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Innovative Control of Noise and Vibration of Industrial Equipments and Machines

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Abstract ***: Noise and Vibration of industrial equipment is the grave factor influencing its production state, working conditions of staff and job safety. In course of technology development the more potent machines are used, it is quite often accompanied by an increase of vibration and noise level. This is experienced by equipment as it is transmitted to building structures, environment and through staffs. The system of equation advocated in this research work has been permitted to evaluate reduction of machine vibrations caused by unbalance movement of its members, thereby transmitting it onto the floor and the environment. A noise problem generally consists of three inter-related elements- the source, the receiver and the transmission path. This transmission path is usually the atmosphere through which the sound is propagated, but can include structural materials of any building containing the receiver. The development of innovative noise control treatments provides opportunities for applying basic physics and engineering procedures.*

Keywords: *Environment, Industrial equipment, Insulation, Noise, Resonance, Vibration.*

I. Introduction

The more potent machine becomes, the more its vibration and noise disturbs people. Vibration and noise is not only harmful to human being but also hinders fulfillment of working operations, both psychologically and physically. A vibration with frequencies between 25-40 and 60-90 Hz is degraded as visual perception. When frequency of vibration is close to the natural frequency of oscillations of the human body, equal about 5 Hz, operating of vibration becomes especially ideal. In different parts of body the natural frequencies make: in pelvic area 4-6 Hz, in abdominal area 4-8 Hz, and the head 30 Hz (Woodson and Conover 1966). The sources of vibrations of industrial equipment are: impulse applied technological forces, impacts, unbalance of rotated parts, inertia forces of parts with periodic motion. Therefore, the problem of reducing harmful vibrations remains actually permanent; one solution to the problem is the improvement of the kinematics, balancing of inertia forces, developing of shock free technological processes. However these actions are not always possible. That is why the other way to reduce harmful effect of vibration is vibration insulation of the equipment, installed on building structures. Vibration insulation is the reduction of the transmission of vibrations which is reached by the installation of pliable members of small stiffness between vibratory units and adjacent structures (Ya, 1974). Millions of workers around the world are exposed to sound levels that are likely to cause permanent hearing loss, even though many of them wear hearing-protection devices. Many people do not realize that these devices and hearing-protection programs are not the preferred way of protecting hearing. The preferred way, often called "engineering controls," is to reduce the noise of machinery or introduce a noise control element between machinery and workers (Rao, 1995). Engineering controls are preferred for many reasons, including permanence, effectiveness with or without worker/supervisor compliance, less absenteeism, easier communication, lower worker compensation costs, and reduced legal costs. In fact, engineering controls are the protection method of choice according to the Occupational Safety and Health Administration. Many of these controls could be integrated into machinery by original equipment manufacturers, nonetheless, for non-engineering reasons; they have been eliminated from machinery design. Since the late 1940s, scientists and engineers have been working on ways to control noise from machinery (Jensen et al 1978). In the 1970s, the emphasis was on engineering controls in the workplace, but since then the focus has shifted because OSHA has not enforced the requirement for engineering controls and because industry leaders have failed to take into account the risk to hearing when purchasing equipment.

II. Occupational Noise-Exposure Regulation

When the daily noise exposure is composed of two or more periods of noise exposure of different levels, their combined effect should be considered, rather than the individual effect of each. If the sum of the fractions: presented in equation (1) exceeds unity, then the mixed exposure should be considered to exceed the limit value.

$C(1)/T(1) + C(2)/T(2) + \ldots + C(n)/T(n) \quad (1)$

Where:

T (n) indicates the total time of exposure permitted at that level.

Exposure to impulsive or impact noise should not exceed 140 dB peak sound pressure level. Note that the regulation calls for engineering controls to be used first to reduce sound levels to within the limits specified in Table (1), and only if the controls do not succeed in bringing the sound levels down, the hearing-protection devices are to be used in addition. The requirement is that "feasible administrative or engineering controls" be tried. Simply explained, "administrative control" means removing workers from noise exposure by rotating them from noisy to quieter areas. "Feasible" means—that it can be done. However, sometimes the cost of a noise control treatment is cited as a reason why it is "not feasible" for a particular company to install it (Franken, 1974).

A number of very simple engineering controls can often be implemented with great success:

- proper maintenance (e.g., fixing steam leaks)
- modified operating procedures (e.g., relocating an operator and equipment controls to a quieter position)
- relocation of noisy vents away from workers
- replacement of equipment (e.g., buying a quieter version of the product)
- modified room treatment (e.g., introducing sound absorption in the space between equipment and worker to reduce noise in the distant reverberant field)
- relocation of equipment (e.g., putting noisy equipment in areas that are often unoccupied)
- proper operating speed (e.g., running equipment at lower speed to reduce noise)

Despite the effectiveness and ease of taking these simple actions, they are often ignored.

When employees are subjected to sound exceeding those listed in Table -1, feasible administrative or engineering controls shall be utilized. If such controls fail to reduce sound levels within the levels of Table -1, personal protective equipment shall be provided and used to reduce sound levels to within the levels of the table.

TABLE 1 Permissible Noise Exposures

Duration per day, hours	**Sound level dBA slow response**
8	90
6	92
4	95
3	97
2	100
1-1/2	102
1	105
1/2	110
1/4 or less	115

2.2 SOME COMMON SOURCES OF NOISE

2.2.1 Noise Sources Involving Fluid Flow: Noise sources that involve fluid flow include fans, compressors, engines, pumps and valves. The most frequent problem is sound from the discharge, but engineering solutions (e.g., lined ducts; dissipative and reactive silencers; and special-purpose silencers) are available for both intake and discharge noise. Ducts can be lined with sound-absorbing material, such as fiber glass or mineral wool. Typical thicknesses are 1 to 4 inches. Thicker materials are used for low-frequency noise. Dissipative silencers involve using sound-absorbing materials, such as mineral wool or fiber glass, to attenuate noise. A simple dissipative silencer would be a series of parallel baffles running lengthwise from a noise source that requires airflow. The absorptive material would likely be covered with glass-fiber cloth to reduce erosion caused by airflow, and perforated or expanded metal might be added to protect it against contact damage. The silencer could be improved by using longer baffles and decreasing the space between them. Reactive silencers operate on the basis of mismatching acoustic impedance. Whenever a sound wave meets a change in the acoustic impedance, some of its energy is reflected back to the source or back and forth within the silencer(Randall et al 2002),. An example of a reactive muffler is shown in Figure 1.

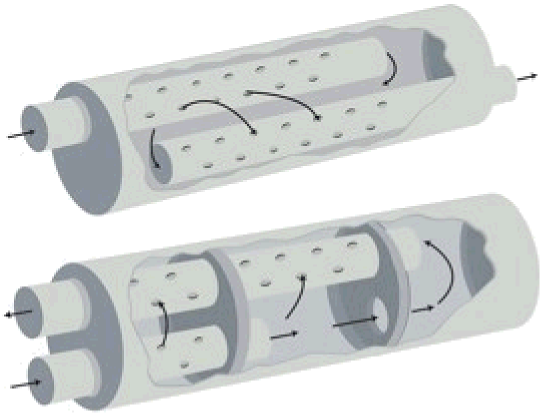


FIGURE 1 Diagram of reactive mufflers showing expansion chambers.

Recently, a special-purpose silencer was developed based on the principles of a Helmholtz resonator (a small neck and a larger cavity, such as a bottle). As sound waves pass over the opening of the neck, a small portion of gas at the neck of the bottle begins to oscillate back and forth. The frequency at which this "slug" of gas oscillates is a function of mass-spring resonance—the slug in the neck acts as the mass, and the volume of gas in the cavity acts as the spring. This resonance is a function of the diameter of the neck, the density of the gas, the length of the neck, the volume of the cavity, and the speed of sound in thegas(Liu,2003). The resonance frequency in Hz of a simple Helmholtz resonator can be calculated from equation (2).

F= $^{1}/_{2\prod(MC)^{0.5}}$ (2)

Where:

M =ρl/A (ρ = the gas density (kg/m^3)

l = the "effective" length of the neck (m), and

A = the cross-sectional area of the neck (m^2).

C = V/c^2 (3)

Where:

V = the volume of the cavity (m^3), = the density (kg/m^3), and

c = the speed of sound (m/s) in the gas).

The ends of the tube influence the resonance frequency by increasing the effective length (l) of the neck. Note that the resonance frequency is independent of density. The resonance frequency of a typical 500 ml water bottle, about 200 Hz, can be excited by blowing across the top of the bottle.

One special-purpose silencer, invented by (Bruce, 2004) uses the Helmholtz concept to reduce centrifugal compressor noise, which often includes a strong tonal component at the blade-passing frequency of the compressor. The resonators for the Dresser-Rand Duct Resonator Array (DRA) are shown in Figure 2.

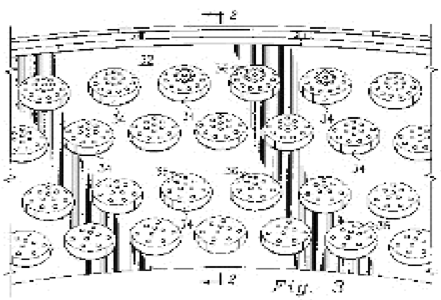


FIGURE 2 Dresser-Rand Duct Resonator Array .

The numerous holes in the resonators, acting as masses in parallel, raise the resonance frequency. The cavity behind these holes acts as the volume for the resonator. The DRA, which consists of numerous resonators positioned at the diffuser of the compressor, can also be located in a pipe spool in discharge piping. The DRA reduces the A-weighted sound level by at least 10 dB—which is similar to "halving" the loudness of the sound. Significant noise radiating from piping in refineries and from forced-draft and induced-draft fan ducts can be reduced simply with layers of treatment around the piping (Figure 3). Typically,

The first layer is 2 to 4 inches of 6 to 8 lbs/ft^3 glass fiber or mineral wool wrapped around the pipe. Next a mass-loaded vinyl or lead sheeting weighing 1 to 2 lbs/ft^2 is wrapped around the glass fiber or mineral wool. Finally, a weatherproof covering is added. This type of treatment can provide 10 to 20 dBA of noise reduction, depending on the details of the installation.

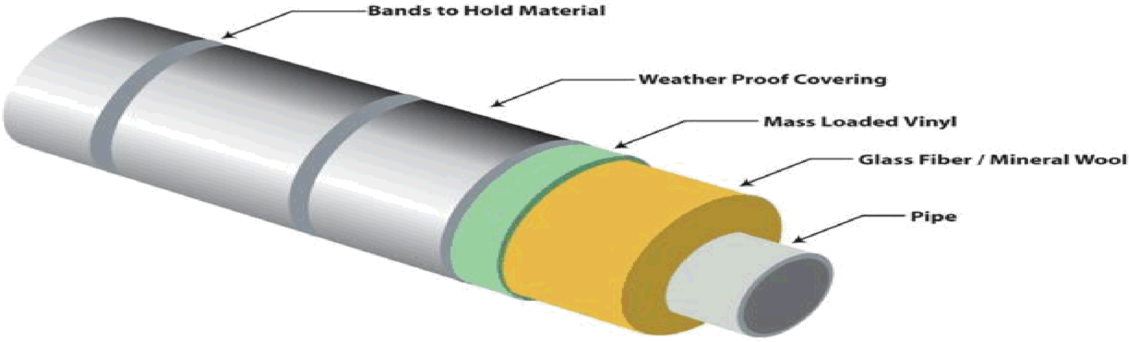


FIGURE 3 Pipe Logging.

2.2.2 Radiated Noise from Machinery Housings: Airborne noise can radiate from any surface. In a piano, for example, the keys strike the hammers, which hit the strings. Although the strings do not produce much sound by themselves, they are attached to the soundboard, which is very large compared to the strings and radiates the sound. In general, the larger the vibrating panel, the more sound is radiated from the surface. Another example is a metal parts bin into which metal parts are dropped. If the bin is made of perforated, rather than solid metal, the radiating area, and thus the radiated sound, is reduced. Of course, materials with high internal damping radiate less noise. If the bin were made of rubber (with high internal damping) rather than metal (with low internal damping), the radiated sound would be reduced accordingly. Sometimes machinery is housed inside an enclosure provided by the original equipment manufacturer. If there is any possibility that the resonance frequencies of the enclosure panels will be excited, it is desirable that the housing be treated with a damping material. If the machine inside the enclosure causes significant vibration of the enclosure housing and structure, then the panels should be "vibration-isolated" from the structure. In addition, it may be useful for the machinery enclosure to be mounted on vibration isolators to minimize the amount of vibration transmitted to the floor.

2.3 Machinery Shields: An acoustical shield may be inserted between the worker and a noisy section of a machine. An acoustical shield, often mounted on the machine, can provide 8 to 10 dB of noise reduction under the following circumstances: the worker is near the noisy operation; the smallest dimension of the shield is at least three times the wavelength of the dominant noise; and the ceiling above the machine is covered with sound-absorptive material (Stewart,2007).Shields can be manufactured from safety glass, quarter-inch thick clear plastic, metal, or wood. Criteria for selecting materials include durability, cost, the need for visual observation of the operation, and the need for physical access to the operation. If possible, oil-resistant, cleanable, sound-absorptive materials should be incorporated into the machine side of the shield. Handles and casters can be provided to facilitate moving of the shield, and hinged sections can be incorporated into the design to provide physical access to the machine. Neoprene can be used to minimize acoustical leaks through joints or hinges. When shields are used to replace less acoustically efficient machine guards, the shield should be fitted carefully to cover all noise leaks and then vibration-isolated to keep the shield from vibrating with the

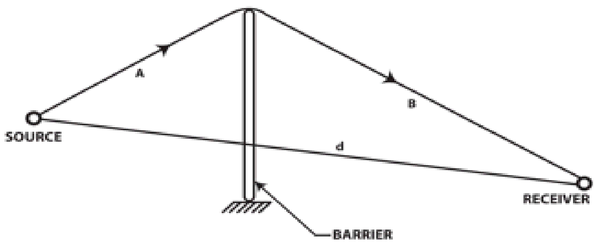


FIGURE 4 Geometry for determining sound attenuation by a noise barrier (d is the straight line distance from source to receiver and A+B is the shortest path length of wave travel over the wall between source and receiver).

2.4 Barriers: Any solid wall that blocks the line of sight between a noise source and an observer will reduce the noise level at the location of the observer. The noise reduction depends on the frequency of the noise, the distance between the source and the wall, the distance between the receptor and the wall, and the height of the wall. Low-frequency sound, which has wavelengths comparable to the size of the wall, diffracts around the ends and over the top. Thus, low-frequency sounds are less attenuated than high-frequency sounds. Typically, low-frequency sounds are attenuated by less than 5 dB, whereas high-frequency sounds can be attenuated by as much as 20 dB. The highest practical value for barrier attenuation is 24 dB. If the noise source is inside a room, then the barrier effect may be reduced, depending on the room absorption and the location of the receivers relative to the barrier wall. Most formulas for calculating the attenuation provided by a barrier assume that the wall is infinitely long. One typical calculation procedure uses the difference between a direct path from the source to the receiver and the path over the barrier (Figure 4). This difference is called the "path length difference."Attenuation = 10 log (3 + 0.12 f d).–P), where f = the frequency in Hz, and P is the path difference in meters (A+B).

Partial Enclosures: A partial enclosure is a series of walls around a machine with the top left open. This treatment can be effective inside a plant for positions near a wall. However, some of the noise radiates out the open top and contributes to the reverberant sound in the room. Reflections from the ceiling increase the sound level at greater distances from the barrier (Figure5).

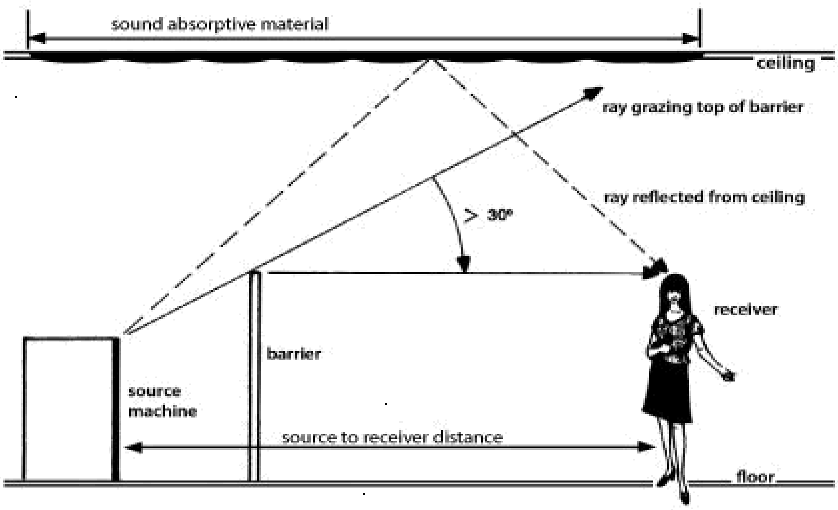


FIGURE 5 Source-barrier-reciever geometry for an indoor noise source.

For a 10 dB reduction, the angle shown in the figure must be greater than 30, and the ceiling must either be sound absorptive or 1.5 times the distance from the source to the receiver. A sound-absorptive ceiling can reduce reflected sounds, thereby increasing the effectiveness of the barrier. For maximum effectiveness, the sound-absorptive ceiling should extend out to the location of the receivers, and the inside of the barrier walls should be sound absorptive.

Total Enclosures: A total enclosure with a closed top will provide better noise reduction than a partial enclosure. However, enclosures usually need openings to provide access for personnel (e.g., visual, maintenance, operator usage); access to materials (e.g., raw materials, product, scrap removal and ventilation. Leaks around doors, windows, and hatches greatly reduce the acoustical effectiveness of enclosures. Closed-cell elastomeric weather-stripping with a pressure-sensitive adhesive can be used to prevent sound leakage from around doors, windows, and hatches. Special acoustical gaskets are available, as well as magnetic-strip gaskets similar to those used on refrigerator doors. If workers need to be able to see inside an enclosure, lighting may be required. If workers evaluate the performance of the machinery by its sound, it may be necessary to retrain them or to place a rugged microphone inside the enclosure that sends a signal to a small adjustable loudspeaker at the worker position. Occasionally, it is possible to develop processors that incorporate workers' knowledge to automatically adjust the machinery for optimal performance. Openings for raw materials, products, and scrap-flow can be tunnels lined with sound-absorptive material. The noise reduction will depend on the length and cross-section of the tunnel, as well as the thickness of the sound-absorptive material.

Ventilation is required for all total enclosures and for some partial enclosures. The amount of air required for cooling is a function of the heat generated within the enclosure, as shown in equation:4

$Q=1.76W/\Delta T$ 4

Q = the flow of cooling air in cubic feet per meter at sea level,

W = the watts of heat generated, and

ΔT = the temperature rise permitted above the ambient temperature (F). Ventilation openings can be acoustically lined ducts, elbows, or mufflers, depending on the severity of the problem. Machinery with special heat-sensitive equipment may require special cooling. Neither the enclosure panels nor the enclosure structure should be in contact with any part of the machinery. If the enclosure is mounted on the machinery, it should be vibration-isolated from the machinery. All holes in the enclosure for electricity, oil, water, steam, air, or hydraulic power must open into a junction box packed and sealed with at least 3 inches of glass fiber. A convenient design can be built on an angle-iron frame to which the enclosure panels can be attached with quarter-turn captive screws. The enclosure should be as small as possible without touching the machinery. Noise reductions of 20 dB can be obtained with careful attention to design and construction.

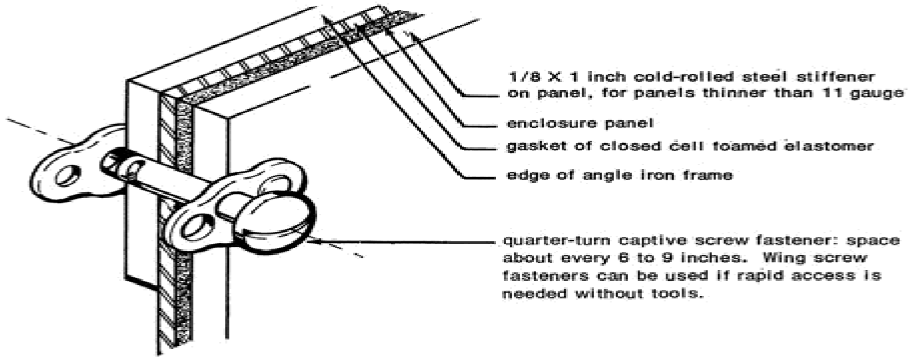


FIGURE 6 Enclosure panels secured to frame by quarter-turn fasteners.

Figure 6 shows the connection of the panels to the angle iron frame. Damping and sound-absorption material are attached to the interior of the panel. The enclosure should also be vibration-isolated from the floor (Figure 7). If the machine vibrates, it may also be important to isolate it using steel springs or elastomers, depending on the circumstances.

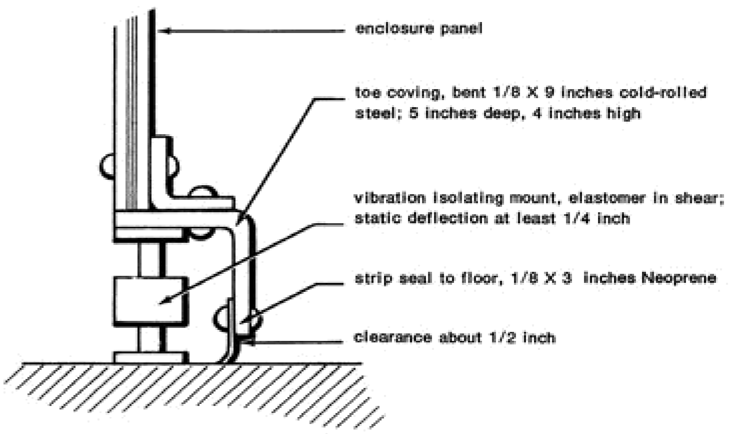


FIGURE-7 Vibration isolation and toe covering.

If the enclosure panels are metal, their resonances can be distributed uniformly in frequency if the panel is reinforced by bolted-on angle irons (bolting and damping). These stiffeners should be placed to divide the panel into smaller areas, no two of which are the same size and shape. Frames for doors, windows, and hatches can also be used as stiffeners. Windows are an acoustical weak link in enclosures. Generally, if the A-weighted sound level must be reduced by more than 20 dB, double-glazing will be necessary. The inside layer, which must withstand rough handling and cleaning to remove oil, grease, and dirt, should be safety glass. All panes should be set into soft elastomeric gaskets. Room-temperature-setting silicone rubbers are useful. The window(s) should be placed carefully to provide the necessary information to the operator. In extreme cases, closed-circuit video monitoring can be used. If the dimensions of the enclosure result in resonance, the enclosure can be driven to high levels of vibration and become a new radiator of these sound components. When the enclosure is driven to high levels of vibration, the sound-pressure level outside the enclosure can be higher than it was before the enclosure was installed. If the noise source occupies a sufficient fraction of the room volume, this effect can be significantly reduced by using absorbent lining on the interior surfaces of the enclosure and by stiffening the panels and lining them with damping materials (Inovich, 1990).

2.5 **Some common noise and their Control in Industry**

Many plants are unnecessarily noisy because the most basic elements of noise control are not applied. The table-2 gives a brief summary of some common noise sources and applicable noise control techniques, materials or devices. Although these measures may reduce noise from the sources treated, remember, to obtain an overall measurable noise reduction requires the noise source to be the dominant source in the area. All new equipment should be purchased to be quiet. In general, encourage buyers to specify that sound levels should meet 80 dBA at 1m. It is unlikely that purchasing equipment meeting 85 dBA at 1m will give plant sound levels under 85 dBA when multiple items are installed.

Noise Source	**Approach**	**Principle**
Air Exhaust	Air Exhaust Muffler	Spreads air exhaust over many small holes to reduce velocity.
Air Jet	Air Thrust Nozzle (for cooling, cleaning, drying or moving)	Entrains air to primary jet to increase airflow at a slower speed, i.e. quieter but with increased thrust.
Fan	Inlet or Outlet Silencer	Absorbs sound in baffles lined with fiberglass or mineral wool. Specification requires some expertise.
New Electric Motor (most often required for 3600 rpm and higher)	Quiet Line Motor	Available in most sizes and speeds. Usually higher efficiency than conventional noisy motor. Much better option than quieting after purchase.
Existing electric motor	Motor silencer/motor mute	Silences motor cooling fan, usually the major noise source. Must be sized not to overheat motor. Better to buy a quiet motor in the first place.
Noise Source	**Approach**	**Principle**
"Singing" motor	Filter electric power supply	Power supplies producing DC or current for variable speed motors often produce audible harmonics in the regular plant power system.
New flow valves	Buy to meet noise specification.	Most valve selection programs will help select quiet valves. These are premium price, but often worth the extra cost.
Existing loud valves (usually high pressure drop)	Quiet trim valve	Quiet trim can be retrofit in existing valves in some situations to make them considerably quieter.
Existing loud valves (usually high pressure drop)	Orifice plate	Introduce an orifice plate across the pipe to reduce pressure drop across the noisy valve.
Flow noise in pipes	Repair leaks and insulate pipes	Acoustical insulation can reduce noise from piping, but in some cases must extend a considerable distance from a noisy valve.

Flow noise in pipes	Wrap acoustical lagging around pipes	Wrap pipes in a composite of mineral wool blanket covered with metal jacket or loaded vinyl
Pump rooms and other similar small equipment rooms	Line with sound absorbing material.	Small industrial rooms can be highly reverberant, increasing sound levels inside.
Separate different circuits		Allows shut down circuit to be worked on under quiet conditions while other circuit continues to operate.
Isolated, noisy, automatic equipment	Noise enclosure with heavy (steel) outer shell and sound absorbing lining (fiberglass)	Must be designed to provide inspection, light, access, maintenance and adequate cooling.
Hydraulic equipment	Isolate from drip trays and tanks, insulate hoses and enclose pump if necessary	Can often be specified quiet
Ventilation openings	Acoustical louvres	Provide about 10 dB attenuation with 50% free area

Table -2A

Control Feature	**Approach**	**Principle**
Operator Booth	Acoustical leaks developed	Check for poor door & window seals, holed or cracked panels, interior absorbent walls & ceiling treatment damaged or removed, booth in mechanical contact with mill.
Machine enclosure	Acoustical leaks developed	Check for poor door & window seals, holed or cracked panels, interior absorbent walls & ceiling treatment damaged or removed infeed/outfeed tunnels without cover flaps, machine in mechanical contact with enclosure wall.
Noisy mobile equipment	Acoustical leaks developed	Muffler missing, damaged, rusted or incorrectly sized

Table -2B

Buying quiet equipment is always better than quieting noisy equipment that is already installed. Applying the above should help this to happen.

3.0 Model of a machine installed on elastic vibration absorbers

The analysis of vibrations of production equipment carried out by many researchers (Ya, 1974) and (Hansan et al 2003). The outcomes of analysis of vibration system is shown on Fig.2 This system corresponds to production machine installed on elastic members. It was considered that the machine consists of fixed and moving part. The last ones are the masses m_1,m_2…….m_m, the motion of which is determined by the machine structure which is considered less than the mass M_o of fixed parts. The model is placed in the systems rectangular coordinates. One of them XYZ is connected to mass M_O, origin of the system O is combined with the center of mass G_O. Otherξ_O, η_O,ξ_O ,with the origin in the same point O, is independent, fixed in space. The axes of the system XYZ coincide with principal axes of inertia of the mass M_O. In the balanced state both of these systems coincide. The connection of systems at the arbitrary moment is determined, the coordinates ξ_O ,η_O,ξ_O, the point $G_O$and angles φ,Ψ,Θ, which are selected to be small for small oscillations of the mass M_O. The mass of the i^{th} traveling part is M_i, coordinates of its center of gravity are X_i, Y_i, Z_i. The sum of parts m, is equal m, its coordinates are X, Y, Z.

1. From the expressions for a kinetic energy after transformations one get the left parts of the Lagrange equations of the second kind:

$$\left|\begin{matrix} (M_o+m)\ddot{\xi}_o & + & & m & \ddot{X} \\ (M_o+m)\ddot{\Pi}_o & + & & m & \ddot{y} \\ (M_o+m)\ddot{\xi}_o & + & & m & \ddot{z} \\ (A+a)\ddot{\varphi} & + & \sum m_i & & (y\ddot{_{\iota}x_{\iota}} - \ddot{x}_{\iota}y_i) \\ (B+b)\ddot{\Psi} & + & \sum m_i & & (x\ddot{_{\iota}z_{\iota}} - \ddot{z}_{\iota}x_i) \\ (C+c)\ddot{\theta} & + & \sum m_i & & (z\ddot{_{\iota}y_{\iota}} - \ddot{y_{\iota}z_{\iota}}) \end{matrix}\right| \ldots\ldots\ldots\ldots\ldots\ldots .(5)$$

Lagrange equations of the second order
In which; m$\ddot{X}$, m$\ddot{Y}$ and m$\ddot{Z}$ are Inertia forces of the linearly moving parts,

$\sum m_i(\ddot{y}_\iota x_i$ - $\ddot{x}_\iota y_i)$, $\sum m_i(\ddot{x}_\iota z_i$ - $\ddot{z}_\iota x_i)$ and $\sum m_i(\ddot{z}_\iota y_i$ + $\ddot{y}_\iota z_i)$ are distributing moments
A, B, C – principal moments of inertia of the mass M_o concerning fixed axes,
a, b, c – total moments of traveling masses concerning axes XYZ, and they will be expressed as shown in equation (6)

$$\left.\begin{matrix} a = I_{1z} + I_{2z} + \cdots + I_{iz} \\ b = I_{1y} + I_{2y} + \cdots + I_{iy} \\ c = I_{1x} + I_{2x} + \cdots + I_{ix} \end{matrix}\right\} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (6)$$

Where I_{ix},I_{iy}, I_{iz} and- moments of inertia of i^{th} mass concerning axes X, Y, Z.
The right hand members of Lagrange differential equations are composed by means of the expressions for the potential energy of the system. The potential energy for n elastic members is given by the expression indicated in equation (7)

$$P = \frac{1}{2}\left(\sum_{i=1}^{n} c_{\xi i}u_i^2 + \sum_{i=1}^{n} c_{ni}v_i^2 + \sum_{i=1}^{n} c_{\xi i}w_i^2 + \sum_{i=1}^{n} k_{\xi i}\varphi^2 + \sum_{i=1}^{n} k_{ni}\Psi^2 + \sum_{i=1}^{n} k_{\xi i}\theta^2\right)\ldots (7)$$

Where:
U, v, and w – Deformation of the elastic members,
C – Linear stiffness of elastic members,
K – Torsion stiffness of elastic members.
The deformation of the elastic members can be expressed as presented in equation 8:

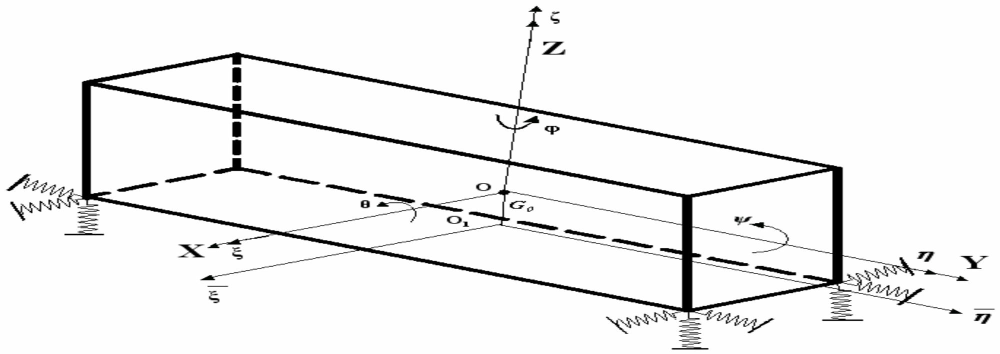
$$\left.\begin{matrix} u_i = \xi_o + \Pi_i\phi - \xi_i\Psi \\ v_i = \eta_o + \xi_i\theta - \xi_i u \\ w_i = \xi_o + \xi_i\Psi - \Pi_i\,\theta \end{matrix}\right\} \ldots\ldots\ldots\ldots\ldots (8)$$

Substitute equation (8) in equation (37we get:

$$P = \frac{1}{2}\left(\sum_{i=1}^{n} c_{\xi i}(\xi_o + \Pi_i\varphi - \xi_i\Psi)^2 + \sum_{i=1}^{n} c_{ni}(\Pi_o + \xi_i\theta - \xi_i\,\varphi)^2 + \sum_{i=1}^{n} c_{\xi i}(\xi_o + \xi_i\,\Psi - \eta_i\theta)^2 + \sum_{i=1}^{n} k_{\xi i}\,\theta^2 + \sum_{i=1}^{n} k_{\Pi i}\,\Psi\theta^2 + \sum_{i=1}^{n} k_{\xi i}\,\varphi^2\right) \ldots\ldots\ldots. (9)$$

Finding of partial derivatives from potential energy on generalized coordinates gives right hand members of the system of differential equations depicting oscillations of the model of the machine installed on the elastic shock- absorbers (Fig.2) as shown in equation 10:

$$\left.\begin{matrix} (M_o+m)\xi_o + \left(\xi_o C_\xi + \varphi u_\Pi - u_\xi\Psi\right) = -m\ddot{X} \\ (M_o+m)\ddot{\Pi}_o + \left(\Pi_o C_\Pi + \theta V_\xi - V_\xi\varphi\right) = -m\ddot{y} \\ (M_o+m)\ddot{\xi}_o + \left(\xi_o C_\xi + \Psi\omega_\xi - \omega_\eta\theta\right) = -m\ddot{z} \\ (A+a)\ddot{\varphi} + \left(\varphi C_{\xi\xi} + \xi_o u_\eta - \Pi_o V_\xi - \Psi C_{\eta\xi} - \theta C_{\xi\xi}\right) = -\sum m_i(\ddot{y}_\iota x_i - \ddot{x}_\iota y_i) \\ (B+b)\Psi + (\ddot{\Psi} C_{\eta\eta} - \xi_o u_\xi + \xi_o\omega_\xi - \theta C_{\xi\eta} - \varphi C_{\xi\eta}) = -\sum m_i(\ddot{x}_\iota z_i - \ddot{z}_\iota x_i) \\ (C+c)\theta\left(\theta C_{\xi\xi} + \Pi_o V_\xi - \ddot{\xi}_o\omega_\eta\right) = -\sum m_\iota(\ddot{z}_\iota y_\iota - \ddot{y}_\iota z_i) \end{matrix}\right\} \ldots\ldots\ldots (10)$$



Fig, 8. Model of the production machine installed on elastic members

In these equations: the linear stiffness's are as indicated in equation 11:

$$\left.\begin{aligned} C_\xi &= \sum_{i=1}^{n} C_{\xi i} \\ C_\eta &= \sum_{i=1}^{n} C_{\eta i} \\ C_\zeta &= \sum_{i=1}^{n} C_{\zeta i} \end{aligned}\right\} \ldots\ldots\ldots\ldots (11)$$

- Linear – rotary stiffness's are shown in equation 12:

$$\left.\begin{aligned} u_\eta &= \sum_{i=1}^{n} C_{\xi i}\eta_i & u_\zeta &= \sum_{i=1}^{n} C_\zeta \xi_i \\ v_\xi &= \sum_{i=1}^{n} C_{\eta i}\xi_i & v_\zeta &= \sum_{i=1}^{n} C_{\eta i}\zeta_i \\ w_\xi &= \sum_{i=1}^{n} C_{\zeta i}\xi_i & w_n &= \sum_{i=1}^{n} C_{\zeta i}\eta_i \end{aligned}\right\} \ldots\ldots\ldots\ldots (12)$$

***The torsion stiffness's are presented in equation 13:**

$$\left.\begin{aligned} C_{\xi\xi} &= \sum_{i=1}^{n}(k_{\zeta i} + C_{\eta i}\xi_i^2 + C_{\zeta i}\eta_i^2) \\ C_{\eta u} &= \sum_{i=1}^{n}(k_{\eta i} + C_{\zeta i}\xi_i^2 + C_{\xi i}\zeta_i^2 \\ C_{\zeta\zeta} &= \sum_{i=1}^{n}(k_{\zeta i} + C_{\xi i}\eta_i^2 + C_{\eta i}\xi_i^2 \end{aligned}\right\} \ldots\ldots\ldots\ldots (13)$$

***** Gyroscopic stiff nesses are given by equation 14:

$$\left.\begin{aligned} C_{\xi\eta} &= C_{\eta\xi} = \sum_{i=1}^{n} C_{\zeta i}\xi_i\eta_i \\ C_{\eta\zeta} &= C_{\zeta\eta} = \sum_{i=1}^{n} \xi_i\,\eta_i\zeta_i \\ C_{\zeta\xi} &= C_{\xi\zeta} = \sum_{i=1}^{n} C_{\eta i}\,\xi_i\zeta_i \end{aligned}\right\} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (14)$$

The given system includes in its right hand members the disturbing forces caused by the unbalanced moving masses. Equations (10) can be used at the designing of vibration dampers (absorber) in order to avoid loads transmitted by the machine onto the floor. In many cases not all the motions of the machine are interdependent, and then the system (10) is simplified. If two principal central axes of stiffness are only principal axes of inertia, but not the central ones ,the principal central axis of inertia will be the third principal central axis of stiffness then all gyroscopic (14) and four of six linear – rotary (12) stiff nesses are equal zero. In this case the system (10) is essentially simplified and becomes as denoted by equation 15.

$$\left.\begin{aligned} (M_o + m)\ddot{\xi}_o + \xi_o C_\xi &= -m\ddot{X} \\ (M_o + m)\ddot{\eta}_o + \eta_o C_\eta &= -m\ddot{y} \\ (M_o + m)\ddot{\zeta}_o + \zeta_o C_\zeta &= -m\ddot{z} \\ (A + a)\ddot{\varphi} + \varphi C_{\zeta\zeta} &= -\sum m_i(\ddot{y}_i x_i - \ddot{x}_i y_i) \\ (B + b)\ddot{\Psi} + \Psi C_{\eta\eta} &= -\sum m_i(\ddot{x}_i z_i - \ddot{z}_i x_i) \\ (C + c)\ddot{\theta} + \theta C_{\zeta\zeta} &= -\sum m_i(\ddot{z}_i y_i - \ddot{y}_i z_i) \end{aligned}\right\} \ldots\ldots\ldots\ldots\ldots\ldots (15)$$

3.0 **Developing of model of machine installed on elastic–dissipative vibration dampers**

The systems (10) and (15) are not taking into account dissipation of energy of oscillations in the vibration dampers. To take into consideration this dissipation one should add a damping to the vibration dampers and simplify the scheme a little, having shown only vertical components of each of four hearings Ilnawafleh, (2001). In this case the machine is presented by the model shown at Fig.3. The origin of the coordinated system is placed in the point of the static equilibrium of the center of masses of the machine, as the axes of coordinates the central axes of inertia of the machine are considered. The model has six degrees of freedom by the way of linear displacement along axes and angular displacements around the last .As a result of a unbalance of mobile masses there is a disturbing force Ilnawafleh, et al (2005) and Shilin, et al (2008) which can be accepted equal *Q= sinωt*, as well as in the already reviewed model. The restoring force of each vibration damper is proportional to its deformation. The force of a viscous strength of vibration damper absent in the model at Fig.2 is proportional to the speed of deformation.

The vibrations of the model (Fig, 3) are described by equations 16.

$$\left.\begin{aligned} m\ddot{x} &= \textstyle\sum_{i=1}^{4} F_{xi} + Q sin\omega t \\ m\ddot{y} &= \textstyle\sum_{i=1}^{4} F_{yi} \\ m\ddot{z} &= \textstyle\sum_{i=1}^{4} F_{zi} - P \\ I_x\ddot{\Psi} &= -F_{zi}l_{y1} + F_{z2}l_{y2} - F_{z3}l_{y1} + F_{z4}l_{y2} + \textstyle\sum_{i=1}^{4} F_{yi}z_b \\ I_y\ddot{\varphi} &= F_{z1}l_{x2} + F_{z2}l_{x2} - F_{z3}l_{x1} - F_{z4}l_{x1} + \textstyle\sum_{i=1}^{4} F_{xi}\, z_b - Q sin\omega t z_Q \\ I_z\ddot{\theta} &= F_{x1}l_{y1} - F_{x2}l_{y2} + F_{x3}l_{y1} - F_{x4}l_{y2} + F_{y2}l_{x2} - F_{y3}l_{x1} - F_{y4}l_{x1} \end{aligned}\right\} \text{………..} \; (16)$$

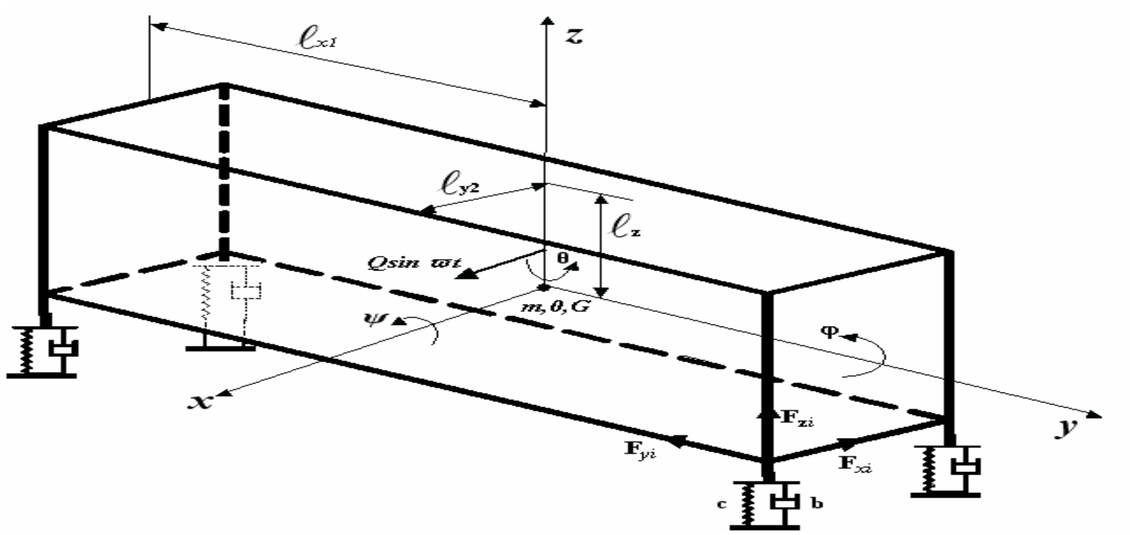


Fig 8. Model of the production machine installed on elastic-damping supports

Where

$$\left.\begin{aligned} F_{x1} = F_{x3} &= -C_x\left(x + z_b + l_{y1}\theta\right) - b_x\left(\dot{x} + z_b\dot{\varphi} + l_{y1}\dot{\theta}\right) \\ F_{x2} = F_{x4} &= -C_x\left(x + z_b\varphi + l_{y2}\theta\right) - b_x\left(\dot{x} + z_b\dot{\varphi} + l_{y2}\dot{\theta}\right) \\ F_{y1} = F_{y2} &= -C_y(y + z_b\Psi + l_{x2}\theta) - b_y(\dot{y} + z_b\dot{\Psi} + l_{x2}\dot{\theta}) \\ F_{y3} = F_{y4} &= -C_y(y + z_b\Psi + l_{x1}\theta) - b_y(\dot{y} + z_b\dot{\Psi} + l_{x1}\dot{\theta}) \\ F_{z1} &= -C_z\left(z + l_{x2}\varphi + l_{y1}\Psi\right) - b_z\left(\dot{z} + l_{x2}\dot{\varphi} - l_{y1}\dot{\Psi}\right) \\ F_{z2} &= -C_z\left(z + l_{x2}\varphi + l_{y2}\Psi\right) - b_z\left(\dot{z} + l_{x2}\dot{\varphi} - l_{x2}\dot{\Psi}\right) \\ F_{z3} &= -C_z\left(z + l_{x1}\varphi + l_{y1}\Psi\right) - b_z\left(\dot{z} - l_{x1}\dot{\varphi} - l_{y1}\dot{\Psi}\right) \\ F_{z4} &= -C_z\left(z + l_{x1}\varphi + l_{y2}\Psi\right) - b_z\left(\dot{z} - l_{x1}\dot{\varphi} + l_{y2}\dot{\Psi}\right) \end{aligned}\right\} \text{………………} \; (17)$$

Substituting (17) in (16) the transformations would lead to the system of linear differential equations of the second order shown in equation 18.

$$\left.\begin{aligned}
&M\ddot{x} + b_{11}\dot{x} + b_{15}\dot{\varphi} + b_{16}\dot{\theta} + C_{11}x + C_{15}\varphi + C_{16}\theta = Q\ sin\omega t\\
&M\ddot{y} + b_{22}\dot{y} + b_{24}\dot{y} + b_{26}\dot{\theta} + C_{22}y + C_{24}\Psi + C_{26}\theta = 0\\
&M\ddot{z} + b_{33}\dot{z} + b_{34}\dot{\Psi} + b_{35}\dot{\varphi} + C_{33}z + C_{34}\Psi + C_{35}\varphi = -P\\
&I_x\ddot{\Psi} + b_{42}\dot{z} + b_{43}\dot{z} + b_{44}\dot{\Psi} + b_{45}\dot{\varphi} + b_{46}\dot{\theta} + C_{42}y + C_{43}z + C_{44}\Psi + C_{45}\varphi + C_{46}\theta = 0\\
&I_y\ddot{\varphi} + b_{51}\dot{x} + b_{53}\dot{z} + b_{54}\dot{\Psi} + b_{55}\dot{\varphi} + b_{56}\dot{\theta} + C_{51}x + C_{55}\varphi + C_{56}\theta = -Qz_b \sin\omega t\\
&I_z\ddot{\theta} + b_{61}\dot{x} + b_{62}\dot{y} + b_{64}\dot{\Psi} + b_{65}\dot{\varphi} + b_{66}\dot{\theta} + C_{61}x + C_{62}y + C_{64}\Psi + C_{65}\varphi + C_{66}\theta = 0
\end{aligned}\right\} \ldots\ldots (18)$$

in which stiffnesses C are described by the relations presented in equation 19.

$$\left.\begin{gathered}
C_{11} = 4C_x;\ C_{15} = C_{51} = 4C_xZ_b;\ C_{16} = C_{61} = 2C_x\,(l_{y1} - l_{y2})\\
C_{22} = 4C_y;\ C_{24} = C_{42} = 4C_yZ_b;\ C_{26} = C_{62} = 2C_y(l_{x2} - l_{x1})\\
C_{33} = 4C_z;\ C_{34} = C_{43} = 4C_z\big(l_{y2} - l_{y1}\big);\ C_{35} = C_{53} = 2C_z(l_{x2} - l_{x1})\\
C_{44} = 2C_z\big(l_{y2}^2 + l_{y1}^2\big) + 4C_yZ_b^2\\
C_{45} = C_{54} = C_z(l_{x2}l_{y2} + l_{x1}l_{y1} - l_{x1}l_{y2} - l_{x2}l_{y1})\\
C_{46} = C_{64} = 2C_yZ_b(l_{x2} - l_{x1})\\
C_{55} = 2C_z(l_{x1}^2 + l_{x2}^2) + 4C_xZ_b^2;\ C_{56} = C_{65} = 2C_xZ_b(l_{y1} - l_{y2})\\
C_{66} = 2C_x\big(l_{y1}^2 + l_{y2}^2\big) + 2C_y(l_{x1}^2 + l_{x2}^2)\\
C_{12} = C_{13} = C_{14} = 0;\ C_{23} = C_{25} = 0;\ C_{36} = 0
\end{gathered}\right\} \ldots\ldots (19)$$

The coefficients of strength are described quite similarly.

IV. SOLUTION OF THE COMPOSED SET OF EQUATIONS

The natural frequencies are found by means of the determinant of the equation system (17) under the condition that for the zero instant at the linear and angular displacements and the speeds at all coordinates are considered equal zero and that the right side members of the equation system (17) also equal zero as shown in equation 20.

$$\begin{vmatrix}
C_{11} - M_w^2 & 0 & 0 & 0 & C_{15} & C_{16}\\
0 & C_{22} - M_w^2 & 0 & C_{24} & 0 & C_{26}\\
0 & 0 & C_{33} - M_w^2 & C_{34} & C_{35} & 0\\
0 & C_{42} & C_{43} & C_{44} - I_xW^2 & C_{56} & 0\\
C_{51} & 0 & C_{53} & C_{54} & C_{55} - I_yW^2 & 0\\
C_{61} & C_{62} & 0 & C_{64} & C_{65} & C_{66} - I_zW^2
\end{vmatrix} = 0$$

As an example consider the production machine of the mass m=1500 kg. Moments of inertia concerning principal axes are I_x=300 Nms^2,I_z =630 Nms^2. Frequency of rotation speed of the main shaft n=230 rpm, frequency of the disturbing force ω= $24s^{-1}$, its amplitude Q=5

$$C_x = C_y = 1011Nms^{-1}\,, C_z = 3 * 10^6 Nms^{-1}$$

Damping factors:

$$b_x = b_y = 2.5 * 10^6 Nms^{-1}\,, b_z = 4.3 * 10^3\ Nms^{-1}$$

Coordinates of supports:

$$z_b = 0.5m\,;\ l_{x1} = 1.0m\,;\ l_{x2} = 0.54m\,; l_{y1} = 1.8m\,; l_{y2} = 0.83m$$

The solution of the set of equations for the case of natural vibrations of the machine in a vertical direction found by means of MATLAB, shows that this natural frequency makes 71 s^{-1}, whereas the disturbing frequency, as was mentioned, equal 24 s^{-1}. Hence the machine runs in under resonance mode, in which natural frequency is far enough from a resonance.

The calculation, also by means of MATLAB, of the enforced vertical vibrations has shown, that their amplitude X_0 =1.2*10^{-4} m. Then the dimensionless dynamic factor is $K_d = {}^{cx_0}/_{Q} = 0.2$

Comparing its value with a unit, we are convinced, that the first one is much less, and one come to the conclusion, that the vibration insulation of the given machine meets the lead requirements.

V. Conclusion

1. The model offered herein of the production machine is installed on the vibration dampers and developed with the account of elastic and dissipative properties of the vibration dampers. The system of equation is permitted to evaluate the reduction of the machine vibrations caused by the unbalance movements of its members and thereby transmitting it onto the floor.
2. By means of the developed system of equations, it is proven that, taken as an example the production machine that runs far from a resonance and its vibration dampers effectively meet the requirements of the working environment.
3. Like any other engineering problem, noise control requires detailed work—first to identify the source, then to determine the most effective noise control technique, and then to determine the most cost-effective solution. However, like any other purchase, quality costs are required.
4. The development of innovative noise control treatments provides opportunities for applying basic physics and engineering procedures.

References

[1]. Bies, D.A., and C.H. Hansen(2003). Engineering Noise Control: Theory and Practice. London, U.K.: Taylor & Francis.Bruce, R.D., and A.S. Bommer (2004). The Occupational Noise Book. NIOSH Contract No. 0000037091. Cincinnati, Ohio: National Institute of Occupational Safety and Health.
[2]. Cyril M. Harris(1994). Noise Control in Buildings: A Practical Guide for Architects and Engineers
[3]. Franken A. Peter (1974) Community Noise Pollution, Industrial Pollution, Van Noistrand
[4]. Reinhold Company, New York,
[5]. Huang, X., Elliott, S. J., Brennan, M. J. (2003) Active isolation of a flexible structure from base vibration, Journal of Sound and Vibration, Vol.263,357-376.
[6]. Jensen, P., C.R. Jokel, and L.N. Miller(1978). Industrial Noise Control Manual. NIOSH Contract No. 210-76-0149. Cincinnati, Ohio: National Institute of Occupational Safety and Health.
[7]. Liu, Z. (2003). Acoustic Liner and a Fluid Pressurizing Device and Method Utilizing Same. U.S. Patent No. US 6,550,574 B2.
[8]. M. A. Alnawafleh, (2000)"Oscilation Analysis in Winding Mechanisms Actuated by Rotating Frictional Drum", proceeding of the 13th European Simulation Symposium, Simulation in Industry. Society for Computer Simulation International, Marseille, France.
[9]. M. A. Alnawafleh and Faisal M. Al-Ghathian,(2005) Dynamical Properties Analysis of Frictional Winding Mechanism with Bobbin Elastic Support. American Journal of Applied Sciences, Scientific Publication, Vol.2, No.4, 827-830.
[10]. O. N. Nizhibitsky, I. V. Ulyanov, A.S. churilin,(1997) Studies of the antivibration properties of structure materials made of waste in textile and light industries. In: Days of science, 97. Papers of technical sciences conf.- SPB, SPGUTD.
[11]. Rao, P.R (1995). Noise Pollution and Control, Encyclopedia of Environmental Pollution and
[12]. Control, Vol.-2, Environmedia Publications, India.
[13]. Randall F Barron and Barron F Barron (2002), Industrial Noise Control and Acoustics, Marcel Dekker, New York.
[14]. Shilin, X.,Siu, W., and Helen, Lai Wan Chan,(2008): Design optimization isolation system through minimization of vibration power flow. Journal of Structural Engineering and Mechanics, Vol. 28, No. 6, 677-694.
[15]. sStewart, William (February 2007). "Perforated metal systems sound absorbing surfaces". Construction Specifier.
[16]. V. A. Inovich, V. Ya. Onishchenko,(1990) Protection from vibrations in machinery construction, - m, Machinery construction.
[17]. W. E. Woodson, D. W. Conover(1966) Human engineering guide for equipment designers. Univ. of California press. Berkeley, Los Angeles.
[18]. Ya. I. korityssky (1974) Vibrations and noise in textile and light industries. M, light industries.