# Structural Matrices in *MDOF* Systems

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## Outline





## Introductory Remarks

Today we will study the properties of structural matrices, that is the operators that relate the vector of system coordinates $\boldsymbol{x}$ and its time derivatives $\dot{\boldsymbol{x}}$ and $\ddot{\boldsymbol{x}}$ to the forces acting on the system nodes, $\boldsymbol{f}_\text{S}$, $\boldsymbol{f}_\text{D}$ and $\boldsymbol{f}_\text{I}$, respectively.

In the end, we will see again the solution of a *MDOF* problem by superposition, and in general today we will revisit many of the subjects of our previous class, but you know that a bit of reiteration is really good for developing minds.

## Structural Matrices



We already met the mass and the stiffness matrix, $\boldsymbol{M}$ and $\boldsymbol{K}$, and tangentially we introduced also the dampig matrix $\boldsymbol{C}$.
We have seen that these matrices express the linear relation that holds between the vector of system coordinates $\boldsymbol{x}$ and its time derivatives $\dot{\boldsymbol{x}}$ and $\ddot{\boldsymbol{x}}$ to the forces acting on the system nodes, $\boldsymbol{f}_\mathrm{S}$, $\boldsymbol{f}_\mathrm{D}$ and $\boldsymbol{f}_\mathrm{I}$, elastic, damping and inertial force vectors.

$$\boldsymbol{M}\,\ddot{\boldsymbol{x}} + \boldsymbol{C}\,\dot{\boldsymbol{x}} + \boldsymbol{K}\,\boldsymbol{x} = \boldsymbol{p}(t)$$
$$\boldsymbol{f}_\mathrm{I} + \boldsymbol{f}_\mathrm{D} + \boldsymbol{f}_\mathrm{S} = \boldsymbol{p}(t)$$

Also, we know that $\boldsymbol{M}$ and $\boldsymbol{K}$ are symmetric and definite positive, and that it is possible to uncouple the equation of motion expressing the system coordinates in terms of the *eigenvectors*, $\boldsymbol{x}(t) = \sum q_i \psi_i$, where the $q_i$ are the *modal coordinates* and the eigenvectors $\psi_i$ are the non-trivial solutions to the equation of free vibrations,

$$\left(\boldsymbol{K} - \omega^2 \boldsymbol{M}\right)\psi = \boldsymbol{0}$$

## Free Vibrations



From the homogeneous, undamped problem

$$\boldsymbol{M}\,\ddot{\boldsymbol{x}} + \boldsymbol{K}\,\boldsymbol{x} = \boldsymbol{0}$$

introducing separation of variables

$$\boldsymbol{x}(t) = \psi\,(A\sin\omega t + B\cos\omega t)$$

we wrote the homogeneous linear system

$$\left(\boldsymbol{K} - \omega^2 \boldsymbol{M}\right)\psi = \boldsymbol{0}$$

whose non-trivial solutions $\psi_i$ for $\omega_i^2$ such that $\left\|\boldsymbol{K} - \omega_i^2 \boldsymbol{M}\right\| = 0$ are the eigenvectors.
It was demonstrated that, for each pair of distint *eigenvalues* $\omega_r^2$ and $\omega_s^2$, the corresponding eigenvectors obey the ortogonality condition,

$$\psi_s^T \boldsymbol{M}\,\psi_r = \delta_{rs} M_r, \quad \psi_s^T \boldsymbol{K}\,\psi_r = \delta_{rs}\omega_r^2 M_r.$$

## Additional Orthogonality Relationships



From

$$\boldsymbol{K}\,\psi_s = \omega_s^2 \boldsymbol{M}\,\psi_s$$

premultiplying by $\psi_r^T \boldsymbol{K}\boldsymbol{M}^{-1}$ we have

$$\psi_r^T \boldsymbol{K}\boldsymbol{M}^{-1}\boldsymbol{K}\,\psi_s = \omega_s^2 \psi_r^T \boldsymbol{K}\,\psi_s = \delta_{rs}\omega_r^4 M_r,$$

premultiplying the first equation by $\psi_r^T \boldsymbol{K}\boldsymbol{M}^{-1}\boldsymbol{K}\boldsymbol{M}^{-1}$

$$\psi_r^T \boldsymbol{K}\boldsymbol{M}^{-1}\boldsymbol{K}\boldsymbol{M}^{-1}\boldsymbol{K}\,\psi_s = \omega_s^2 \psi_r^T \boldsymbol{K}\boldsymbol{M}^{-1}\boldsymbol{K}\,\psi_s = \delta_{rs}\omega_r^6 M_r$$

and, generalizing,

$$\psi_r^T \left(\boldsymbol{K}\boldsymbol{M}^{-1}\right)^b \boldsymbol{K}\,\psi_s = \delta_{rs}\left(\omega_r^2\right)^{b+1} M_r.$$

## Additional Relationships, 2

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From

$$\boldsymbol{M}\,\psi_s = \omega_s^{-2} \boldsymbol{K}\,\psi_s$$

premultiplying by $\psi_r^T \boldsymbol{M}\boldsymbol{K}^{-1}$ we have

$$\psi_r^T \boldsymbol{M}\boldsymbol{K}^{-1}\boldsymbol{M}\,\psi_s = \omega_s^{-2} \psi_r^T \boldsymbol{M}\,\psi_s = \delta_{rs}\frac{M_s}{\omega_s^2}$$

premultiplying the first eq. by $\psi_r^T \left(\boldsymbol{M}\boldsymbol{K}^{-1}\right)^2$ we have

$$\psi_r^T \left(\boldsymbol{M}\boldsymbol{K}^{-1}\right)^2 \boldsymbol{M}\,\psi_s = \omega_s^{-2} \psi_r^T \boldsymbol{M}\boldsymbol{K}^{-1}\boldsymbol{M}\,\psi_s = \delta_{rs}\frac{M_s}{\omega_s^4}$$

and, generalizing,

$$\psi_r^T \left(\boldsymbol{M}\boldsymbol{K}^{-1}\right)^b \boldsymbol{M}\,\psi_s = \delta_{rs}\frac{M_s}{\omega_s^{2b}}$$

## Additional Relationships, 3

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Defining $X_{rs}(k) = \psi_r^T \boldsymbol{M}\left(\boldsymbol{M}^{-1}\boldsymbol{K}\right)^k \psi_s$ we have

$$\begin{cases} X_{rs}(0) = \psi_r^T \boldsymbol{M}\psi_s & = \delta_{rs}\left(\omega_s^2\right)^0 M_s \\ X_{rs}(1) = \psi_r^T \boldsymbol{K}\psi_s & = \delta_{rs}\left(\omega_s^2\right)^1 M_s \\ X_{rs}(2) = \psi_r^T \left(\boldsymbol{K}\boldsymbol{M}^{-1}\right)^1 \boldsymbol{K}\psi_s & = \delta_{rs}\left(\omega_s^2\right)^2 M_s \\ \ldots \\ X_{rs}(n) = \psi_r^T \left(\boldsymbol{K}\boldsymbol{M}^{-1}\right)^{n-1} \boldsymbol{K}\psi_s & = \delta_{rs}\left(\omega_s^2\right)^n M_s \end{cases}$$

Observing that $\left(\boldsymbol{M}^{-1}\boldsymbol{K}\right)^{-1} = \left(\boldsymbol{K}^{-1}\boldsymbol{M}\right)^1$

$$\begin{cases} X_{rs}(-1) = \psi_r^T \left(\boldsymbol{M}\boldsymbol{K}^{-1}\right)^1 \boldsymbol{M}\,\psi_s & = \delta_{rs}\left(\omega_s^2\right)^{-1} M_s \\ \ldots \\ X_{rs}(-n) = \psi_r^T \left(\boldsymbol{M}\boldsymbol{K}^{-1}\right)^n \boldsymbol{M}\,\psi_s & = \delta_{rs}\left(\omega_s^2\right)^{-n} M_s \end{cases}$$

finally

$$X_{rs}(k) = \delta_{rs}\omega_s^{2k} M_s \quad \text{for } k = -\infty, \ldots, \infty.$$

## Flexibility



Given a system whose state is determined by the generalized displacements $x_j$ of a set of nodes, we define the flexibility coefficient $f_{jk}$ as the deflection, in direction of $x_j$, due to the application of a unit force in correspondance of the displacement $x_k$.

The matrix $\boldsymbol{F} = \left[f_{jk}\right]$ is the *flexibility matrix*.

In general, the dynamic degrees of freedom correspond to the points where there is

- application of external forces and/or
- presence of inertial forces.

Given a load vector $\boldsymbol{p} = \left\{p_k\right\}$, the displacemententent $x_j$ is

$$x_j = \sum f_{jk} p_k$$

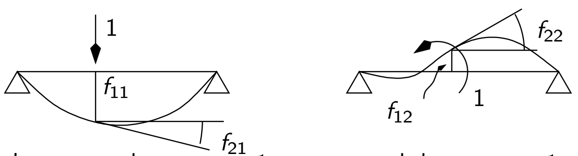
or, in vector notation,

$$\boldsymbol{x} = \boldsymbol{F}\,\boldsymbol{p}$$

## Example




The dynamical system    The degrees of freedom




Displacements due to $p_1 = 1$    and due to $p_2 = 1$.

# Elastic Forces



Momentarily disregarding inertial effects, each node shall be in equilibrium under the action of the external forces and the elastic forces, hence taking into accounts all the nodes, all the external forces and all the elastic forces it is possible to write the vector equation of equilibrium

$$\boldsymbol{p} = \boldsymbol{f}_S$$

and, substituting in the previos vector expression of the displacements

$$\boldsymbol{x} = \boldsymbol{F}\,\boldsymbol{f}_S$$

# Stiffness Matrix



The *stiffness matrix* $\boldsymbol{K}$ can be simply defined as the inverse of the flexibility matrix $\boldsymbol{F}$,

$$\boldsymbol{K} = \boldsymbol{F}^{-1}.$$

As an alternative definition, consider an unary vector of displacements,

$$\boldsymbol{e}^{(i)} = \left\{\delta_{ij}\right\}, \qquad j = 1, \ldots, N,$$

and the vector of nodal forces $\boldsymbol{k}_i$ that, applied to the structure, produces the displacements $\boldsymbol{e}^{(i)}$

$$\boldsymbol{F}\,\boldsymbol{k}_i = \boldsymbol{e}^{(i)}, \qquad i = 1, \ldots, N.$$

# Stiffness Matrix

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Collecting all the ordered $\boldsymbol{e}^{(i)}$ in a matrix $\boldsymbol{E}$, it is clear that $\boldsymbol{E} \equiv \boldsymbol{I}$ and we have, writing all the equations at once,

$$\boldsymbol{F}\left[\boldsymbol{k}_i\right] = \left[\boldsymbol{e}^{(i)}\right] = \boldsymbol{E} = \boldsymbol{I}.$$

Collecting the ordered force vectors in a matrix $\boldsymbol{K} = \left[\vec{k}_i\right]$ we have

$$\boldsymbol{F}\boldsymbol{K} = \boldsymbol{I}, \qquad \Rightarrow \qquad \boldsymbol{K} = \boldsymbol{F}^{-1},$$

giving a physical interpretation to the columns of the stiffness matrix.

Finally, writing the nodal equilibrium, we have

$$\boldsymbol{p} = \boldsymbol{f}_S = \boldsymbol{K}\,\boldsymbol{x}.$$

# Strain Energy



The elastic strain energy $V$ can be written in terms of displacements and external forces,

$$V = \frac{1}{2}\boldsymbol{p}^T\boldsymbol{x} = \frac{1}{2}\begin{cases}\boldsymbol{p}^T\underbrace{\boldsymbol{F}\,\boldsymbol{p}}_{\boldsymbol{x}},\\ \underbrace{\boldsymbol{x}^T\boldsymbol{K}}_{\boldsymbol{p}^T}\,\boldsymbol{x}.\end{cases}$$

Because the elastic strain energy of a stable system is always greater than zero, $\boldsymbol{K}$ is a positive definite matrix.

On the other hand, for an unstable system, think of a compressed beam, there are displacement patterns that are associated to zero strain energy.

# Symmetry

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Two sets of loads $\boldsymbol{p}^A$ and $\boldsymbol{p}^B$ are applied, one after the other, to an elastic system; the work done is

$$V_{AB} = \frac{1}{2}\boldsymbol{p}^{A^T}\boldsymbol{x}^A + \boldsymbol{p}^{A^T}\boldsymbol{x}^B + \frac{1}{2}\boldsymbol{p}^{B^T}\boldsymbol{x}^B.$$

If we revert the order of application the work is

$$V_{BA} = \frac{1}{2}\boldsymbol{p}^{B^T}\boldsymbol{x}^B + \boldsymbol{p}^{B^T}\boldsymbol{x}^A + \frac{1}{2}\boldsymbol{p}^{A^T}\boldsymbol{x}^A.$$

The total work being independent of the order of loading,

$$\boldsymbol{p}^{A^T}\boldsymbol{x}^B = \boldsymbol{p}^{B^T}\boldsymbol{x}^A.$$

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Expressing the displacements in terms of $\boldsymbol{F}$,

$$\boldsymbol{p}^{A^T}\boldsymbol{F}\,\boldsymbol{p}^B = \boldsymbol{p}^{B^T}\boldsymbol{F}\boldsymbol{p}^A,$$

both terms are scalars so we can write

$$\boldsymbol{p}^{A^T}\boldsymbol{F}\,\boldsymbol{p}^B = \left(\boldsymbol{p}^{B^T}\boldsymbol{F}\boldsymbol{p}^A\right)^T = \boldsymbol{p}^{A^T}\boldsymbol{F}^T\,\boldsymbol{p}^B.$$

Because this equation holds for every $\boldsymbol{p}$, we conclude that

$$\boldsymbol{F} = \boldsymbol{F}^T.$$

The inverse of a symmetric matrix is symmetric, hence

$$\boldsymbol{K} = \boldsymbol{K}^T.$$

## A practical consideration



For the kind of *structures* we mostly deal with in our examples, problems, exercises and assignments, that is *simple structures*, it is usually convenient to compute first the flexibility matrix applying the Principle of Virtual Displacements and later the stiffness matrix, using inversion,

$$\boldsymbol{K} = \boldsymbol{F}^{-1}.$$

On the other hand, the PVD approach cannot work in practice for *real structures*, because the number of degrees of freedom necessary to model the structural behaviour exceeds our ability to apply the PVD...
The stiffness matrix for large, complex structures to construct different methods required are.
E.g., the Finite Element Method.

## *FEM*

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The most common procedure to construct the matrices that describe the behaviour of a complex system is the *Finite Element Method*, or *FEM*.
The procedure can be sketched in the following terms:

- the structure is subdivided in non-overlapping portions, the *finite elements*, bounded by *nodes*, connected by the same nodes,
- the state of the structure can be described in terms of a vector $\boldsymbol{x}$ of generalized *nodal displacements*,
- there is a mapping between element and structure *DOF*'s, $i_{\text{el}} \mapsto r$,
- the *element stiffness matrix*, $\boldsymbol{K}_{\text{el}}$ establishes a linear relation between an element nodal displacements and forces,
- for each *FE*, all local $k_{ij}$'s are contributed to the global stiffness $k_{rs}$'s, with $i \mapsto r$ and $j \mapsto s$, taking in due consideration differences between local and global systems of reference.

Note that in the $r$-th *global* equation of equilibrium we have internal forces caused by the nodal displacements of the *FE* that have nodes $i_{\text{el}}$ such that $i_{\text{el}} \mapsto r$, thus implying that global $\boldsymbol{K}$ is a *banded* matrix.

## Example

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Consider a 2-D inextensible beam element, that has 4 *DOF*, namely two transverse end displacements $x_1$, $x_2$ and two end rotations, $x_3$, $x_4$. The element stiffness is computed using 4 shape functions $\phi_i$, the transverse displacement being $v(s) = \sum_i \phi_i(s)\, x_i$, $0 \leqslant s \leqslant L$, the different $\phi_i$ are such all end displacements or rotation are zero, except the one corresponding to index $i$.
The shape functions for a beam are

$$\phi_1(s) = 1 - 3\left(\frac{s}{L}\right)^2 + 2\left(\frac{s}{L}\right)^3, \qquad \phi_2(s) = 3\left(\frac{s}{L}\right)^2 - 2\left(\frac{s}{L}\right)^3,$$

$$\phi_3(s) = \left(\frac{s}{L}\right) - 2\left(\frac{s}{L}\right)^2 + \left(\frac{s}{L}\right)^3 \qquad \phi_4(s) = -\left(\frac{s}{L}\right)^2 + \left(\frac{s}{L}\right)^3.$$

# Example, 2



The element stiffness coefficients can be computed using, what else, the PVD: we compute the external virtual work done by a variation $\delta x_i$ by the force due to a unit displacement $x_j$, that is $k_{ij}$,

$$\delta W_{\text{ext}} = \delta x_i \, k_{ij},$$

the virtual internal work is the work done by the variation of the curvature, $\delta x_i \phi_i''(s)$ by the bending moment associated with a unit $x_j$, $\phi_j''(s) EJ(s)$,

$$\delta W_{\text{int}} = \int_0^L \delta x_i \phi_i''(s) \phi_j''(s) EJ(s) \, \mathrm{d}s.$$

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The equilibrium condition is the equivalence of the internal and external virtual works, so that simplifying $\delta x_i$ we have

$$k_{ij} = \int_0^L \phi_i''(s) \phi_j''(s) EJ(s) \, \mathrm{d}s.$$

For $EJ = \text{const}$,

$$\boldsymbol{f}_S = \frac{EJ}{L^3} \begin{bmatrix} 12 & -12 & 6L & 6L \\ -12 & 12 & -6L & -6L \\ 6L & -6L & 4L^2 & 2L^2 \\ 6L & -6L & 2L^2 & 4L^2 \end{bmatrix} \boldsymbol{x}$$

# Blackboard Time!

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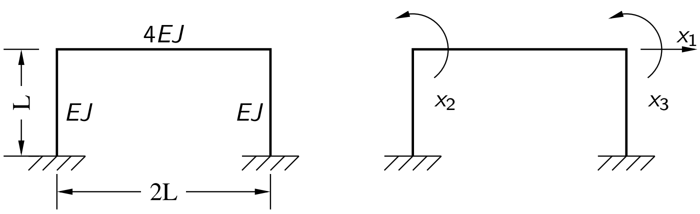

# Mass Matrix



The mass matrix maps the nodal accelerations to nodal inertial forces, and the most common assumption is to concentrate all the masses in nodal point masses, without rotational inertia, computed *lumping* a fraction of each element mass (or a fraction of the supported mass) on all its bounding nodes.
This procedure leads to a so called *lumped* mass matrix, a diagonal matrix with diagonal elements greater than zero for all the translational degrees of freedom and diagonal elements equal to zero for angular degrees of freedom.

# Mass Matrix



The mass matrix is definite positive *only* if all the structure *DOF*'s are translational degrees of freedom, otherwise $\boldsymbol{M}$ is semi-definite positive and the eigenvalue procedure is not directly applicable. This problem can be overcome either by using a *consistent* mass matrix or using the *static condensation* procedure.

# Consistent Mass Matrix



A consistent mass matrix is built using the rigorous *FEM* procedure, computing the nodal reactions that equilibrate the distributed inertial forces that develop in the element due to a linear combination of inertial forces.
Using our beam example as a reference, consider the inertial forces associated with a single nodal acceleration $\ddot{x}_j$, $f_{\mathrm{I},j}(s) = m(s)\phi_j(s)\ddot{x}_j$ and denote with $m_{ij}\ddot{x}_j$ the reaction associated with the $i$-nth degree of freedom of the element, by the PVD

$$\delta\, x_i m_{ij} \ddot{x}_j = \int \delta\, x_i \phi_i(s) m(s) \phi_j(s)\, \mathrm{d}s\, \ddot{x}_j$$

simplifying

$$m_{ij} = \int m(s)\phi_i(s)\phi_j(s)\, \mathrm{d}s.$$

For $m(s) = \overline{m} = \text{const}$.

$$\boldsymbol{f}_\mathrm{I} = \frac{\overline{m}L}{420}\begin{bmatrix} 156 & 54 & 22L & -13L \\ 54 & 156 & 13L & -22L \\ 22L & 13L & 4L^2 & -3L^2 \\ -13L & -22L & -3L^2 & 4L^2 \end{bmatrix} \ddot{\boldsymbol{x}}$$

## Consistent Mass Matrix, 2





### Pro

- some convergence theorem of *FEM* theory holds only if the mass matrix is consistent,
- sligtly more accurate results,
- no need for static condensation.

### Contra

- $\boldsymbol{M}$ is no more diagonal, heavy computational aggravation,
- static condensation is computationally beneficial, inasmuch it *reduces* the global number of degrees of freedom.

## Damping Matrix





For each element $c_{ij} = \int c(s)\phi_i(s)\phi_j(s)\,\mathrm{d}s$ and the damping matrix $\boldsymbol{C}$ can be assembled from element contributions.
However, using the FEM $\boldsymbol{C}^\star = \boldsymbol{\Psi}^T\boldsymbol{C}\,\boldsymbol{\Psi}$ is not diagonal and the modal equations are no more uncoupled!
The alternative is to write directly the global damping matrix, in terms of the underdetermined coefficients $\mathfrak{c}_b$,

$$\boldsymbol{C} = \sum_b \mathfrak{c}_b \boldsymbol{M}\left(\boldsymbol{M}^{-1}\boldsymbol{K}\right)^b.$$

## Damping Matrix





With our definition of $\boldsymbol{C}$,

$$\boldsymbol{C} = \sum_b \mathfrak{c}_b \boldsymbol{M}\left(\boldsymbol{M}^{-1}\boldsymbol{K}\right)^b,$$

assuming normalized eigenvectors, we can write the individual component of $\boldsymbol{C}^\star = \boldsymbol{\Psi}^T\boldsymbol{C}\,\boldsymbol{\Psi}$

$$c^\star_{ij} = \psi_i^T\boldsymbol{C}\,\psi_j = \delta_{ij}\sum_b \mathfrak{c}_b\omega_j^{2b}$$

due to the additional orthogonality relations, we recognize that now $\boldsymbol{C}^\star$ is a diagonal matrix.
Introducing the modal damping $C_j$ we have

$$C_j = \psi_j^T\boldsymbol{C}\,\psi_j = \sum_b \mathfrak{c}_b\omega_j^{2b} = 2\zeta_j\omega_j$$

and we can write a system of linear equations in the $\mathfrak{c}_b$.

## Example

We want a fixed, 5% damping ratio for the first three modes, taking note that the modal equation of motion is

$$\ddot{q}_i + 2\zeta_i\omega_i\dot{q}_i + \omega_i^2 q_i = p_i^\star$$

Using

$$\boldsymbol{C} = \mathfrak{c}_0\boldsymbol{M} + \mathfrak{c}_1\boldsymbol{K} + \mathfrak{c}_2\boldsymbol{K}\boldsymbol{M}^{-1}\boldsymbol{K}$$

we have

$$2\times 0.05\begin{Bmatrix}\omega_1\\ \omega_2\\ \omega_3\end{Bmatrix} = \begin{bmatrix}1 & \omega_1^2 & \omega_1^4\\ 1 & \omega_2^2 & \omega_2^4\\ 1 & \omega_3^2 & \omega_3^4\end{bmatrix}\begin{Bmatrix}\mathfrak{c}_0\\ \mathfrak{c}_1\\ \mathfrak{c}_2\end{Bmatrix}$$

Solving for the $\mathfrak{c}$'s and substituting above, the resulting damping matrix is orthogonal to every eigenvector of the system, for the first three modes, leads to a modal damping ratio that is equal to 5%.

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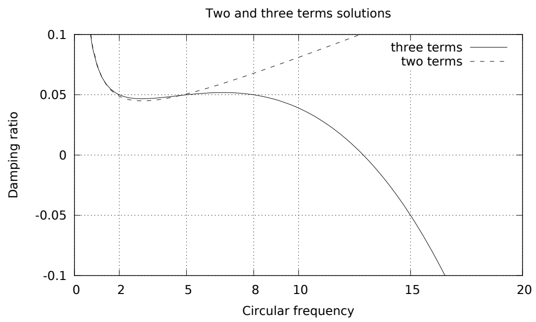
## Example

Computing the coefficients $\mathfrak{c}_0$, $\mathfrak{c}_1$ and $\mathfrak{c}_2$ to have a 5% damping at frequencies $\omega_1 = 2$, $\omega_2 = 5$ and $\omega_3 = 8$ we have $\mathfrak{c}_0 = 1200/9100$, $\mathfrak{c}_1 = 159/9100$ and $\mathfrak{c}_2 = -1/9100$.

Writing $\zeta(\omega) = \frac{1}{2}\left(\frac{\mathfrak{c}_0}{\omega} + \mathfrak{c}_1\omega + \mathfrak{c}_2\omega^3\right)$ we can plot the above function, along with its two term equivalent ($\mathfrak{c}_0 = 10/70, \mathfrak{c}_1 = 1/70$).




Negative damping? No, thank you: use only an even number of terms.

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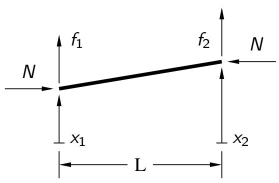
## Geometric Stiffness

A common assumption is based on a linear approximation, for a beam element

$$\boldsymbol{f}_G = \frac{N}{L}\begin{bmatrix}+1 & -1 & 0 & 0\\ -1 & +1 & 0 & 0\\ 0 & 0 & 0 & 0\\ 0 & 0 & 0 & 0\end{bmatrix}\boldsymbol{x}$$




$f_2 = -f_1$

$f_1 L = N(x_2 - x_1)$

It is possible to compute the geometrical stiffness matrix using *FEM*, shape functions and PVD,

$$k_{G,ij} = \int N(s)\phi_i'(s)\phi_j'(s)\,\mathrm{d}s,$$

for constant $N$

$$\boldsymbol{K}_G = \frac{N}{30L}\begin{bmatrix}36 & -36 & 3L & 3L\\ -36 & 36 & -3L & -3L\\ 3L & -3L & 4L^2 & -L^2\\ 3L & -3L & -L^2 & 4L^2\end{bmatrix}$$

## External Loadings



Following the same line of reasoning that we applied to find nodal inertial forces, by the PVD and the use of shape functions we have

$$p_i(t) = \int p(s,t)\phi_i(s)\,\mathrm{d}s.$$

For a constant, uniform load $p(s,t) = \overline{p} = \text{const}$, applied on a beam element,

$$\boldsymbol{p} = \overline{p}L\left\{\tfrac{1}{2} \quad \tfrac{1}{2} \quad \tfrac{L}{12} \quad -\tfrac{L}{12}\right\}^T$$



## Choice of Property Formulation



### Simplified Approach

Some structural parameter is approximated, only translational *DOF*'s are retained in dynamic analysis.

### Consistent Approach

All structural parameters are computed according to the *FEM*, and all *DOF*'s are retained in dynamic analysis.

If we choose a simplified approach, we must use a procedure to remove unneeded structural *DOF*'s from the model that we use for the dynamic analysis.
Enter the *Static Condensation Method*.

## Static Condensation



We have, from a *FEM* analysis, a stiffnes matrix that uses all nodal *DOF*'s, and from the lumped mass procedure a mass matrix were only translational (and maybe a few rotational) *DOF*'s are blessed with a non zero diagonal term. In this case, we can always rearrange

and partition the displacement vector $\boldsymbol{x}$ in two subvectors: *a*) $\boldsymbol{x}_A$, all the *DOF*'s that are associated with inertial forces and *b*) $\boldsymbol{x}_B$, all the remaining *DOF*'s not associated with inertial forces.

$$\boldsymbol{x} = \left\{\boldsymbol{x}_A \quad \boldsymbol{x}_B\right\}^T$$

## Static Condensation, 2

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Example

After rearranging the *DOF*'s, we must rearrange also the rows (equations) and the columns (force contributions) in the structural matrices, and eventually partition the matrices so that

$$\begin{Bmatrix}\boldsymbol{f}_I\\ \boldsymbol{0}\end{Bmatrix} = \begin{bmatrix}\boldsymbol{M}_{AA} & \boldsymbol{M}_{AB}\\ \boldsymbol{M}_{BA} & \boldsymbol{M}_{BB}\end{bmatrix}\begin{Bmatrix}\ddot{\boldsymbol{x}}_A\\ \ddot{\boldsymbol{x}}_B\end{Bmatrix}$$
$$\boldsymbol{f}_S = \begin{bmatrix}\boldsymbol{K}_{AA} & \boldsymbol{K}_{AB}\\ \boldsymbol{K}_{BA} & \boldsymbol{K}_{BB}\end{bmatrix}\begin{Bmatrix}\boldsymbol{x}_A\\ \boldsymbol{x}_B\end{Bmatrix}$$

with

$$\boldsymbol{M}_{BA} = \boldsymbol{M}_{AB}^T = \boldsymbol{0}, \quad \boldsymbol{M}_{BB} = \boldsymbol{0}, \quad \boldsymbol{K}_{BA} = \boldsymbol{K}_{AB}^T$$

Finally we rearrange the loadings vector and write...

## Static Condensation, 3

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... the equation of dynamic equilibrium,

$$\begin{aligned}\boldsymbol{p}_A &= \boldsymbol{M}_{AA}\ddot{\boldsymbol{x}}_A + \boldsymbol{M}_{AB}\ddot{\boldsymbol{x}}_B + \boldsymbol{K}_{AA}\boldsymbol{x}_A + \boldsymbol{K}_{AB}\boldsymbol{x}_B\\ \boldsymbol{p}_B &= \boldsymbol{M}_{BA}\ddot{\boldsymbol{x}}_A + \boldsymbol{M}_{BB}\ddot{\boldsymbol{x}}_B + \boldsymbol{K}_{BA}\boldsymbol{x}_A + \boldsymbol{K}_{BB}\boldsymbol{x}_B\end{aligned}$$

The terms in red are zero, so we can simplify

$$\begin{aligned}\boldsymbol{M}_{AA}\ddot{\boldsymbol{x}}_A + \boldsymbol{K}_{AA}\boldsymbol{x}_A + \boldsymbol{K}_{AB}\boldsymbol{x}_B &= \boldsymbol{p}_A\\ \boldsymbol{K}_{BA}\boldsymbol{x}_A + \boldsymbol{K}_{BB}\boldsymbol{x}_B &= \boldsymbol{p}_B\end{aligned}$$

solving for $\boldsymbol{x}_B$ in the 2nd equation and substituting

$$\boldsymbol{x}_B = \boldsymbol{K}_{BB}^{-1}\boldsymbol{p}_B - \boldsymbol{K}_{BB}^{-1}\boldsymbol{K}_{BA}\boldsymbol{x}_A$$
$$\boldsymbol{p}_A - \boldsymbol{K}_{AB}\boldsymbol{K}_{BB}^{-1}\boldsymbol{p}_B = \boldsymbol{M}_{AA}\ddot{\boldsymbol{x}}_A + \left(\boldsymbol{K}_{AA} - \boldsymbol{K}_{AB}\boldsymbol{K}_{BB}^{-1}\boldsymbol{K}_{BA}\right)\boldsymbol{x}_A$$

## Static Condensation, 4



Going back to the homogeneous problem, with obvious positions we can write

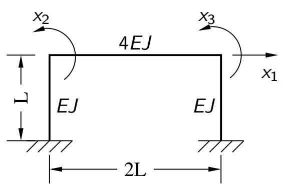
$$\left(\overline{\boldsymbol{K}} - \omega^2 \overline{\boldsymbol{M}}\right) \psi_A = \boldsymbol{0}$$

but the $\psi_A$ are only part of the structural eigenvectors, because in essentially every application we must consider also the other *DOF*'s, so we write

$$\psi_i = \begin{Bmatrix} \psi_{A,i} \\ \psi_{B,i} \end{Bmatrix}, \quad \text{with } \psi_{B,i} = \boldsymbol{K}_{BB}^{-1} \boldsymbol{K}_{BA} \psi_{A,i}$$

## Example






$$\boldsymbol{K} = \frac{2EJ}{L^3} \begin{bmatrix} 12 & 3L & 3L \\ 3L & 6L^2 & 2L^2 \\ 3L & 2L^2 & 6L^2 \end{bmatrix}$$

Disregarding the factor $2EJ/L^3$,

$$\boldsymbol{K}_{BB} = L^2 \begin{bmatrix} 6 & 2 \\ 2 & 6 \end{bmatrix}, \boldsymbol{K}_{BB}^{-1} = \frac{1}{32L^2} \begin{bmatrix} 6 & -2 \\ -2 & 6 \end{bmatrix}, \boldsymbol{K}_{AB} = \begin{bmatrix} 3L & 3L \end{bmatrix}$$

The matrix $\overline{\boldsymbol{K}}$ is

$$\overline{\boldsymbol{K}} = \frac{2EJ}{L^3} \left(12 - \boldsymbol{K}_{AB} \boldsymbol{K}_{BB}^{-1} \boldsymbol{K}_{AB}^T\right) = \frac{39EJ}{2L^3}$$