

<h1 style="background-color: #d9e1f2; padding: 5px;">Generalized Single Degree of Freedom Systems</h1> <p style="margin-top: 20px;">Giacomo Boffi</p> <p style="margin-top: 10px;"><a href="http://intranet.dica.polimi.it/people/boffi-giacomo">http://intranet.dica.polimi.it/people/boffi-giacomo</a></p> <p style="margin-top: 10px;">Dipartimento di Ingegneria Civile Ambientale e Territoriale Politecnico di Milano</p> <p style="margin-top: 10px;">March 22, 2017</p>	<p style="text-align: center;">Generalized SDOF's</p> <p style="text-align: center;">Giacomo Boffi</p> <p>Introductory Remarks</p> <p>Assemblage of Rigid Bodies</p> <p>Continuous Systems</p> <p>Vibration Analysis by Rayleigh's Method</p> <p>Selection of Mode Shapes</p> <p>Refinement of Rayleigh's Estimates</p>
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<h2 style="margin: 0;">Outline</h2>	<p style="text-align: center;">Generalized SDOF's</p> <p style="text-align: center;">Giacomo Boffi</p> <p>Introductory Remarks</p> <p>Assemblage of Rigid Bodies</p> <p>Continuous Systems</p> <p>Vibration Analysis by Rayleigh's Method</p> <p>Selection of Mode Shapes</p> <p>Refinement of Rayleigh's Estimates</p>
<p style="color: #4a7ebb; margin-top: 20px;">Introductory Remarks</p> <p style="color: #4a7ebb; margin-top: 15px;">Assemblage of Rigid Bodies</p> <p style="color: #4a7ebb; margin-top: 15px;">Continuous Systems</p> <p style="color: #4a7ebb; margin-top: 15px;">Vibration Analysis by Rayleigh's Method</p> <p style="color: #4a7ebb; margin-top: 15px;">Selection of Mode Shapes</p> <p style="color: #4a7ebb; margin-top: 15px;">Refinement of Rayleigh's Estimates</p>	

<h2 style="margin: 0;">Introductory Remarks</h2>	<p style="text-align: center;">Generalized SDOF's</p> <p style="text-align: center;">Giacomo Boffi</p> <p>Introductory Remarks</p> <p>Assemblage of Rigid Bodies</p> <p>Continuous Systems</p> <p>Vibration Analysis by Rayleigh's Method</p> <p>Selection of Mode Shapes</p> <p>Refinement of Rayleigh's Estimates</p>
<p>Until now our <i>SDOF</i>'s were described as composed by a single mass connected to a fixed reference by means of a spring and a damper. While the mass-spring is a useful representation, many different, more complex systems can be studied as <i>SDOF</i> systems, either exactly or under some simplifying assumption.</p> <ol style="list-style-type: none"> <li>1. <i>SDOF</i> rigid body assemblages, where the flexibility is concentrated in a number of springs and dampers, can be studied, e.g., using the Principle of Virtual Displacements and the D'Alembert Principle.</li> <li>2. simple structural systems can be studied, in an approximate manner, assuming a fixed pattern of displacements, whose amplitude (the single degree of freedom) varies with time.</li> </ol>	

<h2>Further Remarks on Rigid Assemblages</h2>	<p>Generalized SDOF's</p>
<p>Today we restrict our consideration to plane, 2-D systems. In rigid body assemblages the limitation to a single shape of displacement is a consequence of the configuration of the system, i.e., the disposition of supports and internal hinges. When the equation of motion is written in terms of a single parameter and its time derivatives, the terms that figure as coefficients in the equation of motion can be regarded as the <i>generalised</i> properties of the assemblage: generalised mass, damping and stiffness on left hand, generalised loading on right hand.</p> $m^* \ddot{x} + c^* \dot{x} + k^* x = p^*(t)$	<p>Giacomo Boffi</p> <p>Introductory Remarks</p> <p>Assemblage of Rigid Bodies</p> <p>Continuous Systems</p> <p>Vibration Analysis by Rayleigh's Method</p> <p>Selection of Mode Shapes</p> <p>Refinement of Rayleigh's Estimates</p>

<h2>Further Remarks on Continuous Systems</h2>	<p>Generalized SDOF's</p>
<p>Continuous systems have an infinite variety of deformation patterns. By restricting the deformation to a single shape of varying amplitude, we introduce an infinity of internal constraints that limit the infinite variety of deformation patterns, but under this assumption the system configuration is mathematically described by a single parameter, so that</p> <ul style="list-style-type: none"> <li>▶ our <i>model</i> can be analysed in exactly the same way as a strict <i>SDOF</i> system,</li> <li>▶ we can compute the <i>generalised</i> mass, damping, stiffness properties of the <i>SDOF model</i> of the continuous system.</li> </ul>	<p>Giacomo Boffi</p> <p>Introductory Remarks</p> <p>Assemblage of Rigid Bodies</p> <p>Continuous Systems</p> <p>Vibration Analysis by Rayleigh's Method</p> <p>Selection of Mode Shapes</p> <p>Refinement of Rayleigh's Estimates</p>

<h2>Final Remarks on Generalised SDOF Systems</h2>	<p>Generalized SDOF's</p>
<p>From the previous comments, it should be apparent that everything we have seen regarding the behaviour and the integration of the equation of motion of proper <i>SDOF</i> systems applies to rigid body assemblages and to <i>SDOF</i> models of flexible systems, provided that we have the means for determining the <i>generalised</i> properties of the dynamical systems under investigation.</p>	<p>Giacomo Boffi</p> <p>Introductory Remarks</p> <p>Assemblage of Rigid Bodies</p> <p>Continuous Systems</p> <p>Vibration Analysis by Rayleigh's Method</p> <p>Selection of Mode Shapes</p> <p>Refinement of Rayleigh's Estimates</p>

## Assemblages of Rigid Bodies

- ▶ planar, or bidimensional, rigid bodies, constrained to move in a plane,
- ▶ the flexibility is *concentrated* in discrete elements, springs and dampers,
- ▶ rigid bodies are connected to a fixed reference and to each other by means of springs, dampers and smooth, bilateral constraints (read hinges, double pendulums and rollers),
- ▶ inertial forces are distributed forces, acting on each material point of each rigid body, their resultant can be described by
  - ▶ a force applied to the centre of mass of the body, proportional to acceleration vector (of the centre of mass itself) and total mass  $M = \int dm$
  - ▶ a couple, proportional to angular acceleration and the moment of inertia  $J$  of the rigid body,  $J = \int (x^2 + y^2) dm$ .

Generalized SDOF's

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Introductory Remarks

Assemblage of Rigid Bodies

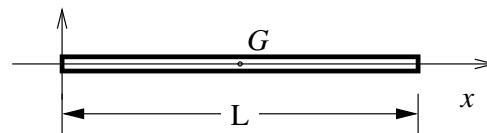
Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## Rigid Bar



Unit mass	$\bar{m} = \text{constant},$
Length	$L,$
Centre of Mass	$x_G = L/2,$
Total Mass	$m = \bar{m}L,$
Moment of Inertia	$J = m \frac{L^2}{12} = \bar{m} \frac{L^3}{12}$

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Introductory Remarks

Assemblage of Rigid Bodies

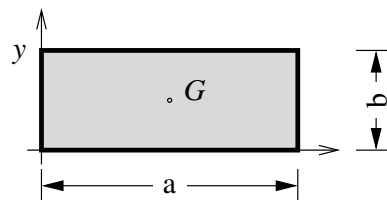
Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## Rigid Rectangle



Unit mass	$\gamma = \text{constant},$
Sides	$a, b$
Centre of Mass	$x_G = a/2, \quad y_G = b/2$
Total Mass	$m = \gamma ab,$
Moment of Inertia	$J = m \frac{a^2 + b^2}{12} = \gamma \frac{a^3 b + ab^3}{12}$

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Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

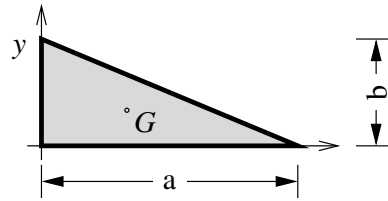
Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## Rigid Triangle

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For a right triangle.

Unit mass	$\gamma = \text{constant,}$
Sides	$a, b$
Centre of Mass	$x_G = a/3, \quad y_G = b/3$
Total Mass	$m = \gamma ab/2,$
Moment of Inertia	$J = m \frac{a^2 + b^2}{18} = \gamma \frac{a^3 b + ab^3}{36}$

Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

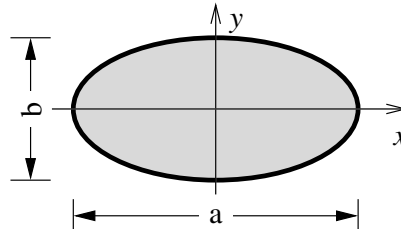
Refinement of Rayleigh's Estimates

## Rigid Oval

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When  $a = b = D = 2R$  the oval is a circle.



Unit mass	$\gamma = \text{constant,}$
Axes	$a, b$
Centre of Mass	$x_G = y_G = 0$
Total Mass	$m = \gamma \frac{\pi ab}{4},$
Moment of Inertia	$J = m \frac{a^2 + b^2}{16}$

Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

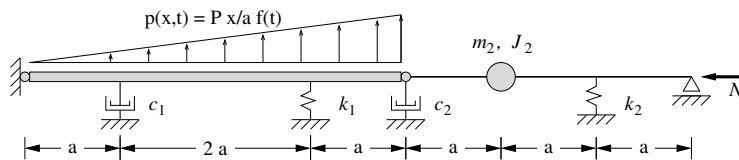
Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## trabacolo1

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The mass of the left bar is  $m_1 = \bar{m} 4a$  and its moment of inertia is  $J_1 = m_1 \frac{(4a)^2}{12} = 4a^2 m_1/3$ .

The maximum value of the external load is  $P_{\max} = P 4a/a = 4P$  and the resultant of triangular load is  $R = 4P \times 4a/2 = 8P a$

Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

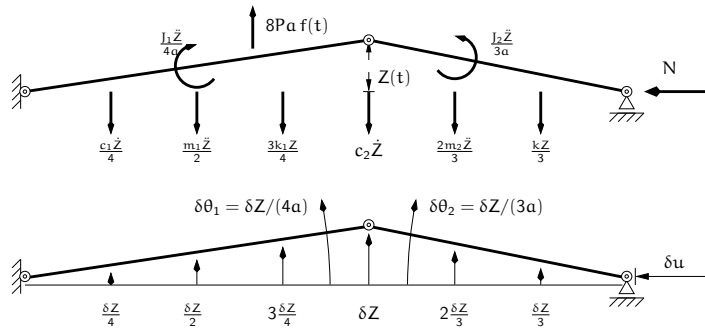
Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## Forces and Virtual Displacements

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$$u = 7a - 4a \cos \theta_1 - 3a \cos \theta_2, \quad \delta u = 4a \sin \theta_1 \delta \theta_1 + 3a \sin \theta_2 \delta \theta_2$$

$$\delta \theta_1 = \delta Z / (4a), \quad \delta \theta_2 = \delta Z / (3a)$$

$$\sin \theta_1 \approx Z / (4a), \quad \sin \theta_2 \approx Z / (3a)$$

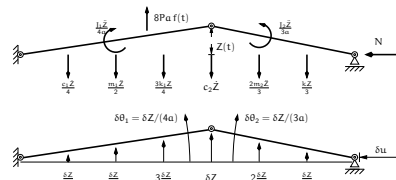
$$\delta u = \left( \frac{1}{4a} + \frac{1}{3a} \right) Z \delta Z = \frac{7}{12a} Z \delta Z$$

Introductory Remarks  
Assemblage of Rigid Bodies  
Continuous Systems  
Vibration Analysis by Rayleigh's Method  
Selection of Mode Shapes  
Refinement of Rayleigh's Estimates

## Principle of Virtual Displacements

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The virtual work of the Inertial Damping Elastic External forces:

$$\delta W_I = -m_1 \frac{\ddot{Z}}{2} \frac{\delta Z}{2} - J_1 \frac{\ddot{Z}}{4a} \frac{\delta Z}{4a} - m_2 \frac{2\ddot{Z}}{3} \frac{2\delta Z}{3} - J_2 \frac{\ddot{Z}}{3a} \frac{\delta Z}{3a}$$

$$= - \left( \frac{m_1}{4} + 4 \frac{m_2}{9} + \frac{J_1}{16a^2} + \frac{J_2}{9a^2} \right) \ddot{Z} \delta Z$$

$$\delta W_D = -c_1 \frac{\dot{Z}}{4} \frac{\delta Z}{4} - c_2 Z \delta Z = -(c_2 + c_1/16) \dot{Z} \delta Z$$

$$\delta W_S = -k_1 \frac{3Z}{4} \frac{3\delta Z}{4} - k_2 \frac{Z}{3} \frac{\delta Z}{3} = - \left( \frac{9k_1}{16} + \frac{k_2}{9} \right) Z \delta Z$$

$$\delta W_{Ext} = 8Pa f(t) \frac{2\delta Z}{3} + N \frac{7}{12a} Z \delta Z$$

Introductory Remarks  
Assemblage of Rigid Bodies  
Continuous Systems  
Vibration Analysis by Rayleigh's Method  
Selection of Mode Shapes  
Refinement of Rayleigh's Estimates

## Principle of Virtual Displacements

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For a rigid body in condition of equilibrium the total virtual work must be equal to zero

$$\delta W_I + \delta W_D + \delta W_S + \delta W_{Ext} = 0$$

Substituting our expressions of the virtual work contributions and simplifying  $\delta Z$ , the equation of equilibrium is

$$\left( \frac{m_1}{4} + 4 \frac{m_2}{9} + \frac{J_1}{16a^2} + \frac{J_2}{9a^2} \right) \ddot{Z} + (c_2 + c_1/16) \dot{Z} + \left( \frac{9k_1}{16} + \frac{k_2}{9} \right) Z = 8Pa f(t) \frac{2}{3} + N \frac{7}{12a} Z$$

Introductory Remarks  
Assemblage of Rigid Bodies  
Continuous Systems  
Vibration Analysis by Rayleigh's Method  
Selection of Mode Shapes  
Refinement of Rayleigh's Estimates

## Principle of Virtual Displacements

Collecting  $Z$  and its time derivatives give us

$$m^* \ddot{Z} + c^* \dot{Z} + k^* Z = p^* f(t)$$

introducing the so called *generalised properties*, in our example it is

$$m^* = \frac{1}{4} m_1 + \frac{4}{9} m_2 + \frac{1}{16a^2} J_1 + \frac{1}{9a^2} J_2,$$

$$c^* = \frac{1}{16} c_1 + c_2,$$

$$k^* = \frac{9}{16} k_1 + \frac{1}{9} k_2 - \frac{7}{12a} N,$$

$$p^* = \frac{16}{3} Pa.$$

It is worth writing down the expression of  $k^*$ :

$$k^* = \frac{9k_1}{16} + \frac{k_2}{9} - \frac{7}{12a} N$$

Geometrical stiffness

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Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

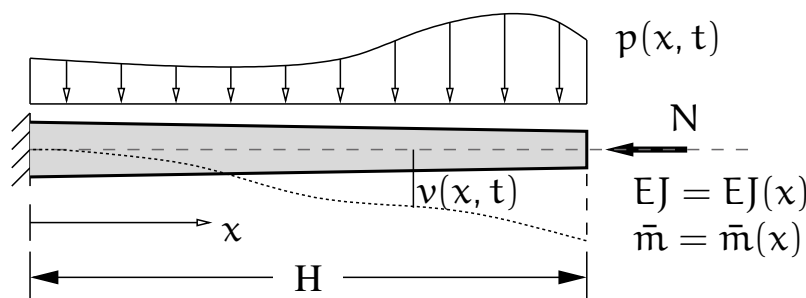
## Let's start with an example...

Consider a cantilever, with varying properties  $\bar{m}$  and  $EJ$ , subjected to a load that is function of both time  $t$  and position  $x$ ,

$$p = p(x, t).$$

The transverse displacements  $v$  will be function of time and position,

$$v = v(x, t)$$



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Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## ... and an hypothesis

To study the previous problem, we introduce an *approximate model* by the following hypothesis,

$$v(x, t) = \Psi(x) Z(t),$$

that is, the hypothesis of *separation of variables*

Note that  $\Psi(x)$ , the *shape function*, is adimensional, while  $Z(t)$  is dimensionally a generalised displacement, usually chosen to characterise the structural behaviour.

In our example we can use the displacement of the tip of the chimney, thus implying that  $\Psi(H) = 1$  because

$$Z(t) = v(H, t) \quad \text{and}$$

$$v(H, t) = \Psi(H) Z(t)$$

Generalized SDOF's

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Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## Principle of Virtual Displacements

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For a flexible system, the PoVD states that, at equilibrium,

$$\delta W_E = \delta W_I.$$

The virtual work of external forces can be easily computed, the virtual work of internal forces is usually approximated by the virtual work done by bending moments, that is

$$\delta W_I \approx \int M \delta \chi$$

where  $\chi$  is the curvature and  $\delta \chi$  the virtual increment of curvature.

Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## $\delta W_E$

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The external forces are  $p(x, t)$ ,  $N$  and the forces of inertia  $f_I$ ; we have, by separation of variables, that  $\delta v = \Psi(x)\delta Z$  and we can write

$$\delta W_p = \int_0^H p(x, t) \delta v \, dx = \left[ \int_0^H p(x, t) \Psi(x) \, dx \right] \delta Z = p^*(t) \delta Z$$

$$\begin{aligned} \delta W_{\text{Inertia}} &= \int_0^H -\bar{m}(x) \ddot{v} \delta v \, dx = \int_0^H -\bar{m}(x) \Psi(x) \ddot{Z} \Psi(x) \, dx \delta Z \\ &= \left[ \int_0^H -\bar{m}(x) \Psi^2(x) \, dx \right] \ddot{Z}(t) \delta Z = m^* \ddot{Z} \delta Z. \end{aligned}$$

The virtual work done by the axial force deserves a separate treatment...

Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## $\delta W_N$

Generalized SDOF's

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The virtual work of  $N$  is  $\delta W_N = N \delta u$  where  $\delta u$  is the variation of the vertical displacement of the top of the chimney.

We start computing the vertical displacement of the top of the chimney in terms of the rotation of the axis line,  $\phi \approx \Psi'(x)Z(t)$ ,

$$u(t) = H - \int_0^H \cos \phi \, dx = \int_0^H (1 - \cos \phi) \, dx,$$

substituting the well known approximation  $\cos \phi \approx 1 - \frac{\phi^2}{2}$  in the above equation we have

$$u(t) = \int_0^H \frac{\phi^2}{2} \, dx = \int_0^H \frac{\Psi'^2(x) Z^2(t)}{2} \, dx$$

hence

$$\delta u = \int_0^H \Psi'^2(x) Z(t) \delta Z \, dx = \int_0^H \Psi'^2(x) \, dx \, Z \delta Z$$

and

$$\delta W_N = \left[ \int_0^H \Psi'^2(x) \, dx \, N \right] Z \delta Z = k_G^* Z \delta Z$$

Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## $\delta W_{Int}$

Approximating the internal work with the work done by bending moments, for an infinitesimal slice of beam we write

$$dW_{Int} = \frac{1}{2} M v''(x, t) dx = \frac{1}{2} M \Psi''(x) Z(t) dx$$

with  $M = EJ(x)v''(x)$

$$\delta(dW_{Int}) = EJ(x)\Psi''^2(x)Z(t)\delta Z dx$$

integrating

$$\delta W_{Int} = \left[ \int_0^H EJ(x)\Psi''^2(x) dx \right] Z\delta Z = k^* Z\delta Z$$

Generalized SDOF's

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Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## Remarks

- ▶ the shape function *must* respect the geometrical boundary conditions of the problem, i.e., both

$$\Psi_1 = x^2 \quad \text{and} \quad \Psi_2 = 1 - \cos \frac{\pi x}{2H}$$

are acceptable shape functions for our example, as  $\Psi_1(0) = \Psi_2(0) = 0$  and  $\Psi_1'(0) = \Psi_2'(0) = 0$

- ▶ better results are obtained when the second derivative of the shape function at least *resembles* the typical distribution of bending moments in our problem, so that between

$$\Psi_1'' = \text{constant} \quad \text{and} \quad \Psi_2'' = \frac{\pi^2}{4H^2} \cos \frac{\pi x}{2H}$$

the second choice is preferable.

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Introductory Remarks

Assemblage of Rigid Bodies

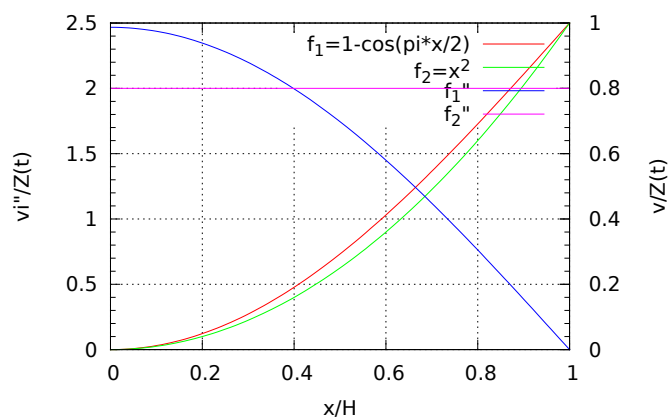
Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## Remarks



Generalized SDOF's

Giacomo Boffi

Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates



## Example

Using  $\Psi(x) = 1 - \cos \frac{\pi x}{2H}$ , with  $\bar{m} = \text{constant}$  and  $EJ = \text{constant}$ , with a load characteristic of seismic excitation,  $p(t) = -\bar{m}\ddot{v}_g(t)$ ,

$$m^* = \bar{m} \int_0^H \left(1 - \cos \frac{\pi x}{2H}\right)^2 dx = \bar{m} \left(\frac{3}{2} - \frac{4}{\pi}\right)H$$

$$k^* = EJ \frac{\pi^4}{16H^4} \int_0^H \cos^2 \frac{\pi x}{2H} dx = \frac{\pi^4 EJ}{32 H^3}$$

$$k_G^* = N \frac{\pi^2}{4H^2} \int_0^H \sin^2 \frac{\pi x}{2H} dx = \frac{\pi^2 N}{8H}$$

$$p_g^* = -\bar{m}\ddot{v}_g(t) \int_0^H 1 - \cos \frac{\pi x}{2H} dx = -\left(1 - \frac{2}{\pi}\right) \bar{m}H \ddot{v}_g(t)$$

Generalized  
SDOF's

Giacomo Boffi

Introductory  
Remarks

Assemblage of  
Rigid Bodies

Continuous  
Systems

Vibration  
Analysis by  
Rayleigh's  
Method

Selection of  
Mode Shapes

Refinement of  
Rayleigh's  
Estimates

## Vibration Analysis

- ▶ The process of estimating the vibration characteristics of a complex system is known as *vibration analysis*.
- ▶ We can use our previous results for flexible systems, based on the *SDOF* model, to give an estimate of the natural frequency  $\omega^2 = k^*/m^*$
- ▶ A different approach, proposed by Lord Rayleigh, starts from different premises to give the same results but the *Rayleigh's Quotient* method is important because it offers a better understanding of the vibrational behaviour, eventually leading to successive refinements of the first estimate of  $\omega^2$ .

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SDOF's

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Introductory  
Remarks

Assemblage of  
Rigid Bodies

Continuous  
Systems

Vibration  
Analysis by  
Rayleigh's  
Method

Selection of  
Mode Shapes

Refinement of  
Rayleigh's  
Estimates

## Rayleigh's Quotient Method

Our focus will be on the *free vibration* of a flexible, undamped system.

- ▶ inspired by the free vibrations of a proper *SDOF* we write

$$\begin{aligned} Z(t) &= Z_0 \sin \omega t & \text{and} \\ v(x, t) &= Z_0 \Psi(x) \sin \omega t, \\ \dot{v}(x, t) &= \omega Z_0 \Psi(x) \cos \omega t. \end{aligned}$$

- ▶ the displacement and the velocity are in quadrature: when  $v$  is at its maximum  $\dot{v} = 0$ , hence  $V = V_{\max}$ ,  $T = 0$  and when  $\dot{v}$  is at its maximum it is  $v = 0$ , hence  $V = 0$ ,  $T = T_{\max}$ ,
- ▶ disregarding damping, the energy of the system is constant during free vibrations,

$$V_{\max} + 0 = 0 + T_{\max}$$

Generalized  
SDOF's

Giacomo Boffi

Introductory  
Remarks

Assemblage of  
Rigid Bodies

Continuous  
Systems

Vibration  
Analysis by  
Rayleigh's  
Method

Selection of  
Mode Shapes

Refinement of  
Rayleigh's  
Estimates

## Rayleigh's Quotient Method

Generalized SDOF's

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Now we write the expressions for  $V_{\max}$  and  $T_{\max}$ ,

$$V_{\max} = \frac{1}{2} Z_0^2 \int_S EJ(x) \Psi''^2(x) dx,$$
$$T_{\max} = \frac{1}{2} \omega^2 Z_0^2 \int_S \bar{m}(x) \Psi^2(x) dx,$$

equating the two expressions and solving for  $\omega^2$  we have

$$\omega^2 = \frac{\int_S EJ(x) \Psi''^2(x) dx}{\int_S \bar{m}(x) \Psi^2(x) dx}.$$

Recognizing the expressions we found for  $k^*$  and  $m^*$  we could question the utility of Rayleigh's Quotient...

Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## Rayleigh's Quotient Method

Generalized SDOF's

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- ▶ in Rayleigh's method we know the specific time dependency of the inertial forces

$$f_I = -\bar{m}(x)\ddot{v} = \bar{m}(x)\omega^2 Z_0 \Psi(x) \sin \omega t$$

$f_I$  has the same *shape* we use for displacements.

- ▶ if  $\Psi$  were the real shape assumed by the structure in free vibrations, the displacements  $v$  due to a loading  $f_I = \omega^2 \bar{m}(x) \Psi(x) Z_0$  should be proportional to  $\Psi(x)$  through a constant factor, with equilibrium respected in every point of the structure during free vibrations.
- ▶ starting from a shape function  $\Psi_0(x)$ , a new shape function  $\Psi_1$  can be determined normalizing the displacements due to the inertial forces associated with  $\Psi_0(x)$ ,  $f_I = \bar{m}(x) \Psi_0(x)$ ,
- ▶ we are going to demonstrate that the new shape function is a better approximation of the true mode shape

Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## Selection of mode shapes

Generalized SDOF's

Giacomo Boffi

Given different shape functions  $\Psi_i$  and considering the true shape of free vibration  $\Psi$ , in the former cases equilibrium is not respected by the structure itself.

To keep inertia induced deformation proportional to  $\Psi_i$  we must consider the presence of additional elastic constraints. This leads to the following considerations

- ▶ the frequency of vibration of a structure with additional constraints is higher than the true natural frequency,
- ▶ the criterium to discriminate between different shape functions is: better shape functions give lower estimates of the natural frequency, the true natural frequency being a lower bound of all estimates.

Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## Selection of mode shapes 2

Generalized SDOF's

Giacomo Boffi

Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

**Selection of Mode Shapes**

Refinement of Rayleigh's Estimates

In general the selection of trial shapes goes through two steps,

1. the analyst considers the flexibilities of different parts of the structure and the presence of symmetries to devise an approximate shape,
2. the structure is loaded with constant loads directed as the assumed displacements, the displacements are computed and used as the shape function,

of course a little practice helps a lot in the the choice of a proper pattern of loading...

## Selection of mode shapes 3

Generalized SDOF's

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Introductory Remarks

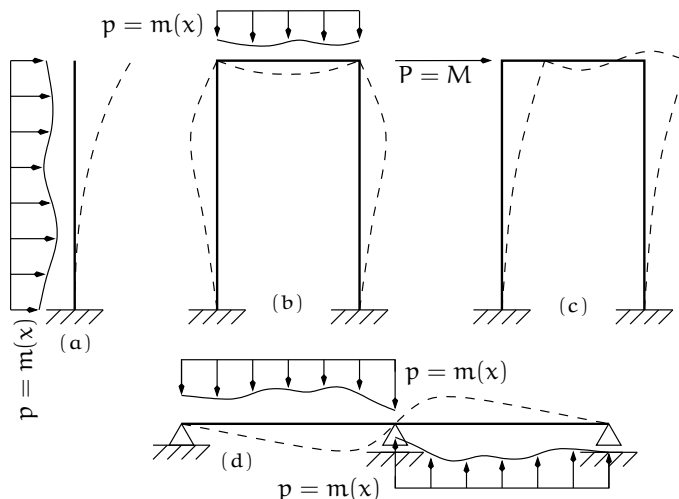
Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

**Selection of Mode Shapes**

Refinement of Rayleigh's Estimates



## Refinement $R_{00}$

Generalized SDOF's

Giacomo Boffi

Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

**Selection of Mode Shapes**

Refinement of Rayleigh's Estimates

Choose a trial function  $\Psi^{(0)}(x)$  and write

$$v^{(0)} = \Psi^{(0)}(x)Z^{(0)} \sin \omega t$$

$$V_{\max} = \frac{1}{2}Z^{(0)2} \int EJ\Psi^{(0)''2} dx$$

$$T_{\max} = \frac{1}{2}\omega^2 Z^{(0)2} \int \bar{m}\Psi^{(0)2} dx$$

our first estimate  $R_{00}$  of  $\omega^2$  is

$$\omega^2 = \frac{\int EJ\Psi^{(0)''2} dx}{\int \bar{m}\Psi^{(0)2} dx}$$

## Refinement $R_{01}$

We try to give a better estimate of  $V_{\max}$  computing the external work done by the inertial forces,

$$p^{(0)} = \omega^2 \bar{m}(x) v^{(0)} = Z^{(0)} \omega^2 \Psi^{(0)}(x)$$

the deflections due to  $p^{(0)}$  are

$$v^{(1)} = \omega^2 \frac{v^{(0)}}{\omega^2} = \omega^2 \Psi^{(1)} \frac{Z^{(1)}}{\omega^2} = \omega^2 \Psi^{(1)} \bar{Z}^{(1)},$$

where we write  $\bar{Z}^{(1)}$  because we need to keep the unknown  $\omega^2$  in evidence. The maximum strain energy is

$$V_{\max} = \frac{1}{2} \int p^{(0)} v^{(1)} dx = \frac{1}{2} \omega^4 Z^{(0)} \bar{Z}^{(1)} \int \bar{m}(x) \Psi^{(0)} \Psi^{(1)} dx$$

Equating to our previous estimate of  $T_{\max}$  we find the  $R_{01}$  estimate

$$\omega^2 = \frac{Z^{(0)} \int \bar{m}(x) \Psi^{(0)} \Psi^{(0)} dx}{\bar{Z}^{(1)} \int \bar{m}(x) \Psi^{(0)} \Psi^{(1)} dx}$$

Generalized SDOF's

Giacomo Boffi

Introductory Remarks

Assemblage of Rigid Bodies

Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## Refinement $R_{11}$

With little additional effort it is possible to compute  $T_{\max}$  from  $v^{(1)}$ :

$$T_{\max} = \frac{1}{2} \omega^2 \int \bar{m}(x) v^{(1)2} dx = \frac{1}{2} \omega^6 \bar{Z}^{(1)2} \int \bar{m}(x) \Psi^{(1)2} dx$$

equating to our last approximation for  $V_{\max}$  we have the  $R_{11}$  approximation to the frequency of vibration,

$$\omega^2 = \frac{Z^{(0)} \int \bar{m}(x) \Psi^{(0)} \Psi^{(1)} dx}{\bar{Z}^{(1)} \int \bar{m}(x) \Psi^{(1)} \Psi^{(1)} dx}$$

Of course the procedure can be extended to compute better and better estimates of  $\omega^2$  but usually the refinements are not extended beyond  $R_{11}$ , because it would be contradictory with the quick estimate nature of the Rayleigh's Quotient method and also because  $R_{11}$  estimates are usually very good ones.

Generalized SDOF's

Giacomo Boffi

Introductory Remarks

Assemblage of Rigid Bodies

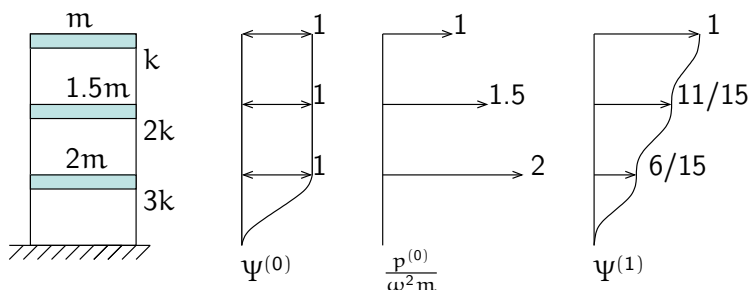
Continuous Systems

Vibration Analysis by Rayleigh's Method

Selection of Mode Shapes

Refinement of Rayleigh's Estimates

## Refinement Example



$$T = \frac{1}{2} \omega^2 \times 4.5 \times m Z_0$$

$$V = \frac{1}{2} \times 1 \times 3k Z_0$$

$$\omega^2 = \frac{3}{9/2} \frac{k}{m} = \frac{2}{3} \frac{k}{m}$$

$$v^{(1)} = \frac{15}{4} \frac{m}{k} \omega^2 \Psi^{(1)}$$

$$\bar{Z}^{(1)} = \frac{15}{4} \frac{m}{k}$$

$$V^{(1)} = \frac{1}{2} m \frac{15}{4} \frac{m}{k} \omega^4 (1 + 33/30 + 4/5)$$

$$= \frac{1}{2} m \frac{15}{4} \frac{m}{k} \omega^4 \frac{87}{30}$$

$$\omega^2 = \frac{\frac{9}{2} m}{m \frac{87}{8} \frac{m}{k}} = \frac{12}{29} \frac{k}{m} = 0.4138 \frac{k}{m}$$