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ANALYSIS OF THE IMPACT OF DIFFERENT TYPES OF VIBRATION ISOLATION ON THE DYNAMIC LOADING OF MACHINES AND THE SURROUNDING ENVIRONMENT

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ABSTRACT

The resulting dynamic loading on machines, the environment and humans generated by vibration and noise is dependent on the vibro-insulating components and the quality of resilient materials used in the mounting of these components. Well-designed vibration isolation of vibrating sources can effectively reduce the transmission of vibro-acoustic energy into supporting and surrounding structures. Based on frequency spectrum, the vibro-isolation efficiency of various vibro-insulating components and their resilient materials is analysed. The solution of this problem is based on theoretical knowledge and methodology of the transmission of vibration-sound waves and measurement of the machines involved. Measurements of vibration at the sources and along the path of transmission, as well as sound measurements, were performed for different vibro-isolators to compare real results with theory. Measured components include; isolation of a recirculation fan in a heating plant, air-conditioning unit, and combustion engine of a passenger vehicle. For the detection of the vibro-acoustic energy the vibration and sound were measured and FFT analysis was applied. Finally, this paper suggests measures which can be taken to reduce undesirable vibro-acoustic energy on machines, the environment and bystanders.

INTRODUCTION

Mechanical vibration and damping by vibration isolators and/or resilient material constitute the two most widely applicable means of vibration control in structure-borne sound, particularly in the low and audio frequency range. Vibration isolation, in essence, involves use of a purpose connection between the source of vibration and the surroundings to be protected so that the surroundings vibrate less than when a rigid connection is used. In some typical situations the source consists of a vibrating mechanical system and the item to be protected is a receiving structure such as the mounting base of a mechanical system, vehicle bodies, nearby surroundings such as other machines, technological and building structures, etc. Many salient features of vibration can be analysed in terms of a simple model consisting of a rigid body that is connected to a support via a linear spring constrained to translate along a single axis (e.g. recirculation fan, air-conditioning units and cogeneration units). More complex models are needed to address situations where the magnitude of excitation depends on the motions involved, where an additional spring-mass system is inserted between the mounting base and its surroundings, and/or at comparatively high frequencies where the isolator mass plays a significant role or where the isolated resilient materials do not behave as rigid bodies (e.g. vibration isolation of a mounting base). Other complications arise because of non-uniaxial motions and nonlinearities (e.g. combustion engine). Damping refers to the reduction of mechanical energy from a particular vibration of concern. This reduction may result in the transfer of energy to structural components, fluids, or vibration modes or from conversion of mechanical energy into other forms. Various measures of damping are currently in use. Most are based on observation of the effects that damping has on the motions of simple systems. Data on the damping inherent in materials cover a wide range, extending from comparatively low damping for high-strength structural materials to very high damping for some viscoelastic materials (typically, plastics or elastomers) with limited strength. Structural components that are acceptably strong and that also exhibit relatively high damping may be obtained by combining high-damping viscoelastic materials with resilient materials in the form of added layers or in sandwich arrangements (1, 2, 3). The damping efficiency of the vibration is significantly dependent on the dynamic transfer stiffness of isolators.

DYNAMIC TRANSFER STIFFNESS OF VIBRATION ISOLATORS

This part of the paper explains why the dynamic transfer stiffness is most appropriate to characterize the vibro-acoustic transfer properties of isolators and/or resilient material for many practical applications. The dynamic transfer stiffness is determined by the elastic, inertial and damping properties of the isolator. The reason for choosing a presentation of test results in terms of stiffness is the practical consideration that it complies with data of static and/or low-frequency dynamic stiffness which are commonly used. The additional importance of inertial forces (i.e. elastic wave effects in the isolators) makes the dynamic transfer stiffness at high frequencies more complex than at low frequencies. Because at low frequencies only elastic and damping forces are important, the low-frequency dynamic stiffness is only weakly dependent on frequency due to material properties. In principle the dynamic transfer stiffness of vibro-acoustic isolators is dependent on static preload and temperature. The vibro-acoustic energy from the base of the vibrating mechanical system (input side) to its support (output side) transferred through isolators or a vibro-isolating layer, depends on the mounting strategy, as well (4, 5, 6).

A familiar approach to the analysis of complex vibratory systems is the use of stiffness – compliance – or transmission matrix concepts. The matrix elements are basically special forms of frequency-response functions; they describe linear properties of mechanical and acoustical systems. On the basis of the knowledge of the individual subsystem properties, corresponding properties of assemblies of subsystems can be calculated. The three matrix forms mentioned above are interrelated and can be readily transformed amongst themselves (4, 7). The general conceptual framework for the proposed isolator characterization and corresponding real test equipment is shown in “Fig. 1”. The system consists of three blocks, which represent the vibration source (mechanical system as a source of vibration), isolator of vibration and the receiving structure respectively.

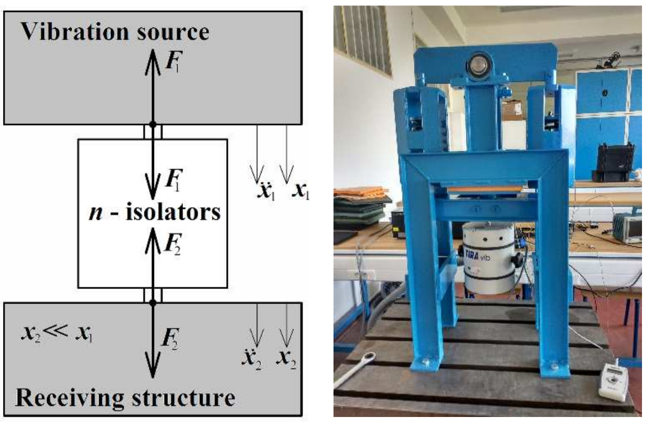

Figure 1. Block diagram of the source – isolators – receiving structure and the corresponding test equipment.

A point contact is assumed at each connection between source and isolator and between isolator and receiver. At each connection point a force vector $\boldsymbol{F}$ containing three components F_x, F_y, F_z and three moment components M_x, M_y, M_z and a displacement $\Delta \boldsymbol{r}$ containing three orthogonal translational and three orthogonal rotational components are assigned. In “Fig. 1” just one component of each of the vectors F_1, x_1, F_2 and x_2 is shown. These vectors contain $6n$ elements, where n denotes the number of isolators.

The blocked transfer stiffness is suitable for isolator characterization in many practical cases, the discussion will proceed from the simplest case of unidirectional vibration to the multidirectional case for a single isolator. For the presented case, the damping force is not necessary. For unidirectional vibration of a single vibration isolator, the isolator equilibrium may be expressed by the following stiffness equations

$$F_1 = k_{1,1}x_1 + k_{1,2}x_2 \quad \text{and} \quad F_2 = k_{2,1}x_1 + k_{2,2}x_2 \qquad (1)$$

where $k_{1,1}$ and $k_{2,2}$ are driving point stiffness’s when the isolator is blocked at the opposite side (i.e. $x_2 = 0$, $x_1 = 0$, respectively) and $k_{1,2}$ and $k_{2,1}$ are blocked transfer stiffnesses, i.e. they denote the ratio between the force on the blocked side and the displacement on the driven side. For passive isolators $k_{1,2} = k_{2,1}$, because passive linear isolators are reciprocal. The matrix form of equations (1) is

$$\boldsymbol{F} = [k]\boldsymbol{x} \qquad (2)$$

with the dynamic stiffness matrices

$$[k] = \begin{bmatrix} k_{1,1}k_{1,2} \\ k_{2,1}k_{2,2} \end{bmatrix} \qquad (3)$$

For excitation of the receiving structure via isolator

$$k_r = \frac{F_2}{x_2} \qquad (4)$$

where k_r, denotes the dynamic driving point stiffness of the receiver. From Eq. (1) and Eq. (4) it follows that

$$F_2 = \frac{k_{2,1}}{1 + k_{2,2}/k_r} x_1 \qquad (5)$$

Therefore, for a given source displacement x_1 the force F_2 depends both on the isolator driving point dynamic stiffness and on the receiver driving point dynamic stiffness.

However, if $|k_{2,2}| \le 0.1|k_r|$, then F_2 approximates the so-called blocking force to within 10 %, i.e.

$$F_2 \approx F_{2,\text{blocking}} = k_{2,1}x_1 \qquad (6)$$

Because vibration isolators are only effective between structures of relatively large dynamic stiffness on both sides of the isolator of vibration and/or resilient material, Eq. (6) represents the intended situation at the receiving end, therefore these conditions have to be respected when setting up the vibration isolator.

As the block diagram of the source/isolator/receiver system in “Fig. 1” shows, the vibro-acoustic transmission through isolators depends on the source vibration, the isolator transfer stiffnesses and the receiver driving point stiffnesses. Usually vibrational sources do not vibrate unidirectional. Therefore, the measurement of isolator stiffnesses for orthogonal directions is of practical interest.

If forces and motions for single isolator at each interface can be characterized by six orthogonal components (three translations, three rotations), the isolator may be described as a 12-port (7), The matrix form of the 12 dynamic stiffness equations is equal to Eq. (2), where now

$$\Delta\boldsymbol{r} = \begin{Bmatrix}\Delta\boldsymbol{r_1}\\ \Delta\boldsymbol{r_2}\end{Bmatrix}; \quad \boldsymbol{F} = \begin{Bmatrix}\boldsymbol{F_1}\\ \boldsymbol{F_2}\end{Bmatrix} \qquad (7)$$

are the vectors of the six displacements, six angles of rotations, six forces and six moments. The 12x12 dynamic stiffness matrix may be decomposed into four 6x6 sub matrices

$$\boldsymbol{r} = \begin{bmatrix}[k_{1,1}] & [k_{1,2}]\\ [k_{2,1}] & [k_{2,2}]\end{bmatrix} \qquad (8)$$

where $[k_{1,1}]$ and $[k_{2,2}]$ are (symmetric) matrices of the driving point stiffnesses; $[k_{1,2}]$ and $[k_{2,1}]$ are the blocked transfer stiffness matrices. Reciprocity implies that these transfer matrices equal their transpose.

Again, if the receiver has relatively large driving point dynamic stiffnesses compared to the isolator, the forces exerted on the receiver approximate the blocking forces

$$F_2 \approx F_{2,\mathrm{blocking}} = [k_{2,1}]\, x_1 \qquad (9)$$

Therefore, the blocked transfer stiffnesses are appropriate quantities to characterize vibro-acoustic transfer properties of isolators, and also in the case of multidirectional vibration transmission.

For the general case the blocked transfer stiffness matrix $[k_{2,1}]$ of a single isolator contains 36 elements. However, structural symmetry causes most elements to be zero. The most symmetrical shapes (a circular cylinder or a square block) have 10 non-zero elements, i.e. five different pairs (7).

In practical situations the number of elements relevant for characterization of the vibro-acoustic transfer is usually even smaller than the number of non-zero elements. In many cases it will be sufficient to take into account only one, two or three diagonal elements for translation vibration, i.e. for only one vibration direction (often vertical see air-conditioning unit) or for two (see recirculating fan) or three perpendicular directions (see combustion engine for further discussion).

For some special technical cases, rotational degrees of freedom also play a significant role for determination of vibro-acoustic transfer properties, e.g. in the flexible coupling in drive shafts, which are often highly dependent on the preload and/or static torque. Although it is not considered as a subject for this paper; reference is made in (7) to literature that describes how rotational elements may be handled in the same way as the translational elements.

The model shown in “Fig. 1” and of Eqs. (1) to (9) is correct under the assumption that the isolators form the only transfer path between the vibration source and the receiving structure. In practice there may be mechanical or acoustical parallel transmission paths which cause flanking transmission. For measurement method of isolator and/or vibro-isolation layer properties, the possible interference of such flanking with proper measurements has to be minimized.

Let a representative case be considered where the translation amplitudes of the source in three orthogonal directions are of the same order of magnitude. Then, a priori, at least three transfer stiffnesses, i.e. the diagonal elements for translation of the transfer stiffness matrix, are of interest for characterizing the isolator. Of course, depending on symmetry, this may be reduced to two transfer stiffnesses, i.e. one vertical and one transverse.

The question of whether it is allowed to neglect transverse vibration depends now on two factors, as are the ratio of the transverse stiffness to the vertical stiffness of the isolators and the ratio of the transverse stiffness to the vertical stiffness of the receiver.

METHODOLOGY OF MEASUREMENT AND INSTRUMENTATION

The goals of the study were to investigate and analyse the transmission of structure-borne vibro-acoustic energy flow of a deterministic excitation in an air-conditioning unit, recirculation fan and internal combustion engines mounted through different types of isolators to their surroundings, then experimentally confirm the theoretical blocking transfer stiffness necessary to implement in the vibro-isolation components. This theoretical analysis was compared with experimental results. Frequency spectrum of the measured vibration signals, before and after the vibro-isolating components, were compared and analysed. The measurement of transmission loss assumes that the vibro-isolating components behave in a linear fashion and that it has negligible mass compared to the mass loading. This condition is satisfied for the investigated frequency range of vibration. The method determines the impedance of the material when loaded by a mass providing a compressive force equivalent to that found when the vibro-isolation material is used between the mechanical system (vibration source, input) and support of receiving structure (output). Hence, the system consists of three blocks, which respectively represent the vibration source, isolators and the receiving structure, which can be a mounting base, floor area of a technological structure, body of a passenger car, etc. A fixed contact is assumed at each connection between the source and isolator and between the isolator and receiver. This is done by measuring the transfer function of the mass-loaded material at all the required frequencies (5). The method uses a vibration excitation system above which the vibro-isolation component is placed with the loading body on top of the vibro-isolation component.

The vibro-acoustic energy from the vibration source (input side) to the receiving structures (output side) transferred through the vibro-isolators and/or resilient material (vibration isolation layer) depends on their loading and implementation (mounting) (see Fig. 1). If two accelerometers measure the vibration of the source $\ddot{x}_1$, and the vibration of the receiving structure $\ddot{x}_2$ (Fig. 1), it is more suitable to determine directly the transmission loss D (in dB) of the isolators and/or vibration isolation layer, which can be calculated by the formula

$$D = 10\lg\frac{\ddot{x}_1^2}{\ddot{x}_2^2} \qquad (10)$$

or, if the material properties of the isolator and contact materials are known, we can calculate the transmission loss from the relationship

$$D = 10\lg\frac{(Z_1+Z_2)^2}{4Z_1Z_2} \qquad (11)$$

where $Z = \rho c$ represents the mechanical impedance, and ρ (kg/m^3) is density, and c (m/s) is phase speed of wave.

A real signal was generated by means of rotating and translation components of the mechanical system and the consequent response on the receiving structures was measured using the FFT analyser PULSE Bruel & Kjaer platform. This portable analyser represents a system which guarantees reliable measurement processes, analysis, and evaluation. The system consists of a piezoelectric accelerometer with a frequency range from 1 Hz to 10 kHz (amplitude ±10 %), seismic accelerometer for low-frequency vibrations from 0.1 Hz to 1.5 kHz, and display and memory module. To identify the energy dominant low-frequencies more precisely, a seismic sensor was attached to a steel cylinder (see "Fig. 2-bottom"). The fast Fourier transform (FFT) analysis was carried out using the FFT analyser PULSE. Sensors mounted on the mechanical systems were attached by means of a magnet in a given position. Measurement of the investigated objects coincided with ISO standard guidelines for accelerometers and with respect to past experiences (8, 9). The goal was to ensure that the sensor correctly reproduces the motion of the analysed components without interfering with the response (check by microphone as well). Other than the frequency range, for the type of signal, it was also very important to select the appropriate type of averaging as well and number of averages per unit time over a suitable time window (10).

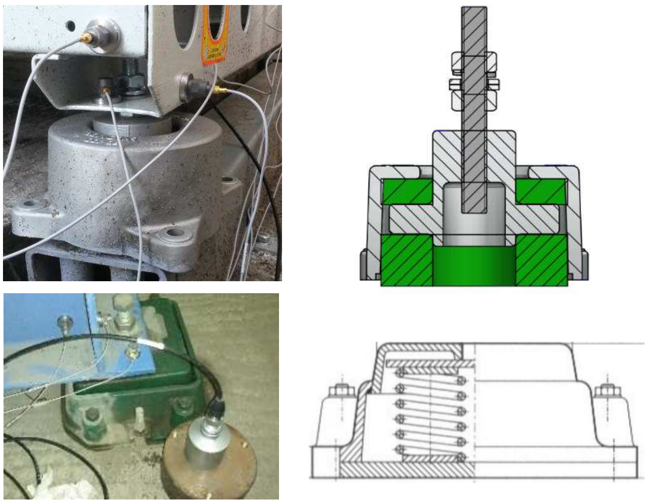

Figure 2. The attachment of the acceleration sensor on the investigated systems and their design.

The microphone was used for the acoustic measurements of the signal generated by sources at a suitable distance from the main source. The handheld B & K Type 2250-S class 1 precision sound analyser was used to do FFT transformation of the time signal into the frequency domain. The use of the acoustic signal follows the confirmation of the vibration measurement results and, in particular, the amplitude of the significant frequencies, since by passing through the discontinuities their value may amalgamate with the background, e.g. 20.8 Hz, 99.3 Hz, 197,7 Hz, etc. (see Fig. 3). The results of FFT analysis can be used to detect the causes of the excessive dynamic load and damage of components of the analysed mechanical system, thereby reducing noise and transmitting oscillations to the foundations (10, 11).

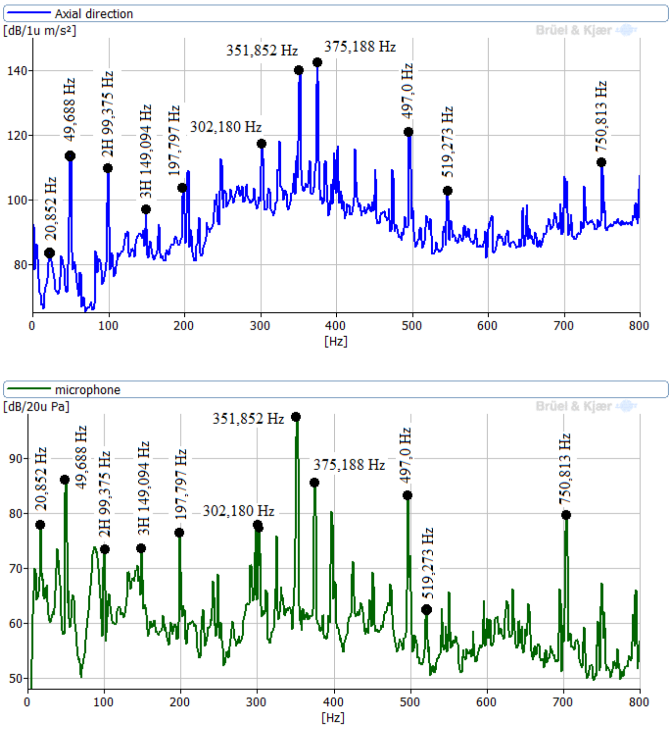

Figure 3. Frequency spectra using the accelerometer (top) and microphone (bottom).

The methodology presented in the paper can also be applied to other excitation sources of low-frequency vibration (12).

ANALYSIS AND EVALUATION RESULTS

Vibration Isolation of the Air-conditioning Unit (Fig. 2-top)**.** Appropriately designed passive vibration isolators for the dynamic load of the air-conditioning unit to protect mechanical systems, building structures, the surrounding environment, and humans are primarily used to reduce unwanted vibration and noise. Therefore, the dynamic analysis is concentrated on the measurement of the transmission of vibration energy through vibro-isolation, and mainly the transmission of strong energy of low-frequency waves through the rubber isolators on which the air-conditioning unit was mounted. The vibration was generated by a screw compressors and the fans of the air conditioning unit.

Based on the comparison of the vibration acceleration time history, it can be stated that the greatest vibration acceleration amplitude is in the vertical direction and the horizontal plane can be neglected (Fig. 4). The rotational frequency is well-observed in time history. This is also confirmed by the frequency spectra that accurately identify the rotational frequency of 49 Hz (Fig. 4). This frequency and its harmonics are necessary to assess the potential resonance of this mechanical system.

By comparing the acceleration frequency spectra measured in the vertical direction in front of the rubber isolator and behind it, it was ascertained by the amplitude that the individual frequencies are transmitted to the base. By comparing the amplitude of the frequency spectra it can be stated that the efficiency of the applied rubber isolators (Fig. 2-top) is negligible up to 200 Hz. From this analysis it follows that the used type of isolators are not suitable for the reduction of low frequency waves although the dynamic stiffness satisfy the requirements of the theory mentioned above.

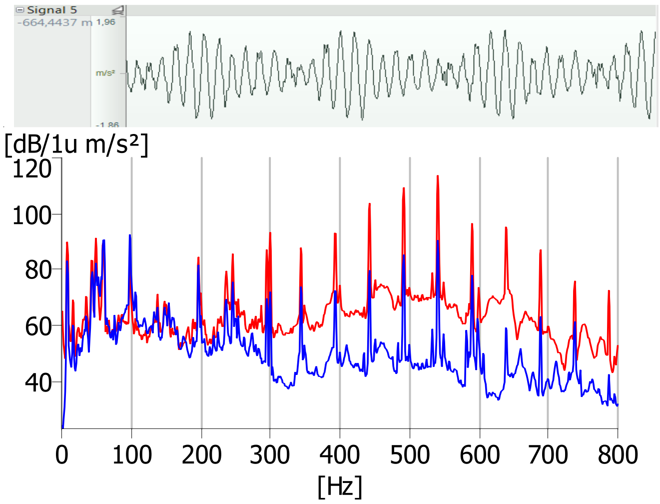

Figure 4. Time history of the dynamic response and the frequency spectrum before the isolator (red curve) and after the isolator (blue curve).

Vibration Isolation of the Recirculation Fan (Fig. 2-bottom)**.** The dynamic loading of the recirculation fan affects its mounting on the base plate. It is important to adjust the spring-loaded housed vibration isolators so that the dynamic stiffness in the direction of the dynamic forces is maximal. In case of an incorrect setting, the isolator can generate unwanted vibrations transmitted to the fan structure and its base (13). The isolators used are designed for vertical dynamic load. With a high dynamic load in a direction perpendicular to the axis of the springs where the dynamic stiffness is neglected, there is contact of the fixed parts at the inlet (before the springs) and also at the outlet (behind the springs) of the isolator (see Fig. 3-bottom), which significantly reduces transmission loss of the isolators.

The transmission loss in vertical direction for the two selected spring isolators loaded with radial dynamic force in the frequency band up to 800 Hz is 16 dB and 19.1 dB respectively (Fig. 5). Relatively low values of isolators transmission loss (typically more than 23 dB) are a result of a small attenuation, even a negative attenuation (-3.2 dB) at frequencies of 375 Hz, which cause excessive axial dynamic loading on this type isolators. If only the axial dynamic load in the frequency band up to 800 Hz (Fig. 5-bottom) is taken into account for the housed vibration isolators measured, the transmission loss is only 9 dB and 12.6 dB respectively, and at a frequency of 375 Hz a negative attenuation of -5.3 dB was measured.

In this case, the blocking dynamic stiffness in the axial direction can be neglected and does not correspond to the requirements of the theory mentioned above.

It can be said, however, that amplitudes of low-frequency waves are sufficiently damped, as is the rotational frequency, because low frequencies have a negative impact on the machine itself, the building and technological structures as well as on bystanders (up to 80 Hz).

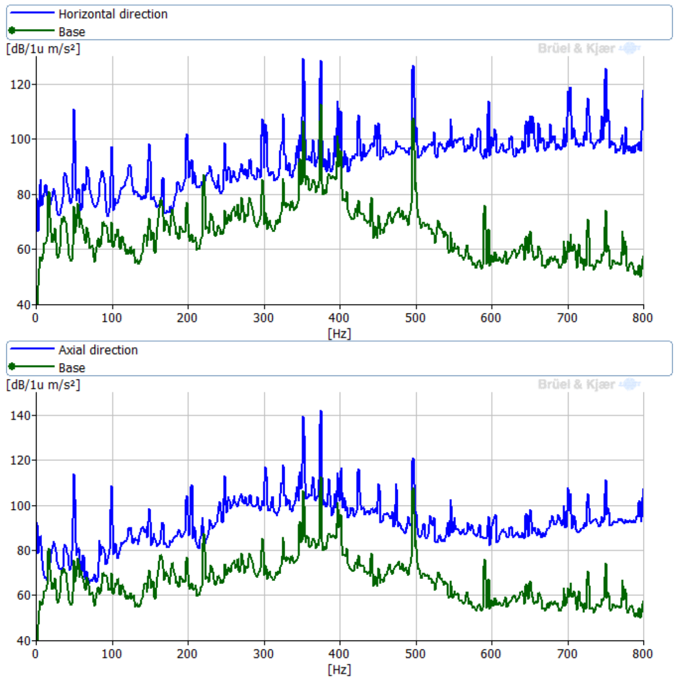

Figure 5. Frequency spectra characterizing the transmission loss through two housed spring vibration isolators from radial forces (top) and axial force (bottom).

Based on the measured results and the theory mentioned above, it can be stated that the transmission loss of the housed spring vibration isolators to the base structure is sufficient for the low frequency amplitudes and can be increased in the frequency range (up to 800 Hz) by reducing the dynamic load of the fan in the axial direction from 300 Hz to 550 Hz frequency range, where the characteristic frequencies of bearings and coupling are significant (13).

Vibration Isolation of a Passenger Vehicle's Combustion Engine (Fig. 6)**.** The load on the car shock absorbers can be assumed in three directions. Measurements, however, have shown that the decisive load is in the vertical direction and the load in the horizontal plane does not have a significant effect on the vibro-isolator transmission loss (5, 14). Due to inevitable internal losses in the vibro-isolation layer, the input energy flow must be larger than the output energy flow, as is the case on engine and transmission shock absorbers (Fig. 6). Determining the frequency components transmitted through the shock absorber are rotational frequency and its harmonics. The maximal attenuation of the shock absorber is within the frequency interval from 300 Hz to 800 Hz. From the FFT analysis before and after the absorber, the attenuation for each dominant or important frequency can be determined. The transmission loss of the engine vibration damper under investigation is more than 32 dB of the frequency range 5 Hz to 800 Hz. The dominant second harmonic of speed frequencies are 23.2 dB and 23.7 dB, respectively (14). The installation of isolators coincides with the conditions for the blocking force and have a relatively large dynamic stiffness on the all sides.

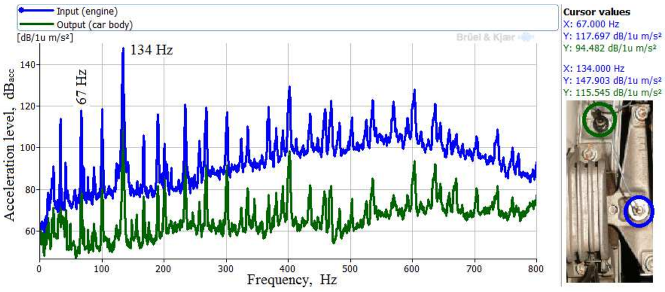

Figure 6. Frequency spectra characterizing the transmission loss at the combustion engine of the passenger vehicle.

DISCUSSION AND CONCLUSIONS

With respect to the isolator properties, geometrical shapes which have large transverse stiffnesses may also display large rotational stiffnesses. When isolators of such shapes are applied in thin-walled structures and for rather high frequency isolation, transfer stiffnesses which include rotational components may become quite important. In such cases the above-mentioned simplification, which takes only one, two or three translational stiffnesses into account, could be inaccurate for the purpose of analysing the vibro-acoustic transmission.

When designing the vibro-isolator of a given mechanical system, it is necessary to know the frequency range of the required attenuation. From examples above-mentioned, it is clear that the viscoelastic isolators do not attenuate sufficiently low frequencies even at the high dynamic stiffness of the base. Similarly it is if the resilient materials is used (5). For amplitude attenuation across the frequency band, spring-loaded silent blocks should be applied, but their loading must be in direction of the spring axes. The transverse loads can lead to contact resonance and the transmission loss will be reduced. Isolator with very good transmission damping characteristics have high dynamic stiffness around their entire circumference, what confirms also the results of the shock damper measurement of a passenger vehicle.

Effective selection, or design of a vibration isolator and/or a vibro-isolation layer, significantly reduces the noise and vibration transmitted to the surrounding environment as well as into the mechanical system itself.

In conclusion, a good judgement of the type of vibration isolators and their typical installation environment is needed to decide on the number of transfer stiffnesses that have to be determined for an isolator.

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