United States Patent

[11] **3,580,357**

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[54] **WAVE INTERFERENCE SILENCING SYSTEM**
4 Claims, 14 Drawing Figs.

[52] **U.S. Cl.** **181/44,** 181/56, 181/67
[51] **Int. Cl.** **F01n 1/06,** F01n 1/12
[50] **Field of Search** ..... 181/33.04, 44—46, 56, 66, 67, 36, 36.2, 33

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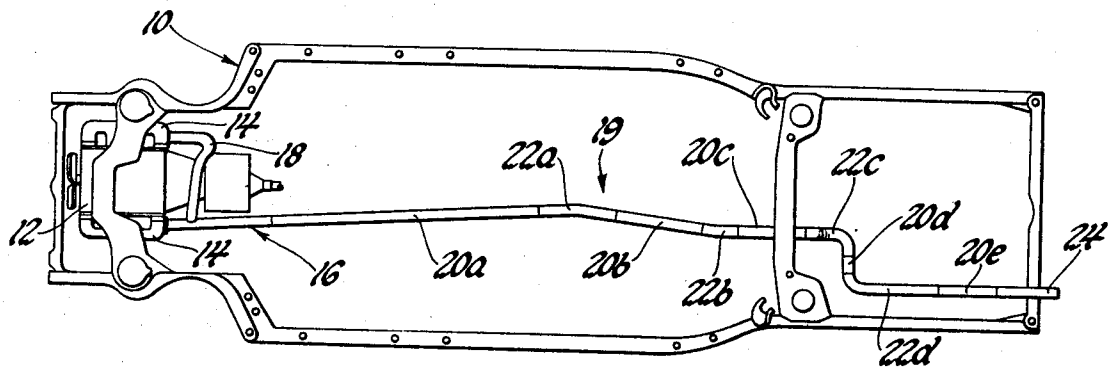
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ABSTRACT: A wave interference silencing system for predictably attenuating sound waves in the frequency noise spectrum of the exhaust gases from an internal combustion engine wherein a plurality of standardized serially connected attenuating sections includes a pair of discrete flow paths of differing acoustical lengths that are energy and flow compensated to establish actual sound propagational velocities so that the pressure waves of the individual paths are in phase opposition at the exit end of the transited attenuating section so as to destructively interfere within a low-impedance coupling volume adjacent thereto. The coupling volume serves to equalize the vibrational energy between the separate flow paths and effect a flow stabilization at the entrance end of a succeeding attenuating section.



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SHEET 1 OF 2

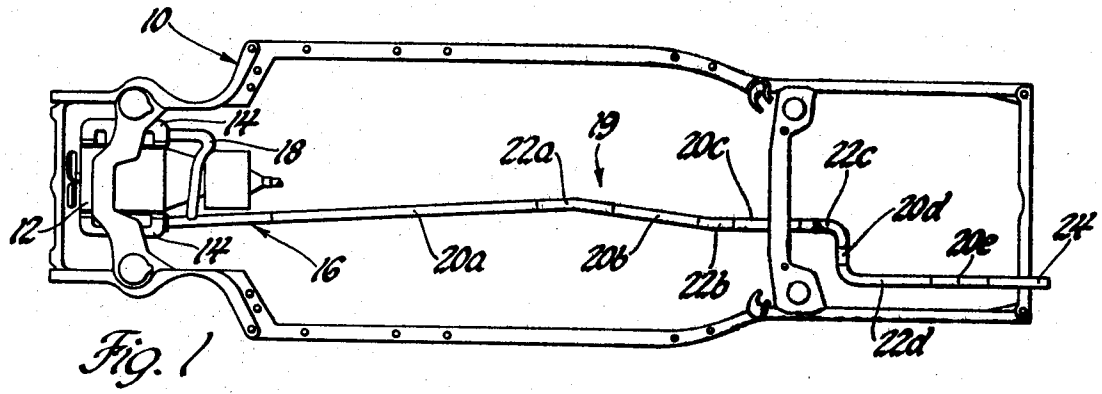


Fig. 1

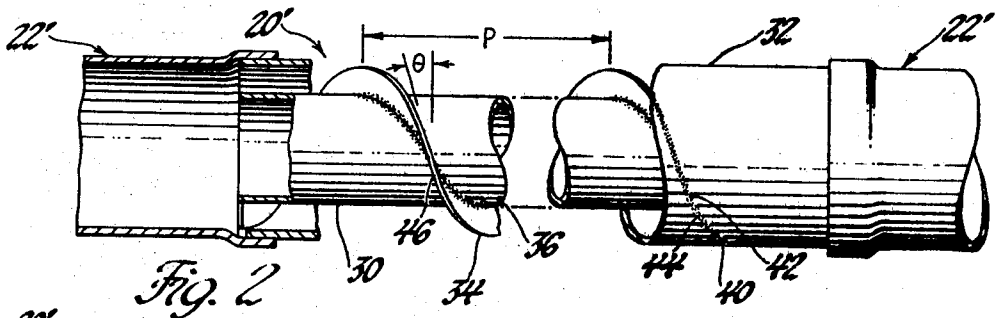


Fig. 2

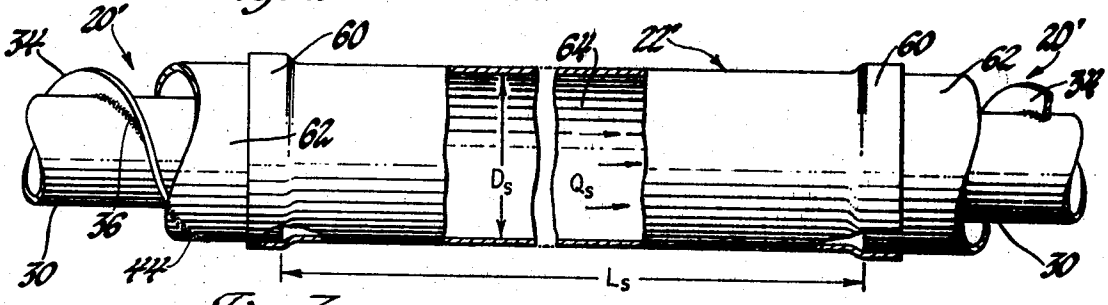


Fig. 3

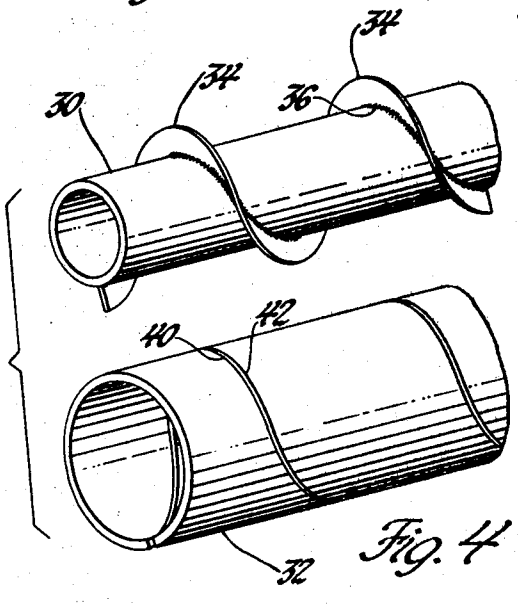


Fig. 4

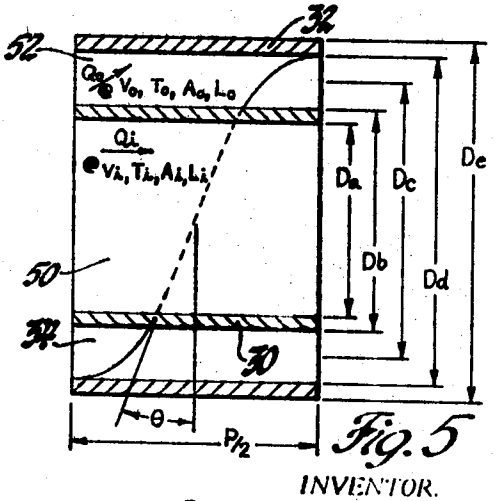


Fig. 5

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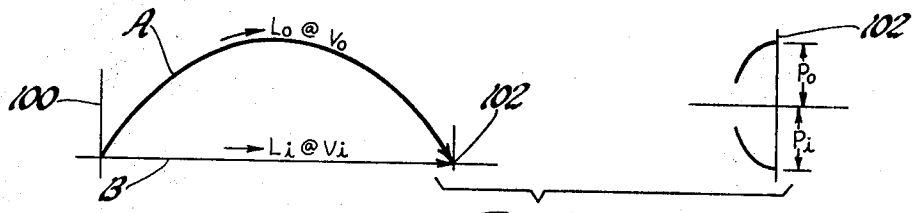


Fig 6a

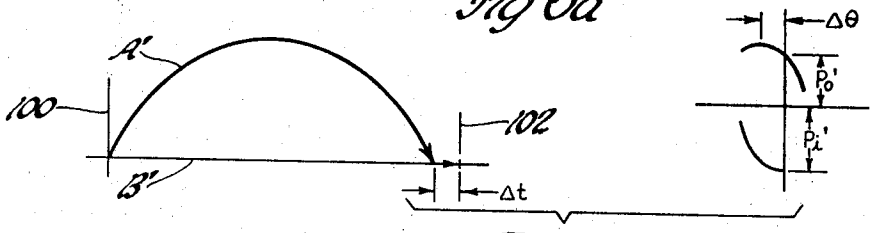


Fig. 6b

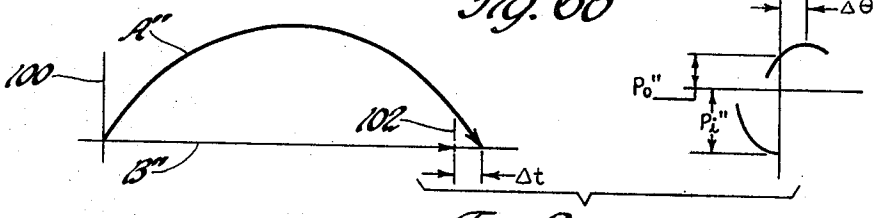


Fig. 6c

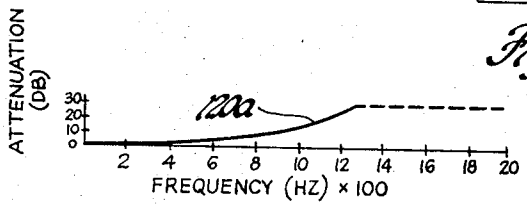


Fig. 7a

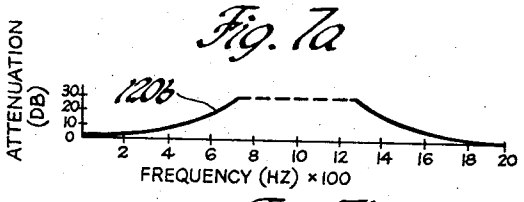


Fig. 7b

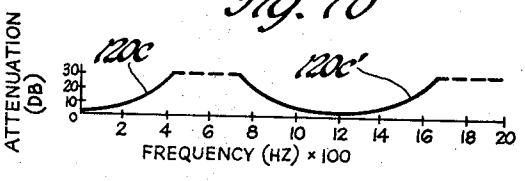


Fig. 7c

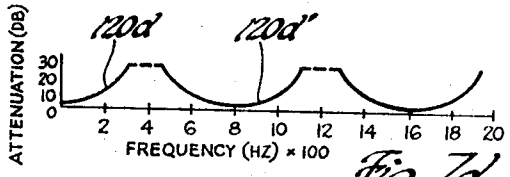


Fig. 7d

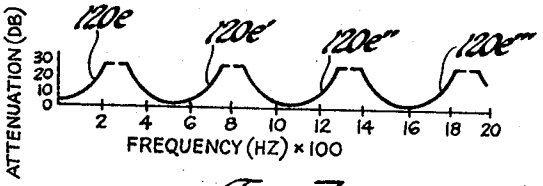


Fig. 7e

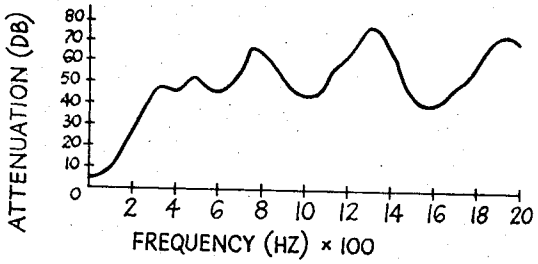


Fig. 7f

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WAVE INTERFERENCE SILENCING SYSTEM

This invention relates to exhaust silencing systems for motor vehicles and, in particular, to wave interference silencing systems incorporating a plurality of serially connected attenuating sections.

The periodic noises produced by an operating motor vehicle are generally comprised of engine-related, low frequency noises ranging around 200 Hz. and high frequency noises ranging around 700 to 2,000 Hz. The low frequency noises emanate from the exhaust tailpipe and are primarily a function of engine-operating conditions, the exhaust system pipe lengths, and the sound velocity in and of the exhaust gas. These noises are conventionally suppressed by nonflow resonant chambers utilizing a spring mass damping concept to silence a particular undesirable frequency. These nonflow chambers, depending on the acoustical nature of the emitted noises and the space availability beneath the vehicle chassis, may be incorporated into the system as separate components or as parts of the vehicle muffler. The high frequency noises are generally a function of the combustion process, the engine speed, restrictions in the gas flow stream, and the higher harmonics of exhaust pipe lengths. These frequencies are normally suppressed by flow sections incorporating baffles or labyrinths within the engine muffler. Of course, other well-known configurations can also be used to supplement the above components in either frequency range.

Regardless of the combination of acoustical elements adopted for these systems, the design for a given system requires a trial-and-error selection and matching of individual attentuating devices of varying sizes and characteristics in order to produce an acceptable silencing for a single-engine size. Due to the unpredictability of the interaction between the silencing components, a completely independent design process is necessary to achieve the desired attenuation for differing engine sizes.

The present invention contemplates providing attenuating sections of a single sectional design that can be serially incorporated into an exhaust system to predictably produce predetermined attenuation in the exhaust gas noise spectrum. Generally, these beneficial results are accomplished by a novel application of the principles of acoustical wave interference. The present arrangement takes the form of a pair of parallel flow paths having differing actual acoustical lengths so as to appreciably place the sound waves in phase opposition at the ends of an individual section. In this manner, the gas streams travel on separate paths and the pressure waves are substantially one-half period or 180° out of phase at the exit end of the transited attenuating section. Therefore, insofar as a given section is concerned, the output pressure due to the mixing of the separate pressure waves produces a complete pressure stabilization and a band of attenuation on either side of the design frequency. Of the wave interference attenuating devices which have been heretofore proposed in the prior art, none have provided devices which adequately compensate for the variety of noises and conditions presented by motor vehicle exhaust lines. For instance, most devices have merely arbitrarily established mechanical length ratios which evidence little appreciation for the environmental conditions under which the device is to be operated. Without adequate regard for operating temperatures, flow processes, and flow energies of the individual flow paths, the attenuation thus provided is as unpredictable in frequency response as conventional systems. Moreover, individual devices of this type without adequate compensation for environment are not consonant in plural usage to form a coherent exhaust silencing system.

The present invention, on the other hand, utilizes a plurality of wave interference attenuating sections having a standardized cross-sectional design varying only in section lengths which are individually and collectively designed to produce a predictable noise suppression. Such sections also provide an exhaust system wherein the several sections are individually flow compensated both with regard to temperature and flow velocity so as to produce discrete flow paths having a maximum phase angle difference for a given system length while, at the same time, maintaining a minimum pressure loss due to flow velocity for a given pipe diameter. This standardized section can be appropriately sized insofar as section length is concerned to provide a predictable attenuation response at select frequency bands within the noise spectrum. Inasmuch as the standardized sections are independent of engine and environmental conditions, the entire system can be analytically designed by subsequent consideration of a single parameter, namely, the section length necessary to individually and collectively produce the requisite attenuation over the entire frequency spectrum.

These and other features will be apparent to one skilled in the art upon reading the following detailed description, reference being made to the accompanying drawings in which:

FIG. **1** is a plan view of a motor vehicle incorporating a wave interference silencing system made in accordance with the present invention;

FIG. **2** is a fragmentary cross-sectional view of an attenuating section;

FIG. **3** is a fragmentary cross-sectional view of an acoustical equalization conduit;

FIG. **4** is an exploded view of an attenuating section;

FIG. **5** is an enlarged view of a portion of an attenuating section;

FIGS. **6***a* through **6***c* are schematic views illustrating the effect of environmental flow conditions on sound propogational velocity and wave interference attenuation;

FIGS. **7***a* through **7***e* are graphs of predominant attenuating bands versus frequency for the individual attenuating sections; and

FIG. **7***f* is a composite graph of attenuation versus frequency for the wave interference silencing system shown in FIG. **1**.

Referring to FIG. **1**, there is shown a frame **10** for a motor vehicle on which there is mounted an internal combustion engine **12** having exhaust manifolds **14**. For the V-type engine illustrated, exhaust gases are discharged from the exhaust manifolds **14** to an exhaust line **16** including a crossover pipe **18**. The exhaust line **16** includes a wave interference silencing system **19** comprising a plurality of serially connected attenuating sections **20***a*, **20***b*, **20***c*, **20***d*, and **20***e* which are axially separated by acoustical equalization conduits **22***a*, **22***b*, **22***c*, and **22***d*. A tailpipe extension **24** is telescoped over the rearward end of the attenuating section **20***e* and constitutes the exhaust end of the system. Those skilled in the art will appreciate that the above components from the exhaust manifolds **14** to the tailpipe extension **24** form an acoustical line through which heated exhaust gases and their resultant pressure waves flow as products of combustion from the engine cylinders.

During operation, the engine **12** generates pressure pulses and sound waves in the exhaust line having plural frequencies which are a function of engine speed. For a given engine, certain exhaust frequencies will predominate in the noise spectrum of the exhaust gases. The present invention predictably attenuates the entire noise spectrum in the manner hereinafter described.

Referring to FIG. **2**, a representative attenuating section **20'** includes an inner conduit **30** and an outer sleeve **32** structurally connected by a helical baffle **34**. While the cross-sectional configuration of the attenuating section **20'** may take various forms and still accommodate the principles disclosed herein, the present embodiment utilizes a cylindrical tube for the inner conduit **30**, a helically wound strip for the baffle **34**, and a helically wrapped sheet for the outer sleeve **32**. The baffle **34** establishes a helical angle θ and an axial pitch P. The inner cylindrical edge of the baffle **34** is seam welded at **36** to the outer surface of the inner conduit **30**. The mating edges **40** and **42** of the outer sleeve **32** are seam welded at **44** to the outer surface **46** of the baffle **34**.

For purposes of future reference and as shown in FIG. **5**, the inner conduit **30** has an inner diameter D_a and an outer diameter D_b; the baffle **34** has a mean diameter D_c; and the outer

sleeve **32** has an inner diameter D_d and an outer diameter D_e. The inner diameter D_a forms an inner flow path **50** handling a gas flow Q_i having an actual propagational sound velocity V_i, a prevailing temperature T_i, a cross-sectional area A_i, and a length L_i. The outer flow path **52** defined between the baffle **34** and the sleeve **32** and the inner conduit **30** handles a gas flow Q_o having an actual propagational sound velocity V_o, a prevailing temperature T_o, a cross-sectional area A_o, and a length L_o.

A representative acoustical equalization conduit **22'**, as shown in FIG. **3**, is in the form of a continuous pipe having diagrammatically enlarged ends **60** which are telescopically received over the end sections **62** of adjacent attenuating sections **20'**. Each conduit **22'** has a length L_s and an inner diameter D_s and forms a flow stabilization and equalization volume **64** handling a gas flow Q_s.

The method employed for attenuating sound waves in the present invention employs a main duct or primary flow path and a reentrant duct or parallel flow path. The flow paths have an acoustical length ratio, as hereinafter described, to establish an appreciable phase angle difference between the sound waves exiting at the equalization conduit. The method provides maximum attenuation or predominant attenuation bands when the phase angle is 180° or at odd-numbered multiples thereof, i.e., 540°, 900°. Supplemental bands of attenuation are provided according to accepted theory when the sum of the respective flow path lengths, L_i and L_o, are related by the formula $L_i+L_o=KA2\pi n$ wherein n is an integer and K is a constant dependent upon frequency. With this arrangement, a pressure peak traveling through the reentrant duct will meet with a rarefaction at the exit end of the attenuating section thereby causing destructive acoustical interference. Theoretically, if the sound waves and energies are equal in both ducts, the attenuation at this frequency will be infinite. Thus, as schematically shown in FIG. **6***a*, separate sound waves will travel a reentrant path A and a main path B between an entrance end **100** and an exit end **102**. When the flow velocities and energies are compensated and equalized, the sound waves will exhaust at the succeeding equalization volume so as to completely destructively interfere. In other words, the pressure waves at the exit end **102** will be in phase opposition and have equal and opposite peak amplitudes P_A and P_B, thereby causing a complete equalization of pressure to produce a resultant ambient pressure amplitude. For the conventional attenuation formula

$$A=20 \log \frac{P_1}{P_A-P_B}$$

wherein

A is the attenuation in decibels;

P_A-P_B is the resulting acoustical pressure at the exit end; and

P_1 is the acoustical pressure at the entrance end; the idealized conditions will produce infinite attenuation. However, the actual attenuating capabilities of any device are dependent on many variables such as system impedance, multiple internal reflections, and manufacturing inaccuracies. Therefore, the reasonable attenuation is generally established at a maximum value which for the purposes of subsequent description will be 20 db. or a pressure ratio of 10 to 1.

In the present invention, the acoustical lengths of the separate flow paths **52** and **50** are selected to have an actual acoustical length ratio J which will establish the aforementioned phase angle relationship at the prevailing environmental conditions.

The ratio of flow path lengths to attenuate a given frequency has been found to require compensation for prevailing operating conditions such as temperature and gas velocity. Also, the inner and outer flow paths must be suitably related to actual operating conditions in order to balance the energy levels therewithin and maximize the wave interference attenuating capabilities of each section. Additionally, proper account must be taken of other factors such as flow resistance or pressure drop within the conduits and their effect on the length ratios between the inner and outer flow paths as well as the acoustical energy flowing within the separate paths.

Referring to FIG. **6***a* where the system is adequately temperature and flow compensated, the gas streams will be divided into two flow paths which have a relationship as follows:

$$J_a \cdot \frac{L_i}{V_i}=\frac{L_o}{V_o}=t_o=t_i \cdot J_a$$

In other words, the pressure wave B flowing down the inner flow path will have an actual sound propagational velocity V_i traveling through the length L_i and a pressure peak P_i exiting at the exit end **102** after a transit time t_i. In the outer flow path, the pressure wave A will have an actual sound propagational velocity V_o traveling through the outer length L_o with a pressure P_0 exiting the exit end **102** in transit time t_o. Where the flow conditions satisfy the above-noted criteria, the pressure wave of the outer flow path will exit at the end **102** at a pressure amplitude P_o while the sound wave of the inner flow path will exit at a pressure amplitude P_i. The pressure waves, in this case, will be of equal magnitude and in phase opposition such that complete destructive acoustical interference will occur. The compensated system, under these conditions, will theoretically achieve infinite attenuation.

Where the inner and outer flow paths are not flow compensated, a time lag Δt will exist between the exiting of the inner sound wave B' and the outer sound wave A'. In such instances, the peak amplitudes of the sound waves will experience a phase shift $\Delta\theta$ such that the terminal pressure amplitude P_o' of the outer sound wave A' will no longer be in exact phase opposition to the pressure amplitude P_i' of the inner sound wave B'. In other words, only partial cancellation of the pressures occur and the resultant pressure at the exit of this attenuating section will be some finite value which, accordingly, reduces the attenuation value.

In instances where an inadequate mixing volume precedes an attenuating section, the sound waves, A'' and B'', as shown in FIG. **6***c*, traveling through the respective flow paths will have differing initial pressure amplitudes. Under these circumstances, the peak pressure amplitude P_o'' from the outer flow path, in addition to having a lesser maximum amplitude than the inner flow path, will also have an exit pressure P_o'' below its maximum amplitude. Additionally, inlet flow conditions may cause the inner sound wave B'' to lag behind the outer sound wave A'' by a time increment Δt or by a phase shift $\Delta\theta$ according to the formula

$$J_a\frac{L_i}{V_i}+\Delta t=\frac{L_o}{V_o}$$

These aforementioned conditions also result in a decrease of attentuating efficiency and represent a significant reduction in performance as compared to the compensated system represented in FIG. **6***a*.

The design of a given exhaust line will be preliminarily controlled by certain vehicle and engine characteristics such as engine size, vehicle length, and vehicle clearance. More particularly, the engine size and speed will determine the maximum gas flow which the exhaust system must handle. The vehicle length, to a large extent, will determine the maximum length of the exhaust system subject to bending and other rerouting procedures. The available space beneath the vehicle body will determine the maximum size of the conduits employed in the exhaust system. Inasmuch as many factors influence the varying spectrum of noises produced by an engine, certain initial conditions must be tentatively prescribed on the basis of prior experience in the art such that remaining variables can thereafter be systematically ascertained to produce acceptable attenuation levels in the system.

Accordingly, the exhaust line length should be roughly established at a value which is substantially equal to the length of the vehicle. Such a figure of course, includes sufficient length to incorporate the required attenuating sections **20** and

equalization conduits **22**. Secondly, the maximum outer diameter of the sleeve **32** is limited to a value which can be fitted to the available clearance with the vehicle body. Thirdly, the cross-sectional flow area of the sleeve should adequately handle the maximum gas flow from the engine **12**. Fourthly, the inner conduit **30** is selected at a commercially available pipe size to thereby further enhance the economics of the present system.

For the system thus far described, the aforementioned actual sound propagational velocities, V_i and V_o, will exist in the inner flow path **50** and the outer flow path **52**, respectively. These actual sound propagational velocities are comprised of sound velocity at the prevailing temperature C_x and the gas flow velocity V_{fx} in the flow path. More particularly, the functional relationship of the velocity of sound C_x as related to temperature can be expressed as follows:

$$C_x=1052\left(\frac{T_x+459}{459}\right)^{.5}$$

wherein

T_x is the average temperature in the flow path **50**.

At the prevailing temperatures in the exhaust line of around 1,100°F. **and a temperature difference of around** 20° F. between the flow paths, the propagational velocity will vary approximately 12 f.p.s. Unless such change is properly recognized in the system design, the decrease in attenuation occurs as outlined with respect to FIGS. **6***b* and **6***c*.

The gas flow velocity V_{fx} in the respective flow paths will be a function of the mass gas flow, the size and characteristics of the flow path including the cross-sectional area and the prevailing operating temperature. For the inner flow path this relationship can generally be empirically set forth as follows:

$$Q_f=5.24\ H^{0.45}\ (D_a)^{2.68}$$

wherein

H is the pressure drop per foot of section length,

in. hg./ft.

For a quantity Q_i flowing through the inner path, a gas flow velocity V_{fi} will exist as related by the following formula:

$$V_{fi}=.060\frac{Q_i(T_i+459)}{D_a^2}$$

The outer path **52** will have a flow area which is the equivalent of a single pipe having an inner diameter S_3 represented as follows:

$$S_3=1.34\frac{(WY)^{.625}}{(W+Y)^{.25}}$$

wherein

W is the width of the mean diameter helix, in.; and

Y is the height of the helix or $D_d-D_b/2$, in.

With the equivalent diameter thus calculated, the mass flow through the outer flow path Q_o can be represented as follows:

$$Q_o=5.24\ S_3^{2.68}\ (H/J)^{0.45}$$

The above equations thus relate three independent variables: J, Q_o, and Q_i. The initial values are based on skilled estimates as to their ultimate value. By iterative computation of the type conventionally performed on digital computers, the final values can be calculated within prescribed system limits as related to total gas flow and the acoustical length ratio.

On the basis of the above calculations, the gas flow velocity in the outer flow path V_o can be established as follows:

$$V_o=.048\frac{Q_o(T_o+459)}{(WY)}$$

Using a preliminary value for an initial helix angle, θ_i, necessary to produce the uncompensated length ratio J_1, the helical pitch P for the baffle will be prescribed according to the formula:

$$P=\frac{\pi(D_b+D_d)}{2\tan(\theta_1)+T_2\cos(\theta_1)}$$

wherein

T_2 is the thickness of the baffle, in.

This calculated pitch P will then establish an outer wrap helix angle θ_1 as follows:

$$\theta_1=\text{Atn.}\ \frac{(P-T_2/\text{Cos}\ (\theta))}{\pi D_d}$$

The acoustical characteristics of the sound waves in the individual sound paths are a function of the actual gas flow velocity and the temperature compensated gas flow. Thus, the length ratio J between the flow paths must be approximately revised. Generally, the aforementioned differences can be accounted for as follows:

$$J_a=J_1\frac{(V_{fo}+C_o)}{(V_{fi}+C_i)}$$

This revised length ratio will also provide secondary representations for the helix angle θ and the pitch P as follows:

$$\theta=\text{Atn.}\ \frac{1}{(J^2-1)^{.5}}$$

and

$$P=3.14\frac{(D_b+D_d)}{2\tan(\theta_1)}$$

Additionally,

$$\theta=\text{Atn.}\ \frac{(P-T_2/\cos(\theta_1))}{\pi\frac{(D_b+D_d)}{2}}$$

Thus, with two representations for the helix angle θ and an intermediate representation for the pitch P, by iterative computation, the design helix angle can be calculated within prescribed limits with the helical pitch P being appropriately revised in accordance therewith. As a result of the aforementioned computations, a basic attenuation section will be established which is standardized in regard to the structural characteristics of the flow paths. Moreover, the basic section produces an acoustical wave interference due to phase opposition at the end thereof which is accurately compensated for prevailing flow and temperature conditions.

On the basis of a standard attenuating section, the total system for silencing objectionable sound noises can be established for a given motor vehicle engine. The permissible length for the exhaust line of a vehicle with such an engine will be roughly established by the length between the exhaust manifolds **14** and the end of the vehicle body together with mechanical bends necessary to route the exhaust line around the vehicle chassis. Moreover, each engine will normally produce objectionable noises at predominate frequencies. The most objectionable of these frequencies forms the basis for establishing the length of a fundamental attenuating section which will influence selected lengths of the remaining sections.

In the preferred embodiment, the longest section is selected as the fundamental attenuating section and has an acoustical length which attenuates a fundamental frequency at the low end of the noise spectrum and, of course, multiple frequencies thereabove. The length L_x of this section is determined by the following formula:

$$L_x=\frac{V_o}{2\cdot F_x\cdot(J_a-1)}$$

wherein

V_o is the propagational sound velocity of gas in the outer flow path;

J_a is the actual acoustical length ratio of the outer flow path to the inner flow path; and

F_x is the frequency to be attenuated.

By way of example, the preferred embodiment in the present invention uses an internal combustion engine having a full throttle actual sound velocity of 2,000 f.p.s., a length ratio of 2.02, and a base attenuating frequency of 250 Hz. Accordingly, the length of the base attenuating section is 3.96 feet.

The attenuation provided by such an attenuating section is dependent on the mass gas flow in the inner and outer flow paths as related to total gas flow and the permissible pressure drop for the exhaust line. Thus, given the specified maximum gas flow Q_T and permissible pressure drop H_T for the entire system, the aforementioned relationships for Q_i, V_i, Q_o, and V_o are iteratively computed to obtain the values for the unknown quantities.

The attenuation for a given section is a function of the input energy to the section in comparison with exit energy and, accordingly, requires due consideration for the prevailing conditions in their respective flow paths. The flow energy in the inner flow path E_i is a function of the flow path area and sound and gas flow velocity as related in the following manner:

$$E_i=K[(V_{fi}+CAi)\cdot 0.78\cdot(D_b)^2]$$

Similarly, the flow energy in the outer flow path E_o is

$$E_o=K[(V_{fo}+CAo)\cdot(W\cdot Y)]$$

As a function of frequency F_x, the attenuating section length L and the acoustical length ratio J_a, the angular difference of path lengths relative to the frequency at velocity of propagation G can be expressed as follows:

$$G=2\pi F(t_o-ti)$$

The output energy E_T at the end of a given attenuating section is a function of the G factor, the attenuation factor A, and the attenuating section length L_s. This relationship can be expressed as follows:

$$E_T=E_i+_o[A^{(L_s J_a)}\cos(G)]$$

While the attenuation as represented by a comparison of the input flow energy and the output flow energy is ascertainable for all frequencies, I have found it practical to arbitrarily limit the expectable attenuation A to specified maximum value which in the present instance is 20 db. However, the attenuation at a given frequency R can be calculated as follows:

$$R=20\log_{10}\frac{E_o+E_i}{E_T}$$

Referring to FIGS. **7***a* through **7***e*, the attenuation versus frequency responses for the individual attenuating sections **20***a* through **20***e* are illustrated for the predominant attenuation bands. In each of these sections, the arbitrary maximum attenuation cutoff has been set at 20 db. As previously mentioned, the attenuating section **20***e* has an output curve **120***e* selected to attenuate a certain low-level noise frequency of around 270 Hz. On either side of the design fundamental frequency, the attenuation decreases in the manner predicated in the formula outlined above. However, for longer length sections, secondary attenuation bands will be provided at higher frequencies corresponding to the odd integer multiples of the fundamental frequency. Thus, the base section **20***a* will have a secondary attenuating frequency response **120***e'* at about 810 Hz., a third response **120***e''* at about 1,340 Hz., and a fourth response **120***e'''* at about 1,890 Hz. Each response has a maximum amplitude bandwidth of around 100 Hz.

With the attenuating capabilities of the base section thus established, the noise spectrum of the engine is reanalyzed and the next objectionable frequency dealt with by selection of appropriate attenuating lengths. For example, the attenuating section **20***b* is selected to attenuate a low frequency noise **120***d* of around 420 c.p.s. Such an attenuating section will additionally contribute to overall attenuation at the supplemental attenuating bands ranging around 1,260 Hz., **120***d'*, and 2,100 Hz., **120***d''*, with each response having a bandwidth of about 160 Hz.

The attenuating response for the exhaust system is then once again reexamined and an intermediate attenuating length selected to provide a noise suppression in the middle frequency range. The third attenuating section **20***c* has a fundamental frequency, **120***c*, of about 625 Hz., a secondary response, **120***c'*, of 1,875 Hz., and a bandwidth of 310 Hz., The fourth section **20***d* is used to provide broad-band attenuation in the middle frequency range and has a first attenuating response, **120***b*, of about 990 Hz., and a bandwidth of about 550 Hz. The fifth attenuating section **20***e* is used for extremely broad-band, high frequency attenuation and has a fundamental attenuating frequency **120***a* of about 1,840 Hz.

The resulting attenuation of the exhaust line is shown in FIG. **7***f* and represents the algebraic summation of the attenuation provided by the individual sections. In the event that the attenuation provided at a particular frequency is greater than necessary or insufficient to meet the desired sound reduction, the individual section lengths can be appropriately revised to provide greater or lesser attenuation response for these portions of the spectrum. Moreover, additional sections can be incorporated within the overall length restriction for the exhaust line. Generally, however, the frequency spectrum, to the extent possible, should attenuate with at least the arbitrary maximum factor A after the low frequency band. Accordingly, in the preferred embodiment, the section lengths are balanced such that the peak amplitude cutoffs for the given attenuating peaks are generally overlapping.

Regardless the sizing of the mixing sections, as previously noted, the mixing volume **64** of the equalization conduits **22** should provide for total mixing of the sound waves which have separately proceeded down the inner and outer flow paths. Moreover, they should provide for pressure and gas flow stabilization in order to minimize differences in vibrational pressures and velocities at the entrance of a subsequent section. While the diameter for the mixing section may be roughly the same as the inner diameter of the outer flow path, variances therein might be desirable to enable routing of the exhaust line around the chassis. Therefore, I have found that minimum permissible flow path is of greatest importance and is generally represented as the equivalent single-pipe diameter for handling combined gas flow of the inner and outer flow paths. More specifically, the inner diameter D_s can be empirically represented as follows:

$$S_5=\left[\frac{Q_T}{(3.4H^{.45})}\right]^{\frac{1}{2.68}}$$

Additionally, experience has shown that the length L_m of the mixing section should be about twice the diameter of S_5 to achieve flow and pressure stabilization mentioned above.

An exhaust line having attenuating capabilities reported in FIGS. **6** through **11** has been successfully built and operated in accordance with the following dimensions:

Section length, feet:	
20a	4
20b	2.62
20c	1.76
20d	1.06
20e	.06
Average temperature degree from outer path, F_o, ° F	1,100
Average temperature degree from inner path, F_i, ° F	1,120
Pressure drop per foot, H, in. hg./ft	1.68
Actual acoustical length ratio, J_a, feet	2.02
Helix angle Θ, radians	.511
Total gas flow:	
Q_t, lb./min	20.0
D_a, inches	1.43
D_b, inches	1.50
D_c, inches	1.93
D_d, inches	2.36
D_e, inches	2.50

Although only one form of this invention has been shown and described, other forms will be readily apparent to those skilled in the art. Therefore, it is not intended to limit the scope of this invention by the embodiment selected for the purpose of this disclosure, but only by the claims which follow.

What I claim is:

1. A silencing system for attenuating exhaust gas sound waves from an internal combustion engine, comprising: a plu-

rality of serially connected attenuating members having low acoustical impedance volumes fluidly coupled thereto; and a pair of discrete flow paths in each of said attenuating members that are individually temperature and flow compensated for the sound waves transiting therewithin so as to establish actual sound propagational velocities for the latter that will cause destructive interference in said volume at the exit end of the transited attenuating member thereby suppressing objectionable noises at predetermined frequencies, each attenuating member further including means for compensating the mass flow through the separate flow paths to equalize vibrational energy arriving at said volumes and thereby minimize vibrational pressures at the entrance end of a succeeding attenuating member.

2. A wave interference exhaust system for attenuating sound waves in the frequency spectrum produced in the exhaust gases of an internal combustion engine, comprising: an exhaust line including a plurality of attenuating sections acoustically coupled in series; conduit means of a low acoustical impedance fluidly succeeding each attenuating section; and a pair of alternate gas flow paths in each attenuating section which handle substantially equal vibrational energy and mass gas flow, and produce destructive interference in the succeeding conduit means, the ratio of the flow path lengths for the individual sections being effective to attenuate a predetermined frequency band in said spectrum, and with the remaining sections, cumulatively reduce objectionable vibrational energy to an acceptable level.

3. A wave interference silencing system for predictably attenuating sound waves at select frequency bands in the noise spectrum of the exhaust gases from an internal combustion engine, comprising: a plurality of attenuating sections acoustically coupled in series; acoustical equalizing conduits fluidly succeeding each attenuating sections; a pair of discrete flow paths in each attenuating section, said flow paths having differing acoustical lengths and being temperature and flow compensated to establish actual sound propagational velocities so that pressure waves transiting therewithin are in appreciable phase opposition at the exit of the transited paths so as to destructively interfere within said equalizing conduits, said sections being standardized in sectional design and being of individual lengths which will provide predictable attenuation responses at said select frequency bands whereby subsequent consideration of a single parameter permits a system design of predictable attenuating response.

4. A method of predictably attenuating sound waves in the exhaust gases of an internal combustion engine, comprising the steps of:

1. separating the exhaust gases into a pair of flow paths;
2. sizing the flow paths to handle equal vibrational energies;
3. dividing the flow paths into discrete sections;
4. establishing a length ratio between the flow paths in the individual sections which is flow and temperature compensated to place the sound waves in phase opposition;
5. selecting a discriminate length for the compensated sections to produce predictable attenuation responses at a select frequency band of the noise spectrum of the exhaust gases; and
6. combining the flow paths at low impedance volumes succeeding each section to cause the pressure waves transiting the individual flow paths to destructively interfere and cumulatively produce an attenuated ambient acoustical pressure for the frequency noise spectrum of the engine.