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Non-linear curling of wide single-flange steel panels

N. Silvestre *

Department of Civil Engineering and Architecture, ICIST- IST, Technical University of Lisbon, Av. Rovisco Pais, 1049-001 Lisbon, Portugal

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ABSTRACT

An original analytical model to study the non-linear flange curling in wide single-flange panels is presented in this paper. Flange curling phenomenon is the tendency of the wide thin flanges (in compression or tension) to move towards the neutral axis, when thin-walled members are subjected to bending. Despite the simplicity of the formula developed in earlier works of Winter to account for the flange curling effects, which is used in current steel codes, recent work has showed that this expression is rather restrictive and does not apply for all cases. The analytical expressions reported here are rather general since they (i) consider the restraining effect provided by the web, (ii) account for the shift of the neutral axis due to curling, (iii) incorporate the decrement of the second moment of area due to curling, and (iv) are fully analytical, thus avoiding iterative techniques. The analytical model is applied to study the curling behaviour of profiled steel decks and cassette-wall panels and is validated by means of comparisons with experimental results available in the literature. Since the EC3 rules state that the tensioned wide flange in liner trays should be reduced if the curling displacement is higher than 5% of the web height, approximate expressions to evaluate the reduced width of the wide thin flange under curling are proposed.

1. Introduction

It is well known that thin-walled circular tubes subjected to pure bending are sensitive to the flattening of the cross-section. As the curvature increases, the bottom and top regions (the most stressed ones) of the cross-section tend to move towards the neutral axis, thus "ovalizing" the tube (see Fig. 1(a)). The growth of the ovalization causes a progressive reduction of the bending stiffness and, eventually, the moment reaches a maximum (limit) value. Since the earlier works of von Karman [1] and Brazier [2], much has been done on this subject — the tube ovalization is, nowadays, also designated by Brazier effect. The notable work of Brazier was based on a simple, but reliable, formulation to evaluate the evolution of the ovalization displacement with the bending curvature and moment. Using the strain energy due to bending and ovalization and applying the Rayleigh–Ritz method, Brazier derived the following expressions,

$$m = \pi\kappa(1 - 1.5\kappa^2) \qquad u = r\kappa^2 \qquad (1)$$
$$m = \frac{M}{M_0} \qquad M_0 = \frac{Ert^2}{\sqrt{1-\nu^2}}$$
$$\kappa = \frac{\chi}{\chi_0} \qquad \chi_0 = \frac{t}{r^2\sqrt{1-\nu^2}}$$

where (i) m and κ are the dimensionless bending moment and curvature, (ii) M_0 and χ_0 are the reference bending moment and curvature, (iii) u is the ovalization displacement, (iv) r and t are the tube radius and thickness and (v) E and ν are the material elastic constants. For a detailed account of these formulae, the interested reader is referred to recent works by Li [3], Karamanos [4] and Guarracino [5]. From expressions (1), it is noticed that the bending moment m displays a non-linear relation with the ovalization displacement u.

Following an independent path from Brazier's work, Winter [6] investigated the non-linear behaviour of wide flange box and I-section beams under bending and noticed that "..., *the flange is not only curved longitudinally in the loaded beam but, under the action of longitudinal bending stresses, also tends to curve in the direction perpendicular to the axis of the beam* ...". This phenomenon is designated by curling and, theoretically, stems from the tendency of wide thin flanges, in compression or tension,[1] to move towards the neutral axis — see Fig. 1(b). Using simple expressions from engineer's theory of structures, Winter developed the formula

$$u = 2 \cdot \left(\frac{\sigma}{E}\right)^2 \frac{b_s^4}{t^2 z} \qquad (2)$$

* Tel.: +351 218418410; fax: +351 21 8497650.
E-mail address: nunos@civil.ist.utl.pt.

[1] In reality, it seems obvious that the wide flange in compression exhibits an increasing loss of stiffness, compared with the tensioned one. In fact, wide flanges in compression are also sensitive to local buckling effects, but that is an issue not considered here.

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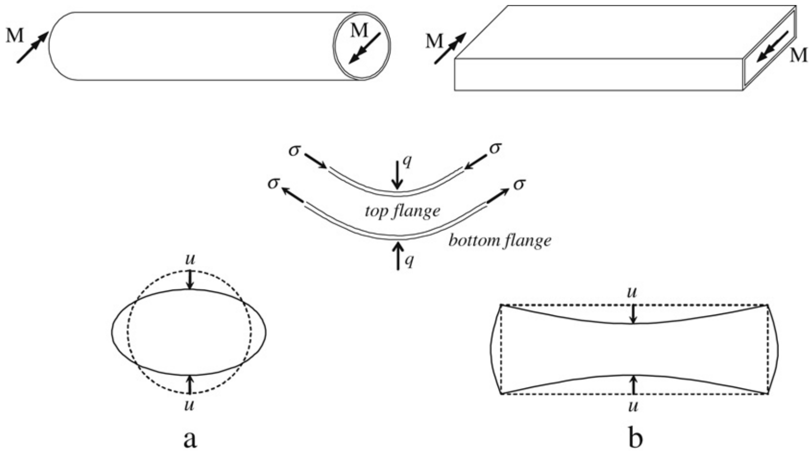

Fig. 1. Similar phenomena: (a) ovalization of tubes and (b) curling of wide flange sections.

Nomenclature

b, t	Wide flange width and thickness
A	Cross-section area
b_{red}	Reduced flange width
c_f	Admissible curling displacement
D	Plate bending stiffness
E	Young modulus
f_{av}	Average stress in the full (unreduced) flange width
f_y	Yield stress
I, I_0	Second moment of area of the deformed and undeformed sections
I_f, I_w	Second moment of area of the flange and web due to curling
I_{f0}, I_{w0}	Second moment of area of the flange and web without curling
I_u	Decrement of second moment of area of the deformed section due to curling
I_{red}	Approximate second moment of area of the reduced cross-section
k	Rotational stiffness
K	Transverse bending stiffness
M, χ	Bending moment and curvature
l, h	Web width and height
q	Flange distributed load
r	Parameter associated with the rotational restraint provided by the web
u	Curling displacement
v	Deflected shape of the curled flange
V	Potential energy
x	Coordinate along the flange width
y	Coordinate measured perpendicularly to the flange
y_0, y_u	Centroid coordinates of the undeformed and deformed section
y_{red}	Approximate centroid coordinate of the reduced cross-section
β	Angle of inclination of the web
ϕ	Flange width reduction factor
λ	Boundary condition parameter
θ	Rotation at the web-flange junction
σ	Normal stress

where (i) u is the curling displacement (see Fig. 1(b)), (ii) σ is the stress in the flange, (iii) b_s is one half of the distance between the webs of a doubly supported flange ($b_s = b/2$), (iv) t is the thickness and (v) y_0 is the distance of the flange to the neutral axis of the cross-section — the derivation details can be found in [6,7]. Introducing $\sigma = M \cdot z/I$ in expression (2), one obtains the relation between the applied moment M and the curling displacement u, given by

$$M = \frac{EI \cdot t}{b_s^2 \sqrt{\lambda \cdot y_0}} \sqrt{u} \tag{3}$$

where λ is a parameter depending on the boundary conditions of the flange and I is the second moment of area of the cross-section. Very recently, Lecce and Rasmussen [8] showed that this parameter may vary significantly: $\lambda = 0.455$ for flanges fully fixed at the web-flange junctions and $\lambda = 2.275$ for flanges simply supported. Notice that Winter adopted an intermediate value ($\lambda = 2$).

Winter's formula is currently adopted in Eurocode 3 — Part 1.3 [9]. The EC3 rules stipulate that the effect of flange curling must be taken into account unless the curling displacement u is less than 5% of the web height. With the exception of structural liner trays, EC3 does not state how the design for flange curling must be performed if the curling displacement u is higher than 5% of the web height. Recently, Davies and Chiu [10] rationally argued that EC3 rules on flange curling are inconsistent because (i) they are missing (at least for standing seam or roofing systems) if the phenomenon is relevant or (ii) they should be avoided if the phenomenon is of minor consequence. In fact, more investigations (experimental, numerical and analytical) on this subject are needed to clarify these issues unveiled by Davies and Chiu [10].

The AISI Specification for Cold-Formed Steel Members [11] also provides a rule concerning flange curling effects, which differs from the EC3 one. It states that, for a given amount of flange curling c_f, the maximum permissible flange width b_s is given by

$$\text{Max } b_s = \sqrt{\frac{0.061 \cdot t \cdot h \cdot E}{f_{\text{av}}}} \sqrt[4]{\frac{100 c_f}{h}} \tag{4}$$

where h is the web height, f_{av} is the average stress in the full (unreduced) flange width[2] and c_f is the admissible curling displacement. AISI Specification does not stipulate the amount of curling that can be regarded as tolerable and states that the corresponding factor c_f must be established by the designer since

[2] If the member is designed by the effective width design procedure, the average stress f_{av} equals the maximum stress multiplied by the ratio of the effective width b_{eff} to the actual width b ($f_{\text{av}} = f_y b_{\text{eff}}/b$).