Jan. 21, 1969 G. E. WARNAKA 3,422,921

SOUND ATTENUATING WALL FOR BLOCKING TRANSMISSION OF INTELLIGIBLE SPEECH

Filed April 25, 1966

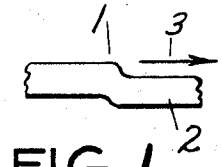


FIG. 1

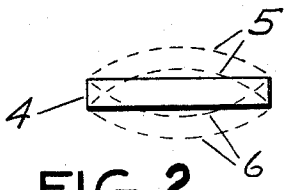


FIG. 2

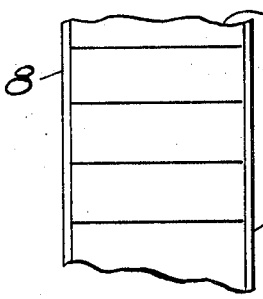


FIG. 3

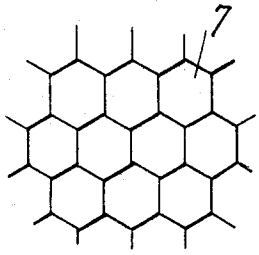


FIG. 4

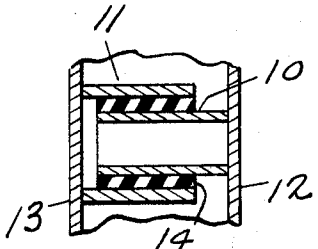


FIG. 5

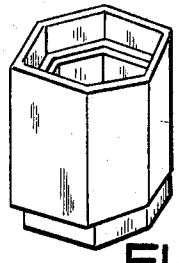


FIG. 6

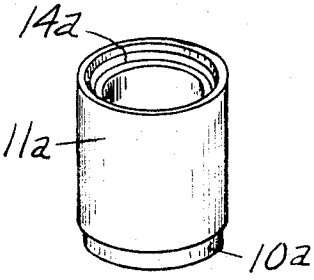


FIG. 7

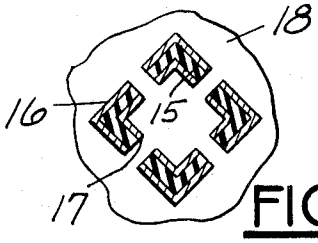


FIG. 8

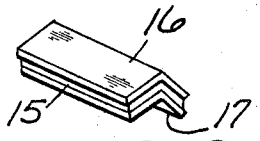


FIG. 9

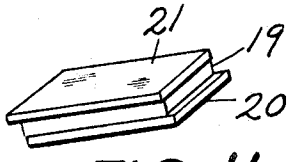


FIG. 11

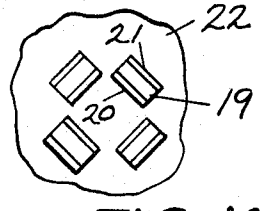


FIG. 10

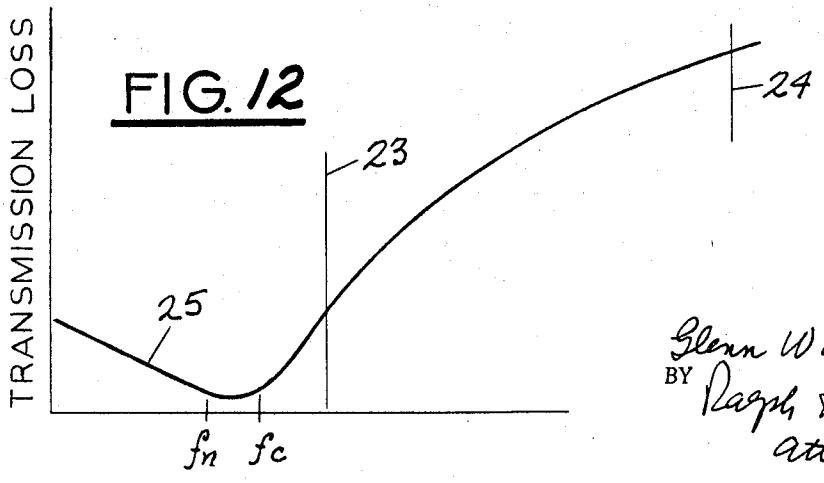


FIG. 12

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SOUND ATTENUATING WALL FOR BLOCKING TRANSMISSION OF INTELLIGIBLE SPEECH

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U.S. Cl. 181—33 12 Claims
Int. Cl. E04b 1/82

ABSTRACT OF THE DISCLOSURE

Transmission of sound in the band of frequencies required for transmission of intelligible speech is prevented by placing the critical frequency for bending waves in or below such band and by damping the bending waves. The foregoing features are applied to an acoustical partition or wall.

In buildings, there is need for walls and partitions substantially impervious to speech. This invention accomplishes this result by a damped wall panel in which the critical frequency for bending waves and the fundamental natural frequency of the panel are both in or below the band of frequencies (400 c.p.s. to 5,000 c.p.s.) required for transmission of intelligible speech. Sound waves striking such a panel are predominantly transmitted by travelling bending waves, and the damping of the bending waves reduces the transmission of intelligible speech through the panel by interferring with the normal propagation of the sound-radiating waves. The damping of bending waves is more effective than the damping of other vibrations with the result that by placing the critical frequency for bending waves at or near the upper end of the bass region, all frequencies above this region are effectively cut off. Cutting off all of the speech frequencies above bass essentially prevents transmission of intelligible speech through the wall panel.

In the drawing, FIG. 1 is a fragmentary diagrammatic edge view of a wall showing the local distortion of the panel due to a bending wave, FIG. 2 is a diagrammatic edge view of the wall panel showing the fundamental mode of vibration, FIG. 3 is an edge view of a honeycomb wall panel, FIG. 4 is a plan view of a portion of the honeycomb for FIG. 3, FIG. 5 is a cross sectional view through a modification using honeycomb sections in telescoping relation to each other, FIG. 6 is a perspective of one of the telescoping honeycomb sections of FIG. 5, FIG. 7 is a perspective of a telescoped tubular section for use in the panel, FIG. 8 is a sectional view in a plane midway between the skins of a modification, FIG. 9 is a perspective of one of the damping sections used in FIG. 8, FIG. 10 is a plan view broken away of another modification, FIG. 11 is a perspective of one of the sections used in FIG. 10, and FIG. 12 is a transmission loss diagram.

In the panel, both the fundamental natural frequency of vibration and the critical frequency for bending waves are below the frequencies to be suppressed. This condition favors the propagation of travelling bending waves on the panel in the frequency band of interest. The amplitude of the bending waves can be effectively reduced as the waves move along the panel, if the panel has high damping in bending. Consequently, the sound initiating the bending waves will not be effectively radiated from the wall because of the rapid reduction of the bending wave amplitude as the wave travels on the damped wall.

FIG. 1 shows diagrammatically a bending wave **1** moving along panel **2** in the direction of arrow **3**. At frequencies equal to and above the critical frequency, the speed of movement of the wave is equal to or greater

than the speed of sound in air as given by the equation

$$C = W^{1/2}\left(\frac{B}{m}\right)^{1/4}$$

where C is the bending wave speed, W is 2π times the excitation frequency, B is the flexural rigidity of the panel per unit width, and m is the mass of the panel unit area. The important characteristic of the panel is not the over-all flexural rigidity which would depend upon the gross dimensions of the panel but rather the flexural rigidity of its cross section.

When the panel is excited by a sound wave, the bending wave length will exactly equal the trace wave length (or projected wave length) of the exciting sound wave at some excitation frequency. This means that a crest of the exciting wave will exactly coincide with the crest of the excited bending wave, and the velocity of propagation of the bending wave is the same as the component of the exciting wave parallel to the plate. At this condition perfect coupling between the air-borne exciting wave and the bending wave excited in the wall exists. As a result, a large portion of the energy of the exciting wave is transferred to the bending wave. Sound is efficiently radiated from the other side of the wall because of the coincidence or matching of the bending wave speed to a component of the speed of sound in air or other elastic medium in contact with the other side of the wall. The lowest frequency for which this wave coincidence can occur is called the critical frequency and is given by the equation

$$f_c = \frac{a^2}{2\pi} \times \sqrt{\frac{m}{B}}$$

where a is the speed of sound in air in inches per second, m is the mass of the panel per unit area in lbs. sec.^{2}/in.^{3} and B is the bending stiffness per unit width of the cross sectional stiffness in lbs. in.^{3}/in.^{2} In the absence of damping, the panel becomes "transparent" to the exciting sound wave. The transmission loss drops to a very low level and the exciting sound wave would pass through the panel readily and be efficiently radiated from the other side as a sound wave. For this reason, it has been customary to place the critical frequency for bending waves above above the frequencies to be transmitted through the panel. The idea has been to prevent the efficient radiation of sound by bending waves above the critical frequency.

Bending waves should not be confused with the over-all resonant vibration of the panel. FIG. 2 shows a panel **4** vibrating in its fundamental mode alternately assuming the position of dotted lines **5** and **6**. The frequency at which this transverse vibration takes place depends upon the over-all stiffness of the panel determined by its dimensions and edge fixation and calculated in accordance with beam theory. The frequency at which the type of vibration illustrated in FIG. 2 takes place can be varied by varying either the over-all panel dimensions or the edge fixation or both and is preferably less than substantially 900 cycles per second. The frequency may be raised by decreasing the panel dimensions. A small panel will vibrate at a higher frequency than a large panel. Similarly, a panel rigidly fixed at its edges will vibrate at a higher frequency than the same panel with its edges free. In addition to the fundamental mode of vibration illustrated in FIG. 2, there are higher modes of vibration which occur at higher frequencies.

In order to make the wall impervious to intelligible speech, both the fundamental natural resonant frequency of the wall panel and its critical frequency of bending waves are placed below the frequencies of intelligible speech and damping is introduced having a component

effective transverse to the plane of the wall panel. Placing the fundamental natural frequency f_n of the panel at or somewhat below the critical frequency insures or favors the propagation of bending waves. Since the critical frequency is less than the frequencies of intelligible speech, it would be expected that the panel would become transparent to speech and would not act as an effective acoustic barrier. However, because of the damping introduced, the transmission loss of the panel is increased and all of the frequency band of intelligible speech (400–5,000 c.p.s.) is effectively cut off. The effectiveness of the damping of the bending waves increases with frequency at the rate of ten or twelve db per octave, substantially twice the rate of increase of transmission loss expected from panels working on the mass law. This high rate of increase of transmission loss continues for at least two octaves above the critical frequency and, therefore, includes the major components of intelligible speech.

FIGS. 3 through 11 show various expedients for damping the bending waves. In each of these constructions, there is a damping component normal to the wall surfaces. There may be damping in other directions. However, for damping the bending waves such as diagrammatically illustrated in FIG. 1, the component normal or transverse to the wall surfaces is most effective.

In FIGS. 3 and 4, there is a honeycomb **7** sandwiched between skins **8** and **9**. The honeycomb is made of extensional damping material having a loss factor at least substantially equal to 0.1. The skins **8** and **9** may be of structural material such as metal (steel, aluminum, magnesium, titanium, beryllium, molybdenum) or glass fiber reenforced plastic. If made of metal, the skins have negligible damping. However, some of the rigid plastics also have high loss factors in the range of 0.1. In a typical wall, the skins **8** and **9** might be made of aluminum 0.020 inch thick and the honeycomb might be of damping material and about six inches thick. This wall will be rigid to forces normal to the skins **8** and **9**. Because the material for the honeycomb **7** is made of extensional damping material, that is, material having high internal friction to tension and compression stresses, there will be high frictional damping of bending waves. The ratio of the stiffness to the mass is such that the critical frequency f_c is 300 to 400 cycles per second. A conventional wall panel four feet by eight feet made of this construction should have a fundamental natural frequency, f_n, below 300 cycles per second. When constructed in a wall panel for 16 inch centers, the fundamental natural frequency f_n of the panel vibrating as a whole would be less than 400 cycles per second.

In the modification of FIGS. 5 and 6, telescoping honeycomb sections **10** and **11** are respectively fixed at one end to skins **12** and **13**. The honeycomb sections are in telescoping relation and are secured to each other by an intermediate layer **14** of highly damped viscoelastic material or commonly known damping materials such as butyl rubber, certain vinyl plastic materials, asphaltic materials, etc. Instead of honeycomb sections **10**, **11**, tubular sections **10***a*, **11***a* connected by viscoelastic material **14***a* may be substituted. The telescoping relation of the honeycomb or tubular sections provides high damping due to shearing of the elastomer **14** or **14***a* upon relative motion of the skins **12**, **13** transverse to the skins. Since the elastomer **14**, **14***a* is essentially incompressible, the panels have great rigidity for shearing forces parallel to the planes of the skins **12**, **13**.

FIGS. 8 through 11 show other arrangements for providing acoustic damping in the direction normal to the skins. In FIGS. 8 and 9 a layer of viscoelastic damping material **15** is sandwiched between and bonded to angular sections **16** and **17** respectively fixed at one end to one of the skins of the panel. In FIG. 8 the angle **16** is shown fixed to the skin **18** and the other skin of the panel has been removed. The angles are arranged in a pattern such that movement parallel to the skins is inhibited while movement perpendicular to the skins shears the viscoelastic layers **15** and thereby introduces damping.

In FIGS. 10 and 11, a layer **19** of highly damped viscoelastic material is sandwiched between and bonded to rigid plates **20** and **21** respectively secured to the skins of the wall. In FIG. 10 the skin of a plate **21** is shown fixed to the skin **22** and the other skin of the panel has been removed. The damping elements **19–21** are arranged in a pattern such as the hollow square shown resisting movement parallel to the skins while introducing damping by relative motion of the skins toward and away from each other.

FIG. 12 is a performance diagram. The frequencies necessary for intelligible speech which range from about 400 to 5,000 cycles per second are indicated by the lines **23** and **24**. The fundamental natural frequency of the panel f_n is about 300 cycles per second. The critical frequency for bending waves f_c is about 350. The transmission loss is shown by curve **25**. Below the fundamental natural frequency f_n the transmission loss has a slope of from 5 to 6 db per octave. Above the critical frequency f_c the transmission loss has a slope of 10 to 12 db per octave at least for the first two octaves above the critical frequency. It will be noted that the frequencies below the critical frequency f_c are only slightly attenuated by the wall. However, at frequencies above the critical frequency not only is the sound attenuated but the rate of increase in attenuation is much faster than would be predicted from the mass law acoustical theory. Because of the transparency of acoustical walls to bending waves of frequency above the critical frequency, ordinarily an effort is made to place the critical frequency as high as possible. Typically, f_c would be above 4,000 c.p.s. and there would be a substantial spread between f_n and f_c. Also, between f_n and f_c there would be a number of higher modes of resonant vibration of the wall. These higher modes are effectively eliminated by placing both f_n and f_c below the band of frequencies of interest.

It is true that there is little attenuation of the low frequency components of speech. However, the attenuation of the speech frequencies above 400 cycles per second practically prevents the transmission of intelligible speech. This is an important advantage in offices, hotels, multifamily dwellings and other places requiring speech privacy.

From the efficiency of transmission and radiation of bending waves disclosed in Patent 3,247,925, it would be expected that placing the critical frequency f_c at substantially 400 cycles per second would result in increased transmission of speech. This does not happen because the panel has damping which is highly effective for bending waves.

What is claimed as new is:

1. A partition or wall for blocking the transmission of sound in the band of frequencies required for the transmission of intelligible speech comprising a panel having its critical frequency for bending waves, as determined by the equation

$$f_c=\frac{a^2}{2\pi}\sqrt{\frac{m}{B}}$$

where a is the speed of sound in the medium in contact with the panel in in./sec., m is the mass of the panel per unit area in lbs. sec.^{2}/in.^{3} and B is the bending stiffness per unit area in lbs. in.^{3}/in.^{2}, substantially a low frequency in the band of frequencies required for transmission of intelligible speech and having a loss factor in directions normal to the surface of the panel at least as large as substantially 0.1 for frequencies above its critical frequency.

2. The wall of claim **1** in which the fundamental resonant frequency of the panel is less than substantially 900 cycles per second.

3. The wall of claim **1** in which the fundamental resonant frequency of the panel is less than its critical frequency.

4. The wall of claim 1 having a honeycomb core and facing skins.

5. The wall of claim 1 in which the panel has spaced skins with rigid members projecting from and transverse to each skin and toward the other skin, the members on one skin having sections transverse to the skins in overlapping and sliding relation in a direction transverse to the skins to the members on the other skin, and a highly damped elastomer sandwiched between and bonded to the overlapping sections of said members.

6. The wall of claim 4 in which the core is of extensional damping material.

7. The wall of claim 5 in which the members on one skin are in telescoping relation to the members on the other skin.

8. The wall of claim 5 in which the overlapping sections of said members are arranged to block relative shearing movement of the skins.

9. The wall of claim 4 in which the skins are selected from the group consisting of glass fiber reenforced plastic and the metals steel, aluminum, magnesium, titanium, beryllium, and molybdenum.

10. The wall of claim 1 in which the critical frequency for bending waves is substantially equal to or less than 400 cycles per second.

11. The wall of claim 10 in which the fundamental resonant frequency of the panel is less than substantially 900 cycles per second.

12. The wall of claim 10 in which the fundamental resonant frequency of the panel is less than its critical frequency.

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ROBERT S. WARD, JR., *Primary Examiner.*