Flutter Analysis of Slender Beams with Variable Cross Sections Based on Integral Equation Formulation

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Abstract—This paper studies a mathematical model based on the integral equations for dynamic analyzes numerical investigations of a non-uniform or multi-material composite beam. The beam is subjected to a sub-tangential follower force and elastic foundation. The boundary conditions are represented by generalized parameterized fixations by the linear and rotary springs. The non-uniform section introduces non-uniformity in the rigidity and inertia of beams and consequently, more complicated equilibrium who governs the equation. Using the boundary element method and radial basis functions, the equation of motion is reduced to an algebro-differential system related to internal and boundary unknowns. A generalized formula for the deflection, the slope, the moment and the shear force are presented. The free vibration of non-uniform loaded beams is formulated using a compact matrix form and all needed matrices are explicitly given. The dynamic stability analysis of slender beam is illustrated numerically based on the coalescence criterion. A realistic case related to an industrial chimney is investigated.

Keywords—Chimney, BEM and integral equation formulation, non uniform cross section, vibration and Flutter.

I. INTRODUCTION

The interest in non-uniform beam due to cross variable section or to non uniformity material has been widespread in recent year due to the structural advantage of such form in some system. The free vibration problem has been studied by a lot of researchers. However, for a non uniform beam, even without any other parameters, the researches on their dynamic behaviors are relatively fewer [1], [2]. For designers and manufacturing systems, the rectangular and the circular cross sections are usually used. As to the free vibration and the dynamic analyses of those non uniform beams carrying multiple parameters (concentrated foundations or masses, fixations, etc.), the information concerned is rare and this is one of the reasons why the problem in this aspect is studied. Literature relating to the free vibration analysis of a non-uniform beam is rare [3]. The numerical assembly method presented in [4], takes the case with any number of concentrated attachments. Instability phenomenon of non-uniformbeam is inexist. Amongst the numerical methods available for thin structure problems, the finite element method is undoubtedly the most versatile. The only problem with this method is that its formula is quite laborious and it takes a large amount of computer storage. A powerful alternative method based on integral equations is the Boundary Element Method (BEM) [5], [6]. The main reason for rapid development of the BEM is a dimensionality reduction possibility of the problem, leading to a reduced set of equations and a smaller amount of data required to calculate. For a part of problem, is using the fundamental solution corresponding to the exact solution, inappropriate terms are moved to the right hand side of governing equation and considered as a fictitious source density. For dynamic instability of a non uniform beams under elastic foundations and sub-tangential follower forces, domain integrals are necessary in formulation. Thus, the main advantage of dimensionality reduction is eliminated. But, the use of Dual Reciprocity Method (DRM), introduced by Nardini and Brebio [7], permits to combine the dimensionality reduction advantage with a simple fundamental solution and to formulate the problem on boundary unknowns only. A comprehensive literature review of the DRM and Multiple Reciprocity Method (MRM) as applied to elasto-dynamics can be found in [8]. Karimi Khorramadi and A.R. Nezamabadi [9] give a studies free vibration of functionally graded beams subjected to axial load that is simply supported at both ends lies on a continuous elastic foundation. Displacement field of beam is assumed based on Engesser-Timoshenko beam theory. The Young’s modulus of beam is assumed to be graded continuously across the beam thickness.

In this paper, we present a general formula based on integral equation formulation for the investigation of dynamic analysis (buckling, vibration, dynamic instability, ...) of a non uniform slender beam with a generalized fixations submitted to an elastic foundation and to a follower force.

II. BASIC BEAM EQUATIONS

Based on the Euler-Bernoulli theory for slender beams and neglecting the axial displacement, the equation of motion is formulated using the transverse displacement only. The fixations are generalized by the introduction of linear and rotary springs at the two ends of the beams as presented in Fig. 1. The partial differential equation governing the motion of beams subjected to sub-tangential force, lateral excitation and boundary generalized fixations is formulated by:

\[ \frac{\partial^2}{\partial t^2} \left[ E(z) \frac{\partial^2 V(z,t)}{\partial z^2} \right] + \rho(z) \left( \frac{\partial^2 V(z,t)}{\partial t^2} \right) + \kappa(z) \frac{\partial V(z,t)}{\partial z} + \lambda(z) \frac{\partial^2 V(z,t)}{\partial z^2} - F(z,t) = 0 \]
where \( V \) is the transverse displacement, \( E, I, S \) and \( \rho \) are the Young's modulus, the inertia, the area and the mass density respectively. \( \lambda \) is the sub-tangential load, \( \kappa \) is the elastic foundation, \( p(z,t) \) is the lateral excitation and \( z \) is the axial coordinate. The boundary fixations will be discussed later. For a slender beam with a non uniform cross section, the section \( S(z) \) and the inertia \( I(z) \) are \( z \)-absicissa dependent. Splitting the inertia and the section into constant and variable parts, the following equations may be used:

\[
E I(z) = E I(0)(1 + KI(z)) \quad \text{and} \quad \rho S(z) = \rho S(0) K2(z)
\]

(2)

where \( S(0) \) and \( I(0) \) are the section and the inertia at the end \((z=0)\), \( KI(z) \) and \( K2(z) \) depend on the type of section considered and translate the inertia and section variation along the beam. Remember that the function \( KI(z) \) depends also on the sense of displacement considered and particularly when the section is rectangular.

**Fig. 1** A non uniform beam with circular section, linear variation and generalized fixations

Equation (1) is a partial differential equation with variable coefficients. The principal part of this equation is the rigidity term \( \frac{\partial^2 E I(z)}{\partial z^2} \left( \frac{\partial^2 V(z,t)}{\partial z^2} \right) \) which can be decomposed using (2) as:

\[
\frac{\partial^2 E I(z)}{\partial z^2} \left( \frac{\partial^2 V(z,t)}{\partial z^2} \right) = EI(0) \frac{\partial^2}{\partial z^2} \left( KI(z) \frac{\partial^2 V(z,t)}{\partial z^2} \right) + EI(0) \frac{\partial^2}{\partial z^2} \left( \frac{\partial^2 V(z,t)}{\partial z^2} \right)
\]

\[
(3)
\]

Then, we obtain the following adimensional equation:

\[
\frac{\partial^2 W}{\partial \tau^2} + \frac{\partial^2 W}{\partial x^2} \left[ KI(x) \frac{\partial^2 W}{\partial x^2} + K2(x) \frac{\partial^2 W}{\partial x^2} \right] = p^* \frac{\partial^2 W}{\partial x^2}
\]

(4)

where

\[
\chi^2 = \frac{\rho S(0) L^4}{EI(0)}, \quad \kappa^* = \kappa \frac{L^4}{EI(0)}, \quad \lambda^* = \frac{\lambda L^2}{EI(0)}, \quad p^* = \frac{p L^3}{EI(0)};
\]

\[
W(x,\tau) = \frac{V(z,t)}{R}, \quad \tau = \frac{t}{X}, \quad R = \sqrt{\frac{\tau}{\chi}} \quad \text{and} \quad x = \frac{z}{L}.
\]

\( R \) is the radius of gyration of the beam, \( 0 \leq x \leq 1 \) and \( \dot{\cdot} \) indicates time derivative.

The aim of this paper is the development of integral equation formulations for numerical solutions of (4) with generalized boundary conditions and dynamical behaviors of non-uniform slender beams subjected to various types of loads.

**A. Cross Section Considered**

In manufacturing engineering beam systems, the case of a constant cross section is very limited and variable cross section offers various performances that are very useful. Most cross sections used in manufacturing beam systems are circular or rectangular. Non uniformity of the section introduces a non uniformity in rigidity and inertia of beams that leads to intricate governing equilibrium equation. In this study, we limit ourselves to beams with circular cross sections and constant young modulus and mass density. This type of section is simply characterized by its radius or diameter. The sense of transverse displacement has no effect in this type of section. For a parabolic variation (Fig. 2), diameter and inertia of section are given by:

\[
d(x) = d_0 \left( r_x x^2 + r_y x + I \right),
\]

\[
d(0) = d_0, \quad d(0.5) = d_2 \quad \text{and} \quad d(1) = d_1
\]

(5)

\[
I(x) = I_0 \left[ I - KI(x) \right]
\]

(6)

where

\[
KI(x) = \left( r_1 x^3 + r_2 x + 1 \right)^2 - 1; \quad K2(x) = \left( r_1 x^2 + r_2 x + 1 \right)^3;
\]

\[
r_2 = \frac{d_1 - d_2}{d_0} \quad \text{and} \quad r_1 = \frac{d_1 - d_2}{d_0} - \frac{d_1 - d_2}{d_0} - 3
\]

(7)

In case of a convex \((d_0 = d_1; \; d_0 < d_2)\) or a concave \((d_0 = d_1; \; d_0 > d_2)\) section, we have:

\[
r = r_1 = -r_2 = \frac{d_1}{d_0} - 1
\]

Inertia and mass functions are reduced to:

\[
KI(x) = \left( r x^2 + r x + 1 \right)^2 - 1 \quad \text{and} \quad K2(x) = \left( r x^2 - r x + 1 \right)^2
\]

(8)

An emphasis is on a particular section corresponding to a circular one with linear variation (Figs. 3 (a) and (b)) obtained for \( d_2 = \frac{1}{2}(d_1 + d_0) \). For this type of section, \( r_1 = 0 \) and \( r_2 = \frac{d_1}{d_0} - 1 \). Inertia and section functions are then given by:

\[
KI(x) = (r x + 1)^4 - 1 \quad \text{and} \quad K2(x) = (r x^2 + r x + 1)^2, \quad r = \frac{d_1}{d_0}
\]

(9)
B. Integral Equation Formulation

Equation (4) can be directly treated by Integral Equation Formulation when fundamental solution of the operator

$$\frac{\partial^2}{\partial x^2} \left( K(x) \frac{\partial^2}{\partial x^2} \right)$$

is analytically known. Unfortunately, this fundamental solution is known only for a few cases of simplified functions $K(x)$. In this study, only fundamental solution of the simplified operator will be used. Let us denote $W^*$ fundamental solution of the following problem:

$$\frac{\partial^4 W^*(x,s)}{\partial x^4} = 0$$

where $\delta$ is Dirac function and $s$ is the source point. This fundamental solution will be used and the partial differential equation (4) will be transformed into an integral equation. Following the boundary element procedure [5], [6], resulting integral equation will be reduced to an algebro-differential equation that will be solved by the dual reciprocity method.

Multiplying (4) by $W^*$ and integrating from 0 to 1, we obtain:

$$\int_0^1 \frac{\partial^4 W(x)}{\partial x^4} W^*(x,s)dx = \int_0^1 \frac{\partial^2}{\partial x^2} \left( K(x) \frac{\partial^2}{\partial x^2} \right) W^*(x,s)dx - \int_0^1 K(x) W^*(x,s)dx$$

$$\int_0^1 \kappa(x) W(x) W^*(x,s)dx - \lambda^2 \int_0^1 \frac{\partial^2 W}{\partial x^2} W^*(x,s)dx + \int_0^1 p(x) W^*(x,s)dx$$

(11)
\[ \int_0^s \frac{\partial^j W^*}{\partial x^j} (s, x) d\tau = W(s) + A_0(s) \]

\[ A_0(s) = \begin{bmatrix} W^* (s, x) \frac{\partial^j W^*}{\partial x^j} (s, x) \frac{\partial^j W^*}{\partial x^j} (s, x) \end{bmatrix} \]

\[ \theta(x) = \frac{\partial W^*}{\partial x} (s, x), M(x) = -K(s) \frac{\partial^3 W^*}{\partial x^3} (s, x) \text{ and } Q(x) = \frac{\partial M}{\partial x} \]

For a concentrated elastic foundation \( \kappa' \) at point \( x_0 \):

\[ \int_0^s \kappa' (s) ((W(x)) (x-x_0)) W^* (s, x) d\tau = \]

\[ L_j (s) = 0 \quad j \neq k \]

\[ p(s) = 0 \]

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\[ p(s) = 0 \]

For a compact equation representation, let us recall that in bending problem of beams, the following variable have physical meaning and may be known at boundaries.

\[ \theta(x) = \frac{\partial W^*}{\partial x} (s, x), M(x) = -K(s) \frac{\partial^3 W^*}{\partial x^3} (s, x) \text{ and } Q(x) = \frac{\partial M}{\partial x} \]

where \( \theta(x) \) is slop, Mo(x) is bending moment and Q(x) is shear force, related to derivatives of deflection \( W \). These physical parameters will be combined with W-equations and boundary conditions in order to get a well posed problem.

III. MATRIX FORMULATION

For simplified expressions, the following notations are introduced:

\[ j(s) = \frac{\partial^j}{\partial s^j} (s), \quad \tilde{j}(s) = K(s) \frac{\partial^3}{\partial s^3} (s) \text{ and } \tilde{j}(s) = \frac{\partial^3}{\partial s^3} (s) \]

Based on these notations, the integral equation formulation (11) is reduced to the following algebro-differential equation at interior points:

\[ W(s) + A_0(s) = \sum_{j=1}^{2s} \alpha_j M_1(s) (s) + \kappa' W, L_j (s) + \]

\[ p(s) \]

In order to present a well posed problem in its general formulation, more equations than (22) related to \( \theta, Mo \) and \( Q \)
are needed. They are obtained by the derivative of (22). The slope equation given by:

\[ \theta_i(s) + \dot{A}_0(s) = \sum_{j=1}^{n} \frac{1}{\alpha_j} \left( \frac{\partial}{\partial s} \dot{M}_j(s) + \kappa' W_j \dot{L}_j(s) + \right) + P_i \]  

(23)

For numerical results, discretization of beam is necessary. This discretization may be uniform or non uniform in order to improve the accuracy in case of a concentrated source of density. After discretization and using following notations, we get:

\[ W_i + A_0 = \sum_{j=1}^{n} \frac{1}{\alpha_j} \left( \frac{\partial}{\partial s} \dot{M}_j(s) + \kappa' W_j \dot{L}_j(s) + \right) + P_i \]

\[ \theta_i + \dot{A}_0 = \sum_{j=1}^{n} \frac{1}{\alpha_j} \left( \frac{\partial}{\partial s} \dot{M}_j(s) + \kappa' W_j \dot{L}_j(s) + \right) + P_i \]

where \( W_i = W(s_i) \), \( s_i = \frac{i}{n+1} \) and \( i = 1 \) and \( i = n + 2 \) correspond to the beam ends and \( i = 2 \) to \( n + 2 \) correspond to interior points. This system of equation contains both unknowns at the interior points and boundaries.

A. Generalized Boundary Conditions

A non uniform beam with circular cross section is considered. Fixations are generalized by the introduction of linear and rotary springs at the two ends of the beams. Linear fixations and follower force acts on the shear force whereas rotary fixations affect the moment. For the used sections, boundary conditions at beam ends \( i = 1 \) and \( i = n + 2 \) are [6], [10]:

\[ (l + K(l)) \frac{\partial^2 W}{\partial s^2}(0) = \kappa'' \theta_1, \]

\[ (l + K(l)) \frac{\partial^2 W}{\partial s^2} (l) = \kappa'' \theta_{n+2} \]

\[ \frac{d}{dx} \left( (l + K(x)) \frac{\partial^2 W}{\partial s^2} (s) \right) |_{s=0} = -\kappa'' W_{0} + (1 - y) \dot{\theta}_1 \]

\[ \frac{d}{dx} \left( (l + K(x)) \frac{\partial^2 W}{\partial s^2} (s) \right) |_{s=l} = -\kappa'' W_{n+2} + (1 - y) \dot{\theta}_{n+2} \]

as \( K(l) = 0 \), (25) can be rewritten as:

\[ \frac{\partial^2 W}{\partial s^2} (0) = \kappa'' \theta_1, \]

\[ \frac{\partial^2 W}{\partial s^2} (l) = \kappa'' \theta_{n+2} \]

\[ \frac{\partial^2 W}{\partial s^2} (0) = \kappa'' W_{0} + (1 - y) \dot{\theta}_1 \]

\[ \frac{\partial^2 W}{\partial s^2} (l) = \kappa'' W_{n+2} + (1 - y) \dot{\theta}_{n+2} \]

(26)

By putting (26) into (12), we get:

\[ A_0(s) = \left( \frac{\partial^2 W^*}{\partial s^2} (s, 0) + \kappa'' W^*(s, 0) \right) W_i + \]

\[ \left( \frac{\partial^2 W^*}{\partial s^2} (s, l) + \kappa'' W^*(s, l) \right) W_{n+2} + \]

\[ + \left( \frac{\partial^2 W^*}{\partial s^2} (s, 0) + \kappa'' W^*(s, 0) \right) + \]

\[ \kappa'' \frac{d K}{dx} (0) W^*(s, 0) \]

\[ + \kappa'' \frac{d K}{dx} (l) W^*(s, l) \]

\[ (1 - y) \dot{\theta}_1 \left( -W^*(s, 0) \theta_1 + W^*(s, l) \theta_{n+2} \right) \]

\[ + \kappa'' \frac{d K}{dx} (0) W^*(s, l) \]

\[ + \kappa'' \frac{d K}{dx} (l) \theta_{n+2} \]

(27)

By this way, the considered boundary conditions are taken into account in present formula. Equation (11) becomes more compact as:

\[ A_0(s) = \left( \frac{R_1(s) W_1(s) + R_2(s) W_{n+2} + \right. \]

\[ + \frac{1}{l + K(l)} \left( \frac{\partial^2 W^*}{\partial s^2} (s, 0) + \right. \]

\[ \left. \frac{\partial^2 W^*}{\partial s^2} (s, l) \right) \theta_1 + \frac{R_4(s)}{l + K(l)} \theta_{n+2} \]

\[ \tilde{A}_0(s) = \tilde{R}_1(s) + \tilde{R}_2(s) \tilde{F}_0(s) \]

(28)

Or in a discretized form:

\[ A_0_i = R_i + \tilde{R}_1(1 - y) \tilde{F}_0_i, \quad i = 1 \text{ to } n + 2 \]

\[ \tilde{A}_0_i = \tilde{R}_i + \tilde{R}_2(1 - y) \tilde{F}_0_i \]

(29)

By putting (29) into (24), leads to:

\[ W_i + R_i + \tilde{R}_2(1 - y) \tilde{F}_0_i = \sum_{j=1}^{n+4} \left( \frac{\partial^2 M_j(s)}{(l + K(l))} \right) \theta_{j}, \quad i = 1 \text{ to } n + 2 \]

\[ \theta_i + \tilde{R}_1(1 - y) \tilde{F}_0_i = \sum_{j=1}^{n+4} \left( \frac{\partial^2 M_j(s)}{(l + K(l))} \right) \theta_{j}, \quad i = 1 \text{ and } i = n + 2 \]

(30)

Finally, we have ‘\( n+2 \)’ equations by applying the first equation of (30) and ‘\( n+4 \)’ unknowns (‘\( n+2 \’ unkowns of \( W, \theta_1 \) and \( \theta_{n+2} \)’). The two additional equations can be explicitly expressed by applying the second equation of (30). The well posed system of equations is formulated by ‘\( n+4 \’ equations:
allows us to investigate buckling, vibration and flutter of beams with variable cross sections under generalized boundary conditions.

IV. EIGENVALUE PROCEDURE

A. Buckling Problem

Buckling problem may be formulated by assuming static displacements and omitting excitation and sub-tangential parameters in (32). The critical buckling loads and associated eigenmodes can be determined for various types of boundary conditions and concentrated elastic foundation \( \kappa \) by solving following eigenvalue problem:

\[
\begin{bmatrix}
X W_1 \\
X W_2 \\
\vdots \\
X W_n
\end{bmatrix} = \frac{1}{\lambda_i} \begin{bmatrix} W_1 \\ W_2 \\ \vdots \\ W_n \end{bmatrix}
\]

This allows the investigation of numerical critical buckling loads and associated eigenmodes at interior points. The corresponding slope, moment and shear force can be numerically computed using following relationships:

\[
[p] = -\left( \begin{bmatrix} \mathbf{R} \end{bmatrix} + \kappa' \begin{bmatrix} \mathbf{L} \end{bmatrix} + \begin{bmatrix} \mathbf{A} \end{bmatrix} \right) \begin{bmatrix} W_1 \\ W_2 \\ \vdots \\ W_n \end{bmatrix},
\]

\[
[M_0] = \left[ \text{diag}(1+K_1) \right] + \left[ \text{diag}(1+K_1) \right] \begin{bmatrix} \mathbf{R} \end{bmatrix} + \kappa' \begin{bmatrix} \mathbf{L} \end{bmatrix} + \begin{bmatrix} \mathbf{A} \end{bmatrix} \begin{bmatrix} W_1 \\ W_2 \\ \vdots \\ W_n \end{bmatrix}
\]

\[
[q] = \left[ \text{diag}(1+K_1) \right] + \left[ \text{diag}(1+K_1) \right] \begin{bmatrix} \mathbf{R} \end{bmatrix} + \kappa' \begin{bmatrix} \mathbf{L} \end{bmatrix} + \begin{bmatrix} \mathbf{A} \end{bmatrix} \begin{bmatrix} W_1 \\ W_2 \\ \vdots \\ W_n \end{bmatrix}
\]

In which \( \lambda_i \) is the \( i^{th} \) critical buckling load associated to the \( i^{th} \) buckling mode shape \( \{W_i\} \).

B. Flutter Problem

Let us assume that displacement field can be written in the following form:

\[
W(x,\tau) = W(x)e^{\omega^*\tau}, \quad \Omega^* = \omega^* + \alpha^* i
\]

Numerical solution of (32) permits, on one hand, to study linear vibration behaviors of beams by omitting the load parameter. On the other hand, linear vibration analysis of beams under axial compression and instability phenomenon can be investigated by numerically solving the following eigenvalue problem for each fixed load parameter:
Remember that matrix $[X]$ is $\lambda^*\text{ and } y^\prime$ dependent and is particularly not symmetric. Generalized boundary conditions are expressed by the $[R]$ matrix. The load-frequency dependence can be investigated and the divergence stability force) required for designers without more computation effort.

\[
[X] \{W\} = \frac{1}{\Omega^2} \{W\} \\
[X] = \left[ \begin{bmatrix} 1 & [F] & [L] & [\lambda][G] & + & \ldots & [F][G] & + (1-y)[F0] \end{bmatrix} \right] [M]
\]

\[
(37)
\]

A. Nonuniform Beam

1. Influence of the Elastic Fixation Parameters on the Natural Frequencies

Considering a beam with linear variation and circular cross section for which taper ratio is $r = d_0/d_0$ (Figs. 3 (a) and (b)). In practice, ideal fixation does not exist and then the parameterization of the fixations is more realistic. Based on the presented formulations, the five natural frequencies according to the linear and rotary stiffness coefficient $\kappa_{10}$ and $\kappa_{00}$ for the first end fixation ($x=0$) and $\kappa_{11}$ and $\kappa_{01}$ for the second end fixation ($x=l$) are given in Tables I and II. Obtained results by 40 internal points are well compared to [1] using Bessel functions. It is demonstrated from these tables that boundary stiffness fixations may largely influence vibration behaviors of the beam. Classical boundary conditions such as simply supported ($\kappa_{1}^* = \infty$ and $\kappa_{2}^* = 0$) and clamped ($\kappa_{1}^* = \infty$, $\kappa_{2}^* = \infty$) are also modeled by these fixations. These generalized boundary conditions are more realistic and allow us to investigate the imperfections in the boundary conditions. The stiffness parameters have a strong influence as shown in Tables I and II.

2. Influence of the Taper Ratio on the Natural Frequencies and the Mode Shapes

Influence of taper ratio ‘$r$’ on the mode shapes is shown in Figs. 4-9 for simply supported-clamped (C-S) beam and in Figs. 10-13 for a cantilever beam (C-F). The lowest five mode shapes for different values of ‘$r$’ are presented. We notice that amplitude of the mode decreases with increasing section. This is due to the fact that rigidity of beam increases with section. The influence of taper ratio ‘$r$’ on the first five natural frequencies is given in Table III for a C-S beam. Effect of ‘$r$’ tends to increase natural frequencies modulus of the C-S beam. Let us note that for $r > l$ there is a critical value $r_c$ of taper ratio for which the first and the second frequencies become complex conjugate. This means that for $r > r_c$, beam will flutter even if no charge acts on it. This critical value $r_c$, depending on the boundary conditions considered, is of great importance for a safe design.

B. Dynamic Instability of a Chimney

Our aim in this subsection is to give an industrial application of the present model. A case studied here corresponds to an industrial realistic chimney (OCP, Morocco) which is very long in order to drain the toxic gas at a very high altitude [11]. This chimney can be assimilated and approximated by a non uniform slender C-F tube with linear variation and constant thickness along the longitudinal axis. The corresponding approximated geometrical and material characteristics are given in Table IV. The vibration analysis of this chimney is investigated and the two first natural frequencies of the considered C-F chimney are given in Table V for various taper ratios $r$. It is demonstrated that there is a critical $r_c (r_c = 1.045)$ under which the chimney becomes...
instable and will flutter. From these results, it appears that a deep investigation of this critical value related to boundary conditions and geometrical parameters is necessary for safe design of chimneys. For flutter analysis, the chimney is assumed to be submitted to a sub-tangential follower force ($\gamma = 1$). Load-frequency dependence is investigated and flutter phenomenon is analyzed for different values of taper ratio '$r$' using the coalescence criterion. The obtained load frequency curves are presented in Figs. 14-18. These figures show a higher dependence between flutter load and taper ratio. The dynamic analysis of the chimney shows first that, for $r < r_c$ ($r_c = 1.045$, Table V), the flutter load increases with increasing taper ratio '$r$' as shown in Figs. 14-18. Chimney will flutter when the non-conservative load exceed flutter load value. For a uniform chimney ($r=1$), flutter curve is presented in Figs. 14-17 for comparison and flutter load is $\lambda^*_f \approx 16 \times 10^5 N$.

For $r=0.5$, the chimney will first diverge at $\lambda^*_j \approx 42 \times 10^5 N$ and then flutter at $\lambda^*_f \approx 81 \times 10^5 N$. For $r=0.1$, the flutter load $\lambda^*_f$ is $8 \times 10^3 N$. These values demonstrate that the chimney with a large basis is more stable than the one with a reduced basis. A deep analysis of dynamic behavior of this chimney is given in Fig. 18 for refined values of '$r$'. It is shown that around the critical value $r_c = 1.045$, dynamic behavior is strongly affected and in some cases the chimney has to be loaded in order to be stabilized.

VI. CONCLUSION

A methodological approach based on integral equation formulations for dynamic analysis of a non-uniform slender beam subjected to a sub-tangential follower force and generalized boundary conditions is presented in simple and compact forms. Required matrices are explicitly given and dynamic analysis is be investigated by solving the resulting eigenvalue problems. The coalescence criterion was used for flutter analysis and many numerical tests are performed. The effect of taper ratio on the natural frequencies, flutter loads and mode shapes is investigated for non-uniform beams and chimneys. The presented model is quite general and obtained results are in agreement with available data. These investigations demonstrated the effectiveness of the developed methodological approach.
Fig. 9 The lowest five mode shapes for a non uniform S-C beam for $r=2$

Fig. 10 The lowest five mode shapes for a non uniform F-C beam for $r=1$

Fig. 11 The lowest five mode shapes for a non uniform F-C beam for $r=1.5$

Fig. 12 The lowest five mode shapes for a non uniform F-C beam for $r=2$

Fig. 13 The lowest five mode shapes for a non uniform F-C beam for $r=1$

Fig. 14 The load-frequency curve for a non uniform slender C-F beam for $r=1$, $\omega_f=1.6 \times 10^6$ N

Fig. 15 The load-frequency curve for a non uniform slender C-F beam for $r=0.2$, $\omega_f=27 \times 10^6$ N
**Fig. 16** The load-frequency curve for a non uniform slender C-F beam for $r=0.5, \lambda_d = 4.2 \times 10^6 \, N, \lambda_f = 8.3 \times 10^6 \, N$.

**Fig. 17** The load-frequency curve for a non uniform slender C-F beam for $r=0.1, \lambda_d = 8.7 \times 10^6 \, N$.

**Fig. 18** The load-frequency curve for a non uniform slender C-F beam for different values of ratio $r$: (a): $r=1$, (b): $r=1.01$, (c): $r=1.02$, (d): $r=1.03$, (e): $r=1.04$, (f): $r=1.045$, (g): $r=1.056$, (h): $r=1.06$.

**TABLE I**

<table>
<thead>
<tr>
<th>$\omega^*, \Omega$</th>
<th>$\omega^*, \Omega$</th>
<th>$\omega^*, \Omega$</th>
<th>$\omega^*, \Omega$</th>
<th>$\omega^*, \Omega$</th>
<th>$\omega^*, \Omega$</th>
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<tbody>
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<td>$\kappa_r^0$</td>
<td>$\kappa_r^0$</td>
<td>$\kappa_r^0$</td>
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<td>$\kappa_r^0$</td>
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<td>13.91</td>
<td>14.04</td>
<td>9.73</td>
<td>9.10</td>
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<td>61.00</td>
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<td>45.25</td>
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<tr>
<td>3</td>
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<td>114.42</td>
<td>133.71</td>
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<td>142.32</td>
<td>126.53</td>
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</tr>
<tr>
<td>4</td>
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<tr>
<td>5</td>
<td>360.62</td>
<td>311.77</td>
<td>363.66</td>
<td>328.60</td>
<td>374.42</td>
<td>342.02</td>
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**TABLE II**

The Five Lowest Natural Frequencies ($\omega^*$) of a Non Uniform Circular Beam for $\kappa^*_r = \kappa^*_l = \infty$ and $r=1.4$, for Various Values of the Non Dimensional Rotational and Linear Stiffness Coefficients $\kappa_r^*$ and $\kappa_l^*$ Obtained by the Present Model ($n=40$) and Compared to [1]

<table>
<thead>
<tr>
<th>Natural Frequencies $\omega^*$</th>
<th>$\omega^*$</th>
<th>$\omega^*$</th>
<th>$\omega^*$</th>
<th>$\omega^*$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\kappa_r^0 = 0$</td>
<td>$\kappa_r^0 = \infty$</td>
<td>$\kappa_l^0 = 0$</td>
<td>$\kappa_l^0 = \infty$</td>
<td>$\kappa_r^0 = 10$</td>
</tr>
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<td>1</td>
<td>19.62</td>
<td>19.53</td>
<td>19.18</td>
<td>19.01</td>
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<tr>
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<td>59.95</td>
<td>55.95</td>
<td>55.62</td>
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<td>3</td>
<td>125.44</td>
<td>124.02</td>
<td>104.45</td>
<td>102.78</td>
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<td>211.24</td>
<td>165.38</td>
<td>164.96</td>
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<tr>
<td>5</td>
<td>320.55</td>
<td>246.83</td>
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<td>246.83</td>
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**TABLE III**

The Five Lowest Natural Frequencies ($\Omega$) of a Non Uniform Circular C-S Beam for Various Values of the Ratio $r$ ($\lambda^* = 0$)

<table>
<thead>
<tr>
<th>Natural Frequencies $\omega^*$</th>
<th>$\omega^*$</th>
<th>$\omega^*$</th>
<th>$\omega^*$</th>
<th>$\omega^*$</th>
<th>$\omega^*$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\omega^*$</td>
<td>$\omega^*$</td>
<td>$\omega^*$</td>
<td>$\omega^*$</td>
<td>$\omega^*$</td>
<td>$\omega^*$</td>
</tr>
<tr>
<td>$r=0.1$</td>
<td>$r=0.5$</td>
<td>$r=1$</td>
<td>$r=1.01$</td>
<td>$r=1.02$</td>
<td></td>
</tr>
<tr>
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<td>10.31</td>
<td>13.06</td>
<td>15.41i</td>
<td>1.92 + 33.55i</td>
<td>30.77 + 46.13 + 35.00i</td>
</tr>
<tr>
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<td>27.40i</td>
<td>40.93i</td>
<td>49.96i</td>
<td>1.92 - 33.55i</td>
<td>30.77 + 46.13 - 35.00i</td>
</tr>
<tr>
<td>3</td>
<td>53.01i</td>
<td>78.70i</td>
<td>104.24i</td>
<td>142.52i</td>
<td>48.63 + 193.33i + 77.64 + 218.04i</td>
</tr>
<tr>
<td>4</td>
<td>87.79i</td>
<td>138.25i</td>
<td>178.27i</td>
<td>184.68i</td>
<td>48.63 - 193.33i + 77.64 - 218.04i</td>
</tr>
<tr>
<td>5</td>
<td>130.65i</td>
<td>203.21i</td>
<td>272.03i</td>
<td>344.72i</td>
<td>52.78 + 93.72 + 465.65i</td>
</tr>
</tbody>
</table>

**Natural Frequencies $\omega^*$**

<table>
<thead>
<tr>
<th>$\omega^*$</th>
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<th>$\omega^*$</th>
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<tbody>
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<td>$\kappa_r^0 = 0$</td>
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</tbody>
</table>

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the general form used is:

Then we have:

Several types of radial basis functions $f_j(x)$ are tested and the general form used is:

The fundamental solution $W^*$ used in this analysis corresponds to

Several types of radial basis functions $f_j(x)$ are tested and the general form used is:

Then we have:

The other terms are null.

The other terms are null.

The other terms are null.

The other terms are null.

The other terms are null.

The other terms are null.

The other terms are null.

The other terms are null.
\[ [F_0] : \text{matrix } ((n+4) \times (n+4)), \]

\[
F_{0,i+1} = F_0(s_i) = -\hat{w}^*(s_i,0)\theta_i + \frac{\hat{w}^*(s_i,1)}{(1 + K(i))}\hat{p}_{i+2}, \\
i = 1, n + 2
\]

\[
F_{0,3,n+3} = \hat{F}_0(0) = -\hat{w}^*(0,0) + \frac{\hat{w}^*(0,1)}{(1 + K(i))},
\]

\[
F_{0,4,n+4} = \hat{F}_0(1) = -\hat{w}^*(1,0) + \frac{\hat{w}^*(1,1)}{(1 + K(i))},
\]

the other terms are null

\[ [P] : \text{vector } (1 \times (n+4)), \]

\[
P_i = P(s_{i-1}) \quad i = 1, n + 2
\]

\[
P_{i+3} = \hat{P}(0)
\]

\[
P_{i+4} = \hat{P}(1)
\]

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The authors wish to greatly thank the ‘Ministère de l’Enseignement Supérieur et de la Recherche Scientifique’ and the center ‘CNRST’ of Morocco and the center ‘CNRS’ of France for financial support from both projects: “PROTARS III (D11/22)” and “Action Intégrée (MA/05/17)”.

REFERENCES