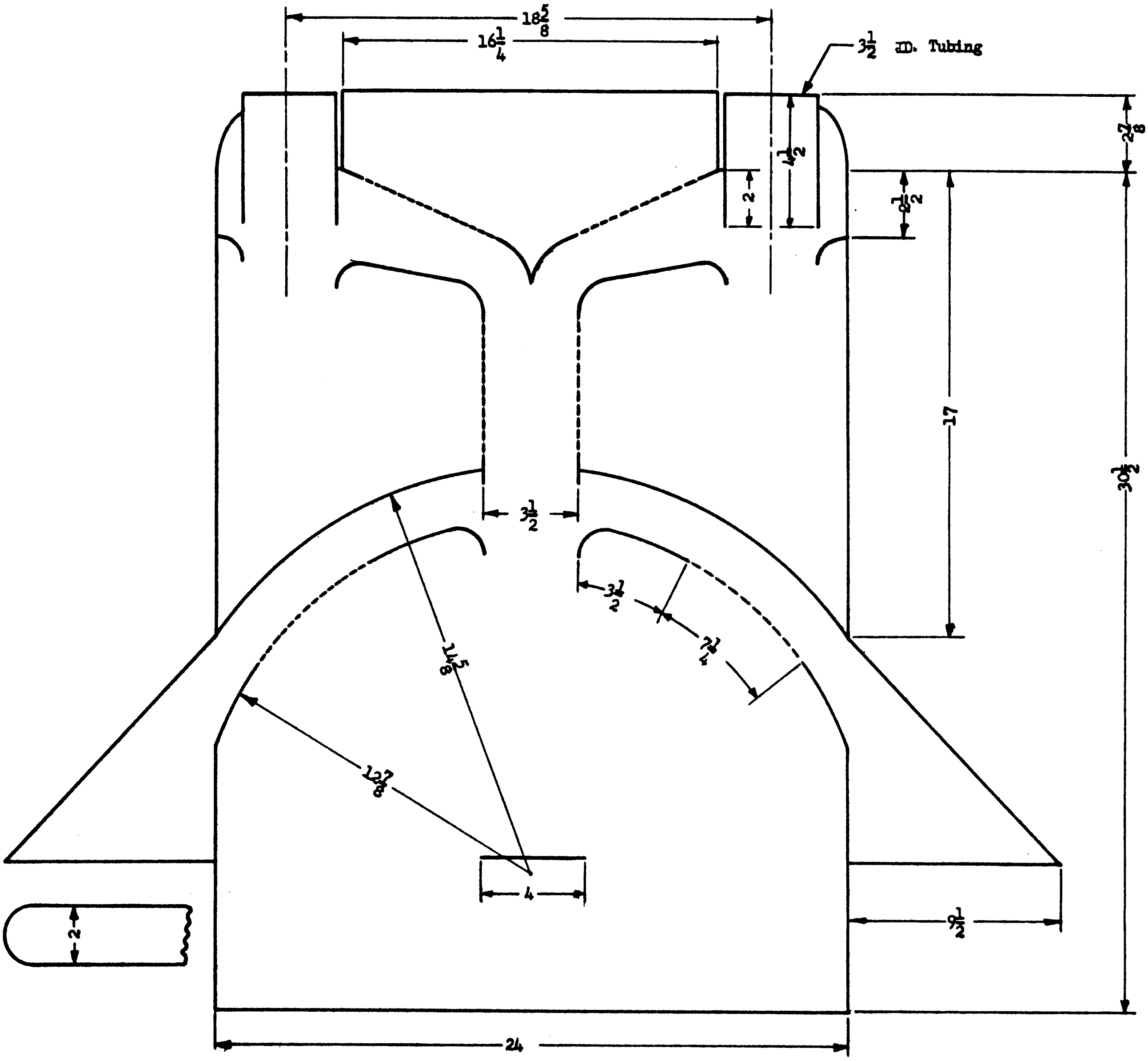
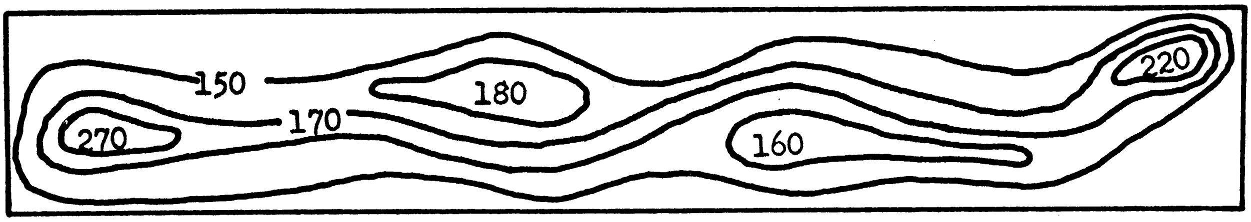
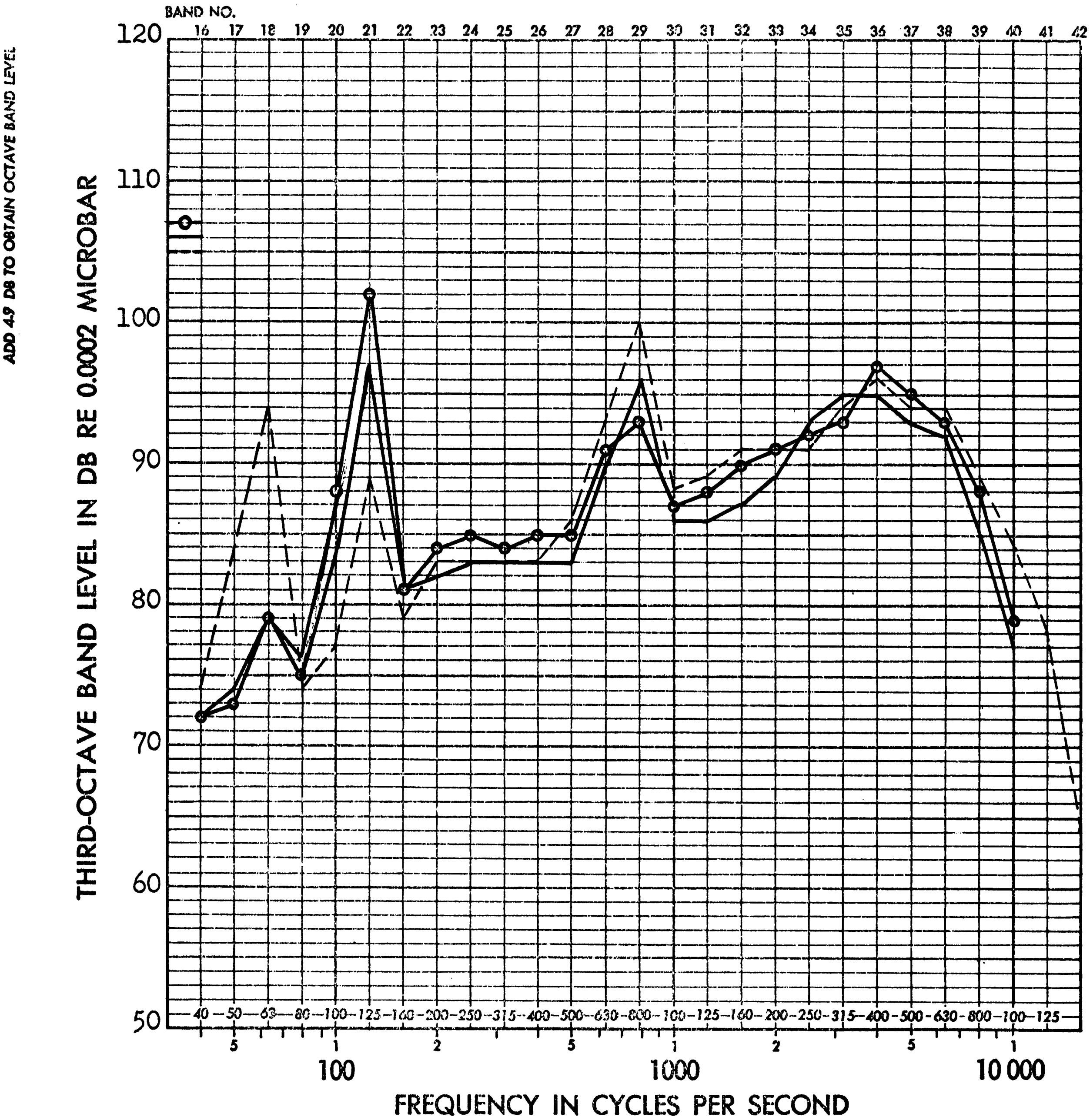


Fig. 20. Mechanical-pulse-absorber design, Modification II (top view).



Air temperature (°F) at duct outlet, 2400 rpm, 350 ft-lb.
Pulse absorber, modification II installed.

Fig. 21. Temperature distribution with pulse absorber, Modification II.



--- Maxim silencer
-o- Pulse absorber, modification III in duct
—— Pulse absorber, modification III out of duct

Corresponding average overall levels are shown on the extreme left.

Fig. 22. Effects of muffler position and comparison of Maxim silencer to pulse absorber, Modification III.

10. Effects of Muffler Position During Tests

Since the Maxim muffler and its ability to silence is quite well known, one of the tests which was run gives a fair comparison of the absorber-type muffler to that of the Maxim.

The muffler mentioned in Section III-B-9 was completely separated into two sections and tested in the normal duct position and also out of the duct in a position at which the Maxims were tested. The position outside the duct caused a reduction of 5 db in the 120-cps amplitude (see Fig. 22). Part of this reduction might have been caused by the fact that the absorber mufflers may not have transmitted quite as much noise because their outlets were oriented at about a 30° greater angle from the microphone. However, because of the low frequencies involved, this effect is probably small. Having the mufflers in this position is the best comparison with the Maxim. Not taking into account the difference in frequencies produced by the absorber and the Maxim, the pulse absorber is about 3 db higher.

11. Summary of Sound Determinations

The sound spectrum of the AOS-895-3 tank engine can be roughly divided into three parts. The low frequencies, up to about 400 cps, are primarily associated with the firing rate of the engine. At low speeds, two peaks, corresponding to the firing rate and its second harmonic, can be observed in this region. The peak in the 630- to 800-cps band was found to be caused by the engine cooling fan. The rate at which the blades pass a given point was calculated to be 800 cps at 2400 rpm. The remainder of the spectrum is attributed to wind noise, generator noise, and valve clatter, although the fundamental valve operating rate is of the same frequency as the firing rate.

The breakdown of the frequency spectrum leads to a simple method for acoustic comparison of mufflers. At this time, it should be pointed out that until maximum silencing of the engine is accomplished, the total sound-level meter reading is only 1 or 2 db higher than the level of the firing-rate peaks. Therefore, if one were to modify the response of an ordinary sound-level meter so that it measured only those frequencies below 400 cps, the relative merit of experimental mufflers could easily be compared by operating the engine at a given speed and measuring the low-frequency sound level at a given distance from the engine exhaust. The sound-level-meter modification can be accomplished with a simple L-R-C filter plugged into the front of the meter. A filter suitable for this work is commercially available from the United Transformer Company. The filter, LMI-400, can be connected directly to the filter jacks on the General Radio Sound Level Meter Model 1551-A.

C. EXHAUST PRESSURE PHENOMENA

Because of the important part that exhaust pressures play in this investigation, the reliability of measurements taken, and of the results obtained by using usual engineering methods, should be considered.

The pressure which is measured in exhaust systems constitutes a dynamic condition, varying greatly from one instant to another. This was mentioned briefly in Section I. A very good explanation is to be found in Chipley.[3] Since it is a dynamic condition, it can be measured accurately only with suitable dynamic instrumentation. Several methods can be used, including strain-gage transducers, which is the method used in our test setup. The problem encountered with all dynamic systems is the difficulty in accurately determining the absolute or reference pressure.

The manometer, on the other hand, is primarily suitable for reasonably steady pressure measurements; consequently its use in our application is very limited. However, it was used in our tests as a source of comparison. Reasonable comparisons may be expected when rpm, manometer pickup position, manometer tube size and length, etc., are kept constant. The error which is introduced when one of these parameters is changed, is shown in Fig. 23. In this figure the solid line shows readings with the manometer direct connected: the dash line shows readings with a one-quart container interposed between the manometer pickup and the manometer, 6 in. from the pickup.

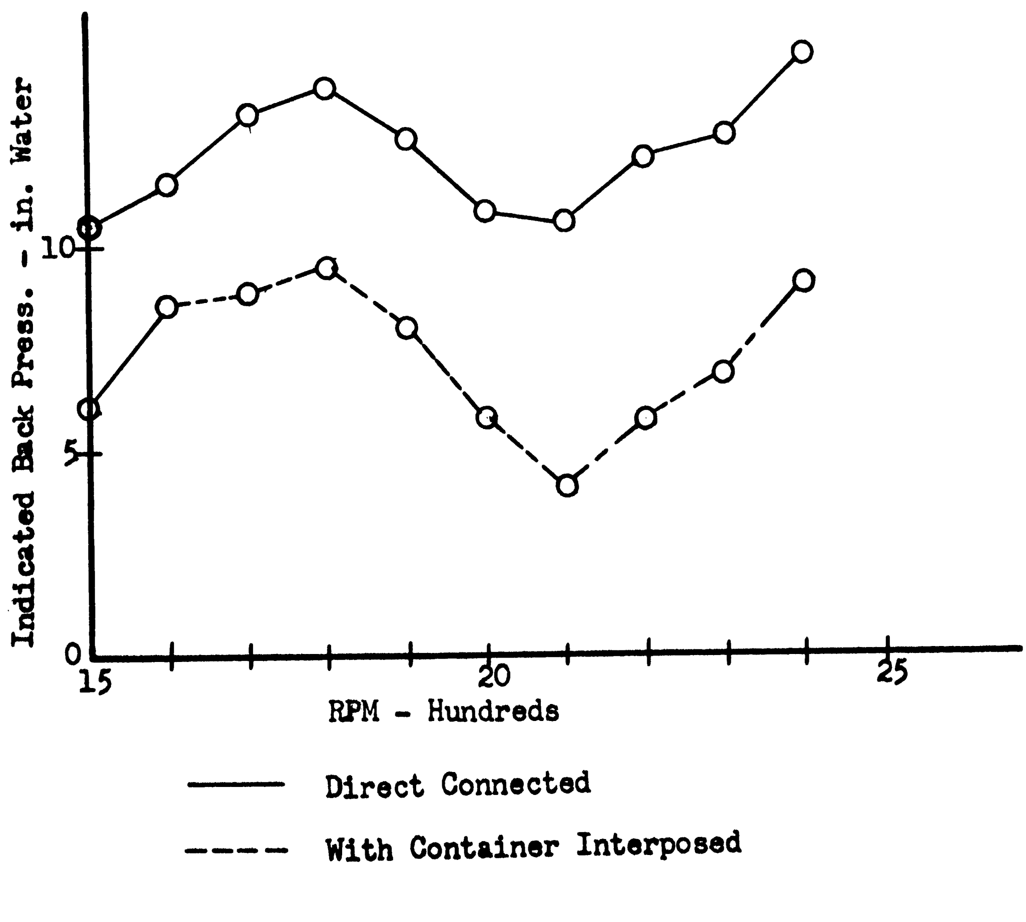


Fig. 23. Manometer phenomena.

A manometer used under these conditions is thus indicating only a "compromise" pressure, with an introduced error, caused by the pulsating pressure of the exhaust gases. This "compromise" pressure reading is merely an indication of muffler restriction, not of the pressure which is opposing the engine exhaust when the valve opens.

An indication of the dynamic pressures in the system is shown in Fig. 24. All pictures were taken to show the three exhaust pulses which are produced during each camshaft revolution in one bank of cylinders of this engine. The pulses are timed with the camshaft motion, shown as the lower line in some cases.

The three larger pulses are the initial exhaust discharges. The smaller pulses result from the resonating tube phenomena explained in Section I. Photographs (a) through (h) of Fig. 24 show the results when a Maxim silencer was installed with approximately 1-1/2 ft of tubing connecting with the standard manifold. Constant speed of 2400 rpm was maintained in photographs (a) through (d) and constant torque was maintained in (e) through (h). The constant-speed pressures show very little change with load. This might be expected since the photographs show only dynamic pressure changes, not the average levels which could increase with load with a muffler restriction.

Observation of the constant torque pulses show the 1500 rpm (Fig. 24) to have very equally spaced resonant pulses between the gas pulses. A rough calculation gives some indication why this is so.

$$\text{Pulse rate fundamental} = \frac{xn}{120} = \frac{1500 \times 3}{120} = 37.5 \text{ cps}$$

The resonant peaks appear evenly spaced between the pulse rate, indicating they are at twice the frequency or 37.5 x 2 = 75 cps.

The velocity of sound in the exhaust system is:

$$C = \sqrt{kg\ RT}$$

$$C = \sqrt{(1.4)(32.2)(53.3)(2000)}$$

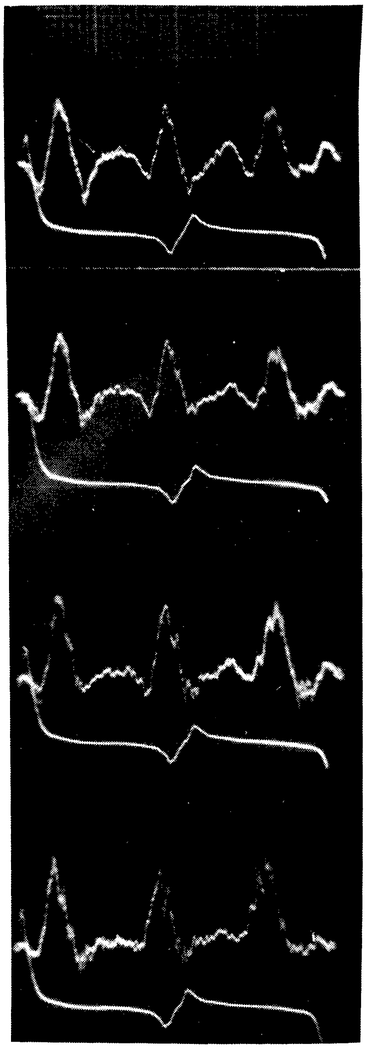
$$C \cong 2190 \text{ ft/sec} ,$$

where

C = velocity of sound, ft/sec
k = $C_p/C_v \cong 1.4$
g = 32.2 ft/sec
R = gas constant = 53.3
T = absolute temperature (°R) of the gas

Maxim Muffler Installed

2400 rpm



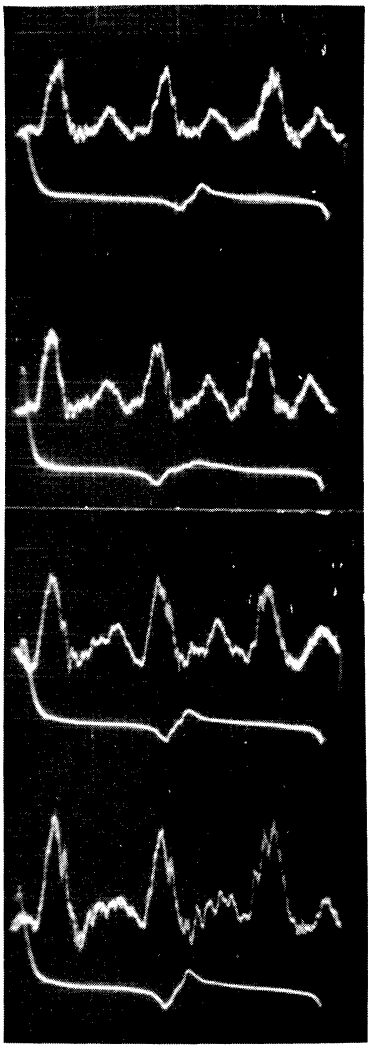
(a)
0 torque

(b)
200 ft-lb

(c)
350 ft-lb

(d)
500 ft-lb

Constant 350-ft-lb Torque



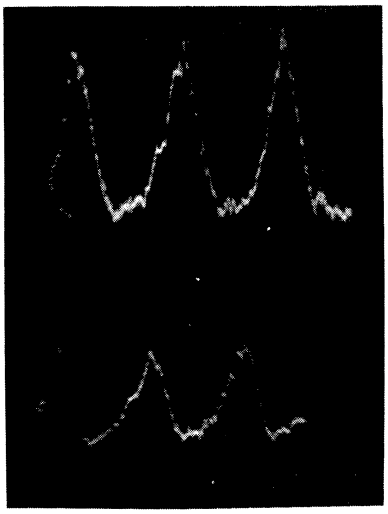
(e)
1500 rpm

(f)
1700 rpm

(g)
2000 rpm

(h)
2400 rpm

Flat Muffler Installed



(i)
2200 rpm
850 ft-lb

(j)
2200 rpm, 350 ft-lb

Fig. 24. Dynamic exhaust pressure photographs.

Since $f = C/\lambda$, where λ = wave length,

$$\lambda = \frac{C}{f} = \frac{2190}{75} = 29.2 \text{ ft} \quad .$$

For a resonating tube, the length equals $1/4\ \lambda = 29.2/4 = 7.3$ ft. This is approximately the combined length of the manifold, 1-1/2-ft extension tube and inlet pipe of the muffler.

Figures 24 (i) and (j) show the results at 2200 rpm, with the flat muffler installed. No secondary peaks are apparent, indicating the resonant condition and gas pulses are at the same frequency, causing a high instantaneous back pressure when the valve is opening.

$$\text{Pulse rate frequency (cps)} = \frac{xn}{120} = \frac{2200 \times 3}{120} = 55 \text{ cps}$$

$$x = \frac{C}{f} = \frac{2190}{55} = 39.8 \text{ ft}$$

$$\text{Tube length} = \frac{1}{4}\lambda = \frac{39.8}{4} = 10 \text{ ft}$$

This again is equivalent to the manifold length and a 5-ft extension tube leading to the muffler installed in the cooling duct.

IV. MUFFLER AND MANIFOLD DESIGN

A. MANIFOLDS AND EXHAUST PIPES

If the exhaust manifold and pipe are not designed properly, the back pressure in these systems may exceed that created in the muffler. The back pressure is due to the friction loss as the gas travels down the pipe. This loss, when expressed as a head (usually inches of mercury) is the restrictive portion of the back pressure and varies with the instantaneous flow. The instantaneous sum of this head, the muffler head, and the resonant tube effect make up the total back pressure.

In the tests that were run here, the loss in the bends was high compared to the loss in the straight pipe. The loss at 2400 rpm, full load, was computed to be about .23-in. Hg per foot of straight pipe. The computed and measured values were about the same. The approximate back pressure per foot of pipe for other conditions can be found from:

$$H_f = 1.26 \times 10^{-2} \frac{V^2}{DT}$$

where

H_f	=	in. of Hg	D	=	diam pipe, in.
V	=	velocity of gas, ft/sec	T	=	absolute temperature, °R

The energy loss in a right-angle bend when the radius of the bend equals the diameter of the pipe is .7 times the velocity head. (This factor is approximately proportional to $\sqrt{D/R}$. The velocity head in inches of mercury is $H_V = .00815\, V^2/T$.) The velocity head at 2400 rpm, full load, was about .57 in. Hg. The product of these (.7 x .57) shows that a back pressure of .4 in. Hg was developed at each bend. (This again is a static value. Instantaneous resistance to pulse flow could be much greater.)

The equations given above were developed for lower velocity gases and only give an approximation of the back pressure of a large engine. Nevertheless, they are useful in determining an initial design for a new system and for modification of an existing system.

B. MUFFLER DESIGN

1. General Design Requirements

Classically, exhaust silencing can be accomplished by acoustic filters which reflect the major portion of the sound to the source. Absorptive treatments have been mentioned but in most practical cases the exhaust gases are too hot for known acoustic absorbing materials. Also carbon and other exhaust products would limit the life of such a muffler.

Design of acoustical filters is discussed in the Appendix. Theoretically, acoustic filters should be the answer. However, as long as there remains a high-velocity gas pulse, there is a noise-generating source. Therefore, one important function of the muffler is to reduce the velocity of the pulses to the extent that they produce little acoustic energy. Then acoustic filtering can be made effective. The pulse absorber provides the best means for accomplishing a smoothing action and the tentative design criterion of such a device will be discussed in detail. Refinements in this technique could not be completely covered in this contract but variations of several parameters were tried and the results are included in this section of the report.

The purpose of the pulse absorber is to trap high-velocity pulses and by-pass the low-velocity gases with minimum restriction. Any physical design of the absorber will, in most cases, also serve the purpose of an expansion chamber. Thus it will reduce the level of the sound created at the valve.

A single-stage absorber is shown in Fig. 25. Chamber A should at least equal the volume of one cylinder, preferably about three times that volume. The inlet to chamber A is flared so that little energy is lost in entering. This flare should be at least 1/5 the pipe diameter, preferably greater. With this radius, more than half of the velocity pressure is lost when the gas tries to expand back out of this orifice. Pressure in the chamber is then relieved by reverse flow through the orifice and the holes which are in the sides. Since this pressure relief takes time, the chamber holds a portion of the gas pulse, so that there will be flow out of the chamber after the original pulse has ceased.

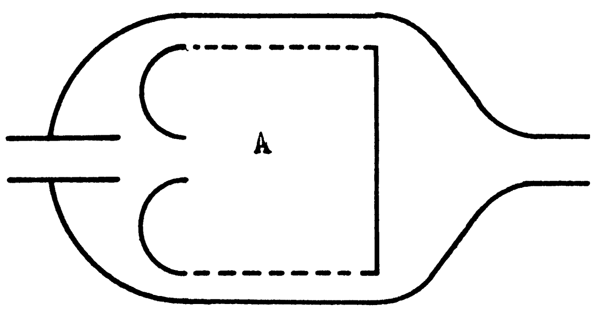


Fig. 25. Single-stage absorber.

The distance between the inlet pipe and the flare is made about 1/3 the diameter of the inlet pipe. This gap acts as an outlet for the low-velocity gases.

The absorber should be followed by a second absorber or an expansion chamber. If an absorber is used, it should be about twice the volume of the first stage but with little more total outlet-hole area.

If an acoustic expansion chamber is used, it should have a cross-sectional area at least four times the area of the inlet pipe. The greater the ratio between the inlet pipe and corresponding area of chamber, the greater the attenuation possible. The length of the chamber will depend on the gas temperature (which determines the sound velocity) and the pulse rate. Since the pulse rate will vary with speed, for a given engine, this pulse rate should be that of maximum speed or possibly normal operating speed. The pass bands will then occur at lower speeds where the noise level is lower and less attenuation is needed (see Appendix).

An estimate of back pressure can be made by considering that all the exhaust flows through the low-velocity path. Most of the static pressure loss will be in the bends. Therefore it is desirable to keep the number of bends to a minimum, which, for a pulse absorber, is one bend for each pulse chamber entrance to separate high- and low-velocity flow.

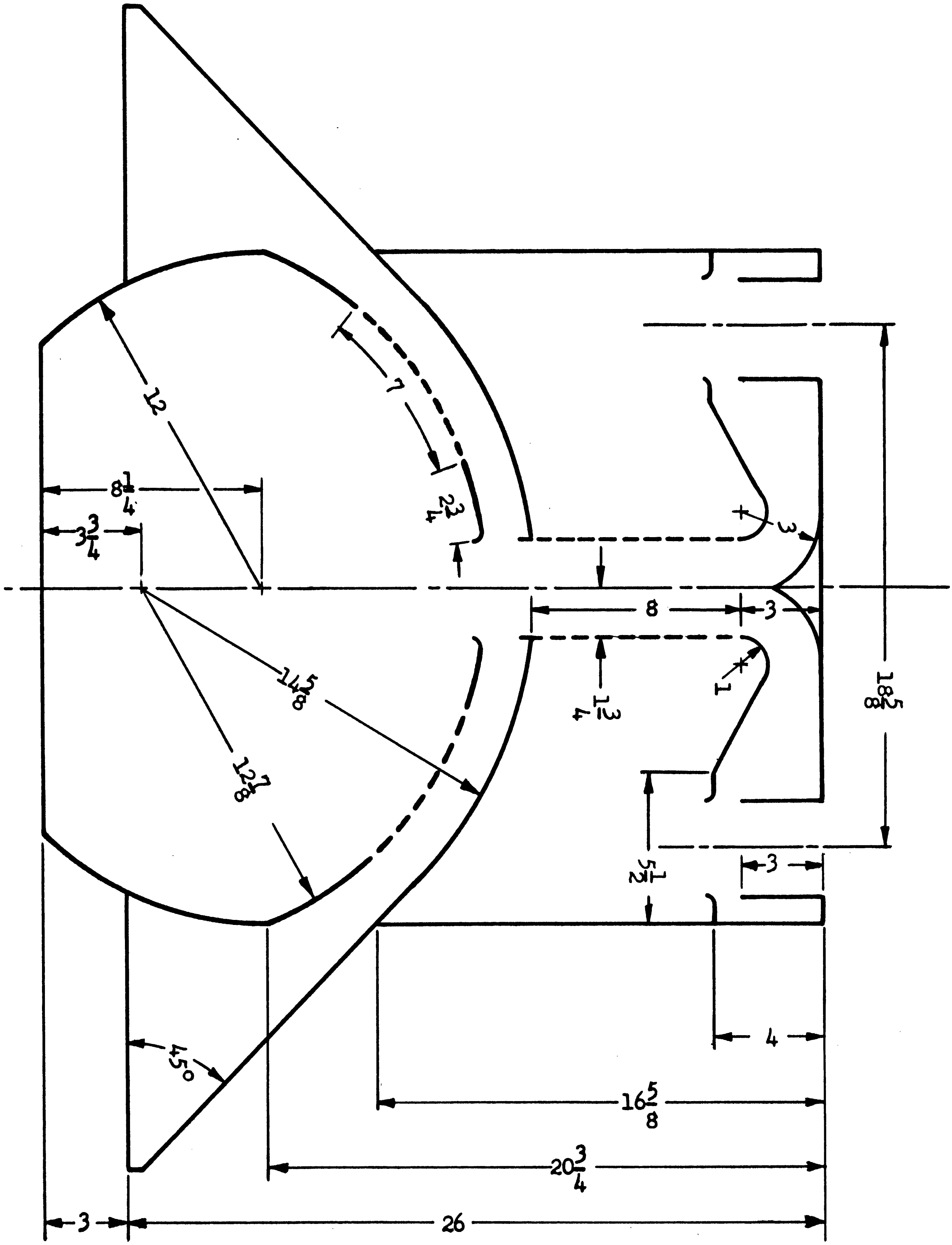
The design of a pulse absorber given here is general and can be used for an engine of any size, although the following example specifically pertains to the AOS-895-3 engine. Once an efficient mechanical arrangement is found, it is felt that the best way to adjust the muffler to engines of different cylinder displacement is simply to increase or decrease the dimensions. For instance, if the cylinder displacement is increased, the volume of the absorber chambers should be increased accordingly. If the number of cylinders is increased, the steady flow path should be enlarged. However, one size of absorber could be used for all engines by the use of absorbers put in parallel. The number required will be determined either by the volume of the absorber chamber or the low-velocity restriction, whichever criterion requires the larger number. This standard absorber would be designed for a particular cylinder displacement and a particular flow. Therefore it would be a simple matter to determine the number of absorbers needed to make the chamber volume large enough and the number of absorbers needed to reduce the back pressure to required limits.

2. Muffler Design for the AOS-895-3 Engine

The results of the sound determinations showed the mechanical pulse absorbers, Modifications II and III, to be the most suitable muffler types. To provide a more streamlined exterior shape for air-flow purposes, a redesign of Modification II was completed as shown in Fig. 26. It was also desirable to eliminate as much of the flat surfaces as possible to reduce vibration and fatigue. A muffler of this type is difficult and expensive to build; therefore the sketches of this muffler are included for reference only.

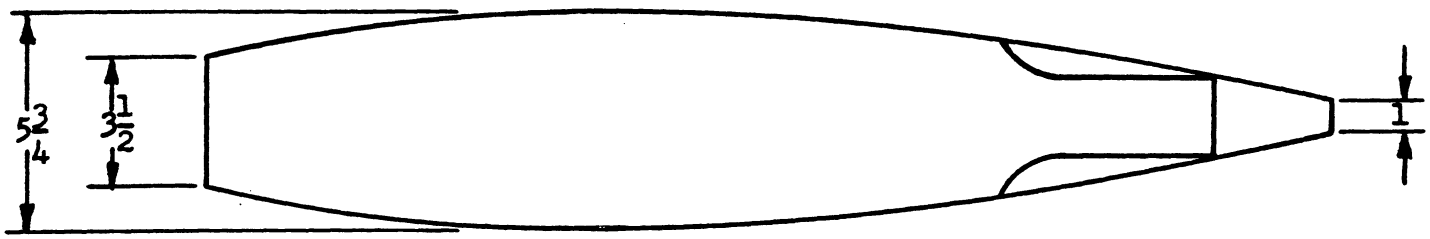
Modification V, which is the final design used here, is a direct outgrowth of the previous design. It is intended to allow simple fabricating techniques and reduce costs as much as possible (see Figs. 27a and 27b).

In an endeavor to clarify the method of muffler design, physical aspects of Modification V will be discussed here in detail.

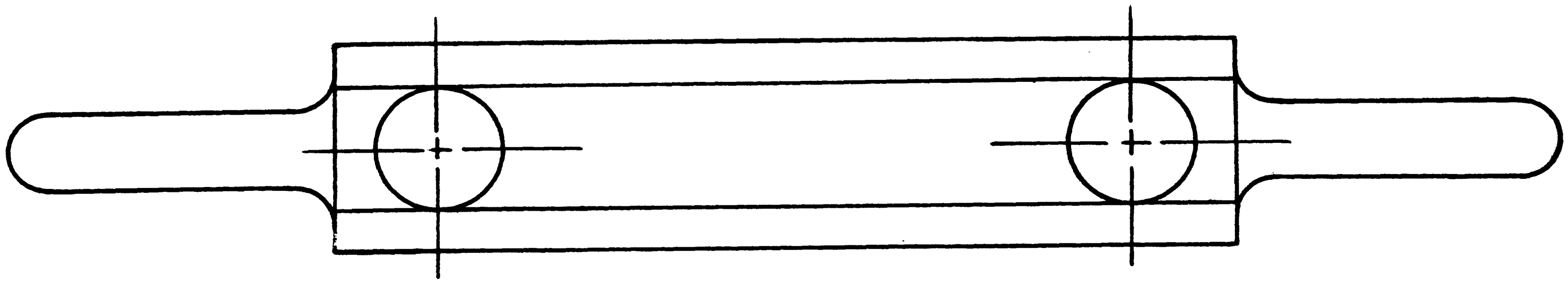


(a)

Fig. 26. Proposed pulse absorber, Modification IV (top view).



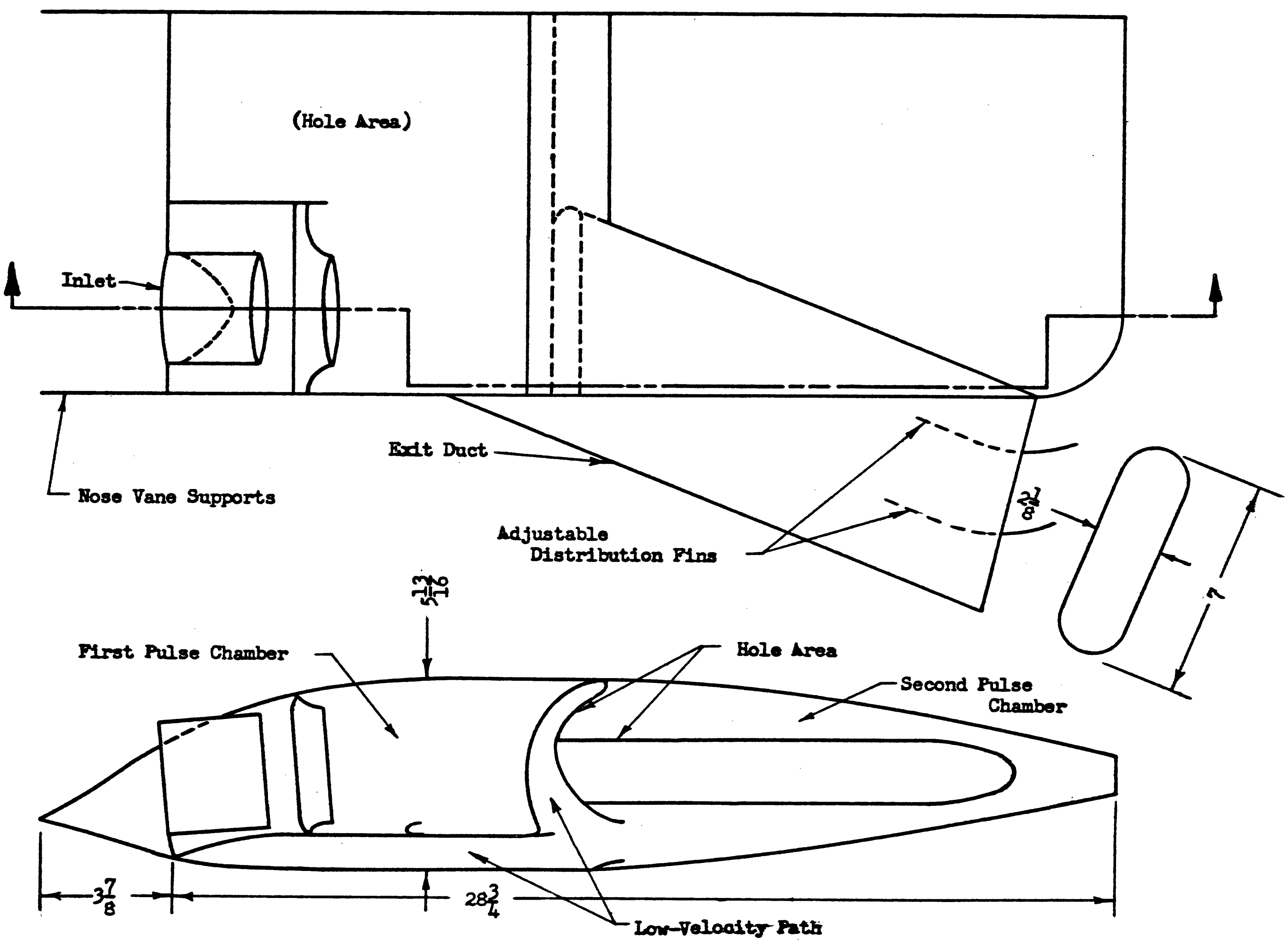
(side view)



(front view)

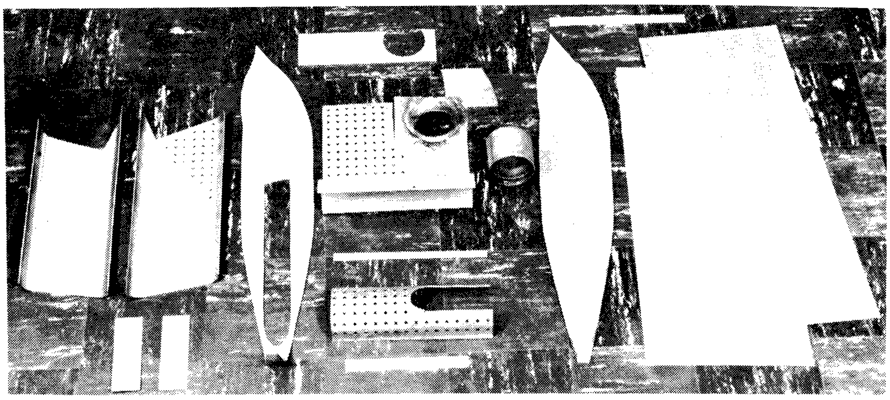
(b)

Fig. 26. Concluded.

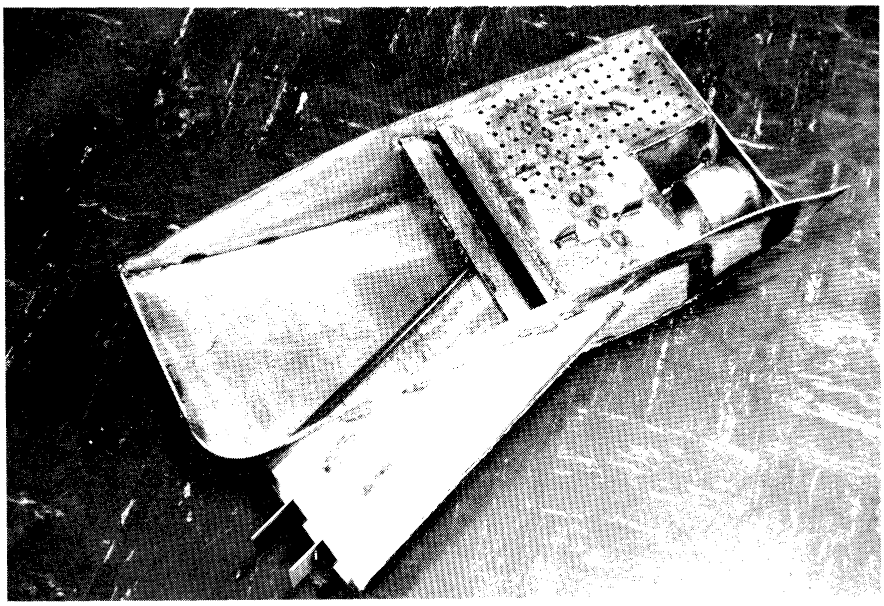


(a)

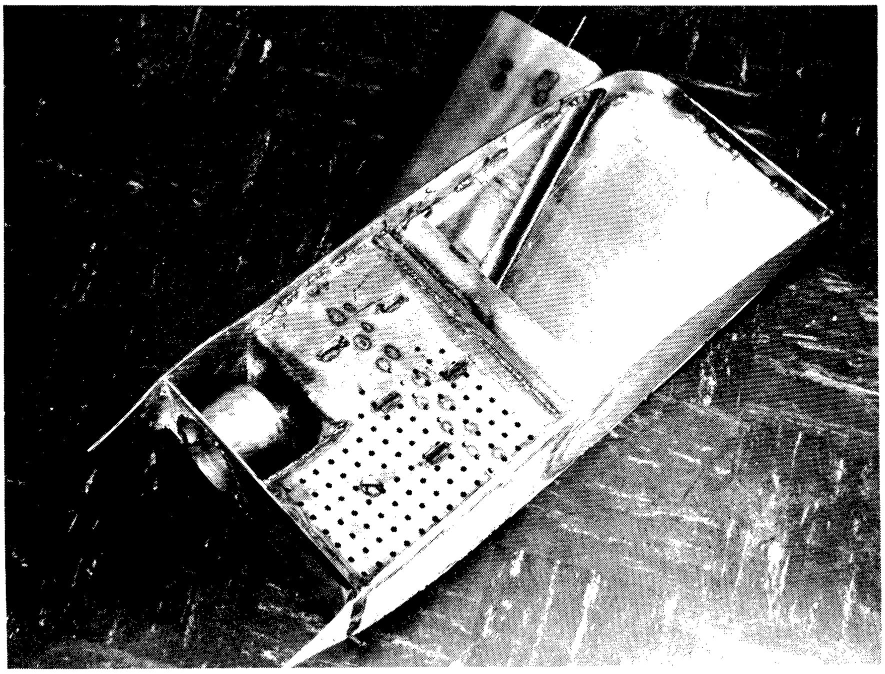
Fig. 27. Construction details and photographs of pulse absorber, Modification V.



Various parts
ready for assembly



Bottom view of
assembled section,
less bottom cover



(b)

Fig. 27. Concluded.

First the maximum cross-sectional area of the muffler is obtained from Fig. 3. This gives an allowable 37% area reduction for the specified maximum one-inch-of-water increase in static fan differential pressure. With a Venturi design, the pressure will be less than this for the 37% area and will give a good margin of safety. The duct, measuring 9 by 40 in. at the small end, will then allow .37 by 9 by 40 or 133 $in.^2$ muffler cross-sectional area. A muffler of 5-3/4 by 24 in. cross section has approximately that area, being 138 $in.^2$

To test the muffler at the manifolds without extension pipes, it was divided into two sections, forming in effect two mufflers similar to Modification III. To obtain the pulse chamber volume, the cylinder displacement has to be considered. Total displacement of the AOS-895-3 engine is 895 $in.^3$ or 150 $in.^3$ for each of its six cylinders. Three times this volume or 450 $in.^3$ determines the approximate pulse chamber volume. This would accommodate the expanded gas contained in a cylinder under a pressure of three atmospheres.

The bleed-off hole area, made equal to one-half the inlet pipe area, would equal

$$\frac{1}{2}\left[\frac{\pi(3.5)^2}{4}\right] \approx 5 \text{ in.}^2$$

With 1/4-in. holes (.049-$in.^2$ area) the number of holes required is 5/.049 = 101. The individual hole size will also have some effect on flow, but for design purposes this effect is not important.

Taking advantage of the full length of the duct, the pulse chamber size available amounted to about 600 $in.^3$ for the first chamber and 700 $in.^3$ for the second. Low-velocity flow path was kept about equal to or greater than the exhaust and pipe area, the minimum being 7.5 $in.^2$ This is increased gradually to about 13 $in.^2$ in the wing exit.

Some modification may be necessary to adapt this to the 1195 engine, for optimum sound reduction or for a decrease in back pressure. We have no specifications of this engine on which to base an opinion. Volume changes can be accomplished most easily by increasing the width of the muffler. Back pressure can be decreased by increasing the low-velocity paths or increasing the chamber-hole area, whichever is presenting the greater restriction to flow.

The model shown in Fig. 27 was constructed of Type 302 Stainless, 11 gage; and Type 304 Stainless, 16 gage. The type of steel was determined by the availability of those steels at the time of construction. For maximum durability the steel should be very resistant to temperature; 25% chrome steels are about the only usable steels in the 2000°F range.

Steels at high temperatures have very low strength, and are subject to creep (a continual elongating condition which is nonexistent at room temperaatures) and corrosion. Corrosion is the main difficulty in muffler design; the strength and dimensional stability fortunately are not important. Corrosion is thought to be caused by a precipitation of carbide which contains considerable amounts of the chromium from the surrounding metal. With a reduced amount of chromium in these areas, they will be more subject to corrosive attack. This problem is magnified in muffler design by exhaust gases containing incompletely burned fuel which carburizes the steel. After a few hours of test runs the steel in the vicinity of the welds was considerably oxidized.

The 16-gage should also be increased for any production mufflers to obtain the maximum stiffness possible to avoid fatigue vibration difficulties. An alternate solution if production costs warrant, would be to form stiffening sections in the metal. An equivalent of this was done on these experimental mufflers by adding two ribs on each muffler surface. The internal surfaces were curved as much as possible to achieve the same effect.

3. Muffler Parameters and Their Effects

The Modification V muffler was installed on the engine and tested. The noise level was high and was accompanied by resonances at several speeds. The back pressure was 1.5 in. Hg at 2400 rpm, 800-ft-lb torque. This muffler obviously had considerably different characteristics from those of the previous models, despite being very similar in construction. Chamber size, low-velocity flow paths, and bleed-off hole areas were nearly alike in volumes and areas. The new muffler, however, had the bleed-off holes distributed over a greater area in the chambers, and the flow paths generally must have had less resistance to flow than those of the previous models, perhaps due to a less turbulent condition of the gases.

Since the muffler had an undesirable noise level, some investigation into the various muffler parameters was necessary to determine their effects on the sound level and on the back pressure. Time and funds did not permit a complete sound analysis on the modifications which were tried, but comparative results were obtained by direct readings on the sound equipment and by listeners' opinions.

a. Low-Velocity Paths.—To determine if the high-velocity gases were entering the low-velocity path below the orifice of the first chamber (see Fig. 27), the path was blocked with a 3/4 by 3/4 angle iron inserted through a slot made in the side of the muffler. About 2-db decrease in the 122-cps tone was obtained at 2400 rpm, 350-ft-lb torque. The back pressure increased from 1.5 in. Hg to 2.7 in. Hg (full load 2400 rpm). Closing this path in this manner also restricted 28 holes in the first chamber, so the back pressure is partially due to this restriction. Later tests will show that this is a minor effect.

A second test was conducted which closed the low-velocity path at the entrance to the second chamber, forcing various amounts of the total flow through the second chamber. The control was affected by a 2- by 1/8-in. bar inserted through a slot cut in the sides of the muffler. The following results were obtained.

2400 rpm, Full Load

% of Path Cut Off	Back Pressure, in. Hg
0	1.5
16.6	1.7
33.4	1.9
50.0	2.0
66.6	2.2
83.4	2.4
100.0	2.5

The number of resonant peaks was reduced from five to three with 100% of the path cut off. The sound level of the firing rate was reduced about 3 db.

b. Hole Area.—The hole areas of both chambers were next reduced by blocking a number of holes in the chambers.

The following results were noted.

	No. of Holes Remaining		Back Pressure, Full Load		Approximate db Level at 350 ft-lb	
	Chamber I	Chamber II	Insert Out	Insert In*	Insert Out	Insert In
1.	101**	101**	1.5	2.7	108	105
2.	74	64	1.5	2.7	107	103
3.	50	64	1.7	3.6	108	104
4.	50	40	1.9	---	105	---
5.	32	40	2.0	---	104	---

*This insert is the 2- by 1/8-in. bar described previously at the entrance to the second chamber.

**Original.

The table indicates that the original design had too much bleed-off area, and that decreasing this area had little effect on the manometer pressure, which is effected more by the low-velocity flow-path restriction. It apparently also had little effect on the gas pulses as the sound level does not decrease until over half of the hole area is closed off. This indicates that the reduction of sound level obtained by reducing the pulse velocity must be accomplished by considerable restriction in the chamber to obtain any results.

The table also supports the theory that the noise is a result of high-velocity flow; a reduction of 4 db can be obtained by closing the low-velocity path at a cost of 1.9-in.-Hg increase in back pressure, while closing the pulse chamber can accomplish this same reduction with a .5-in.-Hg increase in back pressure. There is also some indication that considerable decrease in noise may be had if a higher back pressure may be tolerated. Automobile mufflers have back pressure of from 6 to 15 in. Hg. They also are aided in having large muffler volumes 1.5 to 4.2 times the total engine displacement.[2]

The several resonances of the original muffler reduced to two when Step 4 was tried. The level indicated was the maximum of one of these resonances which occurred at 2415 rpm. Other speeds were about 10 db lower.

At this point (Step 5), the muffler was still somewhat louder than the Modification III version tested. However, that model had 2.25 in. Hg back pressure, and it is felt that equivalent silencing can be obtained in the new model with equivalent back pressure.

Since the maximum back pressure was reached, no further changes in the muffler were made. It would be interesting to continue the hole restrictions to follow the results, but it is desirable first to test the muffler in a tank installation and perhaps continue such tests after field trials are run.

The sketches and templates used in the construction of the final muffler have been given to the Detroit Arsenal.

V. CONCLUSIONS

The most flexible and effective mufflers tested were those based on the principle of mechanical pulse absorption. Outside shape should be based on Venturi principles to keep air flow restriction to a minimum.

The sound-level minimum obtained with one of the pulse absorber mufflers was 103 db for the firing-rate frequency under the laboratory conditions of this report. This was sufficiently low to be masked completely by the fan noise of the engine, and therefore was not audible to the ear. Whether this masking effect will hold in a tank installation is not known.

A second muffler of the absorber type was 3 db higher than the Maxim silencer in the exhaust-tone peaks under similar circumstances. This same muffler when installed in the duct had a level of 102 db (at 125 cps). Back pressure in these cases was 2.25 in. Hg at 2400 rpm, 800-ft-lb torque.

The final muffler design has a back pressure of 2.0 in. Hg but the sound level is higher. With a slight increase in back pressure resulting from additional restriction in the pulse chamber discharge holes, similar quieting should be accomplished.

Manifold restrictions create back pressures equal to or greater than that of the mufflers. These can be reduced by large tubing size, fewer and smoother large radius bends, and minimum lengths. Any allowable back pressure can be put to much better use in the muffler design with the restriction in the manifolds and exhaust pipes kept to the absolute minimum.

Exhaust temperature was held to 300°F when mixed with cooling air at 125°F. Engine load was 2400 rpm, 800-ft-lb torque.

The mechanical-pulse absorber has the best overall characteristics and is the simplest to design. It is also applicable to an engine of any size by proper scaling or possibly by the use of multiple units.

If a particularly loud resonance should appear in a muffler, it is possible to add a resonant chamber to the muffler for increased attenuation. This chamber need not be a part of the muffler; it can be a portion of the engine compartment or framework and connected by a tube to the muffler.

VI. RECOMMENDATIONS

It is recommended that on the initial use of the design suggestions included in this report, a model or models (with various volumes) should be constructed with the following items variable:

1. Pulse chamber size.
2. Outlet hole area in chambers.
3. Low-velocity flow-path area.

The testing which was included in this project was necessarily limited in time and scope, and refinements in technique could not be developed. Variation in design as suggested above and subsequent testing should accomplish this end.

Perhaps the value of exhaust turbine devices for silencing could be investigated. Present turbine designs utilized as power sources have proved effective silencers. It seems possible that they could be modified to act only as a silencer when their use as a power source, and the resulting increase in back pressure, is undesirable. The principle of operation would be similar to the pulse absorber. High-velocity pulses would serve to accelerate the turbine, thus storing the pulse energy in the turbine's rotation. Between pulses the turbine would act as a pump, evacuating the manifold and exhaust system (possibly to below atmospheric pressure), while it gave up energy. Gas flow would thus tend to a uniform low velocity and subsequent silencing.

VII. TEST EQUIPMENT

TEST-CELL EQUIPMENT

AOS-895-3 Continental Air-Cooled Engine
600-hp Midwest Eddy-Current Dynamometer
Link Unibeam with Wallace and Tiernan Pressure Gage
Merian Inclined Water Manometer
U-Tube Mercury and Water Manometers
Brown 48-Point Chromel-Alumel Potentiometer

DYNAMIC PRESSURE EQUIPMENT

DuMont Dual-Beam Oscilloscope
DuMont Record Camera
Control Engineering Corporation Pressure Transducer EP-2000
Amplifier (Laboratory-Built)

SOUND EQUIPMENT

Ampex Model 600 Recorder
Electro-Voice Model 650 Microphone
Davon Attenuator
Bruel and Kjaer 1/3-Octave Filter
Bruel and Kjaer High-Speed Level Recorder
16-in. University Loudspeaker
White-Noise Generator (Laboratory-Built)
Fairchild Audio Power Amplifier
General Radio Audio Analyzer Model 760-A
Bruel and Kjaer Beat Frequency Oscillator
General Radio Sound-Level Meter Type 1551-A
General Radio Transistor Oscillator Type 1307-A
General Radio Sound-Level Calibrator Type 1552-B

VIII. APPENDIX

A program was carried out to find the effect of theoretical acoustic filters. Some information on this subject was found in the literature but the results were based on theoretical calculations and conditions. It was important to know how closely measured results would agree with calculations. To do this, some experimentation was necessary.

The muffler test setup discussed in Section III was used except that a tone from a sweep oscillator was substituted for the white noise. The tone output from the various filters was compared with a pipe of the same length. This in effect gives the insertion loss of the filter. Since these filters were connected with pipes of finite length, they more closely represent the conditions encountered in actual practice.

1. RESONATOR

A resonator filter was tried first. The frequency of resonance should be

$$f_r = \frac{c}{2\pi}\sqrt{\frac{c_o}{V}} ,$$

where

f_r = resonant frequency, cps
c = velocity of sound, ft/sec
V = volume of resonator, ft^3

$$c_o = \frac{\pi r^2}{2rl + \pi} \quad \text{(see Fig. 28).}$$

Therefore the resonance should be sensitive to both volume and connecting tube or gap changes. This was found to be true even though the resonators were terminated in short lengths of pipe. Figure 28 is a curve based on theoretical calculations that should be used to design a resonant chamber, the theoretical attenuation factor being $\sqrt{c_o V}/2S$. The scale of the abscissa is the log of the ratio of input frequency to the resonant frequency of the resonator. From experimental work done, it appears that maximum attenuation obtainable is limited and is constant within broad limits of the factor $\sqrt{c_o V}/2S$, where S = area of main pipe. However, the band width is changed by this factor; so the value should be made as high as possible to obtain attenuation over a greater band of frequencies.

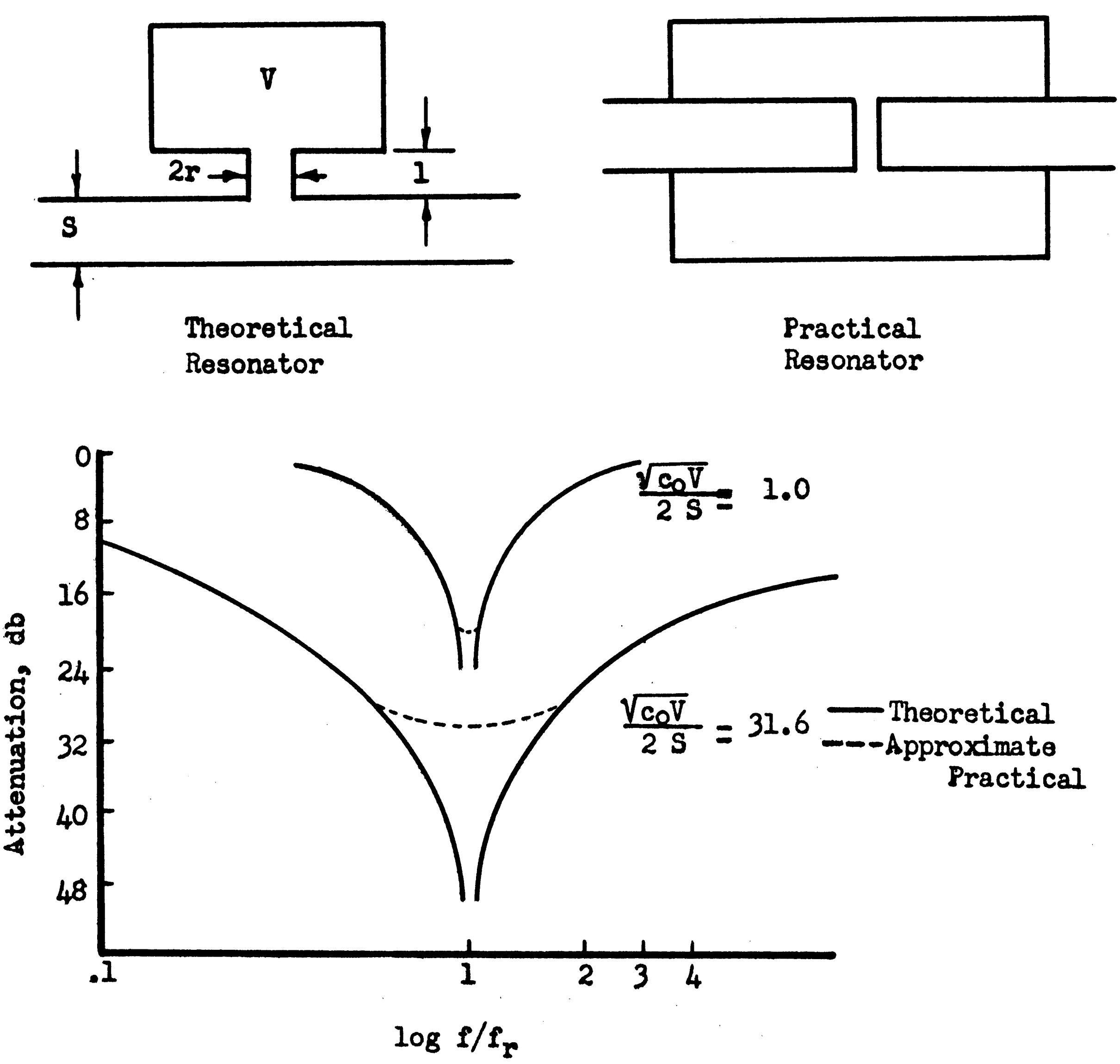


Fig. 28. Resonator normalized attenuation curves.

2. SINGLE EXPANSION CHAMBER

A single expansion chamber was tried next. The attenuation of the chamber agreed quite well with that calculated, considering the complexity of the process and the fact that the input and output pipes were terminated with reflection. Appreciable deviation from the theoretical curves occur only at the inlet- and outlet-pipe resonances. The attenuation pattern in theory is shown in Fig. 29.

3. MODIFICATIONS OF EXPANSION CHAMBERS

A damping treatment was tried to see if the vibration of the panels of the mufflers affected the muffler noise output. This apparently had no effect on the noise output through the outlet pipe. However, unless the muffler is of quite heavy material, a damping treatment is needed to eliminate acoustic energy from being radiated from the muffler shell outward into the air. A light

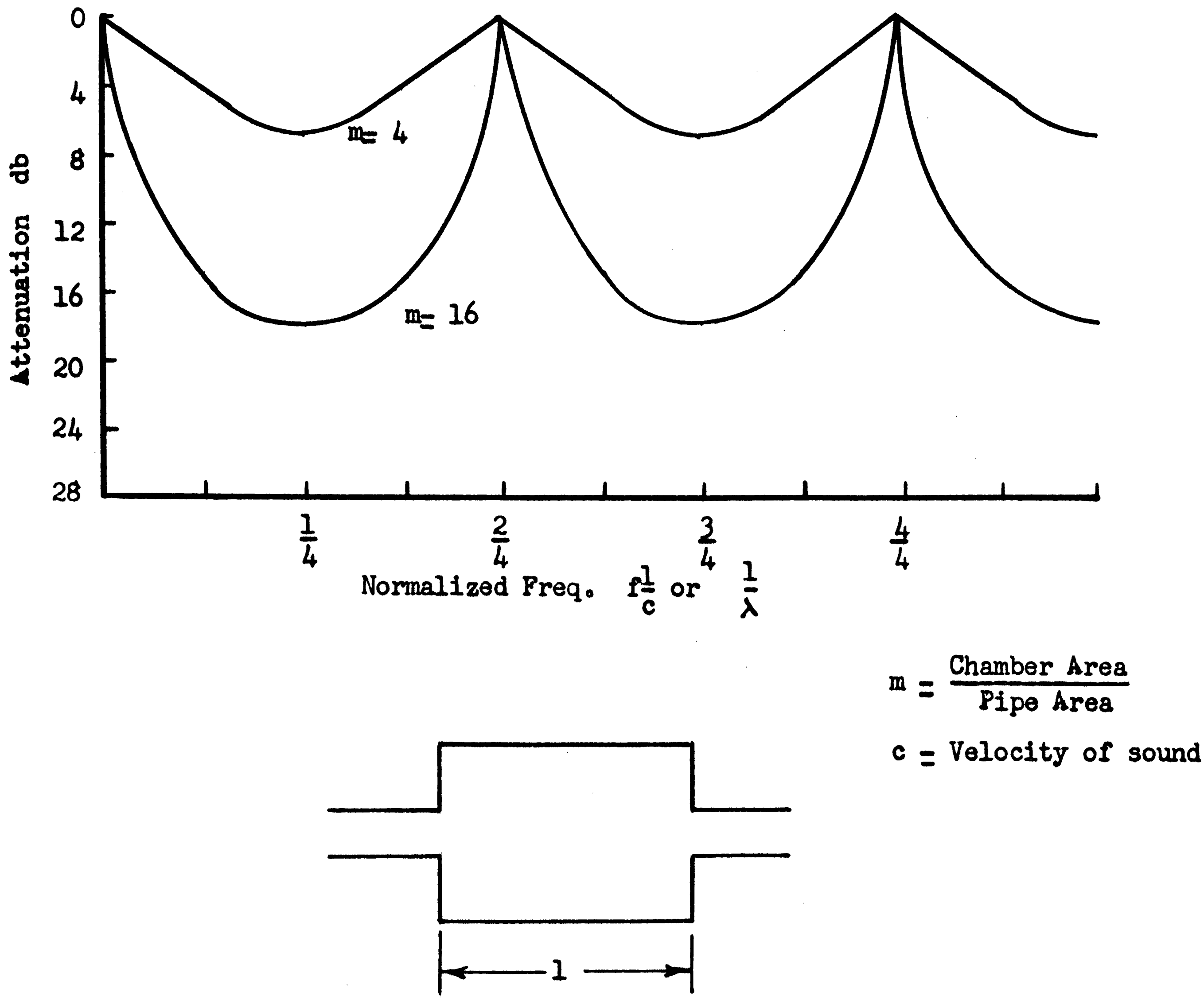


Fig. 29. Single-expansion-chamber attenuation curves.

piece of metal, about 30 gage, wrapped tightly around the muffler, would probably damp sufficiently.

Packing the various walls of the chamber with two inches of Fiberglas was tried to find out whether or not the absorption would enhance the attenuation characteristics. This caused a sharp increase in attenuation over a narrow band width. The increase occurs at frequencies which have a wave length twice a dimension of the chamber, i.e., height, width, or length.

Other modifications, such as putting a baffle between the inlet the outlet pipes, connecting these pipes with perforated tubing, and pipes that extended into the chamber, were tried. None of these had an appreciable effect on the attenuation except the baffle, which gave a zero attenuation point at about half the frequency of the chamber. The baffle probably gives the effect of dividing the chamber into two expansion chambers. These modifications would also have negligible effect on double expansion chambers, discussed below.

From the preceding correlation between calculated attenuation and measured attenuation, it was felt that a muffler attenuation could be predicted very closely from theoretical calculations which neglect input and output pipe terminations.

4. DOUBLE EXPANSION CHAMBER

An attenuation curve is shown for a double expansion chamber in Fig. 30. The best attenuation curve is one in which the inner pipe length is equal to the chamber length. This particular filter has a high attenuation compared to a single expansion chamber, considering the change in area from the exhaust pipe to the muffler, and in addition has attenuation over a wide band of frequencies. It is interesting to note that the Maxim silencer uses this principle. However, in the Maxim silencer, the chambers are of different lengths and the connecting pipe is longer than either chamber.

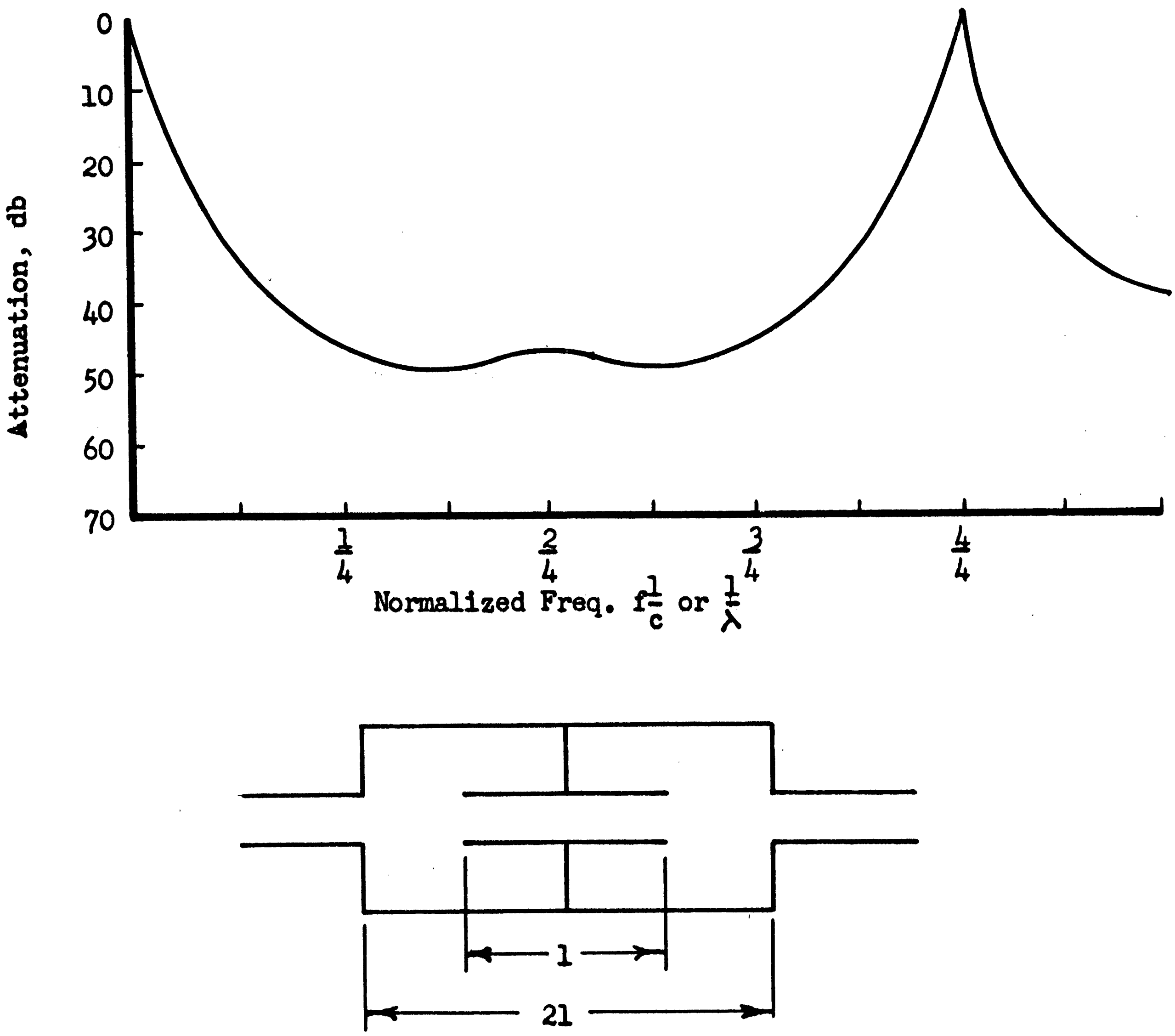


Fig. 30. Double-expansion-chamber attenuation curve.

It is noted in Fig. 30 that the filter has a pass band, i.e., no attenuation, $\ell = \lambda$. It is unfortunate that a resonator cannot be designed on paper, constructed, and put on the end of this expansion chamber to serve as a muffler. The problem arises from the fact that a resonator connected at the filter output would change the impedance enough to make even a rough calculation too inaccurate. However, suitable results could be obtained by a slight tuning of the resonator.

The Maxim silencer had a bad pass band at 400 cps, when tested in the laboratory setup. This was well taken care of with a resonator, as shown in Fig. 28. From an analysis of tank noise, a frequency of 400 cps, coming from the muffler, may be masked by other engine noises, so that the only important frequencies are the low-order harmonics of the firing rate.

IX. REFERENCES

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