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A SUMMARY STUDY OF NOISE AND VIBRATIONS IN THE M113 ARMORED INFANTRY CARRIER EQUIPPED WITH STANDARD OR WITH IMPACT ABSORPTION DRIVE SPROCKETS

BY

DR. FRANZ JAKLITSCH, P.E.

MAY 1964

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CONTENT

A. FOREWORD

Noise and vibration pose a major problem in the M113 vehicle. The reduction of noise and wear obtained in the M48 medium tank by using the impact absorption sprocket test interested F.C. in its application for the M113. When these sprockets were installed, Mr. F. Jaklitsch was invited to attend vehicle testing for both versions: standard and impact absorption sprockets. He participated in rides and observed the testing of the sprocket and track test rig. He also had the opportunity to participate in the discussion of current problems of track drive, recent test results and some approaches to solutions for existing noise and vibration problems.

The impact absorption sprockets showed substantial reduction of noise near the sprocket, but at other locations their performance in this regard was strongly affected by other influences. A preliminary analysis of available measurements shows that a careful control of a number of strongly influencial sources of sound level and vibrations in the vehicle is indispensable to achieve the full benefit of flexible elements in the track drive for noise and vibration control.

The following report is based on available test data to analyze the nature and variety of the many suspected sources of noise and vibration in the vehicle. It ranges from design dimensions and features, to operational parameters. This creates the complex picture of sound and vibration whose present excessive level we witnessed in the M113. It is presented as an analysis of a disturbing and irritating behavior in the M113.

Noise and vibration control should be considered in the design stage when it can be accomplished most efficiently and economically. Little indications are available at the present time as to where and how adjustments in dimensions of components may influence auditory and vibrational behavior of military vehicles. Application of light metals might have some influence on the level also. The M113 provides an excellent object of studies to investigate the origins of and to increase know-how for control of noise and vibration in a military tracked vehicle.

Although the immediate identification of the noise and vibration sources might be difficult, it is, however, most essential for improvement of the new armored infantry carrier or other vehicles.

B. OBSERVATIONS AND CONCLUSIONS

The observations and conclusions #1 and 2, were made during the tests on the 13th and 26th of February, 1964 at the FMC Corporation in San Jose, California:

1. Track and sprocket test rig (motorized track simulation)

a. Considerable noise was observed when the speed of the freely rotating track was increased to approximately 450 rpm (corresponding to 30 mph of vehicle speed)

b. At shut-off before complete stop, a few (2-3) hard metallic blows were heard which seem to originate from the sprocket teeth.

c. It was impossible to pull the track by hand over the corners of the rubber polygon tire of the sprocket. The sprocket bounced back when released, and therefore, induced investigation of the sprocket teeth surfaces. It could be observed that the rear surfaces showed metallic contact also. Thus, it can be concluded that the rubber polygon created a spring system which may contribute to resonance resulting in a movement of the track end-connectors back and forth between the sprocket teeth and thus originating the hard hammering blows.

2. Rides in the M113 vehicle on the test track hard surface level road:

a. At reasonably high speed excessive noise was encountered.

b. At a certain noise level, a deep, strong sound suddenly appeared which was easily discernable in spite of defeaning noise in the crew compartment, and the threshold of discomfort was reached. The sensation was irritating but disappeared when the ears were covered. It is assumed that this sound originated at approximately 25 to 30 mph.

c. Origin and location of the deep sound was not discovered since the ride was made without instruments.

d. There are several structural members in the crew compartment subject to strong vibrations: the torsions bars, the support on the back of the crew seats, the rods to close the rear door, etc. One of these, or other vibrating items, may be a source for resonance and, therefore, the origin of the high noise level in the compartment.

3. Results of the analysis:

a. There are several frequency ranges which require special attention designated in figures 19 and following by:

(1) Upper region of V (vibrations)

(2) Region A (deep audible noises)

(3) Lower region of B (lower medium audible range)

b. In the vehicle speed range from 15 to 30 mph the track shoe motion produces a sequence of impulses which correspond mainly to the frequency region of A and lower B.

c. It results from previous measurements that vibrations of the hull were substantially decreased by reducing the track pitch dimensions from 6 to 4.5 inches, as the tests show in Table 1.

TABLE 1

Comparison of Vibration Loads Between 6 and 4.5 Inch Pitch Track Shoes

	VIBRATION LOADS IN G'S Vehicle Speed - MPH				
	10	20	28	32	PITCH
VERTICAL					
A. right front (lifting eye)	.98	1.68	1.86	3.23	6
	.58	1.39	1.54	1.72	4.5
B. right rear (lifting eye)	2.61	2.08	7.18	7.07	6
	1.32	0.5	1.33	2.57	4.5
C. right sponson (outer edge)	6.05	5.79	17.4	23.6	6
	2.43	6.18	7.72	11.7	4.5
TRANSVERSE					
D. right rear (fuel accessory cover)	3.02	4.23	6.67	9.10	6
	1.15	1.80	3.75	6.89	4.5
AVERAGE RATIO OF INCREASE					
A.	1.69	1.19	1.21	1.88	
B.	1.98	4.2	5.4	2.83	
C.	2.50	0.93	2.25	2.02	
D.	2.62	2.33	1.78	1.32	
AVERAGE, OVERALL	2.2	2.16	2.66	2.01	
TOTAL AVERAGE	2.26				

The overall average for the 6 inch pitch track is, therefore, approximately 2.26 times higher than for the $4\frac{1}{2}$ inch pitch track.

C. RECOMMENDATIONS

(Remarks: The recommendations marked by an asterik were made during the visit at FMC, San Jose, California)

*1. Installation of an elastic rear idler permitting radial displacement of rim in relationship to hub (Sketch of concept: See Fig. 1).

*2. Test of circular sprocket tire (replacing the polygonal) for comparative investigation of vibrations which might be created by the polygon tires.

*3. Variation of durometer values for the rubber inserts of sprockets and idlers (range proposed from 40 to 70).

*4. Establishment of the noise level of the vehicle during operation at 2 mph increments from standstill up to 37 mph to establish potential existance of conditions of resonance. Pick-up at the left and right sprockets, the drivers seat and the cargo area.

*5. Identification of the frequency level of the deep sound observed at 29 to 31 mph in the vehicle cargo area and simultaneously experienced threshold of discomfort. After this frequency is established, checking of the frequencies of vibrating items in the crew and driver compartment to identify a potential source of this irritating deep sound.

*6. Three-dimensional (vertical, longitudinal and transverse) representation of response of rear idler assembly to forced vibration of M113 vehicle based on the test results of 9 August 1961 (Report ORO 673, p.2) to show orientation of vehicle acceleration in space.

*7. Investigation of the surface of the sprocket teeth (front and rear) to prove the existance of a hammering movement of the end connectors between the pairs of teeth.

*8. Taking of motion pictures of the behavior of the tracks (left and right) to establish the potential existance and kinds of standing waves in the tracks. Since length and weight of the tracks are different, both sides should be investigated.

9. Establishment of the natural frequency of the right and left track.

10. Determination of the influence of the sag of the tracks on the natural frequency.

11. Establishment of the natural frequencies (fundamental harmonics) for the following components of the vehicle or the test equipment:

a. Engine

b. Track, frequency of engagement or disengagement of track shoes

c. Track test rig

d. Fan

e. In the crew compartment:

(1) Back support of crew seats

(2) Torsion bars

(3) Rods to close the back door

12. Determination of the relevant frequencies when investigating accelerations and amplitudes of vibrations. Identification of sources.

13. Inclusion of subsonic region into investigation of acoustic and vibrational behavior.

14. Optimization of operating conditions of impact absorption sprockets by variation of durometer hardness and thickness of inserts. Identification and reduction of elimination of impairing influences inside the vehicle on the reduced sprocket noise levels.

15. Adjustment to flexible inserts at sprocket and idler.

16. Emphasize investigation of vibration amplitudes in frequency regions V and A.

D. INFLUENCES ON THE NOISE INTENSITY OF THE M113

The results of a noise test dated December 10, 1963 versus frequency and vehicle speed at the three locations: Near left sprocket, near driver and in cargo area were analyzed to investigate the effectiveness of the use of impact absorption sprockets for reduction of noise (see Attachment #1).

The noise level for constant frequency levels was plotted versus vehicle speed as shown in the figures:

a. Nr. 2 through 4 (near sprocket)

b. Nr. 5 through 7 (near driver)

c. Nr. 8 through 10 (in cargo area)

In a simplified form, the noise level i can be presented as follows:

$$i = i_0 + V \cdot \tan\alpha \qquad (1)$$

where i is the noise level (db) at the vehicle speed V (mph):

i_0 = is the noise level extrapolated for V = 0 mph (Standing vehicle)

$\tan\alpha$ = is the slope of the increase of i versus vehicle speed V

$$= \frac{i_{30} - i_0}{30}$$

This approach is supported by the shape of the curves in Fig. 3 (for 425, 850 and 1700 cps for the impact absorption sprockets), in Fig. 3 for 425 cps and in Fig. 4 for 6800 cps it is valid also for the standard sprockets. At lower frequencies, some deviations become apparent which may be caused by overriding influences.

There are some differences in the manner by which the vehicle speed effects intensity and frequency of noise at the various investigated locations on the vehicle. Although the general pattern expressed by equation (1) is valid, there is an obvious indirect influence of the speed on the noise. Another influence is the type of sprocket installed.

In order to show the variation in trend and level of the noise, the changes of the initial noise level i_0 (as extrapolated for $V = 0$ mph) and the increment Δi with increasing vehicle speed are shown in the following figures:

a. Figs. 11 to 13 (Rate of noise intensity near sprocket)

b. Fig. 14 (Rate of noise intensity near driver)

c. Figs. 15 to 17 (Rate of noise intensity in cargo area)

1. <u>Near Sprocket</u>

The noise level is shown versus vehicle speed in Fig. 2 to 4 and the result of analysis versus frequency in Fig. 11 for the standard sprocket, and in Fig. 12 for the impact absorption sprocket. An obvious reduction in noise intensity was obtained by the flexible sprockets.

The initial noise level i_0 for a standing vehicle is identical in slope for both sprockets but differs in level in favor the impact absorption sprocket (Fig. 13).

The rate of the noise level Δi shows an increasing difference for higher frequencies.

All figures for the low region of octave bands 53 and 106 cps i.e. Figs. 2, 5 and 8 show a superposed disturbance at the lower vehicle speed. The same phenomenon will be found in the discussion of the vibrational behavior of the vehicle.

2. <u>Near Driver</u>

Figs. 5 to 7 show the behavior of the noise versus vehicle speed near the driver for both sprockets and Fig. 14 gives the comparison.

Comparing with Fig. 13, it becomes evident that level and trend of the initial noise level i_0 as well as of the rate of increase

$$\Delta i = i_{30} - i_0$$

is different from the situation near the sprocket and from that in the cargo area.

In the frequency range up to 1000 cps the rate of noise Δi is higher for the impact absorption sprocket but beyond 1000 cps it is identical for both types. The initial noise i_0 for a standing vehicle shows a stronger decrease in the higher frequency range as compared to near the sprocket or the cargo area. The standard sprocket shows an evident change in slope at 1000 cps, but the numerical difference in the order of magnitude is rather small.

3. Cargo Area

Figs. 8 to 10 show level and trend of these values for the two types of sprockets. In both cases, there is a slight reduction of the initial noise level i_0 and the rate of increase Δi with increasing frequency f (Figs. 15 and 16). A comparison between standard and impact absorption sprockets is shown in Fig. 17.

The rate of increase with increasing vehicle speed is nearly the same for both sprockets. Although the initial noise level for a standing vehicle shows lower values, the slope is identical for the impact absorption sprocket. The increment of noise by the vehicle speed shows a divergent behavior: it is decreasing from a higher level for the impact absorption sprocket and increasing for the standard sprocket versus frequency.

E. ANALYSIS OF VEHICLE LOUDNESS

The perceived subjective sensation on the ear is not expressed by the intensity level (decibel) but by the loudness level (phon) or the loudness proper (sone) respectively. While the intensity level by its definition is based on a logarithmic scale, the loudness is based on a linear scale and, hence, makes comparison easier.

The rate of loudness (sone per decibel) varies in wide limits with the decibel level (See Fig. 18).

In Figs. 19 through 23 the decibel readings from the test results shown in Attachment Nr. 1 were converted into loudness sones. This presentation provides a more realistic picture of the acoustic situation and the present accomplishment in noise attenuation in the M113.

For example, in the lower regions A and B, the sound intensity for 53 and 106 cps at 28 mph near the sprocket was 118 and 120 decibel respectively. When converted into loudness which is in reality acting upon the human hearing, the corresponding readings are 242 and 268 sones respectively.

Expression in sones which are not based on a logarithmic scale 've also an immediate possibility in comparison of the attenuation ability of the impact absorption sprocket.

Applying the experience in the analysis of noise phenomena in airplanes (See Chapter I) the presented frequency range in Figs. 19 to 23 was subdivided into characteristic regions designated by the kind of perception:

a.	Subsonic range	f =	to 16 cps
b.	Sonic range	f =	16 to 20000 cps
c.	Ultrasonic range	f =	over 20000 cps

These ranges are subdivided into the following regions which seem to be indicative for the source of sensation:

V = up to 30 cps (Vibrational region)

A = 30 to 75 cps (Low sonic region)

B = 75 to 600 cps (Lower medium sonic region)

C = 600 to 4800 cps (Higher medium sonic region)

D = 4800 to 20000 cps (High sonic region)

The regions A through D are important for sound and noise, the V - range is significant for the vibrational sensations in the vehicle.

Permissible levels of noise intensity are given for the sonic range for wheeled and tracked vehicles in decibels, which differ substantially in magnitude (Fig. 24 curves B_1, B_2 and C).

For the region V, permissible levels of vibration amplitudes also have been established (Curve D.). Acceptable acceleration levels are specific in their relationship between vertical, longitudinal and transverse vibrations. This will be treated in later chapters.

In Fig. 24, the loudness in sones is reported with the loudness levels in phones as a parameter (90, 100, 110 and 120 phons). The threshold of discomfort at 120 phons is shown by Curve A. The decibel reading at 1000 cps are numerically identical with the phon readings. In the same graph the noise limits recommended by HEL (Human Engineering Laboratory of APG) are reported for tracked vehicles (Curves B_1 and B_2) and for wheeled vehicles C. B_2 values apply when a pure sound exists. This is the case in the M113 in the cargo area at a speed of 28 mph and above.

The attenuation of noise by the use of the impact absorption sprockets in comparison to standard sprockets is shown in Fig. 25. It is of the order of 10 to 60% in the tested frequency range.

F. INFLUENCE OF THE IMPACT ABSORPTION SPROCKETS ON NOISE LEVEL

The influence of the impact absorption sprockets on the noise level was inconsistant. Near the sprocket, a substantial reduction of the noise in comparison to standard sprockets could be noted within a considerable range of the vehicle speed.

Inside the vehicle, this reduction was overshadowed by other still unknown influences which adversely affected the reduction of noise at the sprockets and reversed it even slightly to some extent in the high speed range for 25 to 30 mph (Table 2).

TABLE 2

Reduction of Noise	Vehicle Speed (mph) for change of Noise Experienced		
	Near Sprocket	At Driver's Seat	In Cargo Area
Substantial	at 15 mph	at 20 mph	at 28 mph
Little	at 25 mph 28 mph 30 mph		at 15 mph
None		at 15 mph	at 20 mph
Slight increase of Noise		at 25 mph 30 mph	at 25 mph 30 mph

It is interesting to note also, that at 20 mph at the driver's seat and at 28 mph in the cargo area, substantial reduction in noise is experienced. Thus, the effect of suspected resonators in the compartment on the noise created at the sprocket is selective in the compartment.

Elimination of these suspected resonant elements would probably correct the situation, shown in Table 2. For this purpose, it is suggested that potential resonant elements in the hull and compartment be examined to determine frequency level and intensity of noise and vibration and compare this with the noise modification obtained by the use of the impact absorption sprockets.

Fig. 26 demonstrates the influence of the impact absorption sprocket in the cargo area at 28 mph which represents the worst conditions in noise reported in the test report of 10 December 1963.

At about 100 cps, with the standard sprocket, the noise produced exceeds the threshold of discomfort (120 phons), as shown by Curve S. The Curve I for the impact absorption sprocket in the A and the lower B - region (below 200 cps) is substantially below that of the standard sprocket - in the average reduced by about 80 sones.

Beyond 200 cps, no appreciable reduction could be observed but above 1000 cps a slight reduction in loudness takes place.

Fig. 26 shows the recommended limits for noise in accordance with the standard of the HEL-APG report (curves B_1, B_2 and C). Since at 28 mph a deep strong sound was experienced which was perceived as a pure tone the limits designated by B_2 apply.

The loudness exceeds the permissible ("B_2") limits by:

TABLE 3

(See Fig. 27)

at 75 cps	by	2.0 times
100 cps	by	2.8
200 cps	by	3.0
400 cps	by	3.7
800 cps	by	4.3
1000 cps	by	2.9
2400 cps	by	2.8
6000 cps	by	1.7

That means the excess is increasing (up to 800 cps) with the 3rd root of the frequency and decreasing with the square root between 800 and 6000 cps, in the average by 3 times.

The gap of loudness units between the permissible sound limits for wheeled and those for tracked vehicles as determined by the standards of the APG-Human Engineering Laboratory (See Fig. 26) arouses questions about the safe limits of exposure to noise and vibrations in the M113; therefore, it should be determined just what is a "safe" sound level. The environmental living conditions in modern cities, act unfavorably upon the perception ability of the population, in particular, upon higher frequencies which are essential for the distinction of the hiss-sounds c, s, f, etc. This decrease in perception is due to the effect of noise which causes strong mechanical and biochemical stress on the hair cells, the basilar membrane of the cochlea and Corti's rods in the inner ear. These parts degenerate under stress and reduced hearing and understanding is the result. To what extent the civilization process affects the perception of sounds was recently investigated by a team of US, German and Egyptian otologists (2). The result of a comparison with the hearing ability of an African tribe remote to these blessings of civilization is shown in Fig. 28. In this graph the difference in decrease of perception ability for the reported frequency levels is shown versus age of the more than 1400 test persons. But it affects civilized populations to a far greater extent than the African tribe.

The degenerative process in the inner ear due to aging which starts as early as 20 years of age, affects all soldiers. High noise level in the M113 will accelerate this process for those exposed to vehicle noise for long periods. Reduction of noise and exposure is recommended in the interest of the crew.

G. INFLUENCE OF LOCATION OF SPROCKET ON OPERATIONAL BEHAVIOUR OF TRACK

Sprocket in Front

The acceleration of the track shoes in the stressed upper band plus the portion wrapped around the idler is accomplished by the top tooth of the sprocket. The other teeth serve to decelerate the portion of the track wrapped around the sprocket from double vehicle speed to zero on the ground. The engagement of every track shoe accelerated from ground zero creates an impulse in half the mass of the track (stressed band plus portion around the idler).

The resonance speed of the belt in accordance with Leonov[3] is:

$$V_{res} = \sqrt{\frac{1}{m}} \cdot \sqrt{\frac{T}{n^2-1}} \qquad (2)$$

where m is the mass of the unit length of the belt

T is the tension

and n is an integer.

Considering the tension and the centrifugal forces, the reduction of the track mass m would result in an increase of the resonance speed of the track inverse to the square root of the moving track mass. Increase of the tension will result in increase of the resonance speed with the square root of the total tension.

This equation is similar to the propagation speed v_{tb} of the transverse vibrations in the track, as determined by the tension T of the immobile band. The propagation speed v_{tb} is:

$$v_{tb} = \sqrt{\frac{T}{m}} \qquad (3)$$

Sprocket in Rear

The acceleration of the track portion wrapped around the sprocket is accomplished by the sprocket teeth in engagement. The upper track band running at constant double vehicle speed has no basic acceleration requirements. The tension is therefore smaller than in the aforementioned case.

That means that the resonance speed of the track will be decreased by the square root of the ratio of the track tensions including the centrifugal forces which remain unchanged. Resonance and non-resonance vibrations act upon the track.

There are various opinions about the origin and influence of track vibrations on the performance: M. K. Christie[3)] assumed an adverse influence on vehicle speed of track pulsations originated by inadequate positioning of the idler, and to some extent by the sprocket and rear road wheel. Ye D. L'vov concluded that there is an increased pulling force caused by track pulsation when the track tension falls below a certain limit. P. Nagy investigated the rate of the vibration of the upper track band with a rear sprocket. He concluded that in the event of a standing wave there is a substantial increase in the dynamic load which tends to increase the loss in a track system. He insists on the importance of elimination or reduction of the standing wave.

It is recommended that the natural frequency of the M113 track be established by computation and test as well as on the test rig for comparison purposes. Past test results of Leonov show that the maximum amplitude of the standing wave is obtained in the first resonance mode at a relatively low vehicle speed. This supports the recommendation to investigate the track behavior by increments of 2 mph, with an additional test run when a resonance falls between two test points in order to establish precise data for the natural frequency of the track.

H. COMPARISON WITH NOISE IN AIRPLANE PILOT COMPARTMENTS

A survey of noise levels in military airplanes was made by Harvard University (4). The order of magnitude and trend versus airplane speed are shown in Fig. 28. The noise shows a tendency to increase with the 0.4 power of the airplane speed V or

$$i = \text{constant } V^{0.4} \qquad (4)$$

The constant is, therefore, defined by

$$\text{Constant} = \frac{i}{V^{0.4}}$$

These values are shown versus airplane speed in Fig. 29. The speed range is too narrow to permit an unmistakable direction. There is a slight increase below 140 mph. A similar phenomenon is experienced in the M113 with regard to vehicle speed. A plotting of the test results versus vehicle speed similar to that in Fig. 29 is shown in Fig. 30 for both types of sprockets with sound pick-up near the left sprocket for 106 and 6800 cps.

The trend is similar to that of the airplanes (Fig. 30) but exhibits a variation of the exponents. The magnitude of the frequency and type of sprocket makes a mathematical model more complex; the level, however, is the same as in the airplanes.

TABLE 4

Data of the Military Airplanes #1 and #2

		AIRPLANE	
		#1	#2
Cruising conditions		Normal	Normal
Engine speed	rpm	2100	2000
Horsepower per engine		715	625
M. P.		21"	32"
Propeller tip velocity	ft/s	615	735
Speed	mph	220	135
Altitude	ft	11000	8000
i max for 37.5 cps	db	127	105
i for 4800 cps	db	99	59
Remarks:		The excessive high sound levels are believed to be due to the use of ejector exhausts.	Treatment in this plane is excellent.

Reference: L. L. Beranek and al.: Principles of Sound Control in Airplanes. Report OSRD, Nr. 1543

Fig. 31 gives a comparison of the decibel reading in the pilot cabin of a fighter airplane with excessive noise level (airplane #1 of Table 4) and the crew compartment and driver's seat of the M113 at 28 mph and 30 mph respectively. It is shown that the noise intensity of the M113 compartment up to about 1000 cps is higher by 7 to 10 db in the average. Only at frequencies higher than 2000 cps does the noise level in the M113 drop below the level of the airplane cabin.

Applying experience gathered in the acoustic treatment of airplanes, it is suggested in view of probable usefulness, that the fundamental resonances of the engine and tracks be investigated in order to locate their main harmonics in the octave bands. An analysis of track fundamental resonance is tried in Chapter K. In applying it to the M113 vehicle it might be useful to divide the octave bands into the following regions:

A 30 to 75 cps

B 75 to 600 cps

C 600 to 6800 cps

(See Figs. 11 and following)

Seventy-five (75) cps is approximately in the middle between 53 and 106 cps on the frequency line in Fig. 11; so is 600 cps in the middle between 425 and 850 cps. This simplifies the determination of the various regions A, B and C.

It is interesting to note that comparing regions and sound level of airplane #1 and that of the M113 vehicle equipped with standard sprockets, the sound intensity i has the same trend in region B but is about 8 decibels (equivalent 56 sones) higher than the level in the pilot cabin. In the regions A and C, the noise levels in both crafts approach each other. This similarity in the noise intensity, in the pilot or driver compartment, suggests an application of aircraft acoustic experience to the military ground vehicle.

In the airplane pilot cabin the increase in sound intensity level i with the increase of horsepower, can be computed from the reported test results as follows:

$$i = (94 \text{ to } 97.5) + 6.02 \log N \text{ db}$$

where N is the engine horsepower. Doubling the horsepower input from 500 to 1000 hp would, therefore, result in an increase from:

$$i = (94 \text{ to } 97.5) + 16.3 \text{ db} = 110.3 \text{ to } 113.8 \text{ db}$$

to $$i = (94 \text{ to } 97.5) + 18.06 \text{ db} = 112 \text{ to } 115.5 \text{ db}$$

The difference is 1.76 db. Three db would mean the effect of two sound sources equal in intensity.

The equivalent increase in loudness $\mathcal{L}$ would be:

from $\mathcal{L} =$ 166 to 198 sones

to $\mathcal{L} =$ 181 to 215 sones

that is by 15 sones at the lower limit and by 17 sones at the higher limit.

I. INFLUENCE OF DIMENSIONS OF THE COMPARTMENT ON FREQUENCY LEVEL

It was learned from the efforts to reduce the noise level in airplane pilots compartments[4)], that standing waves will be built up which will cause large variations of sound level from one part to the other in the cabin, if its dimensions come close to approximately half of the wave length or multiple thereof.

The following variations were observed in the compartment of airplane #2 whose noise spectrum is shown in Table 5 as a very excellent result of treatment for noize reduction:

TABLE 5

Variation of Noise Level with Location in Pilot Compartment

Frequency cps	i NOISE INTENSITY IN DECIBEL		Difference. db
	Min. Near Center	Max. Near Wall	
4-75	101	113	12
75-150	100	112	12
1200-2400	70	88	18

Since a variation of 12 decibel in noise intensity means a factor of 2 in loudness, it can be concluded that in the low frequency region the loudness in the pilot compartment is twice as high near the wall as it is near the center. In the 1200 to 2400 frequency range the difference is even greater.

J. INFLUENCE OF SOUND ORIGIN ON FREQUENCY LEVEL

During investigation of the noise spectrum in airplane pilot compartments, it was found that in the frequency pattern there is an arbitrary division of the frequency band into 3 regions: A, B and C, which opened a way to locate and identify some source of noise in these crafts.

It was found opportune to have the following ranges:

A from 37.5 to 75 cps

B from 75 to 600 cps

C from 600 to 4800 cps

The sound intensity in region A is caused by the propeller fundamental component (propeller tip passage frequency [illegible] to 75 cps) and the propeller second harmonic component (80 to 150 cps). The propeller tip velocity is a function of the engine horsepower.

The sound intensity in region B is caused by the engine exhaust, the higher harmonics of the propellers and the aerodynamic and turbulence noises are due to the propellers and the wind stream. However, it is difficult to predict the order of importance of the above mentioned causes in the composition of the noise.

The sound levels in region C result mostly from aerodynamic and turbulence noise. Measurements of sound levels in octave bands showed that adding acoustic materials in airplanes was many times more effective in reducing high rather than low frequency sounds. This leads to the conclusion that reduction of noise in the more disturbing low frequency bands in the M113 should first be attempted and the actual cause of noise in region A investigated. This is far better than trying to reduce noise by acoustic treatment which would affect more of the high frequencies of region B and C.

K. VIBRATIONAL BEHAVIOR OF M113

The limits of acceleration levels for wheeled vehicles in accordance with the APG Manual of Standard Practice for Human Factors in Military Vehicle Design[5] are shown in Fig. 33 versus frequency. They rise from 0.14 "G" for 1 cp to 0.28 "G" for 20 cps. The levels shown in Table 6 of Attachment #3 of the FMC test results range from 0.4 to 2.29 "G's" without reference to the level of frequency. They are substantially higher than the manual's recommendations.

TABLE 6

	Average Vibration Load "G" for Vehicle Speed mph			
	15	20	25	30
Standard sprocket				
Vertical	1.56	1.22	2.05	2.29
Longitudinal	0.51	0.53	0.50	0.50
Transverse	-0.92	0.60	0.99	1.38
Impact Sprocket				
Vertical	1.0	1.03	1.7	2.73
Longitudinal	0.5	0.4	0.55	0.75
Transverse	0.65	0.65	0.[illegible]	1.3

The level of vibration loads ("G's" in vertical direction) of the M113 cannot be evaluated due to lack of indication of the relevant frequency level. 2 "G's" might be acceptable for 1 cp but might be intolerable for higher frequencies. It results, however, from the Table on page 1 of Attachment [illegible] (Table 7) that the level of vibration loads in longitudinal and transverse directions is too high with reference to the level of vertical vibration loads. Compare levels of M113 vibration with the levels recommended by [illegible] (See Chapter P).

TABLE 7

	Average Vibration Load in % mph				Indicative Relative Jacklin*) Comfort Index K in Percent	
	15	20	25	30	Disturbing	Uncomfortable
Standard Sprocket						
Vertical	100	100	100	100	100	100
Longitudinal	33	43	24	22	13	18
Transverse	59	49	43	60	7.5	13
K_c average	121	119	111	118	101	102
Impact Sprocket						
Vertical	100	100	100	100	100	100
Longitudinal	50	37	32	28	13	18
Transverse	65	60	47	47	7.5	13
K_c average	129	122	115	114	101	102

*) See Chapter 0

Comparison of the M113 distribution of vibration load levels, with regard to orientation with the distributions of the relevant comfort indices of Jacklin, shows that the shares of longitudinal, and in particular of the transverse origin, are excessive in comparison with the vertical share.

Jacklin and Liddel [6] suggested an overall index K_c when all three directions are involved in accordance with

$$K_c = \sqrt{K_v^2 + K_l^2 + K_t^2} \qquad (4)$$

Assuming that the vertical shares be of the same level for all cases, then K_c would be as shown in Table 7a :

TABLE 7a

Sprockets	Relative level of K_c in percent					
	Vehicle Speed mph				Jacklin-Index K_c Distribution	
	15	20	25	30	Disturbing	Uncomfortable
Standard	121	119	111	113	101	102
Impact Absorption	129	122	115	114	101	102

In Fig. 33 the trend of the average vibration loads versus vehicle speed is shown for the three directions of vibrations. Vertical and transverse vibrations increase with the square of the speed, while the longitudinal increase with the power 1.5 or are constant.

This behavior is only approximated if considered separately for either side:

On the left side, the vibration loads increase with the cube for the transverse vibration loads of either sprocket with an obvious reduction for the impact absorption sprocket (Fig. 34). The vertical vibration load for the standard sprocket increases with the cube and for the impact absorption sprocket with the power 1.75.

It is interesting that the impact absorption sprocket creates longitudinal vibration loads, also increasing with the cube of the speed which result from the impulse energy stored in the rubber inserts which add to propulsion.

The influence of the speed upon the three specific vibration loads on the right side is apparently lower. (Fig. 35) If the measured values are based on the computed values as shown by the trend lines in Fig. 33 to 35, these ratios disclose in Fig. 36 a very good coincidence for the vehicle speed range from 20 mph and above; on the other hand, they diverge greatly from the unity (1) at 15 mph showing, thereby, existance of a strong resonance in the neighborhood of 15 mph.

L. INFLUENCE OF THE TRACK PITCH ON VIBRATION

The reduction of the vibrations by decreasing the pitch of the track shoes is in accordance with the influence of the diminished inertial reactions against rotation of the track shoes as follows:

A track shoe has (under assumption of a flat track) a velocity of zero on the ground at the moment of the approach of the idler in the case of a front sprocket. It is accelerated to double the vehicle speed at the top position of the idler within half a revolution. In a reversed position, however, the shoe made half a rotation. The same holds true at the sprocket, only acceleration is replaced by deceleration. The rotation of the shoe starts and ends suddenly, resulting therefore, in two shocks which are caused by the abruptness of acceleration, and deceleration of the track shoes at every engagement and disengagement.

The full cycle of acceleration and deceleration and deceleration takes placce within the time frame of half a revolution of idler or sprocket respectively. This is within

$$\Delta t = \frac{r \cdot \pi}{12} \cdot \frac{3600}{5280 \cdot V} \quad \text{in seconds} \qquad (5)$$

where r is the idler (or sprocket) radius respectively in inch and V the vehicle speed in mph

For $r = 1/2 \times 19.62$ in. $= 9.81$ of the sprocket

and $V = 30$ mph

The time frame $\Delta t = \frac{9.81}{12}\pi \cdot \frac{3600}{5280 \cdot 30} = 0.059$ seconds

The peripherial velocity of the sprocket for 30 mph:

$$V_{sp} = \frac{30 \cdot 5280}{3600} = 44 \text{ ft/s}$$

and the track speed therefore

$$V_{tr_{max}} = 2\, V_{sp} = 88 \text{ ft/s}$$

Hence, the acceleration

$$a = \frac{88}{0.059} = 1490 \text{ ft/s}^2 \quad \text{or}$$

$$a = 46 \text{ "G's"}$$

For the rated speed of 37 mph, the peripherial sprocket speed is

$$V_{sp} = 54.2 \text{ ft/s}$$

and the track speed

$$V_{tr} = 2 \times 54.2 = 108.4 \text{ ft/s}$$

The relevant time for half a revolution of the sprocket is

$$\Delta t = \frac{19.62}{12 \times 2} \cdot \frac{3600}{5280 \times 37} = 0.047 \text{ seconds}$$

and the deceleration of the shoes on the sprockets is

$$a = \frac{108.4}{0.047} = 2300 \text{ ft/s}^2 \quad \text{or}$$

$$= 71 \text{ "G's"}$$

The diameter of the idler is 17.4 in. Thus, the acceleration of the track shoes over the idler is 1680 ft/s^2 or 52 "G's" for 30 mph and 2600 ft/s^2 or 30 "G's" for the rated speed of 37 mph. These are considerable values.

The acceleration is applied suddenly to the track shoes on the idler by the top tooth of the sprocket transmitted through the moving track band to the track portion wrapped around the idler. It can be easily understood that these jerkingly originated track shoe movements will cause a reaction in track, idler and sprocket. Over and above this acceleration, the track shoe makes half a rotation from zero to top track speed involving inertial forces and a rotational acceleration. Since the sprocket or idler moves in space, there is also a Coriolis-acceleration involved which adds to the acceleration problem.

The uplift of sprocket and idler in the M113 affects little the basic relationship as analyzed for flat tracks by John Eilers (See Attachment #II).

Since there is a difference in the length and mass of the tracks at the left and right side, a difference in their behavior has to be expected. At certain vehicle speeds, resonance can result with repercussion on performance, vibration, noise, stress, wear and endurance.

The flexible idler is intended to help investigation of the fundamental aspect of track movements to guide design improvements.

Since the track shoes wrap around idler and sprocket in a polygonal manner with periodic radial displacement of the end-connectors which form the flexible corners, a periodic variation in track tension is caused which adds to the previously analyzed phenomena.

It is proposed to adapt the idler rim on an intermediate system of plates elastically interconnected. This will neutralize the fluctuations of tensions in the tracks and of the rotational inertial forces created by the changes in acceleration of the shoes (See Fig. 1).

M. POSSIBLE ORIGINS OF VIBRATION

The theory of tracked vehicles provides a formula for the tension of the immobile track as follows(LEONOV [3]) :

$$T\,s = Z^2 \frac{GL}{8} \qquad (6)$$

where T is the tension of the immobile track

s is the sag

G is the weight of the track per unit length

L is the track pitch

Z is the number of links

(All dimensions in kg, m, sec).

Equation (6) provides a means to determine the dynamic tension and its variation by establishment of the dynamic displacement (dynamic sag) of the track. The variation of the dynamic tension creates a periodically variable load of the sprocket shafts and idler axles (which are in essence cantilever beams). The fluctuations of the loads with consideration of the clearances of the joints can enter into the hull in the form of impulses creating hull vibrations.

Under the varying dynamic loads the shafts and axles are exposed to slight deformations which might result into lateral movements of the tracks and, therefore, lateral impulses (transverse vibrations).

A source of longitudinal vibrations of the track is seen in the periodic engagement of the trackshoes on the sprockets and idler wheels. This results in a relevant variation of track tension superposed on other active dynamic phenomenons in the track.

Another source of vibrations of the track is the conditions of the ground, geometrical nature, (holes, bumps, washboard, etc) hardness, etc. The inherent natural frequency of the track is activated to resonance conditions under certain modes of vehicle speed or possibly engine speed.

Natural frequency of other components of the vehicle can exist.

H. TRANSVERSE VIBRATIONS

S. I. Leonov[3)] investigated behaviour of belts in view of transverse vibrations. He defined the speed at which vibrations take place by the following equations

$$v = n \cdot v_{tb} \qquad (7)$$

where n is an integer or a proper fraction with one in the numerator and

$$v_{tb} = \sqrt{\frac{T}{m}} \qquad (8)$$

is the velocity of propagation of the transverse vibration in the track.

T - is the tension of the immobile band

m - is the mass of the belt of the unit length

It was established that vibrations occured when the velocity of propagation in the belt was a multiple of the band speed.

It was suggested that the centrifugal forces be considered for computation of the resonance speed as follows:

$$v = \sqrt{\frac{T + m\,v_b^2}{m}} \qquad (9)$$

where v_w = the propagation velocity of the wave

v_b = the speed of belt motion

For resonance conditions v_w is a multiple of the belt speed v_b thus

$$n \cdot v_{res} = \sqrt{\frac{T + m\,v_{res}^2}{m}} \qquad \text{or}$$

$$v_{res} = \sqrt{\frac{T}{m}} \cdot \sqrt{\frac{1}{n^2 - 1}} \qquad (10)$$

For a track, the tension could be calculated from the theory of tracked vehicles as follows:

$$T = \frac{z^2\,G}{8} \cdot \frac{L}{s} \qquad (6)$$

where z is the number of track links

G is the weight of the track per unit length

L is the pitch of the track

s is the sag of the track

(all units in the metric system: kg, m, sec)

For comparison purpose the following data and results of the track test rig are given:

sag of track	s	=	0.03 m
tension	T	=	121.2 kg
mass of unit length of track	m	=	$5 \frac{kg\ sec^4}{m^2}$

for n = 2 the computed speed v_{res} = 2184 m/sec = 10.2 km/h .

The experimental resonance speed was V = 10 km/h .

At V = 10 km/h the average pull on the working band of the track was about 800 kg while the track expending forces in the upper free track band was increased up to (1300 to 2400) kg (225 to 300%).

This illustrates that vibrations in the track can create substantial dynamic loads with impeding effect on the operation of the vehicle.

It is recommended to investigate the existance and amplitude of track vibration (and eventually resonance conditions) on the M113 vehicle by taking motion pictures simultaneously on the right and left side of the vehicle against a light (white) background on the hull, at at least two significant vehicle speeds.

Formula (10) may give some indication for expected resonance speed ranges.

O. VIBRATION TEST OF M113 VEHICLE EQUIPPED WITH STANDARD OR IMPACT ABSORPTION SPROCKETS

In Attachment #3, the test results of vibration measurements, dated December 10, 1963, versus vehicle speed are shown for the standard and the impact absorption sprockets.

Tests were performed to investigate the effect on crew members during relatively high-amplitude low-frequency vertical vibrations. The influence on decrement of performance in various tasks was to be established.

As a basis of evaluation, the test results of Sternick and al.,[7] Janeway[8] and Snyder[9] can be used. In the following the results obtained by Snyder on five subjects are shown in Fig. 38.

The uneveness in the experimental curves is attributed to existing differences between different subjects in their tracking efficiency during vibration.

For evaluation of the meaning of the various operational curves the following definitions may be useful*)

1. Threshold of perception: By intense concentration and with prior knowledge of the coming vibration, the vibration was perceived by the pilot.

2. Definitely or easily perceptible: This level is the minimum vibration which cannot escape attention during performance of normal in-flight duties. No interference with duties occurs.

3. Irritating or annoying: The vibration has sufficient intensity to divert attention from operational duties or tasks which are beginning to be impaired. Fatigue would be experienced in case of prolonged exposure (over 1 hour). With increased concentration, normal crew duties would be little impaired.

*)Abstract from instructions to test subjects Reference 7) p. 43

4. Maximum Tolerable for continuous operation: Vital duties not requiring fine adjustments could still be performed. Safety of flight could be maintained for limited time (up to 15 min.). Typical bombing navigation equipment (for comparison) would probably be non-operable at this condition.

5. Intolerable: Inability to perform even very gross functional airplane controls. The conditions must be limited to 1 or 2 min of flight without danger. High alarm feeling when experienced during flight.

The range recommended by R. N. Janeway for comfort limits (curve A-B-C-D) is recorded in Fig. 39. It coincides approximately with the test curve characterised by: "Definitely or easily perceptible" which excludes substantial interference with operational duties and with the attention of the pilot. It whould be noted that the range of Janeway's recommendation extends to 50 cps, that is into the lowest frequency of the M113 tests of sound intensity.

Although these experiments were obtained with pilots in airplanes during flight, they can be used for comparative judgement for military vehicle requirements, until more specific experience with these vehicles is available. It is suggested to examine the level of vibration amplitudes in regions A and V in future vehicle tests in view of the comfort limits of Janeway. Military vehicles may not require the smoothness of passenger cars. The limits may therefore lie closer to the curve - "Irritating and Annoying" in Fig. 39.

P. LIMITS OF COMFORT OF SINUSOIDAL VIBRATIONS

The experimental curves of comfort level of Snyder[9] can be approximated within the limits from 3 to 30 cps by the following relationships for the amplitude A of the vibrations:

1. Threshold of perception

$$A = \frac{0.37}{f^{2.25}} \quad \text{in G's} \tag{12}$$

2. Definitely or easily perceptible

$$A = \frac{0.63}{f^{2.25}} \tag{13}$$

3. Irritating or annoying

$$A = \frac{2.0}{f^{2.25}} \tag{14}$$

4. Maximum tolerable for continuous operation

$$A = \frac{3.65}{f^{1.85}} \tag{15}$$

5. Intolerable

$$A = \frac{2.00}{f^{1.53}} \tag{16}$$

Jacklin and Lidell[6] recommended the following equation of subjective sensations relative to vibrations:

$$K = ae^{0,6.f} \tag{17}$$

Where a - is the maximum value of acceleration

f - frequency (cps)

K - comfort index

e - the basis of natural logarithms

The following characteristic levels of K at different vibration conditions were given:

TABLE 8

ORIENTATION	COMFORT LEVEL	COMFORT INDEX K		
		K	%	%
Vertical	Uncomfortable	64.7	100	100
Longitudinal	Uncomfortable	11.73	18	18
Transverse	Uncomfortable	8.21	12.7	12.7
Vertical	Disturbing	31.2	48	100
Longitudinal	Disturbing	4.02	6.2	12.8
Transverse	Disturbing	2.35	3.5	7.5

Based on these values, the limit acceleration for comfort should be determined by

$$a_e = \frac{K}{0.6 \times f} \qquad (17a)$$

Table 8 shows that the tolerance for longitudinal vibration is only 18% of the amount of the vertical, and for the transverse only 12.7% in the uncomfortable range. The reduction is even greater in the disturbing range: 12.8% of the vertical tolerance for the longitudinal and 7.5% for the transverse.

In Fig. 40, Snyder's curves of specific comfort for vertical vibrations and Janeway's recommendation are reported on a straight graph which shows that with increasing frequency the permissible amplitudes decrease beyond about 8 cps very quickly to low amounts; it does not take an appreciable increase to reach the irritation condition.

If these values are brought into relationship with the Jacklin distribution of comfort indices for vertical, longitudinal and transverse vibration, it becomes apparent that the tolerable amplitudes for transverse and longitudinal vibrations are considerably below the experienced values in the M113.

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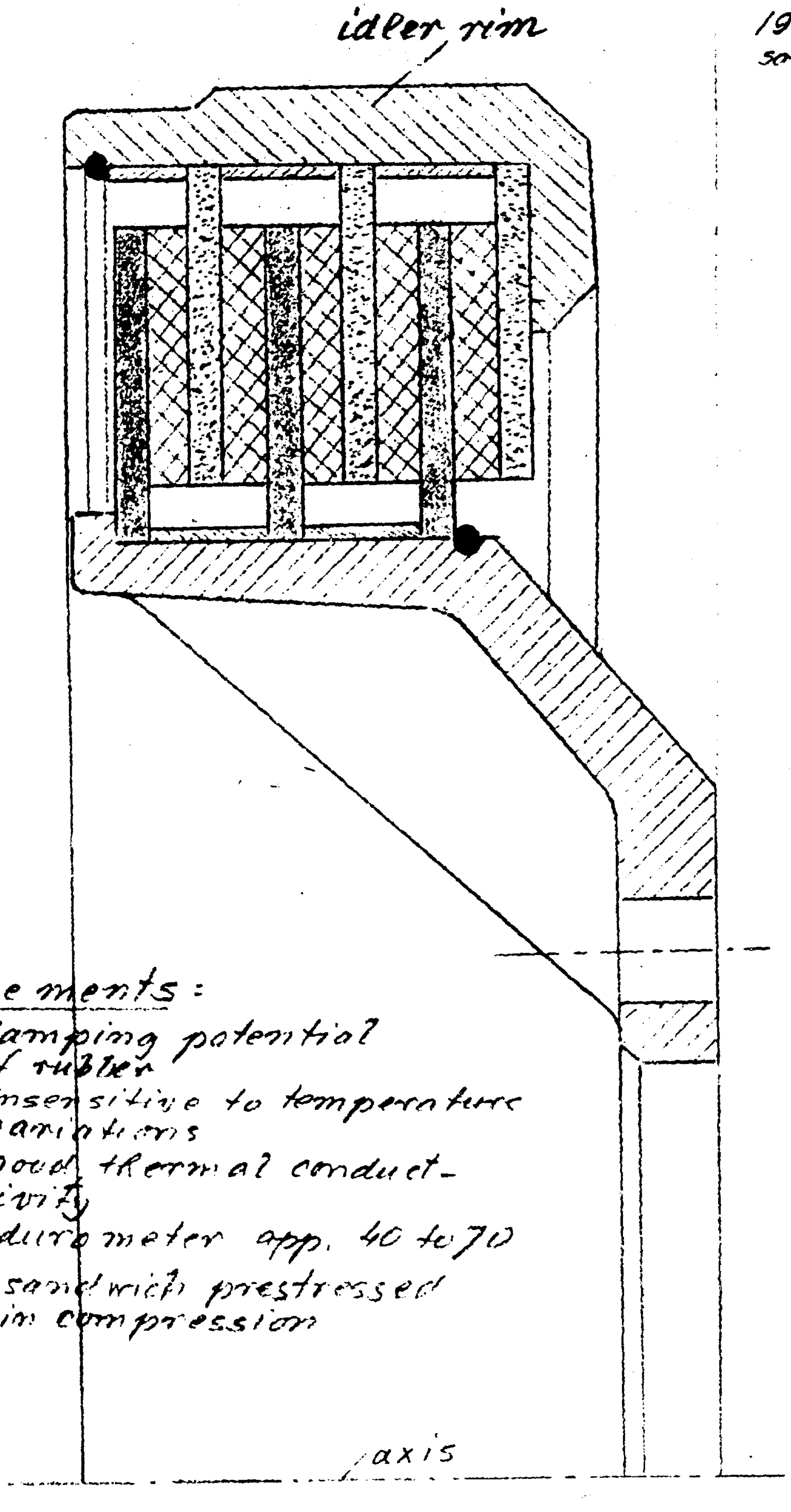


Requirements:

a.) damping potential of rubber
b.) insensitive to temperature variations
c.) good thermal conductivity
d) durometer app. 40 to 70
e) sandwich prestressed in compression

Fig. 1

Noise Level near Sprocket
Versus Vehicle Speed

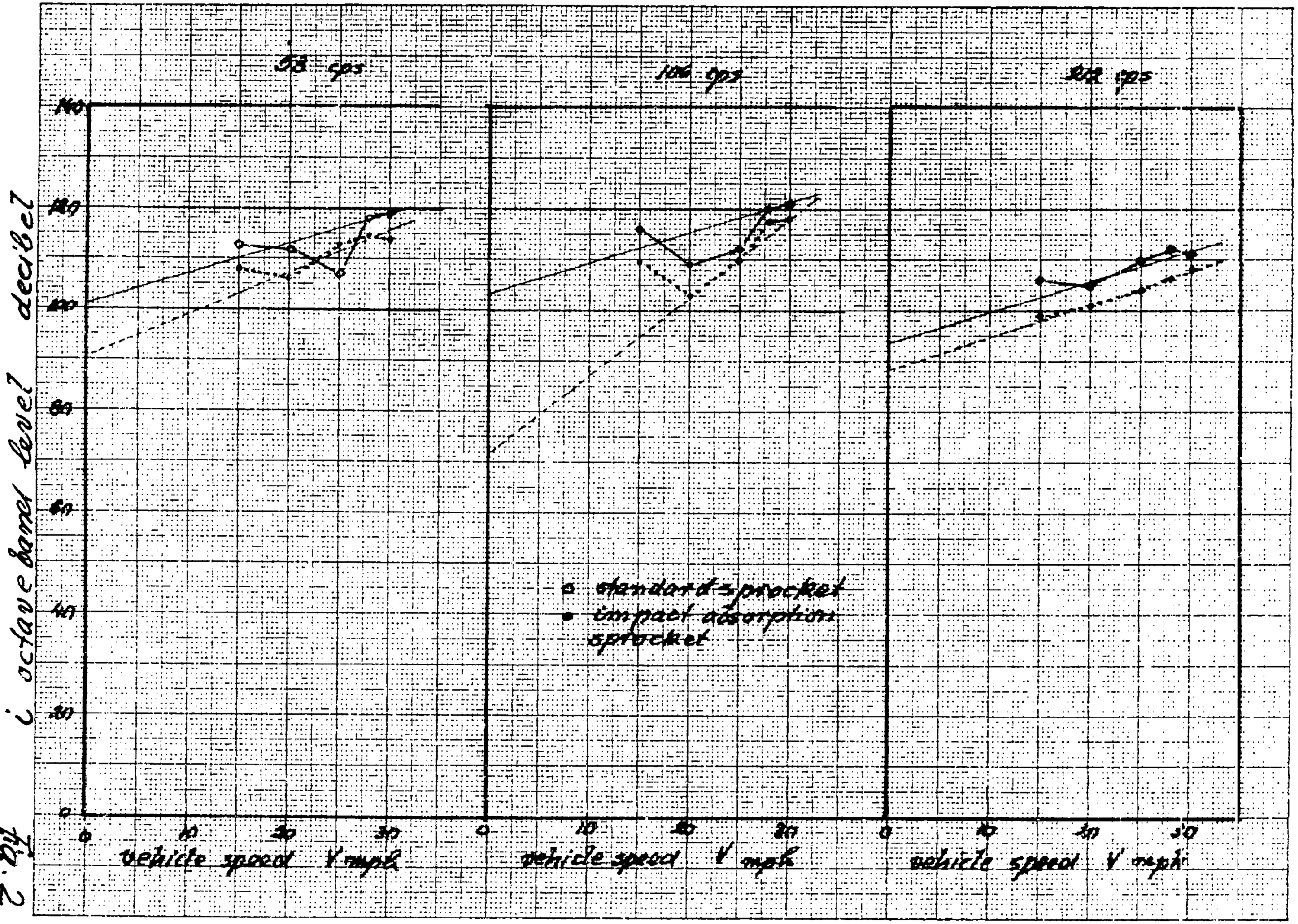


Fig. 2

Noise Level near Sprocket
Versus Vehicle Speed

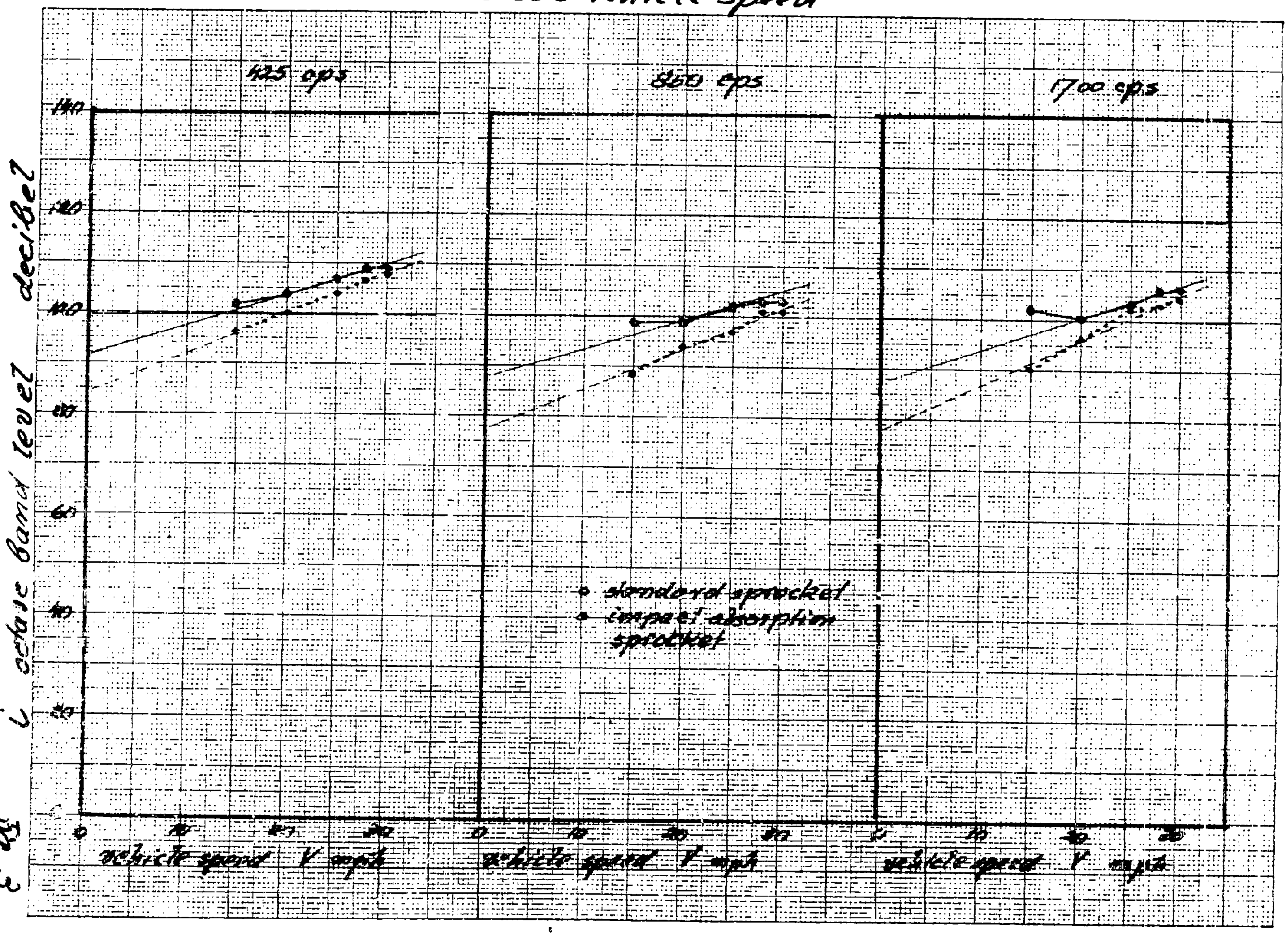


Fig. 3

Noise Level near Sprocket versus Vehicle Speed

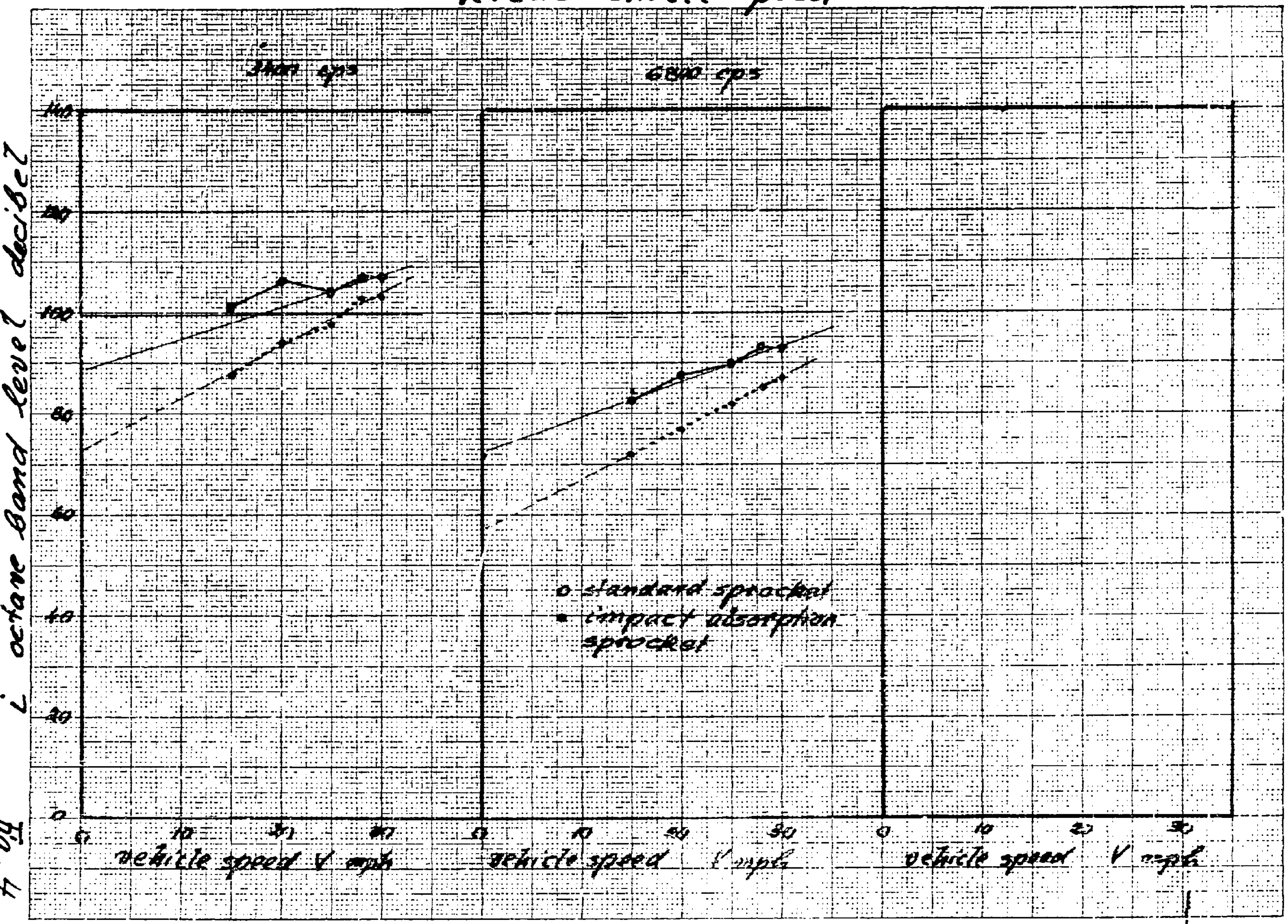


Fig. 4

Noise Level Near Driver

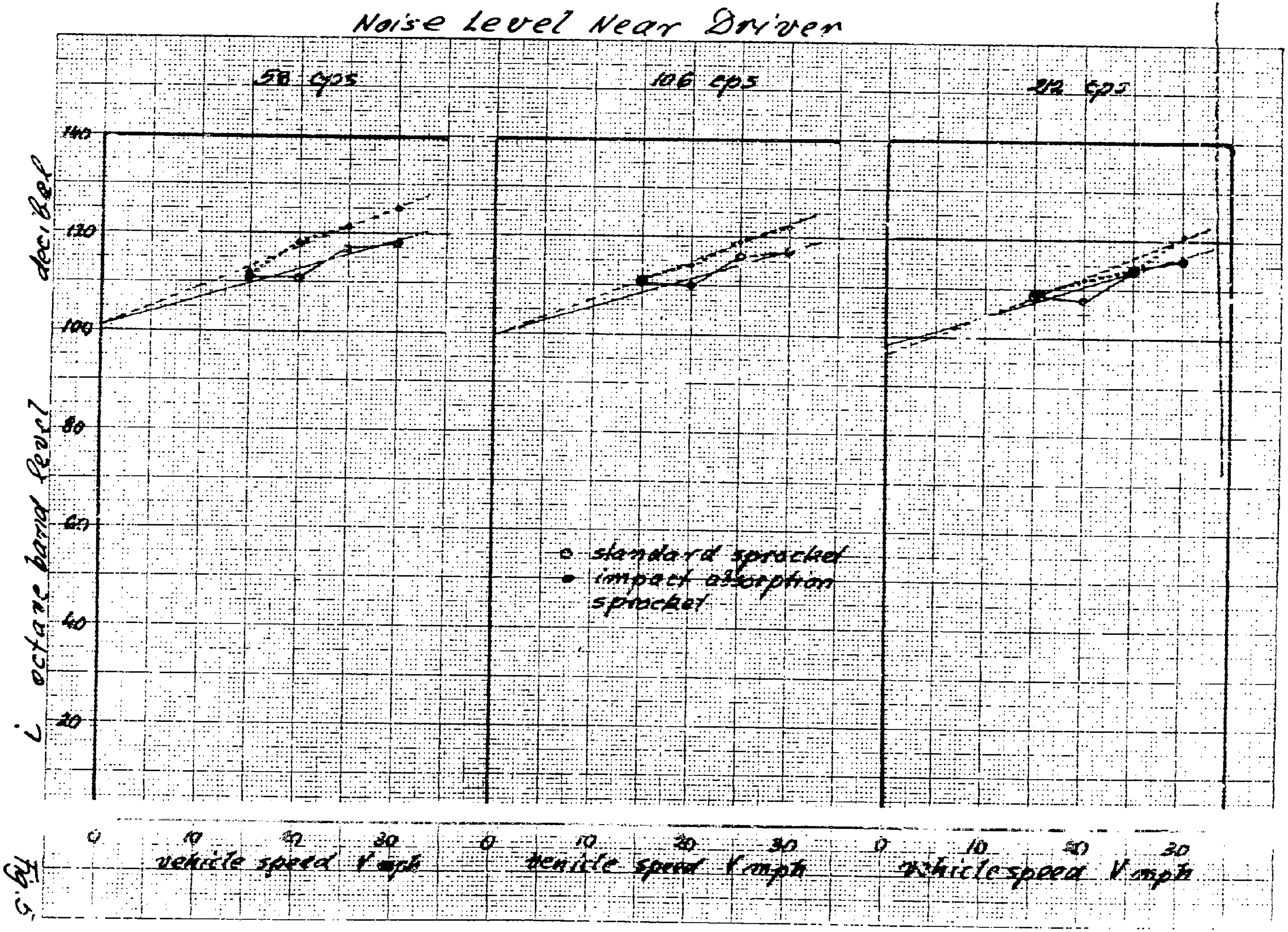


Fig. 5

Noise Level Near Driver

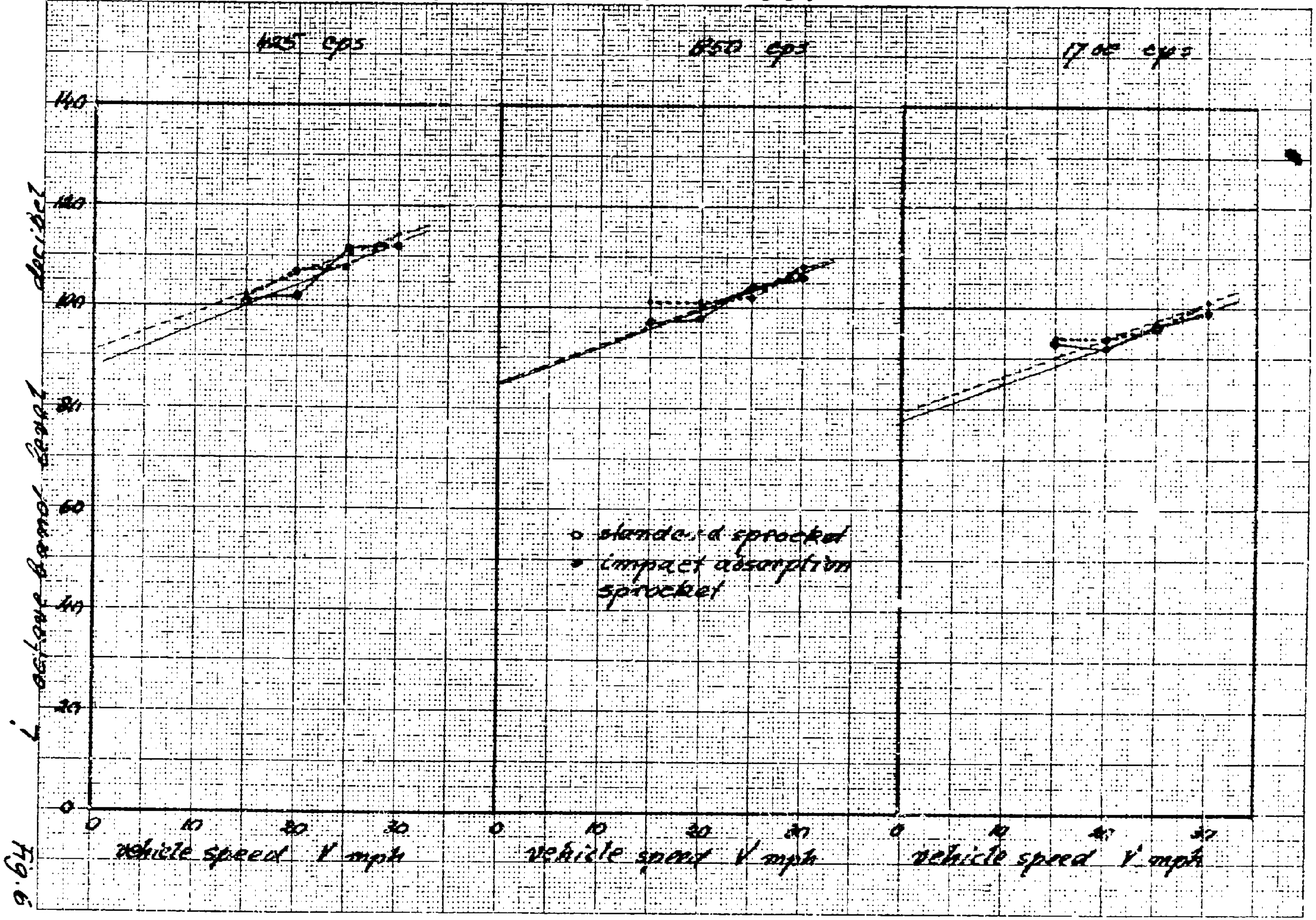


Fig. 6

Noise Level Near Driver

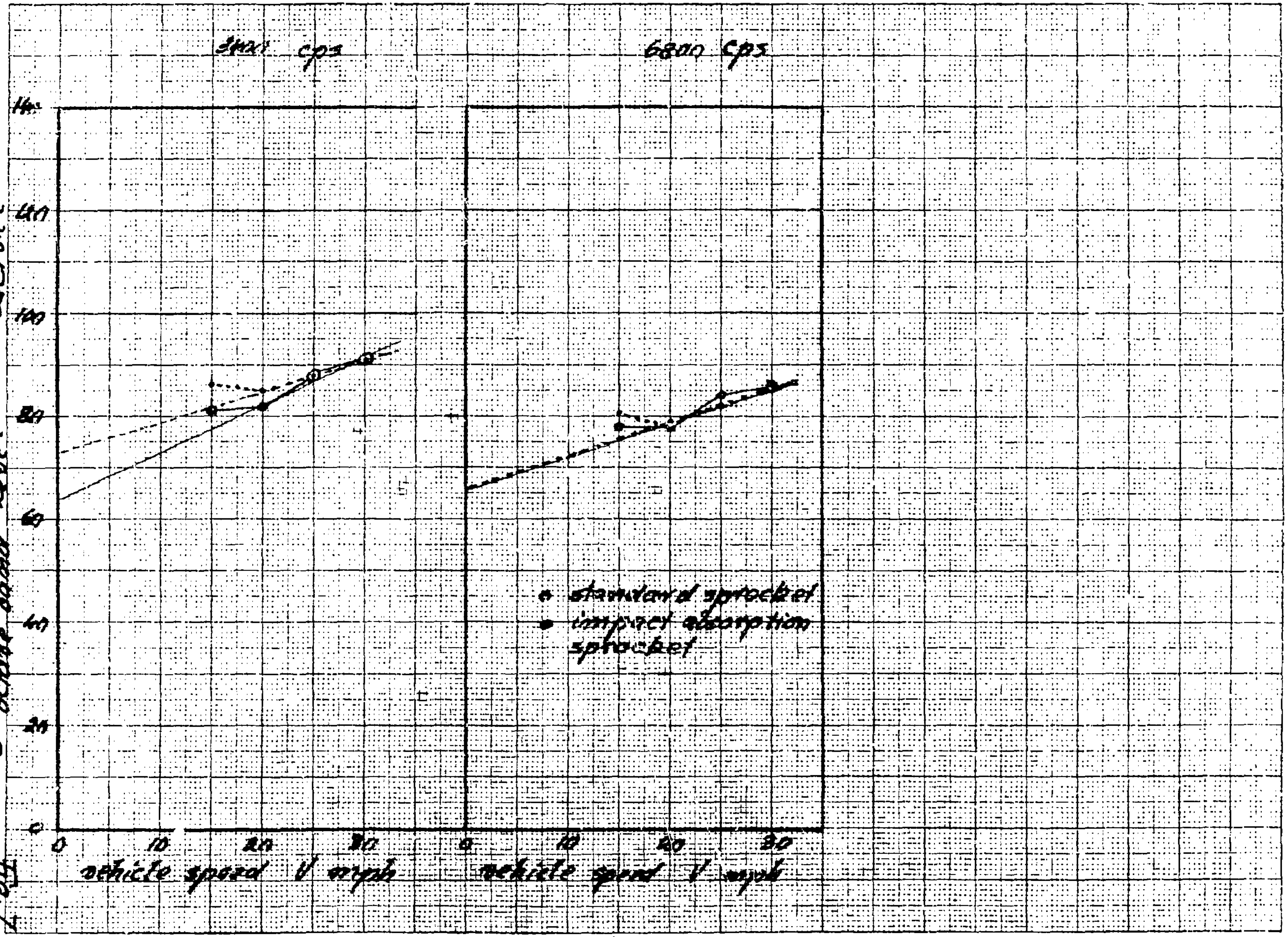


Fig. 7

Noise Level in Cargo Area
Versus Vehicle Speed

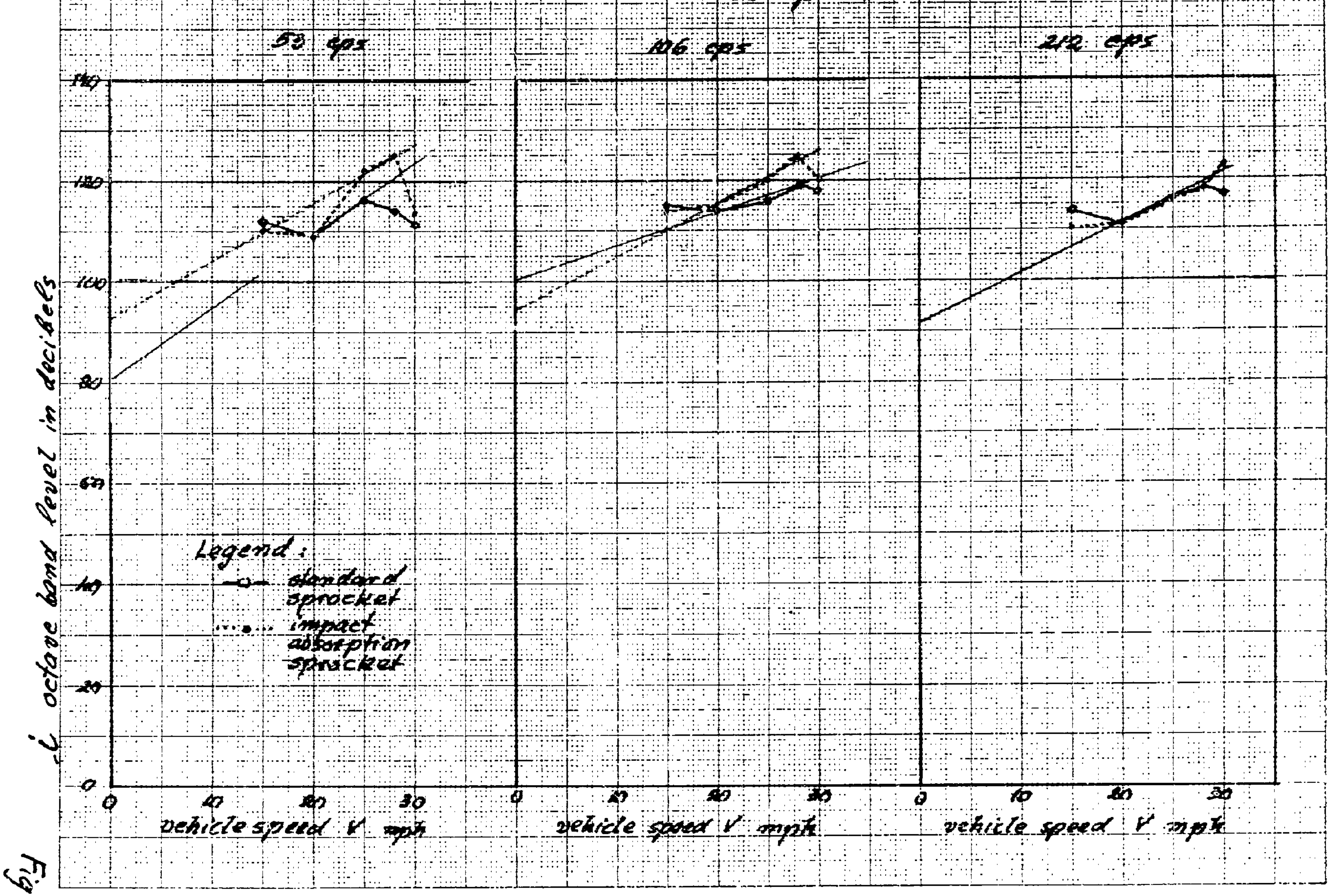


Fig. 8

Noise Level in Cargo Area
Versus Vehicle Speed

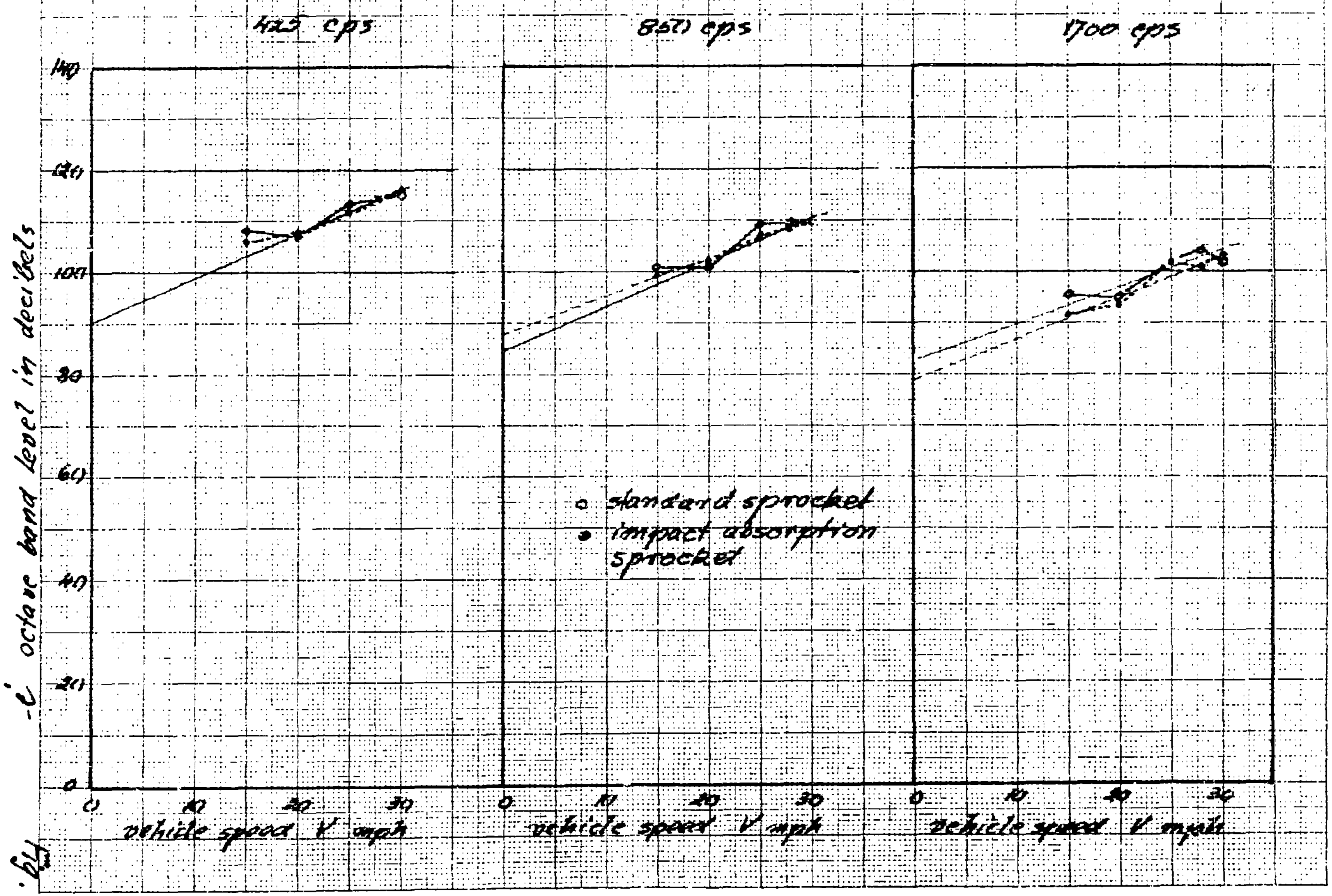


Fig. 9

Noise Level in Cargo Area versus Vehicle Speed

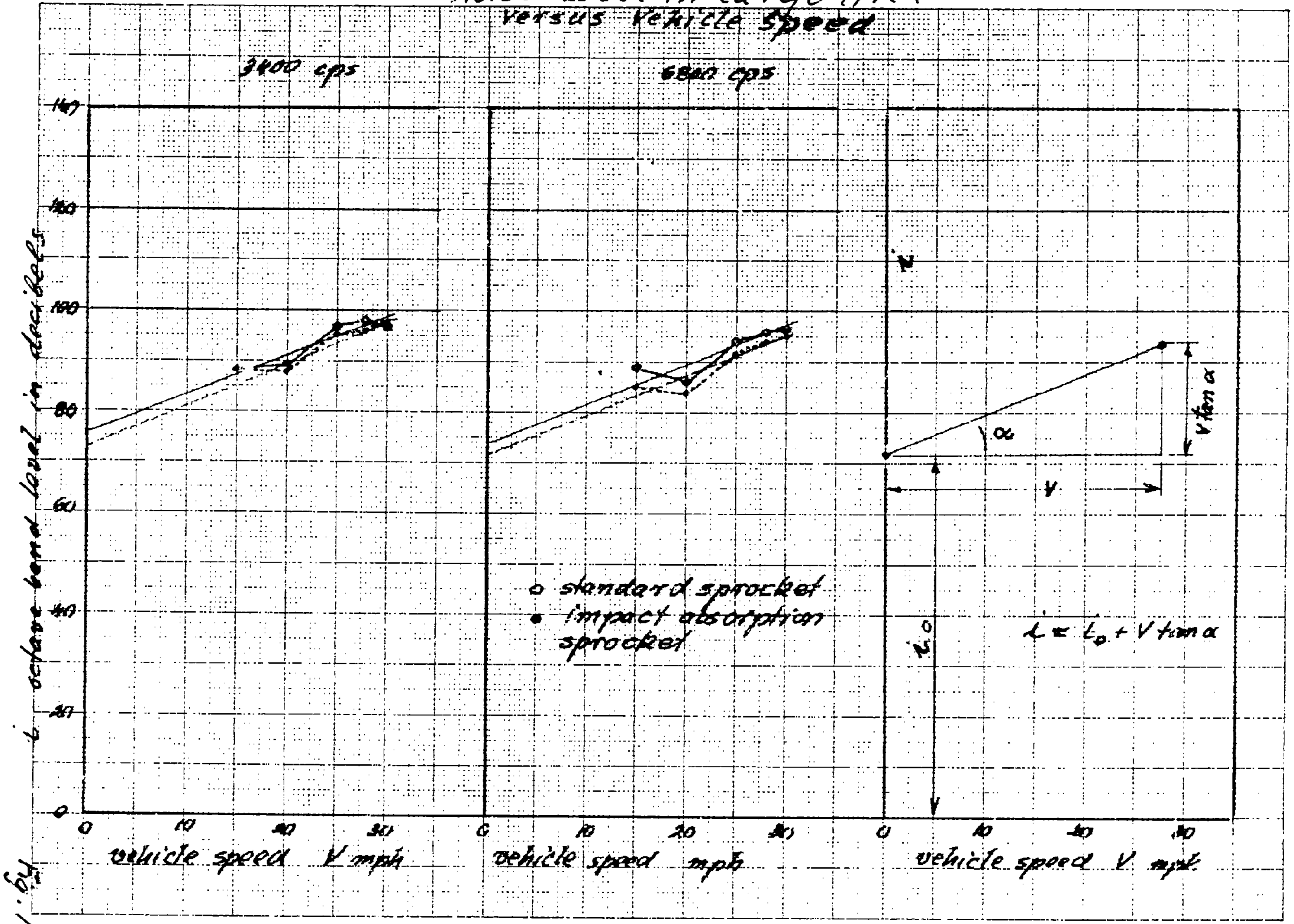


Fig. 10

Rate of Noise Near Sprocket for standard sprocket versus frequency

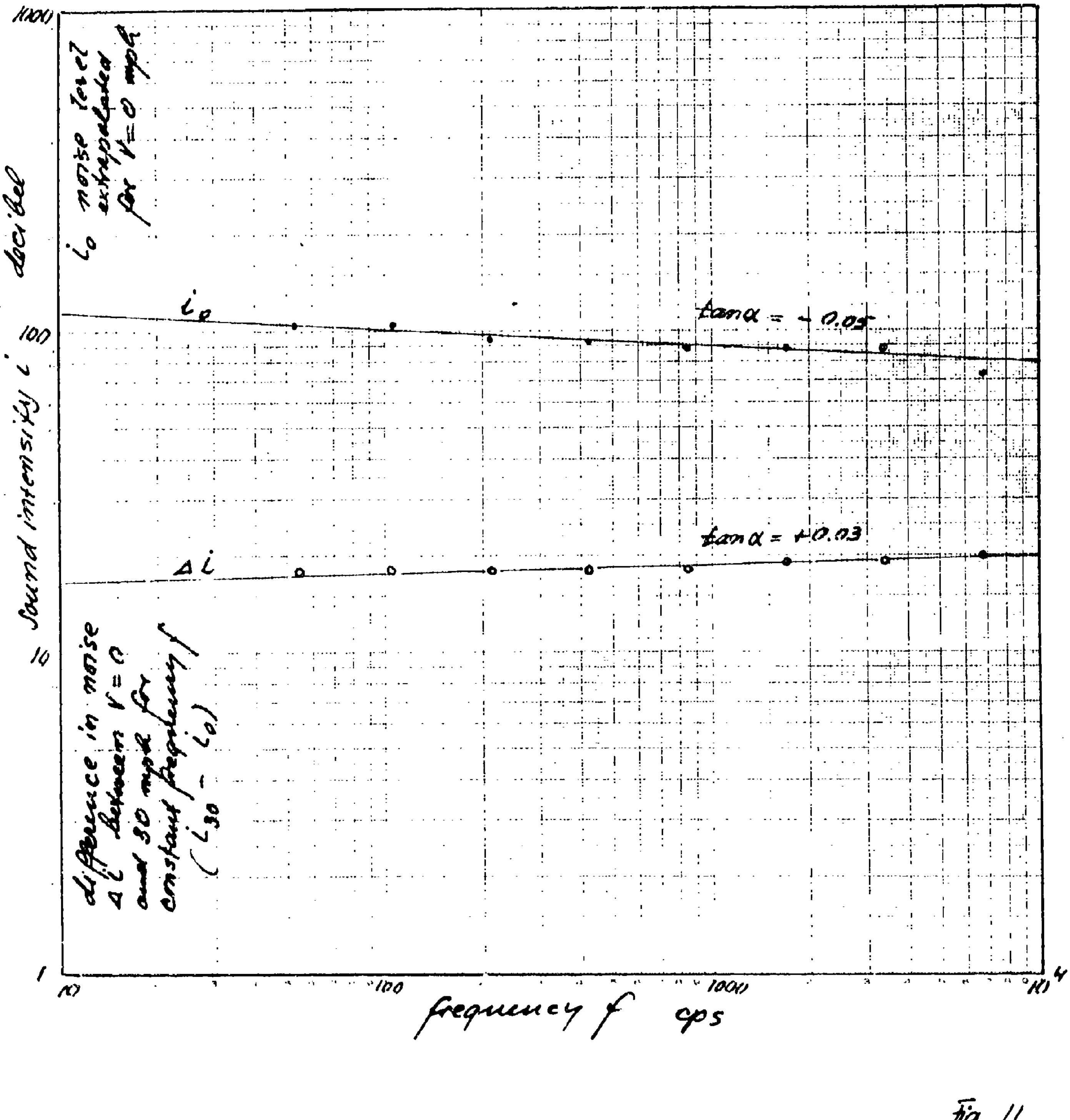


Fig. 11

Rate of Noise near Sprocket
versus frequency
(Impact Absorption Sprocket)

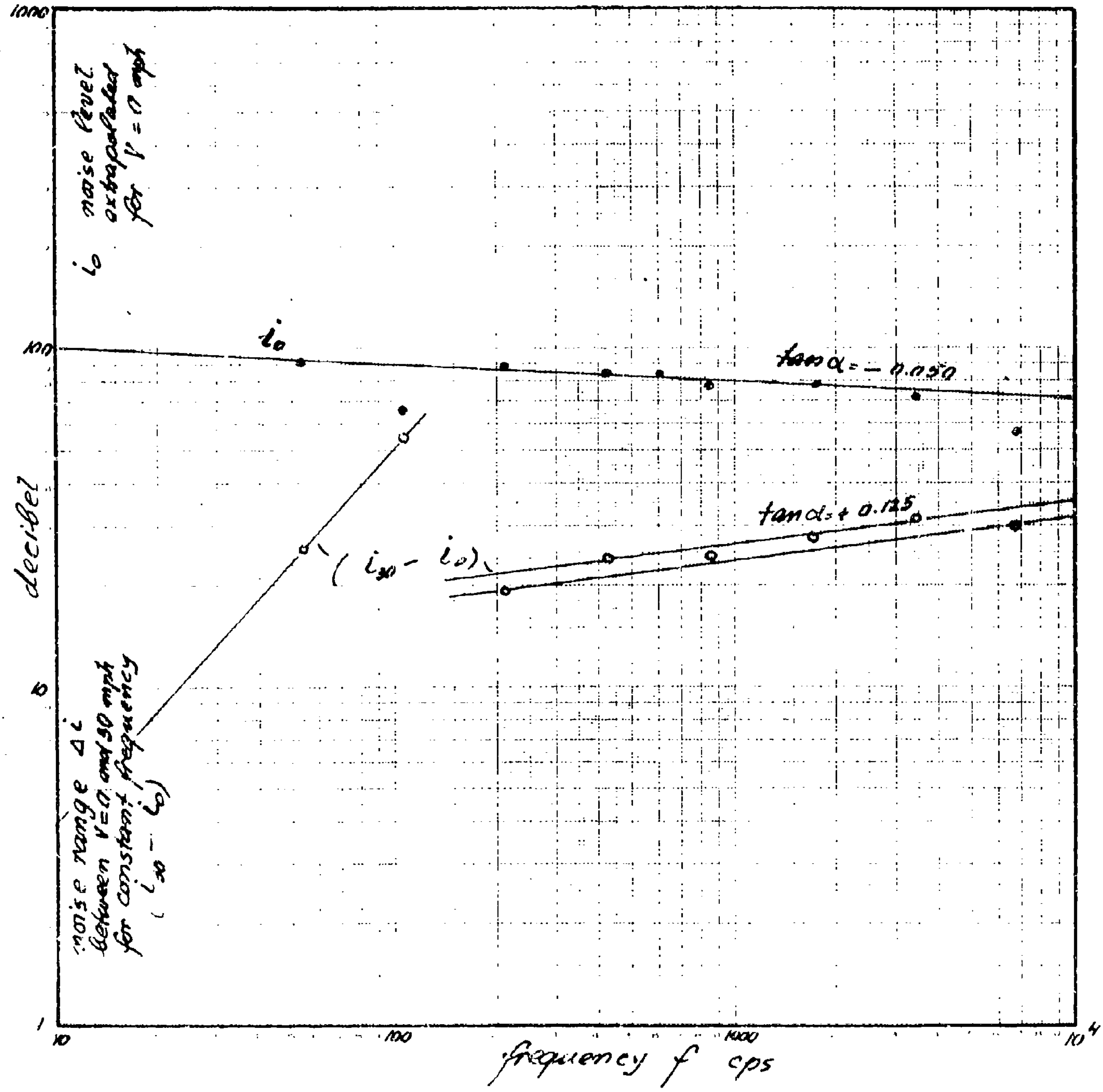


fig. 12

Rate of Noise near Sprocket
Versus Frequency

Comparison of Standard with Impact
Absorption Sprocket

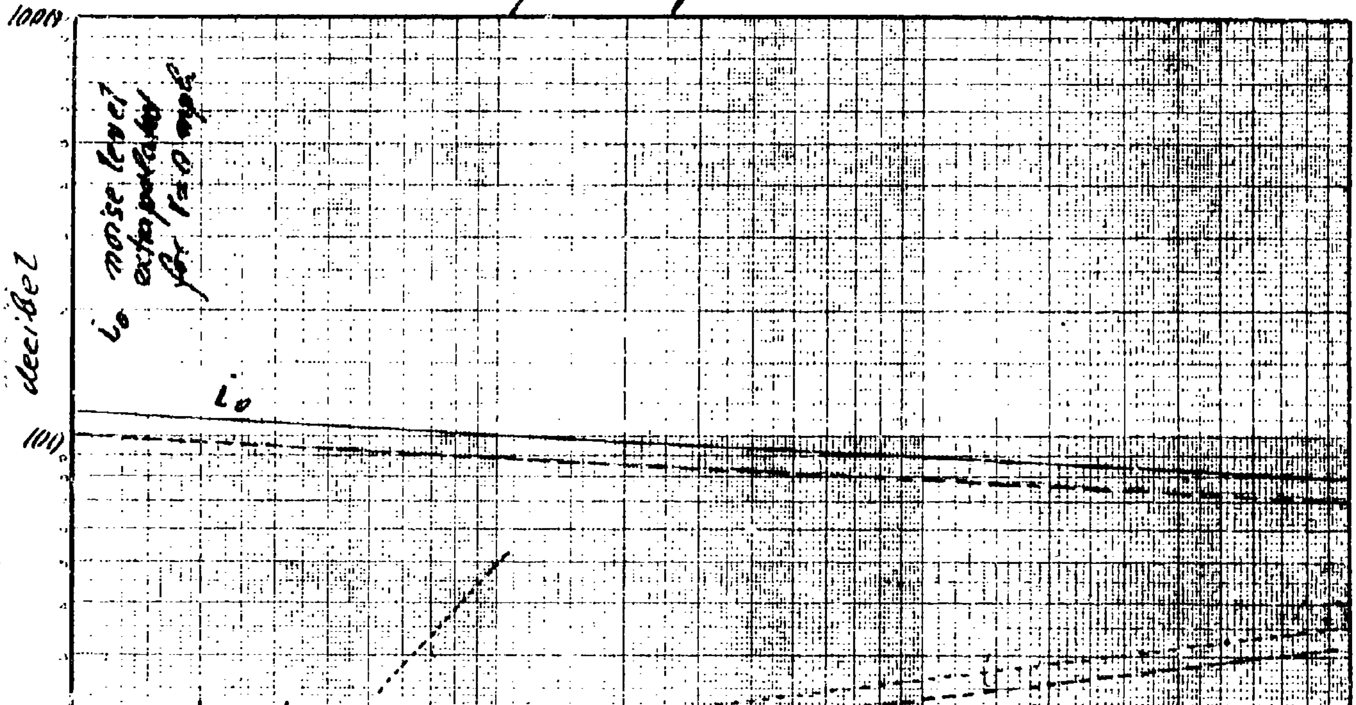


fig. 13

Rate of Noise Near Driver
versus frequency

Comparison of Standard with Impact Absorption Sprockets

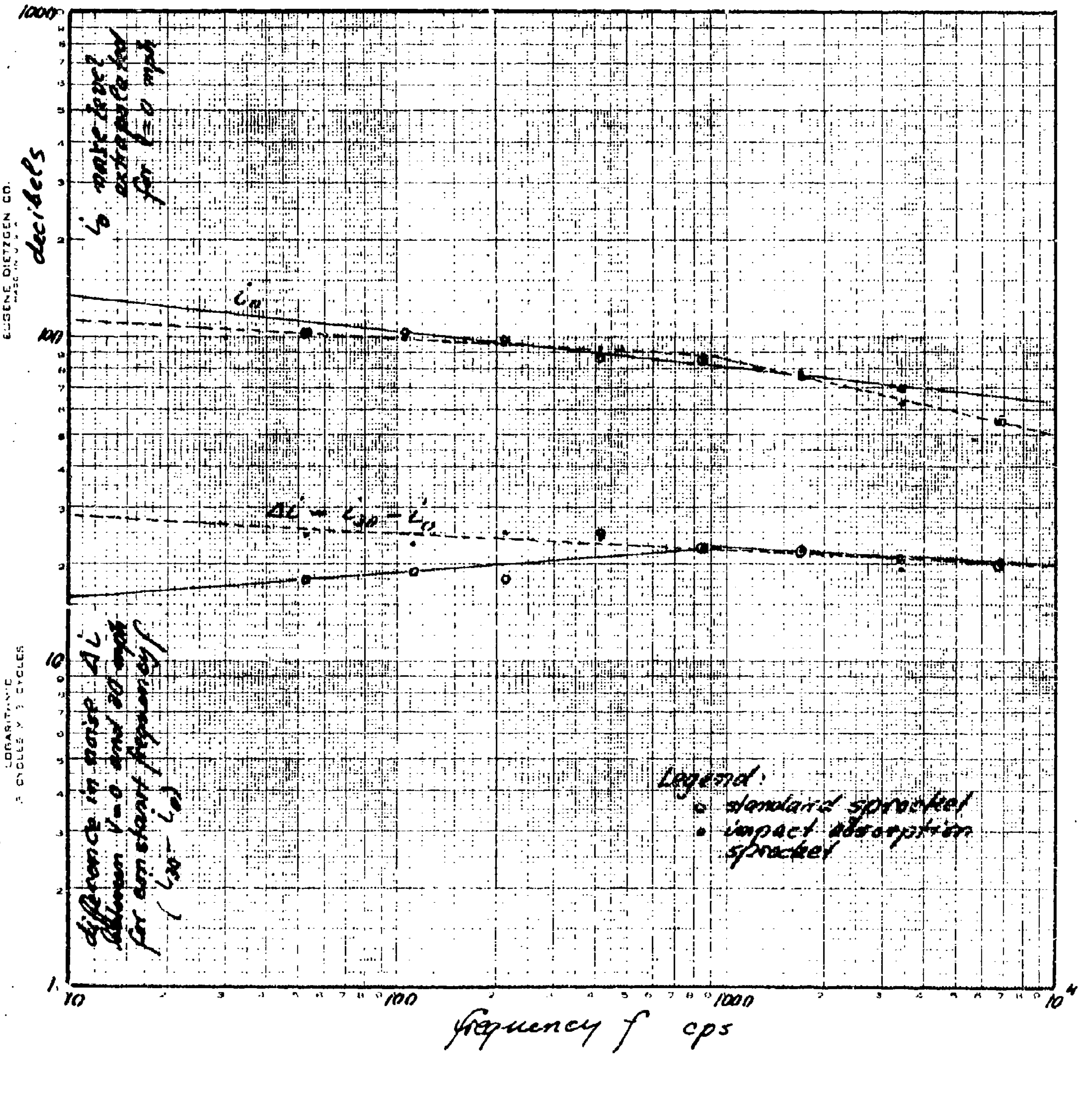


fig. 14

Initial Level i_0 and Rate of Noise in Cargo Area of Standard Sprocket versus Frequency

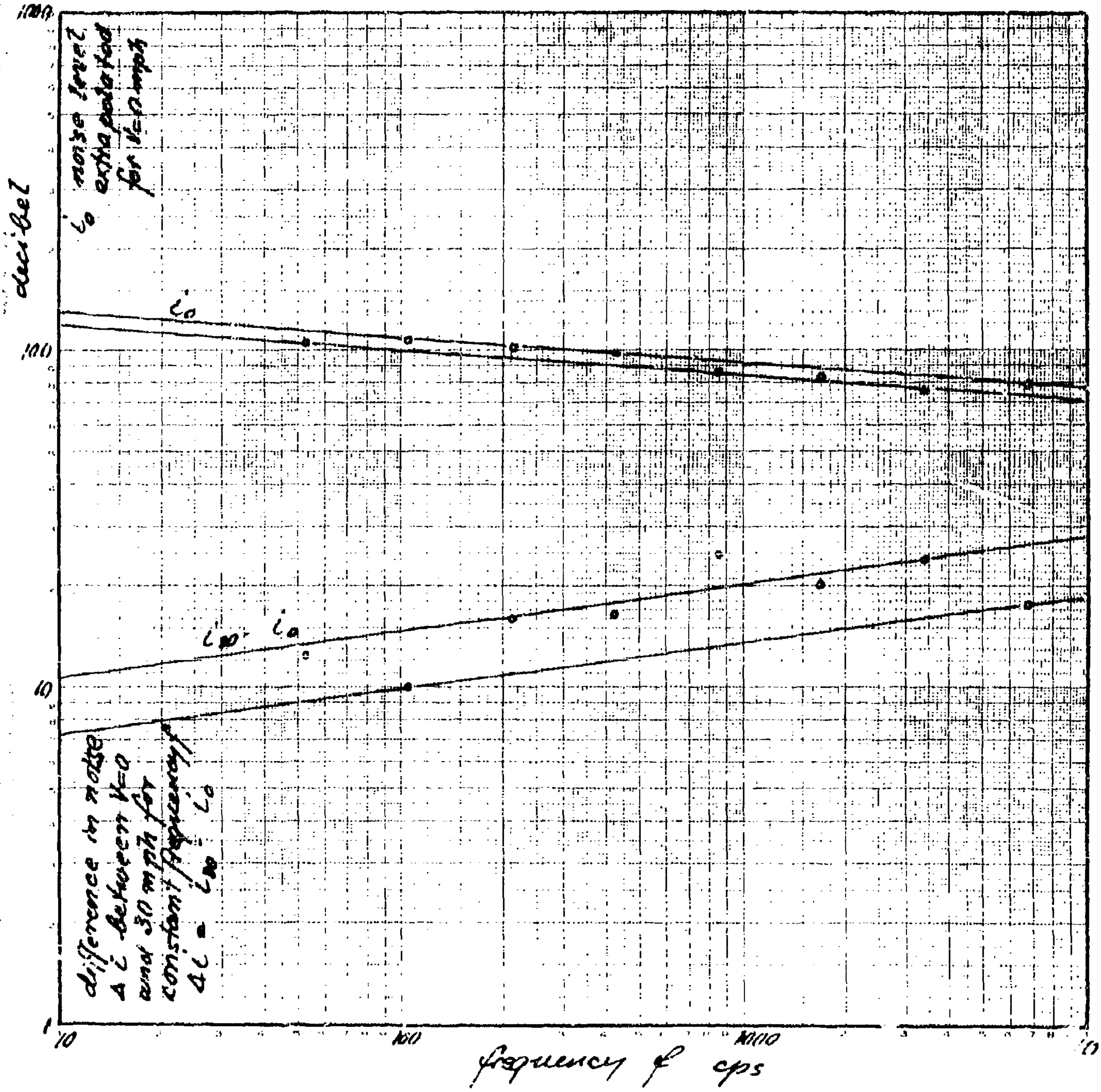


fig. 15

Rate of Noise in Cargo Area
versus Frequency
(Impact Absorption Sprocket)

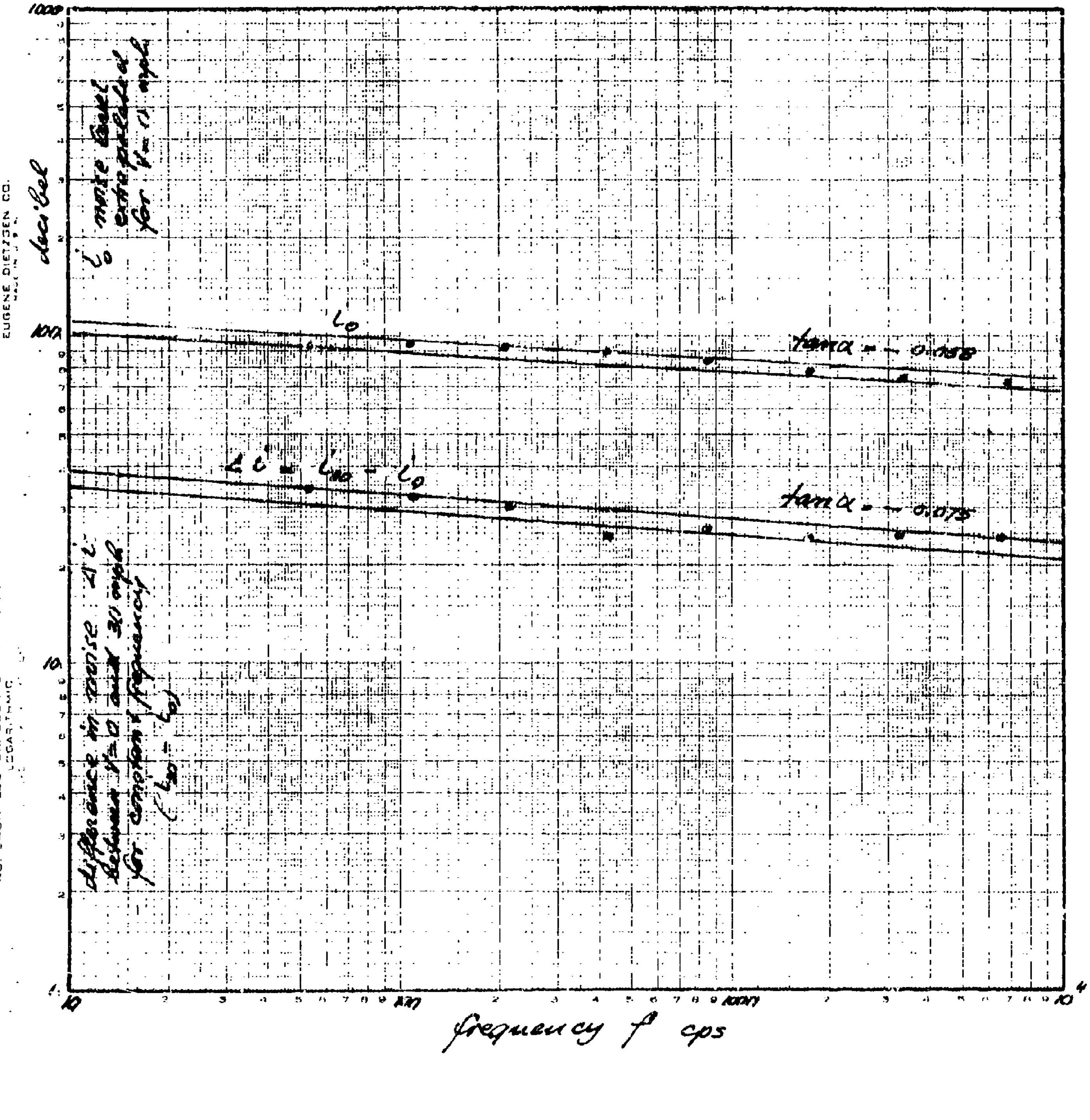


Fig. 16

Initial Level i_o and Rate of Noise in Cargo Area versus Frequency. Comparison of Standard with Impact Absorption Sprocket

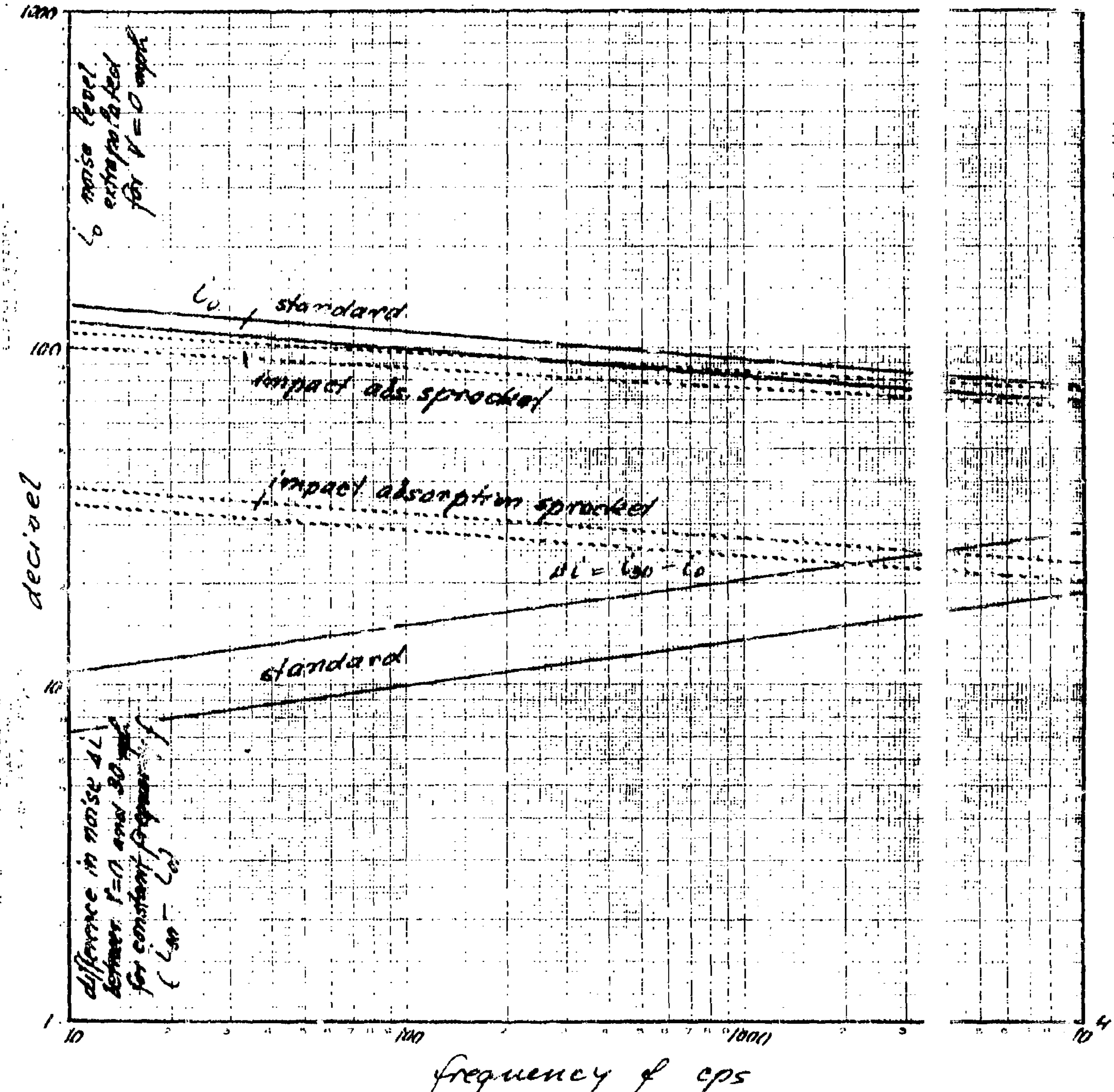
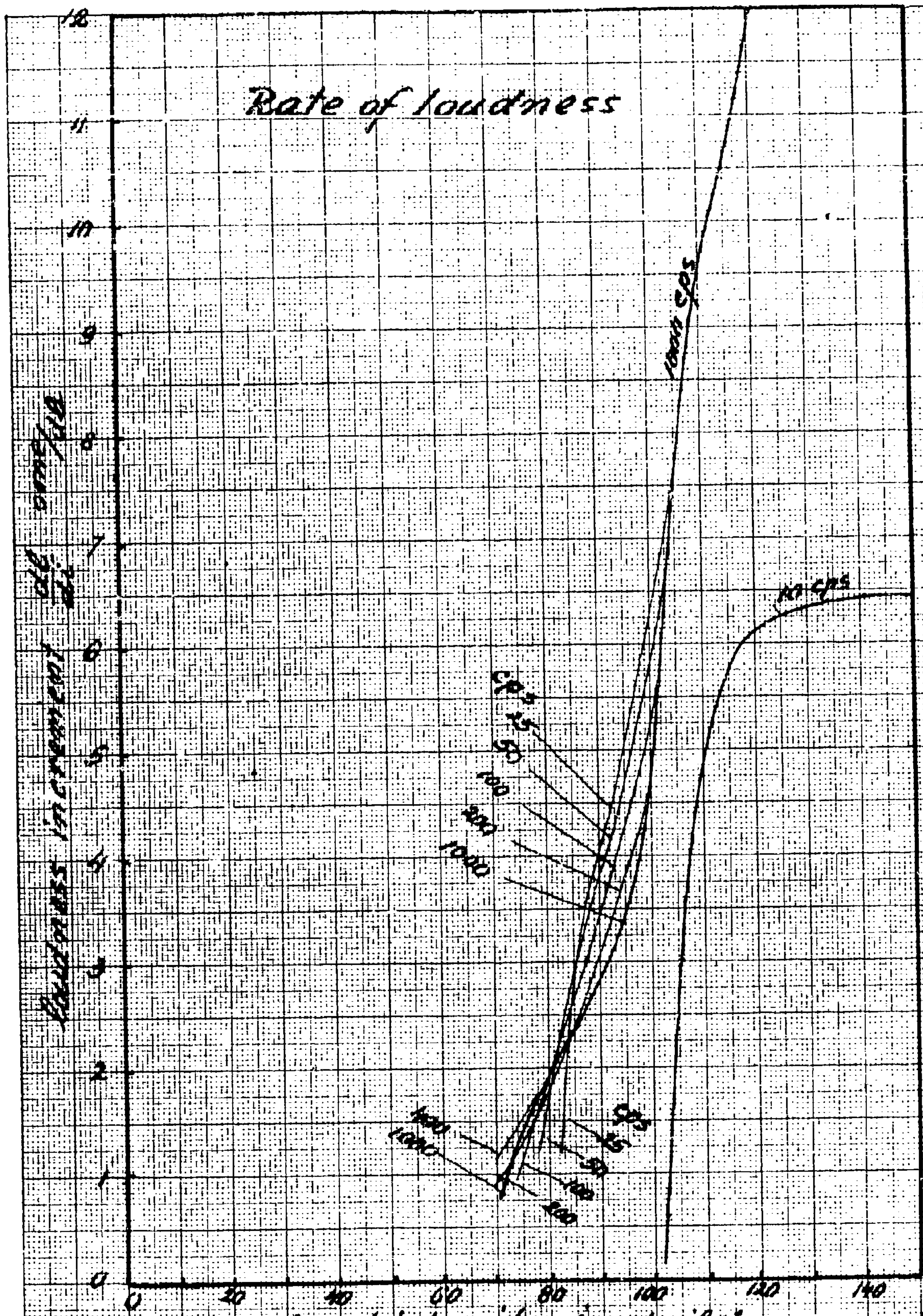
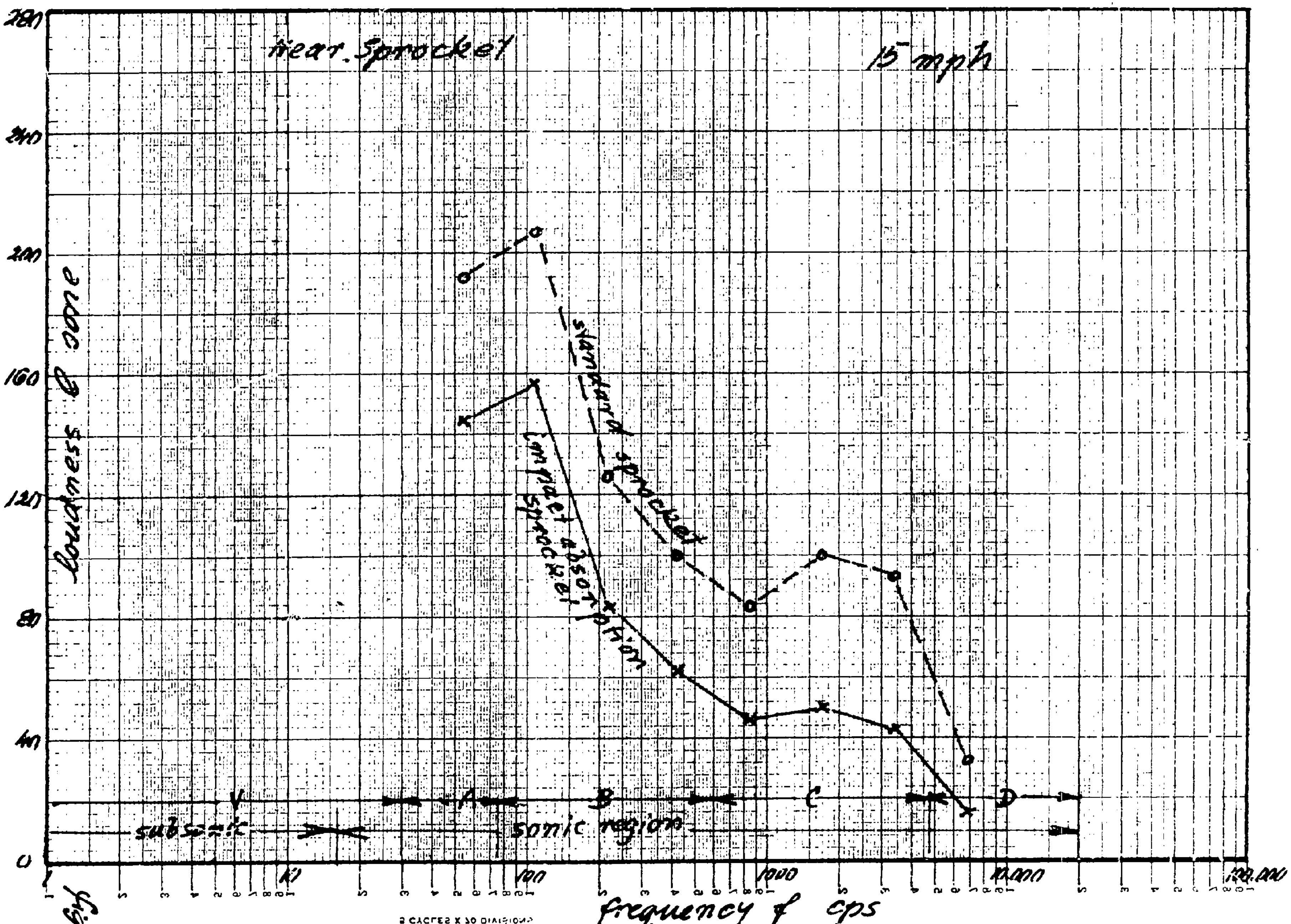


fig. 17



Rate of loudness
loudness increment dL per one dB
Sound intensity i decibel
1000 cps
10 cps
cps 25
50
100
200
1000
cps
25
50
100
200
400
1000
0
20
40
60
80
100
120
140
1
2
3
4
5
6
7
8
9
10
11
12

fig. 18



rear. Sprocket
15 mph
standard sprocket
impact absorption sprocket
loudness @ sone
280
240
200
160
120
80
40
0
1
10
100
1000
10,000
100,000
frequency f cps
subsonic
sonic region
A
B
C
D

fig. 19

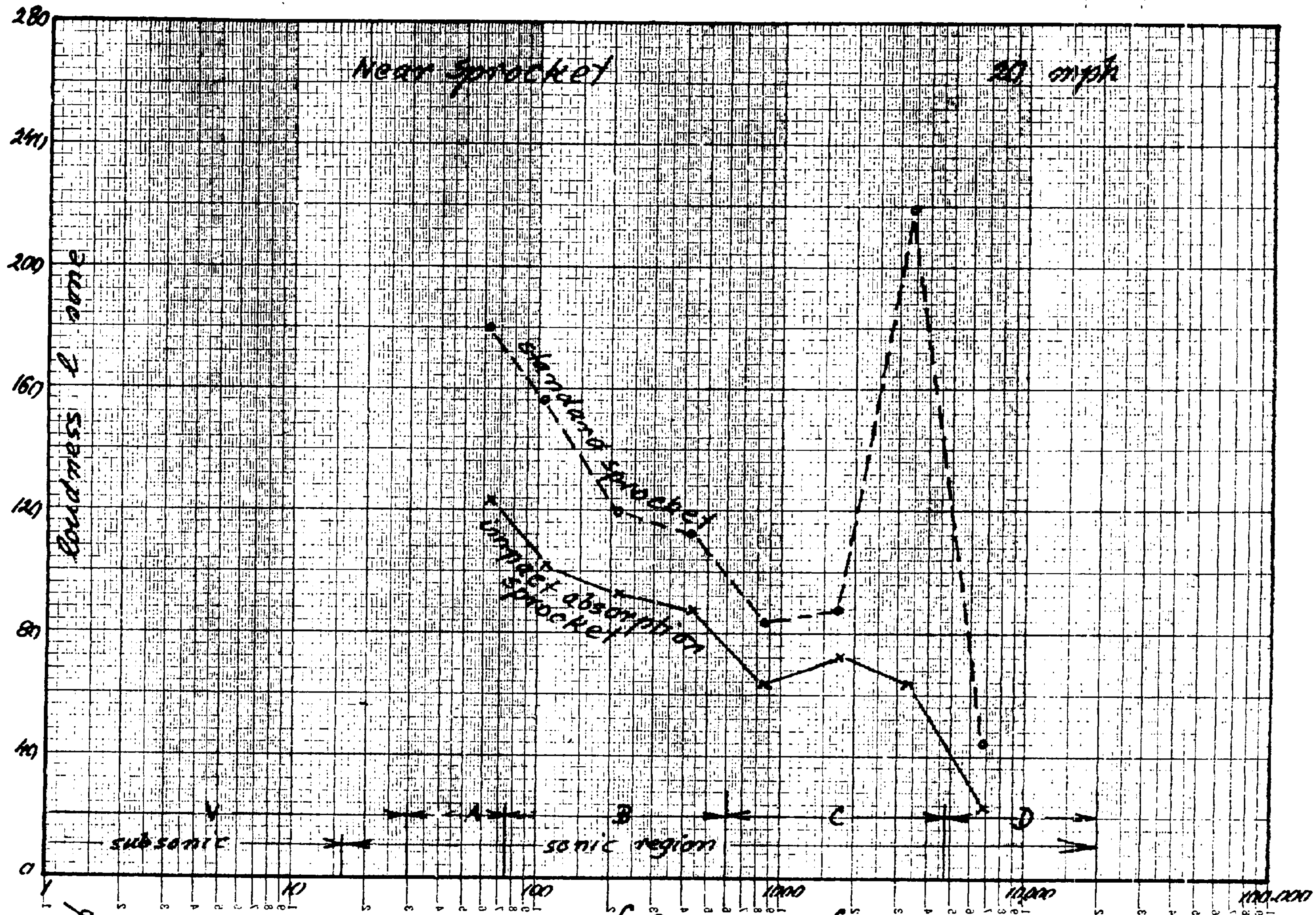

Near Sprocket
20 mph
standard sprocket
impact absorption sprocket
loudness of zone
frequency f cps
subsonic
sonic region
A
B
C
D
280
240
200
160
120
80
40
0
1
10
100
1000
10,000
100,000
K+E SEMI-LOGARITHMIC 359-81G

fig. 21

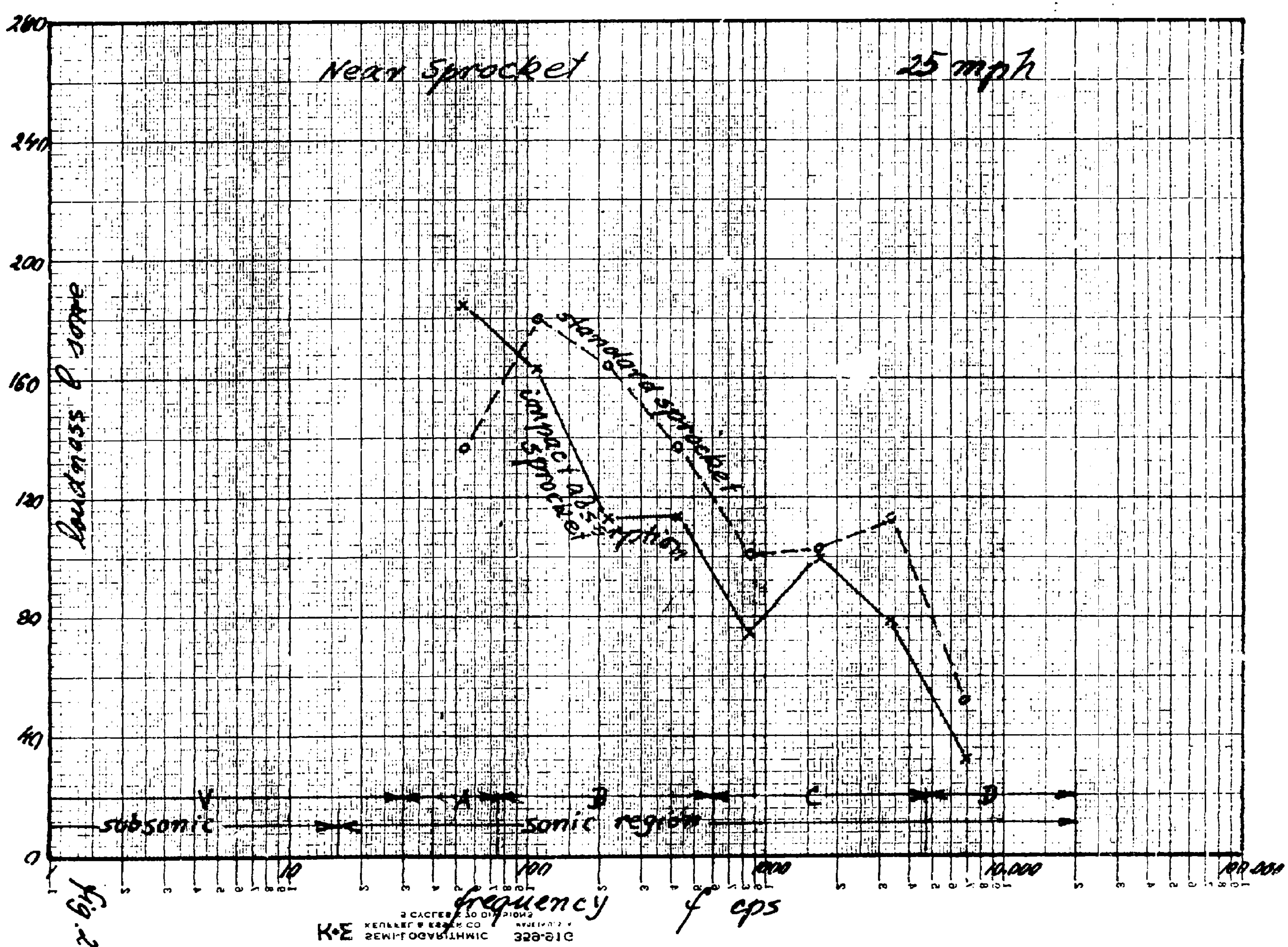

Near Sprocket
25 mph
standard sprocket
impact absorption sprocket
Loudness of tone
frequency f cps
subsonic
sonic region
V
A
B
C
D
0
40
80
120
160
200
240
260
1
10
100
1000
10,000
100,000
K+E KEUFFEL & ESSER CO.
SEMI-LOGARITHMIC 359-51G
5 CYCLES X 10 DIVISIONS

Fig. 21


rear sprocket 20 mph
Loudness L sone
frequency f. cps
subsonic region
sonic region
A
B
C
D
0
40
80
120
160
200
240
280
10
100
1000
10.000
100.000

Fig. 22

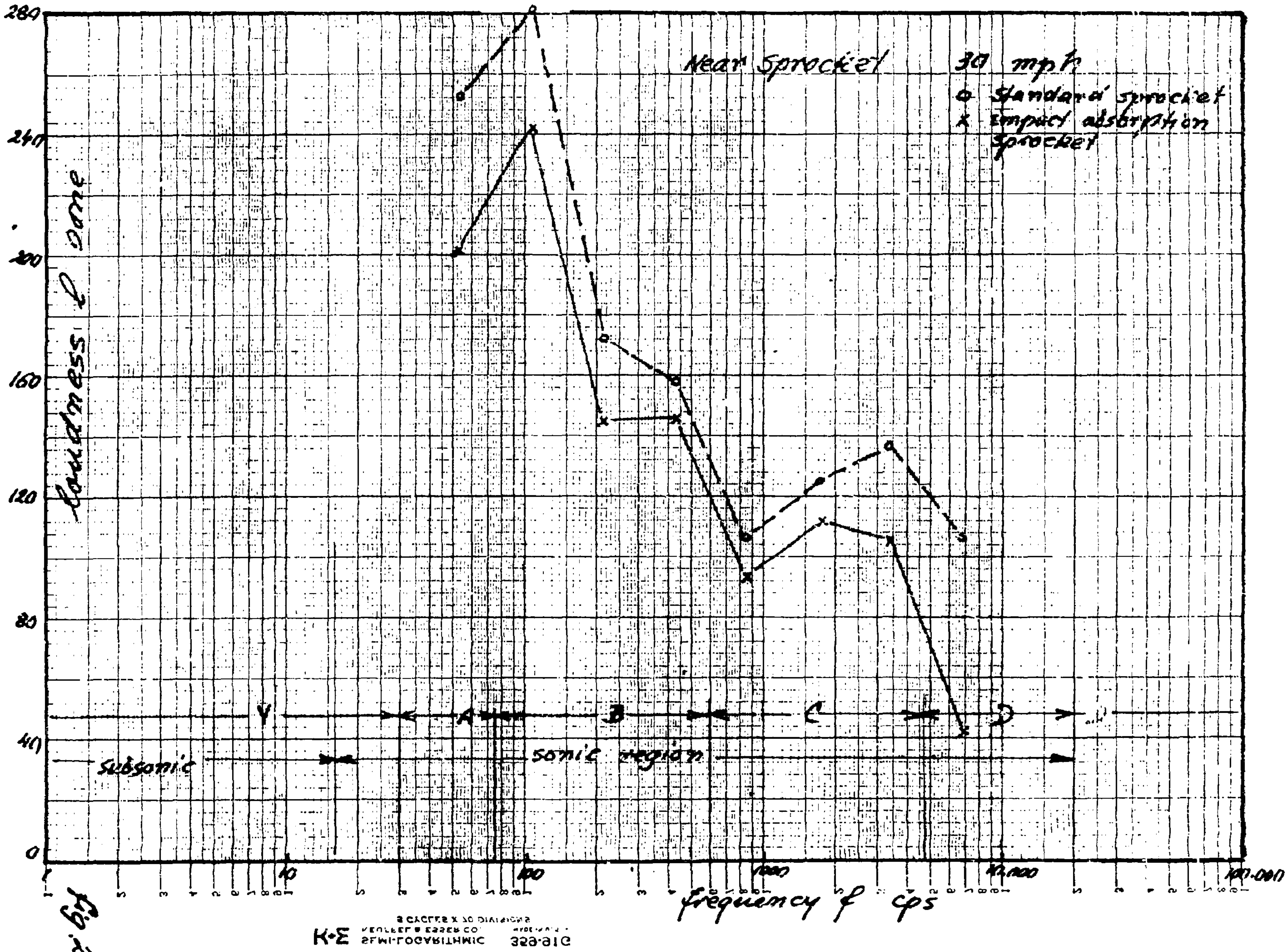
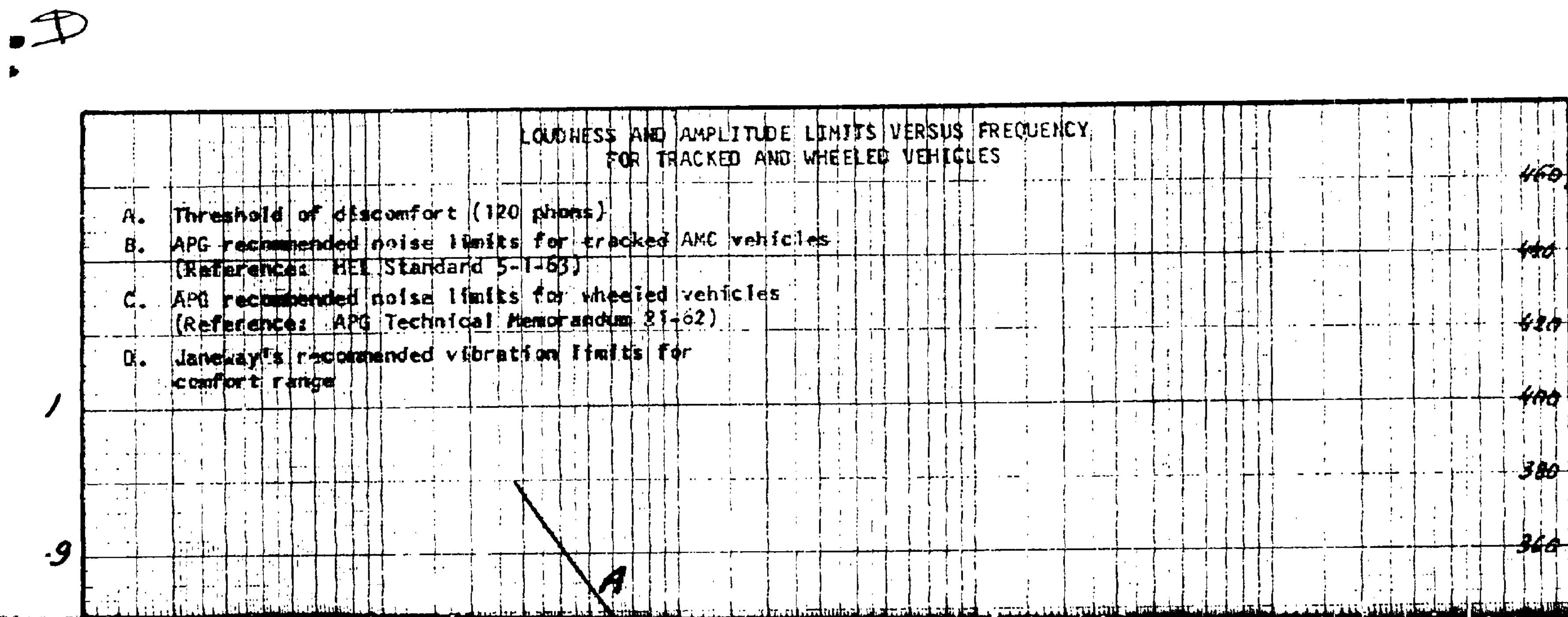

Near Sprocket
30 mph
o Standard sprocket
x Impact absorption sprocket
Loudness of sone
280
240
200
160
120
80
40
0
1
10
100
1000
10,000
100,000
frequency of cps
V
A
B
C
D
subsonic
sonic region
K+E KEUFFEL & ESSER CO. SEMI-LOGARITHMIC 359-31G 5 CYCLES X 10 DIVISIONS

Fig. 23



D
LOUDNESS AND AMPLITUDE LIMITS VERSUS FREQUENCY
FOR TRACKED AND WHEELED VEHICLES
A. Threshold of discomfort (120 phons)
B. APG recommended noise limits for tracked AMC vehicles
(Reference: HEL Standard 5-1-63)
C. APG recommended noise limits for wheeled vehicles
(Reference: APG Technical Memorandum 21-62)
D. Janeway's recommended vibration limits for
comfort range
1
.9
460
440
420
400
380
360
A

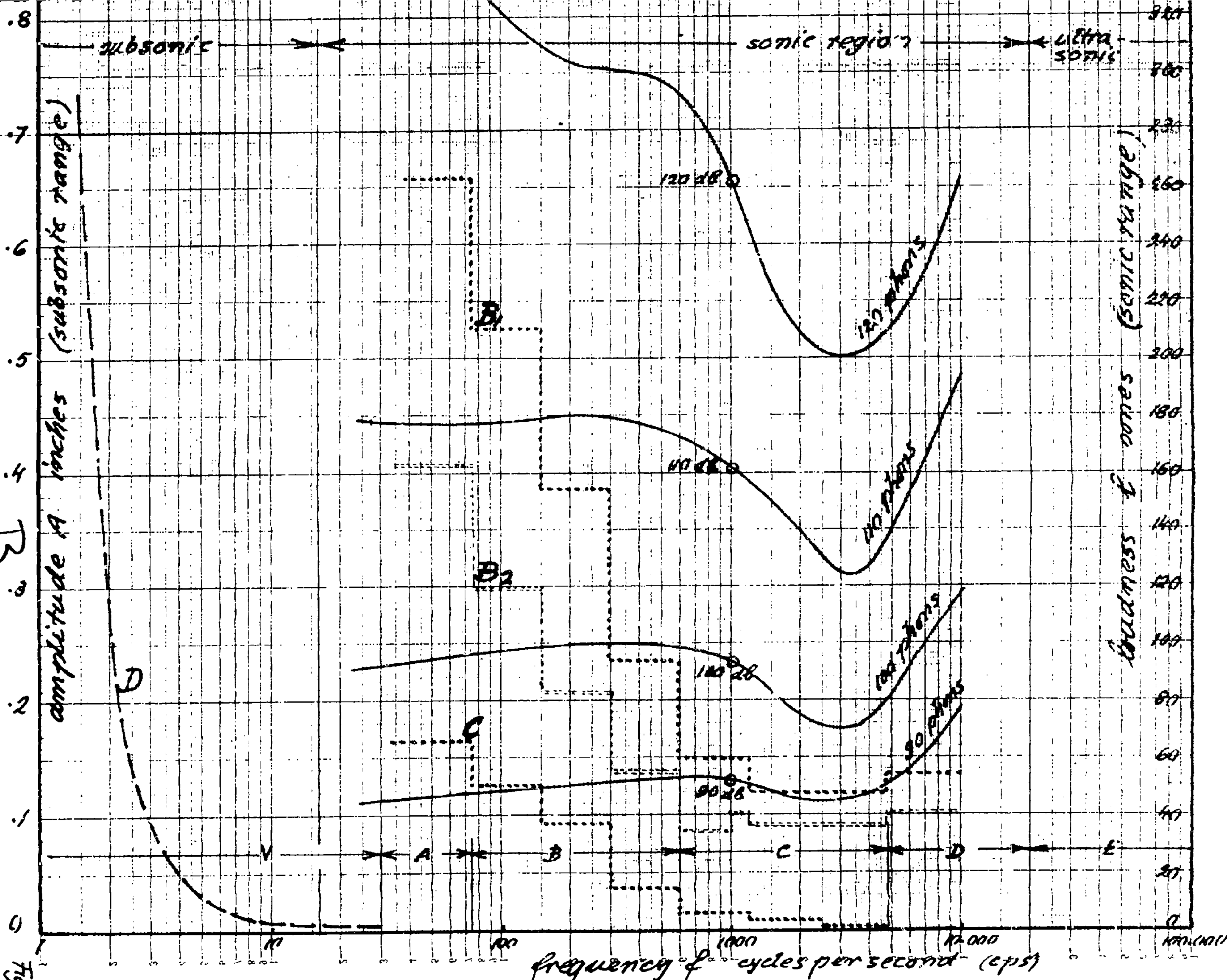
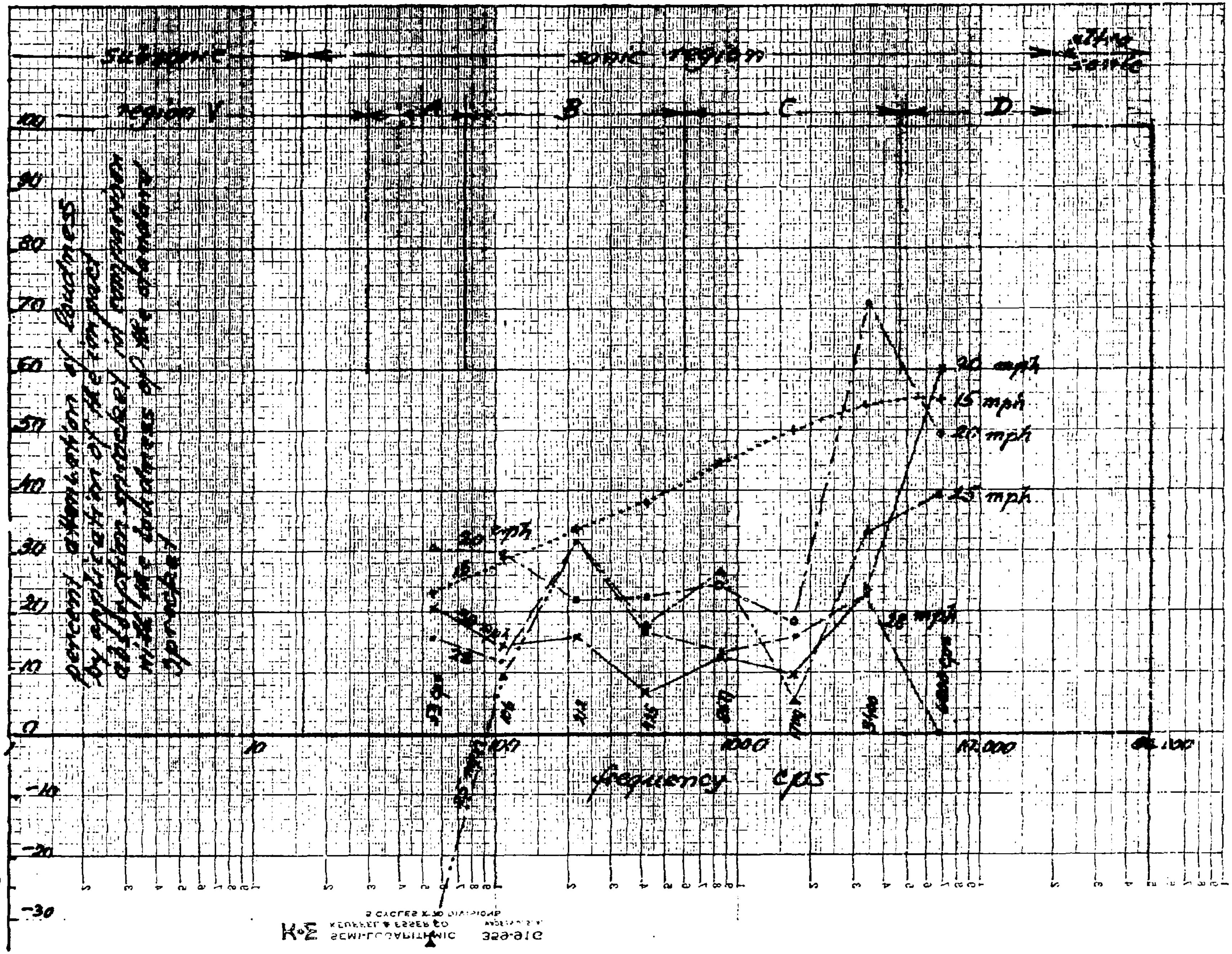
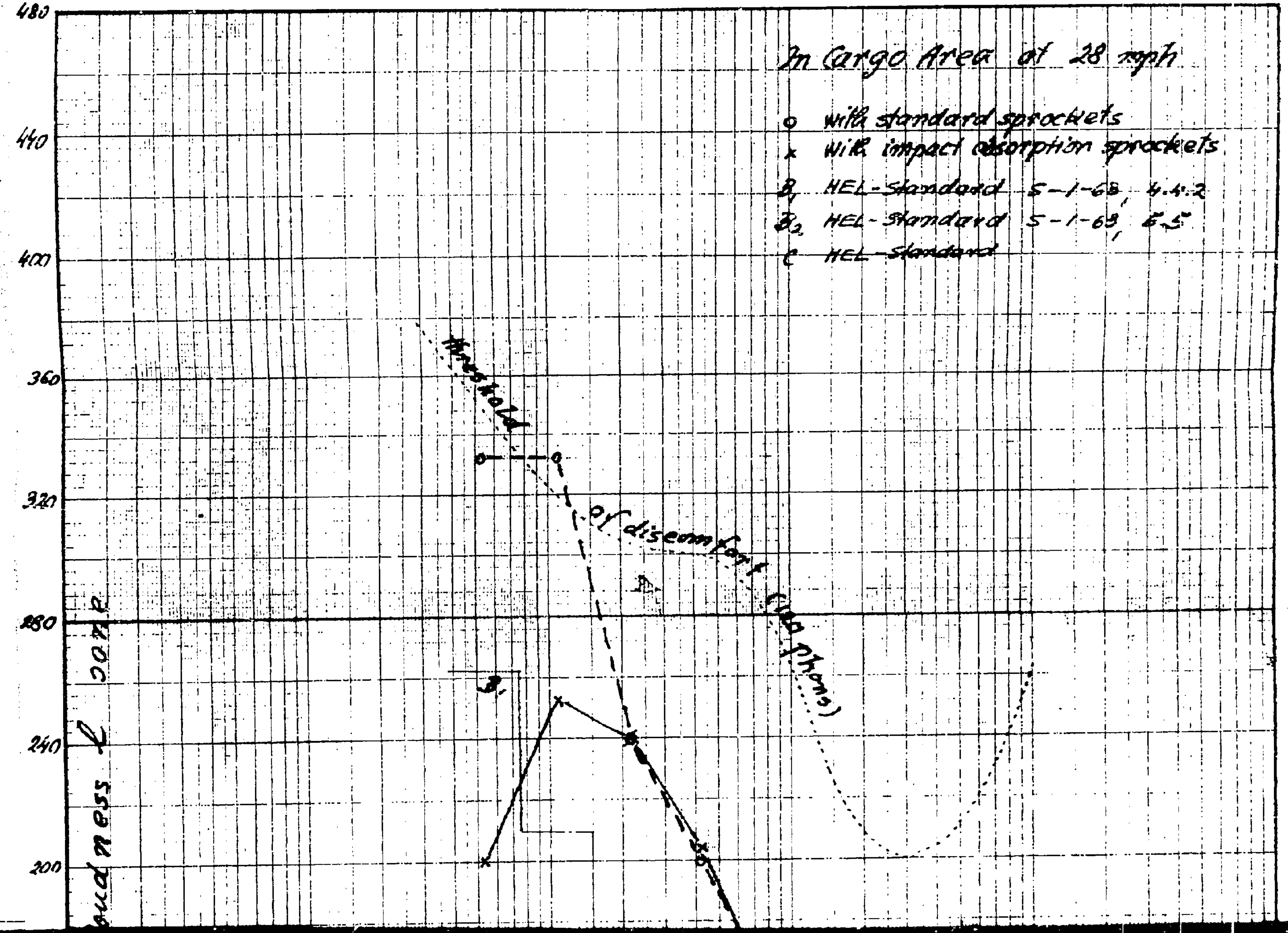

subsonic
sonic region
ultra sonic
amplitude A inches (subsonic range)
loudness of ones (sonic range)
frequency f cycles per second (cps)
120 dB
110 dB
100 dB
90 dB
120 phons
110 phons
100 phons
90 phons
B1
B2
C
D
V
A
B
C
D
E
1
10
100
1000
10,000
100,000
.8
.7
.6
.5
.4
.3
.2
.1
0
200
160
140
120
100
80
60
40
20
0
B
K+E SEMI-LOGARITHMIC 359-91G

Fig. 24



sonic region
region V
A
B
C
D
percent attenuation of loudness
by application of the improved
attenuation sprocket, as compared
with the loudness of the standard
sprocket
100
90
80
70
60
50
40
30
20
10
0
-10
-20
-30
10
100
1000
12,000
frequency cps
30 mph
15 mph
20 mph
25 mph

fig. 25



In Cargo Area at 28 mph
o with standard sprockets
x with impact absorption sprockets
B_1 HEL-Standard 5-1-68, 4.4.2
B_2 HEL-Standard 5-1-68, 5.5
C HEL-Standard
threshold of discomfort (120 phons)
A
B_1
480
440
400
360
320
280
240
200
Loudness L sone

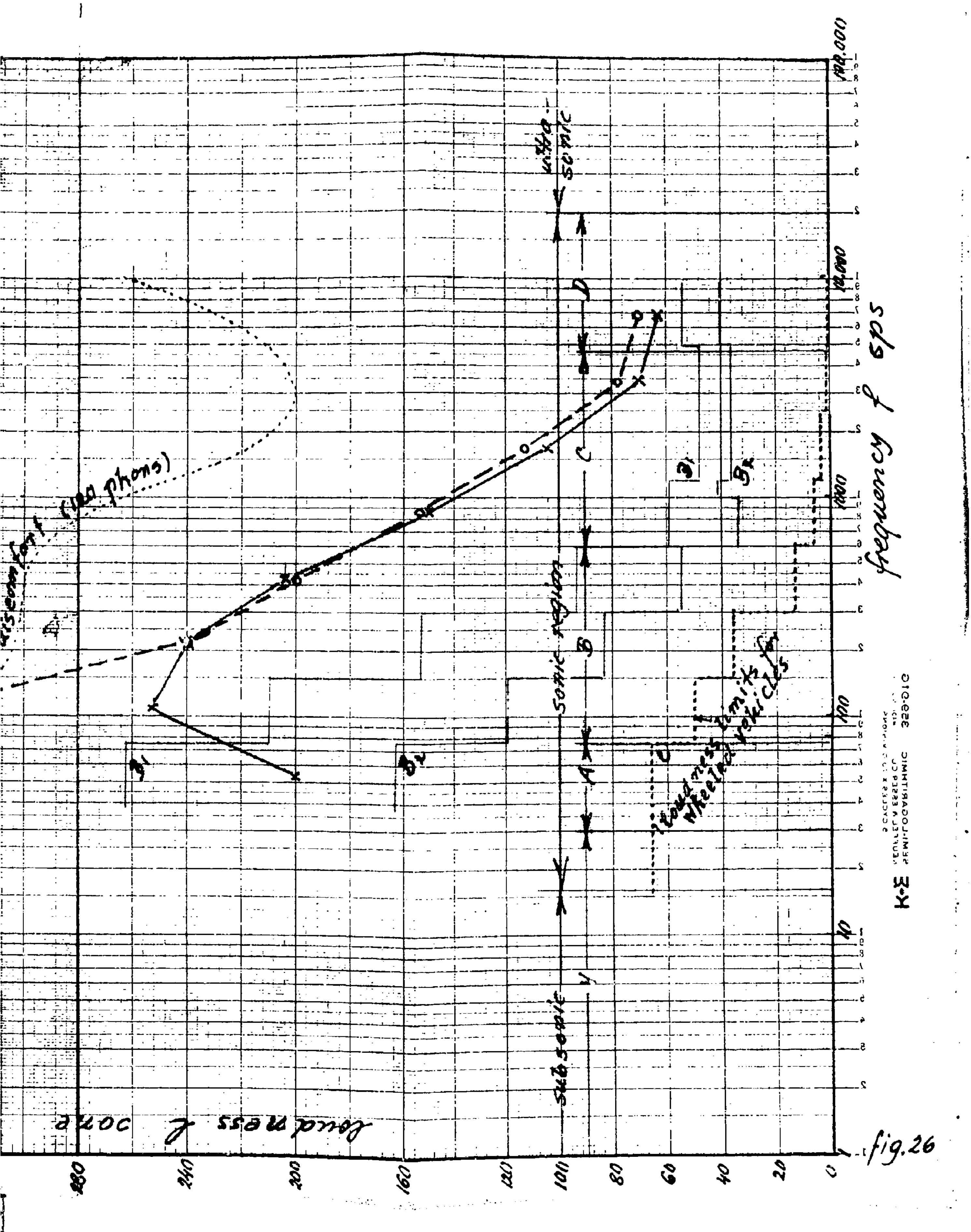

loudness of zone
frequency of cps
discomfort (120 phons)
sub sonic
sonic region
ultra sonic
loudness limits for wheeled vehicles
A
B
C
D
0
20
40
60
80
100
120
160
200
240
280
10
100
1000
10,000
100,000

fig.26

Excess of Loudness with Respect to
Permissible B_2 - Level

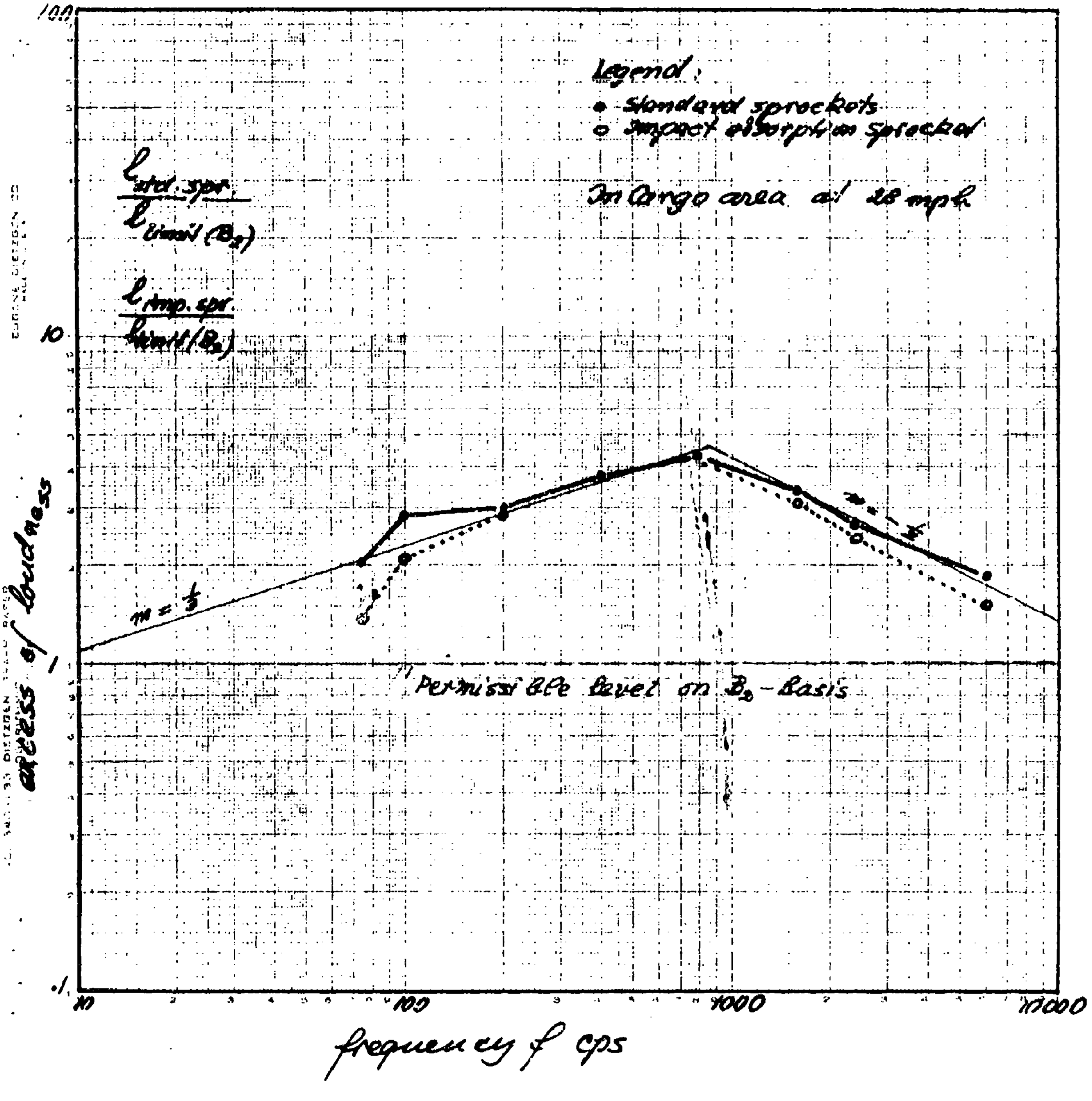


fig. 27

Hearing perception ability test

Dr. S. Rosen, [illegible] and A. El-Mofty
Archives of Otolaryngology
(Chicago Jan. 1964)

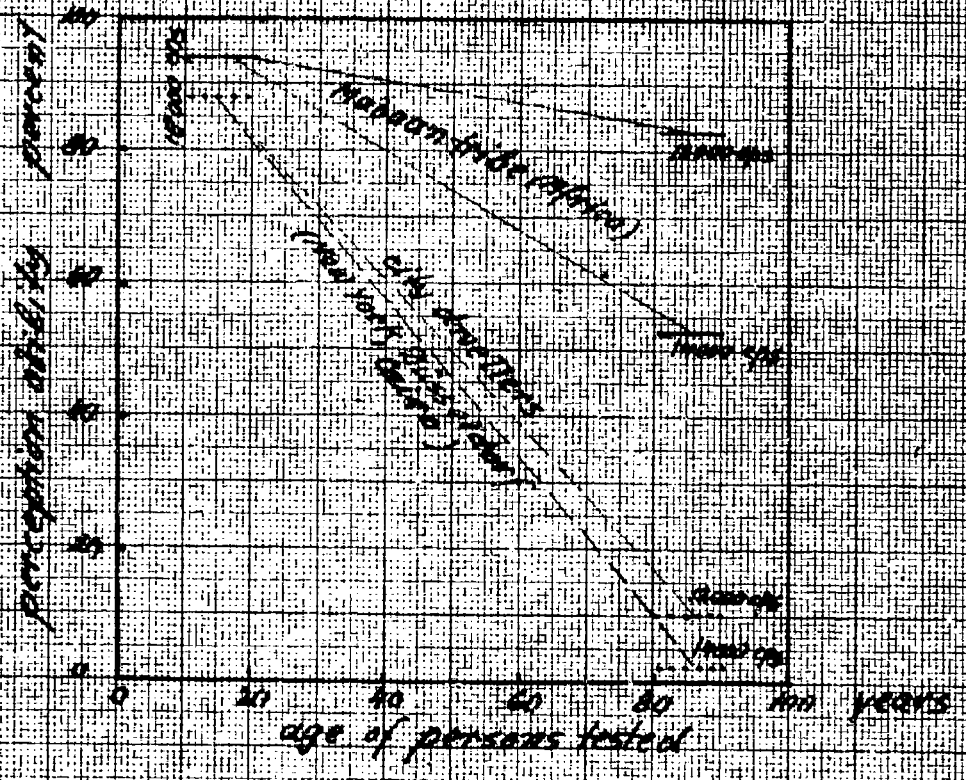


fig 28

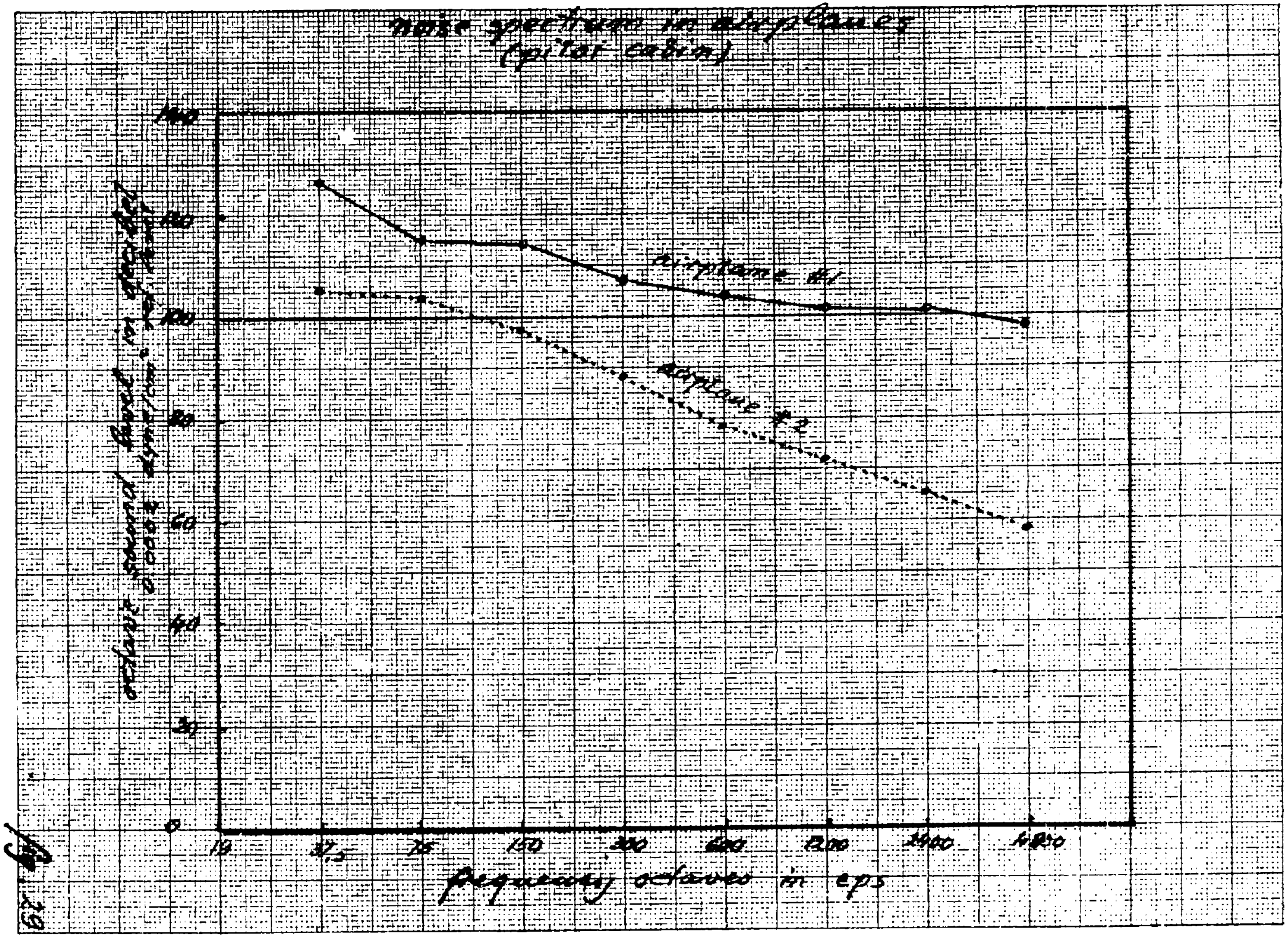

Noise spectrum in airplanes
(pilot cabin)
Airplane #1
Airplane #2
octave sound level in db re 0.0002 dyne/cm²
140
120
100
80
60
40
20
0
19
37.5
75
150
300
600
1200
2400
4800
frequency octaves in cps

Fig. 29

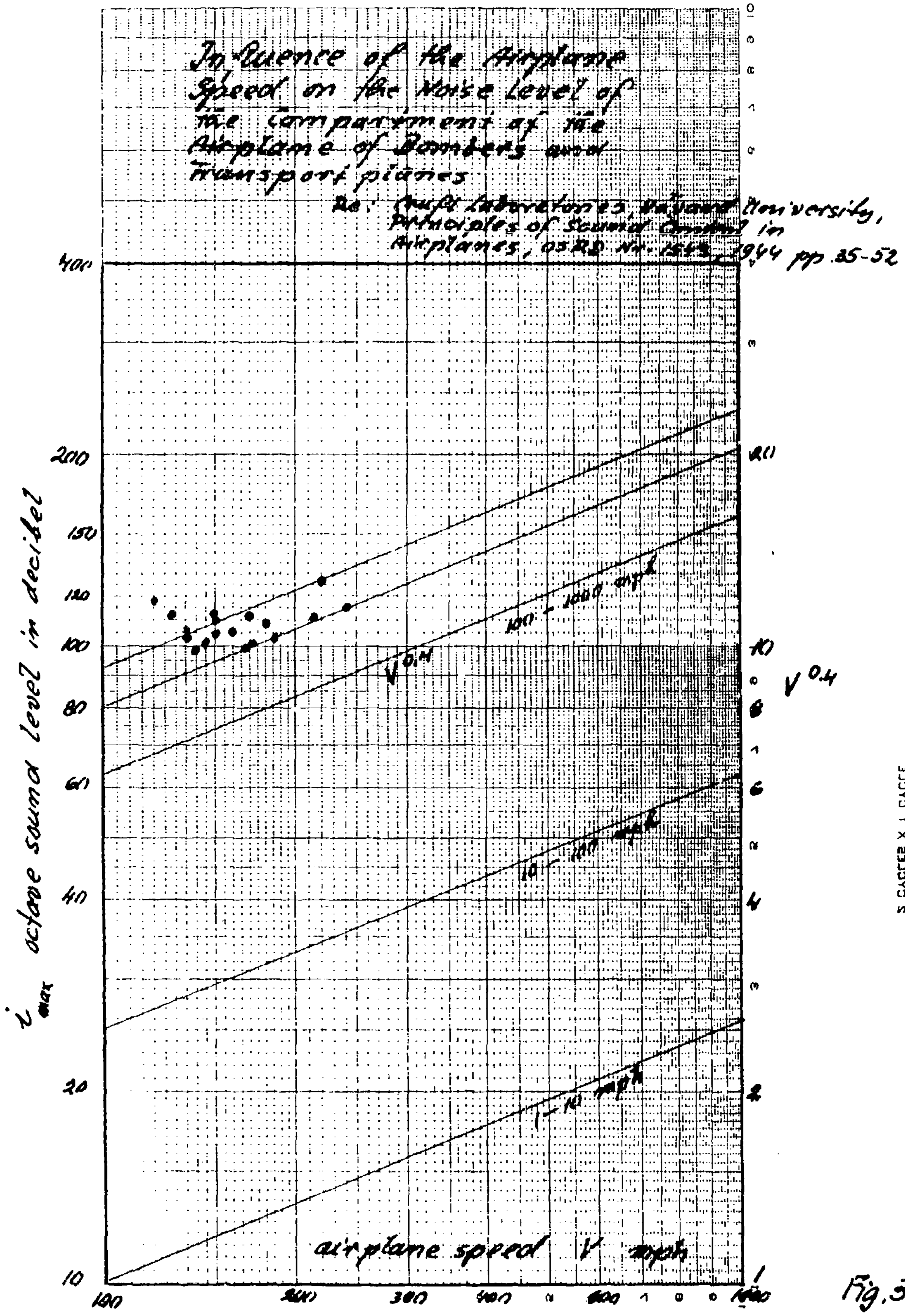
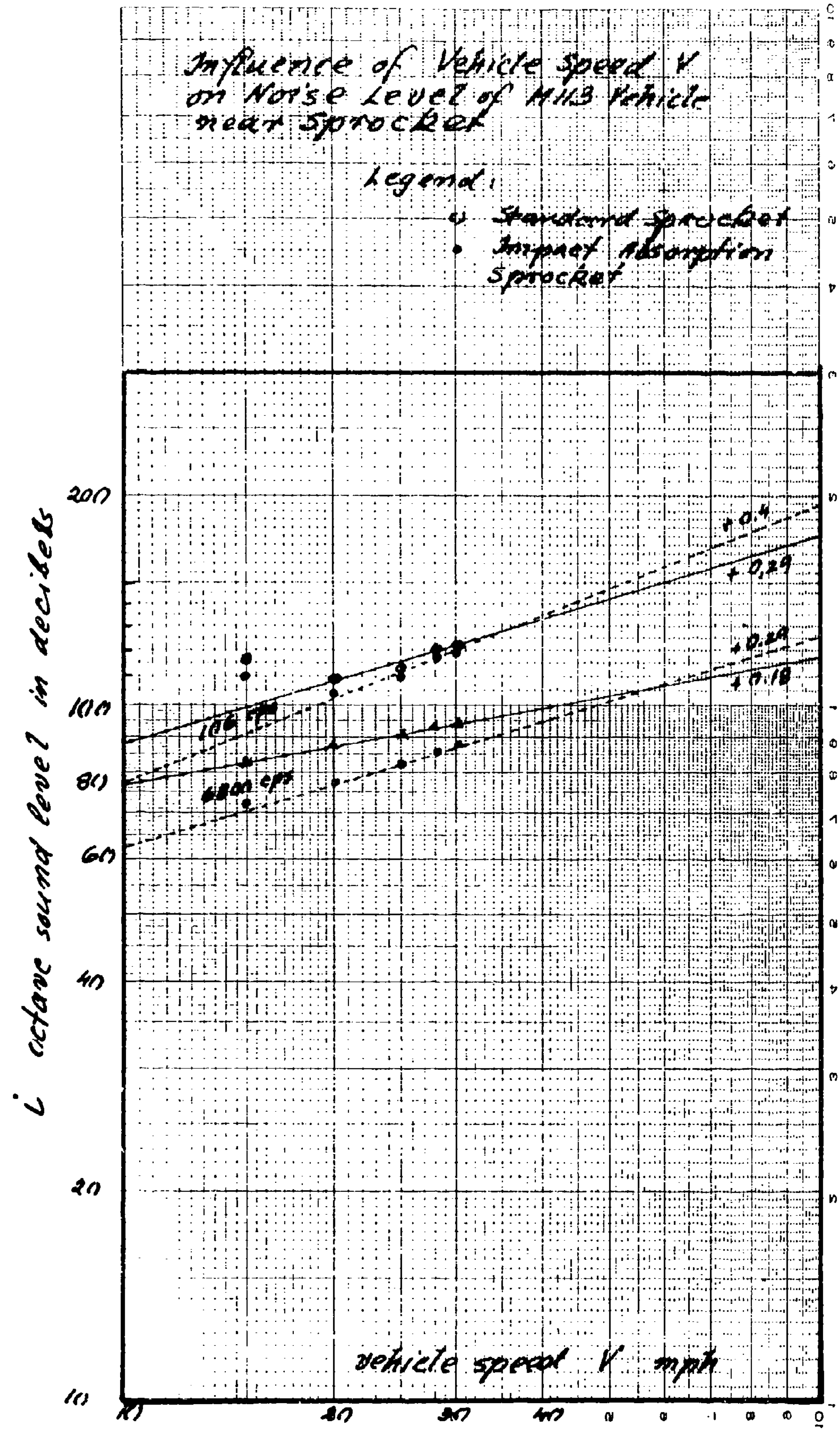

Influence of the Airplane Speed on the Noise Level of the Compartment of the Airplane of Bombers and Transport planes
Re: Cruft Laboratories, Harvard University, Principles of Sound Control in Airplanes, OSRD Nr. 1543, 1944 pp. 35-52
i_{max} octave sound level in decibel
airplane speed V mph
100 - 1000 mph
10 - 100 mph
1 - 10 mph
$V^{0.4}$
400
200
150
120
100
80
60
40
20
10
300
400
600
1000

Fig. 30



Influence of Vehicle Speed V
on Noise Level of M113 Vehicle
near Sprocket
Legend:
Standard Sprocket
Impact Absorption
Sprocket
+0.4
+0.29
+0.24
+0.18
100 cps
4800 cps
200
100
80
60
40
20
10
10
20
30
40
octave sound level in decibels
vehicle speed V mph

Fig. 31

COMPARISON OF M113 NOISE LEVEL WITH THAT
IN A PILOT CABIN OF A FIGHTER AIRPLANE WITH EXCESSIVE NOISE LEVEL

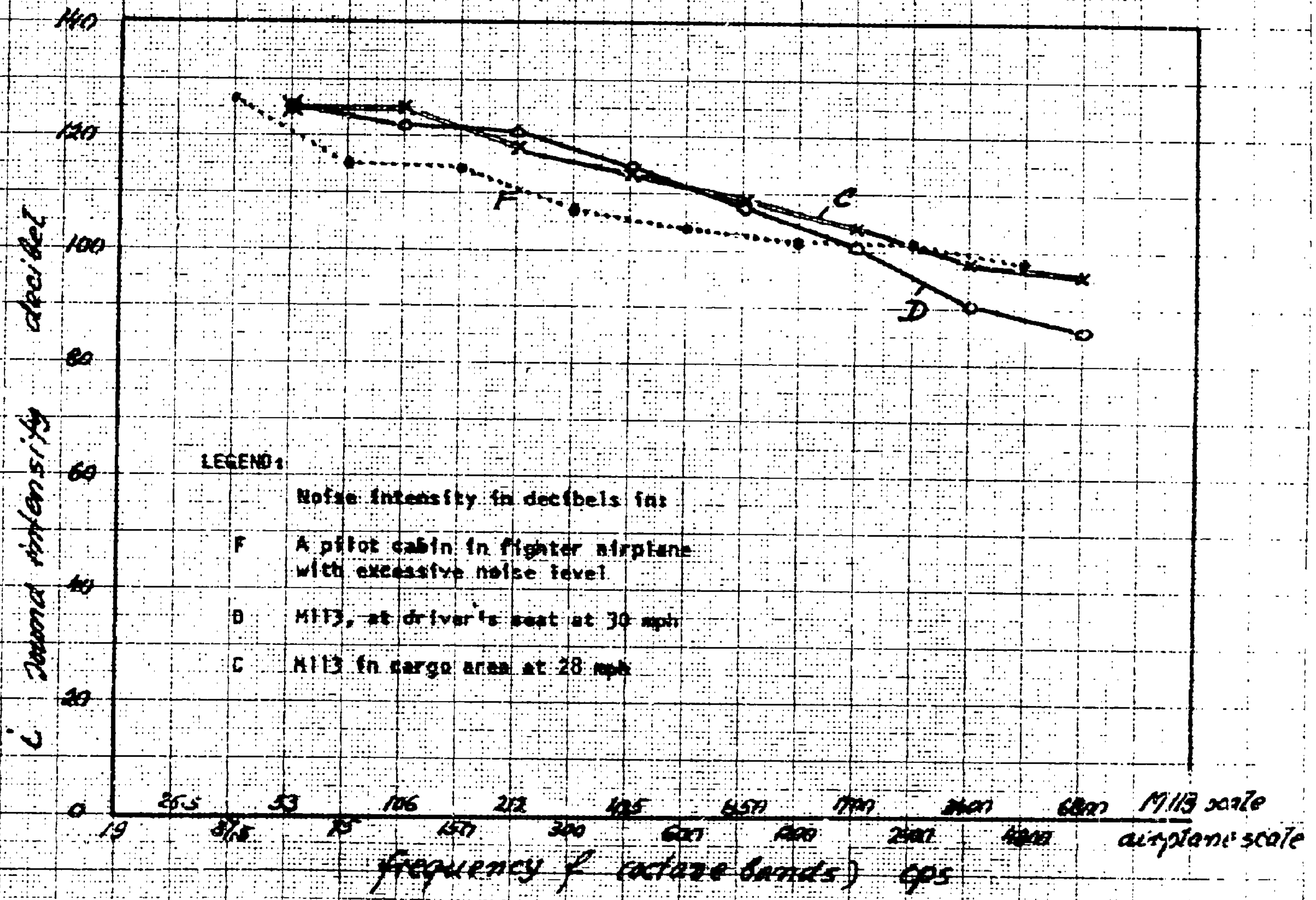


Fig. 32

VIBRATION LIMITS FOR WHEELED VEHICLES

Reference: U. S. Army, Technical Memorandum 21-62, "Manual of Standard Practice for Human Factors in Military Vehicle Design", dated August, 1962, P. 97, Fig. 6, Human Engineering Laboratories, Aberdeen Proving Ground, Maryland

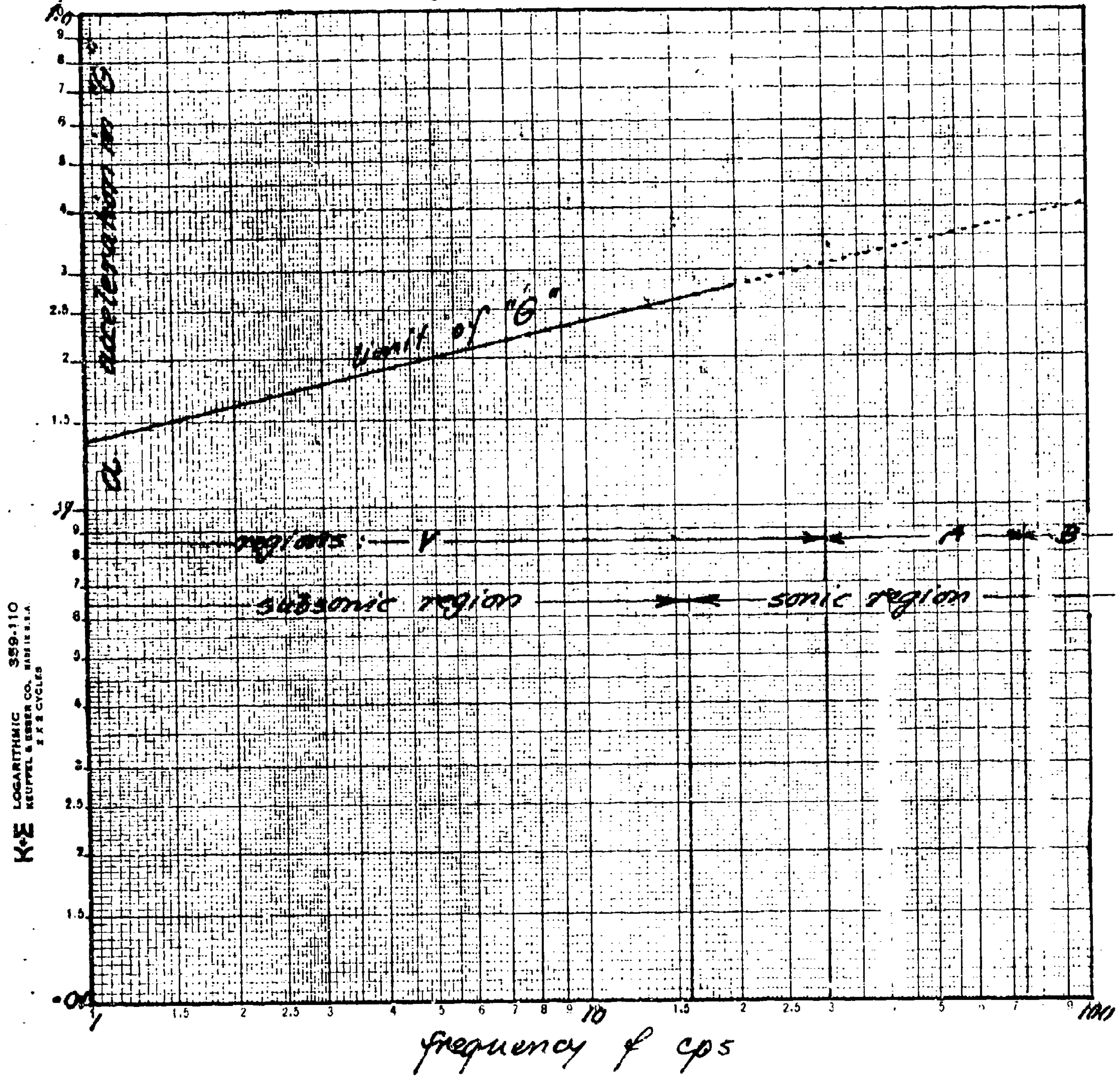
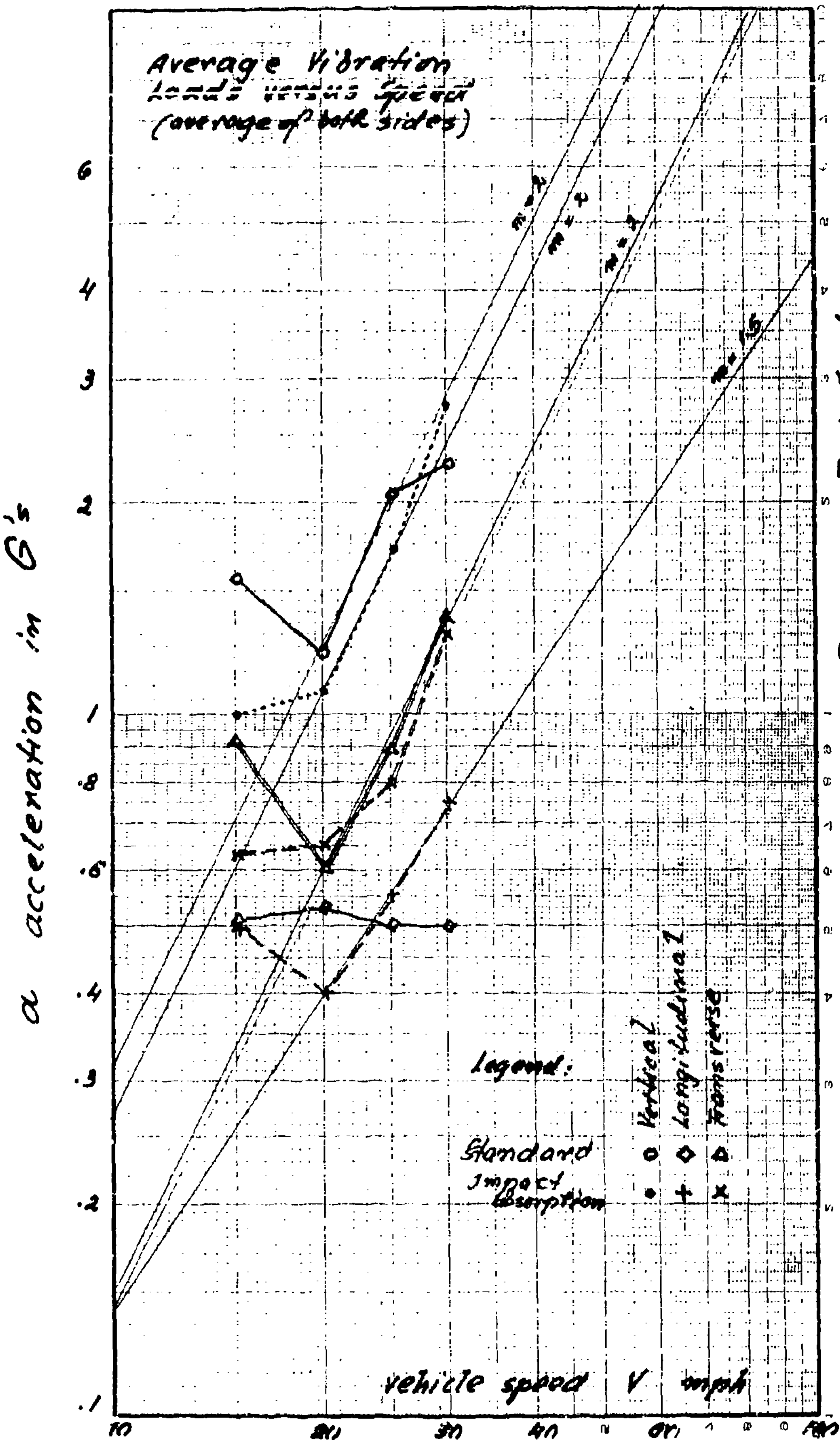
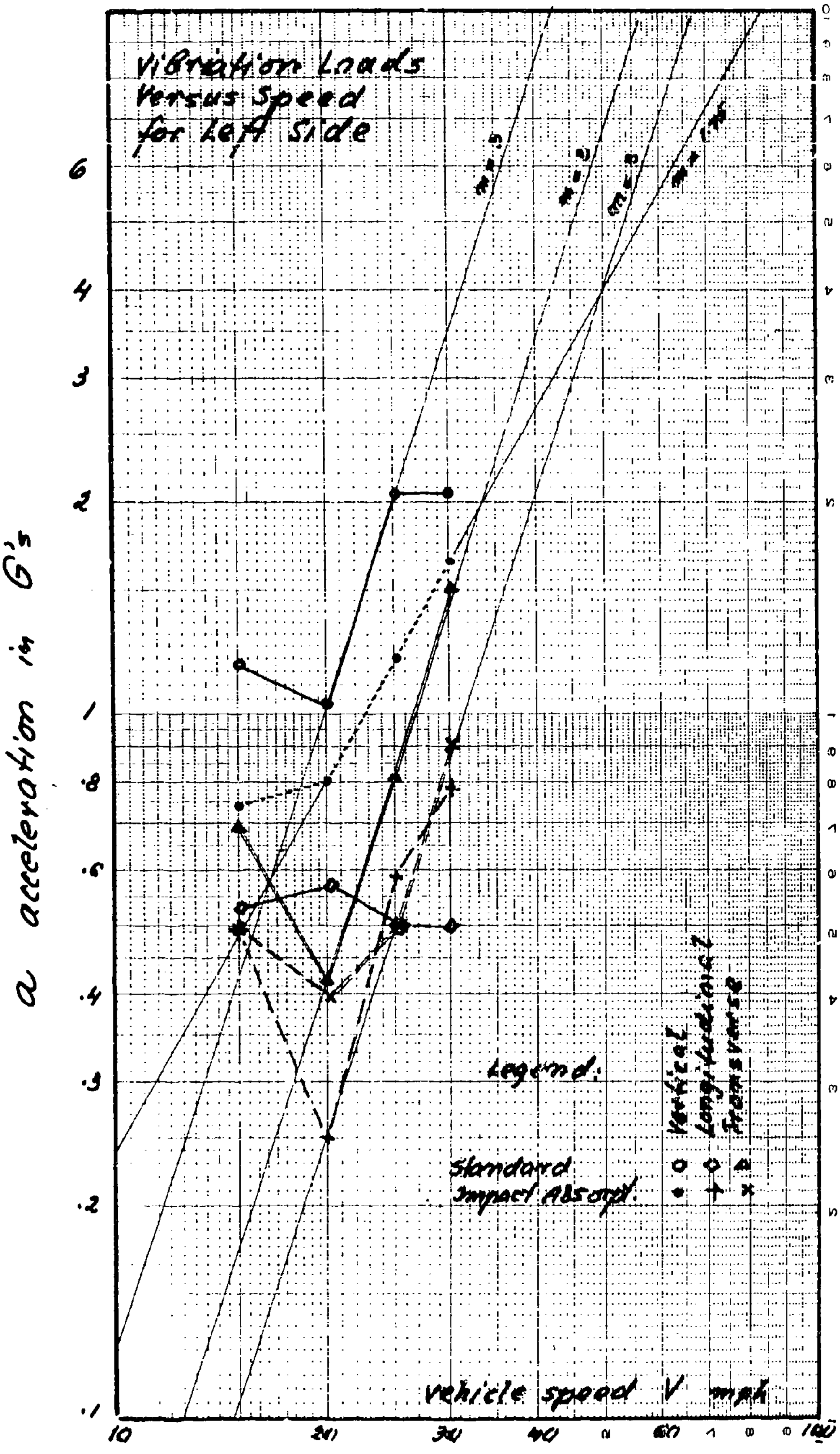


fig. 33



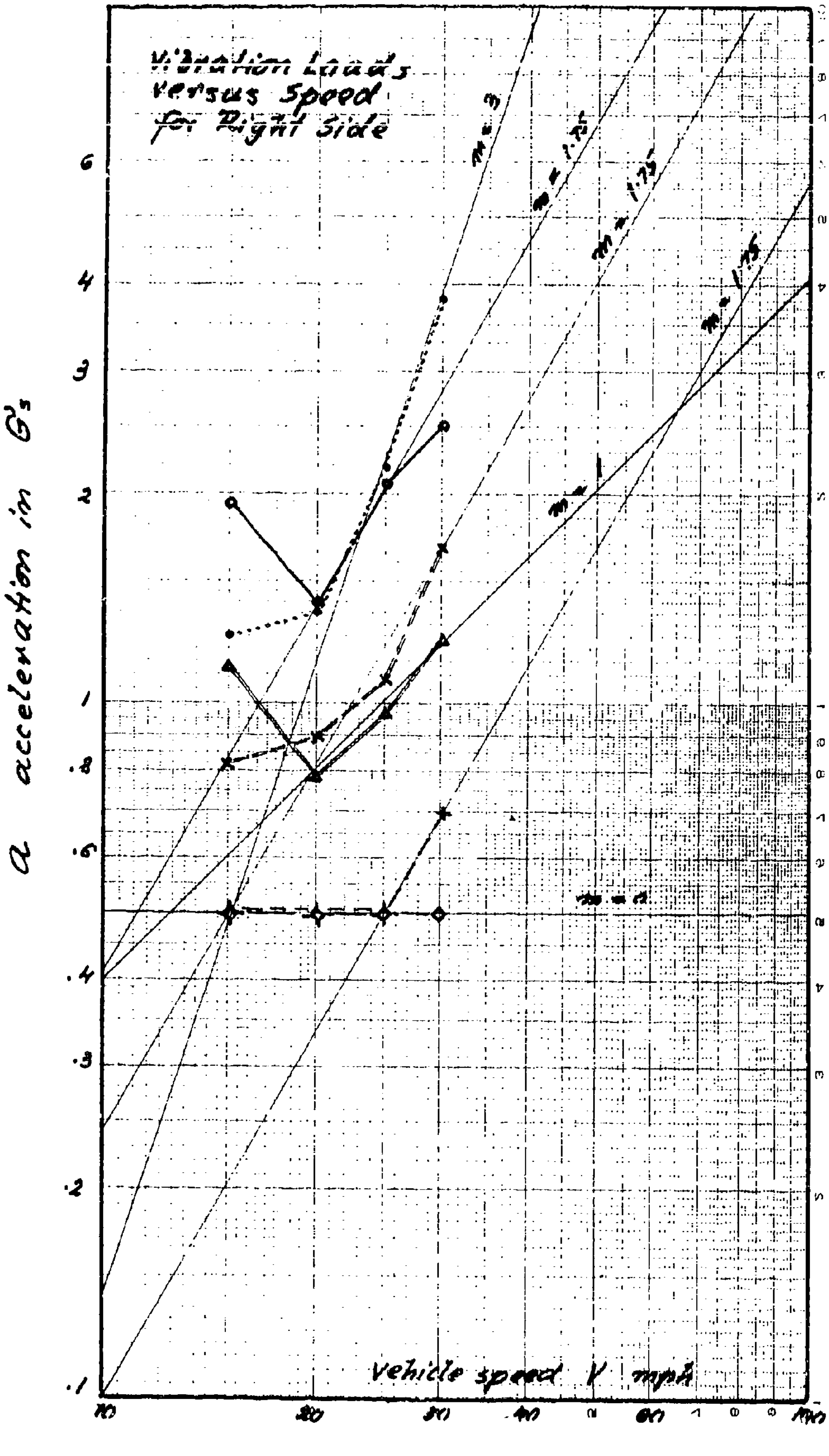
Re.: FMC Test sheet
Impact Absorption Sprocket
Dated 12-10-1963
Project Nr. 411-1-326

Fig. 34



Re: FMC Test sheet
Impact Absorption Sprocket
Dated 12-10-1963 (FASANO)
Project 411-1-426

Fig. 35



Re.: FMC Test sheet
Impact Absorption Sprocket
Dated 12-10-1963 (Fasano)
Project Nr. 411-1-426

Fig. 36

Vibration Loads Versus Speed

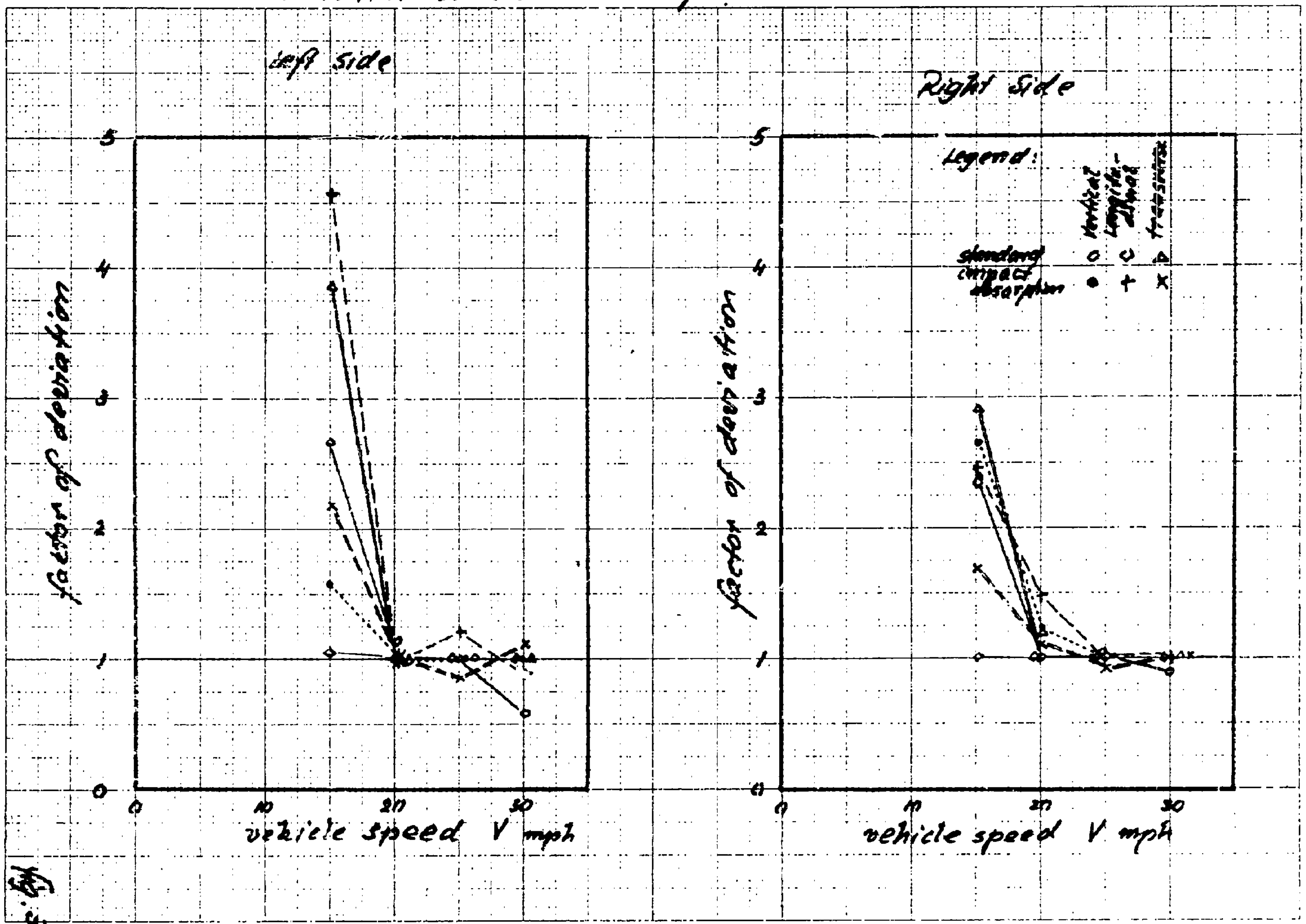
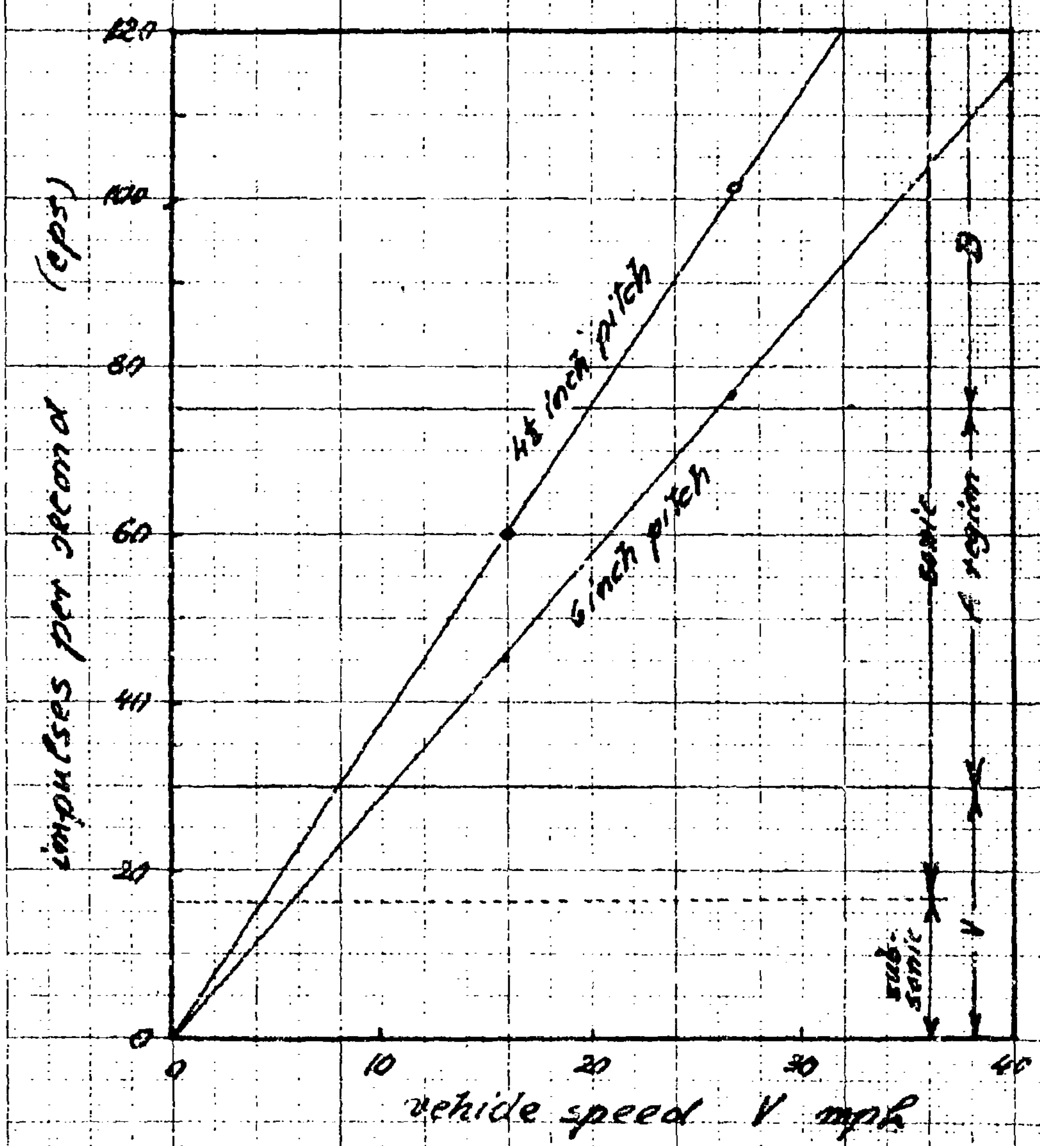
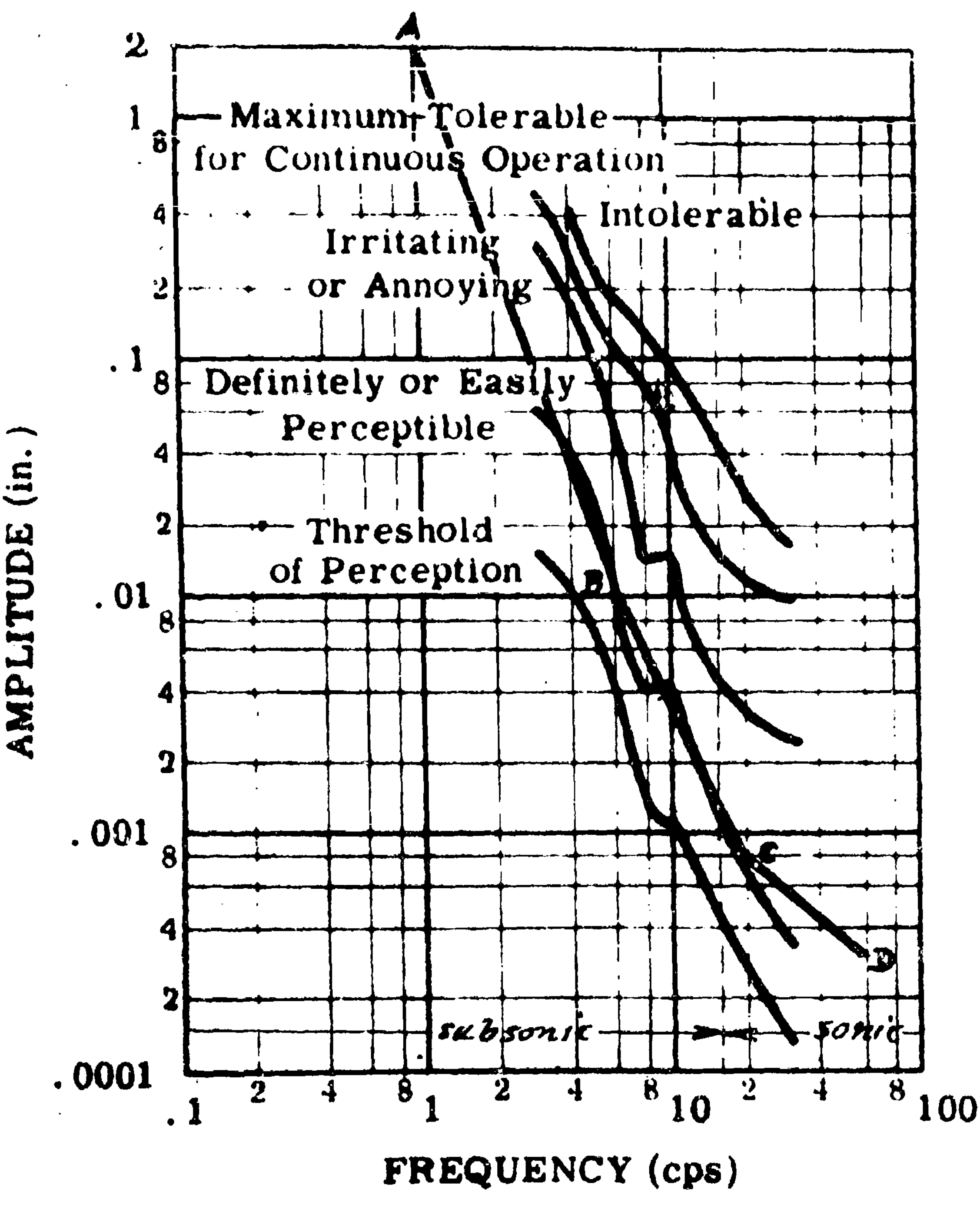


Fig. 37


impulses per second (cps)
120
100
80
60
40
20
0
0
10
20
30
40
vehicle speed V mph
4½ inch pitch
6 inch pitch
sub-sonic
sonic
region
Fig. 38



REFERENCE:

1. [illegible] of Human [illegible] Vi[illegible] Behavior: [illegible], [illegible] and [illegible] Sattinger: "Human [illegible]." Report of Institute of Science and Technology of the University of Michigan, [illegible], Figure [illegible], page 45.

2. [illegible] Comfort [illegible]: [illegible] Janeway: "Vehicle Vibration Limits to Fit the Passenger" Report [illegible], SAE [illegible], March 1948, Abstract in SAE Journal, August [illegible].

Fig. 39

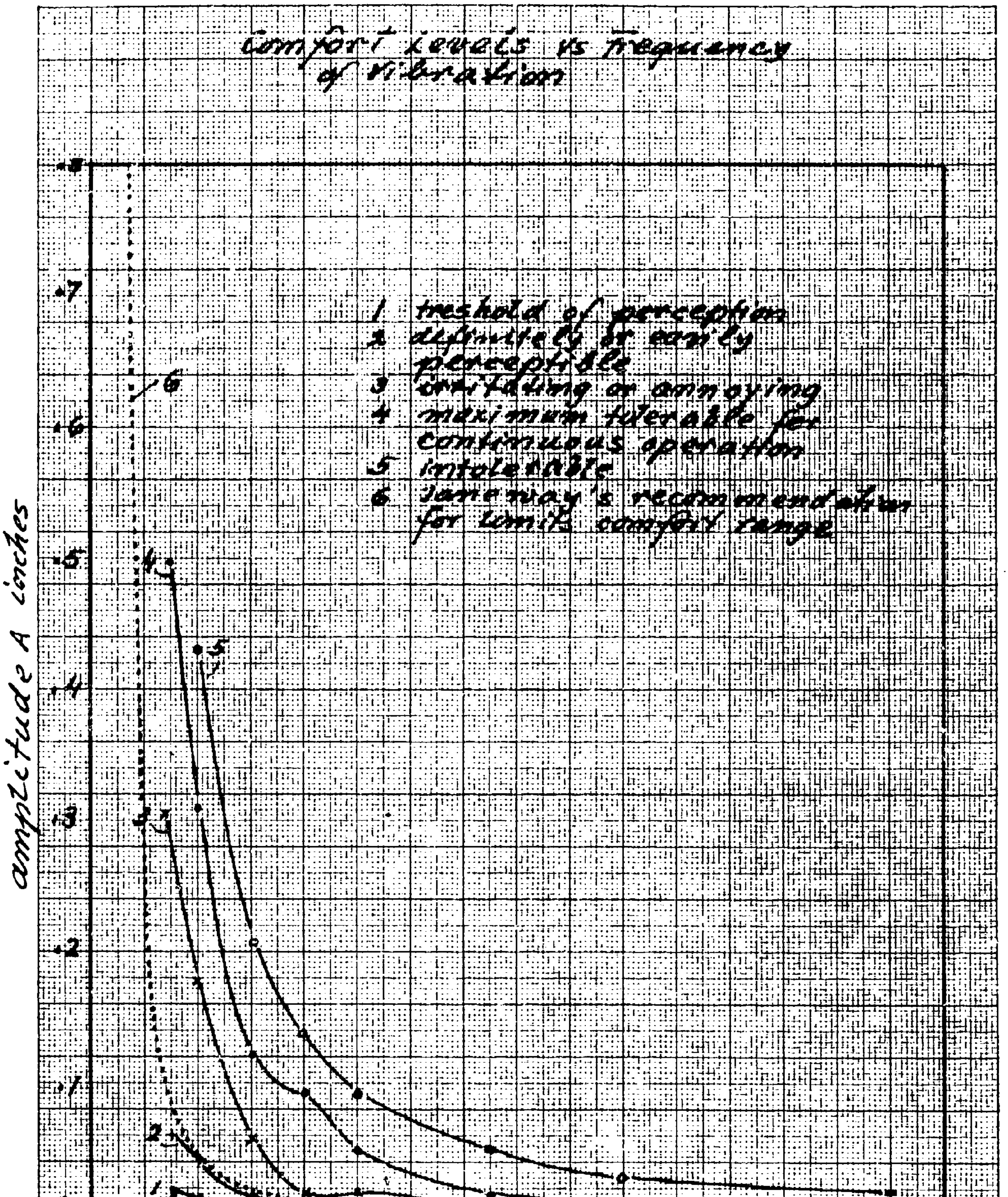

comfort levels vs frequency of vibration
1 treshold of perception
2 definitely or easily perceptible
3 irritating or annoying
4 maximum tolerable for continuous operation
5 intolerable
6 Janeway's recommendation for limits comfort range
amplitude A inches
.8
.7
.6
.5
.4
.3
.2
.1
0
0
2
4
6
8
10
15
20
25
30
frequency f cps

Fig. 40

Attachment #1

FMC Corporation

Ordnance Division, San Jose, California

Sound Levels Versus Frequency Octaves of The M113 Armored Infantry Carrier Equipped with Standard or with Impact Absorption Sprockets

Test Results dated 10 December 1963

By R. FASANO

Subject: IMPACT ABSORBTION SPROCKET TEST

Prepared by R. FASANO Date 12-10-63 Code No.

Checked by Dwg. No. Project No. 411-1-426

VEHICLE SPEED (MPH)	AVG.* VIBRATION JAD G's					
	VERTICAL		LONGITUDINAL		TRANSVERSE	
	STD.	EXP.	STD.	EXP.	STD.	EXP.
15	1.56	1.0	0.51	0.5	0.92	0.63
20	1.22	1.03	0.53	0.4	0.60	0.65
25	2.05	1.7	0.5	0.55	0.89	0.8
30	2.29	2.73	0.5	0.75	1.38	1.3

*MEASUREMENTS ON RIGHT & LEFT SIDES WERE AVERAGED

Subject: ______

Prepared by ______ Date ______ Code No. ______

Checked by ______ Dwg. No. ______ Project No. ______

VEHICLE SPEED (MPH)	RIGHT SIDE VIBRATION LOAD G's					
	VERTICAL		LONGITUDINAL		TRANSVERSE	
	STD	EXP	STD	EXP	STD	EXP
15	1.94	1.25	0.5	0.5	1.13	0.82
20	1.39	1.36	0.5	0.5	0.78	0.89
25	2.03	2.19	0.5	0.5	0.96	1.08
30	2.50	3.81	0.5	0.7	1.28	1.69

VEHICLE SPEED (MPH)	LEFT SIDE VIBRATION LOAD G's					
	VERTICAL		LONGITUDINAL		TRAVERSE	
	STD	EXP	STD	EXP	STD	EXP
15	1.18	0.75	0.53	0.5	0.7	0.5
20	1.04	0.81	0.57	0.25	0.42	0.4
25	2.07	1.2	0.50	0.6	0.82	0.5
30	2.08	1.65	0.50	0.8	1.52	0.9

ATTACHMENT #2

THE MOTION OF A TRACK SHOE

BY

JOHN A. EILERS

6 MARCH 1964

ECONOMIC ENGINEERING STUDY BRANCH
ADVANCED SYSTEMS & CONCEPT RESEARCH DIVISION
RESEARCH AND ENGINEERING DIRECTORATE
U. S. ARMY TANK AUTOMOTIVE CENTER

SYMBOLS

$\mathbf{S}$ (vector) Instantaneous position of track shoe

$\mathbf{V}$ (vector) Instantaneous velocity of track shoe

$\mathbf{A}$ (vector) Instantaneous acceleration of track shoe

$\mathbf{A}_T$ (vector) Instantaneous acceleration of track shoe in the direction tangential to the curve of motion

$\mathbf{A}_R$ (vector) Instantaneous acceleration of track shoe in the direction perpendicular to the curve of motion

$\mathbf{i},\mathbf{j}$ Base coordinate vectors

n RPM of idler

ϕ Angle of rotation of idler

t Time elapsed (minutes)

$\boldsymbol{v}$ (vector) Forward velocity of vehicle

r Radius of idler

THE MOTION OF A TRACK SHOE

With the ground as reference, a track shoe follows a cycloidal curve when passing over the idler or sprocket (Fig. 1). The equation of a cycloid is:

$$\mathbf{S} = r(\phi - \sin\phi)\,\mathbf{i} + r(1 - \cos\phi)\,\mathbf{j}$$

Each track shoe will follow this curve. The velocity and acceleration of the shoe are:

$$\mathbf{V} = \frac{\delta \mathbf{S}}{\delta \phi} \cdot \frac{\delta \phi}{\delta t} = r\left[(1-\cos\phi)\,\mathbf{i} + (\sin\phi)\,\mathbf{j}\right]\frac{\delta\phi}{\delta t}$$

$$|\mathbf{V}| = r\sqrt{2(1-\cos\phi)}\,\frac{\delta\phi}{\delta t}$$

$$\mathbf{A} = \frac{\delta \mathbf{V}}{\delta \phi} \cdot \frac{\delta \phi}{\delta t} = r\left[(\sin\phi)\,\mathbf{i} + (\cos\phi)\,\mathbf{j}\right]\left(\frac{\delta\phi}{\delta t}\right)^2$$

$$|\mathbf{A}| = r\left(\frac{\delta\phi}{\delta t}\right)^2$$

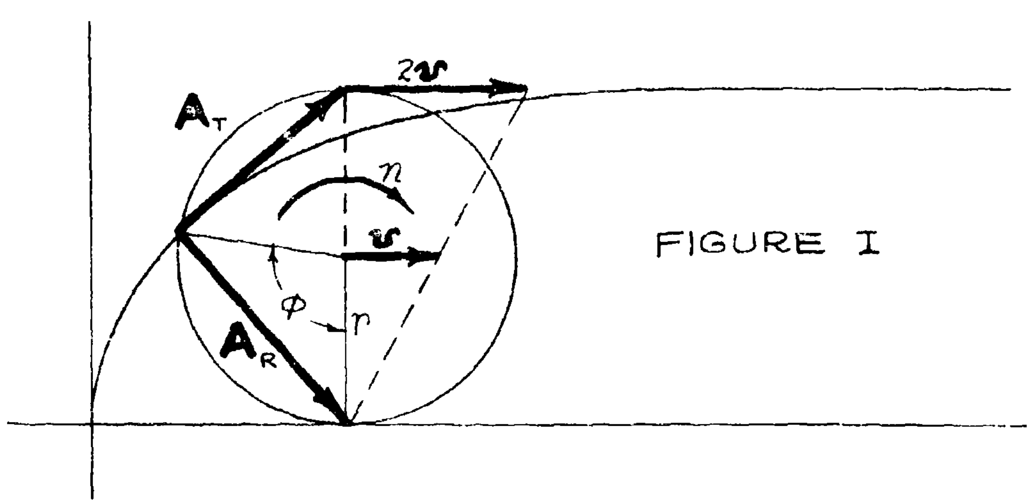


FIGURE I

From Figure 1, $\phi = 2\pi n t$ radians, where n is the RPM of the idler and t is the time in minutes. Therefore:

$$\frac{\delta\phi}{\delta t} = 2\pi n$$

The acceleration $\mathbf{A}$ has to be decomposed into tangential and radial components ($\mathbf{A}_T$ and $\mathbf{A}_R$):

$$|\mathbf{A}_T| = |\mathbf{A}|\cos\frac{\phi}{2} = 4\pi^2 n^2 r \cos\frac{\phi}{2}$$

$$|\mathbf{A}_R| = |\mathbf{A}|\sin\frac{\phi}{2} = 4\pi^2 n^2 r \sin\frac{\phi}{2}$$

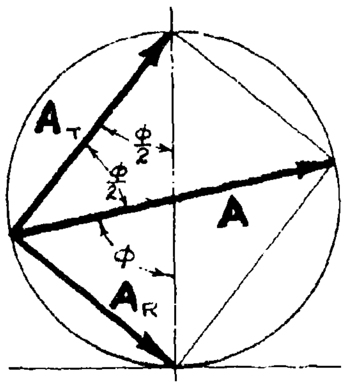


FIGURE II

Substituting for ϕ and $\frac{\delta\phi}{\delta t}$, the equations become:

$$|\mathbf{A}_T| = 4\pi^2 n^2 r \cos(\pi n t)$$

$$|\mathbf{A}_R| = 4\pi^2 n^2 r \sin(\pi n t)$$

$$|\mathbf{V}| = 2\pi n r \sqrt{2[1 - \cos(2\pi n t)]}$$

Dimensionless graphs of these functions are shown in Figure III.

Note that with a front-driving sprocket, all the teeth which contact the track (except the first one) are decelerating the track to zero velocity. Only the first tooth at the top of the sprocket is available to transfer energy to the track and accelerate it over the rear idler. In other words, all the energy which propels the vehicle forward is transfered to the tracks by one tooth on each sprocket. Although this analysis concerns a flat track (idler and sprocket on the ground), the same principles are used with any type of track and the same conclusions apply. (See Figure V)

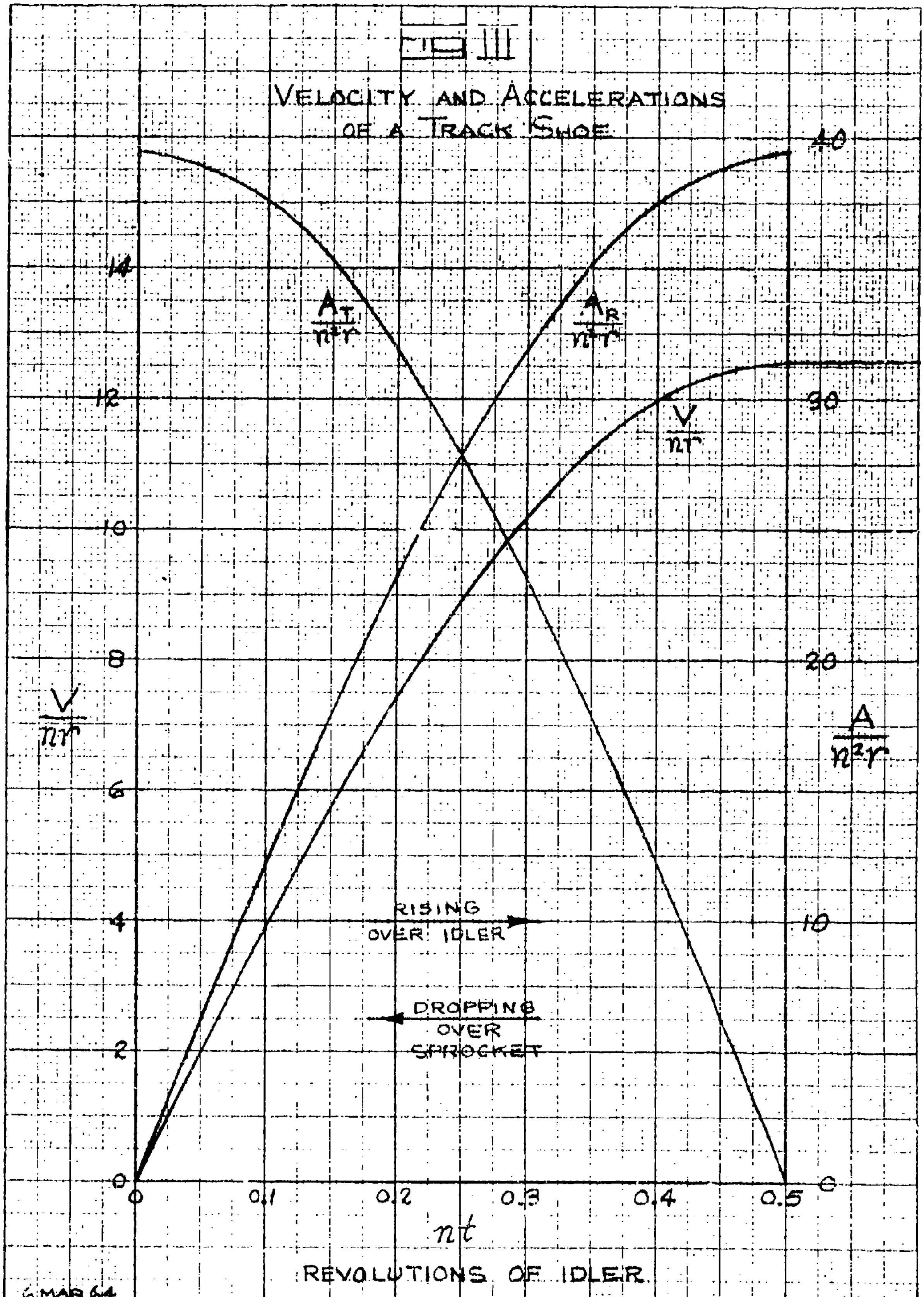


FIG III
VELOCITY AND ACCELERATIONS
OF A TRACK SHOE
$\frac{A_T}{n^2r}$
$\frac{A_R}{n^2r}$
$\frac{V}{nr}$
RISING
OVER IDLER
DROPPING
OVER
SPROCKET
$\frac{V}{nr}$
$\frac{A}{n^2r}$
0
2
4
6
8
10
12
14
10
20
30
40
0
0.1
0.2
0.3
0.4
0.5
nt
REVOLUTIONS OF IDLER
6 MAR 64

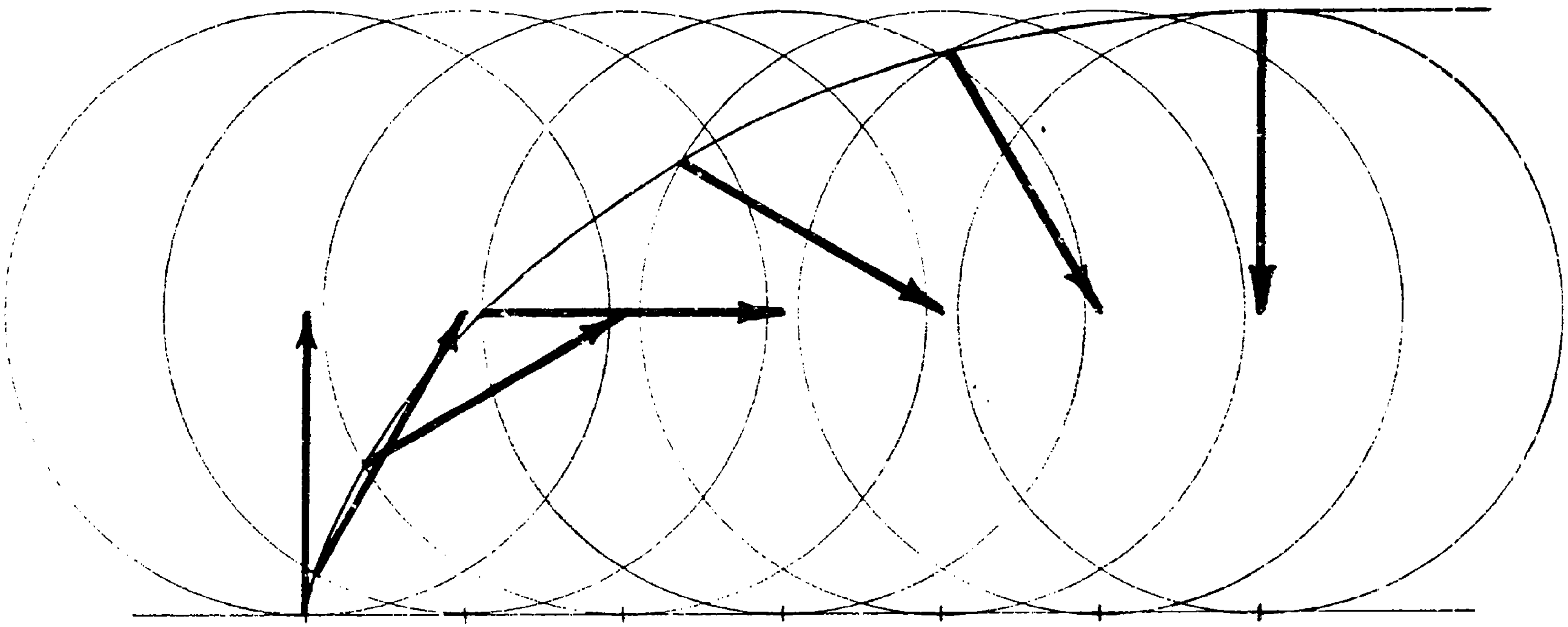


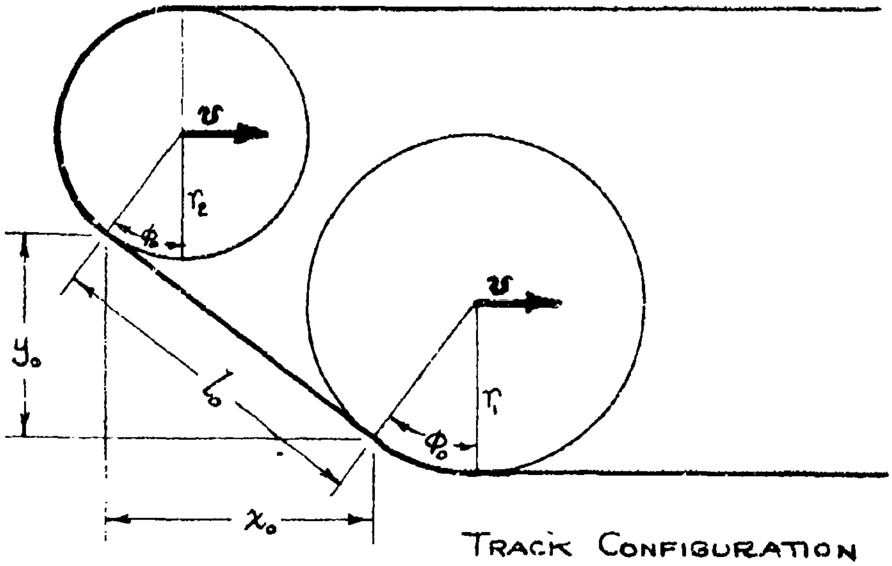
FIGURE IV

ACCELERATION (A) ACTING ON A TRACK SHOE

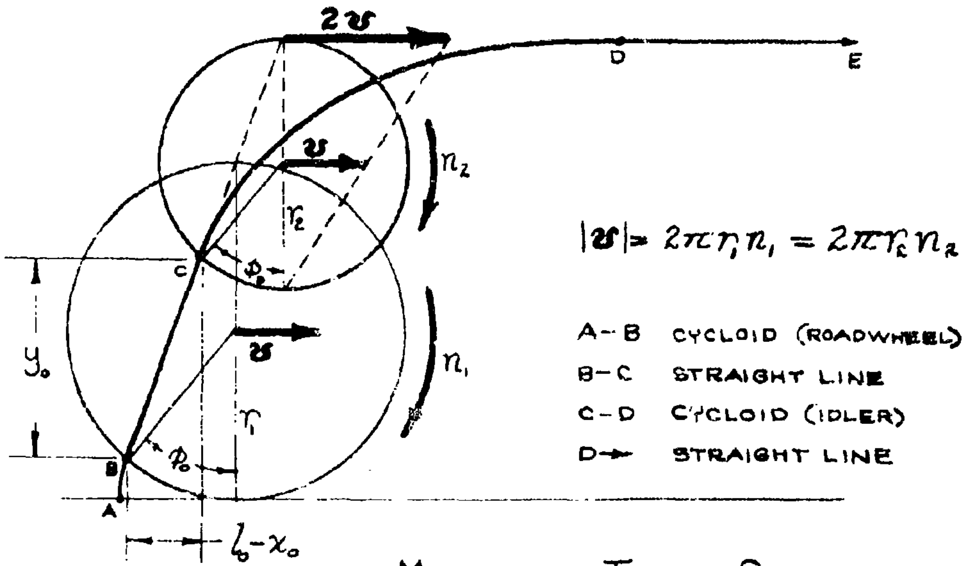
NOTE THAT IT IS CONSTANT IN MAGNITUDE BUT CHANGES DIRECTION CONTINUOUSLY

Figure V

Dynamics of Normal Track Configuration



Track Configuration



Motion of Track Shoe Relative to the Ground

When a track shoe is picked up and carried to the top of the idler, it undergoes a rotation of 180° about its center of gravity (See Fig. VI). This rotation is superimposed upon the cycloidal motion of the shoe as a mass.

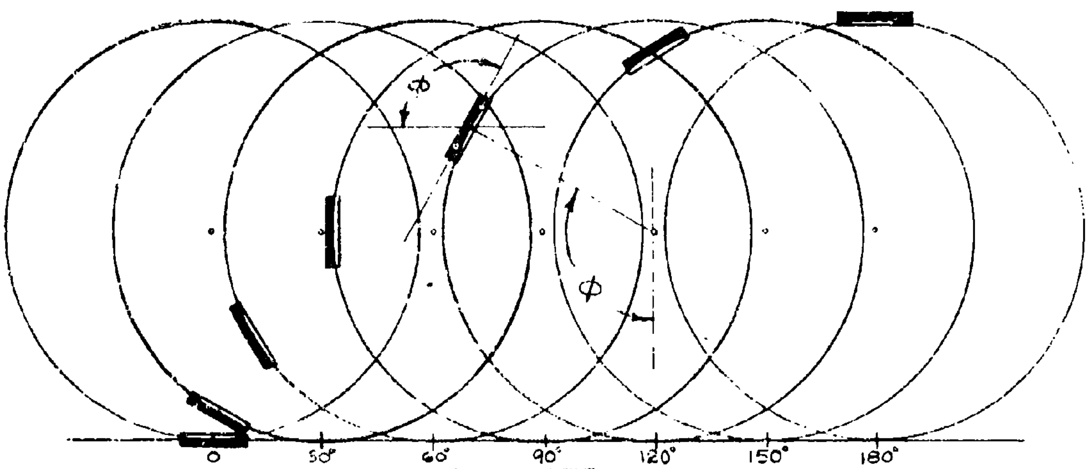


FIGURE VI

Since the shoe moves with the idler (neglecting small impulse-induced fluctuations), the angular velocity of this rotation is constant and is equal to:

$$\omega = \frac{d\phi}{dt} = 2\pi n \text{ radians per minute}$$

This rotation must be started instantaneously when the shoe is lifted off the ground, and must be dissipated rapidly at the top of the idler when the shoe starts down the free section of track. Thus, large angular accelerations will be induced as each shoe is picked up and released. These accelerations should cause vibration in the track, whose wavelength is comparable to the track pitch.

Attachment #3

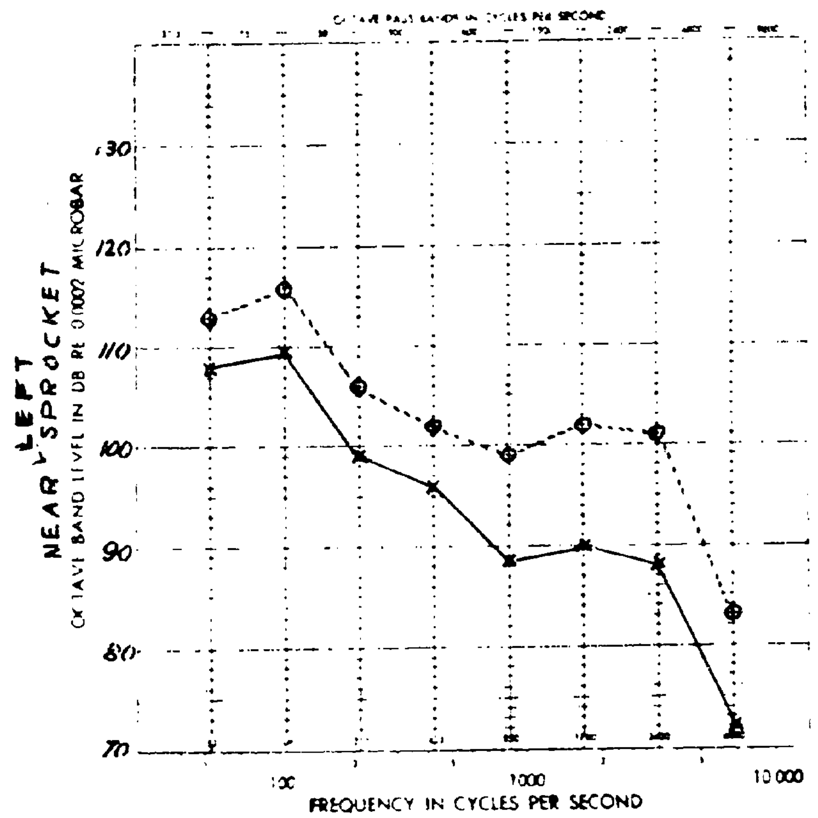
FMC Corporation

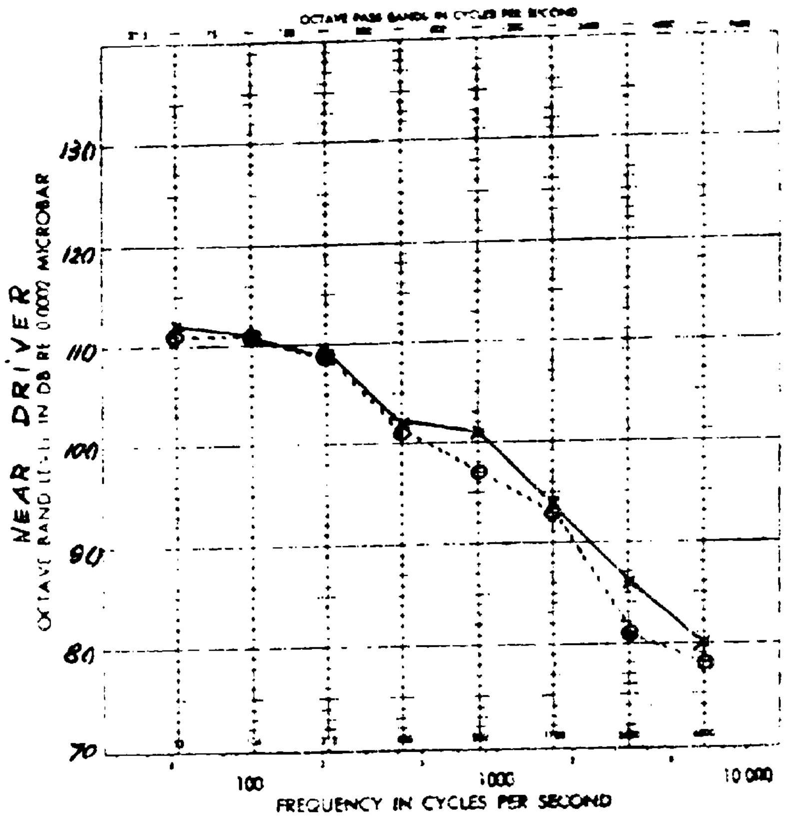
Ordnance Division, San Jose, California

Vibration Loads Versus Vehicle Speed of the M113 Armored Infantry Carrier Equipped with Standard or with Impact Absorption Sprockets

Test Results dated 10 December 1963

By R. FASANO



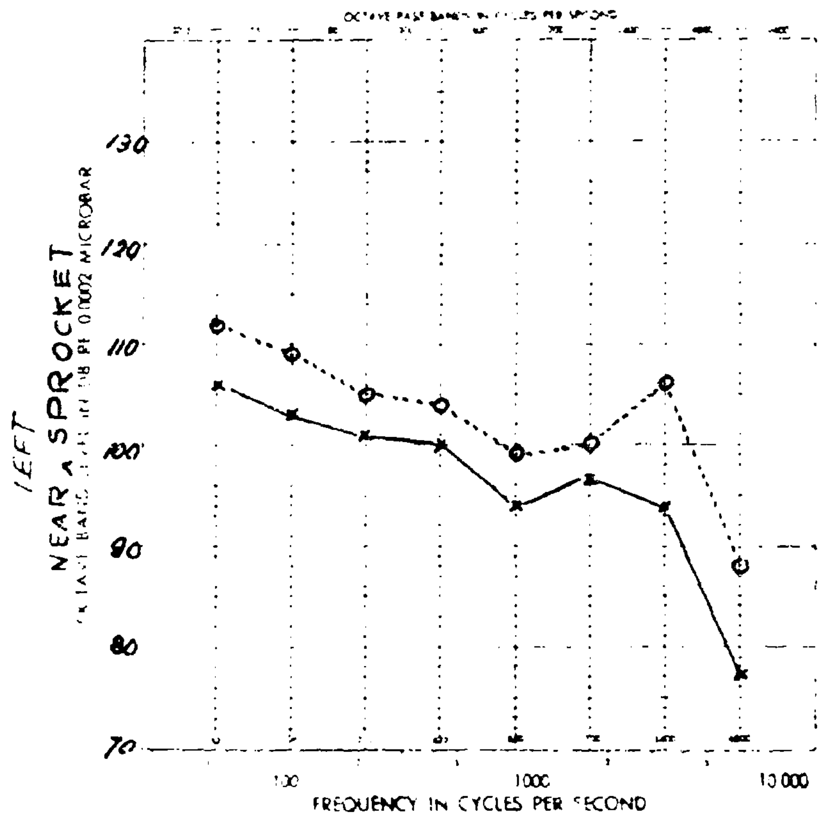


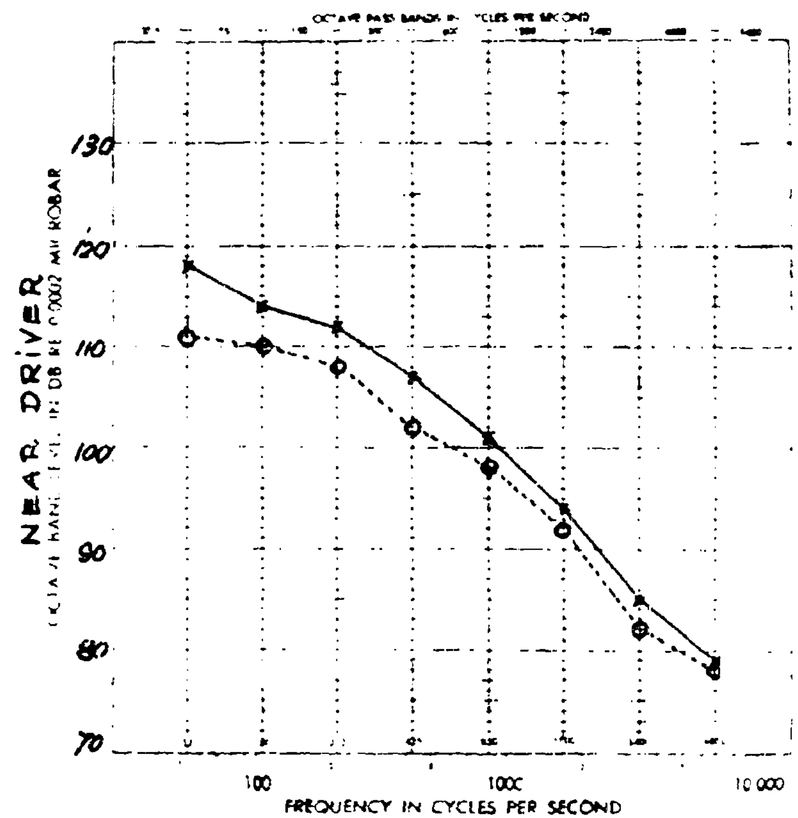
15 MPH

NOISE LEVELS NEAR SPROCKET & NEAR DRIVER'S EAR

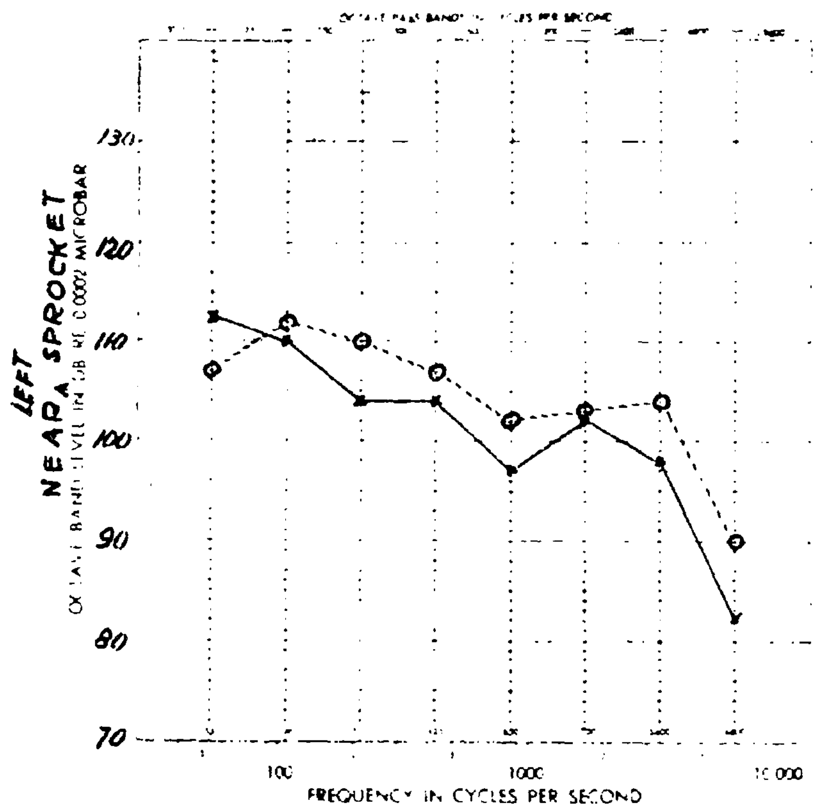
M113 VEHICLE WITH STANDARD DRIVE SPROCKETS (--O--) & WITH IMPACT ABSORPTION SPROCKETS (—x—)

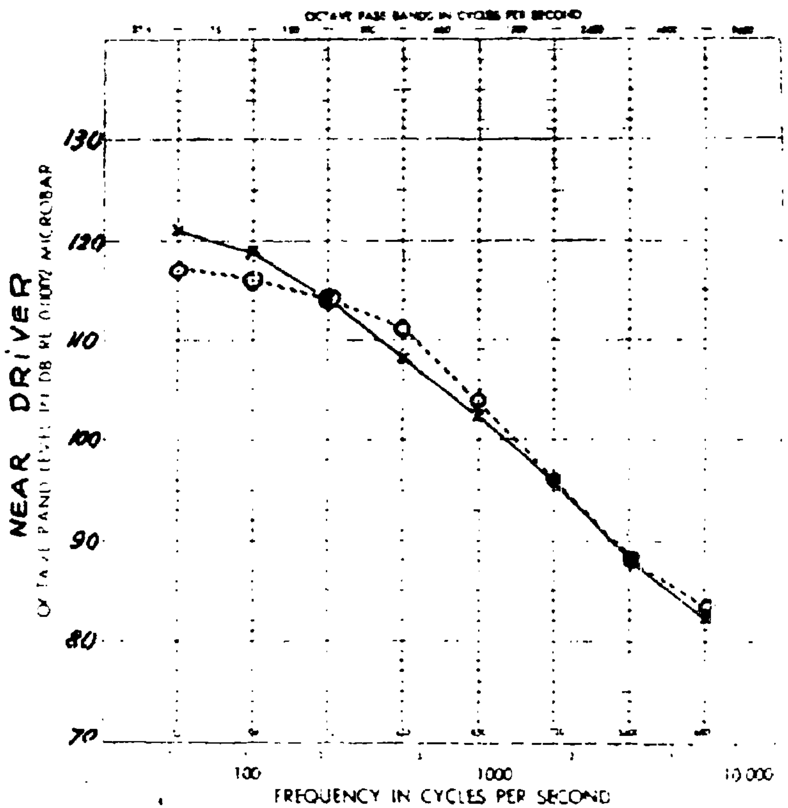
DEC. 10, 1963 R. FASANO


LEFT
NEAR A SPROCKET
OCTAVE PASS BANDS IN CYCLES PER SECOND
130
120
110
100
90
80
70
100
1000
10 000
FREQUENCY IN CYCLES PER SECOND

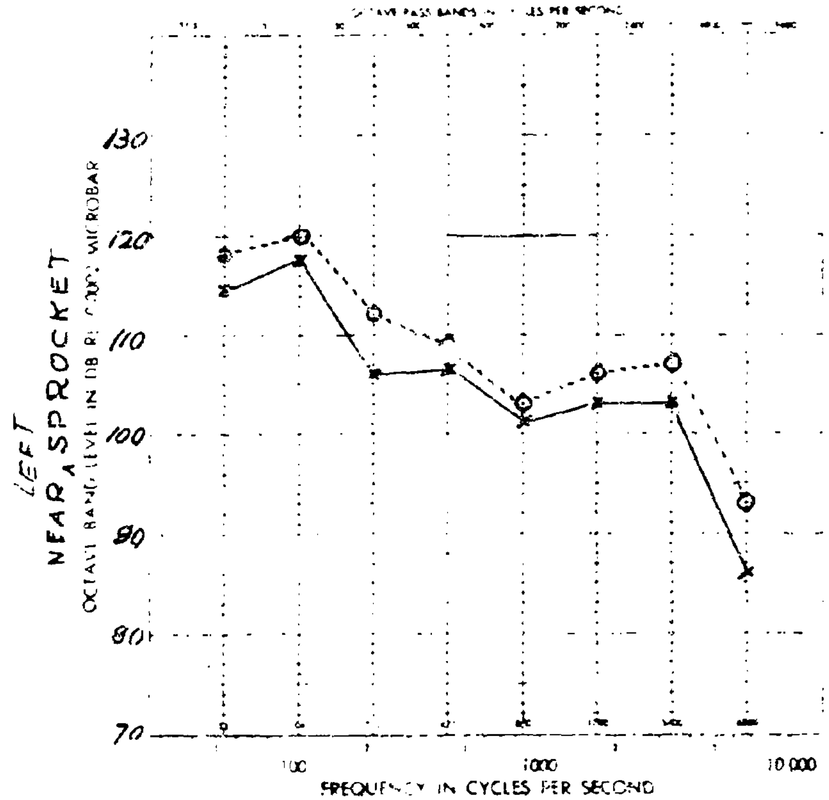

NEAR DRIVER
OCTAVE BAND LEVEL IN DB RE 0.0002 MICROBAR
OCTAVE PASS BANDS IN CYCLES PER SECOND
130
120
110
100
90
80
70
100
1000
10 000
FREQUENCY IN CYCLES PER SECOND

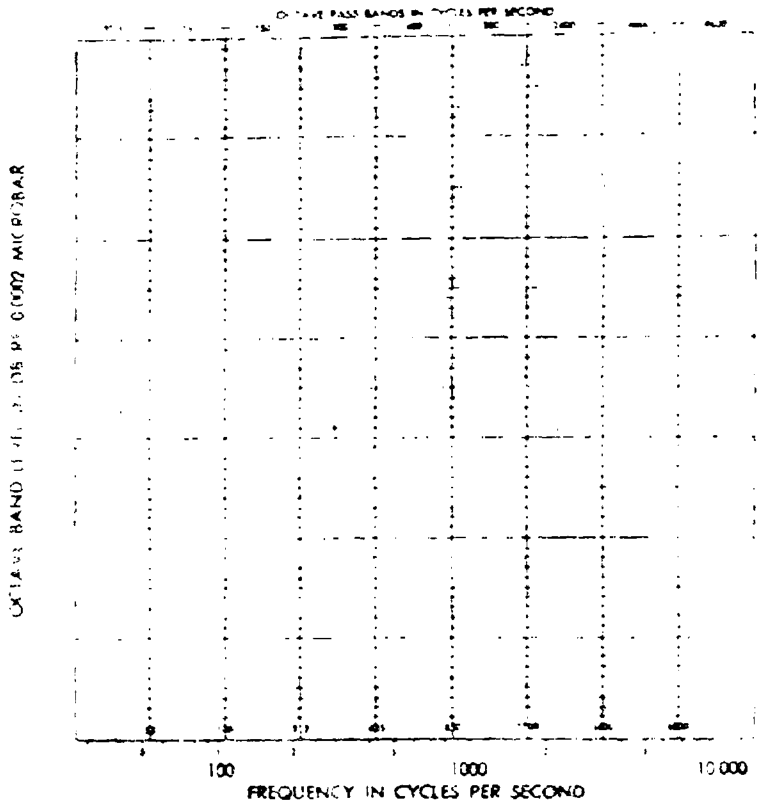
20 MPH


LEFT
NEAR SPROCKET
OCTAVE PASS BANDS IN CYCLES PER SECOND
OCTAVE BAND LEVEL IN DB RE 0.0002 MICROBAR
130
120
110
100
90
80
70
100
1000
10 000
FREQUENCY IN CYCLES PER SECOND

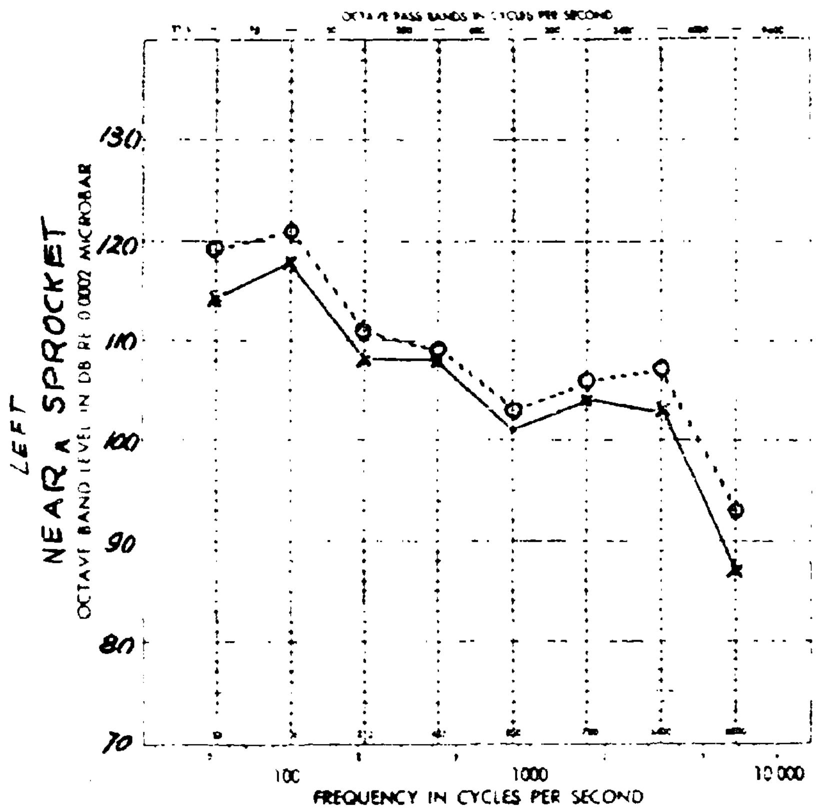

NEAR DRIVER
OCTAVE PASS BANDS IN CYCLES PER SECOND
OCTAVE BAND LEVEL IN DB RE 0.0002 MICROBAR
130
120
110
100
90
80
70
100
1000
10 000
FREQUENCY IN CYCLES PER SECOND

25 MPH

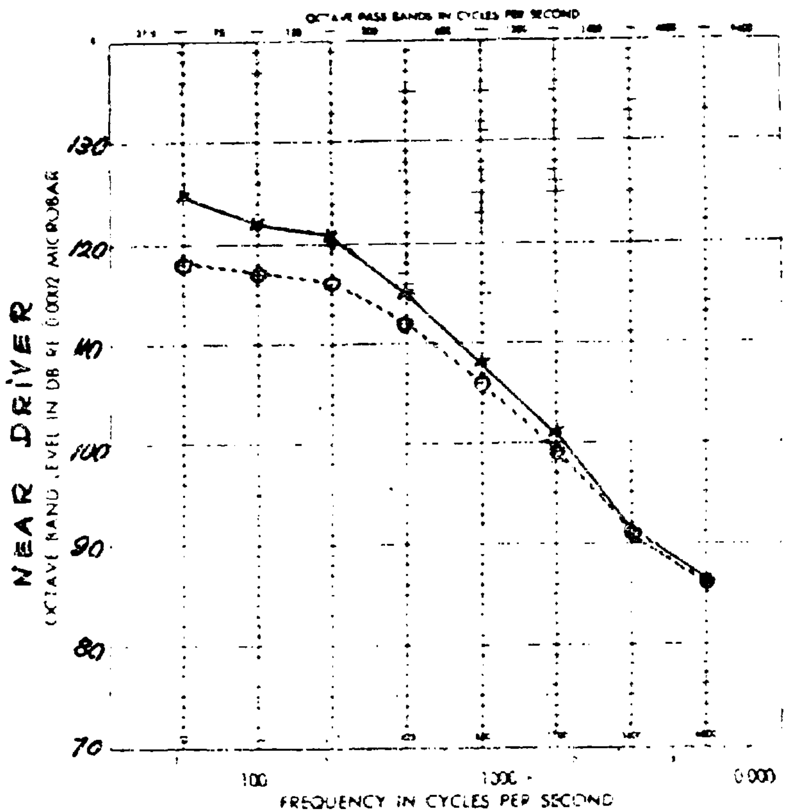

LEFT
NEAR SPROCKET
OCTAVE BAND LEVEL IN DB RE 0.0002 MICROBAR
130
120
110
100
90
80
70
100
1000
10 000
FREQUENCY IN CYCLES PER SECOND


OCTAVE PASS BANDS IN CYCLES PER SECOND
100
1000
10 000
FREQUENCY IN CYCLES PER SECOND

28 MPH

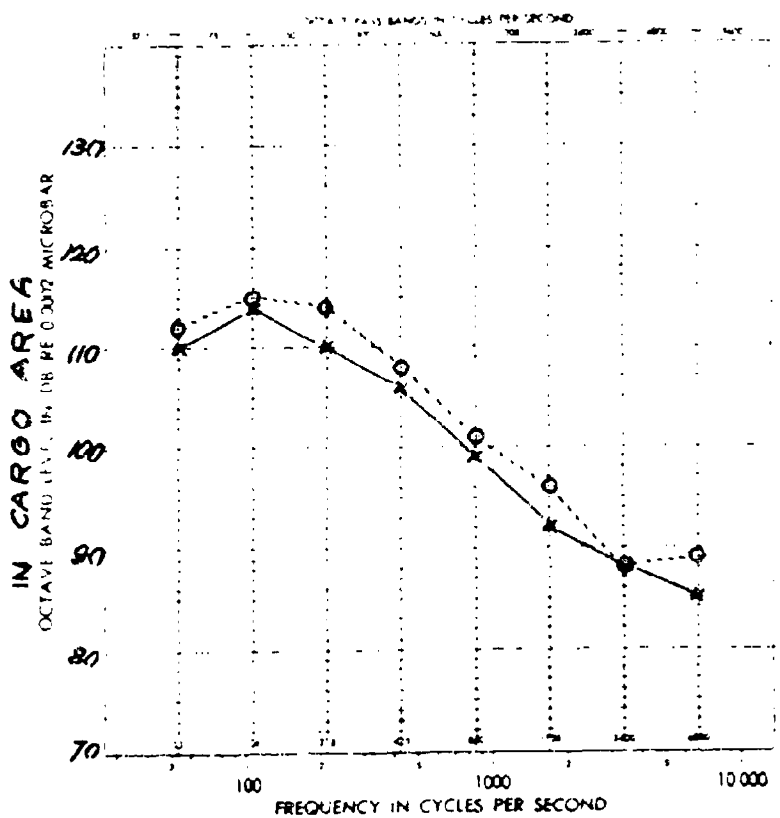


LEFT
NEAR SPROCKET
OCTAVE BAND LEVEL IN DB RE 0.0002 MICROBAR
OCTAVE PASS BANDS IN CYCLES PER SECOND
130
120
110
100
90
80
70
100
1000
10 000
FREQUENCY IN CYCLES PER SECOND

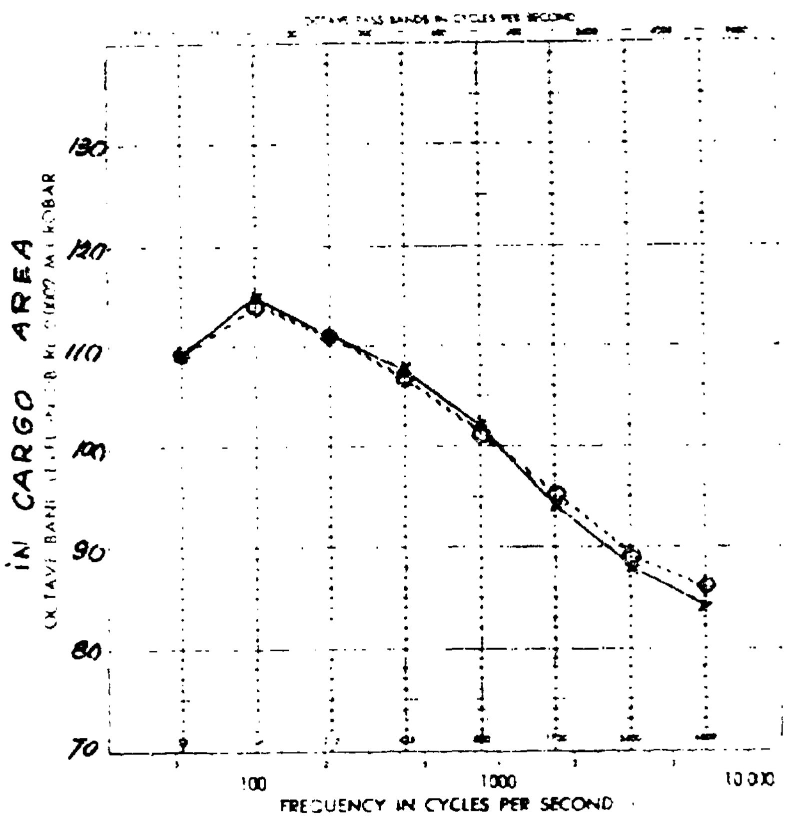


NEAR DRIVER
OCTAVE BAND LEVEL IN DB RE 0.0002 MICROBAR
OCTAVE PASS BANDS IN CYCLES PER SECOND
130
120
110
100
90
80
70
100
1000
10 000
FREQUENCY IN CYCLES PER SECOND

30 MPH



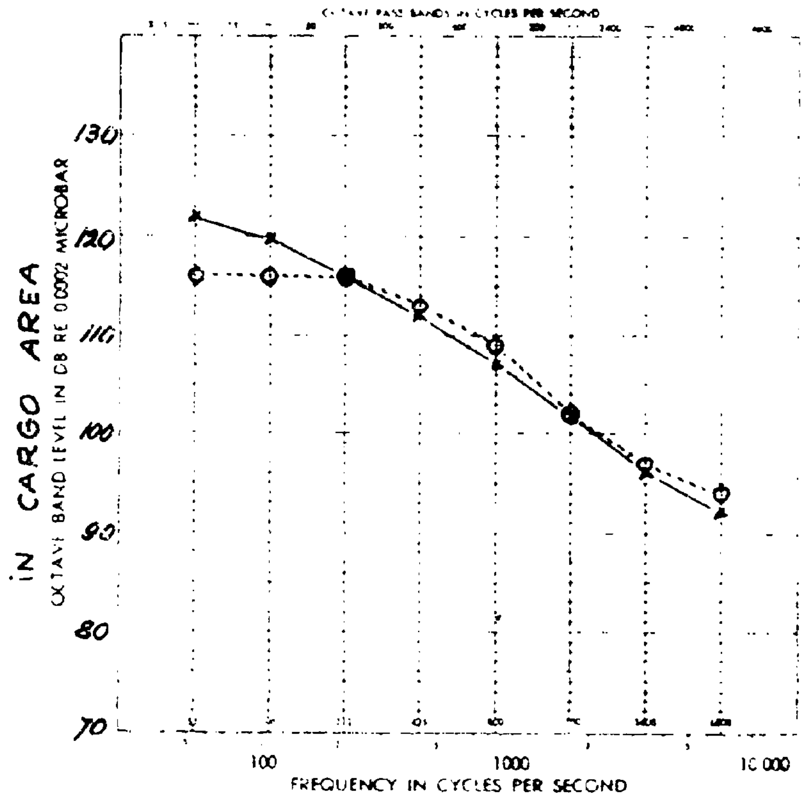
15 MPH



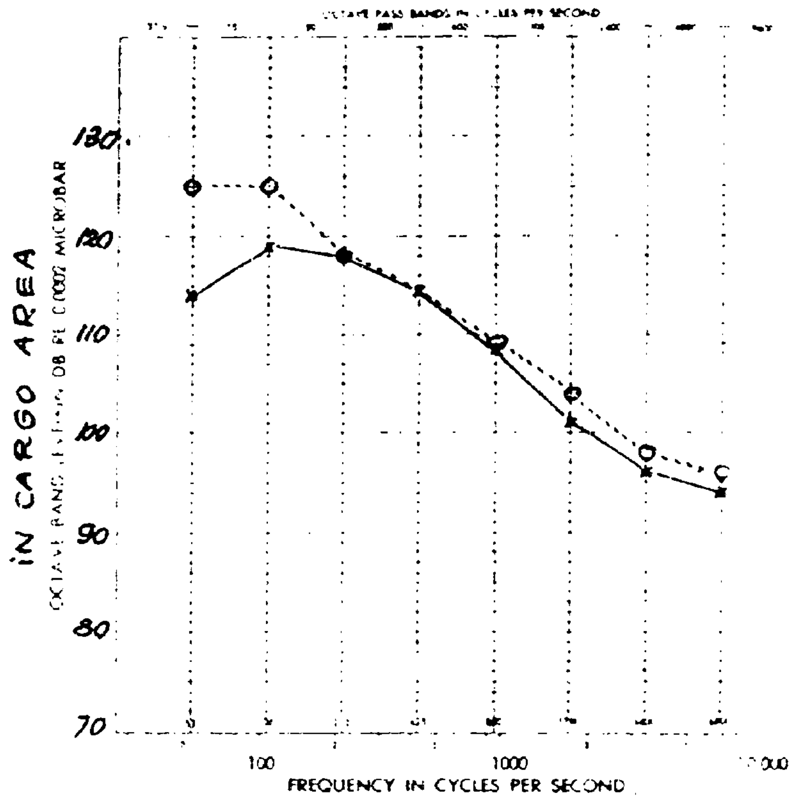
20 MPH

NOISE LEVELS IN CARGO AREA
OF M113 WITH STANDARD DRIVE
SPROCKETS (··O··) AND IMPACT
ABSORBTION SPROCKETS (—X—)

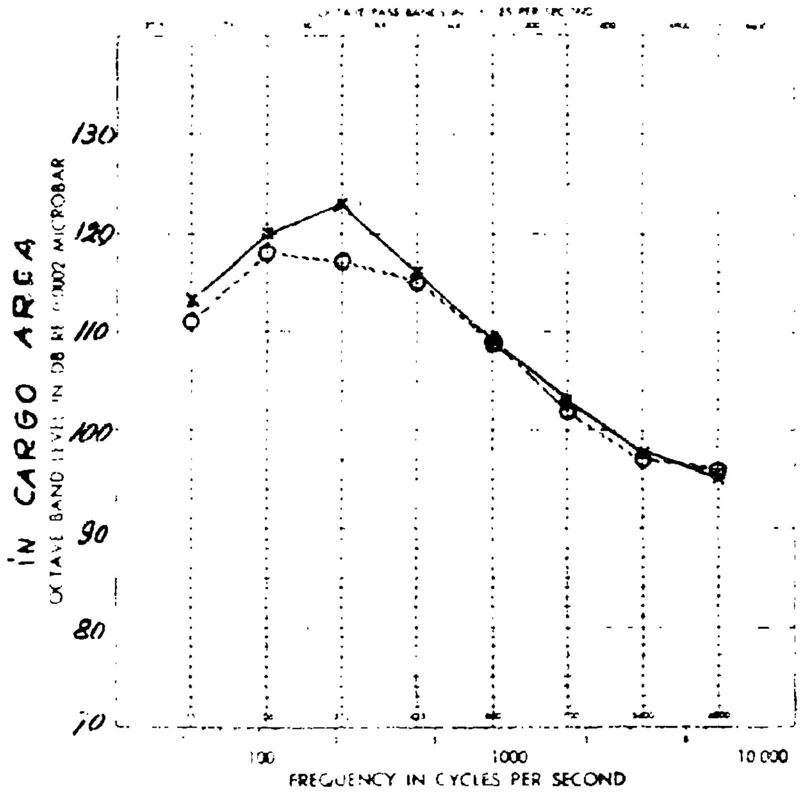
DEC 10, 1963 R. FASANO

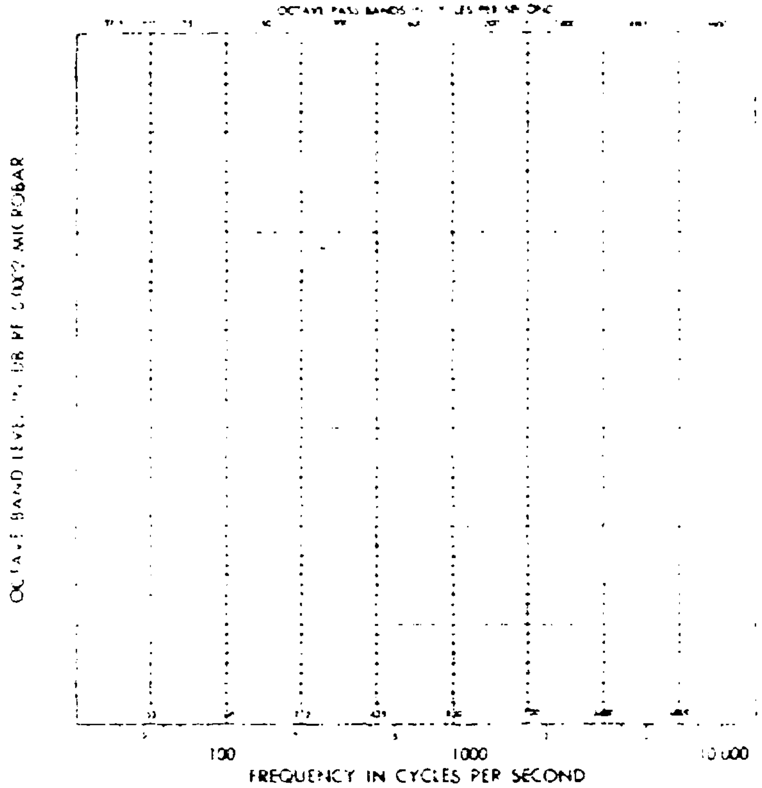

IN CARGO AREA
OCTAVE BAND LEVEL IN DB RE 0.0002 MICROBAR
OCTAVE PASS BANDS IN CYCLES PER SECOND
130
120
110
100
90
80
70
100
1000
10 000
FREQUENCY IN CYCLES PER SECOND

25 MPH


IN CARGO AREA
130
120
110
100
90
80
70
100
1000
FREQUENCY IN CYCLES PER SECOND

28 MPH


IN CARGO AREA
OCTAVE BAND LEVEL IN DB RE 0.0002 MICROBAR
130
120
110
100
90
80
70
100
1000
10 000
FREQUENCY IN CYCLES PER SECOND


OCTAVE PASS BANDS
100
1000
FREQUENCY IN CYCLES PER SECOND

30 MPH

ATTACHMENT #4

SOUND AND VIBRATION UNITS

SUMMARY

Units of the different sound and vibration phenomena used in the Memorandum Report on Noise and Vibration in the M113, Armored Infantry Carrier equipped with impact absorption sprockets, are discussed and defined as based on the explanations, definitions and properties compiled from the references:

1. L. J. Fogel: Biotechnology, Concepts and Applications, Prentice-Hall, 1963.

2. L. L. Beranek and al.: Principles of Sound Control in Airplanes Cruft Laboratory, Harvard University, 1944.

3. J. M. Bowsher and D. W. Robinson: Calculation of Zwicker Phons on a Digital Computer

Nature, Vol. 200, Nr. 4906, 9 November 1963, pp. 553 to 555.

Compiled by: Dr. F. Jaklitsch

CONTENT

TERMS	UNIT
1. Intensity of Sound	decibel
Signal Intensity	
Intensity Level	
Sensation Level	
2. Standard Reference Level	$dyne/cm^2$
3. Loudness Level	phon
4. Loudness	sone
5. Pitch of Sound	mel
6. Precision of Perception	jnd
7. Vibration Intensity	pal, trem

INTENSITY OF SOUND (decibel db)
(Sensation level,
signal intensity,
intensity level)

Sound intensity is expressed in decibels. The decibel is a logarithmic unit based on the ratio between two values of power or two values of energy:

The difference in decibels between two values of energy E_1 and E_2 is

$$i = 10 \log_{10} \frac{E_1}{E_2}$$

Thus, the difference in decibels between two sound intensities i_1 and i_2 is

$$i = 10 \log_{10} \frac{i_1}{i_2}$$

Inasmuch as the intensity is proportional to the square of the sound pressure p or to the square of the particle velocity v:

$$i = 10 \log_{10} \frac{i_1}{i_2} = 10 \log_{10} \frac{p_1^2}{p_2^2} = 10 \log_{10} \frac{v_1^2}{v_2^2}$$

or

$$i = 20 \log_{10} \frac{p_1}{p_2} = 20 \log_{10} \frac{v_1}{v_2}$$

The pressure p and particle velocity v are measured at the same points or at points where the accoustical conditions are similar.

The sound intensity in a given direction at a point is defined as the rate of flow of sound energy through a unit area perpendicular to the given direction at the point.

Sound intensity is measured in watt/cm^2 or in dyne/cm^2. The human ear responds to a wide range of intensitie . The maximum tolerable intensity is 10^{12} time as great as the least audible intensity. The decibel scale compresses this very wide range of 10^{12} in intensity to a scale length of 120 db.

The intensity of a sound as defined is a physical property and, thus, subject to objective measurements.

STANDARD REFERENCE LEVEL

Since the decibel scale expresses only how much greater one sound intensity is than another, it is advisable to adopt a certain standard intensity as a reference level so that all sound intensities can be expressed in terms of decibels above or below the standard.

The standard reference level as established by the American Standard Association (ASA) is:

$$0.0002 \text{ dyne/cm}^2$$

and it corresponds roughly to the minimum audible intensity at 1000 cycles per second or HERTZ. This reference level corresponds approximately to an energy transmission of 10^{-16} watt/cm^2 for the case of a freely traveling plane sound wave in air.

For a temperature of 80°F (26.7°C) and a barometer pressure of 74 cm Hg (1kg/cm^2, 10m of water, or 14.22 psi) this relation is exactly true.

PRECISION OF PERCEPTION

The auditory perception is most sensitive to changes in the signal intensity over the frequency range between 500 and 10,000 cps.

LOUDNESS LEVEL L (Phons)

The property of a sound most important for noise reduction work is its loudness level, or the strength of the sensation in the human ear.

The strength of sensation is found to depend in a rather complex manner upon the frequency and intensity level of the sound. A plot of curves for sounds appearing equally loud provides the system of curves shown in Figure 1. All points on a given curve represent sounds which appear to be of equal loudness level to the average human ear.

Thus, a sound at the frequency f and the intensity i has the loudness level L as follows:

for: f = 80 cps and i = 60 db: L = 30 phons

f = 9000 cps and i = 40 db: L = 30 phons

f = 1000 cps and i = 30 db: L = 30 phons

hence, they are of equal loudness level L.

The loudness level L of a sound is defined as the intensity level i in db of an equally loud tone of 1000 cps. The numbers on the curves of Fig. 1 are the loudness levels in phons as defined by the ASA.

L = 0 phons represents the threshold of hearing, provided the threshold is measured in terms of the minimum audible pressure at the ear drum.

The highest curve (120 phons) represents the threshold of discomfort at which the sound sensation is accompanied by a sensation of annoyance or vibration in the ear. The threshold of pain is experienced at approximately 140 db.

The phenomenon of hearing is based on three different ways of conduction:

1. the aero-tympanic
2. the cranio-tympanic
3. the osseous conduction

Probably there exists a distinction of sensitivity for frequencies.

The sonic range reaches from about 16 to 20000 cps for the "average" human ear. In this range the sound is mostly related to a certain pitch.

The subsonic region comprises the range below 16 cps, the ultrasonic extends beyond 20000 cps. Sounds in these regions may still be perceived but without a definite pitch (see "Pitch of Sound") (Fig. 2).

The phon is not a convenient unit for general use, since we need a team of "NORMAL" observers to make any measurements. So over the years, several people have tried to find ways of calculating phons from the sound's physical characteristics. The methods are all based on manipulating the data obtained from a spectrum analysis and allowing for the different response of the ear to sounds of different pitches.

The most sophisticated method is that derived by D. E. Zwicker[1] in 1959, which takes into account the phenomenon known as masking.

[1] K. E. ZWICKER: FREQUENZ (13), 1959 p. 234 and foll.

LOUDNESS I (sones)

The loudness-level contours (i.e. curves in Fig. 1) and the sound level meter readings indicate in terms of level and frequency what tones will sound equally loud. However, they do not furnish us with a scale for measuring loudness, which is that subjective quality of a sound which determines the magnitude of the auditory sensation produced by the sound.

Loudness is a subjective, or psychological attribute of a sound, whereas loudness level is partly physical partly psychological, and intensity level is purely a physical quantity. Sound level which is the reading taken on a sound level meter, is approximately equivalent to loudness level if a proper weighing network is used.

The loudness level contours makes possible to determine what sounds will appear equally loud to the ear, but they do not indicate what sound will appear twice as loud or half as loud or 100 times as loud as a given sound. For this type of information we refer to curves of loudness versus intensity level which are shown in Fig. 3.

Loudness is expressed in "sones" (units of perceived loudness), which are indicated on the coordinate scale. These numbers are proportional to the loudness of a sound as heard by the "Normal" human ear.

For example, we see from the curve that a 1000 cps tone of 40 db intensity has a loudness of 1 sone, and a tone of 98 db intensity has a loudness of 80 sones. From this we know that the second tone sounds 80 times as loud as the first.

In the frequency range from 1000 to 8000 cps, a change of 12 db is required to change the loudness by a factor of 2. For instance, in order to change the loudness from 150 sones to 75 sones the intensity level must be changed from 110 to 98 db. For a reduction in loudness of 10% the intensity level must be reduced by about 1.2 db.

Loudness is the subjective judgement of the intensity of the perceived sound stimulus. This judgement can be achieved over an extreme wide range of physical sound intensities. The normal ear responds above zero db as shown in Fig. 2 which corresponds to 0.0002 dynes/cm^2. The auditory channel can respond to sound pressure levels which produces a force as small as three millionth of a gram.

At the other extreme it is sensed:

Discomfort at about 120 db (threshold of discomfort)
Feeling at about 130 db (threshold of annoyance or disturbance)
Pain is induced at about 140 db (threshold of pain)

The usable sound intensities extend over a range of several million times the smallest pressure.

It is often necessary to estimate the relative intensity of sounds. In such a case, it is appropriate to use the sone scale wherein one (1) sone is taken to be the perceived loudness of a 1000 cps pure tone 40 db above auditory threshold. If a given sound is judged to be twice as loud as the reference, it is called a two sone sound.

To ease the conversion from physical sound measurements (sound intensity in db) to physiological sensation (perceived loudness in sones) a conversion table was provided (Table 1). Comparison in loudness gives a more realistic judgement of the noise behaviour of a vehicle than compared decibels.

The reason that sound intensity which is a logarithmic energy reading does not give an immediately usable gauge for the judgement of auditory problems can be explained by Fig. 4 whose curves were deducted from Fig. 3.

TABLE

Relationship Between Sound Intensity I (db) Frequency f (cps) and Loudness l (sone)

Intensity I in db	Loudness l in Sones at the Frequency f in cps						
	1000	400	200	100	50	25	10
70	12.8	12.5	9	5	2		
71	14	14	10.2	5.5	2.5		
72	15	15	11.4	6.2	3		
73	16	16	12.6	7	3.5		
74	17.2	17.2	14	8.2	4		
75	18.4	18.4	15.2	9.5	4.6		
76	19.7	19.7	16.7	10.9	5.2		
77	21.2	21.2	18.4	12.4	6.4		
78	23.4	23.4	20	14	7.6	0.5	
79	25	25	21.8	15.7	8.9	1.0	
80	27	27	23.8	17.5	10.3	1.5	
81	29	29	25.6	19.7	11.9	2.2	
82	31	31	27.6	21.5	14.5	3.2	
83	33.2	33.2	30	23.5	17	4.8	
84	35.5	35.5	32.5	26	19.5	6.4	
85	37.8	37.8	35	28.5	22	8.5	
86	40	40	37.5	31.3	25	11.2	
87	42.5	42.5	40	34	28	14	
88	45	45	43	37.5	32	18	
89	48	48	46	41	36	22	
90	51	51	49	44	40	27	
91	54	54	52	48	44	32	
92	57	57	56.5	51.5	48	38	
93	60	60	60	55	52	44	
94	63	63	63	59	57	50	
95	66.5	66.5	66.5	63	61	55	
96	70	70	70	68	65	61	
97	74	74	74	73	70	67	
98	78	78	78	78	75	73	
99	83	83	83	83	81	79	

Table Cont'd

Intensity db	Frequency f cps						
	1000	400	200	100	50	25	10
100	98	98	98	98	87	86	
101	94	94	94	94	94	93	
102	100	100	100	100	100	100	
103	106	106	106	106	106	106	0.4
104	112	112	112	112	112	112	1
105	119	119	119	119	119	119	3
106	127	127	127	127	127	127	6
107	136	136	136	136	136	136	10
108	145	145	145	145	145	145	14
109	154	154	154	154	154	154	19
110	163	163	163	163	163	163	24
111	172	172	172	172	172	172	29
112	181	181	181	181	181	181	34
113	190	190	190	190	190	190	39
114	200	200	200	200	200	200	45
115	210	210	210	210	210	210	51
116	220	220	220	220	220	220	57
117	230	230	230	230	230	230	64
118	241	241	241	241	241	241	69
119	252	252	252	252	252	252	75
120	263	263	263	263	263	263	81
121	276	276	276	276	276	276	87
122	290	290	290	290	290	290	93
123	304	304	304	304	304	304	100
124	317	317	317	317	317	317	107
125	332	332	332	332	332	332	113
126	347	347	347	347	347	347	119
127	363	363	363	363	363	363	125
128	379	379	379	379	379	379	131

Table Cont'd

Intensity db	Frequency f cps						
	1000	400	200	100	50	25	10
129	397	397	397	397	397	397	136
130	412	412	412	412	412	412	142
135							171
140							198
145							230
150							263

PITCH OF SOUND (mel)

Pitch is a single-valued subjective summary of the sensed spectral properties of the sound-stimulus. The listener describes a sound as being "high" or "low".

Generally, the pitch of a note roughly corresponds to the frequency of the predominant sinusoidal components which might be used to synthesize the pressure amplitude waveform. Pitch can run the gamut of sensed sound frequencies from about 20 to an upper limit of about 16000 to 20000 cycles per second, dependent on the listener. As the spectral property of a sound-signal is made more diffuse over a band of frequencies, it becomes more difficult for the listener to distinguish pitch. He hears the note clearly but is unable to offer a singular representation for the spectrum.

Some sounds, such as WHITE SOUND (equal amplitude components over a wide frequency range) and a very short sinusoid (where there is insufficient time to establish a spectral identity) can have no pitch at all.

Sounds as low as 5 cps (subsonic)(Fig. 2) or as high as 100000 cps (ultrasonic) have been heard due to the nonlinearity of the auditory channel which serves to convert portions of the impinging sound energy into frequencies within the normal auditory range. The reported cases are without specific sensations of pitch.

Pitch appears as a two-dimensional attribute:

1. Pitch proper

2. Intensity of pitch sensation measured by the ease of the listener to distinguish it.

Timbre, brightness, fullness, etc. are other spectral-dependent perception attributes (but of secondary importance).

The perceived pitch of a sound is related to other aspects of the physical stimulus. Pitch is lowered by an increase in intensity of pure tones in the range below 500 cps: That is low notes appear lower when they are stronger.

In a similar manner, high tones, above 4000 cps appear higher upon increase of the stimulus intensity.

The octave (the range between tones which are related by a frequency factor of two (2)) is the common unit of measure in music. Pitch, however, is nonlinearly related to frequency in such a way that octaves towards the ends of the musical scale appear to be of shorter range than octaves in the middle range.

To overcome this difficulty, a scale has been developed using the unit of mel. Mel is defined as the unit of equal pitch as judged by an average listener.

Arbitrary reference has been established as a 1000 cps sinusoidal tone stimulus heard at 40 db above the threshold of audition. This is said to correspond to the 1000 mel point. Pitch of a given signal might be judged to be one-half that of the reference in which case it would be described as being 500 mels.

PRECISION OF PERCEPTION (jnd)

The precision of perception can be stated in terms of the size of the "just-noticable difference" (jnd) that minimum amount of change in the stimulus which will be detected with a probability of (50%).

The frequency jnd is smaller for tones toward the low end of the frequency spectrum, the actual amount being dependent upon the sensation level. Below 20 db sensation level, the average human looses his ability to perceive change in frequency. Above that intensity level a frequency difference of three (3) cps is significant for tones below 1000 cps. Above that frequency the jnd is approximately three-tenth of one percent (0.3%) of the tones frequency.

In the loudness range below 20 db, the detectable difference is from 2 to 6 db dependent on frequency of the sound, while in the range above 20 db 1/2 to 1 db is sufficient. At extremely high frequencies a larger increment is required to assure detection of the change.

VIBRATION INTENSITY (pal)

The Weber-Fechner Law states that a sensation is proportional to the logarithm of the stimulus intensity. This statement serves as a basis for the establishment of a gauge of vibration intensity as follows:

The standard vibration intensity unit "pal" is defined by

$$\text{pal} = 10 \log \frac{V}{V_o} \qquad (1)$$

This is similar to the definition of the sound intensity "decibel"

$$\text{decibel} = 10 \log \frac{E_1}{E_2} = 10 \log \frac{I_1}{I_2} \qquad (2)$$

which is the logarithm of energy or intensity ratios.

It means in equation (1):

The rms velocity V of simple harmonic motion (SHM):

$$V = \frac{2\pi f A}{2}$$

with f - frequency in cycles per second (cps)

A - amplitude of vibration in inches

With: $V_o = 0.0125$ in/sec

as a threshold velocity, the following steps of subjective levels of sensation are expressed by pal;

TABLE 2

Sensation	Vibration Intensity pal
just detectable	to 5
quite perceptible	5 to 10
troublesome	10 to 20
unpleasant	20 to 40

Ten Cate[x] defines another vibration unit "trem":

The "trem" is a frequency sensitive scale. This kind of scale compares easily with Janeway's curves throughout his frequency range.

---

x) Reference: W. TEN CATE: "Vibration Nuisance", Royal Aircraft Establishment, Great Britain (AD 159-701) October 1957, Library Translation by R. C. Murray, #693 from Institute TNO VOOR WERKTUIG KINDIGE CONSTRUCTIES, Report #147, May 1953

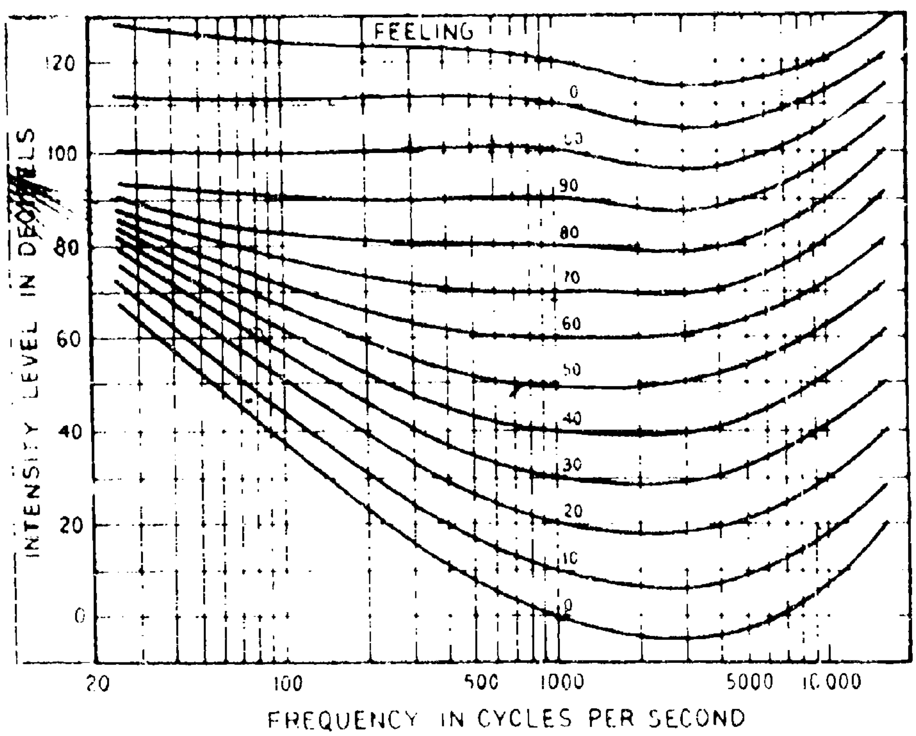
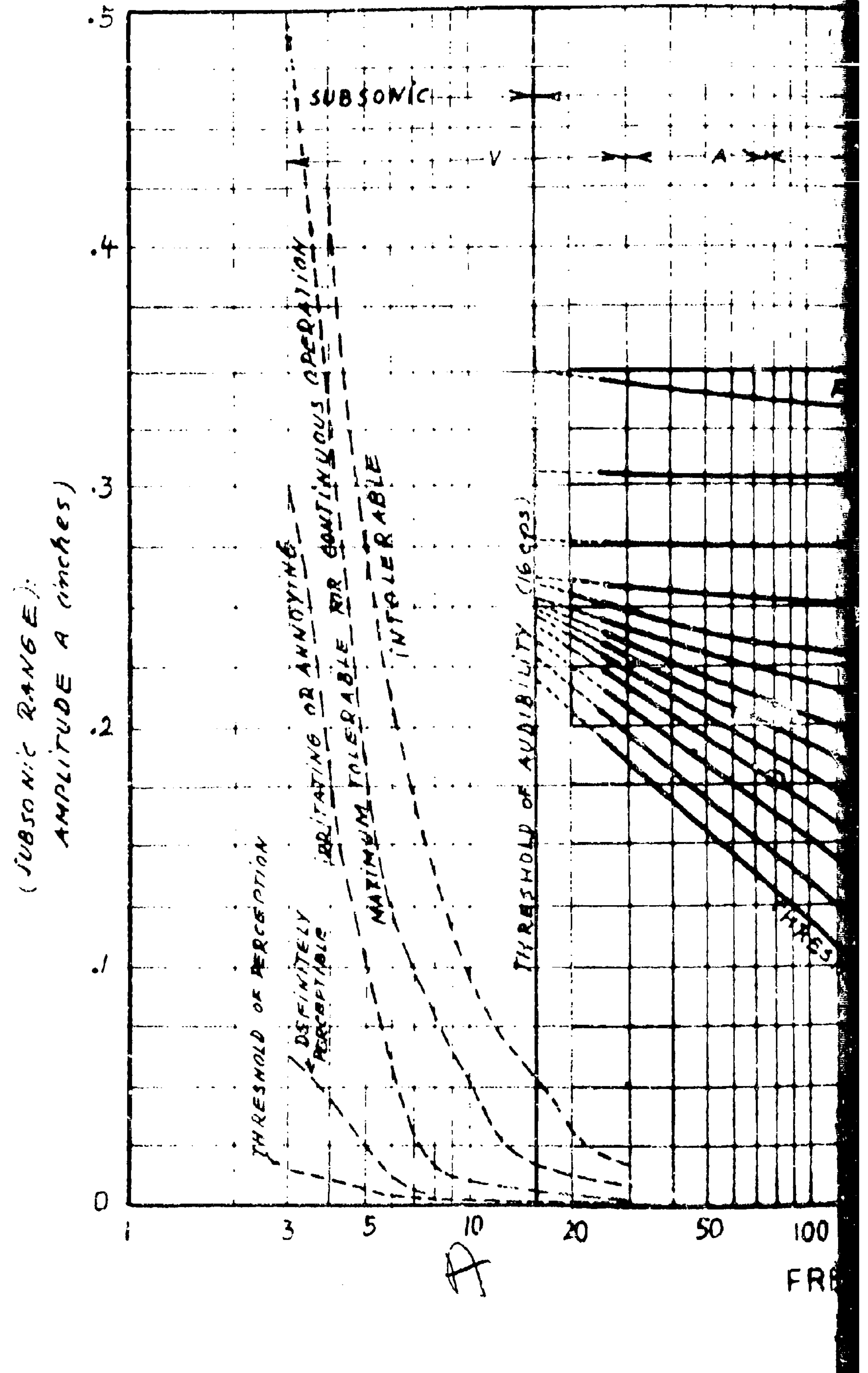

FEELING
120
100
80
60
40
20
0
INTENSITY LEVEL IN DECIBELS
90
80
70
60
50
40
30
20
10
0
20
100
500
1000
5000
10,000
FREQUENCY IN CYCLES PER SECOND

Fig. 1



SUBSONIC
V
A
(SUBSONIC RANGE):
AMPLITUDE A (inches)
.5
.4
.3
.2
.1
0
THRESHOLD OF PERCEPTION
DEFINITELY PERCEPTIBLE
IRRITATING OR ANNOYING
MAXIMUM TOLERABLE FOR CONTINUOUS OPERATION
INTOLERABLE
THRESHOLD OF AUDIBILITY (16 CPS)
1
3
5
10
20
50
100
FR

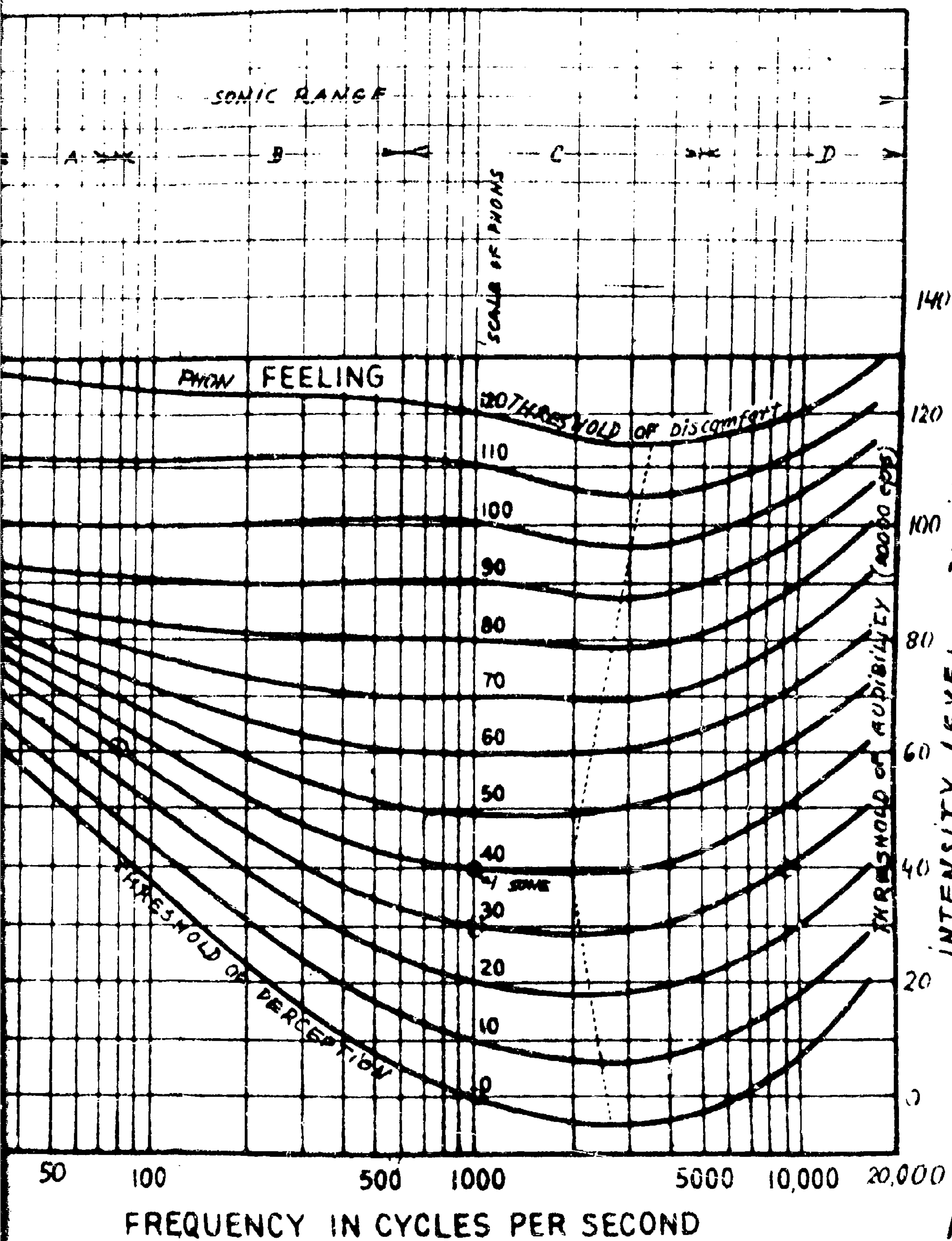
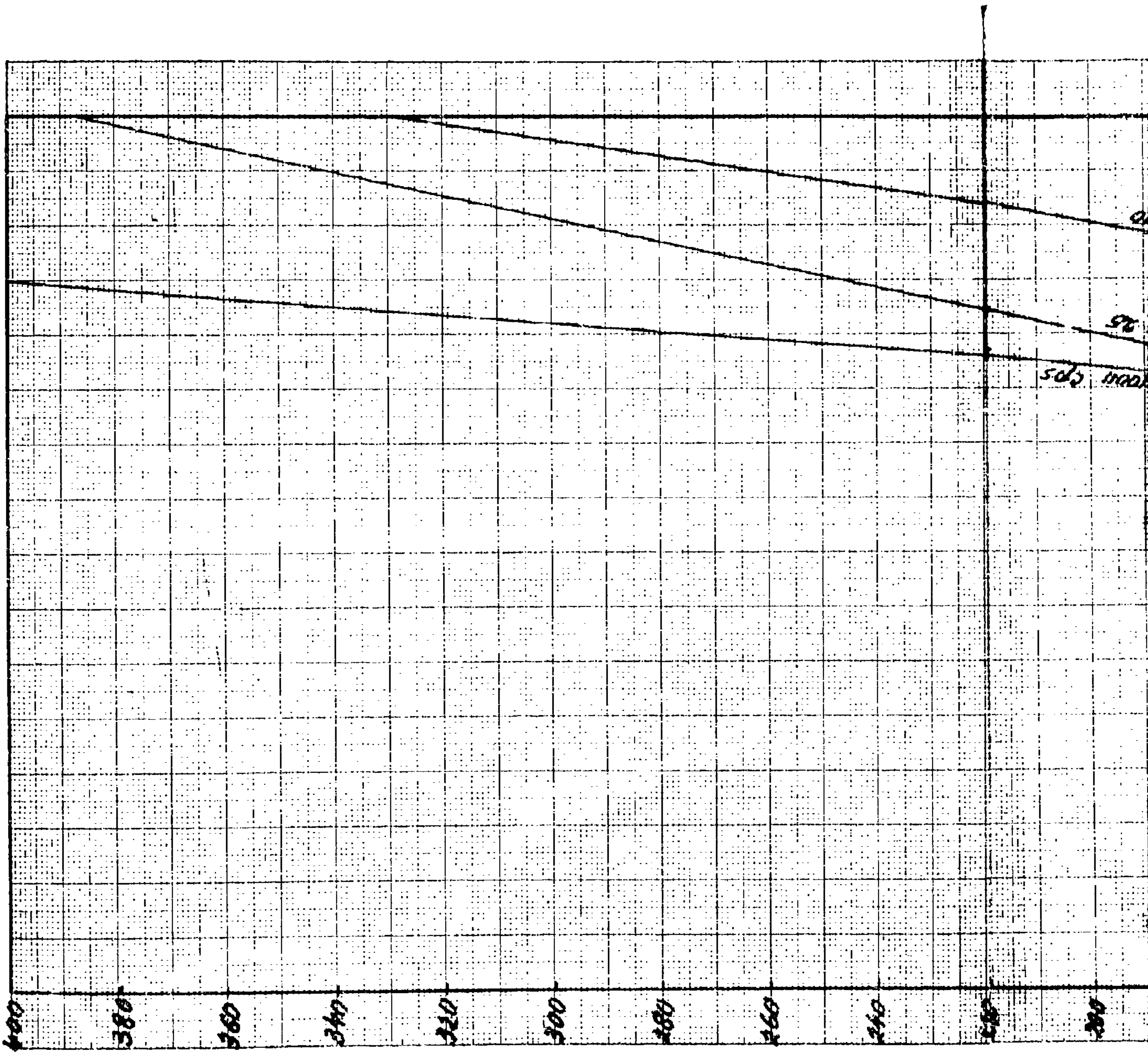

SONIC RANGE
A
B
C
D
SCALE OF PHONS
PHON
FEELING
120 THRESHOLD OF DISCOMFORT
110
100
90
80
70
60
50
40
30
20
10
0
1 SONE
THRESHOLD OF PERCEPTION
THRESHOLD OF AUDIBILITY
140
120
100
80
60
40
20
0
INTENSITY LEVEL DECIBEL
(SONIC RANGE)
50
100
500
1000
5000
10,000
20,000
FREQUENCY IN CYCLES PER SECOND

Fig. 2


400
380
360
340
320
300
280
260
240
220
200

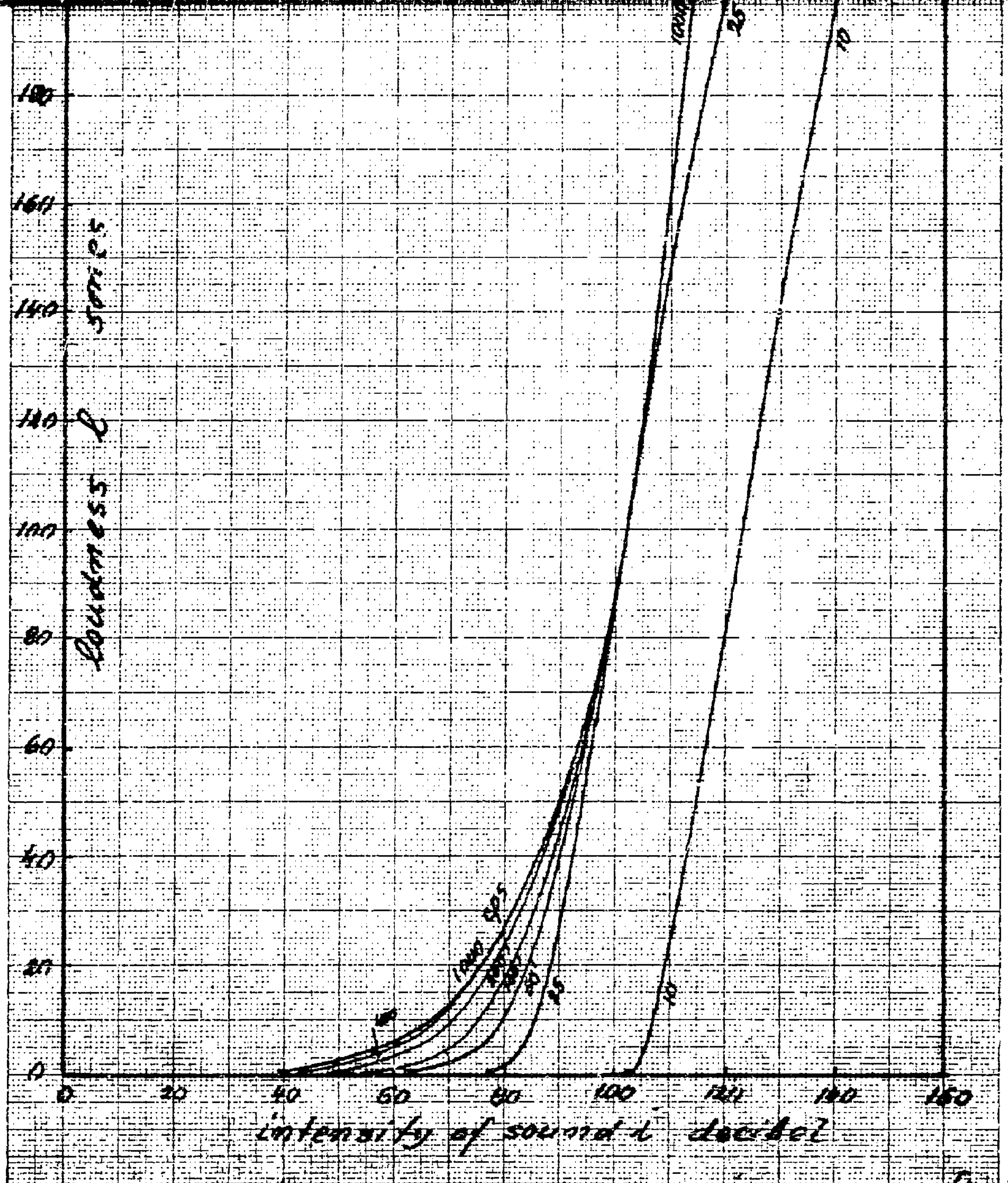

Loudness L sones
intensity of sound i decibel
0
20
40
60
80
100
120
140
160
180
10
25
1000

Fig. 3

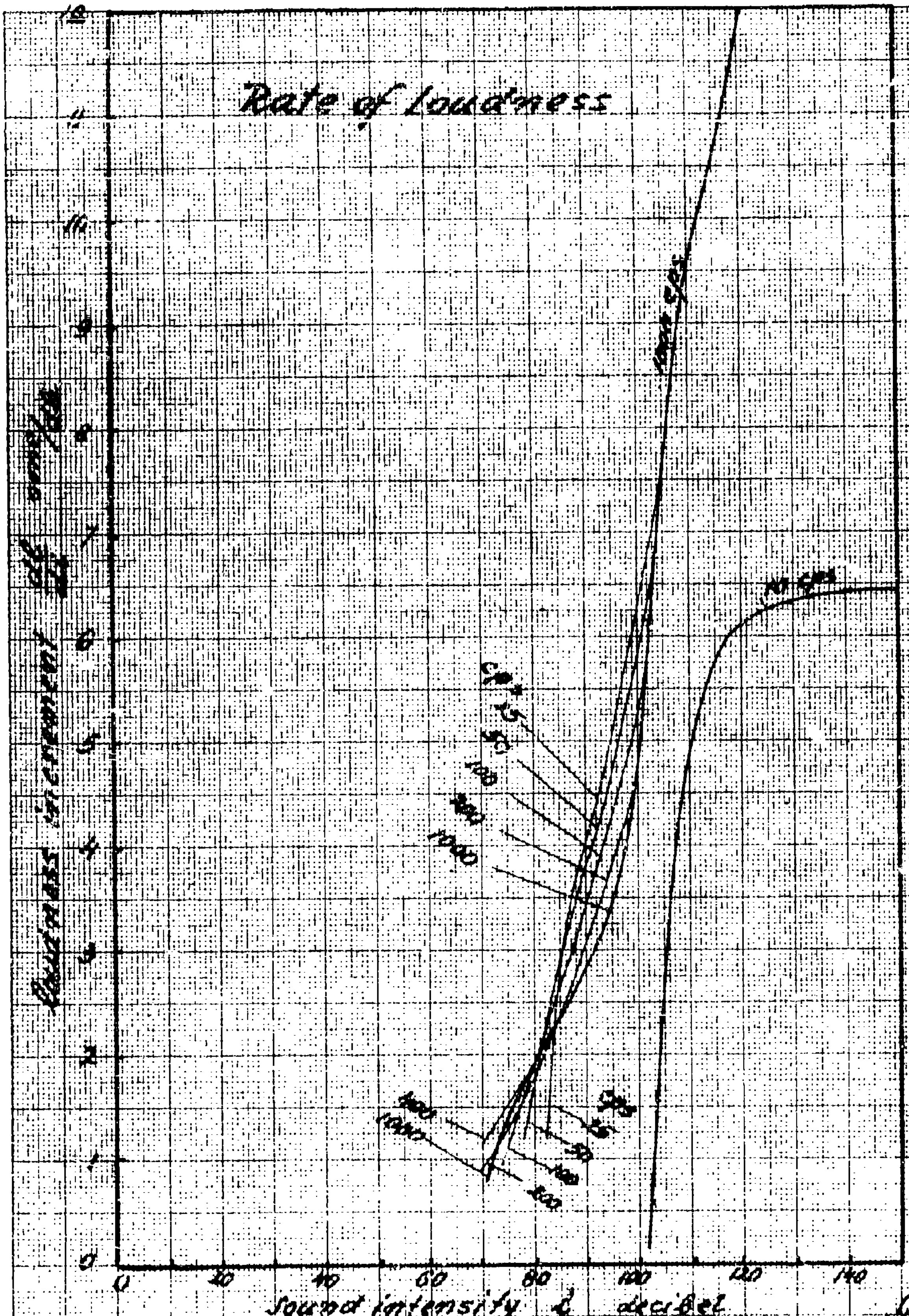

Rate of Loudness
Loudness increment
Sound intensity I decibel
0
20
40
60
80
100
120
140
1
2
3
4
5
6
7
8
9
10
11
12
1000 cps
10 cps

fig. 4