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(72) Inventor GLENN ALLAN ANDERKAY

(19)

(54) ASSEMBLY FOR VIBRATION ISOLATION

(71) We, DEERE & COMPANY, a corporation organised and existing under the laws of the State of Delaware, United States of America, of Moline, Illinois 61265, United States of America, do hereby declare the invention, for which we pray that a patent may be granted us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The present invention relates generally to an assembly for vibration isolation of a member from a base and particularly to such an assembly for isolating a cover from an internal combustion engine.

Generally, conventional vibration isolation systems rely on trial and error methods of isolation which involve adding different materials between a cover and its associated vibratable base. It has been proposed to reduce bolt transmitted vibrations by using the same resilient material in washers between the bolt heads and the cover as in the gaskets.

According to the present invention there is provided an assembly for vibration isolation of a member from a base comprising the base, the member mounted on the base, a plurality of securing means, each having a head, securing the member to the base, a plurality of resiliently compressible vibration isolating elements each compressed between a head and the member, and a resilient vibration isolating gasket compressed between the member and the base, the static spring rate of the plurality of resiliently compressible elements being within the range one half to twice that of the gasket.

A resilient vibration isolating means can be compressed between the securing means and the member, the means being such as to maintain a natural frequency of the member perpendicular to the axis of the securing means which is lower than the natural frequency of the member parallel to the axis of the securing means.

Embodiments of the invention will now be described with reference to the accompanying diagrammatic drawings in which:

Figure 1 is a cross-sectional view of a preferred embodiment of a portion of a vibration isolation assembly of the present invention;

Figure 2 is an isometric view of a grommet of the assembly of Figure 1;

Figure 3 is a top view of a gasket of the assembly of Figure 1;

Figure 4 is a transmissibility plot;

Figure 5 is a cross-sectional view of an alternative embodiment of a portion of an assembly of the present invention;

Figure 6 is a top view of a gasket of the assembly of Figure 5; and

Figure 7 is a cross-sectional view of a further alternative embodiment of a portion of an assembly of the invention.

Referring now to Figure 1, the assembly is generally designated by the numeral 10. Also shown in Figure 1 is a vibratable base 12, which can be the engine of a conventional vehicle power plant (not shown), and a cover 14 spaced therefrom which can be an oil pan, rocker arm cover, or intake air cover.

The base 12 has a passageway 16 provided therein which is encircled by a plurality of threaded holes 18 (only one shown) and the cover 14 is provided with a plurality of clearance holes 20 (only one shown) matching the pattern of threaded holes 18. The cover 14 is secured to the base 12 against a mounting surface 21 by stripper bolts 22 (only one shown) passing through the clearance holes 20 and threaded into the threaded holes 18.

A typical stripper bolt 22 has a head 24 at one end and a shank portion 26 of a predetermined length terminating in a thread 28 at the other end. The shank portion 26 has an annular groove 30 provided medially therein for receipt of a conventional O-ring 32 acting as resilient vibration isolating means. A washer 34 is disposed around the shank portion 26 in abutting relationship with the head 24 to provide a uniform loading surface for a resilient isolating grommet 36 which may be of an elastomeric material such as rubber. The grommet 36 is of generally cylindrical toroidal configuration as may be seen in Figure 2 and has a concentric hole 38 provided therein having a slight outward taper 40 to match the clearance hole 20.

The stripper bolt 22, with the washer 34 and grommet 36 thereon, is inserted through the clearance hole 20 of the cover 14 and then through a resilient isolating and sealing gasket 42 which may be of an elastomeric material such as nitrile, or silicone, rubber. The gasket 42, as may be seen by reference to Figures 1 and 3, is of generally rectangular cross-section in the direction perpendicular to the mounting surface 21 and is shaped so as completely to encircle the passageway 16. As may be seen in Figure 3, the gasket 42 has a plurality of clearance holes 43 therein, spaced so as to match the pattern of threaded holes 18 in the base 12.

When assembled, the predetermined length of the shank portion 26 of the stripper bolt 22 is slightly less than the uncompressed length of the grommet 36 and the gasket 42 plus the thickness of the washer 34 and the cover 14 so as to cause compression of the grommet 36 and the gasket 42, the gasket being compressed by 10% or more of its uncompressed thickness. Furthermore, the O-ring 32 is centrally positioned in the clearance hole 20 of the cover 14 so as to prevent metal to metal contact between the stripper bolt 22 and the cover 14.

To determine the optimum geometrical and physical characteristics of the grommets 36 and the gasket 42, the excitation and/or response frequency spectrum is measured for the base 12 and/or the cover 14. A preferred natural frequency for the cover 14 is selected which will fall between major resonance frequencies while being as low as possible in the frequency spectrum. Where the base 12 is a vehicle engine, it has been determined that the preferred natural frequency will fall within a range extending from 50 to 350 hertz and thus that the ideal equivalent static system spring rate will be in the range extending from approximately 32,800 seconds^{-2} to 1,613,000 seconds^{-2} times the mass of the cover 14 as derived from the equation:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{aK}{m}}$$

where: f_n is the natural frequency

a is the dynamic system spring rate factor (1 for steel springs) (3 for elastomeric materials)

K is the equivalent static system spring rate which is equal to the sum of the spring rates of the grommets 36 and the gasket 42

m is the mass of the cover 14.

In the preferred embodiment, the preferred natural frequency has been determined to be approximatly 100 hertz and thus the equivalent static system spring rate is 132,000 seconds^{-2} times the mass of the cover 14 from the above equation.

In the ideal situation, to obtain maximum length vibration preference paths, the static spring rates of the grommets 36 should equal that of the gasket 42; i.e. there should be equal spring rates on both sides of the cover 14. Accordingly, the ideal static spring rate would be half the equivalent static system spring rate or in the range of 16,450 seconds^{-2} to 806,500 seconds^{-2} times the mass of the cover 14 and preferably approximately 66,000 seconds^{-2} times the mass of the cover 14.

Starting with the rectangular cross-section gasket 42, and the ideal static spring rate, the effective dimensions and the compressive modulus of elasticity of the gasket 42 are derived from the equation:

$$k = \frac{PWE}{T}$$

where: k is the spring rate of the gasket 42

P is the effective perimeter

W is the width

E is the compressive modulus of elasticity

T is the thickness.

In the preferred embodiment, P is constrained by the bolt pattern, W by the flange size on the cover 14, E by the desirability of having the softest material possible so as to have the lowest frequency possible, and T by the durometers available and thicknesses desirable. With particular regard to T, a 10% compression of the gasket 42, regardless of material hardness, will provide a seal which has been experimentally determined to prevent the leakage of a fluid such as oil.

Based on the design criteria for the gasket 42, a new gasket spring rate is calculated which is then used to derive the geometrical and physical characteristics of the cylindrical toroidal grommet 36 according to the equation:

$$k' = \frac{(D^2-d^2)\ \pi\ E'N}{4t}$$

where: k′ is the spring rate of the grommet 36

D is the major mean outside diameter

d is the minor mean inside diameter

E′ is the compressive modulus of elasticity

N is the number of grommets

t is the thickness.

The spring rate for the grommet 36 is determined with D constrained by the washer diameter, d by the bolt 22 diameter, E′ by the desirability of having the softest material possible, and t by the desirability of having equal compression. As would be evident to those skilled in the art, an iterative process yields the closest spring rates based on geometry and durometer.

In the case of the grommets and the gasket having substantially similar vertical cross sections laterally, the compressive modulus of the grommets is substantially different from that of the gasket. If the cross sections are substantially dissimilar the compressive moduli are substantially the same.

In the preferred embodiment, it has been determined that a "40" durometer gasket 42 and "70" durometer grommets 36 will provide the most nearly equal spring rates with the softest practical durometers.

While the above is the preferred embodiment, reference should be had to Figure 4 which shows a transmissibility versus frequency ratio plot for a given material and which defines the permissable deviations from the preferred embodiment. As known to those skilled in the art, transmissibility (TR) is defined as the ratio of transmitted force to impressed force through a material, frequency ratio is defined as the ratio of impressed frequency to natural frequency, and the plot is for a material having a specific damping factor ratio which in the preferred embodiment is approximately 0.1. Upon examination of Figure 4, it will be noted that an inflection of the curve above a frequency ratio of 1 occurs at approximately 1.2. This indicates that a variation in the designed natural frequency from the actual natural frequency of 20% (impressed frequency = 1.2 natural frequency) will result in a system having a transmissibility close to the preferred system. When utilizing the first mentioned equation and solving for spring rate, it will be found that the 20% variation in frequency will allow the spring rate of either the gasket 42 or the grommets 36 to be 1.44 or approximately one and one-half times the other without substantially affecting the isolation capabilities of the system. The static spring rate of the grommets is thus preferably within the range two thirds to one and a half that of the gasket.

Upon further examination of Figure 4, it will be noted that the transmissibility becomes less than 1 at approximately 1.4. This indicates a variation in designed and actual natural frequency beyond which the desired isolation capabilities of the system will be lost. When utilizing the first mentioned equaton and solving for spring rate, bearing in mind that the static system spring rate is equal to the sum of the spring rates of the grommets and gasket and also that the ideal situation is of the latter spring rates being equal, it will be found that a 40% variation in frequency will allow the spring rate of either the gasket 42 or the grommets 36 to be approximately twice the other before the desired isolation effect will be lost.

The effect of the shear spring rate in directions perpendicular to the compressive spring rate must be checked to ensure that the natural frequency parallel to the mounting surface 21 is not greater than the natural frequency perpendicular thereto. If it is greater, the above calculations must be made to arrive at a design suitable for vibrations parallel to the mounting surface and a check made of the natural frequency perpendicular to the mounting surface 21 to ensure that it has not been increased by the design change.

In Figure 5 there is shown an alternative embodiment having a gasket 44 disposed between the cover 14 and the member 12, with the same reference numerals as used in the previously described embodiment to designate the same parts. As may be seen in Figures 5 and 6, the gasket 44 has a generally H-shaped configuration.

In certain instances where it is desirable to utilize the same durometer material for the gasket 44 and the grommets 36, it is possible to obtain substantially equal spring rates by changing the cross-sectional configuration of the gasket or grommets. The same equations as given above will apply except that k and k' will be determined by an equation dependent upon the gasket's or grommets' cross-section perpendicular to the surface of the member 12. The gasket 44 will provide the substantially same spring rate as the grommets 36 where both are "70" durometer.

Referring now to Figure 7, there is shown a further alternative embodiment having a resiliently compressible element in the form of a metal spring 46 disposed between the cover 14 and the washer 34, with the same reference numerals as used in the previously described embodiments to designate the same parts. The metal spring 46 would function similarly to the grommet 36 to provide substantially equal spring rates.

WHAT WE CLAIM IS:—

1. An assembly for vibration isolation of a member from a base comprising the base, the member mounted on the base, a plurality of securing means, each having a head, securing the member to the base, a plurality of resiliently compressible vibration isolating elements each compressed between a head and the member and a resilient vibration isolating gasket compressed between the member and the base, the static spring rate of the plurality of resiliently compressible elements being within the range one half to twice that of the gasket.

2. An assembly according to claim 1 in which the gasket is compressed by 10% or more of its uncompressed thickness.

3. An assembly according to claim 1 or 2 in which the resiliently compressible elements and gasket have substantially similar vertical cross sections laterally, and the compressive

modulus of elasticity of the said elements is substantially different from that of the gasket.

4. An assembly according to claim 1 or 2 in which the resiliently compressible elements and gasket have substantially dissimilar vertical cross sections laterally, and the compressive modulus of elasticity of the said elements is substantially the same as that of the gasket.

5. An assembly according to claim 4 in which the gasket in vertical cross section taken laterally is H-shaped.

6. An assembly according to any preceding claim in which the said range is two thirds to one and a half.

7. An assembly according to any preceding claim in which the spring rates of the said elements and the gasket are substantially equal.

8. An assembly according to any preceding claim in which resilient vibration isolating means are compressed between the securing means and the member and are such as to maintain a natural frequency of the member perpendicular to the axis of the securing means which is lower than the natural frequency of the member parallel to the axis of the securing means.

9. An assembly according to any preceding claim in which the said elements or gasket have a spring rate within the range 16,450 seconds^{-2} to 806,500 seconds^{-2} times the mass of the member.

10. An assembly according to claim 9 in which the spring rate of the said elements or gasket is substantially 66,000 seconds^{-2} times the mass of the member.

11. An assembly according to any preceding claim in which the base is an internal combustion engine, the member is a cover member, and the securing means comprise a plurality of bolts.

12. An assembly according to claim 1 in which the said elements are in the form of helical springs.

13. An assembly for vibration isolation substantially as described herein with reference to, and as illustrated in, Figures 1 to 3, or Figure 5, or Figure 7 of the accompanying diagrammatic drawings.

GEOFFREY S. COLLINS,
Chartered Patent Agent,
c/o Reddie & Grose,
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London, WC1X 8PL.

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1582147 COMPLETE SPECIFICATION

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Sheet 1

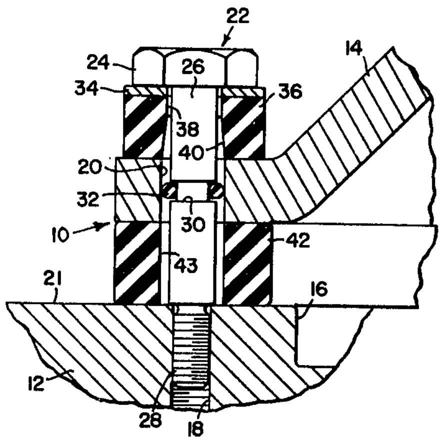


FIG. 1

FIG. 2

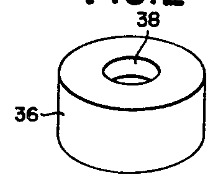
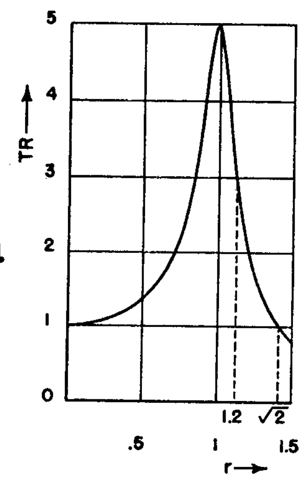


FIG. 4



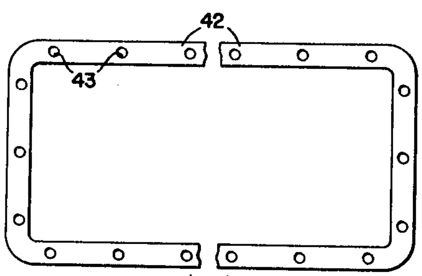


FIG. 3

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2 SHEETS *This drawing is a reproduction of the Original on a reduced scale*

Sheet 2

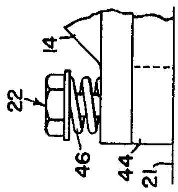

FIG. 7

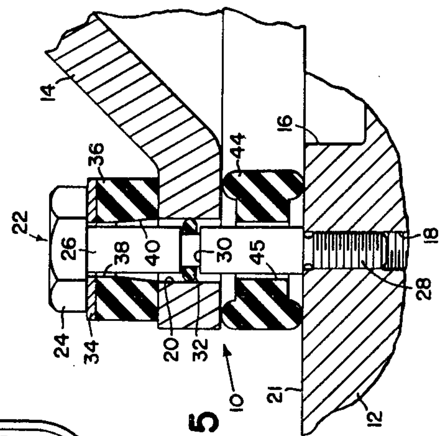

FIG. 5

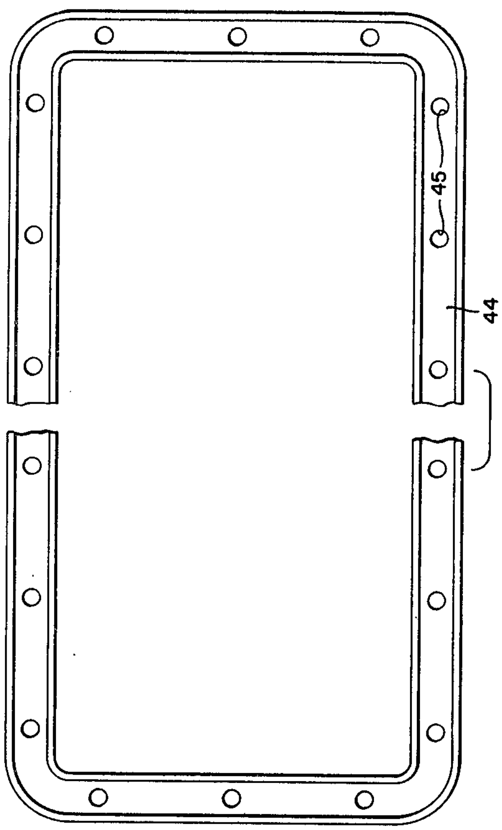

FIG. 6