JUSTIFICATION OF THE SCHEME FOR CALCULATING PROTECTIVE EQUIPMENT SOUNDPROOF CABINS.

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The well-known acoustic schemes [1,2,3,4], which are typical for various cars, and mathematical expressions that allow describing the process of the formation of the sound field, convincingly show that the sound field on the outer surfaces of the cab rails is unevenly distributed. All known methods and formulas for calculating the effectiveness of soundproof cabins for various purposes are given in [1,2,3,4], take this unevenness into account when determining the total value of the sound intensity in the cab. However, the works listed above did not consider the development of a sound and vibration-insulating cab, taking into account the unevenness of the sound and vibration fields.

There are no criteria for acoustic design of a soundproof booth, taking into account the unevenness of the sound field and the logarithmic nature of the decibel scales.

As a result, as experience shows, with equal values and sound insulation of various cab fences, its reduced total efficiency does not provide the required noise reduction at the workplace or in the cab. Currently in the literature there is no unified approach to solving problems associated with increasing the noise-insulating efficiency of the cabin, because there is no method for calculating the protective properties of the cabin for airborne noise based on acoustic design criteria.

Let's consider a way to solve this problem using the example of an MTA cabin. Let us assume that the intensity of the sound field in the cockpit is completely determined by the air component of sound energy. All (air) channels for sound energy to enter the cab are through the fence surfaces or are located on the fence surface.

The butt joints of the cab railing are acoustically sealed. The cabin contains fences, as a rule, for cotton MTA cabins this number is 6.

It is assumed that the sound field on the outer surfaces of the cab fences is unevenly distributed, and the intensity of the sound field is known. A diagram of a soundproof cabin with the accepted designations is shown in Fig.1.

The total intensity of the sound field in the cabin is formed as a result of the energetic summation of the intensities of sounds penetrating through the longitudinal railings of the cabin.

Taking into account the accepted assumptions of a soundproof cabin with increased efficiency and corresponding in terms of the value of the soundproofing properties of individual fences, the energy characteristics of the sound field of the area where the operator's workplace is located, it is proposed to evaluate the efficiency of the soundproof cabin of cotton MTA by the following expressions:

(1)

(2)

(3)

L1, L2 …Lm - sound pressure levels on the outer surfaces of the cab fences at the rated frequencies of the octave bands, dB; Zi1, ЗZi2, ЗIm - sound insulation of fences in the normalized frequencies of octave bands, dB; □Ln□ - maximum permissible sound pressure level, dB; i = 1,2,3 …… m-number of fences;

The fulfillment of conditions (1), (2) and (3) was proposed by the authors of this work as the main criteria for the acoustic design of a cabin installed or installed in an uneven sound field and intended to reduce airborne noise.

Analyzing expressions (1), (2) and (3). Condition (1), which proposes the equality of the ratios of the sound pressure levels and sound insulation of individual cabin barriers, shows that the sound insulation properties of the barriers must correspond to the load on their surface with sound energy.

If this condition is met, the cabin will fully meet (in terms of sound insulation efficiency) the characteristics of the sound working area of the installation. Consider the main ways to ensure condition (2).

According to works [3,5], the average actual sound insulation of the cab guard can be determined as follows:

(4)

Where, ZI is the average actual sound insulation of the fence, dB; □ is the average sound absorption coefficient of the fence from the source side; f is the geometric mean frequency of the octave band, Hz; m is the mass of a unit area of the fence, kg / m2; S is the area of the fence, m2; Akab - cabin sound absorption, m2;

Picture1 To the calculation of the soundproof booth. B-sound vibration

From formula (4) it can be seen that the actual sound insulation of the cab fence depends on many different parameters: the physical and mechanical properties of the fence materials (□, m); the geometric dimensions of the fences and the design of the cabin itself (S, A);

spectrum of acoustic signals from sources and sound field of the cab installation location (f).

With previously known sound pressure levels (initial characteristics of the sound field) on the surfaces of the cab fences, a scientifically based choice of various parameters of the fences and the cab, using formula (4), can always find a rational value (ZI) that ensures the fulfillment of conditions (1) for the entire cab.

This problem was solved by the example of a cotton-growing tractor with the calculation of all parameters on a computer. When developing a soundproof cabin that meets condition (1), its efficiency increases by 6-7 dB (A) without increasing metal consumption.

Condition (2) follows from the energy method and shows that the total sound pressure level in the cabin should not exceed the standard noise value.

Condition (2) makes it possible to set the required values of the average actual sound insulation of the cab fences, which can reduce the sound pressure level at the workplace to the levels of the maximum permissible GOST values, taking into account the uneven sound field.

Condition (3) is a condition for the expediency of installing the cab or a condition that makes it possible to exclude the i-th cab fence from condition (2), since with a difference in sound pressure levels of 10-15 dB, the sound pressure level of the i-th cab fence does not significantly affect formation of the total level of sound pressure in the cabin.

On cotton MTA there are various sources (engine, clutch, gearbox, MTA gearboxes, cardan shafts, etc.) that cause sound vibration in the structure of the skeleton and frame, which spreads along them and is transmitted to the cab elements.

In this case, the noise in the cab is determined by the sound emission of the cab fences and its elements, through which sound vibrations propagate from the sources.

Cab controls (steering column, levers and pedals) are often rigidly connected to the active elements of the cotton MTA, and they also form bridges for transmitting vibration energy to the attachment points (floor, instrument panel, etc.).

The vibrating surfaces of these elements become sources of sound, the contribution of the sound vibration of fences and cabin elements to the formation of total noise in the cabin of cotton MTAs has been determined experimentally.

The results of experimental studies have shown that on basic cotton tractors in the frequency range of 62-2000 Hz, the noise in the cabin is formed equally from airborne and structure-borne noise.

It is known that the most characteristic way of transmitting sound vibration to the cab elements is the vibration-isolating cab mount (if any), controls and fittings of the tractor hydraulic system.

The main part of the vibration energy to the fences penetrates through the vibration isolating mountings of the cab, since it often has a low vibration isolation efficiency, which is due to resonance phenomena, the type of elastic element (spring, rubber), the fastening design, operating conditions, loading and the design of the vibration isolators.

In general, the effectiveness of vibration isolation is characterized by several criteria 6.7, the most important of which is vibration isolation (VI). For the harmonious flow of the process of transferring the movement of speed, acceleration, vibration isolation is determined by the ratio 8.

(5)

Where

Vzh, Vv - vibrational speeds of fastening the cabin (or averaged over them), respectively, with rigid fastening of the cabin to the skeleton or MTA frame and through vibration isolators, m / s.

If the cabin perceives a harmonious force or moment from the side of the MTA skeleton or frame, then vibration isolation is determined by the ratio 8:

(6)

Fzh, Fv are the amplitudes of dynamic forces transmitted to the cab with a rigid vibration-insulated installation on the tractor frame, N.

The expressions make it possible to evaluate the effectiveness of vibration isolation of the cabin, but do not determine the required value of the effectiveness of vibration isolation in specific structures at the design stage of vibration isolation means according to their installation schemes.

Well-known methods for calculating specific structural schemes of vibration isolation are given in the work "4". On domestic and foreign tractors, single-link (one-stage) vibration isolation schemes are used, such a scheme is calculated as follows.

The cabin is offered absolutely rigid. Between it and the frame, inertia-free insulators are installed in parallel (as usual, from 4 to 8), which are replaced by one vibration insulator with total stiffness.

Therefore, it is considered that the cab is installed on one vibration isolator. If the vibration isolator is an elastic element with internal friction, then the vibration isolation is determined by the formula □8□:

All results

(7)

f - frequency of exciting force or speed, Hz;

f0 - natural frequency of vibrations of the cabin, Hz;

□ - vibration isolator loss factor

Vibration isolation of an object is determined by formula (7) and has three characteristic areas. In the case of f□□f0, vibration isolation is practically zero, i.e. VI = 0 - the vibration level does not decrease, the movement of the mass is inversely proportional to the stiffness of the vibration isolator.

For this reason, the region where f□□f0 is called the controlled rigidity region. When f = f0, resonance □8□

(8)

In this area, an increase in damping decreases both the primary force and the movement of the mass, and conversely, a decrease in damping increases them.

Therefore, the area where f = f0 is called the controlled damping area. And finally, with a further increase in the frequency of the exciting force or speed, i.e. f >> f0 and in fact, starting f / f0> 3, vibration isolation increases at a rate of 12 dB per octave.

In this area, the movement of a vibration-insulated object can be inertial; it is called an area with a controlled mass.

Thus, vibration isolators operate effectively in the frequency range f / f0> 3 and must be designed so that the resonant frequency f0 lies below the range of exciting frequencies, in which it is necessary to reduce the vibrations of the cabin.

It is recommended that, as far as possible, the vibration isolation system should not be used in the frequency range where "9". To calculate the vibration isolation of the cab using the formula (7), it is required to determine all the frequencies of free vibrations. In general, there are six of them.

They correspond to translational vibrations of the cabin in the directions of three coordinate axes and rotary vibrations around these axes (Fig. 1).

There are three practically important cases of the location of elastic supports and the calculation of the frequencies of free vibrations of vibration-insulated objects. They are given in the work No. 3.

It also contains the basic formulas for calculating all six frequencies of free vibrations of an object. It is especially noted that when choosing a layout for vibration isolators, it is necessary, if possible, to avoid schemes that lead to the appearance of free vibrations, connected in three and in two.

The effectiveness of the vibration isolator layout is determined by their mutual arrangement of the centers of stiffness of elastic supports and inertia of the cabin, as well as their main axes of inertia and stiffness, since. these parameters are due to free vibrations of the cabin, connected by two and three Free vibration frequencies of the cabin on vibration isolators with uncoupled vibrations are determined by the following formulas □3□:

(10)

(11)

Where,

-

frequency of free translational vibrations of the cabin along the axes x, y, z;

- translational stiffness of vibration isolation of the cab along the axes x, y, z, n/m;

- operating weight of the cotton MTA cab, kg;

- Frequencies of free rotary vibrations of the cab along the axes x, y, z;

-

rotary stiffness of vibration isolation of the cabin along the x, y, z axes, n / m;

- moments of inertia of the operating weight of the cab along the x, y, z axes, kg / m2

The translational stiffness of the cab vibration isolation along the three axes is □3□.

(12)

where,

- stiffness of individual vibration isolators along the axes, n / m;

н - number of vibration isolators , I = 1,2,3 … n.

Rotary stiffness of vibration isolation of a cab made of identical vibration isolators relative to the main axes of rigidity D x, y, z are equal to □3□.

(13)

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arms of individual vibration isolators along the axes, m

Considering that и (where G is the bottom of the cab, - radius of gyration along the axes,); then formulas (9) and (10) after simple transformations can be reduced to the following form:

(13)

(14)

where,

δ x,y,z - static settlement (deformation) of vibration isolators under the influence of the cabin gravity, m.

Thus, the frequencies of natural (translational, rotary) vibrations of the cab, installed on vibration isolators, depend on the static settlement of the latter.

For example, when δ = 0.001 m f o = 16 Hz, and when δ = 0.010 m f o = 5 Hz. Formulas (13) and (14) show that a small difference in the static settlement of vibration isolators sharply increases the width of the range of natural vibration frequencies of the cabin on vibration isolators.

Known methods for calculating the vibration isolation of the cabins of cotton-growing tractors are carried out without taking into account this relationship, and as a result, the actual and calculated values of vibration isolation (VI) have a large discrepancy.

The difference at 1000 Hz is approximately 50-60 dB. Therefore, it is necessary to introduce additional conditions for evaluating the effectiveness of vibration isolation of the cotton MTA cab into the increased efficiency of vibration isolation.

Such experience in relation to vibration isolation of machines is available in the USA "4". However, as our research has shown, taking into account the static settlement of vibration isolators, providing it with equal values for all vibration isolators in the cabin, only a slight reduction in sound vibration can be achieved.

In this case, the unevenness of the vibration levels of the points of attachment of vibration isolators to the core or to the frame also leads to a decrease in the effectiveness of vibration isolation.

It (decrease) is due to the fact that, due to the unevenness of the vibration field, the place of installation of the MTA cabin with the same values of vibration isolation (VI) of the vibration isolators of the cabin, the dynamic settlement (deformation) of the latter takes on different values.

Since the rubber of the vibration isolator is an elastic-viscous material, and its dynamic modulus. When a vibration isolator of periodic force acts on a rubber mass, the deformation of the rubber mass does not have time to follow the force due to the so-called effect of the 9□ inherent in elastic materials.

For this reason, the "instantaneous" dynamic stiffness of the vibration isolator is greater than the static one and the difference between the stiffness values depends on the unevenness of the vibration field.

Coupled vibrations appear and the natural frequency range of the cockpit expands dramatically. The existing known methods for calculating the vibration isolation of cotton MTA cabins are performed without taking into account this phenomenon, since there are no conditions for assessing the vibration isolation of the cab, taking into account the unevenness of the vibration field and the installation site, which takes place on all cotton MTA. On row-crop cotton MTA, as shown by the results of the experiment, the unevenness of the vibration field is 3-12 dB.

Based on the foregoing, the authors proposed (for the first time) new conditions for assessing the vibration isolation of a cotton MTA cabin when installed on a certain number of vibration isolators (three, four or more). The offered conditions are described by the following expressions:

(15)

(16)

where,

F1, F2 …Fn - cabin gravity applied to each vibration isolator, H;

L1, L2 …Ln - vibration velocity levels of the vibration isolator attachment point to the island or MTA frame, dB

1,2,3..m - the number of vibration isolators.

Fulfillment of condition (15) makes it possible to evenly distribute the gravity forces of the cab and to combine the centers of stiffness of the vibration isolators in the center of gravity of the tractor cab. This ensures the equality of values as the frequencies of natural vibrations for all vibration isolators, i.e.

$\delta 1 = \delta 2 = \ldots = \delta n$ and according to (2.13) $f1 = f2 = \ldots = fn$

In practice, where vibration isolators of the same type are used, i.e. with the same stiffness values along the main axes, the fulfillment of conditions (15) is achieved by the inclination of the vibration isolator relative to the reference plane. When the vibration isolator is tilted, its rigidity (vertical) decreases in comparison with a conventional installation, which makes it easy to achieve the fulfillment of conditions (2.15) using the same vibration isolators.

A decrease in stiffness, in turn, leads to a decrease in the values of the natural frequencies of the vibration isolation of the cab, and this leads to good vibration isolation, starting from low frequencies. When the vibration isolator is tilted, the distribution of shear and rotation oscillations is achieved, and therefore, it becomes possible to select the frequencies of these oscillations independently of one another [10]. Condition (16) is also achieved by tilting the vibration isolators, i.e. tilting the vibration isolator changes the efficiency of its vibration isolation. Let's consider this phenomenon.

Formula (7) для

$f >> f0$ can be written as follows [10].

, дБ (17)

Substituting f0 in (17) by formula (9) and changing the mass Mk with the gravity G, we obtain dB (18)

Formula (18) shows that, all other things being equal, changes in the value of the vibration isolator rigidity R and the gravity force G applied to the vibration isolator, the values (VI) vary within wide limits. When the vibration isolator is tilted, the value of R changes, as well as the value of G, and accordingly the value of the vibration isolation (VI) of the vibration isolator changes according to (18).

Based on the above calculations, a new layout of the vibration isolators of the cotton MTPA cabin has been developed, taking into account the uneven vibration field of the installation site and the unevenness of the gravity along the vibration isolators.

In the proposed scheme for the first time a six-point combined (vertical and inclined) installation of a vibration isolator of the AKSS-160M type was used. The installation scheme fully ensures the fulfillment of conditions (15) and (16).

An experimental assessment of the effectiveness of the proposed vibration isolation scheme for a cotton tractor cab showed an increase in vibration isolation (VI) by an average of 7-12 dB in the frequency range 20-1000 Hz.

The proposed structure of the frame of a cotton tractor allows to eliminate and minimizes the unevenness of the vibration field of the place of installation of the cab of the cotton tractor. It is equipped with a lattice metal vibration isolator with a description of the structural features of the frame with a vibration isolating lattice.

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