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Vibration in the building with machines on the floors

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At present, with the progress of science technology materials appeared construction materials with high mechanical properties (high-strength, high-damping and light..etc) and a sophisticated vibration isolation systems. It is also accompanied by the development of methods of analysis and structural design and prosperity aware software and electronics all contributed significantly to the in-depth study of vibration buildings more comprehensive than its predecessors and at the same time meet the requirements for the design of civil - architecture.

In case of machines founded on the floor there exist a dilemma of simultaneous solution of the active and passive vibration isolation problem. Complex approach to design of vibration isolation of machines should take into account transmission path of vibration in the building; from the source to vibration sensitive objects. Relationships between signals in the „dynamic machines-building-precision machines” system can be determined based on the model built of rigid finite elements. Based on the equations of motion of the system its dynamic structure, subsystems transfer functions and cross-couplings between subsystems were determined. The influence of these couplings on transfer function which describes relationship between displacements of the precision machine and forces generated in the dynamic machine was examined.

Keywords: vibration isolation, passive vibration isolation, active vibration isolation, transfer function , cross-coupling

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1. Introduction

Vibrations occur in the buildings as a result of several stimuli exposed structural elements. Of the most important and the most common of these effects are the work of industrial machines and what caused the mechanical or acoustic vibrations spread through the structure or rules.

There are two cases: - initial case be machines in the inside of the building and placed directly above the slab floors of the building, the second case shall be placed above the foundation, separate from the building foundation[1,10,12].

In the first case as a result of the movement of the machinery parts hesitant and variable force is made up in time to influence the rules of moving the machines and then to the floor slab and the columns, walls and rules. This force spread in all directions and take pictures or reciprocating random variable in time. Characteristic of these vibrations as a result of the ongoing work of the machines is that the frequency depends entirely on the frequency of such machines. And in the case of several machines with different speed, the image of the vibrations of the slab or the structure become blurry.

In the case of the machines set up separate foundations for the foundations of the building, the vibration is transmitted through the soil to the foundation and building the foundations thereof to all its parts. Next reason is that the vibrations caused by traffic and heavy transport vehicles or acts of roads and tunnels and others. Characteristic of these vibrations it depends on system and the structure and construction and materials as well as soil incorporation and less on the source of vibrations. There are many ways to avoid vibration. The best method for these are active and passive isolation (figure1).

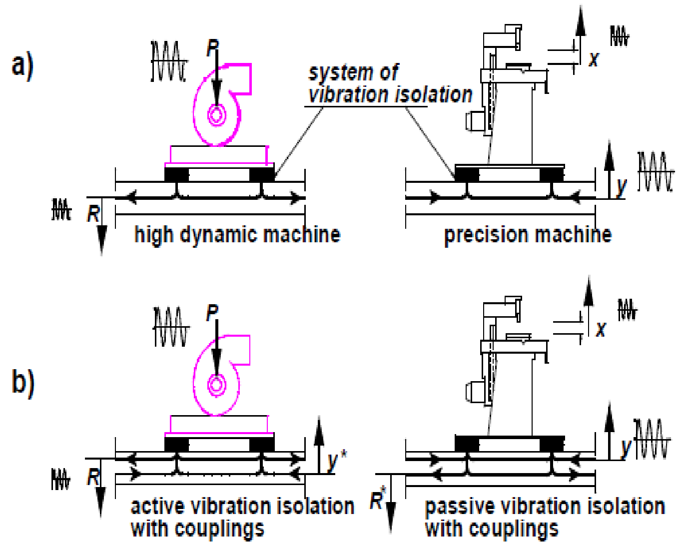


Figure 1 Coupling and uncoupling vibration isolation, a)active isolation, b)passive isolation

Despite the wide recognition of the literature and comprehensive publication on issues modeling and computational methods of mechanical systems, in the case of machines erected flexibly, is so far the issue of transmission of vibrations between machines supported on the floors of the same building has not been widely studied[1,12]. This fact stems from the fact that often the approach to the issue of isolation is omitted transmission path of vibrations (figure1a), presenting the model of the transmission of vibration in the vibration isolation systems as a model of one-way transmission (figure1b). Examine the transmission path of vibration and impact of feedback loop in the global system (Dynamic machines-Building-Precision machines) **DBP** (while applying vibration isolation of power and displacement), is crucial - the dynamic properties of the bearing structure of the building involved in the assessment of the dynamism of the whole system **DBP**

To minimize the bad effect of vibration is needed to study the characteristic of source-transmission path-receiver.

2. The effect of vibrations on the environment

2.1 The effect of vibrations in buildings

Evaluation of vibration damage in Structural Systems (frames, slabs, columns, beams, etc.) generally requires careful analysis and information on the position, as well as knowledge of the structural dynamic loads acting on them. To do this assessment is the process is not easy. And attic should identify the ways in which illustrate the effect of vibrations on the civil-rights and construction activity, which he is doing.

The effect of vibrations on the construction appear in the structural sections stresses caused by dynamic loads added to the static stresses caused by static loads, thereby causing material fatigue and structural failure and attic must reduce wave vibration amplitude to the threshold values until the total does not exceed the strains resistant material, taking into account the effect of fatigue stresses.

Ranges of structural response for various sources					
Vibration source	**Frequency range** **Hz**	**Amplitude range** $[\mu m \mu m]$	**Particle velocity range** $[mm/s\ mm/s]$	**Particle acceleration range** $[m/s^2 m/s^2]$	**Time characteristic**
Traffic Road, rail, ground-bone	1-100 1-80	1-200	0.2-50	0.02-1	
Blasting vibration Ground-bone	1-300	100-2500	0.2-100 0.2-500	0.02-50	T
Air over pressure	1-40	1-30	0.2-3	0.02-0.5	T
Pile driving Ground-bone	1-100	10-50	0.2-100 0.2-50	0.02-2	T
Machinery outside Ground-bone	1-100 1-300	10-1000	0.2-100 0.2-50	0.02-1	C/T
Machinery inside	1-300 1-1000	1-100	0.2-30	0.02-1	C/T

Human activities inside Impact Direct	0.1-30 0.1-100 .1-12	5-500 100-500 100-5000	0.2-20 0.2-20 0.2-5	0.02-0.2 0.02-5 0.02-0.2	T
Earthquakes	0.1-30	10-100000	0.2-400	0.02-20	T
Wind	0.1-10	10-100000	-	-	C
Acoustic	5-500	-	-	-	C/T

Table 1 Ranges of structural response for various sources

The effect of vibrations caused by the machines work on the resistance of concrete and metal structures usually less important if the reduction of vibrations allowed for the work of those machines, as well as people working and sensitive located near the machines. As for the mud buildings or mud brick the effects catastrophic.

Line	**Type of Structure**	**Foundation Frequency**			**Plane of Floor of Uppermost Storey**
		Less than 10 Hz	**10 to 50 Hz**	**50 to 100 Hz**	**Frequency Mixture**
1	Buildings used for commercial purposes, industrial buildings and buildings of similar design	20	20-40	40-50	40
2	Dwellings and buildings of similar design and/or use	5	5-15	15-20	15
3	Structures that, because of their sensitivity to vibration, do not correspond to those listed in lines 1 and 2 and are of great intrinsic value (eg buildings that are under a preservation order)	3	3-8	8-10	8

Table2 Limit velocity, for evaluating the effects of short-term vibration (DIN4150)[7]

A series of possibilities and measures in place (code) to assess the risks to the buildings of the vibrations resulting from the work of the machines are located. German code norm to assess the impact of the vibrations transmitted through the soil into the building is the largest value for the outcome of the speed measured or described in the rules-based.

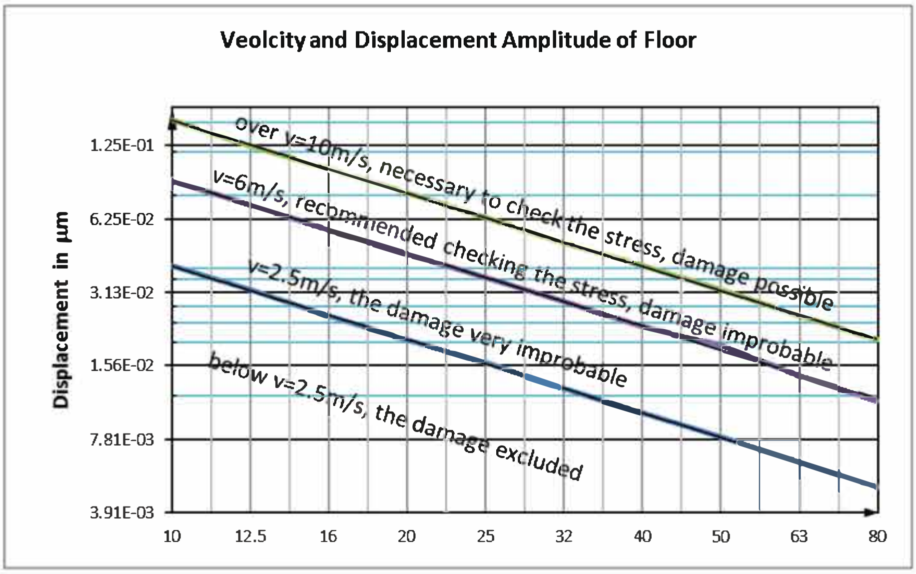


Figure2 limits for structural integrity for long-term continuous vibration, DIN4150-3:1999[8]

When velocity $V < 2\,mm/sV < 2\,mm/s$ no damage on the building and not even in the architectural components (glass, gypsum, ornaments .... etc). When the horizontal and continuous velocity $V < 5\,mm/s\,V < 5\,mm/s$ no damage on the building . The continued velocity effect is described in (figure 2).

2.2 The effect of vibrations on machines

The effect of vibrations on the industrial production explains as follows: elements that make the product are not unexpected movements in the event of their work and in some cases reflected adversely on the product and quality requirements.

RMS Velocity mm/s	0.18	0.28	0.45	0.71	1.12	1.8	2.8	4.5	7.1	11.2	18	28	45
Good, Small machine, up 15kW	Good			Allowable		Just tolerable		Not permissible					
Good, Medium Machines 15-75 kW, or up to 300kW on special foundations	Good				Allowable		Just tolerable		Not permissible				
Good, Large machines with rigid and heavy foundation whose natural frequency	Good					Allowable		Just tolerable		Not permissible			
Good, Large machines operating at speeds above foundation natural frequency (turbo machines)	Good						Allowable		Just tolerable		Not permissible		

Good (Zone A): The vibration o • f newly commissioned machines would normally fall within this zone.

• Allowable (Zone B): Machines with vibration within this zone are normally considered acceptable for unrestricted long-term operation.

• Just tolerable (Zone C): Machines with vibration within this zone are normally considered unsatisfactory for long-term continuous operation. Generally, the machine may be operated for a limited period in this condition until a suitable opportunity arises for remedial action.

• Not permissible (Zone D): Vibration values within this zone are normally considered to be of sufficient severity to cause damage to the machine.

These guidelines are especially useful where sensitive equipment is required to operate in the vicinity of highly vibrating equipment.

Table3 Machine vibration criterion chart (AS2625)[4].

The degree of sensitivity of this effect depends on the sensitivity of the production elements to external influences and this sensitivity depends on the requirements imposed on the element of quality and precision hand. And therefore these harmful movement of up to industrial machines by their rules and attic must reduce vibrations hyphenation rules for these machines.

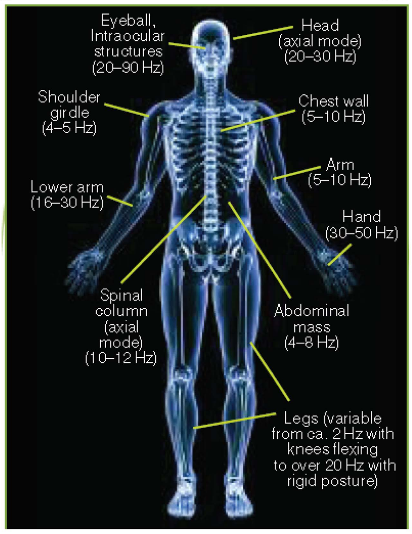
You can not put a definitive answer to the following question: Is velocity or the acceleration is certainly impressive? Here we had to set the standard sensitivity which is described in the (table4).
Germany code **Din 4150** and **AS2625** have been ranked machines according to velocity (table 3)

The degree of sensitivity	**The description**	**The machine name**	**Acceptable values for the speed of vibrations in the most damaging way** $[{}^{mm}/_{s}][{}^{mm}/_{s}]$
I	Very sensitive	Microscope optical instruments, sensitive lathes, electronic machinery, precision measurement of the few machines micrometers	0.1
II	Medium	Turning and grinding machines and lathes automatic accuracy tens of micrometers	1
III	Low sensitivity	Lathes and drill with regular precision, some types of sewing machines	3
IV	Insensitive	Electricity generators, electric shears, yarn, textile machines	5
V	Zero sensitivity	Ventilation machines and milling, molding concrete, machinery road works	12

Table 4 Approximate division of machines on sensitivity class to vibrations [14]

2.3 The effect of vibrations on human

Figure4 The tolerance limits of the body to vibration



some of the likely health effects:

• **spinal column disease and complaints** are perhaps the most common diseases associated with the long-term exposure to whole-body vibration, where the back is especially sensitive to the 4-12Hz vibration range (figure.4)

• **digestive system diseases** are often observed in persons exposed to whole-body vibration over a long period of time. This is associated with the resonance movement of the stomach at frequencies between 4 and 5Hz (figure 4)

• **and cardiovascular system effects** resulting from prolonged exposure to whole-body vibration at frequencies below 20Hz. These result in hyperventilation, increased heart rate, oxygen intake, pulmonary ventilation and respiratory rate.

The vibrational energy waves, much the same as noise, are transferred from the energy source – a hand tool or vehicle – into the body of the exposed operator. This is then transmitted through the body tissues, organs and skeletal systems of the individual before it is dampened and dissipated. Fortunately the human body can tolerate certain levels of vibrational energy but when exposed over a long period of time it begins to deteriorate and fail causing a disruption in the body's natural processes and systems. The health effects experienced by employees vary considerably and factors such as situation, age, lifestyle (smokers), posture, ergonomic design and resonance all have an influence on the ill health effects of the vibration exposure.

Each part of the human body has its own natural frequency of vibration. The extent to which the human body is affected depends on the vibration frequency to which it is exposed.

This resonant response to the vibration will cause symptoms ranging from simple motion sickness to severe discomfort, organ failure or tissue degeneration. The most pronounced and common effect is lower back pain. This can be linked to the vibration acting on the musculo-skeletal system of the body, causing the degeneration of the small cartilage (intervertebral) discs, allowing tissues and nerves to be strained and pinched leading to various back and neck problems. B. Zeller (1932, 1933, 1949) the experiments on human sensitivity to vibration and 8 degrees given the sensitivity of the human. The Rausch (1959) was given 7 degrees and described as in (table 5): -

Feeling vibrations extent, depends on the status of the body	1-0
Everyone feels Vibrating	2-1
Relatively unobtrusive movement street, can be felt inside the building	3-2
Felt like a man inside a moving vehicle and muffled, cumbersome machinery of the human person as a movement or inconvenience of traffic, little effect inside the building	4-3
Vibrations in the vehicle, a serious impact on buildings	5-4
Man can afford a very short time, the vibrations of heavy vehicles, damage to buildings, crash in traditional old buildings	6-5
Not borne human, unbalance myself, sea circulation disease	8-6

Table5 Scale perceptibility vibration by human[13]

Due to the action of physiologically Sorokin[14] proposes to accept any value as a meaningful acceleration (for frequency less than 10 Hz) and velocity (for frequency greater than 10 Hz) [table 6]

Description of the effect on human	$\ddot{u}\left[{}^{cm}/_{s^2}\right]$	$\dot{u}\left[{}^{mm}/_{s}\right]$
Imperceptible to humans	10	0.16
Felt poorly	40	0.64
Felt well	125	2

Harassed at work	400	6.4
Harmful for a long time to work	1000	16
Very harmful		>16

Table6 Characteristics of vibration influence on people [14]

Description Construction	**Time**	**KB value**	
		Continuous and intermittent >2h	**Casual and rare three daily oscillations**
Residential area	Day	0.2	4
	Night	0.15	0.15
Mixed residential neighborhood	Day	0.3	8
	Night	0.2	0.2
A service district And offices	Day	0.4	12
	Night	0.3	0.3
Industrial neighborhood	Day	0.6	12
	Night	0.4	0.4

Table7 Basic values KB to evaluate shocks in residential buildings [7]

With respect to human activity vibration limit described in (table 8)[2,3], according to building function limitation of velocity was described by [10] in (table 9).

The quality of human activity	$\ddot{u}\left[{}^{cm}/_{s^2}\right]$ **Acceleration affecting human**		
	1hour		24hour
All jobs if there is no additional requirements	236	63	24
Work next to complex machines that require concentration	118	32	12
Accounting Offices	37	10	5
High concentration of desired locations, for example, laboratory	5	5	5

Table8 Base acceleration for differing working environments[7]

The function of the building		$[{}^{mm}/_{s}][{}^{mm}/_{s}]$ Veolcity
Special care rooms in hospitals		0.4
Residential buildings	night	0.5
	day	0.8
Offices		1.5
Industrial building		3

Table 9 Recommendation of ISO 2631-74 is described by [10]

Way assessment of appreciability vibration by human is shown in (figure 5)

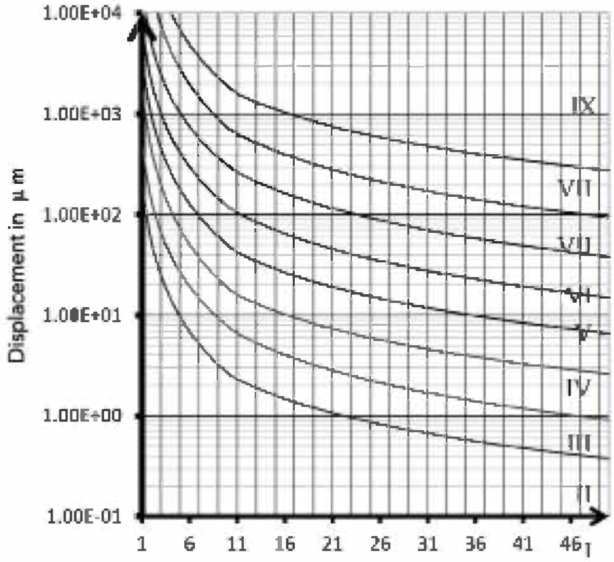


Figure 5 Degrees of perceptibility vibration by human [5], I-.undetectable, II- barely perceptible in silent, III- perceptible, IV- clearly perceptible, V- strongly perceptible, VI- very strongly perceptible, VII- very strongly perceptible and
Interferences, VIII- unbearable, IX- intolerable

3 System of high Dynamic-Building- Precision (DBP-System)

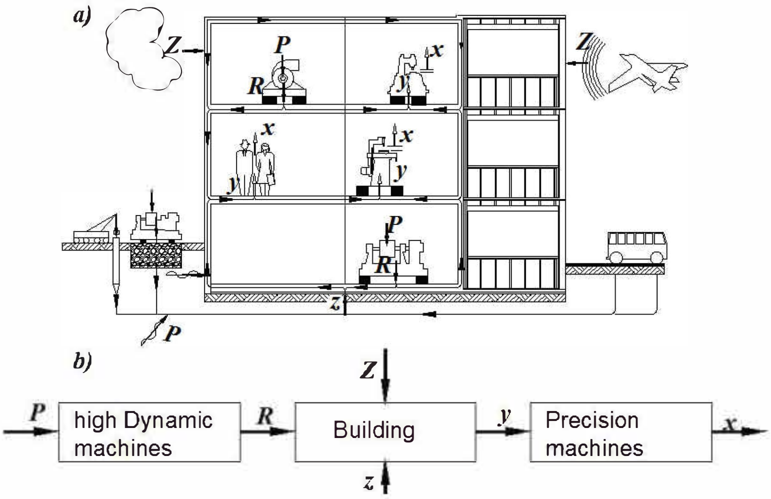


Figure 6 Building with machines,
a) loading on the building, b) scheme of signals

In terms of the dynamics of mechanical systems building with founded on floors machines and devices is a continuous system of mass-damper-spring. The layout of the work force power and kinematic , continuous or short-term, which may take the form of extortion determined or random (Figure 6a). On the supporting structure of the building vibrations are transmitted in the form of extortion kinematic ground **z**. Ground vibrations are caused for example by moving a heavy motor vehicles and rail vehicles, land seismic etc. In addition, the extraneous force **Z**, which are derived from a variable pressure air masses caused e.g. gale force winds or low flying aircraft. In both cases, the distortions usually arise randomly and are often difficult to accurately describe. Once inside the building there are machines and equipment with high dynamic, that when their jobs are created time-varying force P that the force strength **R** are transferred to the ceilings. **R** induced by vibration slabs are then transferred (transmitted) by the supporting structure of the building and are forcing kinematic **y**, which vibrates the **x** human and precision machinery. Relations, occurring throughout the system between the forces and displacements can be presented in the form of a flowchart diagram (Figure.6b). On the basis of the general relationship can be formulated

$$\boldsymbol{x} = \boldsymbol{W}_{xy}(\boldsymbol{W}_{yR}\boldsymbol{W}_{RP}\boldsymbol{P} + \boldsymbol{W}_{yZ}\boldsymbol{Z} + \boldsymbol{W}_{yz}\boldsymbol{z}) \qquad (1)$$

where W_{xy}, W_{yR}, W_{RP}, W_{yZ}, W_{yz} multidimensional transfer functions, which allow to determine the response of the system to enforce ***P, Z, z***. Transfer functions W_{xy}, W_{yR}, W_{RP}, W_{yZ}, W_{yz} may be determine analytically based on a model that with sufficient accuracy describes the structure and parameters of the real system, and which consists of the following subsystems:

- subsystem **D**, modeling high dynamic machines ,
- subsystem **B**, modeling building construction,
- subsystem **P**, which contains precision machines,
- subsystems **DB**, **BD**, **BP** and **PB**, which describe interactions between subsystems D, B, P.

Dynamic properties of real **DBP** system can be described on the basis of discrete model. Motion of **DBP** system built of rigid finite elements (**RFE**) [Kruszewski et al. 1975] is described by matrix equation

On the basis of experience gained during the calculation of isolation machine [1] assumes that modeling of **DBP** can be successfully applied the method of rigid finite element (Rigid Finite Element) **RFE**-Method [11]. Rigid finite elements are obtained by natural (for construction), and conceived of the constant division non-deformable elements (figure.7). The system $\mathbf{X}_{R1}$, $\mathbf{Y}_{R2}$, $\mathbf{Z}_{R3}$, which coincides with the main axis of inertia **RFE** r, describes a generalized displacement q and force p_r

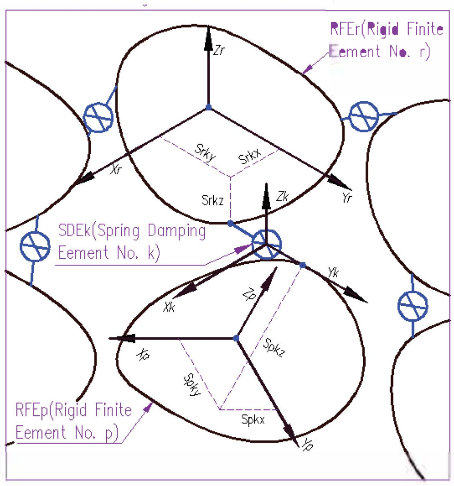


Figure 7 Discretization of Solid body by RFE

$$\boldsymbol{q}_r = col\{q_{ri}\}\,;\ \boldsymbol{p}_r = col\{p_{ri}\}\ for\ i=1,2,...,6 \qquad (2)$$

Inertia **RFE** *r* describes the coefficient matrix of inertia

$$\boldsymbol{M}_r = diag[m_{ri}]\,; for\ i = 1,2,...,6 \qquad (3)$$

Rigid finite elements are interconnected and the mainstay of linear elements of elastic-damping - **SDE** (Spring-Damped Element). Properties **SDE** $_k$ connecting **RFE** $_r$ and **RFE** $_p$ describe matrices damping factors $\mathbf{B_k}$ and $\mathbf{C_k}$ stiffness

$$\boldsymbol{B}_k = diag[b_{ki}]\,;\ \boldsymbol{C}_k = diag[c_{ki}] \qquad (4)$$

Location **SDE**$_k$ terms of **RFE**$_r$ on the array

$$\boldsymbol{F}_{rk} = \boldsymbol{\Theta}_{rk}\ \boldsymbol{S}_{rk}$$

(5)

where

$$\boldsymbol{\Theta}_{rk} = diag\left[[cos\varphi_{rkij}],[cos\varphi_{rkij}]\right] \qquad for\ ij = 1,2,3$$

(6)

$$\boldsymbol{S}_{rk} = \begin{bmatrix} \boldsymbol{I} & \boldsymbol{S}^*_{rk} \\ 0 & \boldsymbol{I} \end{bmatrix} \qquad \mathrm{S}^*_{rk} = \begin{bmatrix} 0 & s_{rkz} & -s_{rky} \\ -s_{rkz} & 0 & s_{rkx} \\ s_{rky} & -s_{rkx} & 0 \end{bmatrix} \qquad (7)$$

4 The equations of motion of the DBP

The movement of the **DBP** modeled based on the finite element method of the rigid describes the matrix equation

$$\mathrm{M}\ddot{\mathrm{q}} + \mathrm{L}\dot{\mathrm{q}} + \mathrm{Kq} = \mathrm{p}$$

(8)

where

$$\mathrm{M} = diag[\mathrm{M}_r] \qquad for\ \ r = 1,2,...,n_{FSE} \qquad (9)$$

It is the matrix of inertia (**RFE** - The number of n_{RFE}). Damping matrix **L** is composed of blocks which is determined based on the relationship

$$\mathrm{L}_{rr}=\sum_{k=1}^{n}\mathrm{F}_{k}^{T}\mathrm{B}_{k}\mathrm{F}_{k}\qquad\qquad \mathrm{L}_{rp}=-\sum_{k=1}^{n}\mathrm{F}_{k}^{T}\mathrm{B}_{k}\mathrm{F}_{pk}$$

$$\mathrm{L}_{pp}=\sum_{k=1}^{n}\mathrm{F}_{pk}^{T}\mathrm{B}_{k}\mathrm{F}_{pk}\qquad\qquad \mathrm{L}_{pr}=\mathrm{L}_{rp}^{T} \tag{10}$$

With analogous according to (10) determines blocks stiffness matrix **K** vectors generalized displacements q and p generalized forces of the entire system built with the vector (2), ie.

$$\mathrm{q}=col\{\mathrm{q}_r\};\quad \mathrm{p}=col\{\mathrm{p}_r\}\qquad for\ \ r=1,2,...,n_{FSE} \tag{11}$$

The equation of motion (8) of the **DBP** after a division of the subsystems **D**, **B**, **P**, **DB**, **BP**, after the **G** sub-system associated with a vehicle of a known traffic takes the form

$$\begin{bmatrix}\mathrm{M}_D & & \\ & \mathrm{M}_B & \\ & & \mathrm{M}_P\end{bmatrix}\begin{bmatrix}\ddot{\mathrm{q}}_D\\ \ddot{\mathrm{q}}_B\\ \ddot{\mathrm{q}}_P\end{bmatrix}+\begin{bmatrix}\mathrm{L}_D+\mathrm{L}_{DB} & -\mathrm{L}_{DB}^{*} & \\ -\mathrm{L}_{DB}^{*} & \mathrm{L}_B+\mathrm{L}_{GB}+\mathrm{L}_{DB}+\mathrm{L}_{BP} & -\mathrm{L}_{BP}^{*}\\ & -\mathrm{L}_{B}^{*} & \mathrm{L}_P+\mathrm{L}_B\end{bmatrix}\begin{bmatrix}\dot{\mathrm{q}}_D\\ \dot{\mathrm{q}}_B\\ \dot{\mathrm{q}}_P\end{bmatrix}+$$

$$+\begin{bmatrix}\mathrm{K}_D+\mathrm{K}_{DB} & -\mathrm{K}_{DB}^{*} & \\ -\mathrm{K}_{DB}^{*} & \mathrm{K}_B+\mathrm{K}_{GB}+\mathrm{K}_{DB}+\mathrm{K}_{BP} & -\mathrm{K}_{BP}^{*}\\ & -\mathrm{K}_{B}^{*} & \mathrm{K}_P+\mathrm{K}_B\end{bmatrix}\begin{bmatrix}\mathrm{q}_D\\ \mathrm{q}_B\\ \mathrm{q}_P\end{bmatrix}=\begin{bmatrix}\mathrm{p}_D\\ \mathrm{p}_B+\left(\mathrm{L}_{GB}^{*}\,\dot{\mathrm{q}}_G+\mathrm{K}_{GB}^{*}\,\mathrm{q}_G\right)\\ \mathrm{p}P\end{bmatrix} \tag{12}$$

After the multiplication unit equation (12) is replaced by the system of three matrix equations

$$\mathrm{M}_D\ddot{\mathrm{q}}_D+\left(\mathrm{L}_D+\mathrm{L}_{DB}\right)\dot{\mathrm{q}}_D+\left(\mathrm{K}_D+\mathrm{K}_{DB}\right)\mathrm{q}_D=\mathrm{p}_D+\left(\mathrm{L}_{DB}^{*}\,\dot{\mathrm{q}}_B+\mathrm{K}_{DB}^{*}\,\mathrm{q}_B\right)$$

$$\mathrm{M}_B\ddot{\mathrm{q}}_B+\left(\mathrm{L}_B+\mathrm{L}_{GB}+\mathrm{L}_{DB}+\mathrm{L}_{BP}\right)\dot{\mathrm{q}}_B+\left(\mathrm{K}_B+\mathrm{K}_{GB}+\mathrm{K}_{DB}+\mathrm{K}_{BP}\right)\mathrm{q}_B=$$

$$=p_B+\left(\mathrm{L}_{GB}^{*}\,\dot{\mathrm{q}}_G+\mathrm{K}_{GB}^{*}\,\mathrm{q}_G\right)+\left(\mathrm{L}_{DB}^{*}\,\dot{\mathrm{q}}_D+\mathrm{K}_{DB}^{*}\,\mathrm{q}_D\right)+\left(\mathrm{L}_{BP}^{*}\,\dot{\mathrm{q}}_P+\mathrm{K}_{BP}^{*}\,\mathrm{q}_P\right) \tag{13}$$

$$\mathrm{M}_P\ddot{\mathrm{q}}_P+\left(\mathrm{L}_P+\mathrm{L}_B\right)\dot{\mathrm{q}}_P+\left(\mathrm{K}_P+\mathrm{K}_B\right)\mathrm{q}_P=\mathrm{p}_P+\left(\mathrm{L}_B^{*}\,\dot{\mathrm{q}}_B+\mathrm{K}_B^{*}\,\mathrm{q}_B\right)$$

The first and third equation in the system of equations (13) describe the motion subsystems **D** and **P** with extortion forces $\mathbf{P_D}$, $\mathbf{P_P}$ and kinematic $\mathbf{q_B}$, coming from the sub-system **B**. The second equation describes the motion sub-system **B** at the impact on him of extortion external force $\mathrm{P}_{\mathbf{B}}$ and kinematic $\mathbf{q_G}$, and internal extortion strength, which were caused by vibration $\mathbf{q_B}$ and $\mathbf{q_P}$ subsystems **D** and **P**.

As a result, the Laplace transform of equations (13) and after substituting s = iω obtained by the equation of motion of the D-B-P frequency-domain

$$\left[\left(\mathrm{K}_D-\omega^2\,\mathrm{M}_D\right)+i\omega\,\mathrm{L}_D\right]\mathrm{q}_D(i\omega)+\left[\mathrm{K}_{\boldsymbol{B}}+i\omega\,\mathrm{L}_{\boldsymbol{B}}\right]\mathrm{q}_D(i\omega)=\mathrm{p}_D(i\omega)+\left[\mathrm{K}^*_{\boldsymbol{B}}+i\omega\,\mathrm{L}^*_{\boldsymbol{B}}\right]\mathrm{q}_B(i\omega)$$

$$\begin{aligned}&\left[\left(\left(\mathrm{K}_B+\mathrm{K}_{\boldsymbol{B}}\right)-\omega^2\mathrm{M}_B\right)+i\omega\left(\mathrm{L}_B+\mathrm{L}_{\boldsymbol{B}}\right)\right]\mathrm{q}_B(i\omega)+\left[\mathrm{K}_{\boldsymbol{B}}+i\omega\mathrm{L}_{\boldsymbol{B}}\right]\mathrm{q}_D(i\omega)+\left[\mathrm{K}_{\boldsymbol{B}}+i\omega\mathrm{L}_{\boldsymbol{B}}\right]\mathrm{q}_B(i\omega)=\\&=\mathrm{p}_B(i\omega)+\left[\mathrm{K}^*_{\boldsymbol{B}}+i\omega\mathrm{L}^*_{\boldsymbol{B}}\right]\mathrm{q}_D(i\omega)+\left[\mathrm{K}^*_{\boldsymbol{B}}+i\omega\mathrm{L}^*_{\boldsymbol{B}}\right]\mathrm{q}_D(i\omega)+\left[\mathrm{K}^*_{\boldsymbol{B}}+i\omega\mathrm{L}^*_{\boldsymbol{B}}\right]\mathrm{q}_P(i\omega)\end{aligned}\tag{14}$$

$$\left[\left(\mathrm{K}_P-\omega^2\,\mathrm{M}_P\right)+i\omega\,\mathrm{L}_P\right]\mathrm{q}_P(i\omega)+\left[\mathrm{K}_B+i\omega\,\mathrm{L}_B\right]\mathrm{q}_P(i\omega)=\mathrm{p}_P(i\omega)+\left[\mathrm{K}^*_B+i\omega\,\mathrm{L}^*_B\right]\mathrm{q}_B(i\omega)$$

The system of equations (14) can be represented as a flowchart. To this end, a multidimensional function is defined transition subsystems **D**, **B** and **P**, which describe the relations between the input $\mathbf{P}^*_{\alpha}$ and the output $\mathbf{q}_{\alpha}$ (α = D, B, P), i.e.

$$\mathrm{q}_D(i\omega)=\left[\left(\mathrm{K}_D-\omega^2\,\mathrm{M}_D\right)+i\omega\,\mathrm{L}_D\right]^{-1}\mathrm{p}^*_D(i\omega)$$

$$\mathrm{q}_B(i\omega)=\left[\left(\left(\mathrm{K}_B+\mathrm{K}_{\boldsymbol{B}}\right)-\omega^2\,\mathrm{M}_B\right)+i\omega\left(\mathrm{L}_B+\mathrm{L}_{\boldsymbol{B}}\right)\right]^{-1}\mathrm{p}^*_B(i\omega)\tag{15}$$

$$\mathrm{q}_P(i\omega)=\left[\left(\mathrm{K}_P-\omega^2\,\mathrm{M}_P\right)+i\omega\,\mathrm{L}_P\right]^{-1}\mathrm{p}^*_P(i\omega)$$

The remaining blocks in the matrix equation (14) describe the internal and external feedback system **DBP**, the structure of which is shown in Figure 8.

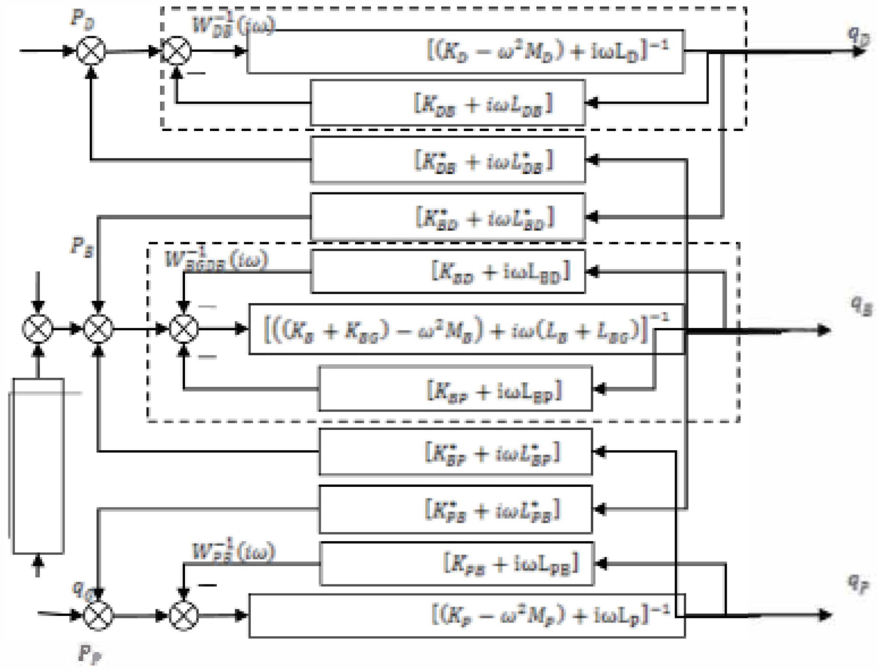


Figure 8 Control block diagram for DBP System

Such a representation of the relationship between the signals allows a better understanding of the phenomena occurring in the system and processes. It should be emphasized that, of the transfer function (15), the other one, can be calculated using the equations of motion subsystem **B**. Other i.e. The first and third are merely lip of the transfer function of the free. Due to the lack of constraints imposed on these subsystems, there is a peculiarity of the complex stiffness matrix. The inclusion of internal stresses in subsystems **D** and **P**, equivalent to joining the bonds defining the combination of these subsystems with a subsystem **B**, leading to define a transfer function, $W_{B}^{-1}(i\omega)$, $W_{BGDP}^{-1}(i\omega)$ and $W_{P}^{-1}(i\omega)$. The transition functions are marked in Figure 9 locks limited by a broken line.

In writing structural in Figure 9 shows all of the feedback system **DBP**. Generally when designing vibration isolation, some of them ignored, as the feedback weak. The active vibration isolation does not include extortion strength of the substrate, which describes the expression

$\left[\mathrm{K}_{B}^{*}+i\omega\,\mathrm{L}_{B}^{*}\right]\mathrm{q}_{B}(i\omega)$. Similarly, passive vibration isolation does not include extortion strength coming from machine vibrations, which describes the expression $\left[\mathrm{K}_{P}^{*}+i\omega\,\mathrm{L}_{B}^{*}\right]\mathrm{q}_{P}(i\omega)$.

If you omit these feedbacks will be introduced and a new record transfer function subsystems P_Df the bonds, the structural diagram of the system can be presented in a simplified form as in figure 9.

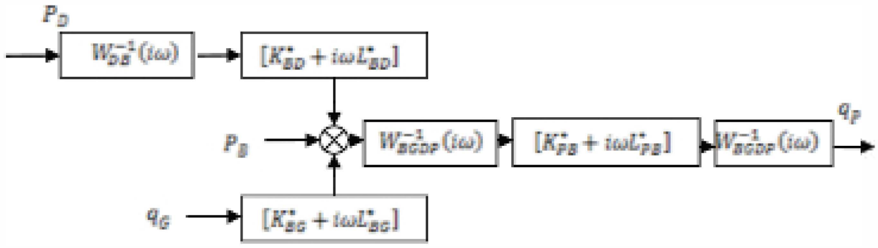


Figure 9 Control block diagram without feedback

On the basis of such a simplified scheme can be determined transition functions between the signals. It is also a record of mathematical transfer function shown in general figure.1b.

5 Calculation of the transfer function between machines founded on the floors

The assessment of the coupling significance in the **DBP** system was conducted on the example of isolation of the fan and grinding machine located in the two-story building. Building was substituted by the model built of 114 rigid finite elements of 684 **DOF** (figure 10). The fan was placed on the second floor and described by the one body model of 6 **DOF**. Plane grinding machine was founded on the first floor. The dynamic properties of the grinding machine were described by the model built of 4 rigid finite elements which enables analysis of relative vibrations of the grinding wheel and workpiece [Witek 1992]. To determine the **DBP** system transfer function it was assumed that the centrifugal force generated in the fan enforces vibrations within the frequency range up to 200Hz

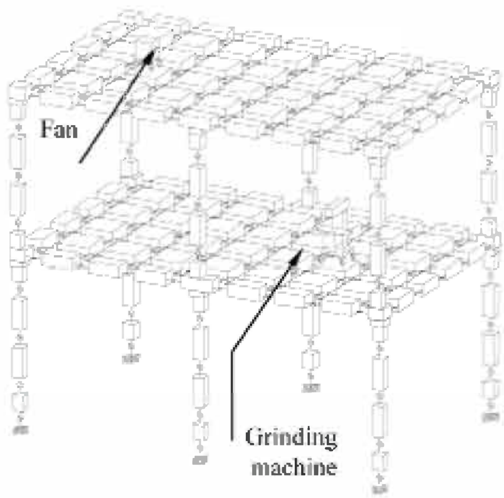


Figure 10 RFE model of DBP system

Hz. From such a determined **DBP** system model it's transfer function was realized

$$W_{xQ}(i\omega) = \frac{x(i\omega)}{Q(i\omega)} \tag{9}$$

where: $x(i)$ – vertical displacements of the grinding wheel, $Q(i\omega)$ – centrifugal force generated in the fan.

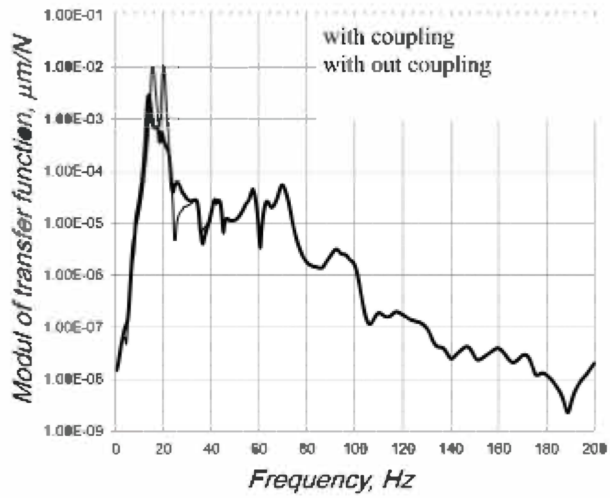


Figure 11 Transfer Function DBP system, with coupling and without coupling

Computations of the transfer function were conducted according to the block diagram, which means that all couplings were considered (Figure 8) and two feedbacks were neglected (figure 9). Results are presented in Figure 10.
Within the frequency range up to 60 Hz significant differences of transfer function determined on the base of uncoupled model were observed. It means that in the lower frequencies range transfer function should be determined on the base of the model which considers all couplings.

6 Summary

The paper was shown that the complex modeling of the dynamic system formed by building up the necessary machines method can be divided into subsystems with couplings defined there between. This approach to modeling enables you to effectively analyze and modify the system through proper analysis and modification of its subsystems. A breakdown of the system on subsystems will among other things, analyze the sensitivity of partial criteria of isolation to change the selected parameters (i.e. To minimize the partial transfer function), which should ensure rapid achievement of a global order of isolation, which is to minimize the vibration of machinery and precision equipment.

It was proved that a priori assumptions which consisted in neglecting couplings existing in the real **DBP** system can lead to the transfer function determination charged with significant errors. As it results from the computational example significant differences of the transfer function can occur in the low frequency band which is very important in the evaluation of efficiency of vibration isolation of machine.

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