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REDUCTION OF NOISE LEVEL OF EXHAUST GAS EJECTORS AND THE EFFECTS OF A BALLISTIC GRILL AND OF WIND VELOCITIES ON EJECTOR PERFORMANCE, WITH AN AOS-895-3 ENGINE

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ABSTRACT

This report deals with the silencing of exhaust and gas ejectors in their application on an AOS-895-3 engine.

A reliable method of acoustic measurement was devised utilizing 1/3-octave-band analyses, and space and time averaging techniques. Broad band measurements were useless in this case because of the very dominant noise from the engine cooling fan.

Comparisons were made of sound levels with various modifications using mufflers and absorptively lined ducts.

Pumping efficiency tests were conducted on a variety of combinations of manifold lengths and nozzle types in addition to those used during the acoustic tests.

The test results indicate that appreciable noise reduction is possible using suitable mufflers and absorptive silencers, but probably not without losing some pumping efficiency.

The possibility of adding a ballistic grill to the ejector system was investigated and this can be accomplished with no difficulty.

The effect of wind on ejector pumping were also investigated and reverse flow is a problem only at very slow-speed—low-power conditions in the face of strong winds.

OBJECTIVE

The object of this investigation is to extend the "Investigation of Exhaust Gas Ejectors for the AOS-895-3 Engine" beyond that covered by the Final Report on Project 2109, dated May, 1955, Army Contract No. DA-20-089-ORD-36259, so as to include:

1. Reduction of the noise level of an exhaust ejector in operation.
2. Determination of the effect of military vehicle ballistic requirements on an ejector.
3. Determination of maximum velocity head against which the ejector can operate without danger of carbon monoxide flow into the vehicle's engine compartment.

INTRODUCTION

This report on modifications of exhaust gas ejectors to reduce noise levels and to adapt them to armored tank application, is of necessity closely allied with the original ejector development completed at The University of Michigan in May, 1955. The results were published in Report 2109-14-F, sponsored by the Detroit Arsenal, Department of the Army, under Contract No. DA-20-089-ORD-36259.[1] The present study is one covering Recommendations 3 and 4, listed on page vi of that report.

Because of the related nature of these two projects, it is assumed that Report 2109-14-F will be available for reference, enabling us to omit certain details of theory and description of engine and related equipment for test from the present report.

All the work has been in fulfillment of Contract No. DA-20-018-ORD-14681 between the Detroit Arsenal and the Regents of The University of Michigan, wherein The University of Michigan, under the supervision of the Detroit Arsenal, is obligated to furnish all labor, material, equipment, and facilities to effect and perform the object of the contract.

The assistance of Mr. James H. Prout in collecting acoustic data and performing acoustic analyses during the latter phases of this project is gratefully acknowledged.

GENERAL DESCRIPTION OF TEST

PHYSICAL SETUP

The investigations recorded in this report were conducted in Building 22 at The University of Michigan, Willow Run Laboratories, Ypsilanti, Michigan.

The AOS-895-3 engine, coupled to a 600-hp, Midwest, eddy-current dynamometer, was located in a test cell of this building in the manner shown in Fig. 1. The test cell enclosed the engine on all sides but one with 8-in. concrete block walls and an insulated composition roof. The open side was a doorway 12 ft wide by 8 ft 6 in. high which faced an open field.

Torque, rpm, temperatures, and pressures were recorded in an adjacent control room similar to the 2109 investigation.

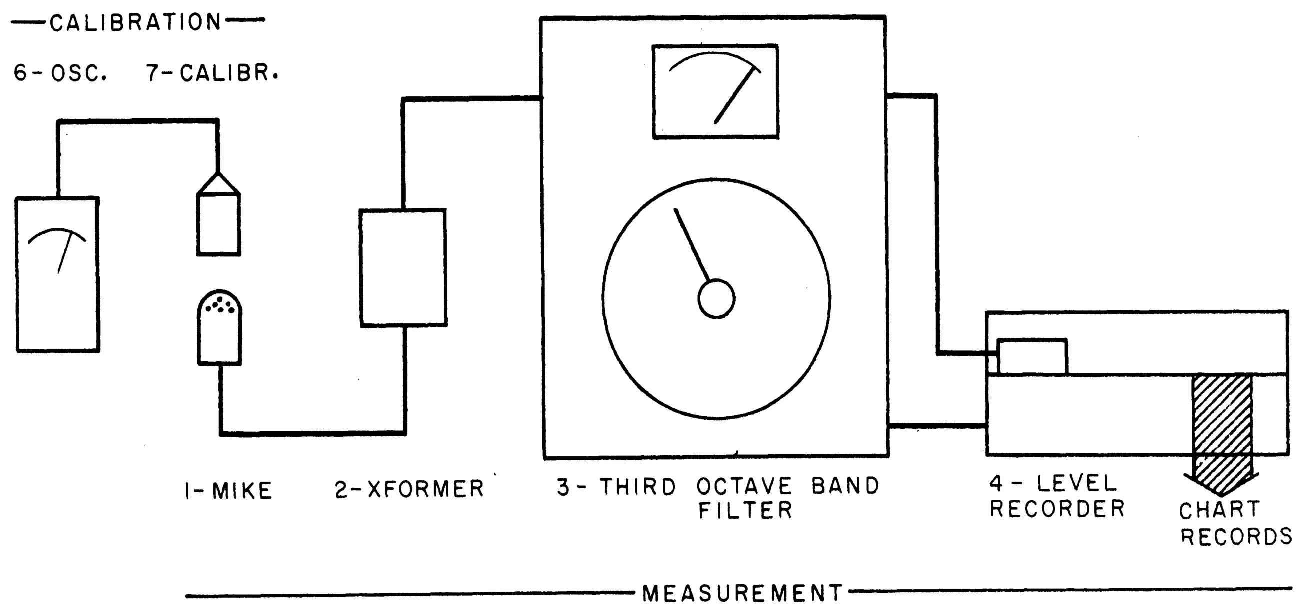
Some air-flow modifications were made over the previous 2109 setup to reduce fluctuations in cooling air-flow temperature and also to cause the air flow to follow more closely conditions which may be encountered in a tank installation. A major change was incorporated in moving the ejector about 5 ft from the engine to enable various mufflers to be installed between the engine and the nozzle, and also to allow the ejector to discharge the gases outside of the building.

TEST EQUIPMENT

The following sound-measuring equipment was used:

1. Altec 633A Dynamic Microphone
2. Altec Microphone Transformer Type K-221-Q
3. Bruel and Kjaer Audio Frequency Spectrometer Type 2109
4. Bruel and Kjaer Level Recorder Type 2304
5. General Radio-Sound Level Meter Type 1551-A
6. General Radio-Transistor Oscillator Type 1307-A (for calibration)
7. General Radio-Sound-Level Calibrator Type 1552-B (for calibration)

A simplified block diagram of the acoustic instrumentation is shown below.



Other test equipment is identical with that used in Project 2109:

1. Midwest Eddy Durrent Dynamometer - 600 hp
2. Link Unibeam Torque Indicator with a Wallace Tierman pressure gage for torque indication
3. Berkeley EPUT Meter Model 521-A for rpm indication

4. Meriam inclined water manometer for measurement of cooling air flow
5. Brown Electronic Precision Indicator 48 Point chromel-alumel for for temperature recording.

Standard U-tube mercury and water manometers were used where necessary and ASME flow nozzles were used for engine and cooling air (M_a) flow measurements.

TEST PROCEDURE AND RESULTS

DEVELOPMENT OF A METHOD OF ACOUSTIC ANALYSIS

With the major portion of the noise being radiated through the open doorway of the test cell, it was decided to conduct the acoustic measurements in the open area outside of the building. A preliminary survey of the noise levels encountered in this area with the engine running full throttle at 2400 rpm was conducted with a General Radio Type 1551-A Sound Level Meter. It was determined that the sound pressure levels occurring at a distance of 30 ft from the diffuser exit were sufficiently weak to prevent damage to the measuring microphone but still high enough to provide good discrimination against extraneous noises.

For the initial survey, 19 evenly spaced measuring locations at a 30-ft radius were selected as shown in Fig. 2. Locations 1 and 18 were on the verge of being acoustically shadowed by portions of the building while location 19 was deliberately placed well inside the shadow area. This was done to ascertain the magnitude of such effects which might be encountered in this particular environment. At each location, measurements were obtained for three different heights of the microphone above the ground, namely, 38, 57, and 69 in. These particular heights were chosen for convenience and a nonuniform spacing was selected to avoid possible undesirable effects at specific wave lengths. (Later, to confirm some observed changes in sound pressure levels in the horizontal plane, a few supplementary measurements were taken with the microphone located 16 ft above the ground.) This total of 57 microphone positions was felt to constitute a fairly reliable space integration of the sound propagating away from the front of the engine test chamber and, together with some time integration provided by the acoustic instrumentation, to be capable of yielding reliable and reproducible acoustic evaluations in the face of the existing nonideal acoustic environment.

At each microphone position, a noise spectrum was recorded using the Bruel and Kjaer 1/3 octave spectrometer and the Bruel and Kjaer Level Recorder equipped with a 50-db potentiometer. The automatic scan was adjusted to obtain one complete spectrum record in about 83 sec, dwelling about 2.7 sec on each 1/3-octave band. A block diagram of this acoustic measuring instrumentation was shown on page 2.

An initial set of measurements were obtained at all 57 microphone positions for the engine operating full throttle at 2400 rpm. An analysis of these data confirmed that the measured levels obtained for microphone location 19 (see Fig. 2) had been drastically effected by acoustic shadowing. Consequently, data taken at this location were ignored but the averages of the sound-pressure levels in each of the 1/3-octave bands were calculated for the remaining 54 microphone positions. The resulting average 1/3-octave spectrum is plotted in Fig. 4. The average over-all sound-pressure level is indicated at the extreme righthand edge of the graph. It is emphasized that the plotted points indicate average sound-pressure levels in 1/3-octave bands and by themselves indicate nothing of the constitution of the noise comprising each 1/3-octave band. The straight lines connecting adjacent points have no physical meaning.

The measuring instrumentation was calibrated at frequent intervals by applying a 400-cps pure tone signal of 116 db re 0.0002 dynes/cm^2 sound-pressure level directly to the microphone. No correction factors derived from individual instrument characteristics have been inserted into the data which were read directly from the level-recorder records. Such corrections were unnecessary since the same instrumentation was used throughout and because the acoustic invironment itself permits comparisons only. Actually, the microphone is the only part of the instrumentation chain which departs significantly from a flat frequency response. Over most of the frequency range, the variation of microphone sensitivity is also comparatively unimportant. However, the dropoff exhibited at 6300 cps and above is probably almost entirely due to decreasing microphone sensitivity. Differences between the levels of the same 1/3 octaves in this high frequency region are nevertheless valid. Indirect evidence seems to indicate that the actual noise spectrum is probably quite flat to considerably higher frequencies than is apparent from the graphs, but this is of little consequence to these investigations.

Figure 5 repeats the average spectrum shown in Fig. 4, but in addition displays the extremes encountered in the original data obtained at the 54 microphone positions. The individual 1/3-octave extreme values occurred in more or less random fashion among the 54 microphone positions. Thus Fig. 5 presents graphic evidence of the need for a careful space-averaging or space-integrating measuring technique.

The collection of spectra at all 54 microphone positions for each set of test parameters represents a prohibitive amount of labor. Consequently, the initial data were examined carefully to see how much the procedure could be simplified before the scatter of the data became excessively large. It was found that 15 microphone positions, i.e., five locations at three heights each, would suffice to yield nearly as good an average spectrum as when the whole collection of 54 positions was used, the difference in general being less than one decibel. Therefore a set of 15 microphone positions consisting of locations 2, 6, 10, 14, and 18 (see Fig. 2) and the three original heights were selected as standard for all survey tests.

A second survey was subsequently conducted under engine conditions identical with those of the initial test to determine the over-all experimental repeatability and to validate the selection of 15 microphone positions as standard. The circles plotted in Fig. 6 show the average spectrum obtained on this repeat run to be in quite good agreement with the original data. (The dashed curve represents the average for 15 standard microphone positions extracted from the original set of 54 microphone positions.)

DEVELOPMENT OF A SOUND LEVEL STANDARD

After a standard procedure for analysis was completed, it became necessary to have a basis for comparison of noise generated during various tests. Four test set-ups were then analyzed:

Ejector Systems 3 by 9, 15S + 24D No Sound Absorption

Run
1. 2400 rpm full throttle CFM (M_a) 1350 M_a/M_e 4.37
2. 2400 rpm part throttle CFM (M_a) 1330 M_a/M_e 5.97
3. 2000 rpm full throttle CFM (M_a) 1350 M_a/M_e 3.50

No Ejector System, No Sound Absorption

4. 2400 rpm full throttle

Run 1 corresponds to the repeat survey just described and the associated noise has already been presented in the discussion of Fig. 6. Run 4 had standard manifolds installed on both banks of cylinders and discharging gases into a 10-in. tube which finally exhausts through the side of the building remote from the acoustic measuring area. (During normal testing with an ejector, the other bank of cylinders exhaust via this route.) This test then established the lowest noise levels readily obtainable with this test cell, and the resulting minimum noise spectrum is presented as a thin solid line in Fig. 6. The pronounced increase in noise in the neighborhood of the 800-cps band is caused by the engine cooling fan which throughout all tests was exhausting air through the large duct shown in Fig. 21.

The average spectra for the engine with the ejector installed as displayed in Figs. 4, 5, and 6 exhibit one unusual characteristic, namely, the abnormally large amount of high-frequency noise above, say, 800 cps. Although the nonfree field acoustic environment could be partly responsible, the noise generated by a normal, unmuffled piston engine falls off quite rapidly with increasing frequency above a few hundred cps.[2] This leads one to suspect that the excess high-frequency noise in the present case may be due to the ejector and may occur as a consequence of turbulent high-velocity gas flow.

Test runs 1 and 2 (Fig. 7) permit comparison of the noise generated by the engine operating at the same speed but different power levels. Since the engine rpm remains constant, all engine tones will remain at the same frequencies and probably will change only slightly in level. In Fig. 7, very small change is evidenced at low frequencies but a drop in sound-pressure level of about 5 db is observed above 4000 cps. This decrease is probably caused by the reduction in exhaust gas (M_e) velocity.

Test run 3 (Fig. 8), at an engine speed of 2000 rpm, produced a noticable shift toward lower frequencies at the low-frequency end of the spectrum and a negligible effect at medium and high frequencies. Actually, this lower engine speed of 2000 rpm was selected to cause the formerly prominent 63-cps 1/3-octave band to be replaced by the next lower 50-cps band on the assumption that the observed prominence was due to a single discrete-frequency tone related to engine speed.

Tape recordings were taken of the noise generated under the conditions of runs 1 and 3. Discrete frequency analyses were made of the recorded noise which completely confirmed the picture of the noise which had been gradually developing from the 1/3-octave spectra and from experience with unrelated acoustic studies.

The lower frequency noise is dominated by a harmonic series of pure tones whose fundamental corresponds to 1/2 crankshaft rpm. However, because of the evenly spaced firing intervals and the fact that three cylinders are coupled together into the ejector manifold, every third harmonic is relatively prominent. In the case of 2400-rpm operation, the fundamental occurs at about 20 cps and the 60, 120, etc. tones are prominent accounting for the observed prominence of the 63- and 125-cps bands and the discrete harmonics could be followed at least through the 36th at 720 cps. Similarly, for 2000-rpm operation, the corresponding first prominent 1/3-octave bands fall as expected at 50 and 100 cps.

Actually, practically all the noise below, say, 800 cps is due to pure tones, and the observed levels of the 1/3-octave bands are the result of the presence or absence of these tones and represent typical piston-engine noise. The observed 1/3-octave band levels in the neighborhood of 800 cps, as displayed by the solid line in Fig. 6, are mainly due to the engine cooling-air blower. Above 800 cps, the noise is largely continuous in frequency and appears to be generated by the turbulent action of the high-velocity exhaust gases issuing from the ejector nozzle. Thus the high-frequency noise is predominently of the type generated by jet aircraft engines. The sound-pressure levels generated by high-velocity gas jets are known to increase as a high power of the gas exit velocity.[3, 4, 5] Thus one would expect the high-frequency noise levels observed in the present tests to be strongly influenced by any alteration of the exit velocity of the exhaust gases through the ejector nozzle, e.g., dependent on engine load and rpm. The amount of air being mixed and pumped by the ejector would be expected from aircraft studies to have relatively small acoustic consequence.

As the result of these preliminary acoustic evaluations, it now appears that quieting an engine equipped with an ejector consists of two discrete and typical acoustical problems. At low frequencies, the problem is that of silencing a typical piston engine with the added criterion that such silencing or muffling must not destroy the ejector action. At high frequencies, the problem is that of silencing a high-velocity jet of hot gases. This can perhaps be accomplished in this case by means of acoustically absorptive ducting surrounding the jet noise source.

Now it can be appreciated that the cooling air blower noise at about 800 cps falls fortuitously in the region between the two characteristic "noises" which are under investigation here and thus does not interfere materially with the analyses. The blower noise does, however, have a very pronounced influence on the over-all noise levels. Since all three noise sources are initially about equally prominent, marked reduction in either or both the engine and ejector noises would result only in rather small reductions in over-all noise which would remain dominated by the blower noise. Thus over-all noise levels cannot be employed to follow the course of this research investigation satisfactorily. An analysis using at least a moderately narrow band spectrum is essential.

EFFECTS OF MODIFICATIONS ON EJECTOR PUMPING-PART I

The development of the ejector system in Project 2109 was conducted entirely with an exhaust manifold system of 2-in. tubing. The nozzle at that time was located at the point where the 55-1/4-in. extension and muffler are attached in Fig. 3. The addition of the 55-1/4-in. extension using 2-in. tubing (actually 2-1/16-in. ID) increased the engine back pressure an indicated 2.3-in. Hg at 2400 rpm full throttle. This resulted in less than 1% additional loss in engine hp but the cooling air M_a decreased 11% at $3.8\Delta p$.

The replacement of the extension with a 2-in. ported Riker muffler (straight-through design) increased the flow resistance from 2.3-in. Hg to 11.3-in Hg and reduced the M_a 25% from that of the extension case above. The engine power loss in the three cylinders feeding the ejector increased from 1.2% to 4.5%, thus decreasing M_e which would account for some of this additional pumping loss.

The effect of an absorptively lined duct replacing the original mixing duct was also considerable. The 15-in. straight section of the ejector was replaced with 16-gauge expanded metal having a 45.5% open area. Around this, a larger 1/4-in. thick steel-plate chamber was fashioned giving a 2-in. deep spacing around the expanded metal. In this area Owens-Corning fiberglas PF 334, 2-1/2-in. thick, 0.5 lb per cu ft, was packed, giving the equivalent of a 3 by 9, 15S mixing duct as before. The pumping loss in the expanded metal system was 22.6% at $3.2\Delta p$.

EFFECTS OF MODIFICATIONS ON SOUND LEVELS-PART I

Despite the large losses in pumping efficiency with the systems studied, the acoustic tests verified the need for both a muffler to control the engine exhaust noise and an absorptive duct to silence the ejector noise. This is shown clearly by the graphs of the respective acoustic spectra.

Figure 9 illustrates the acoustic effect produced by the installation of the

2-in. ported Riker muffler.* Reductions in sound-pressure level are noted both above and below the 800-cps band. The reductions at low frequencies are attributed to the attenuations of the engine exhaust tones by normal muffler action while the reductions at high frequencies result from lower exhaust-gas velocity at the ejector due to increased back pressure and loss of heat. Thus the high-frequency noise reductions evidenced in this test resulted from the undesirable performance characteristics of the particular muffler employed.

Figure 10 demonstrates the acoustic effect of absorptively lining the ejector mixing duct. Large attenuations are observed above the 800-cps band indicating that the ejector noise has been very effectively reduced. In fact, it is doubtful whether greater improvement could be demonstrated under the existing acoustical environment since the high-frequency noise has been reduced almost to the minimum experimental noise level (compare with solid curve, Fig. 6). The actual reductions may have been even greater than illustrated in Fig. 10. (Also note that, despite these large high-frequency noise reductions, the over-all noise levels are scarcely affected, indicating the unsuitability of over-all measurements in studies of this type.) In any event, these large noise reductions observed in the high-frequency portion of the spectrum are interpreted to mean that absorptively lined ducts can exert effective control over ejector noise.

Very little effect is observed at low frequencies in Fig. 10. This was expected since the Riker muffler had been removed and since the absorptive treatment was known to be really effective only at comparatively high frequencies.

The loss of pumping efficiency was due to altered flow conditions probably induced by the expanded metal used to retain the fiberglas. An examination of the erosion suffered by the fiberglas lining indicated that the expanding cone of exhaust gases impinged on the expanded metal about 6 in. away from the nozzle and produced local circulation and reversed flows.

The Riker muffler, the particular fiberglas used for lining the duct, and the design of the absorptive duct section all were chosen for the expedient elucidation of the particular acoustical points in question. Thus their parameters were not optimized in terms of final engineering application unless by accident.

*The missing point at 3150 cps in Fig. 9 and at 1000 cps in Figs. 11, 12, 13, 14, and 15 results from an electrical malfunction of the 1/3-octave-band filter which caused some loss of data; consequently, valid averages could not be computed. However, from the limited data available, nothing of drastic acoustic consequence happened in these bands.

Nozzles with 2-in. Manifold.—With the completion of the first group of tests (Part I), it was obvious that there was too much loss in the system and that attempts should be made to reduce the engine back pressure and also to increase the efficiency as much as possible. It was thought that some further work on nozzle design to supplement the work on Project 2109 might improve the mixing of exhaust gas and cooling air.

The nozzle exit area of 2.44 in. with an aspect ratio of 12:1 developed on Project 2109 was used as the basis for the various modifications. Some of these nozzles are shown in the photographs, Fig. 22, which are as follows:

a. Nozzle used in Project 2109
b. Constant reduction of area nozzle
c. Multiported nozzle

The constant reduction of the area of the nozzle (b) was designed to permit the exhaust-gas velocity to increase uniformly through the nozzle length by having the inlet area of 3.34 sq in. decrease in a linear fashion to the nozzle area of 2.44 sq in.

The multiport nozzle was constructed with the hope of getting better mixing which would increase the efficiency in pumping and also reduce the sound level. This type has shown considerable promise in studies on aircraft jet engines.[5]

The curves in Fig. 16 show the differences between a short length of 2-in.-diameter tubing from the engine manifold to the nozzle and one 55-1/4-in. longer. There is also a comparison of the three nozzle types; the performance of the linear nozzle is very similar to that of the 2109 nozzle.

Manifold Comparisons.—To reduce the engine back pressure, the manifolding was increased from 2-in. (actually 2-1/16 ID) to 3-in. (actually 2-7/8 ID). Two-inch tubing was retained at the exhaust ports as they have openings of about this diameter. However, at the point where the exhaust gas of the rear cylinder joins that of the center cylinder, the tubing was gradually increased in size from 2-in. diameter to 3-in. diameter at the joining of the front cylinder tubing (see Fig. 19). This diameter was continued in the 55-1/4-in. extension and the nozzles.

Results indicate that the back pressure produced in the extension at 2400 rpm, full throttle, was negligible. A direct comparison of tubing diameters vs pumping efficiency is not possible because the ejector nozzles were not interchangeable on the two sizes of manifold.

Nozzles with 3-in. Manifold.—The increase in manifold tubing size to 3-in. necessitated the construction of new nozzles. Since previous tests (see section on nozzles with 2-in. manifold) seemed to indicate a nozzle of the type shown in Fig. 22a was one of the best, a nozzle of similar shape was constructed.

This nozzle was modified several times to indicate the effect a nozzle can have on a system.

The original fabrication had an opening area 9/16-in. high by 5-1/4-in. with a circular segment on each end making the total width 5-9/16, giving an aspect ratio between 10 and 11. The height was made greater than needed for a 12:1 aspect ratio as it was desired to decrease the height gradually to observe the results. The following table indicates some of the effects of nozzle height, which change the aspect ratio, back pressure, and exhaust-gas velocity. Most runs are with the nozzle about 2 in. removed from the throat of the duct (see remarks column).

NOZZLE CHARACTERISTICS
At 2400 rpm Full Throttle
with 55-1/4-in. Extension 3-in. Diameter

No.	Nozzle Height (in.)	Nozzle Back Pressure (in. Hg.)	M_a/M_e	cfm	Δ p (in. H_2O)	Remarks
1	9/16	5	6.20	1933	2.7	3x9, 15S+24D Noz. 2-in. back
2	9/16	5	5.90	1850	3.0	3 5/8x10 3/4 21S+24D, 2-in. back
3	7/16	8.5	6.50	2000	4.2	"
4	13/32	9.2	6.07	1880	3.8	"
5	13/32	9.9	6.62	2050	4.1	"
6	13/32	10.2	6.61	1900	3.8	3 5/8x 10 3/4 Noz. at throat
7	13/32	9.2	6.17	1880	3.8	3x9 15S+24D Noz. 2-in. back
8	3/8	11.0	---	1965	---	"
9	1/2 at center 5/16 at ends	8.8	5.27	1650	2.8	"

The table points out the similarities of the many arrangements rather than the differences in the results. It is apparent that refinements in technique can lead to improvements, but the gain is small. There are two trends which seemed to be indicated in the tests: 1) that the cooling air pumped can be increased 100-200 cfm by proper positioning of the nozzle with regard to the throat, this position varying with duct size and air pumped; and 2) the results with a nozzle with parallel edges on the exit area are much better than those obtained with a nozzle which is higher in the center and narrows at the ends (No. 9 in table)(see Fig. 23a).

Figure 23b shows a second model constructed of the same type with two reinforcing spacers in it. These prevented the nozzle from opening under the high gas temperatures. The material for these nozzles was Type 302 stainless steel.

Mufflers. —The effect of a 2-in. ported Riker muffler was seen in Part I. Changing to a 3-in. ported muffler of the same type gave much less loss (Fig. 18). Flow resistance amounted to .4-in. Hg. Figure 24 shows this installation.

The Maxim Silencer No. 7-3-1/2-in., Type 10649-1 had some interesting results. It was connected to the engine with the standard tank manifolds and then connected to the nozzle with a considerable length of 3-in. tubing (Fig. 25). The results are shown on Fig. 17. Thermocouple measurements at the input to the muffler and at the nozzle input indicated a 370°F drop through the muffler and connecting tube. A straight tube (55-1/4-in. extension) has about 100°F drop. The total pressure drop of the Maxim and the extension was 2-in. Hg.

Absorptively Lined Mixing Duct.—The effect of the absorptively lined mixing duct consisting of 16-gauge expanded sheet metal has been discussed on page 7. This system provided excellent acoustic attenuation but it was undesirable with respect to pumping efficiency.

Research on sound-absorbing treatments, e.g., acoustic ceiling tile, has shown that when the absorbing material is covered with a perforated rigid surface facing, only about 17% open area is required to retain most of the absorptive effectiveness of the underlying material.[16] Thus, in the present case of a sheet-metal facing to retain the acoustic treatment and project it from the destructive action of the high-velocity gases, considerable design flexibility exists without seriously impairing acoustic performance.

Along this line, it was thought that the total open area presented by the sheet-metal lining might be reduced to evaluate the effectiveness to absorption vs amount of open area and placement along the duct of the open area. To retain maximum pumping efficiency, the open areas were drilled first near the nozzle end in the first 4 in. of the mixing duct. The holes were 1/2-in. in diameter spaced 1-in. apart. A total of 60 holes were drilled, giving a 3.3% open area in the duct, but a localized hole/area ratio of 15%. This test revealed no significant decrease in pumping. A second test was then run with an additional 50 holes distributed throughout the remaining mixing duct area spaced on 2-in. centers. This gave a 6% hole/area ratio, and reduced the pumping about 1%.

EFFECTS OF MODIFICATIONS ON SOUND LEVELS-PART II

Sound analyses, similar to those described in Part I, were conducted using the new 3-in. manifold system and nozzle. Figure 11 presents the results obtained without an exhaust muffler or other acoustic treatment. A slight increase of noise levels occurred in the mid-frequency bands as compared to the sound levels generated by the corresponding 2-in. manifold and ejector system. However, no particular importance is attached to this increase in view of the extensive modifications to the manifold and ejector. Moreover, this change in noise levels is in the expected direction since improved flow conditions within the manifold would raise the ejector-nozzle velocity for similar engine operating conditions and consequently cause the generation of greater noise. No attempt is made to explain the finer details of the observed noise variations since, if they are not due solely to system variability, the mechanical modifications to the exhaust system simultaneously altered far too many acoustic parameters to permit such detailed analyses.

In the next test, a 3-in. ported Riker muffler was inserted into the exhaust system with the acoustic results displayed in Fig. 12. Only marginal quieting occurred in the low-frequency range of the spectrum where a muffler should be most effective. Apparently this particular muffler is not well matched to this exhaust system and consequently is rendered acoustically ineffective. (This result is not surprising since it is well known that, at the current state of the engineering art, mufflers have to be carefully optimized experimentally for each system.)

At the high-frequency end of the spectrum (Fig. 12), above 1000 cps, the installation of the Riker muffler has caused a considerable decrease in the sound levels. Since a normal unmuffled piston engine generates comparatively little noise in this frequency range, and since an exhaust muffler directly affects the characteristically lower frequency exhaust noise, the observed reductions at high frequencies are attributed to the indirect effects of installing the Riker muffler.

There are several recognized indirect acoustical effects which either individually or collectively could account for the observed behavior. A decrease in jet velocity results in reduced generation of noise and, in the present case, a reduction in exhaust velocity can result from increased back pressure and additional thermal losses as a consequence of the muffler installation. It might be objected that since this 3-in. muffler caused only a slight increase in back pressure, this change could not account for the large acoustic changes. However, the sound generated by a jet depends on a higher power of the jet velocity. Furthermore, when the local jet velocities reach the velocity of sound, large and rather irregular increases in noise level result, and at higher velocities, certain flow conditions can generate extreme amounts of noise. In the specific case of the exhaust ejector tested here, the nozzle velocities are known to be high enough to be susceptible to large variations in noise levels resulting from apparently trivial variations in exhaust flow.

In addition to the above, there is the possibility that an increase in the amount of cooling air (M_a) pumped could be responsible for the observed high-frequency noise reductions. In the earlier program with a 2-in. manifold, a test was started to evaluate the acoustic effect of completely closing the air intake gate, thereby reducing M_a to zero. Unrelated difficulties interrupted this test, but the partial data appear to substantiate a considerable increase, perhaps by 6 db, of high-frequency ejector noise with the cooling air completely shut off. The mixing of cooling air with the ejector jet gases appears to be capable of some noise reduction but of course not enough to silence the ejector adequately.

With the 3-in. Riker muffler still installed, the absorptively lined mixing duct containing 3.3% hole area was tested with the result shown in Fig. 13. A slight additional high-frequency attenuation occurred. Increasing the hole area to 6% brought about a marginal reduction in the mid-frequency range of the spectrum as shown in Fig. 14.

The above results, although disappointing, can be explained readily in terms of the known behavior of absorptive ducts and jet noise sources. The noise generated by a jet actually originates some distance downstream of the nozzle where turbulence is fully developed. Thus when surrounding such a noise source with an absorptive duct, the most effective placement of the absorbing material along the duct is in the immediate vicinity of the actual sources and then extending along the duct in the direction of propagation.

In the tests reported above, only a small amount of absorption was present and this was located close to the ejector nozzle, probably upstream of most of the noise "sources". This particular placement of the absorptive lining was tried because it presented a minimum of mechanical design and flow-control problems, and thus would have constituted a good preliminary design for operational equipment if it had been sufficiently effective acoustically. From the spectra it is obvious that this arrangement actually has too seriously compromised acoustic effectiveness.

The first tests with an absorptive lining (Fig. 10) showed 1/3-octave band reductions as great as 15 db. That is, in such bands, the sound pressure was reduced to only 18% of its original value. With only minor reservations, it may be assumed that at least this much reduction can be accomplished with the 3-in. manifold system by exposing more of the fiberglas surface farther downstream of the nozzle. However, design problems of protecting the acoustic lining from mechanical damage and of preserving reasonably high pumping efficiency are posed. Both the mixing chamber and the diffuser section could be acoustically treated with absorptive linings and the length of the absorptive duct increased so that almost any desired amount of noise reduction could be accomplished, although perhaps only in unwieldy configuration.

To obtain a qualitative comparison between the noise produced by the ejector systems and the probable noise associated with present operational tank applications, a noise survey was conducted with the Maxim Silencer installed. (Valid quantitative comparison cannot be accomplished because of inherent limitations of the acoustical measurements involved.) Figure 15 shows large reductions in both low- and high-frequency noise. In fact, a comparison with the minimum noise curve (solid line, Fig 6) shows that in many portions of the spectrum the Maxim Silencer reduced the noise to the lowest levels detectable in these experiments. Again, the high-frequency decrease can be attributed to a change in exhaust gas (M_e) flow, i.e., large heat losses, lower velocity.

At frequencies below 800 cps, the reductions in sound levels are attributed to the normal muffler action of the Mixim Silencer on the various engine-exhaust tones. Thus Fig. 15 shows this Maxim Silencer to be very effective for the most part. However, it does let the lowest major exhaust-noise component pass with only slight attenuation, i.e., the 60-cps tone is only attenuated by perhaps 6 db (compare Figs. 11 and 15), whereas during the minimum noise tests, the reduction of this component amounted to about 20 db(see Fig. 6).

EFFECTS OF EXHAUST AND DUCT BENDS

Several results were noted in ejector efficiency when the ejector system was modified in various ways to include bends in exhaust manifolds and ducts.

A 45-degree bend was added in the ejector between the mixing chamber (15S) and the diffuser (24D) resulting in a 40% decrease in (M_a). From this it would seem that the mixing chamber and diffuser must be kept straight for proper mixing. An exception may be made of this if properly designed directing vanes were used at the point of the bend.

Bends placed in any manner in cooling air ducts before the cooling air enters the mixing chamber have little effect on pumping efficiency. The requirements here are mainly minimum resistance to flow and a design of the duct entering the nozzle area which will give uniform distribution about the nozzle.

The effects of bends on the exhaust manifold and tubing conducting the exhaust gas (M_e) are more difficult to determine. The results of several types of setups involving standard manifolds, large radius bend manifolds, and the many types of extensions seem to indicate that the conservation of energy is most important. Keeping the manifold free of constrictions and sharp bends will keep the back pressure to a minimum and obtain resultant maximum horsepower. The other main factor is heat loss from radiation. Long lengths and large surface areas, with high radiation of energy, reduce the kinetic energy appreciably. With the Maxim muffler and 4 ft of manifold tubing, the gas temperature drops about 370°F. In a tank installation, the heat loss indicated by this temperature drop would probably transfer to the cooling air which in turn would increase the total volume of cooling air to be pumped.

The effect of a bend beyond the diffuser is discussed in the next section.

EFFECTS OF BALLISTIC REQUIREMENTS

Two tests were conducted in this regard. The first involved a multiple louvered grill shown in Fig. 26. The second used a hood-type cover shown in Fig. 20. The tests proved that a ballistic arrangement can be designed but not without some losses. The prime requirement is not to reduce the exit area of the diffuser. The louvered grill did reduce the area and the pumping reduced accordingly. The hood-type deflector, however, had a somewhat larger exit opening than the diffuser, and the drop in cooling air (M_a) was 3% at minimum Δp. See Fig. 18.

The hood type of ballistic grill can serve two purposes, as it can serve as a sound deflector and abosrber as well as for armament. With the sound deflected toward the ground, greater attenuation of the sound radiated away from the tank can be expected. Also, the interior surfaces of the hood, a location known to enhance sound-absorbing effectiveness, can be lined with a sound-absorbing material.

WIND EFFECTS ON THE EJECTOR

The wind tests were conducted on the 3 by 9 15S+24D ejector. An axial flow fan was mounted on a movable carriage and the fan outlet ducted to dimensions slightly larger than the diffuser outlet. After a check on the outlet velocities vs discharge area of the fan, a pitot tube was installed in the fan duct facing into the fan discharge.

The fan was then placed in front of the ejector so that the fan was 7 in. away from the diffuser exit with the ejector gases and fan discharge directly opposed.

With the ejector velocity opposing the fan velocity, the pitot tube then measured the fan velocity and the static pressure head created by opposing exhaust gases. Converting these results into effective wind velocity, the following data were obtained:

WIND EFFECTS ON THE EJECTOR

Effective Wind Velocity mph	Engine rpm	bhp	Cooling Air M_a			
			Without Wind		With Wind	
			lb/sec	cfm	lb/sec	cfm
47.2	1700	65	1.00	786	.824	648
43.1	1580	30	.68	535	.392	308
42.2	1600	0	.607	477	Slight Flow	Slight Flow
42.2	1500	0	.448	352	Alternating Flow	

This table is assuming the wind direction and the ejector gases are directly opposed to one another, and the resultant flow is dispersed 90 degrees from the original direction. With the ejector facing downward and protected by portions of the tank, higher wind velocities could be tolerated without reverse flow.

Revolutions per minute below those shown may also be possible in a vehicle installation if the transmission always has some load on the engine.

CONCLUSIONS

The acoustic tests on the exhaust-ejector systems have demonstrated a method for securing valid acoustic data from test-cell installations. These tests have delineated the noise problem at hand as arising from two typical and independent causes, i.e., piston-engine exhaust noise and high-velocity jet noise. Further, it has been demonstrated that these exhaust-ejector systems can be silenced by a two-step silencing process. The low-frequency engine-generated sound can be attenuated between the engine and the ejector nozzle by means of a properly designed exhaust muffler. The high-frequency nozzle-generated sound can be attenuated by absorptive linings in the mixing chamber and subsequent areas through which the $(M_a + M_e)$ gas flows. Such a two-step treatment should be able, within reasonable size limitations, to provide silencing of ejector systems comparable to that now attained with operational engine and muffler installations.

It is impossible to predict accurately from the acoustical data obtained in these tests what free field noise levels would result from the actual installation of an ejector system in a tank.

Although the reported tests have defined the noise problem and demonstrated an effective approach toward silencing, it has not been possible, within the limitations of the available time and funds, to achieve a final design which can be considered as a production prototype. The problems of simultaneously achieving good acoustical performance and high pumping efficiency appear to be quite complex, particularly in view of the large number of engineering parameters involved. The several additional acoustic tests performed indicate not the impossibility of achieving the desired silenced ejector system, but rather the need of additional careful research to arrive at an optimum design.

Muffler and absorptive duct silencing systems can result in very high losses in pumping efficiency. A muffler causing high back pressure and large heat losses seriously reduces pumping efficiency. Likewise, an absorptively lined duct which restricts or improperly directs the gas flow diminishes pumping efficiency.

A ballistic grill can be added to the ejector and can consist of a single-hood arrangement or multiple small curved sections. It is important that the open area of the grill be as large or larger than the diffuser-exit area to prevent flow restriction.

Wind will not be a problem in an ejector-cooled system except when the engine is operating at low load conditions with considerable wind from an unfavorable direction.

RECOMMENDATIONS

To reduce the sound levels produced by an ejector system of this type, both an exhaust muffler and an absorptively lined duct should be used. It is conceivable that a long or tortuous absorptive duct might satisfactorily attenuate both the exhaust noise and the jet noise, but the combination of exhaust muffler and absorptive duct appears to provide a more compact arrangement.

The muffler should provide as little interference with steady gas flow as possible while still attenuating the exhaust tones. Because of this, it may be necessary to restrict muffler designs to the straight-through types. The muffler should be designed to minimize thermal losses, and for this purpose small radiating surfaces and thermal insulation around both the muffler and the manifolds should be exploited as much as possible. This will require construction from high-chrome stainless steel or other heat- and corrosion-resisting alloys to provide extended operational life.

Since the muffler will be followed by an absorptive duct optimized for high-frequency attenuation, the highest frequency exhaust tones may be neglected in the muffler design. Thus muffler design may be optimized to provide maximum attenuation of the lower frequency exhaust tones.

The absorptive lining to control the jet noise and any of the highest frequency exhaust tones escaping the muffler can be accomplished in the mixing duct, diffuser, and in the ballistic grill, particularly if it is a hood-type arrangement. The type, amount, and placement of the sound-absorbing material remains subject to investigation. In addition to placing the sound-absorbing material on the wall surfaces of the ducts, it is also possible to erect the sound-absorbing materials in the form of splitters or guide vanes, particularly in those portions of the duct system where flow velocities are comparatively low.

The type of fiberglas used in the above tests can withstand the gas temperatures at which the tests were conducted. If higher temperatures are encountered, better heat-resistent materials are available which probably have adequate acoustic qualities.

To prevent mechanical deterioration of the sound-absorbing materials, it is generally necessary to protect them from impingement of high-velocity gases. The sheet metal with drilled holes and the expanded metal tried in some of the above tests caused excessive loss in pumping effect and necessitate alternate configurations. The use of slots, louvered openings, etc., is suggested for combining minimum flow interference with adequate mechanical protection of the sound-absorbing material.

It is often possible to cover sound-absorbing materials such as fiberglas with thin, impervious surface coatings, e.g., plastic or aluminum foil, and achieve adequate acoustic effectiveness. This protects the absorbing material mechanically and prevents the undesired adsorption of liquids, dirt, and dust. Such surface treatments should be seriously considered for this application.

It is also recommended that the ejector be directed toward the ground. This will automatically provide some attenuation of radiated noise as well as providing better physical protection of the ejector.

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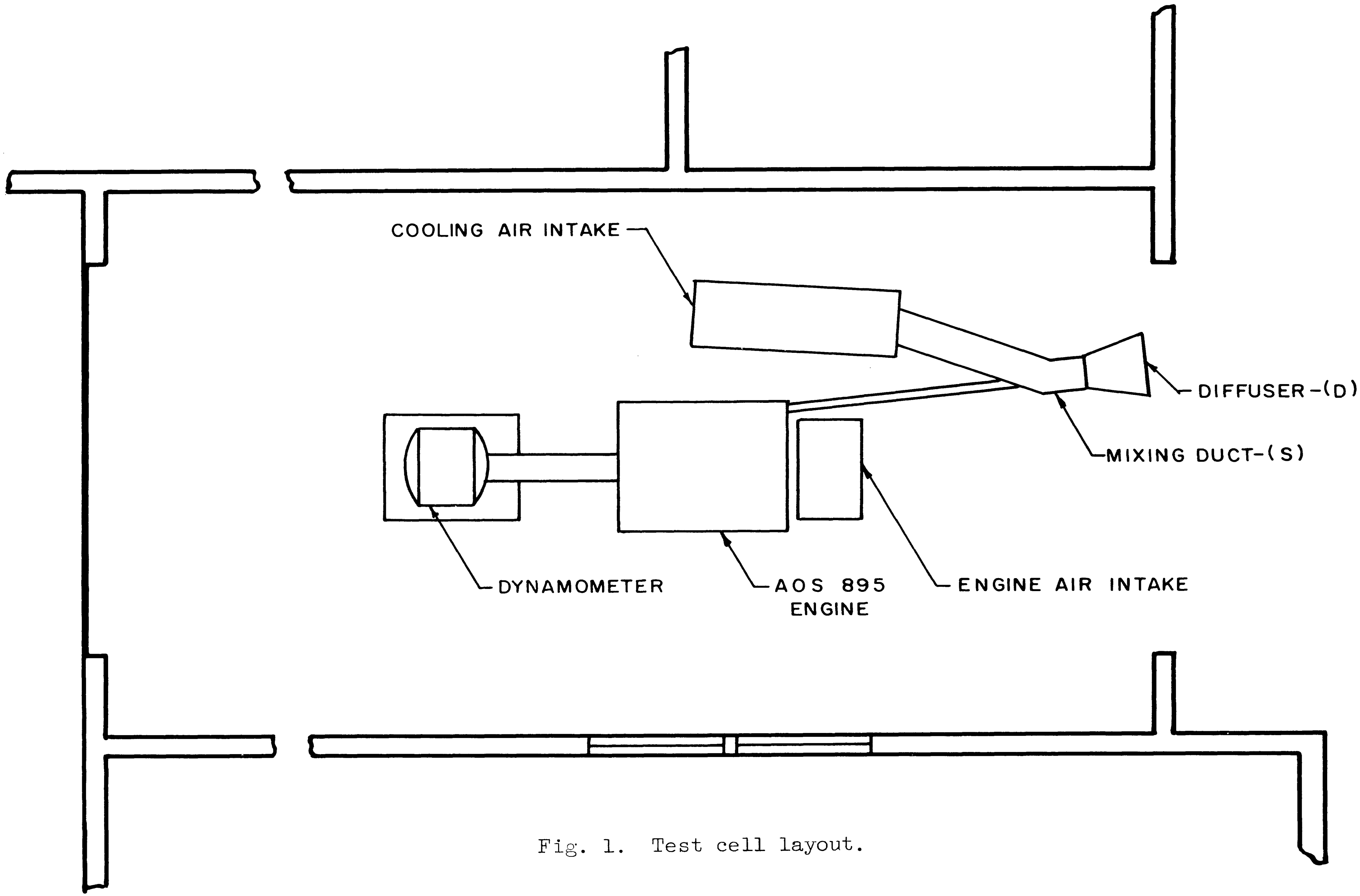


Fig. 1. Test cell layout.

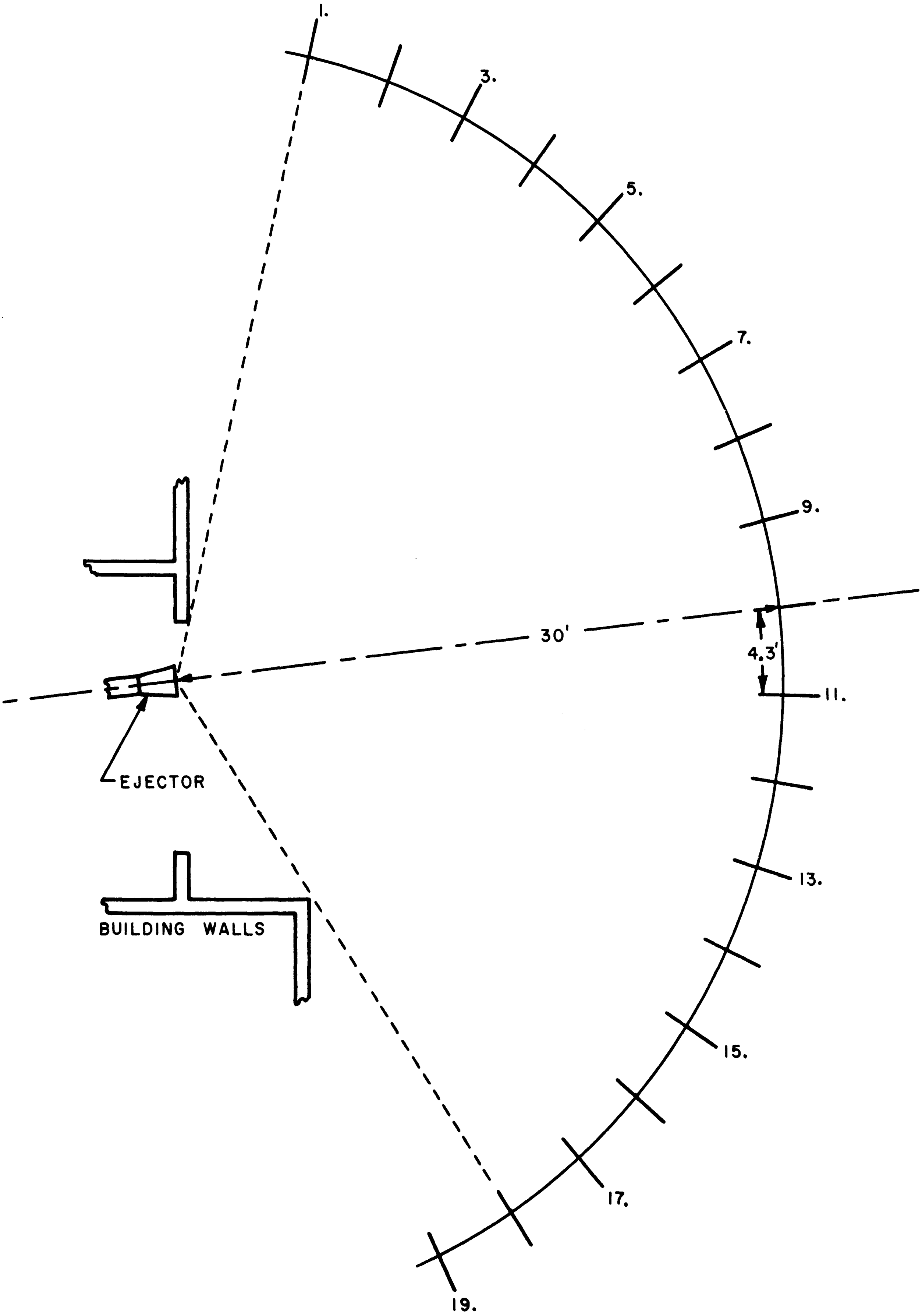


Fig. 2. Microphone locations.

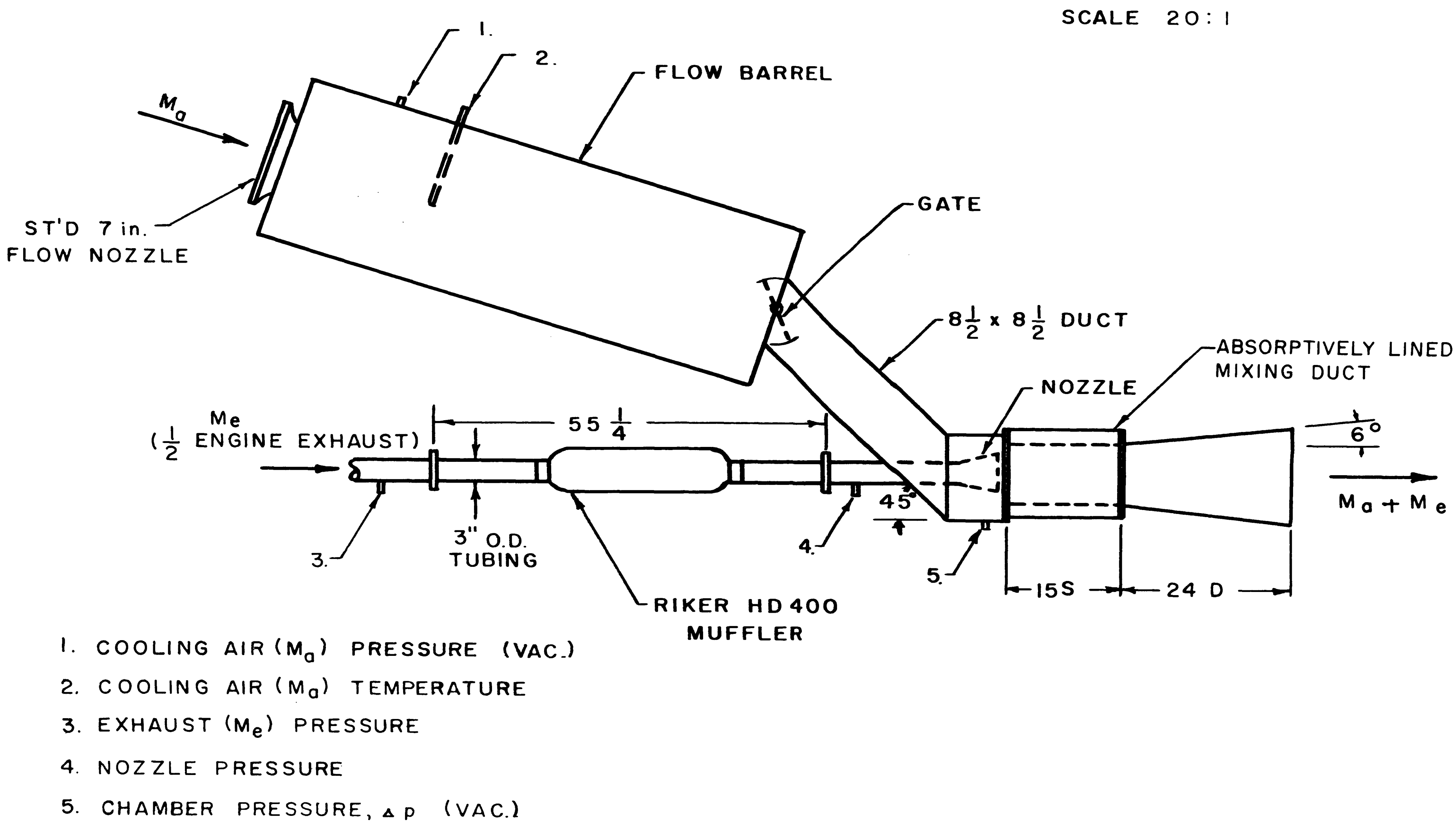
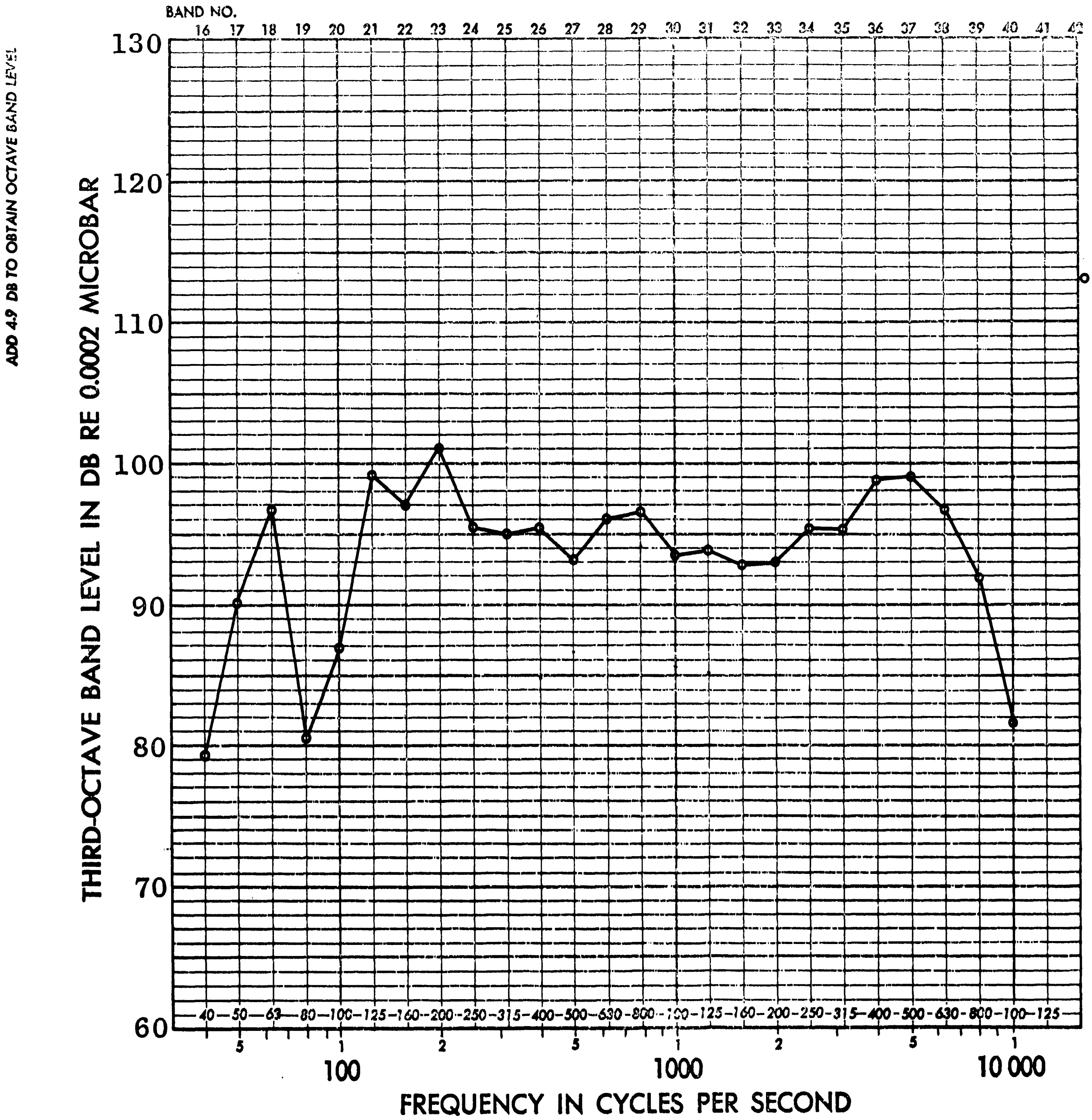


Fig. 3. Top view of ejector with 3 in. x 9 in.—15S + 24D, muffler and sound chamber installed.

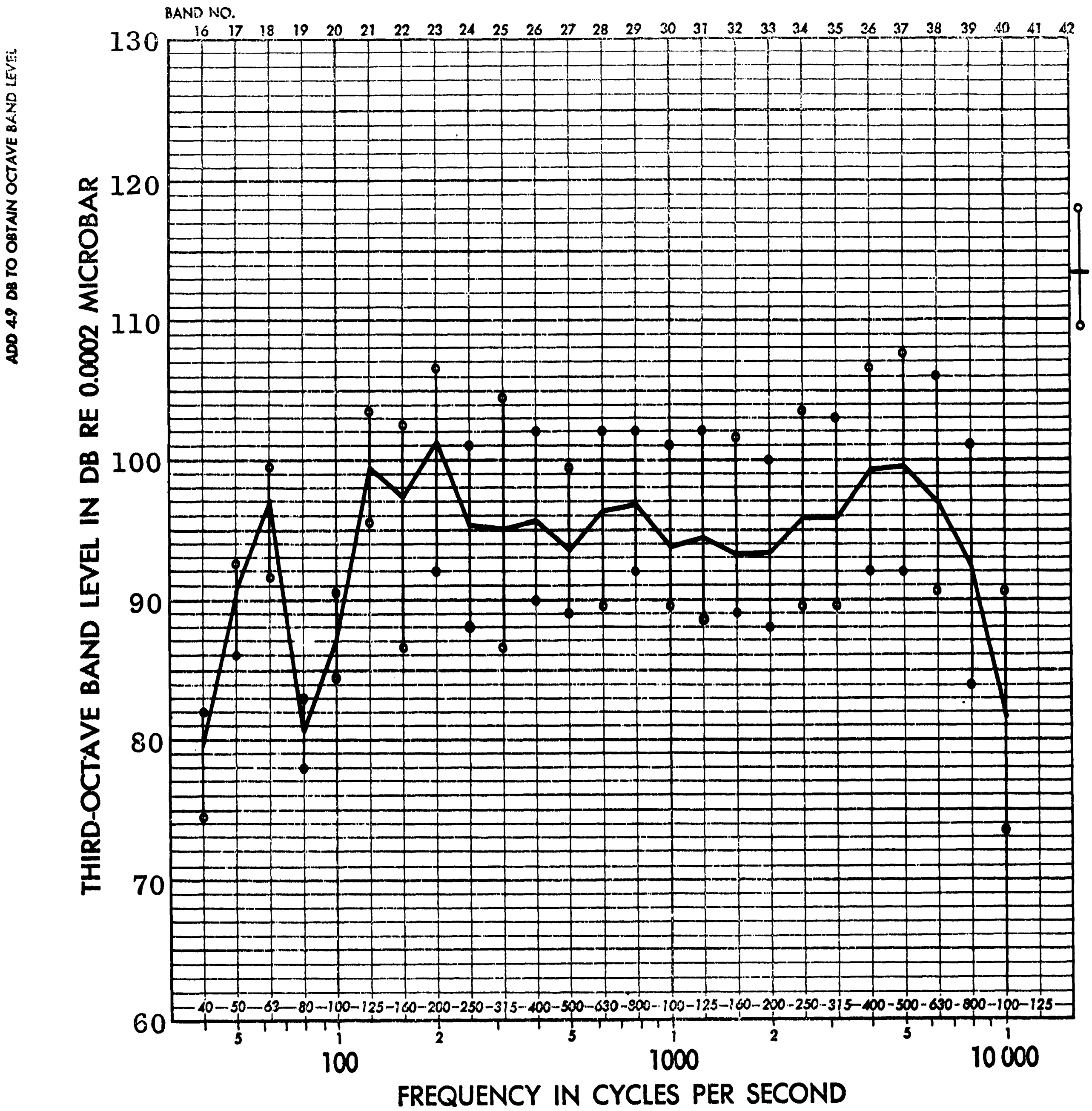


2400 rpm, full throttle, torque 770-785 ft-lb, with ejector, 3 in. x 9 in., 15S + 24D with 2-in. manifold installed. No acoustic treatment installed.

Curve is plot of the average sound-pressure level of 54 microphone positions.

Circle at extreme right indicates average over-all sound-pressure level.

Fig. 4. Noise level versus frequency.

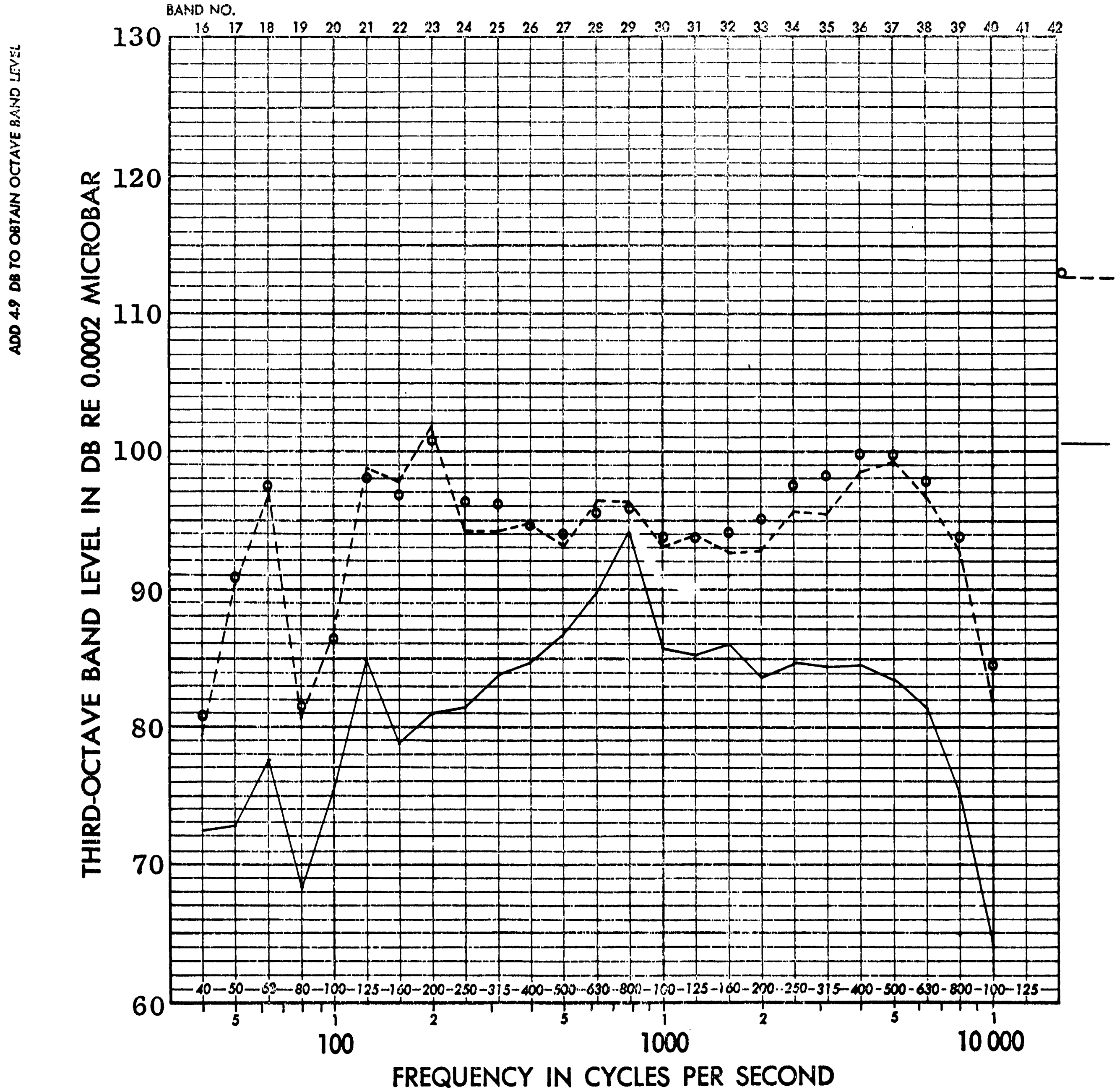


2400 rpm, full throttle with ejector, 3 in. x 9 in., 15S + 24D, 2-in. manifold installed.

This curve is the same as Fig. 4 but also shows the extreme variations in level within the 54 microphone positions compared to average.

Circle at extreme right indicates average over-all sound-pressure level with corresponding extreme variations.

Fig. 5. Noise level versus frequency.



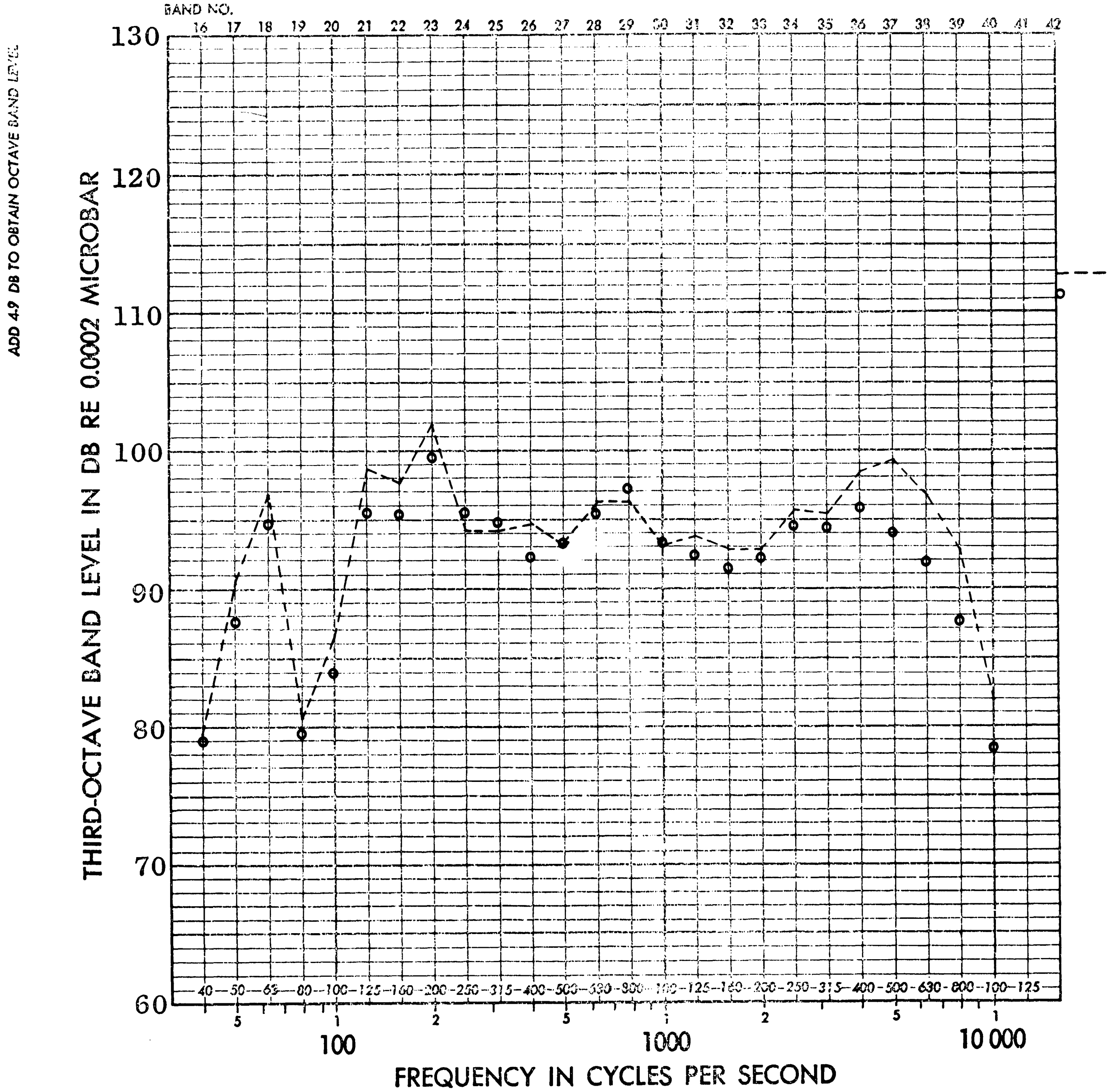
ooooo 2400 rpm, full throttle with ejector, 3 in. x 9 in., 15S + 24D, 2-in. manifold installed. This is a repeat test under same conditions as original survey for the purpose of checking repeatability.

----- Original average of 15 microphone positions extracted from data for Figs. 4 and 5.

——— Minimum noise test without ejector system.

Corresponding average over-all levels shown at extreme right.

Fig. 6. Noise level versus frequency.

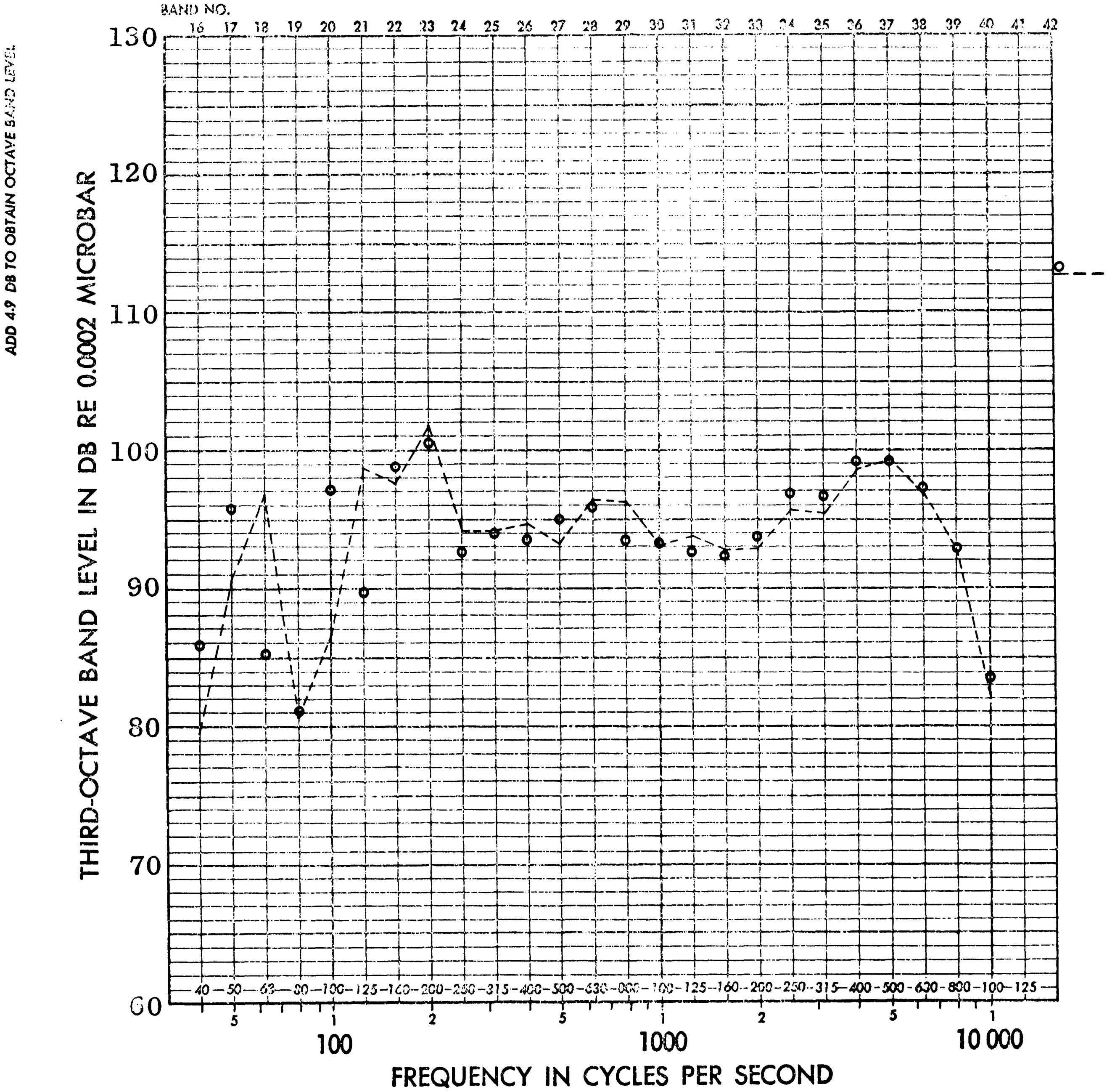


ooooo 2400 rpm, part throttle with ejector, 3 in. x 9 in., 15S + 24D, 2-in. manifold installed.

----- Original average spectrum from Fig. 6.

Corresponding average over-all levels shown at extreme right.

Fig. 7. Noise level versus frequency.

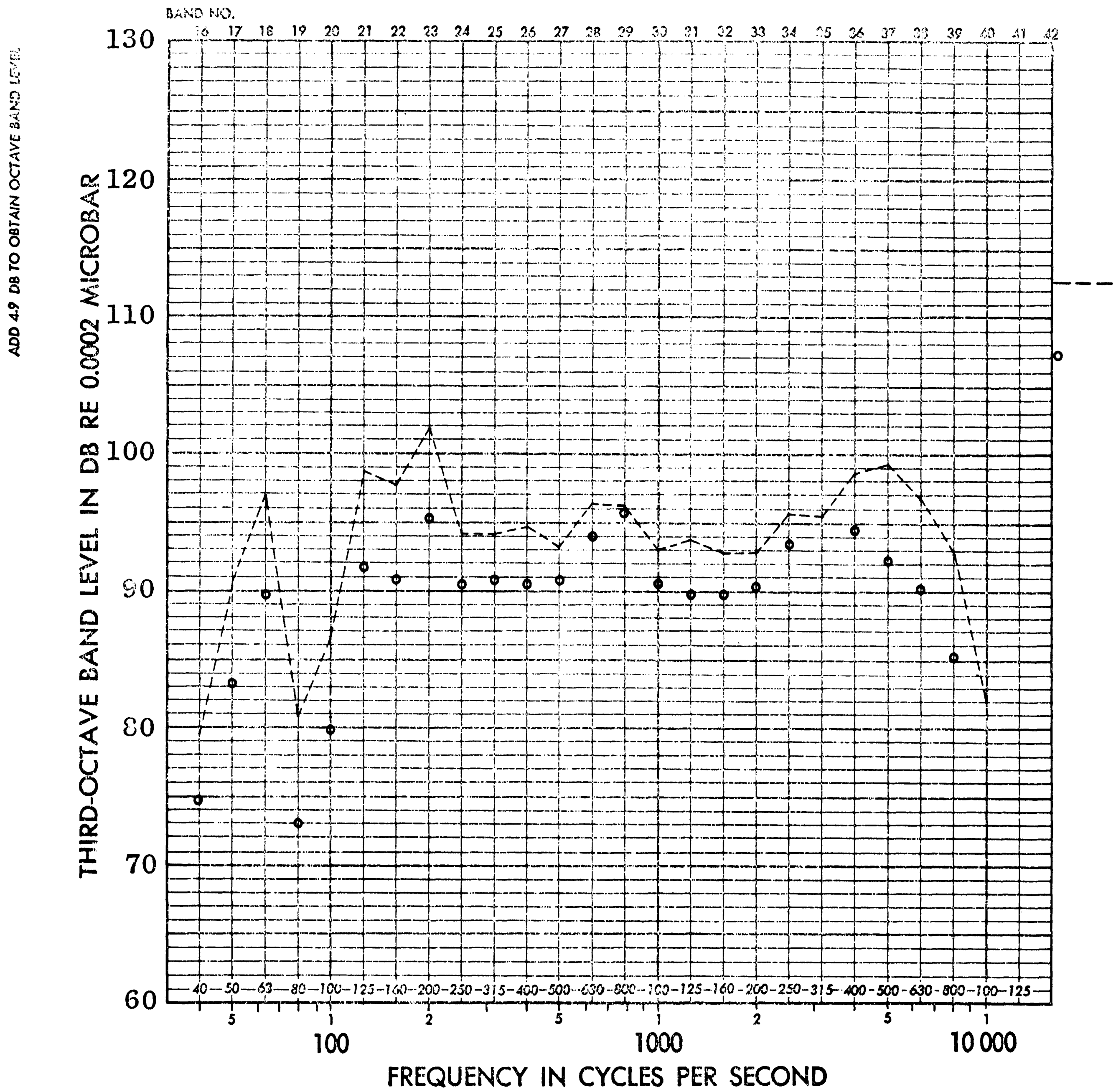


ooooo 2000 rpm, full throttle with ejector, 3 in. x 9 in., 15S + 24D, 2-in. manifold installed.

----- Original average spectrum from Fig. 6.

Corresponding average over-all levels shown at extreme right.

Fig. 8. Noise level versus frequency.

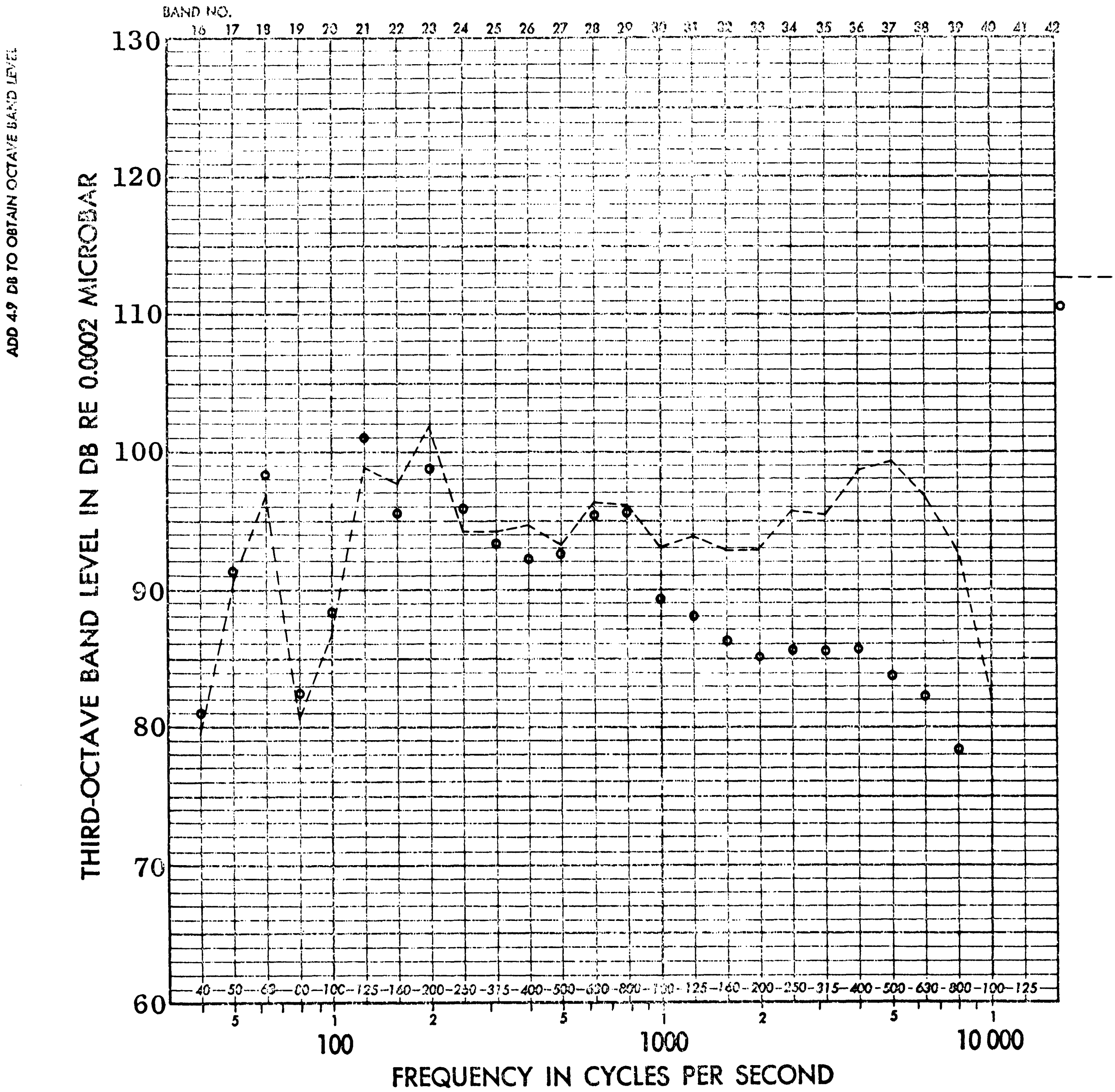


ooooo 2400 rpm, full throttle with ejector, 3 in. x 9 in., 15S + 24D, 2-in. ported Riker muffler installed.

----- Original average spectrum from Fig. 6.

Corresponding average over-all levels shown at extreme right.

Fig 9. Noise level versus frequency.

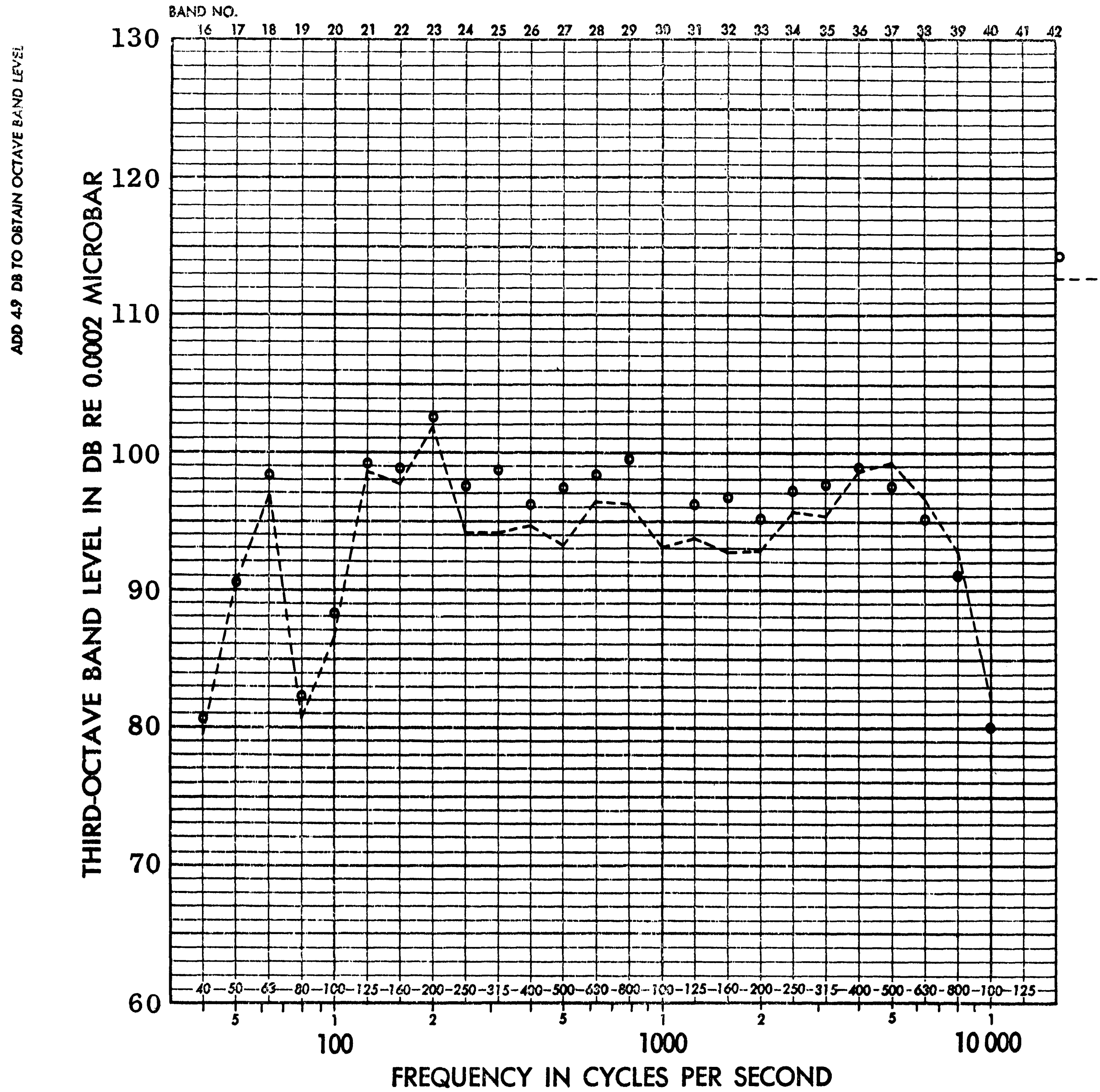


ooooo 2400 rpm, full throttle with ejector, 3 in. x 9 in., 15S + 24D, 2-in. manifold. 15S solid mixing duct surface replaced with 45.5% open area expanded metal and backed by PF 334 Fiberglas.

----- Original average spectrum from Fig. 6.

Corresponding average over-all levels shown at extreme right.

Fig. 10. Noise level versus frequency.

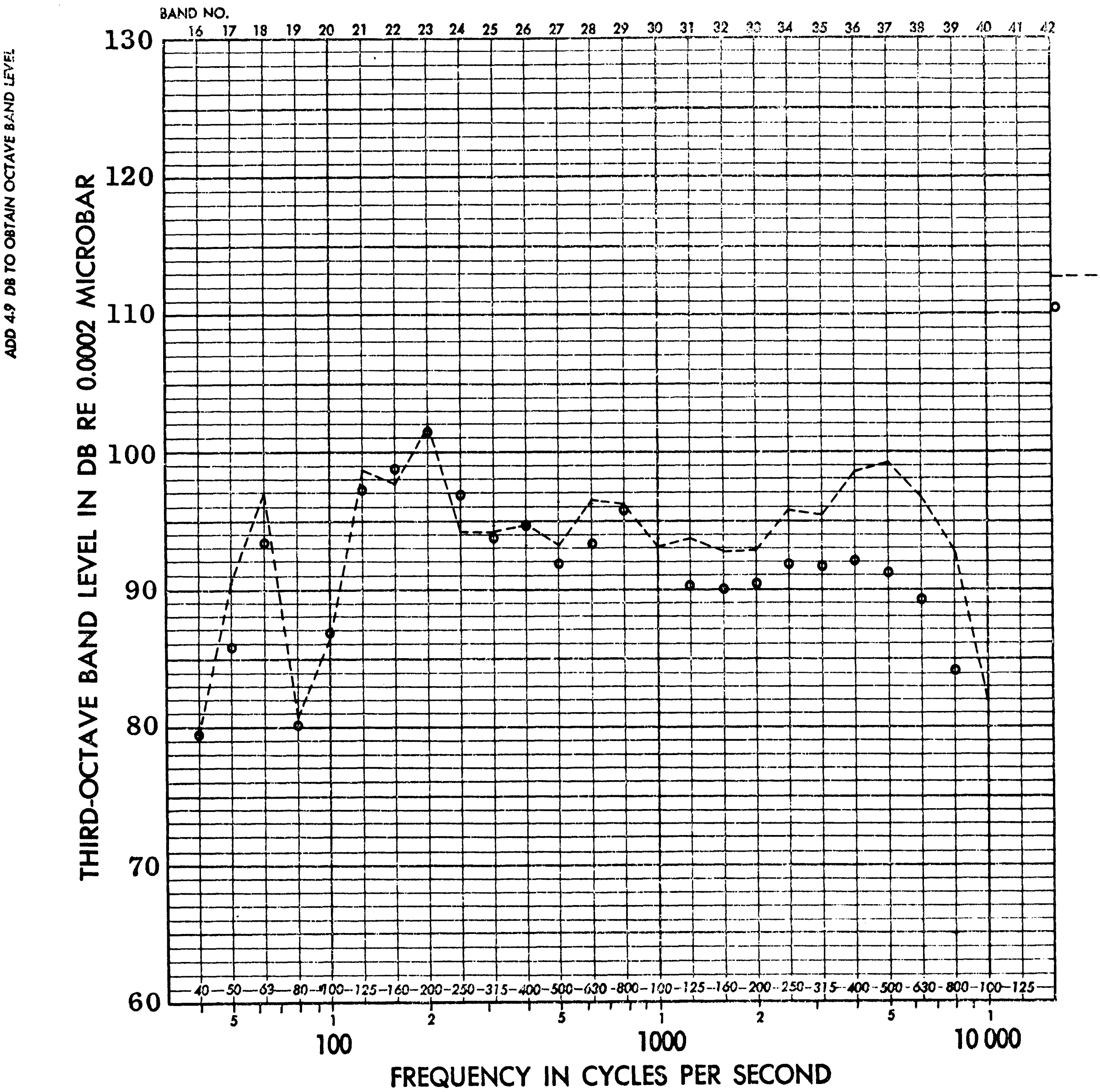


ooooo 2400 rpm, full throttle with ejector, 3 in. x 9 in., 15S + 24D, 3-in. manifold. No acoustic treatment installed.

----- Original average spectrum from Fig. 6.

Corresponding average over-all levels shown at extreme right.

Fig. 11. Noise level versus frequency.

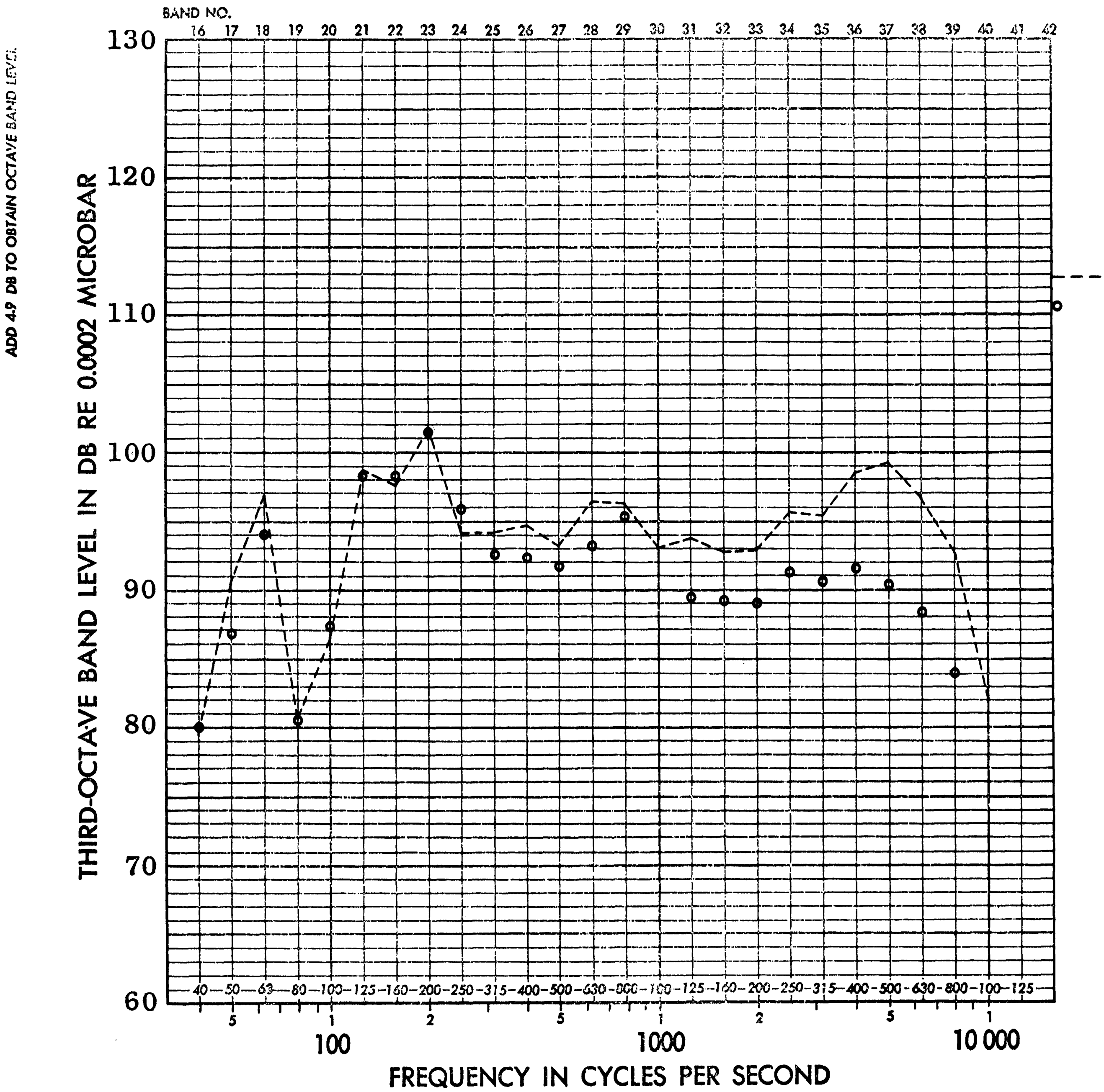


ooooo 2400 rpm, full throttle with ejector, 3 in. x 9 in., 15S + 24D, 3-in. manifold, 3-in. ported Riker muffler installed.

----- Original average spectrum from Fig. 6.

Corresponding average over-all levels shown at extreme right.

Fig. 12. Noise level versus frequency.

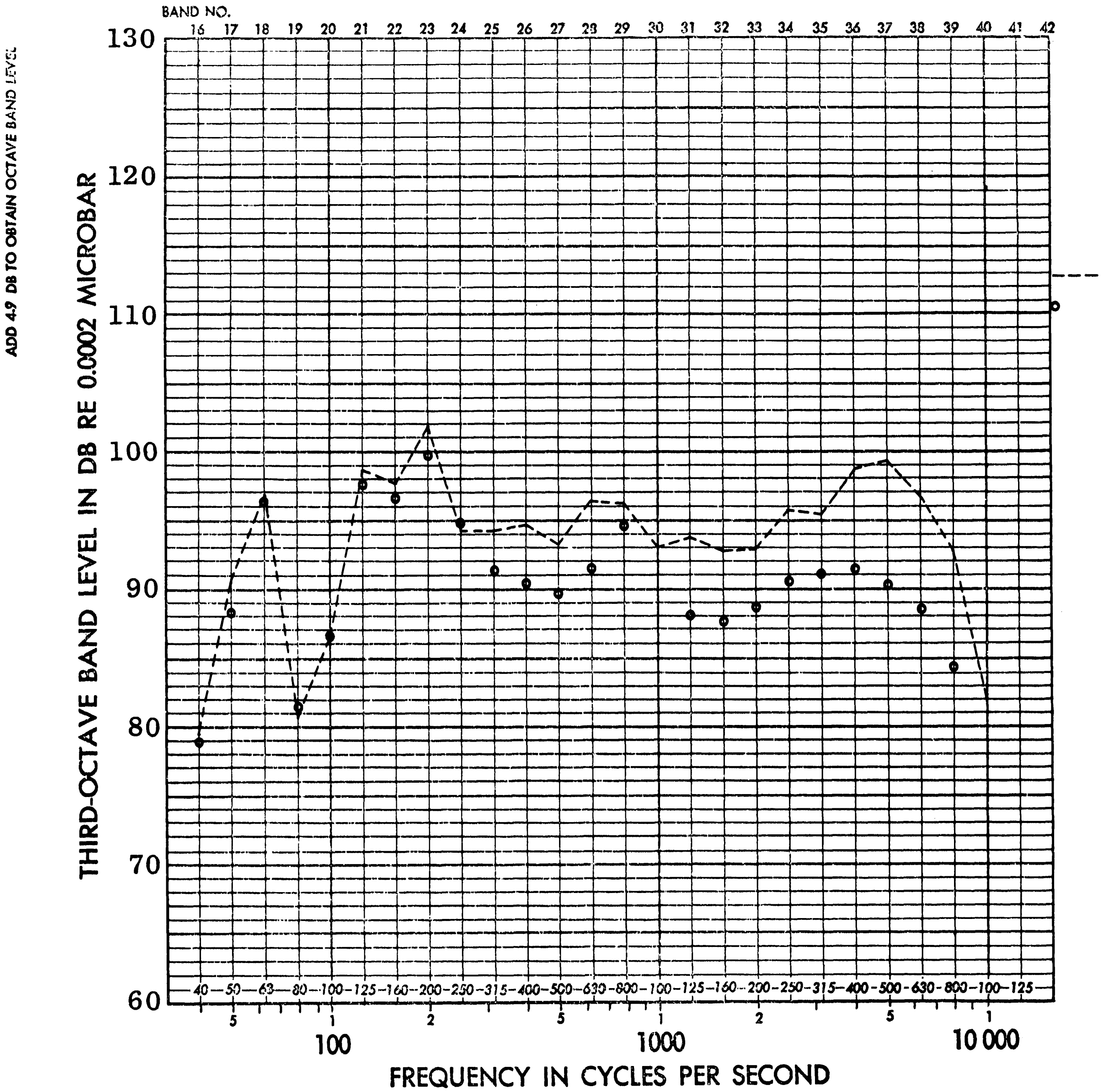


ooooo 2400 rpm, full throttle with ejector, 3 in. x 9 in., 15S + 24D, 3-in. manifold, 3-in. ported Riker muffler, absorptive mixing duct with 60 1/2-in. holes installed.

----- Original average spectrum from Fig. 6.

Corresponding average over-all levels shown at extreme right.

Fig. 13. Noise level versus frequency.

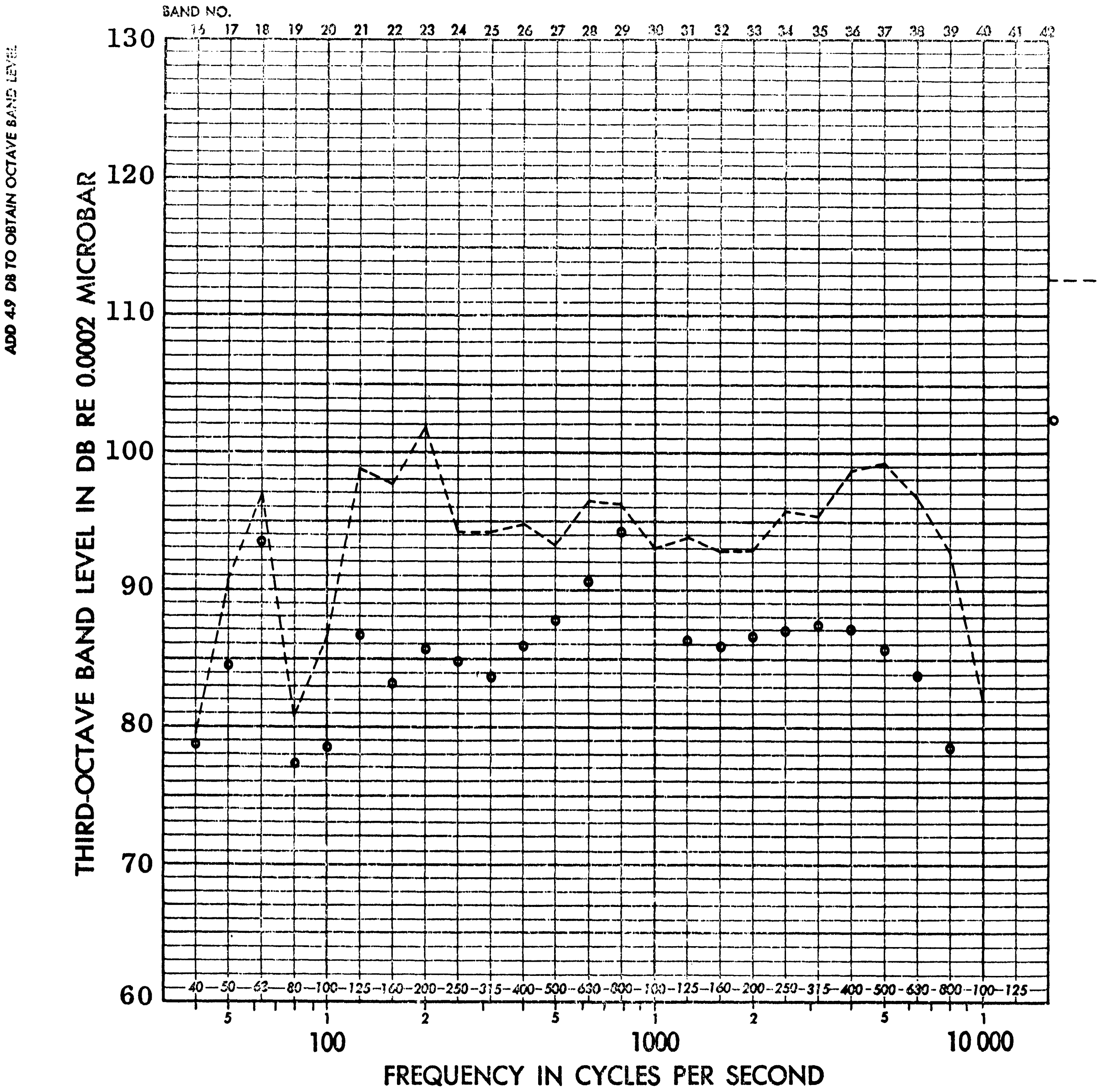


ooooo 2400 rpm, full throttle with ejector, 3 in. x 9 in., 15S + 24D, 3-in. manifold, 3-in. ported Riker muffler, absorptive mixing duct with 110 1/2-in. holes installed.

----- Original average spectrum from Fig. 6.

Corresponding average over-all levels shown at extreme right.

Fig. 14. Noise level versus frequency.



ººººº 2400 rpm, full throttle with ejector, 3 in. x 9 in., 15S + 24D, 3-in. manifold, Maxim Silencer installed.

----- Original average spectrum from Fig. 6.

Corresponding average over-all levels shown at extreme right.

Fig. 15. Noise level versus frequency.

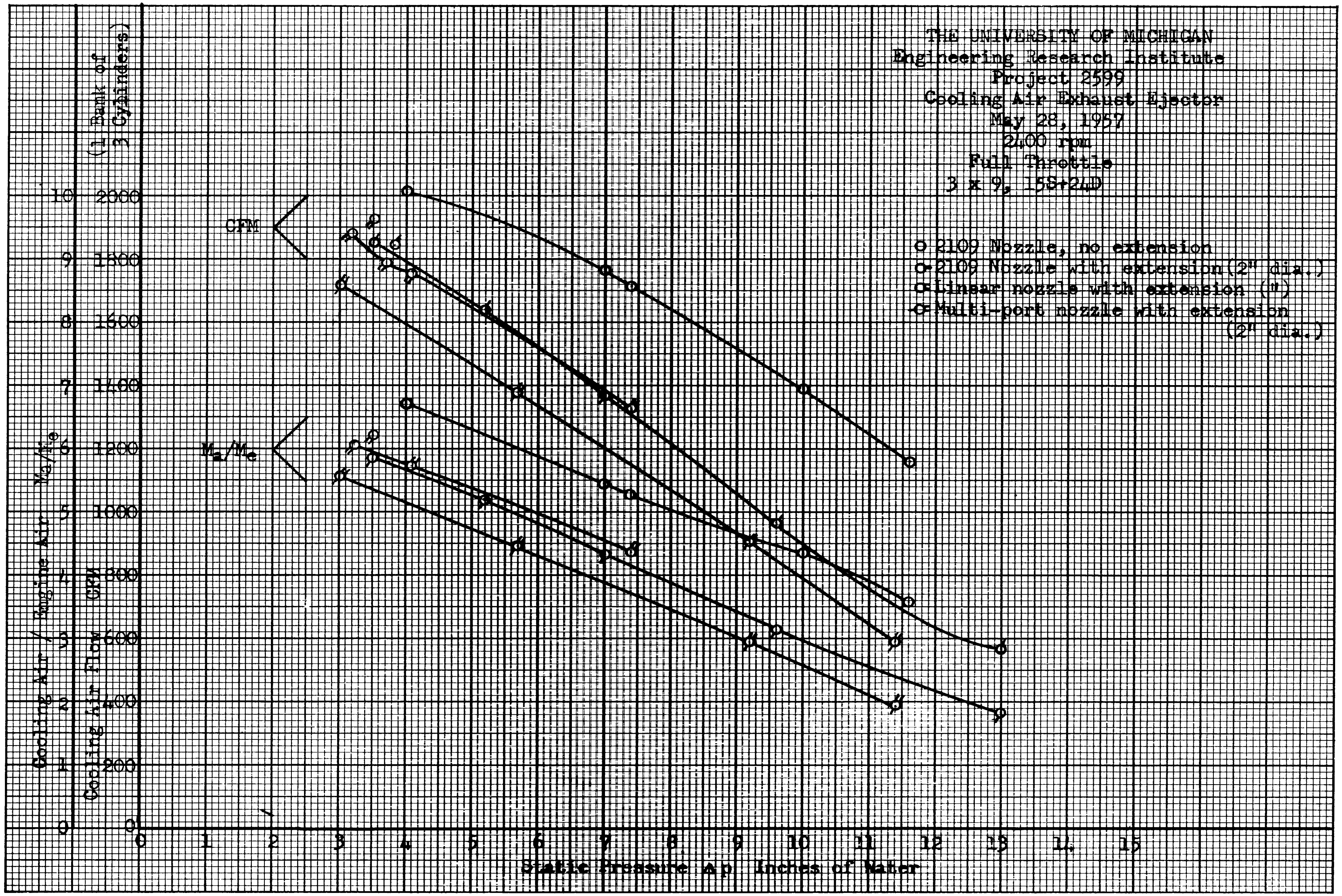
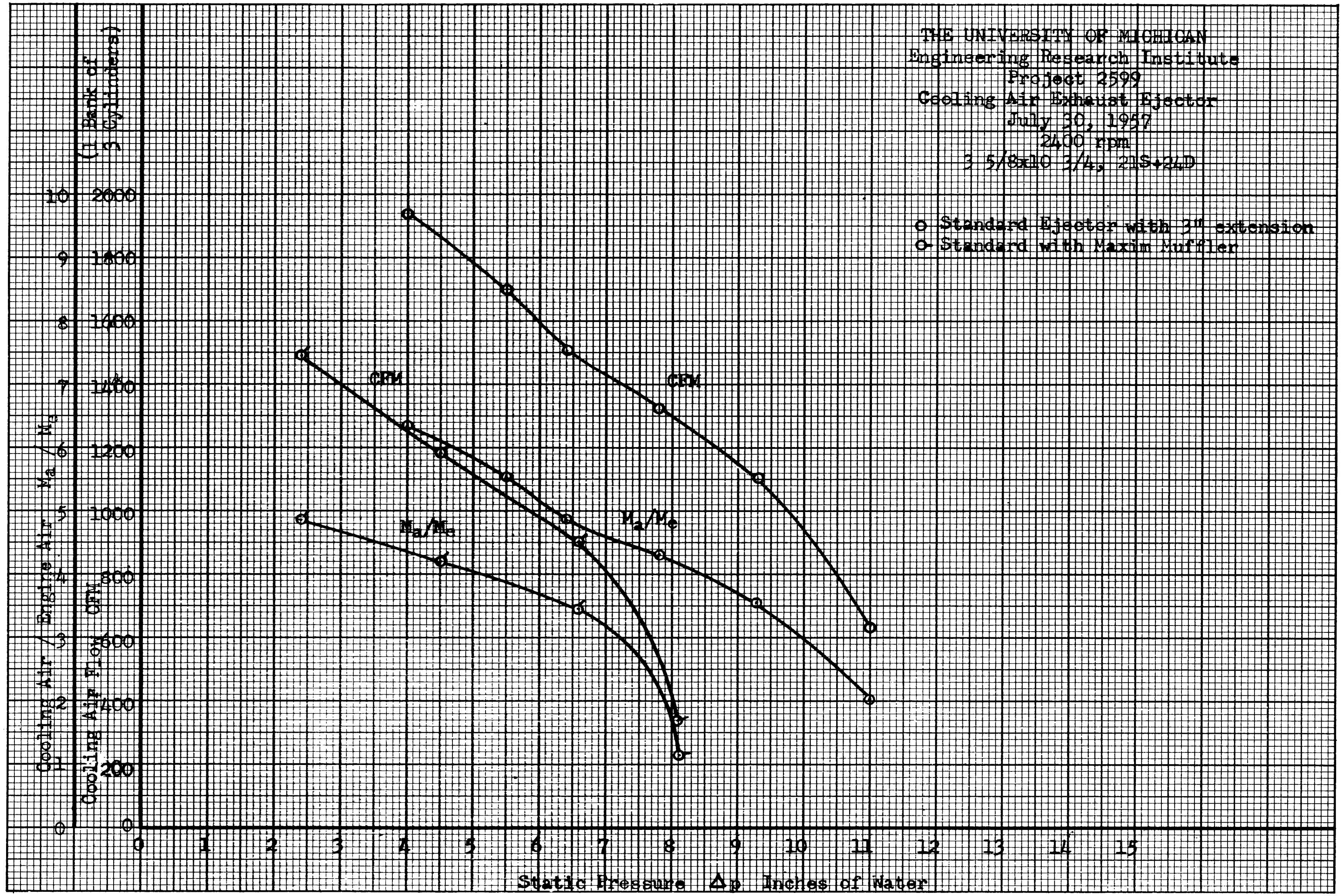

THE UNIVERSITY OF MICHIGAN
Engineering Research Institute
Project 2599
Cooling Air Exhaust Ejector
May 28, 1957
2400 rpm
Full Throttle
3 x 9, 15S+24D
2109 Nozzle, no extension
2109 Nozzle with extension (2" dia.)
Linear nozzle with extension (")
Multi-port nozzle with extension (2" dia.)
CFM
M_a/M_e
Cooling Air / Engine Air M_a/M_e
Cooling Air Flow CFM
(1 Bank of 3 Cylinders)
Static Pressure Δp Inches of Water
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
200
400
600
800
1000
1200
1400
1600
1800
2000

Fig. 16



THE UNIVERSITY OF MICHIGAN
Engineering Research Institute
Project 2599
Cooling Air Exhaust Ejector
July 30, 1957
2400 rpm
3 5/8x10 3/4, 21S+24D
Standard Ejector with 3" extension
Standard with Maxim Muffler
CFM
CFM
M_a/M_e
M_a/M_e
Cooling Air / Engine Air M_a / M_e
Cooling Air Flow CFM
(1 Bank of 3 Cylinders)
Static Pressure Δp Inches of Water
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
200
400
600
800
1000
1200
1400
1600
1800
2000

Fig. 17

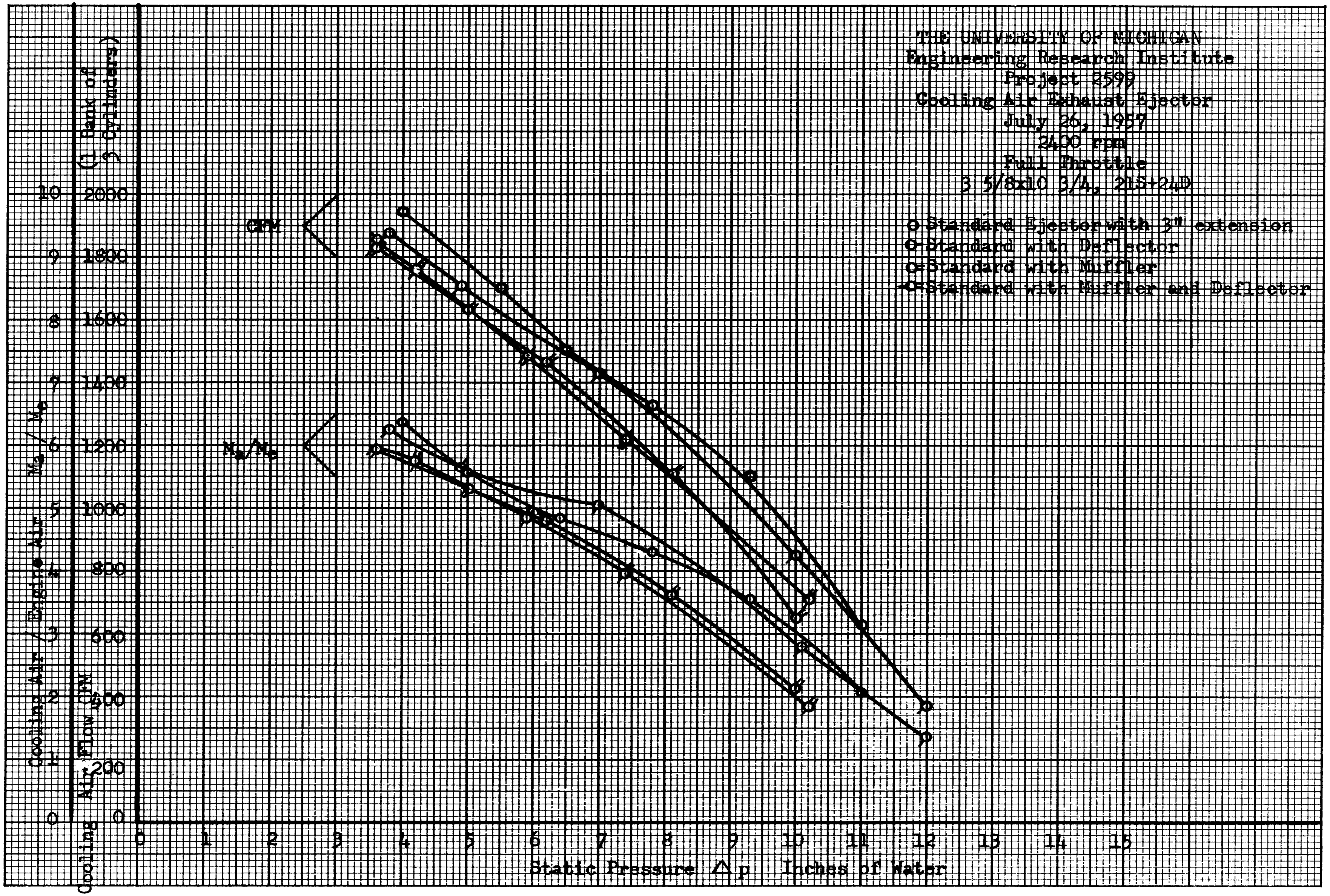

THE UNIVERSITY OF MICHIGAN
Engineering Research Institute
Project 2599
Cooling Air Exhaust Ejector
July 26, 1957
2400 rpm
Full Throttle
3 5/8x10 3/4, 21S+24D
Standard Ejector with 3" extension
Standard with Deflector
Standard with Muffler
Standard with Muffler and Deflector
CFM
M_a/M_e
Cooling Air / Engine Air M_a / M_e
Cooling Air Flow CFM
(1 Bank of 3 Cylinders)
Static Pressure Δp Inches of Water
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
200
400
600
800
1000
1200
1400
1600
1800
2000

Fig. 18

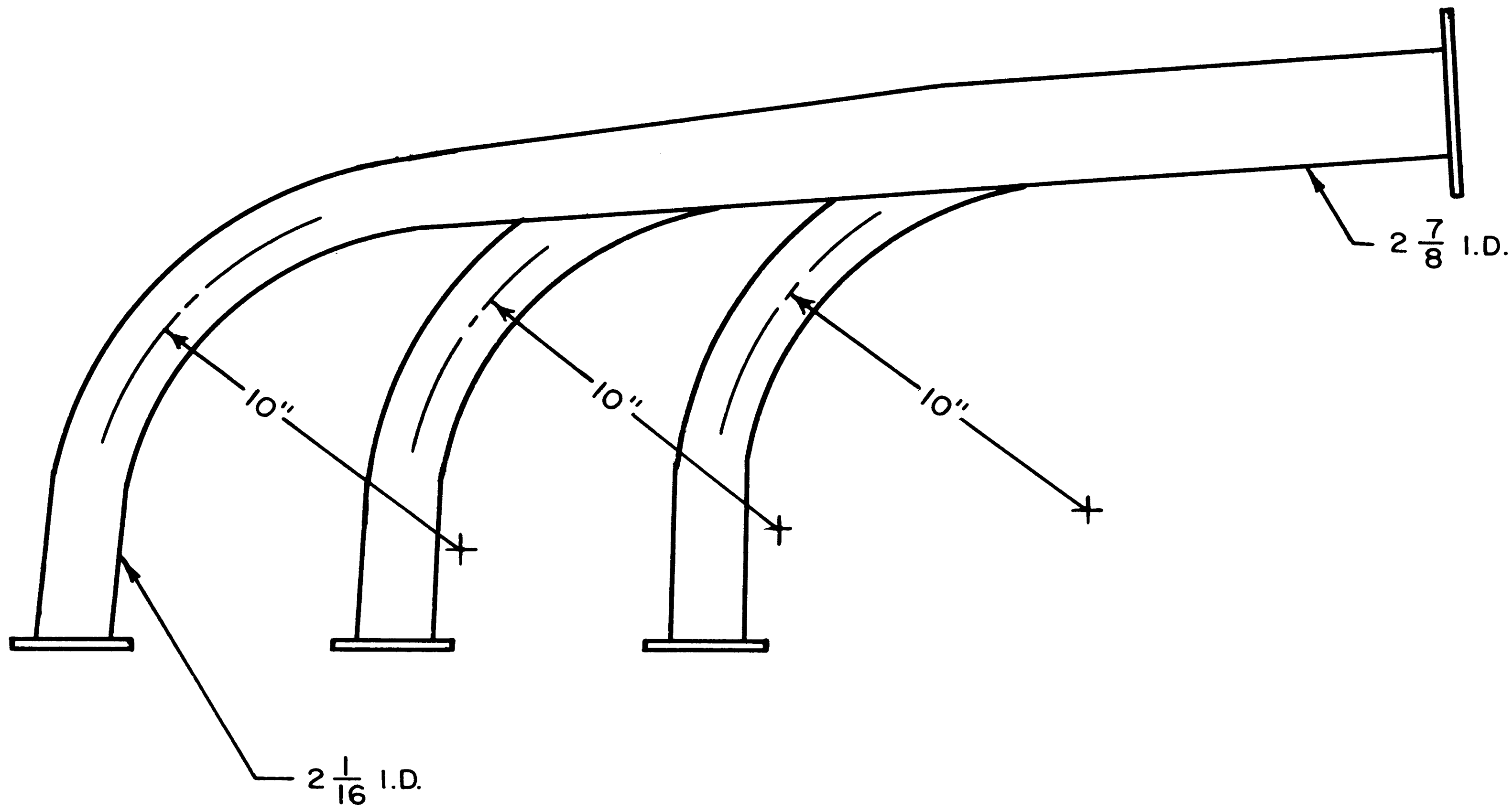


Fig. 19. Three-inch exhaust manifold.

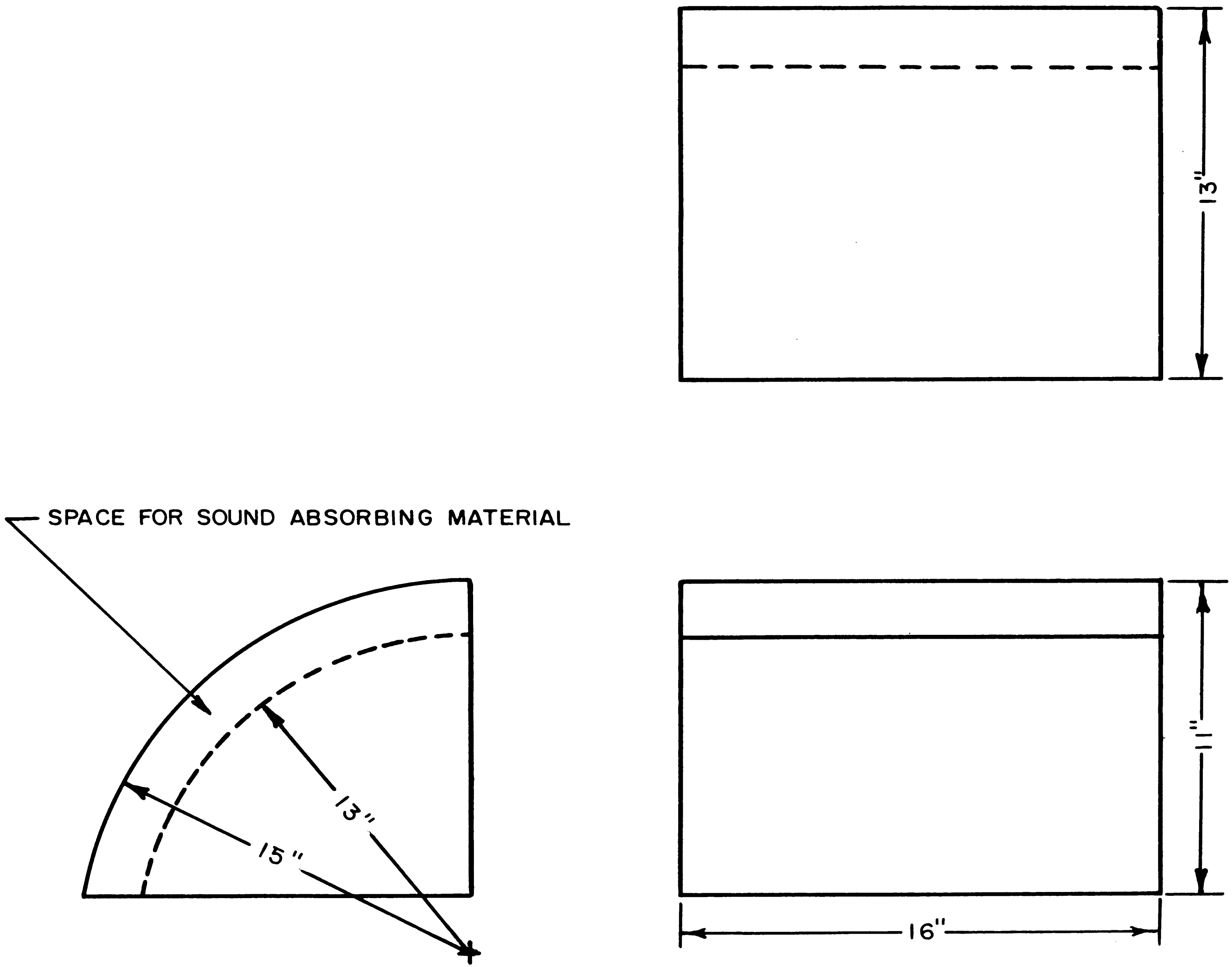


Fig. 20. Hood-type ballistic cover.

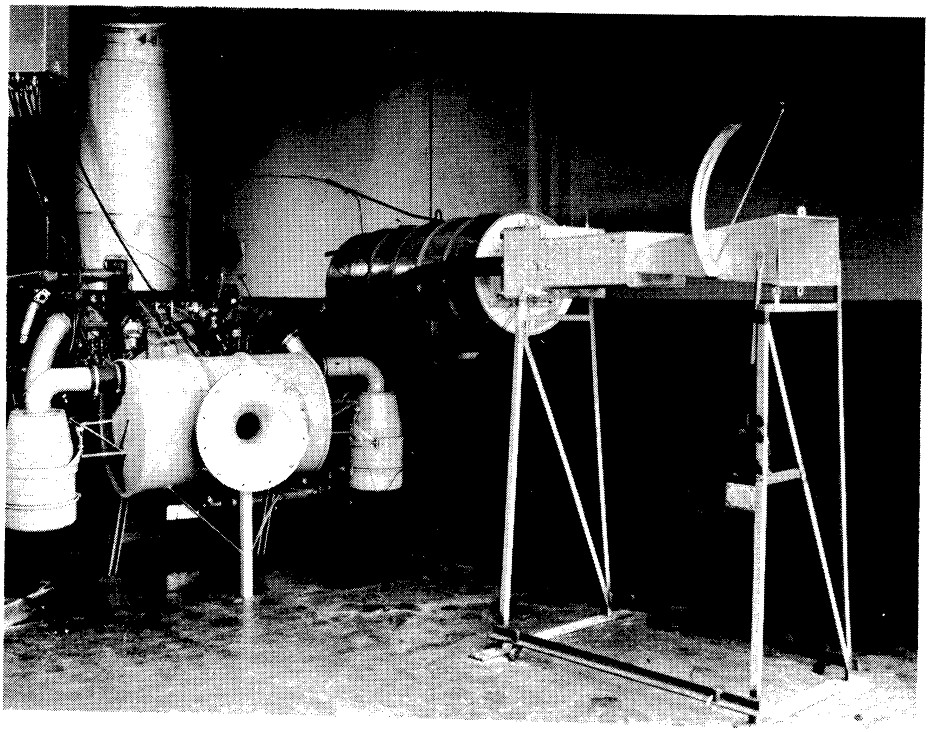
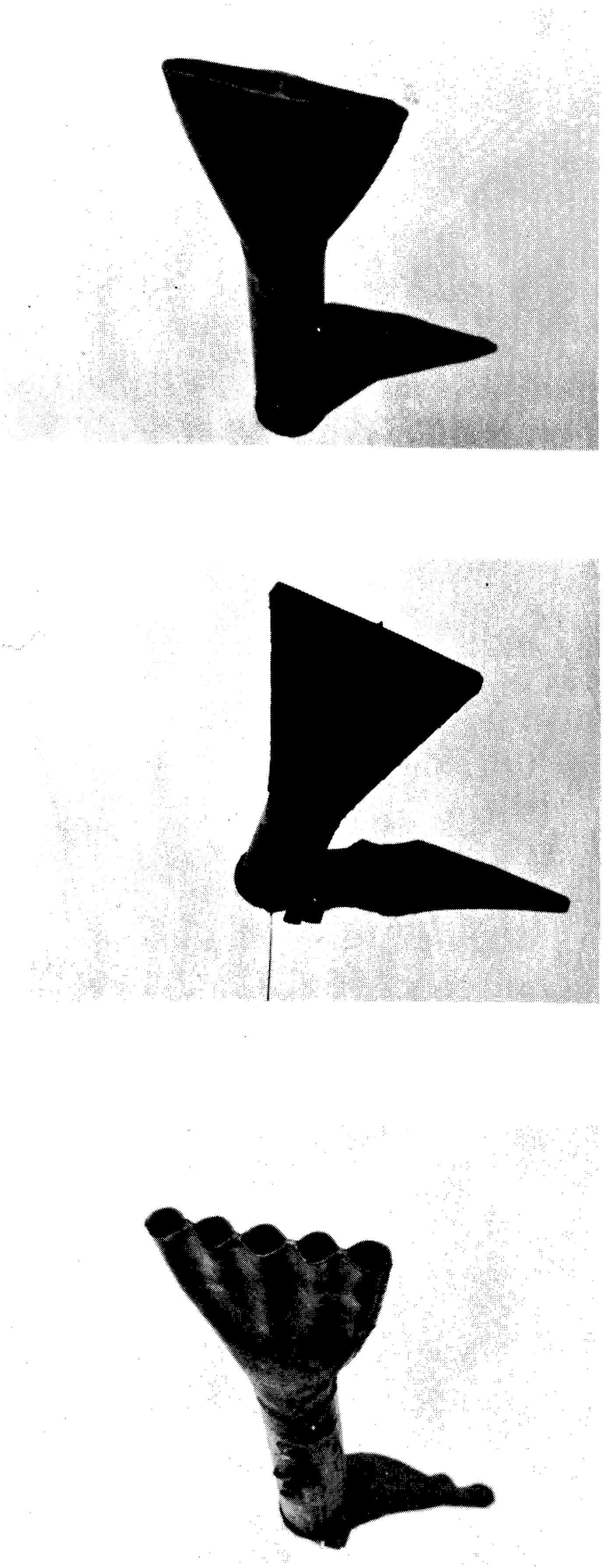


Fig. 21. Front view of ejector test setup.

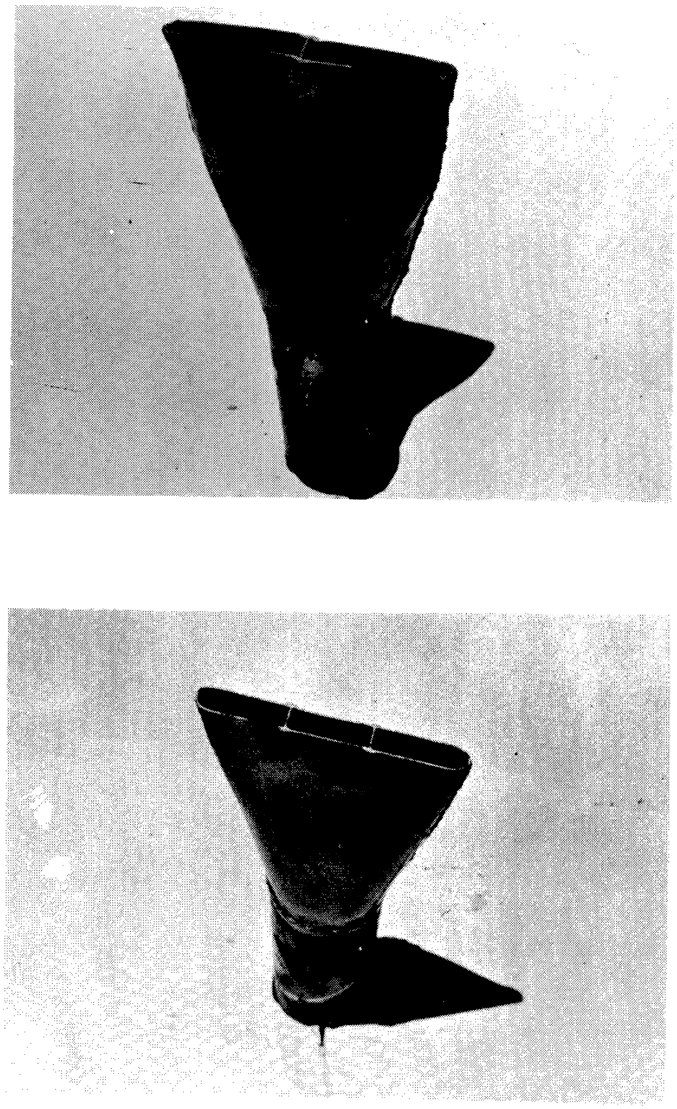


(a)

(b)

(c)

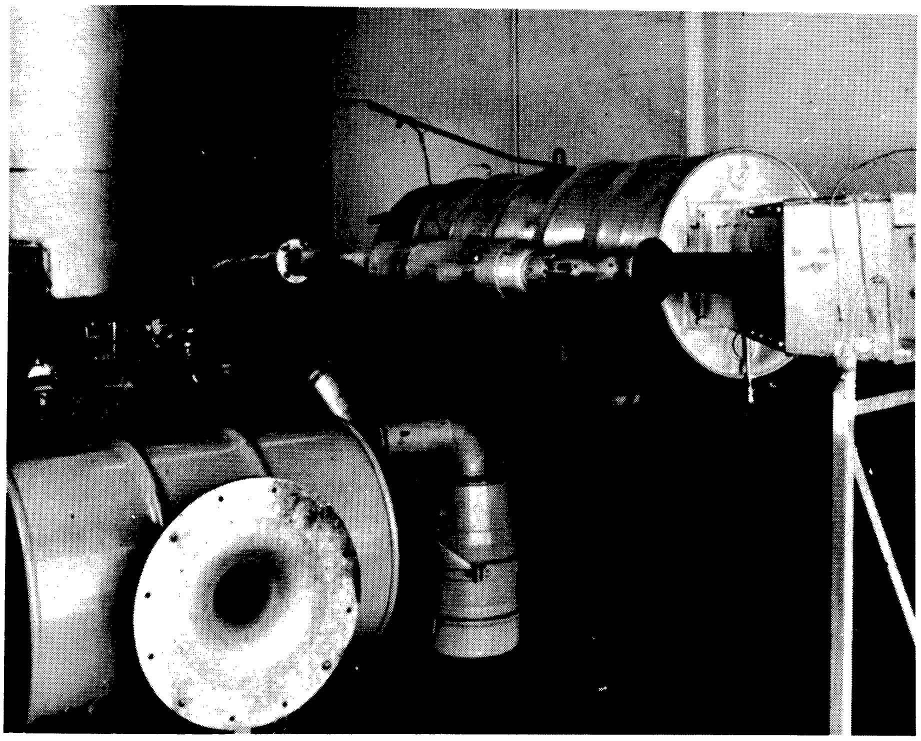
Fig. 22. Two-inch manifold nozzle designs.



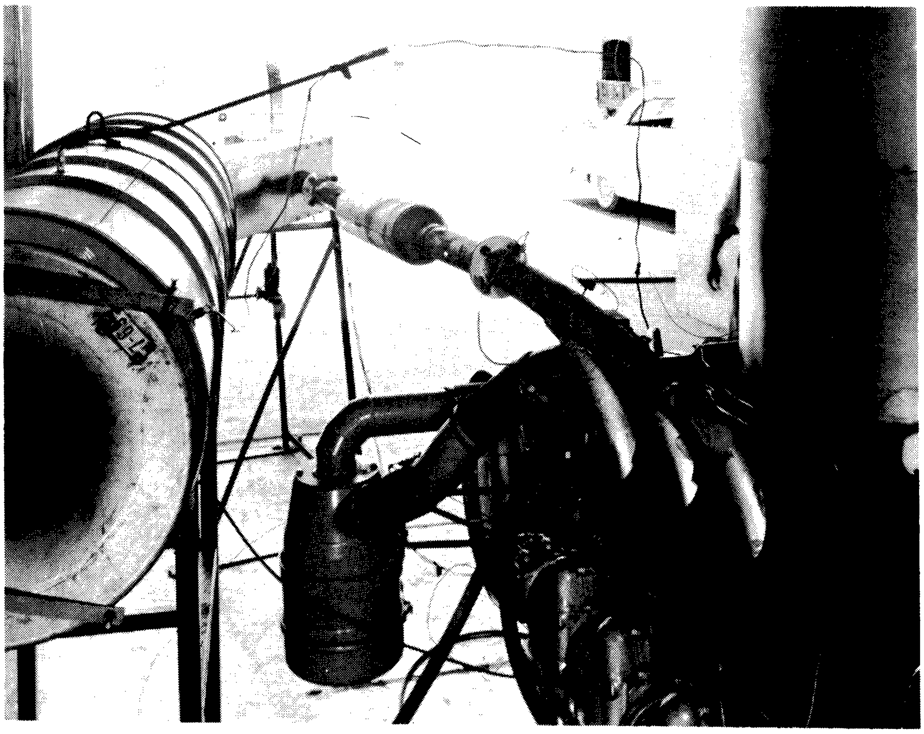
(a)

(b)

Fig. 23. Three-inch manifold nozzle designs.

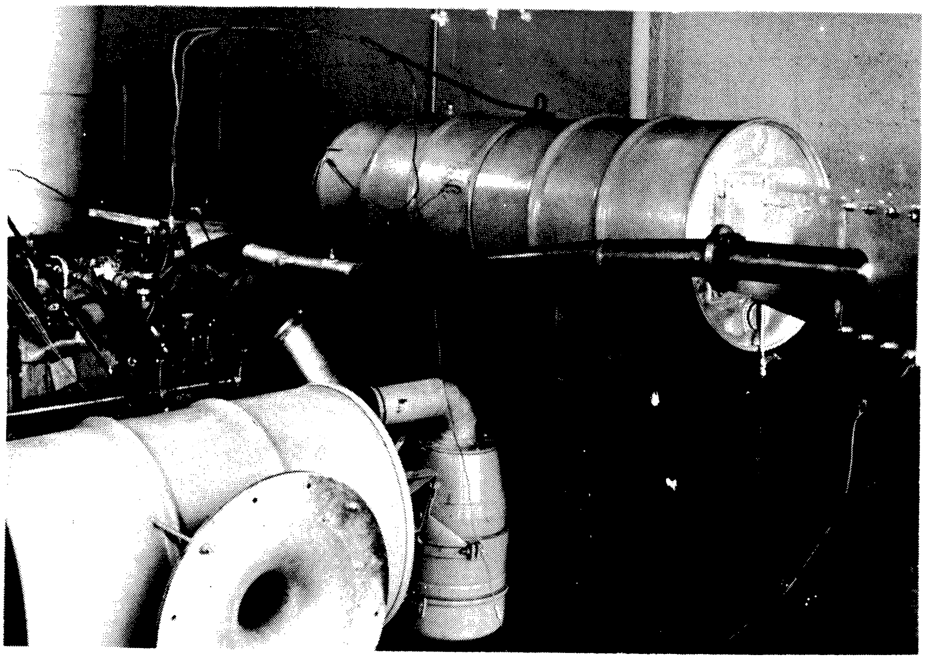


Front View

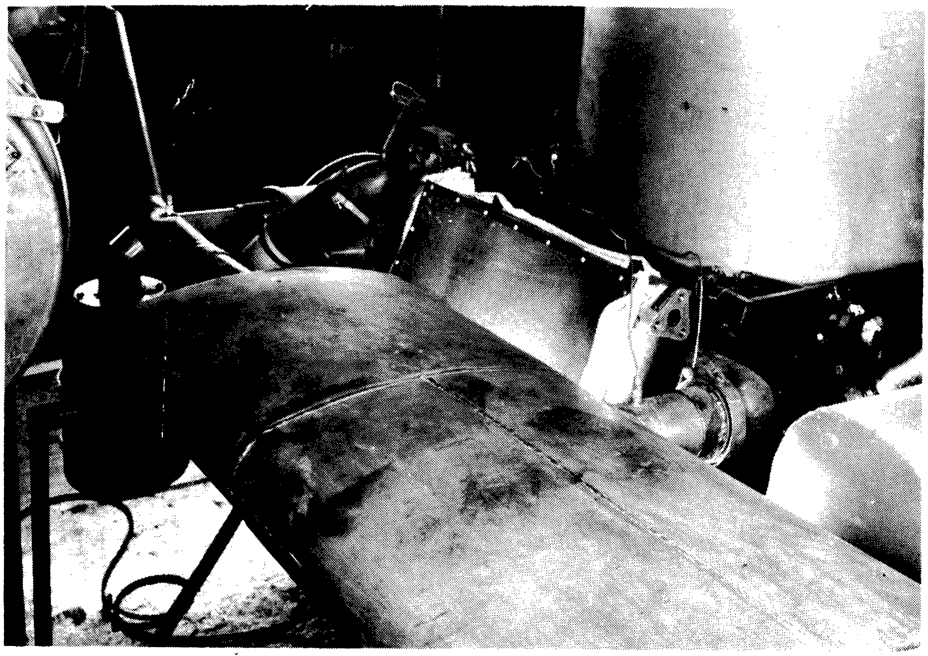


Rear View

Fig. 24. Three-inch manifold system with Riker muffler installed.



Front View



Rear View

Fig. 25. Three-inch manifold with Maxim muffler installed.

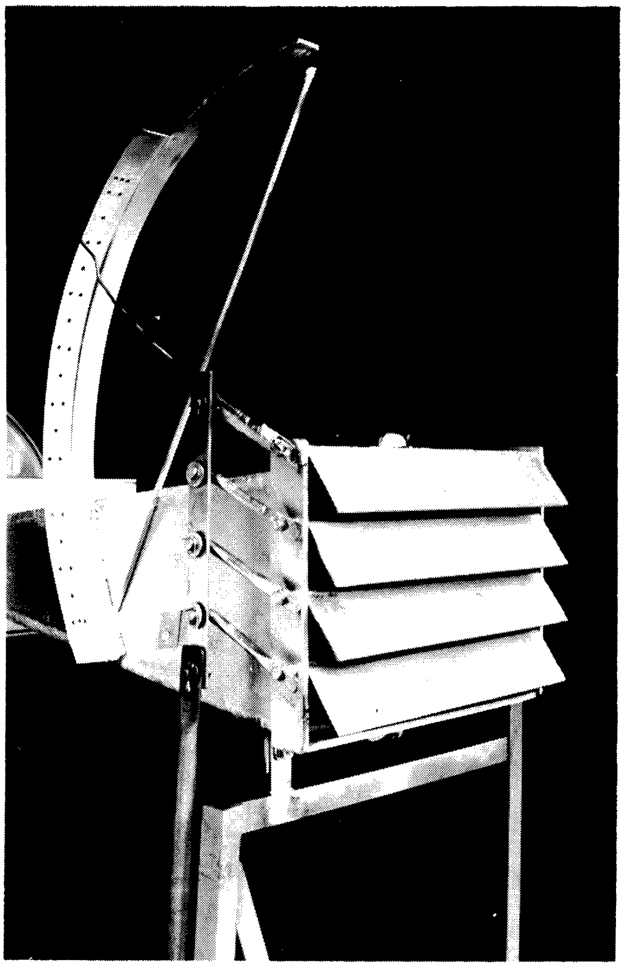


Fig. 26. Multiple louvered grill (right front view).