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भारतीय मानक

मशीन नीवों के लिए कम्पन-विलगन संबंधी मार्गदर्शी सिद्धान्त

Indian Standard

VIBRATION ISOLATION FOR MACHINE FOUNDATIONS — GUIDELINES

UDC 624·15 : 628·517·4

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Price Group 3

Foundation Engineering Sectional Committee, CED 43

FOREWORD

This Indian Standard was adopted by the Bureau of Indian Standards, after the draft finalized by the Foundation Engineering Sectional Committee, had been approved by the Civil Engineering Division Council.

This code is meant to provide necessary information and assistance in the choice of vibration isolators in machine foundation so as to ensure a smooth working of the machinery supported by it as well as to reduce the transmitted vibration into the surrounding environment. It does not imply, however, that use of external isolators is obligatory in a machine foundation.

For the purpose of deciding whether a particular requirement of this standard is complied with, the final value, observed or calculated, expressing the result of a test or analysis, shall be rounded off in accordance with IS 2 : 1960 'Rules for rounding off numerical values (*revised*)'. The number of significant places retained in the rounded off value should be the same as that of the specified value in this standard.

Indian Standard

VIBRATION ISOLATION FOR MACHINE FOUNDATIONS — GUIDELINES

1 SCOPE

1.1 This standard lays down general guidelines for vibration isolation for machine foundation.

2 REFERENCE

2.1 IS 5249 : 1991 'Method of test for determination of dynamic properties of soil (*second revision*)' is a necesary adjunct to this standard.

3 TERMINOLOGY

3.0 For the purpose of this standard, the following definitions shall apply.

3.1 Active Isolation

Reduction of the periodic or shock type of forces transmitted by a machine installation into the surroundings by the working of the machinery itself.

3.2 Passive Isolation

Isolation of a sensitive installation against ambient vibrations emanating from external sources and already existing in the vicinity.

3.3 Transmissibility

The ratio of the peak amplitude of the transmitted force to the applied dynamic force in the case of active isolation.

The ratio of the amplitude of the sensitive equipment to that prevailing at the base in the case of passive isolation.

3.4 Frequency Ratio

Ratio of operating frequency of the machine to the natural frequency of an elastic system.

3.5 Damping Ratio

Ratio of the damping present in a system to that of critical damping for the same system.

4 TYPES OF VIBRATION ISOLATORS

Table 1 gives an approximate range of natural frequencies that can be obtained with different types of vibration isolators. For an idealized single degree freedom system, the natural frequency 'f_n' may be obtained using the relation

$$f_n = \frac{1}{2\pi}\sqrt{\frac{g}{\delta}}$$

where g is the acceleration due to gravity and δ, is the static deflection suffered under the influence of the supported weight of the system in the direction of vibration considered.

For effective vibration isolation, the natural frequency shall preferably be less than 0·4 f_m under harmonic excitation where f_m is the frequency of operation of the machine.

Table 1 Effective Frequency Range for Vibration Isolators

Sl No.	Type	Range of Natural Frequencies (f_m in Hz)
1)	Metal helicals	2-10
2)	Rubber	5-30
3)	Cork	25-60
4)	Air (pneumatic type)	0·5-3·0

5 DYNAMIC PROPERTIES OF CERTAIN MATERIALS USED IN VIBRATION ISOLATION

5.1 Coil Springs

The vertical stiffness of closely coiled helical springs is given by

$$k_v = \frac{1}{n} \quad \frac{Gd^4}{8D^3}$$

where

k_v = vertical stiffness,

G = shear modulus of the spring material,

d = diameter of wire,

D = diameter of coil, and

n = number of coils.

The damping in steel may be taken in the range of 0 to 0·5 percent of critical unless more appropriate values based on actual test data are available. The horizontal stiffness of the spring is given by

$$K_h = k_v \cdot R$$

where

$$R = \left[\frac{1\cdot061\,3}{\nu_0}\ \varepsilon \tan (0\cdot942\,2\ \nu_0\varepsilon) - \left(\frac{\beta_0}{\nu_0} - 1\right)\right]$$

$\varepsilon = [(\beta_0/\nu_0) - 0\cdot614\,2\ ]^{1/2}$

$\nu_0 = \delta_v/D$

$p_0 = h/D$

δ_v = vertical deformation, and

h = height of spring.

5.2 Rubber Springs

Rubber springs can be used either under compression or shear. The stiffness of the rubber pad under axial compression can be obtained from the relation

$$\frac{1}{k_c} = \frac{t}{A}\left[\frac{1 \cdot 0}{E(1+2\alpha A^2_r)} + \frac{1}{B}\right]$$

where

k_c = vertical stiffness under axial compression,

t = thickness of the rubber pad,

A = bearing area over the pad,

A_r = area ratio defined as the ratio of the force free surface area to the bearing area,

E, B and α = constants given in Table 2.

Table 2 Properties of Natural Rubber Compounds

Shore Hardness (S°)	Young's Modulus E N/sq. mm	Shear Modulus G N/sq. mm	Bulk Modulus B N/sq. mm	α
40	1·53	0·46	1 019·4	0·85
45	1·84	0·55	1 019·4	0·80
50	2·24	0·65	1 019.4	0·73
55	3·31	0·83	1 111·1	0·64
60	4·54	1·08	1 172.2	0·57
65	5·96	1·40	1 233·4	0·54
70	7·49	1·76	1 294·6	0·53

The horizontal stiffness is given by

$$k_h = \left(\frac{GA}{t}\right)$$

where

k_h = horizontal stiffness,

A = bearing area, and

t = thickness.

The damping ratio in rubber generally varies between 2% and 10%. A figure of 5% is recommended for design practice for preliminary designs.

5.3 Other Materials

The stiffness of other elastic materials such as cork, felt, etc, which are also available in the form of pads can be obtained using the relations

$$k_v = (EA/t)$$

$$\text{and } k_h = (GA/t)$$

where

E = Young's modulus of the material,

G = shear modulus of the material,

A = bearing area,

t = thickness, and

k_v, k_h = vertical and horizontal stiffness, respectively.

The dynamic modulus of cork shows a high degree of scatter and generally lies in the range 10 to 40 N/mm². The damping ratio lies in the range of 2·5 to 7·5%. A figure of 6% is recommended in design practice for preliminary designs.

Felt has a Young's modulus of around 80 N/mm² and has a damping factor nearly same as cork.

6 DESIGN OF VIBRATION ISOLATORS

6.1 Under Steady State Loads

The term transmissibility (T) under a steady state excitation for an idealized single degree freedom system may be written as

$$T = \frac{(1 + 4\,\eta^2\,\xi^2)^{1/2}}{[(1 - \eta^2)^2 + 4\eta^2\,\xi^2]^{1/2}}$$

where

T = transmissibility,

η = the frequency ratio (ω/ω_n)

ω = operating frequency,

ω_n = natural frequency, and

ξ = damping ratio.

Figure 1 shows a plot using which the static deflection required for the supported weight of the system to obtain any given level of transmissibility in the desired direction for various disturbing frequencies of the machinery may be derived. The ragion below the shaded line indicates amplification while that above this line suggests isolation. For effective isolation, the frequency ratio shall be greater than $\sqrt{2}$ (Fig. 2).

6.2 Shock Loading

The natural period shall be at least:

a) 6 times the duration for rectangular pulse,

b) 3·75 itmes the duration for the sinusoidal pulse, and

c) 3·00 times the duration for the triangular pulse to achieve transmissibility less than unity.

The variation of transmissibility in the case of an undamped system for different pulse-shapes

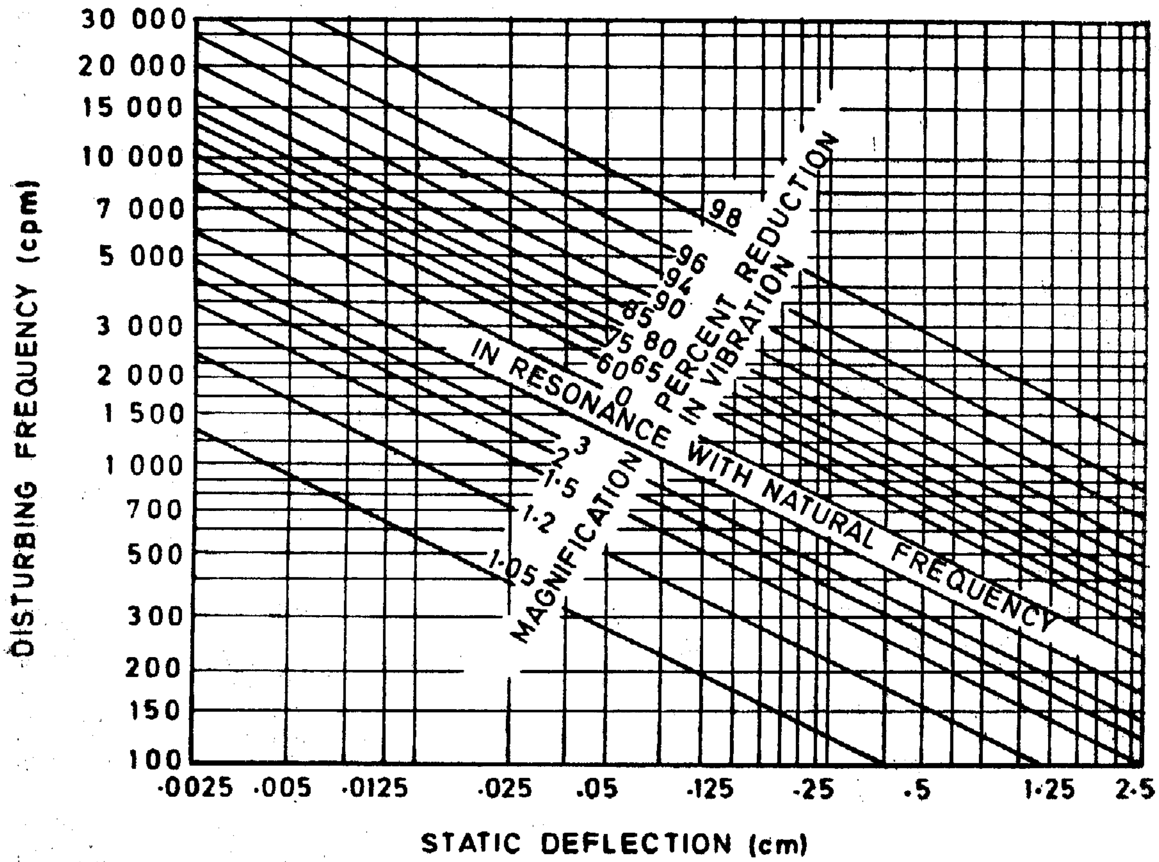


Fig. 1 Isolation Efficiency of Resiliently Mounted Systems

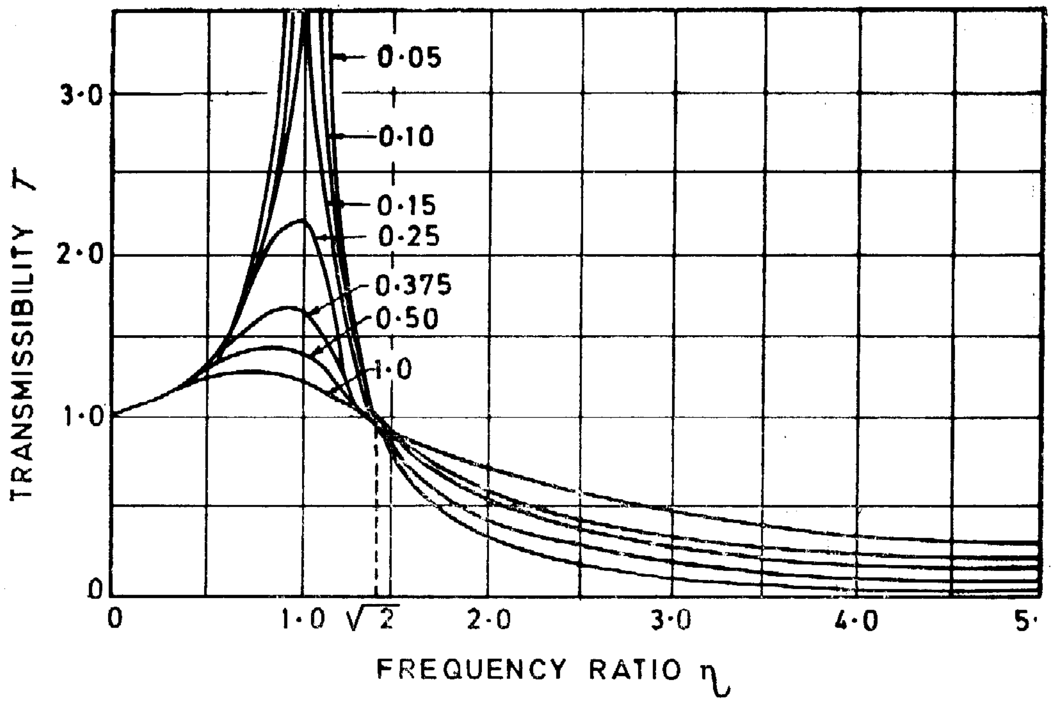


Fig. 2 Variation of Transmissibility (T) with Frequency Ratio (η) for Steady State Dynamic Loading

is shown in Fig. 3. The notation used in Fig. 3 are:

P_T = transmitted force,

P = peak force,

t = duration of the pulse, and

T_n = natural period.

7 OTHER DESIGN CONSIDERATIONS

7.1 Metal Springs

a) ***Strength*** — The shear stress in a closely coiled helical spring under axial loading can be obtained from the relation

$$\tau_v = \left(\frac{8\,\alpha_v\,PD}{\pi d^3} \right)$$

where

τ_v = shear stress,

P = applied load,

D = diameter of coil,

d = diameter of wire, and

$\alpha_v = 1 + 1{\cdot}25\ (d/D) + 0{\cdot}875\ (d/D)^2 + (d/D)^3$.

The shear stress under horizontal loading is given by

$$\tau_h = \left(\frac{8\alpha_h\,HD}{\pi d^3} \right)$$

where

τ_h = horizontal shear stress,

H = applied horizontal load, and

$\alpha_h = (\nu_0/u) + \beta_0 - \nu_0$.

b) *Stability* — To avoid instability of coiled springs, the axial deformation shall be limited to 0·5 h and the buckling stability factor '*S*', to be evaluated from the following expression, shall be greater than 1·5.

$$S = 1{\cdot}296 \left[\left\{ (\beta_0/\nu_0 - 1)^2 + \frac{4{\cdot}29}{\nu_0^2} \right\}^{1/2} - (\beta_0/\nu_0 - 1) \right]$$

The factors β_0 and ν_0 are defined in **5.1**.

7.2 Rubber Spring

a) *Allowable Bearing Pressure* — The allowable bearing pressure shall be specified by the manufacturer. For preliminary designs, however, linear variation in allowable bearing pressure between 0·8 N/mm² and 1·6 N/mm² may be assumed in the range of shore hardness values between 40 and 70 degrees.

b) *Allowable Shear Stress* — The allowable shear stress is also required to be specified by the manufacturer. As in the earlier case, a linear variation in allowable shear stress between 0·3 to 0·5 N/mm² may be assumed for preliminary designs for shore hardness values lying between 40 and 70 degrees.

c) From stability considerations, the thickness of the rubber pad shall be limited to one-fifth of its width.

d) Tests have shows that the dynamic characteristics of rubber pads exhibit a

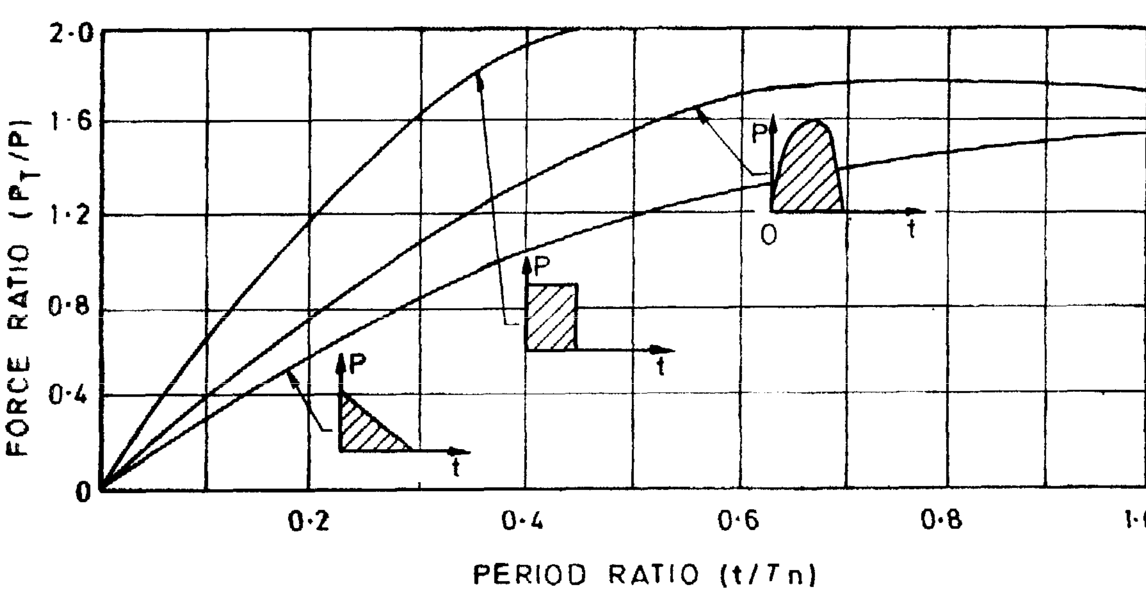


FIG. 3 VARIATION OF TRANSMISSIBILITY WITH PERIOD RATIO FOR PULSE LOADING

non-linear character. Further, the stiffness of the rubber pad depends on the level of the static stress and the amplitude of vibration (or dynamic strain). Laboratory tests in the form of steady state resonance tests are, therefore, recommended on randomly chosen product samples under the expected static stress and dynamic strain levels. This will provide the true picture of the dynamic stiffness and damping present in vibration isolators, that are being commercially marketed today.

e) Care shall be taken to ensure unrestrained free sides of the pad type isolators where used.

7.3 Cork Pads

a) *Bearing Pressure* — The allowable bearing pressure on cork pads usually varies between 1 and 4 kg/cm². The true value shall be ascertained from the manufacturers' recommendation based on tests.

b) Cork sheets lose their strength under compressive loads if the edges of the pads are left free. Hence, the side faces have to be enclosed in steel frames to prevent their lateral expansion.

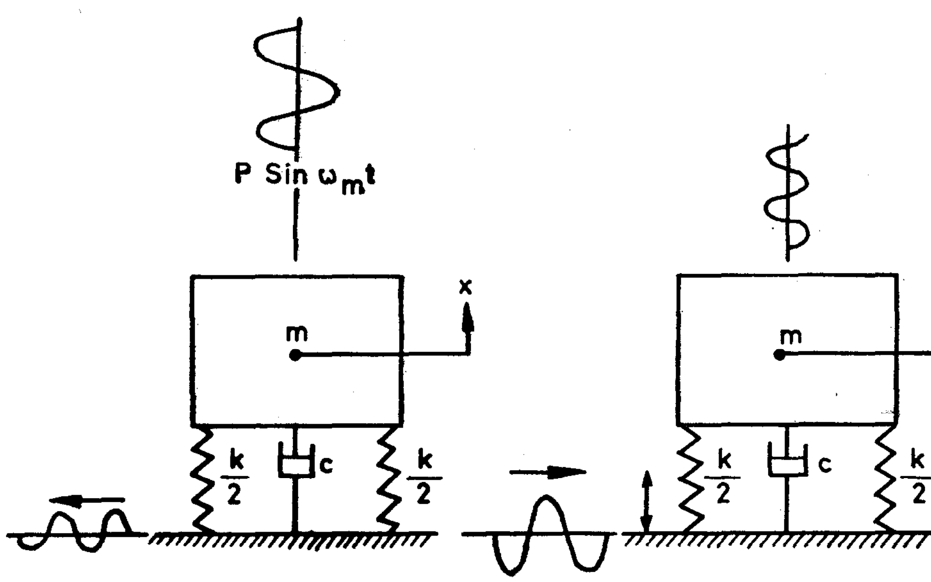
c) Contact with oil or water reduces the efficiency of cork pads and hence shall be treated with suitable preservatives before use.

d) The dynamic characteristics of cork pads show considerable scatter and non-linearity. Tests have shown that the thickness of the cork pad, the static stress level and the amplitude of vibration influence its dynamic properties. Besides, considerable creep deformation occurs under a given static stress level and this tends to increase the stiffness and reduce isolation efficiency. All the above factors are required to be considered in the experimental evaluation of the dynamic properties of cork pads before they are used in important machinery installations.

8 TRENCH ISOLATION

Trench isolation can be effectively used for active isolation in an industrial environment (Fig. 4).

For active isolation, the depth of the trench shall at least be 0·6 L, where L is the length of the Rayleigh wave which is nearly equal to the length of the shear wave (L_s). The latter is given by $[(G/\rho)^{1/2}/f]$ where G is shear modulus, ρ is the mass density of the soil, and f is frequency (Hz) of incoming wave. L is obtained from *in-situ* wave propagation tests as in IS 5249 : 1991.



(a) Active Type (b) Passive Type

FIG. 4 VIBRATION ISOLATION

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