Damping Characteristics of Timber Flooring Systems with Respect to Low-Frequency Vibration Modes

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Summary

The vibration serviceability of flooring structures has been of concern for decades. To date, design criteria classifying the dynamic behaviour of different types of timber flooring structures to an acceptable degree have not been well established. Parameters commonly used to investigate and assess vibrational performances of timber floors include natural frequencies, damping ratios, dynamic displacements, velocities, accelerations, mode shapes and static point load deflections. The damping ratio is also used for design purposes, but the value of damping ratio for design is controversial. This paper presents experimental evaluation of the damping ratios on full-scale timber flooring systems for different modes of vibration and details the variation in damping under different dead weights. The test results in this study highlight the need for further research in selecting appropriate values of damping parameters for design purposes.

1. Introduction

In timber constructions there has been an increasing trend of using engineered timber joists to build large-span flooring systems. While these systems usually provide satisfactory static performance, they can infringe serviceability due to undue vibration, creating discomfort to the occupants. The ongoing research programme at Napier University has focused on this serviceability aspect. Extensive experimental and analytical investigations have been conducted on a variety of floor configurations to assess their dynamic performances. Parameters investigated include natural frequencies, damping ratios, mode shapes and unit point load deflections. This paper focuses on the damping characteristics of the timber flooring systems.

Damping, often expressed by the damping ratio, plays an important role in research about floor vibrations as it influences the vibration amplitude of structures and is used as part of design criteria in standards and guides [1-3]. The amplitude damps out more quickly and occupants will become less sensitive to initial vibration velocities if the damping is increasing [3][4]. There is an uncertainty of the damping ratios chosen for design, e.g. 1% suggested in the Eurocode 5 (EC5) [1] and 2% in the UK National Annex to EC5 [2]. The choice of a damping ratio can largely influence the judgement of classifying flooring structures as satisfactory or unsatisfactory regarding their dynamic performances.

This paper presents a range of damping ratios corresponding to the first three vibration bending modes of 24 two-side supported full-scale timber flooring systems with different configurations including three floor sizes, tested in laboratory conditions. Furthermore, the damping ratios for varied dead weight on a flooring system are illustrated in detail. Advanced analysis technique was used for estimating damping ratios (see section 3.1).

2. Literature Review

In 1966 Lenzen published his research work on the vibrational behaviour of composite steel joist - concrete slab floors and the human sensitivity to it [4]. He observed that the main influencing factor on human beings from vibrations was the damping and that the presence of occupants increased the damping of the flooring system. However, other kinds of loads would not raise the damping values. The damping of a floor was strongly decreased when being loaded with concrete cylinders.

Rainer and Pernica investigated the effect of two impulse excitation methods and two continuous ones on the modal damping ratios of a large-scale floor fabricated with open-web steel joists and a concrete slab [5]. They further confirmed that humans on a flooring structure add damping to the system and thus that damping ratios obtained from heel-drop tests usually were higher than those measured on bare floors. In addition, the location of the person performing the heel-drop test affected the damping ratio.

Ohlsson explained the effect of dead weight on damping [6]. The damping ratio ζ is defined as the ratio of the damping coefficient c to the critical damping c_{cr} for a system, $\zeta = c/c_{cr}$, so c_{cr} is proportional to $\sqrt{k \cdot m}$ and thus ζ is consequently reduced as a result of added dead weight. He suggested a damping ratio of 1% for the design of normal light-weight floors [3]. It could be reduced to 0.8% for floors of large span or large weight (>150 kg/m^2). The vibration serviceability design criteria in the EC5 are based on the work of Ohlsson.

Smith and Chui recommended a value of 3% to be used as damping ratio for light-weight floors [7]. They found a material damping ratio of ~1% for solid timber (Canadian White Spruce) and conclude that the total damping of a floor would increase with increasing mass, justifying the use of their proposed damping ratio [8]. By summarising varied terms and expressions for damping, they described the dependence of damping on testing and analysing methods and suggested using the viscous damping ratio to express damping magnitude of timber structures.

In general, the damping is often referred to the first mode of vibration, but higher modes may also be influential on the disturbance sensed by occupants [9]. The most annoying vibration mode could also correspond to one of the higher natural frequencies, dependent on the location of the disturbed person on the floor.

Damping is found to be an important factor influencing the response of humans to floor vibrations. However, the determination of the damping is complex, depending on test procedures and data analyses. Humans on floors increase the damping ratio. If another type of dead weight is increased, a negative effect on damping can often be observed. The damping ratios suggested for use in serviceability design of timber flooring structures vary considerably. Inspection of damping ratios of higher modes is also of interest since these higher modes can contribute to annoying dynamic performances.

3. Experimental Investigations and Signal Processing

The experimental work carried out for the investigations in this study comprised dynamic tests on 24 composite timber flooring systems. A floor set-up is shown in Fig.1. All floors were constructed with I-joists and wood-based panels and supported along two edges, in which half the floors were built with JJI-joists and the other half with TJI-joists. The main difference of the joist types was the flange material, either solid timber (JJI-joists) or Laminated Veneer Lumber (LVL) (TJI-joists), whereas the web was always Oriented Strand Board (OSB). The JJI-joist floors had a size of 3.5×2.44 m, six TJI-joist floors had a size of 3.7×4.4 m and the other six had a size of 5.0×4.4 m. Board types used for the decking were particleboard and OSB, which were connected to the joists using screws or glue and screws together. Furthermore, variations were made to the joist arrangements, whereas the joist spacing was kept at 300, 400 and 600 mm. Double beams were also adopted [10]. The floors were simply supported or fixed to the supports by screws. I-joist blocking elements were inserted in two cases, at mid-span or 1/3 spans. The effect of dead weight on a flooring system was also examined.

Fig. 1 - Two side supported timber floor prepared for dynamic tests

For the experimental work, a continuous excitation method was adopted, with the input magnitude unknown. The duration of each measurement was 100 sec in minimum. At least 27 points, including two reference points, were selected per flooring structure, on which the responses due to light artificial excitation were measured. The measurement points were equally distributed on the floor surface including locations along the central lines and edges. The selected number of measurement points allowed for identifying all mode shapes and thus natural frequencies of interest after signal processing. Furthermore, the number and location of points on the surface provided a good representation of the overall floor responses. This means that each damping ratio analysed for each floor is the mean value of the measured responses.

Two different analysing methods, the Enhanced Frequency Domain Decomposition (EFDD) and the Stochastic Subspace Identification (SSI) of the ARTeMIS Extractor software package, were used. In the EFDD, the signals are processed by a Fast Fourier Transform (FFT) to obtain the spectral densities in the frequency domain and an inverse FFT is applied to the spectral densities for modal parameter estimation, while the SSI is a time domain approach [11][12]. The application of both methods allowed for verifications of the results. The damping ratios identified from the EFDD regarding the first vibration modes were found to be normally higher than the estimates of the SSI. For higher modes, the results from the two methods were close in most cases. Since the damping estimation of the EFDD may include the measurements on nodes, where no useful response or no response at all can be measured, some damping ratios could be estimated inaccurately. The SSI allows for setting boundaries for the modal parameter estimation and thus unrealistic values can be excluded. Another advantage of the SSI method is that the leakage problem, a bias error, inherent with a Discrete Fourier Transform can be avoided. Leakage can lead to overestimation of damping ratios [13]. Hence, the SSI method has been selected for further analysis towards representation of the results in this paper.

3.1 The Stochastic Subspace Identification Method

The structure tested is assumed to behave linearly and be time-invariant, which basically means that it does not change its dynamic properties during the time of measurement. Furthermore it is assumed that the structure is subjected to multiple broad-banded random excitation sources that persistently excite the frequency range of interest, and that the system response is discretely sampled with a sampling interval denoted T.

Under these assumptions the dynamical system can be described by the stochastic state space system in discrete-time [14-16]:

$$\begin{aligned} \boldsymbol{x}_{t+1} &= \boldsymbol{A}\boldsymbol{x}_t + \boldsymbol{w}_t \\ \boldsymbol{y}_t &= \boldsymbol{C}\boldsymbol{x}_t + \boldsymbol{v}_t \end{aligned} \tag{1}$$

The first part of this model structure is the state equation and models the dynamic behaviour of the physical system. The second equation is the observation or output equation, since this equation controls the part of the dynamic system that can be observed in the output of the model. In this model of the physical system, the measured system response $\boldsymbol{y}_t$ is generated by two stochastic processes $\boldsymbol{w}_t$ and $\boldsymbol{v}_t$. These are called the process noise and the measurement noise and are assumed to be Gaussian white noise. The process noise is the input that drives the system dynamics whereas the measurement noise is the direct disturbance of the system response.

The philosophy is that the dynamics of the physical system is modelled by the $n \times n$ state matrix $\boldsymbol{A}$. Given an $n \times 1$ input vector $\boldsymbol{w}_t$, this matrix transforms the state of the system, described by the $n \times 1$ state vector $\boldsymbol{x}_t$, to a new state $\boldsymbol{x}_{t+1}$. The dimension n of the state vector $\boldsymbol{x}_t$ is called the state space dimension. The observable part of the system dynamics is extracted from the state vector by forward multiplication of the $p \times n$ observation matrix $\boldsymbol{C}$. The $p \times 1$ system response vector $\boldsymbol{y}_t$ is a mixture of the observable part of the state and some noise modelled by the measurement noise $\boldsymbol{v}_t$.

3.2 Determination of modal parameters

Assuming that Eq.(1) has been determined using some estimation method, it is straight forward to obtain the natural frequencies, damping ratios and mode shapes by eigenvalue decomposition of the state matrix $\boldsymbol{A}$:

$$\boldsymbol{A} = \boldsymbol{V}[\mu_i]\boldsymbol{V}^{-1} \tag{2}$$

where the columns of $\boldsymbol{V}$ are the n eigenvectors of $\boldsymbol{A}$ and $[\mu_i]$ is a diagonal matrix containing the associated n eigenvalues of $\boldsymbol{A}$. If the system only contains modes with damping less than the critical damping, then the eigenvalues and eigenvectors are organised in complex conjugated pairs, one pair for every mode. In this case the number of modes the system contains is $n/2$.

The natural frequency f_i and damping ratio ζ_i of each of the $n/2$ modes are determined directly from the eigenvalues as:

$$\left.\begin{aligned} \lambda_i &= \frac{\log(\mu_i)}{T} \\ f_i &= \frac{|\lambda_i|}{2\pi} \\ \zeta_i &= -\frac{\mathrm{Re}(\lambda_i)}{|\lambda_i|} \end{aligned}\right\} \quad j = 1, 3, 5, \ldots, n \tag{3}$$

where λ_i is the continuous time equivalent eigenvalue. $|\lambda_i|$ is the absolute value of λ_i, and $\mathrm{Re}(\lambda_i)$ is the real part.

The mode shape vectors $\boldsymbol{\Phi}_i$ of the $n/2$ modes is given by $\boldsymbol{\Phi}_i = \boldsymbol{C}\,\boldsymbol{V}_i$ where $\boldsymbol{V}_i$ is the ith column of $\boldsymbol{V}$.

3.3 Estimation of the state space system

In order to estimate the system (1), system theory is applied to establish the predictor of the model [17]:

$$\begin{aligned} \hat{\boldsymbol{x}}_{t+1} &= \boldsymbol{A}\hat{\boldsymbol{x}}_t + \boldsymbol{K}\boldsymbol{e}_t \\ \boldsymbol{y}_t &= \boldsymbol{C}\hat{\boldsymbol{x}}_t + \boldsymbol{e}_t \end{aligned} \tag{4}$$

This system is called the innovation state space system. The major difference between this system and the system (1) is that the state vector has been substituted with its prediction, and that the two input processes of the system have been converted into one input process – the innovations. This state space system is widely used as model structure in operational modal analysis [15][16].

From Eq.(4) it can be seen that if sufficiently many states of the system (1), e.g. j states, can be predicted, i.e. $\hat{\boldsymbol{x}}_i$ and $\hat{\boldsymbol{x}}_{i+1}$, then the $\boldsymbol{A}$ and $\boldsymbol{C}$ matrices can be estimated from the following least regression problem:

$$\begin{bmatrix} \hat{\boldsymbol{x}}_{i+1} & \hat{\boldsymbol{x}}_{i+2} & \cdots & \hat{\boldsymbol{x}}_{i+j} \\ \boldsymbol{y}_i & \boldsymbol{y}_{i+1} & \cdots & \boldsymbol{y}_{i+j-1} \end{bmatrix} = \begin{bmatrix} \boldsymbol{A} \\ \boldsymbol{C} \end{bmatrix} \begin{bmatrix} \hat{\boldsymbol{x}}_i & \hat{\boldsymbol{x}}_{i+1} & \cdots & \hat{\boldsymbol{x}}_{i+j-1} \end{bmatrix} + \begin{bmatrix} \boldsymbol{\rho}_w \\ \boldsymbol{\rho}_v \end{bmatrix} \tag{5}$$

This is a valid approach since the innovations are assumed to be Gaussian white noise. Since $\boldsymbol{A}$ and $\boldsymbol{C}$ are assumed to be time-invariant this regression approach will be valid even though the predicted state $\hat{\boldsymbol{x}}_i$ and $\hat{\boldsymbol{x}}_{i+1}$ originates from a non-steady state Kalman filter [17].

Hence, the fundamental problem to solve in the stochastic subspace identification technique is to extract the predicted states $\hat{\boldsymbol{x}}_{\mathbf{i}}$ from the measured system response $\boldsymbol{y}_t$. When these states are estimated, $\boldsymbol{A}$ and $\boldsymbol{C}$ are estimated by Eq.(5) and the modal parameters by Eq.(3).

It is beyond the scope of this paper to explain the details of the estimation of the states in the Stochastic Subspace Identification method, but a detailed description can be found in [17].

4. Results

4.1 Damping Ratio Variation for Different Types of Timber Flooring Systems

The damping ratios of the first three vibration bending modes have been dealt with separately. A potential correlation in the damping ratios with modifications of the structural system was examined. Some of the results were rather conclusive. Since damping ratios cannot be always determined very accurately, even when much care was taken during the analysis, the damping ratios have been divided into groups with an increment of 0.5%. The ranges of the frequencies f_i of all 24 structures for the first three modes are listed in Table 1a, and the lowest, highest and mean values for the damping ratios ζ_i in Table 1b. The minimum and maximum damping may not correspond to the minimum and maximum frequencies.

Fig. 2 shows the damping ratios as histograms for each of the modes. The columns present the numbers of damping ratios in a specified range, and the curves represent the density functions for normal distribution based on the mean value and standard deviation of the exact damping ratios determined. This normal distribution has to be considered with care, since a variety of flooring structures were included in the figures. However, it indeed indicates the range of typical damping ratios of timber floors.

Table 1a - Minimum and maximum natural frequencies for the 24 test floors

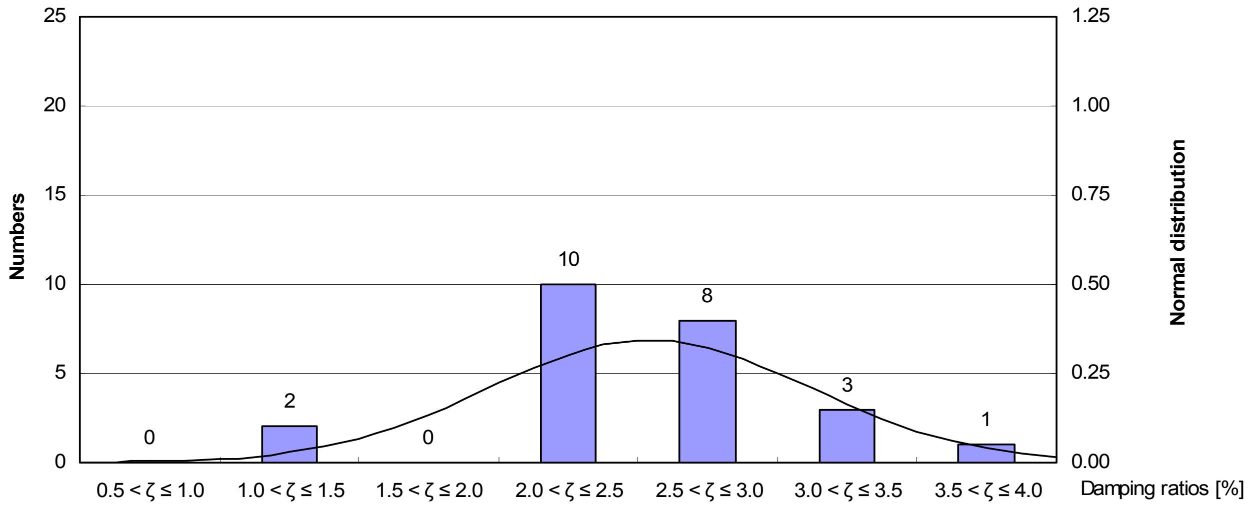
	Mode 1	Mode 2	Mode 3
Minimum modal frequency $f_{i,min}$ [Hz]	12.17	13.18	15.35
Maximum modal frequency $f_{i,max}$ [Hz]	27.19	33.97	48.22

Table 1b - Minimum, maximum and mean damping ratios for the 24 test floors

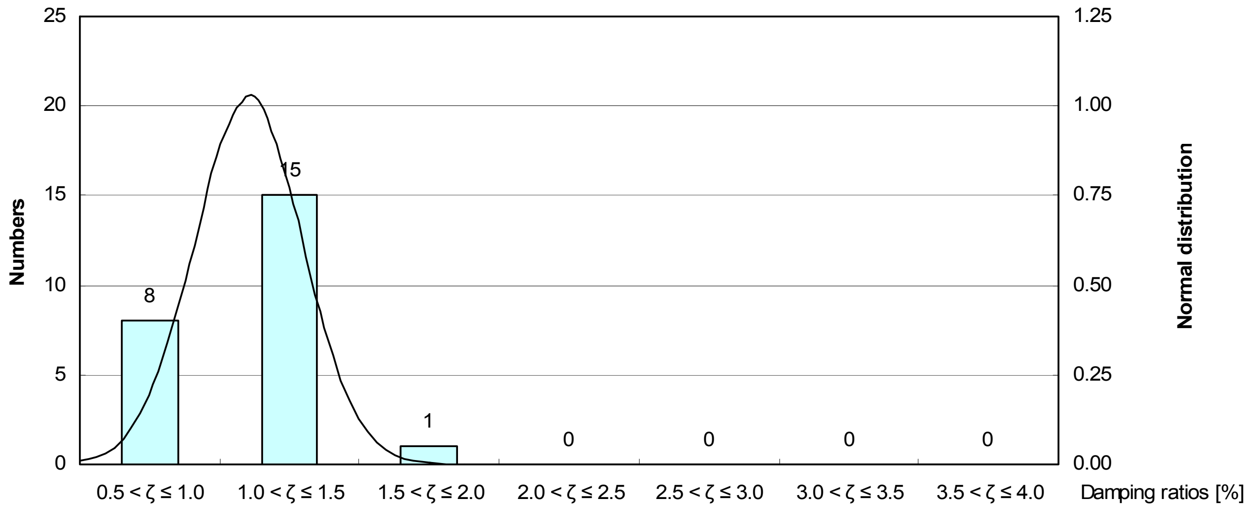
	Mode 1	Mode 2	Mode 3
Minimum damping ratio $\zeta_{i,min}$ [%]	1.16	0.84	0.81
Maximum damping ratio $\zeta_{i,max}$ [%]	3.95	1.72	1.63
Mean damping ratio $\zeta_{i,mean}$ [%]	**2.54**	**1.10**	**1.18**

It has been found that the damping ratios of the first mode usually spread more widely than those of higher modes. Fig. 2 shows that the maximum damping ratios are exceptional since only one value laid in the specified range for each individual mode. The ratios for the first mode usually were below 3.5% and those for the second and third modes usually below 1.5%. Fig. 2a) also shows two very low damping ratios below 1.5% for the first mode. These damping ratios belong to the floors where the dead weight was increased without increasing flooring stiffness. Where only structural modifications took place, the damping ratios are all above 2%.

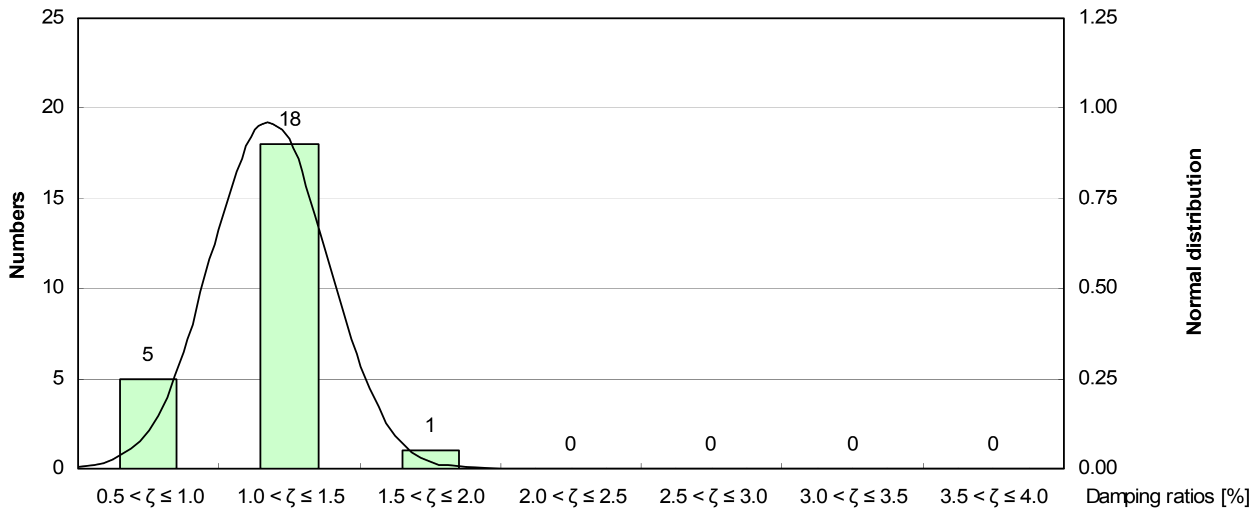
Since only the non-structural modifications with added dead weights yielded damping ratios below 2% (even below 1.5%) for the first mode, a separate analysis has been conducted to study the degree of the effect of added weight. A relatively high value of ~4% was found for one system, but no special consideration is given to this value here. Further separation of the structural systems to identify the effect on damping in more detail is identified as an area for future investigations.



a) Damping ratios corresponding to the first bending mode



b) Damping ratios corresponding to the second bending mode



c) Damping ratios corresponding to the third bending mode

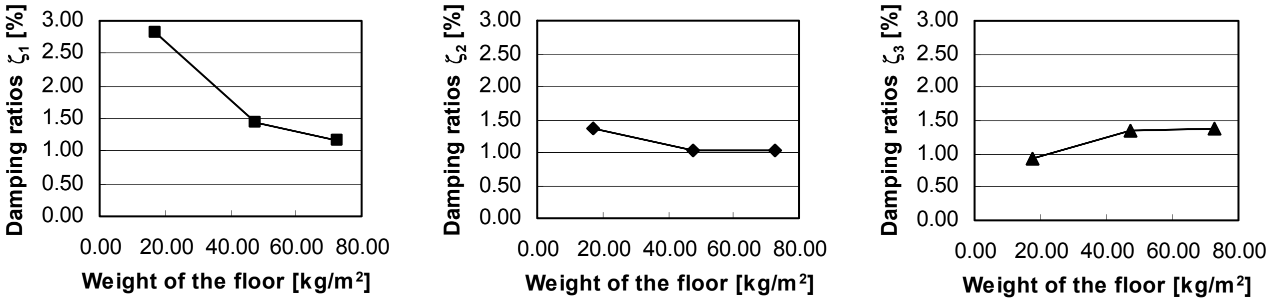
Fig. 2 - Histograms of damping ratios for individual vibration modes

4.2 The Effect of Additional Dead Weight on Damping

The increase in dead weight of the floor was achieved by equally distributing sand bags on the top surface of the floor. The original floor mass of ~17 kg/m² was first increased to ~47 kg/m² by 30 kg/m² and then to ~72 kg/m² by another 25 kg/m². The effect of the raised weight on the damping ratios ζ_i is illustrated in Fig. 3. The test data is listed in Table 2, including natural frequencies f_i and the corresponding damping ratios ζ_i.

Table 2 - Natural frequencies f_i and the corresponding damping ratios ζ_i for variation in weight

Floor weight [kg/m²]	f_1 [Hz]	**ζ_1 [%]**	f_2 [Hz]	**ζ_2 [%]**	f_3 [Hz]	**ζ_3 [%]**
17.36 kg/m²	22.16	**2.83**	27.58	**1.37**	33.79	**0.94**
47.46 kg/m²	14.66	**1.45**	15.96	**1.02**	18.38	**1.34**
72.46 kg/m²	12.17	**1.16**	13.18	**1.02**	15.35	**1.39**



a) Mode 1 b) Mode 2 c) Mode 3

Fig. 3 - Damping ratios for increasing floor weight

It can be seen that there was a monotonic reduction in damping for the first mode with the increasing dead weight. Compared to the light-weight floor with a mass of ~17 kg/m² the damping ratios decreased by 49% and 59% respectively for the first and second increment in mass. The second mode responded similarly with a reduction of 26% to the weight increment initially and then became unchanged. The third mode reacted reversely. The reduction trend in damping of the first mode for increasing dead weight observed by other researchers can be confirmed by this study.

5. Discussion, Conclusions and Further Work

It has been shown that for the tested light-weight floors, the damping ratios corresponding to the first mode of vibration are normally higher than those of the second or third mode, mainly in the range of 2.0-3.5% with a mean value of 2.5%. The damping ratios for the second and third modes usually lie in the range of 0.8-1.5% with mean values of 1.1% and 1.2%, respectively. If the mass of the floor is not significantly changed, the damping ratios for the first mode are above 2%.

Since the weight plays a major role for the first vibration mode, damping ratios used to assess dynamic performances of bare floors for design should closely depend on dead weight. The use of a damping ratio of 2% for design is just reasonable for light-weight structures (say ~20 kg/m² or below) if only the first mode is considered, but may not always be on the very safe side. It should be mentioned that natural frequencies higher than the first one can also contribute to annoying vibrations. For heavier floors lower limits for the damping ratio should be adopted.

The effect of weight on dynamic performances of floors, in particular on damping ratios, needs to be further investigated, since heavier floors clearly exhibit reduced damping capabilities with respect to the first natural frequency, which is often considered to be the most important one regarding design.

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