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C. HUBRICH

3,279,560

CYCLONE SOUND ABSORBER

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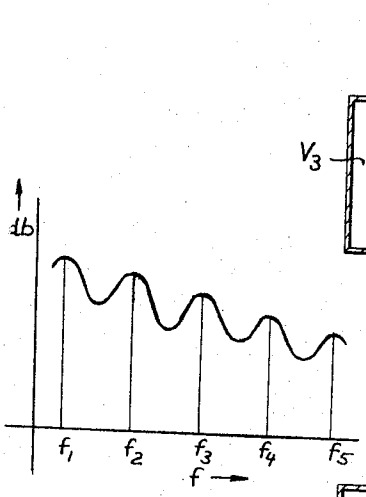


FIG-4

FIG-1

FIG-3

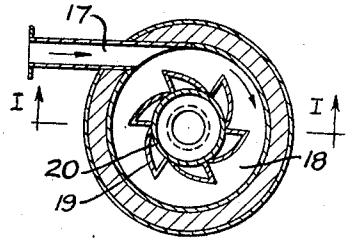


FIG-2

INVENTOR.
Christoph Hubrich

BY

Walter Bubey

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CYCLONE SOUND ABSORBER

Christoph Hubrich, Offenbach (Main), Bieber, Germany, assignor to Polysius G.m.b.H., Neubeckum, Germany

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The present application is a continuation-in-part of my copending application Serial No. 345,933, filed February 19, 1964, now abandoned, and entitled Cyclone Sound Absorber.

The present invention relates to a method and apparatus for damping sound and, in particular, to such a method and apparatus wherein a cyclone absorber is provided with one or more resonator stages.

In the above-mentioned art and also in connection with other flow technical problems, it sometimes occurs that, particularly with high pressure gases, vibratory or flow energy is developed not only by the pressure of said gases, or by a corresponding heat treatment, but that the said vibrator or flow energy is additionally created by mechanical delivery machines. In such instances, the pure flow sound spectrum is overlapped by a delivery sound spectrum which is characterized by the number of the feeding or delivering elements, such as blades, pistons, and the like, multiplied by the speed of the machine. This delivery frequency is in most instances considerably lower than the flow frequencies so that it is not absorbed by the absorption portion of the cyclone sound absorber, which is essentially a high frequency sound absorber.

Resonance chambers located between the expansion stages of the cyclone sound absorber are in most instances for structural reasons and also in view of the above-mentioned difference in frequency, too small to be able to affect the relatively low frequencies of the machine sound.

It is, therefore, an object of the present invention to provide a cyclone sound absorber with one or more resonator stages, which will overcome the above-mentioned drawbacks.

It is further an object of this invention to provide a cyclone sound absorber which will include means adapted to damp the above-mentioned lower delivery frequencies.

These and other objects and advantages of the invention will appear more clearly from the following specification in connection with the accompanying drawing, in which:

FIGURE 1 diagrammatically illustrates partly in elevation and partly in section the assembly of a sound damper installation according to the present invention and, at least as to the cyclone sound absorber, is a section on line I—I of FIGURE 2;

FIGURE 2 is a section taken along line II—II of FIGURE 1;

FIGURE 3 is a diagram showing the sound damping process by way of graphs;

FIGURE 4 diagrammatically illustrates a sound spectrum produced by a machine and to be damped by the resonator according to the present invention.

According to the present invention, there is provided a special resonance system which may be connected to the cyclone sound absorber and may be coupled thereto. In this system, the cyclone chamber of the cyclone sound absorber may likewise be dimensioned in conformity with a Helmholtz resonator for the occurring machine sound. In connection with a further development of the present invention, the said resonator may have connected thereto an accoustic barrier tuned to said resonator, which barrier will prevent the gas volume vibrating in resonance in the resonator from escaping toward the outside as sound.

Referring now to the drawing in detail and FIG. 1

thereof in particular, the installation shown therein comprises an intake pipe **1** of a gas delivery machine **2** into which a gas under more or less pressure enters. The machine **2** imparts velocity or pressure energy upon the gas, depending on the particular construction of the machine. In this way, a certain definite delivery frequency is impressed upon the gas flow from the discharge pipe **3** on, which delivery frequency is mainfested as disturbing sound in the adjacent conduit and/or perhaps also at the exit into the atmosphere. Connected to discharge pipe **3** is a cyclone sound absorber **4** for expanding the gas flow and absorbing the flow sound. Absorber **4** includes a casing **4***a* and has a closed bottom. The absorber is generally conical, as will be seen in FIG. 1, and is lined with sound absorbing material **4***b*. This cyclone absorber may be basically of the type shown in U.S. Patent 3,130,812.

FIG. 2 shows a section taken along the line II—II of FIG. 1 through the cyclone sound absorber **4** and clearly shows the inlet pipe **17** tangentially leading into the cyclone sound absorber **4** and, more specifically, into the annular chamber **18**. Connected to the inner wall of chamber **18** are serially arranged plates **19** forming expansion nozzles and narrowing the passage through which the gas flows. Coaxially arranged with regard to chamber **18** is a funnel-shaped pipe **20** which flares in upward direction, i.e. in the direction of flow of gas and which pipe extends through the top wall of the absorber and forms the said inner wall of chamber **18**.

Referring back to FIG. 1, it will be noted that the cyclone sound absorber **4** has connected thereto a resonance sound dampener or chamber **6** from which leads a conduit **7**, which is designed to damp a certain range of the low frequency machine sounds. Sound damper **6** with the conduit **7** may briefly be called a resonator stage. Chamber **6** is in a manner known per se designed as a resonator, the natural frequency of which equals the peak frequency of the machine sound, while the barrer resonator length **7** has a natural frequency having a wave length corresponding to ¼, ¾ or 5/4 or more of the wave length of the natural frequency of chamber **6**. The design and construction of elements **6** and **7** can be arrived at by observing the known laws of Helmholtz.

Normally, such resonator arrangements have a plurality of chambers in order to be able to damp the sound of higher orders, i.e. the overtones or harmonics of the machine sound. Another resonator stage is designated by reference numerals **21** and **22**.

Thus, in conformity with the persent invention, the inner chamber of the cyclone sound absorber **4** may be equipped with such inner volume that the natural frequency thereof will be identical to the delivery frequency of the machine **2** so that the interior of the cyclone sound absorber **4** will already take over the function of a first resonance chamber. The discharge pipe **5** connected by flange means **5***a* to the upper end of pipe **20** will be so dimensioned that it forms an acoustic barrier length for this first resonance chamber represented by the cyclone sound absorber **4**. This discharge pipe **5** may then be followed by further resonance systems as shown in the drawings, each stage or system damping or forming a barrier for a particular narrow range of frequencies.

The sound damping process is illustrated in FIG. 3 in which over the abscissa there is plotted the sound intensity, whereas over the ordinate there is plotted the time. Inasmuch as the inner chamber of the cyclone sound absorber **4** has been so calculated that its gas volume has a natural frequency which is identical to the disturbing machine frequency, it will be appreciated that in this chamber the calculated gas vibration will be in

resonance. This will assure that at the start of the cyclone pipe the vibration will have its greatest amplitude. This has been shown in the diagram of FIG. 3. The vector circle **11** of the gas vibration has been shown around the point **8** of the coordinate system with the abscissa **9** and ordinate **10**. This vector circle **11** serves to ascertain the individual vibration points when drawing up the diagram. The simplest case in which the acoustic barrier pipe **5** ends at the zero point of the vibration **13** will be obtained when omitting the resonator system **6**, **7**.

In this instance, the sound vibration **13** will be contained in the resonance chamber of the cyclone sound absorber **4**. If, however, the interior chamber of cyclone sound absorber **4** is to contain, for instance, a vibration as it is designated in the drawing with the reference numeral **15**, which is at ½ the frequency of vibrator **13**, it is possible either to give the discharge pipe a length of from **8** to point **16** or the said discharge pipe may be interrupted at a favorable point, for instance in the center, and the interrupting point may be placed in a further resonance chamber **6**. In this instance, the several related frequencies **12**, **13**, **14** and **15**, and various other harmonics or resonant frequencies would be damped which have their zero points at the interrupting point as indicated, for instance, in connection with the vibration **12**. It is thus possible to intensitfy the damping effect and expand the spectrum of absorption by simple means.

More particularly, FIG. 3 illustrates the course of various sound waves **12**, **13**, **14** and **15** in a sound intensity time graph in which the sound intensity is plotted on the abscissa and the time on the ordinate. This graph shows the origin of the abscissa for the curves, which corresponds to the lowermost point of the tube **5** of the cyclone sound absorber. The intersection of curve **13** with the abscissa is located on the same level as the upper end of tube **5**. The intersection **16** of the sound wave curve **12** with the ordinate is located where the conduit **7** ends. The sound curve **14** starts approximately in the middle of the ordinate very steeply and rises to a height which equals the intensity of the sound as it leaves the tube **5**. Its amplitude equals that of the sound in the resonator chamber of the cyclone sound absorber **4**, i.e., equals the starting amplitude of the above-mentioned sound curve **13**. In addition thereto, also the second harmonic **12** has its second zero point coinciding with the zero point of the curve **14** at **16**.

The simplest case, for instance, with an unequivocally pure sound vibration would be obtained when the interior of the cyclone absorber **4** would be designed as resonator, i.e. has such an inner volume that its natural frequency is identical to the delivery frequency of the machine. In this way the sound carrying gas stream is forced to employ or use up its sound energy at the respective frequency (i.e. at the delivery frequency of the machine) for causing the inner cyclone volume to vibrate. Thus, in the cyclone inner chamber there will prevail a gas vibration vibrating within the resonance range. The discharge pipe **5** tuned and precisely calcuated as to its length in conformity with the just-mentioned resonator volume, forms an acoustic barrier which has a length of ¼ of the wave length (or ¾, or 5/4) of the machine frequency. This acoustic barrier permits the stationary sound vibration in the resonator chamber to escape only with the sound intensity of zero. In other words, the said sound barrier prevents the sound intensity from leaving the resonator chamber.

If the gas or air escapes into the free atmosphere, the discharge pipe **5** must have the just-mentioned acoustic barrier length. If, however, the gas or air does not yet escape into the atmosphere, it is necessary at this point considerably to widen the pipe, for instance into a chamber which, of course, should have a favorable reasonator volume adapted to contain the sound. The sound in the resonator chamber **4** will not be able to skip this widening of the pipe or could do so only at a rather poor degree of

efficiency. The connecting pipe **7** will then convey an expanded gas column which, to a great extent, is freed from the flow sound and from the machine sound.

Another situation finally would be encountered when the conduit **7** does not lead into the free atmosphere but instead leads into a pipe line **22** leading to a point of use of the gas. In this instance, the system does not end at point **16** but continues. In this case it is necessary at the zero point of the disturbing vibration, for instance at **16**, to provide a widening of the pipe, for instance by the provision of a chamber similar to the chamber **21**, preferably in the form of a favorable resonator. This chamber will be such that the disturbing sound will not be able to skip over the same, or at best, only at a poor degree of efficiency.

By way of more specific designation of the manner in which the present invention is practiced, the following formulae are employed in calculating volumes and lengths of conduit in the system.

Volume of a resonator chamber:

$$V=\frac{C^2\mu}{4\pi 2f_x^2}$$

wherein:

C=sound velocity
f_x=frequency to be damped in resonator
μ=an acoustic guiding value for the outlet opening of the resonator.

$$\mu=\frac{r^2}{l+\pi/2r}$$

wherein:

r=radius of the outlet opening
l=length of overflow slot, namely, the gap between the inlet and outlet conduits of the resonator, for example, the gap between conduits **5** and **7** of resonator chamber **6**.

$$L=\frac{1}{4}\cdot\lambda=\frac{c}{4f_x}$$

wherein:

L=length of the sound barrier
λ=the wave length blocked by the particular barrier conduit
C=sound velocity
f_x=frequency to be blocked.

The invention may be explained in further detail in connection with FIGS. 3 and 4. FIG. 4 diagrammatically illustrates a sound spectrum produced by machine **2**. Over the abscissa representing the frequency there is plotted the sound intensity in decibels (db). Machine **2** produces a sound spectrum containing the peak frequencies f_1 to f_5. The volume V_1 of cyclone sound absorber **4** is so selected that it possesses, as a resonator volume, a natural frequency equally the disturbing frequency f_1. In this way disturbing frequency f_1 is forced to consume its sound energy for causing volume V_1 to vibrate. Volume V_1 vibrates in resonance, i.e. with its largest possible amplitude, and thereby consumes the maximum possible sound energy.

The resonator length L_1 is so selected that it amounts to ¼ or ¾ of the wave length corresponding to frequency f_1. Therefore, the sound wave corresponding to f_1 at the gap l_1 has the amplitude zero, i.e. the line representing this oscillation intersects the zero line. Therefore, the vibration produced in the resonator chamber can leave gap L_1 only with the amplitude zero, i.e. is unable to leave this gap.

The frequencies f_2 and f_3 are overtones, i.e. integer multiples of f_1. They are, therefore, also adapted to cause resonator chamber V_1 to vibrate. However, the wave lengths corresponding to frequencies f_2 and f_3 are different and it is, therefore, necessary to provide different resonator barrier lengths L_2 and L_3.

The frequency f_4 produced by machine **2** is so different from frequency f_1 that it is unable to cause a vibration of resonator volume V_1. Therefore, it leaves gap l_1 in undisturbed manner. In order to dampen this frequency f_4 a resonator volume V_2 is provided at the gap l_1, and frequency f_4 is adapted to vibrate volume V_2 in resonance in the same manner as frequency f_1 was able to vibrate resonator volume V_1 in resonance. The sound wave corresponding to frequency f_4 can leave the resonator length L_4 at gap l_2 again only with the amplitude zero. Thus, also frequency f_4 is damped in the sound damping installation according to the present invention and can cause no further disturbances. It is, of course, to be understood that the same applies for volume V_3 which serves for damping frequency f_5.

The diameter of the inlet opening of pipe **20** is so selected that the opening does not disturb the flow of the gases in the cyclone sound absorber. This is normally the case if the diameter of the inlet opening amounts to about 0.5 to 0.7 of the inner diameter of the cyclone sound absorber in the plane of the inlet opening.

It is, of course, to be understood, that the present invention is, by no means, limited to the particular arrangements shown in the drawing, but also comprises any modification within the scope of the appended claims.

What I claim is:

1. A sound damping installation which includes: a cyclone type sound absorber having an upper annular chamber, a plurality of serially arranged plate means connected to one of the walls of said chamber and extending into and forming gas expansion nozzle means in the passage formed by said annular chamber, a feeding conduit tangentially leading into said annular chamber for feeding a gaseous medium thereinto, funnel-shaped tubular discharge means forming the inner wall of said chamber and extending into said annular chamber from the top in coaxial relationship therewith, said funnel-shaped tubular means tapering inwardly toward the bottom and having an axially arranged inlet at the narrow end thereof and having an outlet at the opposite end externally of said sound absorber, the volume of said cyclone type sound absorber having the same natural frequency as the maximum peak of the disturbing machine sound to be damped, the said tubular means having a length equalling an odd-numbered multiple of ¼ of the wave length of the natural frequency of the volume of the said cyclone type sound absorber.

2. A sound damping installation which includes: a cyclone type sound absorber having an upper annular chamber, a plurality of serially arranged plate means connected to one of the walls of said chamber and extending into and forming gas expansion nozzle means in the passage formed by said annular chamber, a feeding conduit tangentially leading into said annular chamber for feeding a gaseous medium thereinto, funnel-shaped tubular discharge means forming the inner wall of said chamber and extending into said annular chamber from the top in coaxial relationship therewith, said funnel-shaped tubular means tapering inwardly toward the bottom and having an axially arranged inlet at the narrow end thereof and having an outlet at the opposite end externally of said sound absorber, and additional resonance damping means connected to said outlet, the volume of said cyclone type sound absorber having the same natural frequency as the maximum peak of the disturbing machine sound to be damped, the said tubular means having a length equalling an odd-numbered multiple of ¼ of the wave length of the natural frequency of the volume of the said cyclone type sound absorber.

3. A sound damping installation according to claim **2** in which said resonance damping means includes more than one resonance chamber.

4. A sound damping installation which includes: a cyclone type sound absorber having an upper annular chamber, a plurality of serially arranged plate means connected to one of the walls of said chamber and extending into and forming gas expansion nozzle means in the passage formed by said annular chamber, a feeding conduit tangentially leading into said annular chamber for feeding a gaseous medium thereinto, conduit means coaxial with said absorber and leading from inside the absorber out through the top thereof, said conduit means being interrupted at points therealong which are spaced from the inlet end of the conduit means distances equal to odd multiples of ¼ of the wave length of a vibration to be damped, and resonance chambers through which said conduit means extend and which are located along said conduit means so as to enclose said points of interruption of said conduit means.

References Cited by the Examiner

UNITED STATES PATENTS

3,130,812	4/1964	Hubrich	181—58

FOREIGN PATENTS

1,060,597	11/1953	France.
802,205	2/1951	Germany.
97,758	1/1940	Sweden.

RICHARD B. WILKINSON, *Primary Examiner.*