Continuous Systems, Infinite Degrees of Freedom

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Outline

Intro

Discrete models

Until now the dynamical behavior of structures has been modeled using discrete degrees of freedom, as in the Finite Element Method procedure, and in many cases we have found that we are able to reduce the number of *dynamical degrees of freedom* using the static condensation procedure (multistory buildings are an excellent example of structures for which a few dynamical degrees of freedom can describe the dynamical response).

Intro

Continuous models

For different type of structures (e.g., bridges, chimneys), a lumped mass model is not an option. While a *FE* model is always appropriate, there is no apparent way of lumping the structural masses in a way that is obviously correct, and a great number of degrees of freedom must be retained in the dynamic analysis.
An alternative to detailed *FE* models is deriving the equation of motion, in terms of partial derivatives differential equation, directly for the continuous systems.

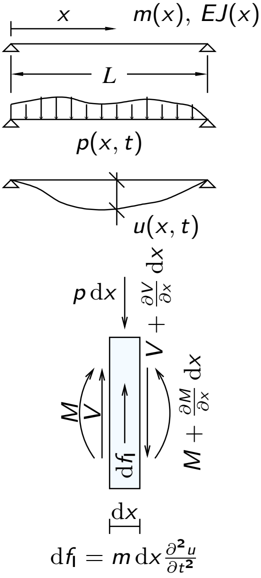
Continuous Systems

There are many different continuous systems whose dynamics are approachable with the instruments of classical mechanics,

- taught strings,
- axially loaded rods,
- beams in flexure,
- plates and shells,
- 3D solids.

In the following, we will focus our interest on beams in flexure.

EoM for an undamped beam



At the left, a straight beam with characteristic depending on position x: $m = m(x)$ and $EJ = EJ(x)$; with the signs conventions for displacements, accelerations, forces and bending moments reported left, the equation of vertical equilibrium for an infinitesimal slice of beam is

$$V - (V + \frac{\partial V}{\partial x}\,\mathrm{d}x) + m\,\mathrm{d}x\frac{\partial^2 u}{\partial t^2} - p(x,t)\,\mathrm{d}x = 0.$$

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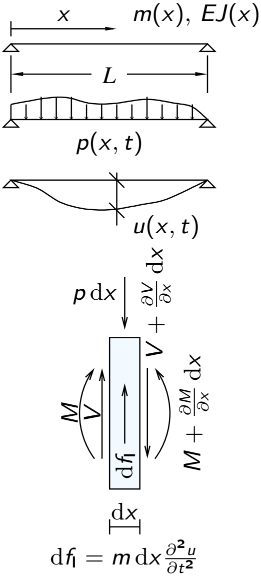

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Rearranging and simplifying $\mathrm{d}x$,

$$\frac{\partial V}{\partial x} = m\frac{\partial^2 u}{\partial t^2} - p(x,t).$$

Equation of motion, 2

The rotational equilibrium, neglecting rotational inertia and simplifying $\mathrm{d}x$ is

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Substituting in the equation of vertical equilibrium and rearranging

$$m(x)\frac{\partial^2 u}{\partial t^2} - \frac{\partial^2 M}{\partial x^2} = p(x, t)$$

Equation of motion, 3

Using the moment-curvature relationship,

$$M = -EJ\frac{\partial^2 u}{\partial x^2}$$

and substituting in the equation above we have the equation of dynamic equilibrium

$$m(x)\frac{\partial^2 u}{\partial t^2} + \frac{\partial^2}{\partial x^2}\left[EJ(x)\frac{\partial^2 u}{\partial x^2}\right] = p(x,t).$$

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Partial Derivatives Differential Equation

In this formulation of the equation of equilibrium we have

- one equation of equilibrium
- one unknown, $u(x, t)$.

It is a partial derivatives differential equation because we have the derivatives of u with respect to x and t simultaneously in the same equation.

Effective Earthquake Loading

If our continuous structure is subjected to earthquake excitation, we will write, as usual, $u_{\text{TOT}} = u(x,t) + u_{\text{g}}(t)$ and, consequently,

$$\ddot{u}_{\text{TOT}} = \ddot{u}(x,t) + \ddot{u}_{\text{g}}(t)$$

and, using the usual considerations,

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Only a word of caution, in every case we must consider the component of earthquake acceleration *parallel* to the transverse motion of the beam.

Free Vibrations

For free vibrations, $p(x,t) \equiv 0$ and the equation of equilibrium for an infinitesimal slice of beam is

$$m(x)\frac{\partial^2 u}{\partial t^2} + \frac{\partial^2}{\partial x^2}\left[EJ(x)\frac{\partial^2 u}{\partial x^2}\right] = 0.$$

Using separation of variables, with the following notations,

$$u(x,t) = q(t)\phi(x),\ \frac{\partial u}{\partial t} = \dot{q}\phi,\ \frac{\partial u}{\partial x} = q\phi'$$

etc for higher order derivatives, we have

$$m(x)\ddot{q}(t)\phi(x) + q(t)\left[EJ(x)\phi''(x)\right]'' = 0.$$

Free Vibrations, 2

Dividing both terms in

$$m(x)\ddot{q}(t)\phi(x) + q(t)\left[EJ(x)\phi''(x)\right]'' = 0.$$

by $m(x)u(x,t) = m(x)q(t)\phi(x)$ and rearranging, we have

$$-\frac{\ddot{q}(t)}{q(t)} = \frac{[EJ(x)\phi''(x)]''}{m(x)\phi(x)}.$$

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The left member is a function of time only, the right member a function of position only, and they are equal... this is possible if and only if both terms are constant, let's name this constant ω^2 and write

$$-\frac{\ddot{q}(t)}{q(t)} = \frac{[EJ(x)\phi''(x)]''}{m(x)\phi(x)} = \omega^2,$$

Free Vibrations, 3

From the previous equations we can derive the following two

$$\ddot{q} + \omega^2 q = 0$$

$$\left[EJ(x)\phi''(x)\right]'' = \omega^2 m(x)\phi(x)$$

The first equation, $\ddot{q} + \omega^2 q = 0$, has the homogeneous integral

$$q(t) = A \sin \omega t + B \cos \omega t$$

so that our free vibration solution is

$$u(x, t) = \phi(x) \left(A \sin \omega t + B \cos \omega t\right),$$

the free vibration shape $\phi(x)$ will be modulated by a harmonic function of time.

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To find something about ω's and ϕ's (that is, the eigenvalues and the *eigenfunctions* of our problem), we have to introduce an important simplification.

Eigenpairs of a uniform beam

With $EJ = \text{const.}$ and $m = \text{const.}$, we have from the second equation in previous slide,

$$EJ\phi^{\text{IV}} - \omega^2 m\phi = 0,$$

with $\beta^4 = \frac{\omega^2 m}{EJ}$ it is

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a differential equation of 4$^{\text{th}}$ order with constant coefficients.
Substituting $\phi = \exp st$ and simplifying,

$$s^4 - \beta^4 = 0,$$

the roots of the associated polynomial are

$$s_1 = \beta,\ s_2 = -\beta,\ s_3 = i\beta,\ s_4 = -i\beta$$

and the general integral is

$$\phi(x) = \mathcal{A}\sin\beta x + \mathcal{B}\cos\beta x + \mathcal{C}\sinh\beta x + \mathcal{D}\cosh\beta x$$

Constants of Integration

For a uniform beam in free vibration, the general integral is

$$\phi(x) = \mathcal{A} \sin \beta x + \mathcal{B} \cos \beta x + \mathcal{C} \sinh \beta x + \mathcal{D} \cosh \beta x$$

In this expression we see 5 parameters, the 4 constants of integration and the wave number β (further consideration shows that the constants can be arbitrarily scaled).
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In general for the transverse motion of a segment of beam supported at the extremes we can write exactly 4 equations expressing boundary conditions, either from kinematic or static considerations.
All these boundary conditions

- lead to linear, homogeneous equation where
- the coefficients of the equations depend on the parameter β.

Eigenvalues and eigenfunctions

Imposing the boundary conditions give a homogeneous linear system with coefficients depending on β, hence:

- a non trivial solution is possible only for particular values of β, for which the determinant of the matrix of coefficients is equal to zero and
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In the case of *MDOF* systems, the determinant's equation is an algebraic equation of order N, giving exactly N eigenvalues, now the equation to be solved is a transcendental equation (examples from the next slide), with an infinity of solutions.

Simply supported beam

Consider a simply supported uniform beam of length L, displacements at both ends must be zero, as well as the bending moments:

$$\phi(0) = \mathcal{B} + \mathcal{D} = 0, \qquad \phi(L) = 0,$$
$$-EJ\phi''(0) = \beta^2 EJ(\mathcal{B} - \mathcal{D}) = 0, \qquad -EJ\phi''(L) = 0.$$

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The conditions for the left support require that $\mathcal{B} = \mathcal{D} = 0$
Now, we can write the equations for the right support as

$$\phi(L) = \mathcal{A} \sin\beta L + \mathcal{C} \sinh\beta L = 0$$
$$-EJ\phi''(L) = \beta^2 EJ(\mathcal{A} \sin\beta L - \mathcal{C} \sinh\beta L) = 0$$

or

$$\begin{bmatrix} +\sin\beta L & +\sinh\beta L \\ +\sin\beta L & -\sinh\beta L \end{bmatrix} \begin{Bmatrix} \mathcal{A} \\ \mathcal{C} \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix}.$$

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The determinant is $-2\sin\beta L \sinh\beta L$, equating to zero with the understanding that $\sinh\beta L \neq 0$ if $\beta \neq 0$ results in

$$\sin\beta L = 0.$$

All positive β solutions are given by

$$\beta L = n\pi$$

with $n = 1, \ldots, \infty$. We have an infinity of eigenvalues,

$$\beta_n = \frac{n\pi}{L} \text{ and } \omega_n = \beta^2\sqrt{\frac{EJ}{m}} = n^2\pi^2\sqrt{\frac{EJ}{mL^4}}$$

and of eigenfunctions

$$\phi_1 = \sin\frac{\pi x}{L},\ \phi_2 = \sin\frac{2\pi x}{L},\ \phi_3 = \sin\frac{3\pi x}{L},\ \cdots$$

Cantilever beam

For $x = 0$, we have zero displacement and zero rotation

$$\phi(0) = \mathcal{B} + \mathcal{D} = 0, \qquad \phi'(0) = \beta(\mathcal{A} + \mathcal{C}) = 0,$$

for $x = L$, both bending moment and shear must be zero

$$-EJ\phi''(L) = 0, \qquad -EJ\phi'''(L) = 0.$$

Substituting the expression of the general integral, with $\mathcal{D} = -\mathcal{B}$, $\mathcal{C} = -\mathcal{A}$ from the left end equations, in the right end equations and simplifying

$$\begin{bmatrix} \sinh\beta L + \sin\beta L & \cosh\beta L + \cos\beta L \\ \cosh\beta L + \cos\beta L & \sinh\beta L - \sin\beta L \end{bmatrix} \begin{Bmatrix} \mathcal{A} \\ \mathcal{B} \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix}.$$

Cantilever beam, 2

Imposing a zero determinant results in

$$(\cosh^2 \beta L - \sinh^2 \beta L) + (\sin^2 \beta L + \cos^2 \beta L) + 2 \cos \beta L \cosh \beta L =$$
$$= 2(1 + \cos \beta L \cosh \beta L) = 0$$

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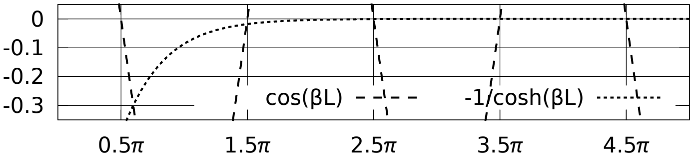
Rearranging, $\cos\beta L = -(\cosh\beta L)^{-1}$ and plotting these functions on the same graph

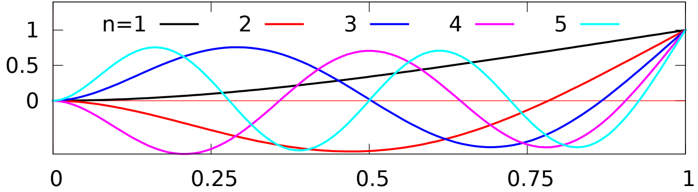


it is $\beta_1 L = 1.8751$ and $\beta_2 L = 4.6941$, while for $n = 3, 4, \dots$ with good approximation it is $\beta_n L \approx \frac{2n-1}{2}\pi$.

Cantilever beam, 3

Eigenvectors are given by

$$\phi_n(x) = C_n \left[(\cosh \beta_n x - \cos \beta_n x) - \frac{\cosh \beta_n L + \cos \beta_n L}{\sinh \beta_n L + \sin \beta_n L} (\sinh \beta_n x - \sin \beta_n x) \right]$$



Above, in abscissas x/L and in ordinates $\phi_n(x)$ for the first 5 modes of vibration of the cantilever beam.

n	1	2	3	4	5
$\beta_n L$	1.8751	4.6941	7.8548	10.9962	$\approx 4.5\pi$
$\omega\sqrt{\frac{mL^4}{EJ}}$	3.516	22.031	61.70	120.9	$\cdots$

Other Boundary Conditions

It is possible that

- the beam is supported not by a fixed constraint but by a spring, either extensional or flexural,
- the beam at its end supports a lumped mass, with inertia and possibly rotatory inertia.

Elastic Support

Consider the right end, $x = L$, supported by an extensional spring k. An infinitesimal slice of beam is subjected to two discrete forces, the shear $V(L,t) = -EJ\phi'''(L)q(t)$ and the spring reaction, $ku(L,t) = k\phi(L)q(t)$. With our sign conventions, the equilibrium is written $-V - ku = 0$ or, simplifying the time dependency,

$$\phi(L) = \frac{EJ}{k}\phi'''(L) = \frac{EJ}{kL^3}(\beta L)^3 f(\beta L; \mathcal{A}, \mathcal{B}, \mathcal{C}, \mathcal{D})$$

where we have shown that the right member depends only on βL. The equation of equilibrium is an homogeneous equation in $\mathcal{A}, \mathcal{B}, \mathcal{C}$ and $\mathcal{D}$.

Supported Mass

The beam end supports a lumped mass M and it is subjected to shear $V(L, t)$ and an inertial force, $f_I = -M\frac{\partial^2 u(L,t)}{\partial t^2}$.
Considering that in free vibrations we have harmonic time dependency, it is

$$f_I = -M\phi(L)\frac{\partial^2 q(t)}{\partial t^2} = M\omega^2\,\phi(L)q(t) = M\beta^4\frac{EJ}{m}\phi(L)q(t).$$

and the equation of equilibrium is, simplifying EJ and the time dependency

$$\beta^3 f(\ldots) + \frac{M}{mL}\beta^4 L\phi(\ldots) = 0$$

eventually dividing by β^3 we have an homogeneous equation in $\mathcal{A}\ldots$ as well,

$$f(\ldots) + \frac{M}{mL}\beta L\phi(\ldots) = 0/$$

Mode Orthogonality

We will demonstrate mode orthogonality for a restricted set of of boundary conditions, i.e., disregarding elastic supports and supported masses. In the beginning we have, for $n = r$,

$$\left[EJ(x)\phi_r''(x)\right]'' = \omega_r^2 m(x)\phi_r(x).$$

Pre-multiply both members by $\phi_s(x)$ and integrate over the length of the beam gives you

$$\int_0^L \phi_s(x)\left[EJ(x)\phi_r''(x)\right]'' \mathrm{d}x = \omega_r^2 \int_0^L \phi_s(x)m(x)\phi_r(x)\,\mathrm{d}x.$$

Mode Orthogonality, 2

The left member can be integrated by parts, two times, as in

$$\int_0^L \phi_s(x) \left[EJ(x)\phi_r''(x)\right]'' \mathrm{d}x =$$

$$\left[\phi_s(x) \left[EJ(x)\phi_r''(x)\right]'\right]_0^L - \left[\phi_s'(x)EJ(x)\phi_r''(x)\right]_0^L +$$

$$\int_0^L \phi_s''(x)EJ(x)\phi_r''(x)\,\mathrm{d}x$$

Mode Orthogonality, 2

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$$\int_0^L \phi_s(x)\left[EJ(x)\phi_r''(x)\right]''\,\mathrm{d}x = \left[\phi_s(x)\left[EJ(x)\phi_r''(x)\right]'\right]_0^L - \left[\phi_s'(x)EJ(x)\phi_r''(x)\right]_0^L + \int_0^L \phi_s''(x)EJ(x)\phi_r''(x)\,\mathrm{d}x$$

but the terms in brackets are always zero, the first being the product of end displacement by end shear, the second the product of end rotation by bending moment, and for fixed constraints or free end one of the two terms must be zero. So it is

$$\int_0^L \phi_s''(x)EJ(x)\phi_r''(x)\,\mathrm{d}x = \omega_r^2\int_0^L \phi_s(x)m(x)\phi_r(x)\,\mathrm{d}x.$$

Mode Orthogonality, 3

We write the last equation exchanging the roles of r and s and subtract from the original,

$$\int_0^L \phi_s''(x) EJ(x) \phi_r''(x)\,\mathrm{d}x - \int_0^L \phi_r''(x) EJ(x) \phi_s''(x)\,\mathrm{d}x = \\ \omega_r^2 \int_0^L \phi_s(x) m(x) \phi_r(x)\,\mathrm{d}x - \omega_s^2 \int_0^L \phi_r(x) m(x) \phi_s(x)\,\mathrm{d}x.$$

This obviously can be simplified giving

$$(\omega_r^2 - \omega_s^2) \int_0^L \phi_r(x) m(x) \phi_s(x)\,\mathrm{d}x = 0$$

implying that, for $\omega_r^2 \neq \omega_s^2$ the modes are orthogonal with respect to the mass distribution, $\int \phi_s \phi_r\, m\,\mathrm{d}x = \delta_{rs} m_r$.
It is then easy to show that $\int \phi_s'' \phi_r'' EJ\,\mathrm{d}x = \delta_{rs} m_r \omega_r^2$.

Forced dynamic response

With $u(x, t) = \sum_1^\infty \phi_m(x) q_m(t)$, the equation of motion can be written

$$\sum_1^\infty m(x)\phi_m(x)\ddot{q}_m(t) + \sum_1^\infty \left[EJ(x)\phi_m''(x)\right]'' q_m(t) = p(x, t)$$

pre-multiplying by ϕ_n and integrating each sum and the loading term gives the equation

$$\sum_1^\infty \int_0^L \phi_n(x) m(x)\phi_m(x)\ddot{q}_m(t)\,\mathrm{d}x +$$
$$\sum_1^\infty \int_0^L \phi_n(x)\left[EJ(x)\phi_m''(x)\right]'' q_m(t)\,\mathrm{d}x = \int_0^L \phi_n(x) p(x, t)\,\mathrm{d}x.$$

Forced dynamic response, 2

By the orthogonality of the eigenfunctions this can be simplified to

$$m_n \ddot{q}_n(t) + k_n q_n(t) = p_n(t), \qquad n = 1, 2, \ldots, \infty$$

with

$$m_n = \int_0^L \phi_n m \phi_n \, \mathrm{d}x, \qquad k_n = \int_0^L \phi_n \left[EJ \phi_n'' \right]'' \mathrm{d}x,$$

and

$$p_n(t) = \int_0^L \phi_n p(x,t) \, \mathrm{d}x.$$

For free ends and/or fixed supports, $k_n = \int_0^L \phi_n'' EJ \phi_n'' \, \mathrm{d}x$.

Earthquake response

Consider an effective earthquake load, $p(x,t) = m(x)\ddot{u}_\mathrm{g}(t)$, with

$$\mathcal{L}_n = \int_0^L \phi_n(x) m(x)\,\mathrm{d}x, \qquad \Gamma_n = \frac{\mathcal{L}_n}{m_n},$$

the modal equation of motion can be written (with an obvious generalization)

$$\ddot{q}_n + 2\omega_n\zeta_n\dot{q}_n + \omega_n^2 q = -\Gamma_n \ddot{u}_\mathrm{g}(t).$$

The modal response, analogously to the case of discrete models, is the product of the modal participation factor and the pseudo-displacement response,

$$q_n(t) = \Gamma_n D_n(t).$$

Earthquake response, 2

Modal contributions can be computed directly, e.g.

$$u_n(x,t) = \Gamma_n \phi_n(x) D_n(t),$$
$$M_n(x,t) = -\Gamma_n EJ(x) \phi_n''(x) D_n(t),$$

or can be computed from the equivalent static forces,

$$f_s(x,t) = \left[EJ(x) u(x,t)'' \right]''.$$

Earthquake response, 3

The modal contributions to equiv. static forces are

$$f_{sn}(x,t) = \Gamma_n \left[EJ(x)\phi_n(x)''\right]'' D_n(t),$$

that, because it is

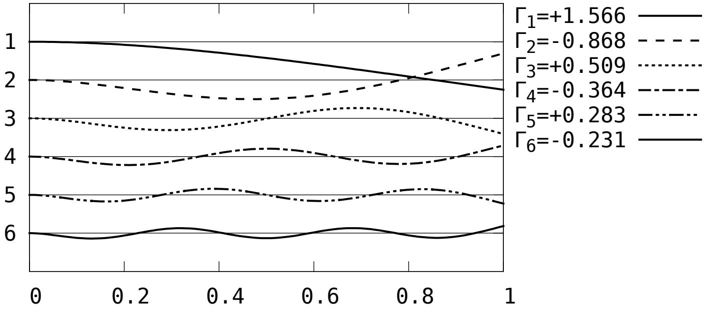
$$\left[EJ(x)\phi''(x)\right]'' = \omega^2 m(x)\phi(x)$$

can be written in terms of the mass distribution and of the pseudo-acceleration response $A_n(t) = \omega_n^2 D_n(t)$

$$f_{sn}(x,t) = \Gamma_n m(x)\phi_n(x)\omega_n^2 D_n(t) = \Gamma_n m(x)\phi_n(x)A_n(t).$$

Earthquake response, 4

The effective load is proportional to the mass distribution, and we can do a modal mass decomposition in the same way that we had for *MDOF* systems, $m(x) = \sum r_n(x) = \sum \Gamma_n m(x)\phi_n(x)$



Above, the modal mass decomposition $r_n = \Gamma_n m\phi_n$,for the first six modes of a uniform cantilever, in abscissa x/L.

EQ example, cantilever

For a cantilever, it is possible to derive explicitly some response quantities,

$$V(x), \qquad V_\mathrm{B}, \qquad M(x), \qquad M_\mathrm{B},$$

that is, the shear force and the base shear force, the bending moment and the base bending moment.

$$V_n^{\mathrm{st}}(x) = \int_x^L r_n(s)\,\mathrm{d}s, \qquad V_\mathrm{B}^{\mathrm{st}} = \int_0^L r_n(s)\,\mathrm{d}s = \Gamma_n\mathcal{L}_n = M_n^\star,$$

$$M_n^{\mathrm{st}}(x) = \int_x^L r_n(s)(s-x)\,\mathrm{d}s, \qquad M_\mathrm{B}^{\mathrm{st}} = \int_0^L s r_n(s)\,\mathrm{d}s = M_n^\star h_n^\star.$$

$M_n^\star$ is the *participating modal mass* and expresses the participation of the different modes to the base shear, it is $\sum M_n^\star = \int_0^L m(x)\,\mathrm{d}x$.
$M_n^\star h_n^\star$ expresses the modal participation to base moment, $h_n^\star$ is the height where the participating modal mass $M_n^\star$ must be placed so that its effects on the base are the same of the static modal forces effects, or $M_n^\star$ is the resultant of s.m.f. and $h_n^\star$ is the position of this resultant.

EQ example, cantilever, 2

Starting with the definition of total mass and operating a chain of substitutions,

$$\begin{aligned} M_{\mathrm{TOT}} &= \int_0^L m(x)\,\mathrm{d}x = \sum \int_0^L r_n(x)\,\mathrm{d}x \\ &= \sum \int_0^L \Gamma_n m(x)\phi_n(x)\,\mathrm{d}x = \sum \Gamma_n \int_0^L m(x)\phi_n(x)\,\mathrm{d}x \\ &= \sum \Gamma_n \mathcal{L}_n = \sum M_n^\star, \end{aligned}$$

we have demonstrated that the sum of the participating modal mass is equal to the total mass.
The demonstration that $M_{\mathrm{B,TOT}} = \sum M_n^\star h_n^\star$ is similar and is left as an exercise.

EQ example, cantilever, 3

For the first 8 modes of a uniform cantilever,

n	$\mathcal{L}_n$	m_n	Γ_n	$V_{B,n} = \mathcal{L}_n\Gamma_n$	h_n	$M_{B,n}$
1	0.391496	0.250	1.565984	0.613076	0.726477	0.445386
2	-0.216968	0.250	-0.867872	0.188300	0.209171	0.039387
3	0.127213	0.250	0.508851	0.064732	0.127410	0.008248
4	-0.090949	0.250	-0.363796	0.033087	0.090943	0.003009
5	0.070735	0.250	0.282942	0.020014	0.070736	0.001416
6	-0.057875	0.250	-0.231498	0.013398	0.057875	0.000775
7	0.048971	0.250	0.195883	0.009593	0.048971	0.000470
8	-0.042441	0.250	-0.169765	0.007205	0.042442	0.000306

The convergence for M_B is faster than the convergence for V_B because V_B is proportional to a higher derivative of displacements.