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Autor: Suzuki, Yasyhiro / Okumura, Toshie

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Influence of Cross-Sectional Distortion on Flexural-Torsional Buckling

Influence de la torsion dans la section sur le flambage combiné flexion-torsion

Einfluß der Querschnittsverdrehung auf das Biegedrillknicken

YASYHIRO SUZUKI
Research Engr.
Sakurada Iron Works Co. Ltd.
Tokyo, Japan

TOSHIE OKUMURA
Prof. of Civ. Engrg.
University of Tokyo
Tokyo, Japan

1. Introduction

The influence of cross-sectional distortion on the flexural-torsional buckling of members with thin-walled open cross sections is studied applying the assemptions used in the folded plate theory.

The buckling theories of thin-walled members have been separatedly developed as the primary buckling and as the local buckling. The flexural-torsional buckling is involved in the former. The former is based on the fundamental assumption that the cross section is non-deformable at the instant of buckling, and the later is related to cross-sectional distortion. It is, however, reasonable to consider that the both of the above buckling phenomena actually take place simultaneously. The influence of cross-sectional distortion is increasing its importance in the analysis of buckling due to a tendency of using the more thin-walled members with the appearance of high-strength steel.

In relation to this problem, F. Bleich studied the flexural-torsional buckling of T-shaped stiffener considering the deformation of its web.[(1)] In Japan, T. Okumura,[(2)] and T. Naka et al.[(3)] studied the lateral buckling of I-shaped beams with the same method as Bleich, where the web plate is considered as an assembly of narrow transverse strips. Recently, R. Schmied[(4)] and M. Fischer[(5)] studied the buckling of I-shaped members considering the complete plate action of web plate. E. Goldberg et al[(6)] presented a systematical buckling analysis for members with arbitrary cross-sectional forms considering the cross-sectional distortion and starting from the usual plate equation and the membrane equation.

In this study, the members which consist of many flat plate elements are treated by energy method, and so it is difficult to take the complete plate action of each element into the consideration due to the complexity. For this reason, the thin-walled members are replaced by mechanical models of folded plate system, and thus the plate action of each element is simplified as Bleich's method.

The buckling stress is calculated with the use of the energy method. Expressions for the internal strain energy and the potential energy of external loads are derived for a thin-walled member as a folded plate system. Assuming proper buckling modes and introducing into the total energy expression, the critical condition for buckling is obtained by the concept of stational energy. Numerical results are obtained with the help of an electronic computer.

2. General Equation for Analysis

2.1 Assumptions and Symbols

In this study, only the member with a prismatical open section is treated. The cross section considered here consists of one series of flat plate elements as shown in Fig. 1.

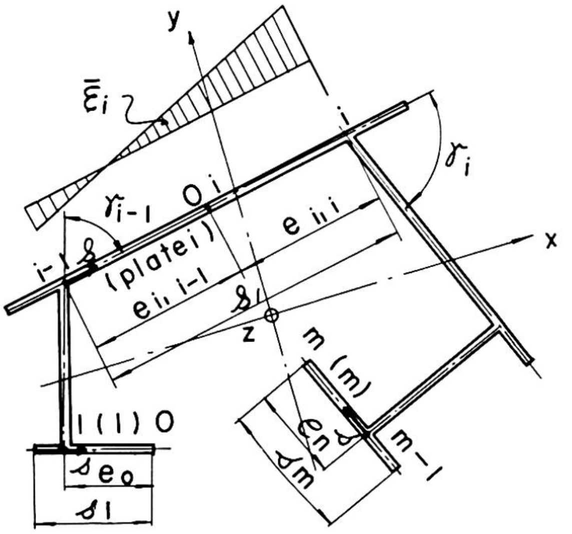

Fig. 1 Typical cross section

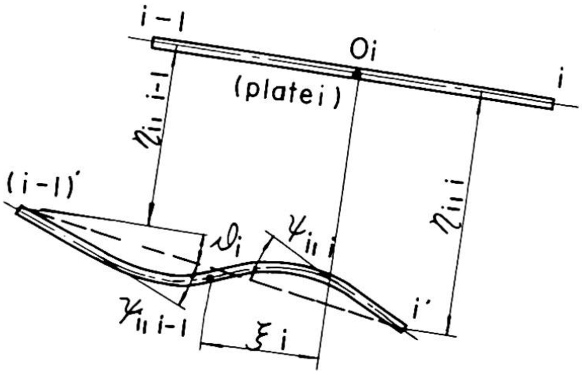

Fig. 2 Deformation of plate element

The following assumptions are also the basis of the analysis.

(1) At any point in the section, the longitudinal normal stresses due to the external loads do not exdeed the proportional limit at the instant of buckling.

(2) The longitudinal normal stresses are linearly distributed only within each plate element. That is, Navier's hypothesis remains valid for each plate element, but not for the whole section.

(3) Each plate element is subjected only to bending moment in the direction perpendicular to the longitudinal axis and to shear force accompanied to this bending moment. That is, the thin-walled member is replaced by an assembly of transverse frames of unit width, and between these frames normal and tangential stresses in the plane of each element are transmitted from one frame to another.

The symbols used hereafter are shown in Fig. 1 and 2. Fig. 2 shows the displacement components. They are

ξ_i : displacement of a plate element in its plane

$\eta_{i,i}$ $\eta_{i,i-1}$: displacements of nodal points perpendicular to the plate to which they belong

ϑ_i : bar rotation of a plate

$\psi_{i,i}$ $\psi_{i,i-1}$: tangential angles at the edge of a plate element

φ_i : rotation of a nodal point

and among these the following relations exist;

$$\vartheta_i = \frac{1}{s_i}(\eta_{i,i} - \eta_{i,i-1}) = \frac{1}{s_i}\left(\frac{\xi_{i-1}}{\sin\gamma_{i-1}} - \left(\frac{1}{\tan\gamma_{i-1}} + \frac{1}{\tan\gamma_i}\right)\xi_i + \frac{\xi_{i+1}}{\sin\gamma_i}\right) \quad \cdots\cdots\cdots (1)$$

$$\left.\begin{aligned} \eta_{i,i-1} &= \left(\frac{\xi_{i-1}}{\cos\gamma_{i-1}} - \xi_i\right)\frac{1}{\tan\gamma_{i-1}} \\ \eta_{i,i} &= \left(\xi_i - \frac{\xi_{i+1}}{\cos\gamma_i}\right)\frac{1}{\tan\gamma_i} \end{aligned}\right\} \quad \cdots\cdots\cdots (2)$$

$$\left.\begin{aligned} \varphi_{i-1} &= \vartheta_i + \psi_{i,i-1} \\ \varphi_i &= \vartheta_i - \psi_{i,i} \end{aligned}\right\} \quad \cdots\cdots\cdots (3)$$

From the above relations, it can be concluded that the necessary and sufficient components to descrive the whole deformation of the section are ξ_i (i =1,2,... m) and φ_i (i =1,2,... m-1).

2.2 Internal Strain Energy

The internal strain energy V as a folded plate system is separated into three parts.

$V = V_1 + V_2 + V_3$

where V_1 : sum of the strain energy by the beam action (simplified membrane action) of each element

V_2 : sum of the strain energy by the frame action (simplified plate action) of each element

V_3 : sum of the strain energy by the pure torsion of each element

These energies are to be expressed in terms of the independent deflection ξ_i and φ_i .

( )' means hereafter the differentiation by z-coordinate.

i) V_1 As the linearlity of the longitudinal normal strain is hold within each element, V_1 can be expressed in the following form for all m plate

$$V_1 = \frac{1}{2}\int_\ell \sum_{i=1}^{m} (EI_i\, \xi_i''^2 + EF_i\, \bar{\varepsilon}_i^2)\, dz \quad \text{------------}(5)$$

where I = moment of inertia of a plate element

F = cross area of a element

$\bar{\varepsilon}_i$ = longitudinal normal strain at the centroid of a element

E = Young's modurus

ξ_i and $\bar{\varepsilon}_i$ are not independent of each other. The longitudinal normal strains in both plates must be equal at a point where two plates meet

$$\bar{\varepsilon}_i + e_{i,i}\, \xi_i'' = \bar{\varepsilon}_{i+1} - e_{i+1,i}\, \xi_{i+1}'' \quad \text{------------}(6)$$

Introducing the average strain ε , and the difference ε_i between the strains ε and $\bar{\varepsilon}_i$, as follows

$$\varepsilon = \frac{1}{F}\sum_{i=1}^{m} F_i\, \bar{\varepsilon}_i \qquad F = \sum_{i=1}^{m} F_i \quad \text{------------}(7)$$

$$\varepsilon_i = \bar{\varepsilon}_i - \varepsilon \quad \text{------------}(8)$$

Eq.(6) can be solved in the following forms

$$\varepsilon_1 = \sum_{i=1}^{m} a_j\, \xi_j'' \quad \text{------------}(9a)$$

$$\varepsilon_i = \sum_{j=1}^{m} a_j\, \xi_j'' + \sum_{j=1}^{i} c_{ij}\, \xi_j'' = \sum_{j=1}^{m} A_{i,j}\, \xi_j'' \quad (i \geq 2) \quad \text{------------}(9b)$$

where

$$a_1 = -\frac{e_{11}}{F}(F - F_1) \qquad a_j = -\frac{1}{F}\Big(e_{j,j-1} F_j + s_j \sum_{j'=j+1}^{m} F_{j'}\Big) \quad \text{------------}(10a)$$

$$\left.\begin{array}{lll} i = 1 & c_{11} = 0 & \\ i = 2 & c_{2,1} = e_{11} \quad c_{22} = e_{21} & \\ i > 3 & c_{i1} = e_{11} \quad c_{i,j} = s_j \ (i \neq j) & c_{i,i} = e_{i,i-1} \end{array}\right\} \quad \text{------------}(10b)$$

$$\left.\begin{array}{ll} j \leq i & A_{i,j} = a_j + c_{i,j} \\ j > i & A_{i,j} = a_j \end{array}\right\} \quad \text{------------}(10c)$$

Substituting Eq.(9) into Eq.(5), the expression for V_1 is obtained in the following form

$$V_1 = \frac{1}{2}\int_\ell \sum_{i=1}^{m} \{ EI_i\, \xi_i''^2 + EF_i (\varepsilon_i + \varepsilon)^2 \}\, dz$$

$$= \frac{1}{2}\int_\ell \Big\{\sum_{i=1}^{m} (EI_i\, \xi_i''^2 + EF_i\, \varepsilon_i^2) + EF_i\, \varepsilon^2 \Big\}\, dz$$

$$= \frac{1}{2}\int_\ell \Big[\sum_{i=1}^{m} \Big\{ EI_i\, \xi_i''^2 + EF_i \sum_{j=1}^{m} (A_{i,j}\, \xi_i'')^2 \Big\} + EF\varepsilon^2 \Big]\, dz$$

$$= \frac{E}{2}\int_\ell \Sigma R_{i,i}\, \xi_i''^2\, dz + \frac{E}{2}\int_\ell \sum_{\substack{i=1 \\ i'=1 \\ (i \neq i')}}^{m} 2R_{i,i'}\, \xi_i''\, \xi_{i'}''\, dz + \frac{E}{2}\int_\ell F\varepsilon^2 dz \quad \text{------}(11)$$

where $R_{ij} = R_{ji}$, and

$$\left.\begin{aligned} R_{ii} &= I_i + \sum_{j=1}^{m} F_j A_{j,i}^2 \\ R_{i',i''} &= \sum_{j=1}^{m} F_j A_{j,i'} A_{j,i''} \quad (i' \neq i'') \end{aligned}\right\} \text{-----------------} (12)$$

ii) V_2 Replacing the thin-walled member into an assembly of frames in the transverse direction according to the assumption in the folded plate theory, the expression for strain energy V_2 can be derived using the following relations for transverse moments Y and Eq.(3)

$$\left.\begin{aligned} Y_{i-1} &= \frac{2D_i}{s_i}(2\psi_{i,i-1} - \psi_{i,i}) \\ Y_i &= \frac{2D_i}{s_i}(-\psi_{i,i-1} + 2\psi_{i,i}) \end{aligned}\right\}$$

The strain energy V_2 stored in the plate element i becomes as follows

$$\begin{aligned} \Delta V_2 &= \int_{s_i} \frac{Y^2}{2D_i} ds = \frac{s_i}{6D_i}(Y_{i-1}^2 + Y_{i-1} \cdot Y_i + Y_i^2) \\ &= \frac{2D_i}{s_i}\{\varphi_{i-1}^2 + \varphi_{i-1}\varphi_i + \varphi_i^2 - 3\vartheta_i(\varphi_{i-1} + \varphi_i) + 3\vartheta_i^2\} \end{aligned}$$

where $D_i = Et_i^3/12(1-\nu^2)$

Since one edge is free for the edge plates 1 and m, no bending moment exist in these element, and thus no contribution to the strain energy may be assumed. Consequently the strain energy V_2 for the whole section is expressed as follows excluding the contribution from the edge plates

$$V_2 = \int_{\ell} \sum_{i=2}^{m-1} \frac{2D_i}{s_i}\{\varphi_{i-1}^2 + \varphi_{i-1}\varphi_i + \varphi_i^2 - 3\vartheta_i(\varphi_{i-1} + \varphi_i) + 3\vartheta_i^2\}\, dz \text{ ------ } (13)$$

ϑ_i in this expression is a function of ξ_i as defined in Eq.(1).

iii) V_3 This energy is defined as the sum of the energies stored in each element due to pure torsion, that is, St. Venant's torsion. This strain energy is expressed as a function of ϑ_i for each interior plate element, while the contribution for the two edge plates are expressed by the twisting angles at the nodal points 1 and m-1, and thus the strain energy V_3 is

$$V_3 = \frac{1}{2}\int_{\ell} \sum_{i=2}^{m-1} GJ_i \vartheta_i'^2 dz + \frac{1}{2}\int_{\ell}(GJ_1\varphi_1'^2 + GJ_m\varphi_{m-1}'^2)\, dz \text{ -------------- } (14)$$

where J_i is St. Venant's torsional constant and G is the elastic modulus for shear.

2.3 Potential Energy of External Loads

The potential energy of external loads U is equal to the sum of negative products external forces and displacements of their points of application in the didection of the forces. Considering the fully loaded but undeflected state as the reference position for the potencial energy, U represents the change of the potential energy due to buckling:

$$U = -\int_F \bar{\sigma}\,\delta\, dF \text{ ---------------------------- } (15)$$

Provided that no cross-sectional distortions take place at the end of the member, the external stresses on the end surfaces are given as follows for the nodal points

$$\bar{\sigma}_i = \frac{P}{F} + \frac{M_x}{I_x} y_i + \frac{M_y}{I_y} x_i$$

The stresses vary linearly between the nodal points and therefore

$$\bar{\sigma} = \bar{\sigma}_{i-1} + (\bar{\sigma}_i - \bar{\sigma}_{i-1})\left(\frac{s}{s_i}\right) \text{ ---------------- } (16)$$

Neglecting the change of the external stresses and the change of the fiber strains due to the change of stresses before and after the buckling, the displacement δ is due only to the curvature of the fiber by the buckling deflection:

$$\delta = \frac{1}{2}\int_\ell \left(\left(\frac{d\Delta x}{dz}\right)^2 + \left(\frac{d\Delta y}{dz}\right)^2\right) dz \text{ ---------------------- } (17)$$

where Δx and Δy are the two displacements of a fiber in the mutually perpendicular direction. Considering the distortion of the plate elements, they become as follows for interior elements

$$\left.\begin{aligned} \Delta x &= \xi_i \\ \Delta y &= \eta_{i,i-1} + s_i\varphi_{i-1}\left(\frac{s}{s_i}\right) - s_i(2\varphi_{i-1} + \varphi_i - 3\vartheta_i)\left(\frac{s}{s_i}\right)^2 + s_i(\varphi_{i-1} + \varphi_i - 2\vartheta_i)\left(\frac{s}{s_i}\right)^3 \end{aligned}\right\} \text{ -------- } (18)$$

Substituting the displacement δ obtained from Eq.(17) and (18), and the external stresses of Eq.(16) into Eq.(15), the portion of potential energy of external loads for the internal plates U_1 is given as follows

$$\begin{aligned} U_1 = -\frac{1}{2}\int_\ell \sum_{i=2}^{m-1} \Big[& \frac{\bar{\sigma}_{i-1}F_i}{840}\big(420\xi_i'^2 + 420\eta_{i,i-1}'^2 + 5s_i^2\varphi_{i-1}'^2 - 6s_i^2\varphi_{i-1}'\varphi_i' + 3s_i^2\varphi_i'^2 + 72s_i^2\vartheta_i'^2 \\ & + 252 s_i\eta_{i,i-1}'\vartheta_i' + 24s_i^2\vartheta_i'\varphi_{i-1}' - 28s_i^2\vartheta_i'\varphi_i' + 84s_i\eta_{i,i-1}'\varphi_{i-1}' - 56s_i\eta_{i,i-1}'\varphi_i'\big) \\ & + \frac{\bar{\sigma}_i F_i}{840}\big(420\xi_i'^2 + 420\eta_{i,i-1}'^2 + 3s_i^2\varphi_{i-1}'^2 - 6s_i^2\varphi_{i-1}'\varphi_i' + 5s_i^2\varphi_i'^2 + 240s_i^2\vartheta_i'^2 \\ & + 588s_i\eta_{i,i-1}'\vartheta_i' + 28s_i^2\vartheta_i'\varphi_{i-1}' - 60s_i^2\vartheta_i'\varphi_i' + 56s_i\eta_{i,i-1}'\varphi_{i-1}' \\ & - 84s_i\eta_{i,i-1}'\varphi_i'\big)\Big] dz \end{aligned} \qquad (19)$$

This potential energy for the edge plates become as follows considering the following relations

$$\begin{array}{llll} \text{for plate} & 1 & \Delta x = \xi_1 & \Delta y = \eta_{11} - \varphi_1 \cdot s \\ \text{for plate} & m & \Delta x = \xi_m & \Delta y = \eta_{m,m-1} + \varphi_{m-1} \cdot s \end{array}$$

$$\begin{aligned} U_2 = -\frac{1}{2}\int_\ell \Big[& \bar{\sigma}_0 F_1\Big\{(\xi_1'^2 + \eta_{11}'^2)\left(1 - \frac{s_1}{2e_0}\right) - 2\eta_{11}'\varphi_1'\left(e_0 - s_1 + \frac{s_1^2}{2e_0}\right) + \frac{\varphi'^2}{4}\left(2 - \frac{s_2}{e_0}\right)(2e_0^2 - 2e_0 s_1 + s_1^2)\Big\} \\ & + \bar{\sigma}_1 F_1\Big\{(\xi_1'^2 + \eta_{11}'^2)\frac{s_1}{2e_0} - \eta_{11}'\varphi_1' s_1\left(1 - \frac{2}{3}\cdot\frac{s_1}{e_0}\right) + \frac{\varphi_1'^2}{12e_0}(6s_1 e_0 - 8e_0 s_1^2 + 3s_1^3)\Big\} \\ & + \bar{\sigma}_{m-1}F_m\Big\{(\xi_m'^2 + \eta_{m,m-1}'^2)\frac{s_m}{2e_m} + \eta_{m,m-1}'\varphi_{m-1}' s_m\left(1 - \frac{2}{3}\cdot\frac{s_m}{e_m}\right) + \frac{\varphi_{m-1}'^2}{12e_m}(6s_m e_m^2 - 8e_m s_m^2 + 3s_m^3) \\ & + \bar{\sigma}_m F_m\Big\{(\xi_m'^2 + \eta_{m,m-1}'^2)\left(1 - \frac{s_m}{2e_m}\right) + 2\eta_{m,m-1}'\varphi_{m-1}'\left(e_m - s_m - \frac{s_m^2}{2e_0}\right) + \frac{\varphi_{m-1}'^2}{4}\left(2 - \frac{s_m}{e_m}\right) \\ & \times(2e_m^2 - 2e_m s_m - s_m^2)\Big\}\Big] dz \end{aligned} \qquad (20)$$

The potentical energy of the whole system π is the sum of the expressions as derived and thus $\pi = V + U = V_1+V_2+V_3+U_1+U_2$, in which all the terms are the function of η and ϑ. The deflection components ξ_i and φ_i involved in the above expression are also expressid in terms of ξ_i and φ_i by Eq. (1) and (2).

Assuming the buckling mode, the critical equation for this buckling is derived from the concept of stationary potential energy, that is

$$\delta\pi = \delta\,(V_1+V_2+V_3+U_1+U_2) = 0$$

The terms which includes the strain ε in the energy V_1 is ommitted, because we consider here only the change of the energy before and after the buckling.

3. Numerical Examples on Hat-Shaped Columns

3.1 Critical Equations

For an example, lateral-torsional buckling of columns subjected to exccentrical compression on the axis of symmetry as shown in Fig. 3 is treated.

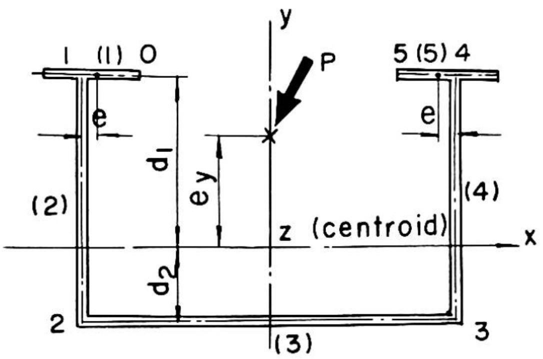

Fig. 3 General hat-shaped section

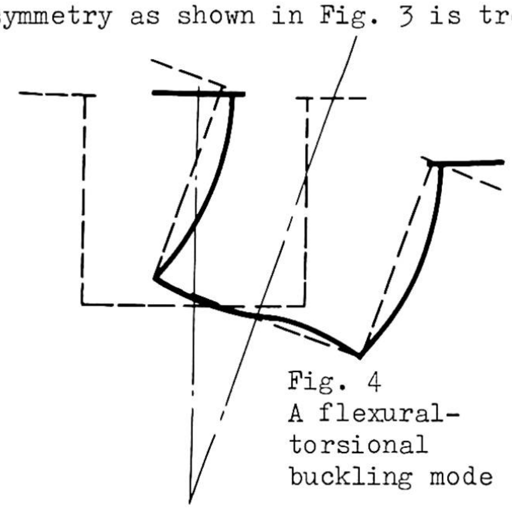

Fig. 4 A flexural-torsional buckling mode

Introducing the following parameters

$$Z_1 = 1 + \frac{Fd_1}{I_x} e_y \quad , \quad Z_2 = 1 - \frac{Fd_2}{I_x} e_y$$

the external stresses at nodal points are expressed by one stress $\bar{\sigma}$ as follows

$$\bar{\sigma}_0 = \bar{\sigma}_1 = \bar{\sigma}_4 = \bar{\sigma}_5 = \bar{\sigma}\cdot Z_1 \quad , \qquad \bar{\sigma} = P/F$$

$$\bar{\sigma}_2 = \bar{\sigma}_3 = \bar{\sigma}\cdot Z_2$$

The flexural-torsional buckling mode is assumed to be symmetric on the axis of symmetry of the cross section as shown in Fig. 4, then there exist the following relationship

$$\xi_1 = \xi_5 ,\ \xi_2 = \xi_4 ,\quad \varphi_1 = \varphi_4 ,\quad \varphi_2 = \varphi_3$$

ξ_1, ξ_2, ξ_3 and φ_1, φ_2 remains as the independent components. In the direction of the column axis, they are further assumed to vary as

$$\xi_i = C_i \sin \lambda z \qquad \lambda = \pi/\ell$$

$$\varphi_i = B_i \sin \lambda z$$

After introducing the above relations into the energy expressions derived in the previous sections, the potential energy for the whole system can be expressed in terms of C_i and B_i, and differentiation of thus obtained total energy by C_i and B_i reduces to a system of linear equations about C_i and B_i. The buckling criteria is the condition that the determinant of this system vanishes;

$$\begin{vmatrix} K_{11} & K_{12} & K_{13} & K_{14} & K_{15} \\ K_{21} & K_{22} & K_{23} & K_{24} & K_{25} \\ K_{31} & K_{32} & K_{33} & K_{34} & K_{35} \\ K_{41} & K_{42} & K_{43} & K_{44} & K_{45} \\ K_{51} & K_{52} & K_{53} & K_{54} & K_{55} \end{vmatrix} = 0$$

The elements of this determinant take the form $K_{ij}=K_{ij,1}+K_{ij,2}\times\bar{\sigma}$. Numerical evaluation of $\bar{\sigma}$ satisfying this condition can be attained with the help of an electronic computer. And in this case the theoretical value with no consideration of cross sectional destortion can be used as the starting value.

3.2 Numerical Results and Comparison with Usual Theory

Some numerical results on hat-shaped columns are shown in Fig. 5 through 8, in which the vertical axis is the reduction of the flexural-torsional buckling stress due to the influence of cross-sectional distortion compared with the usual theory where the cross section is assumed to be non-deformable----- cf. (1) or (7).

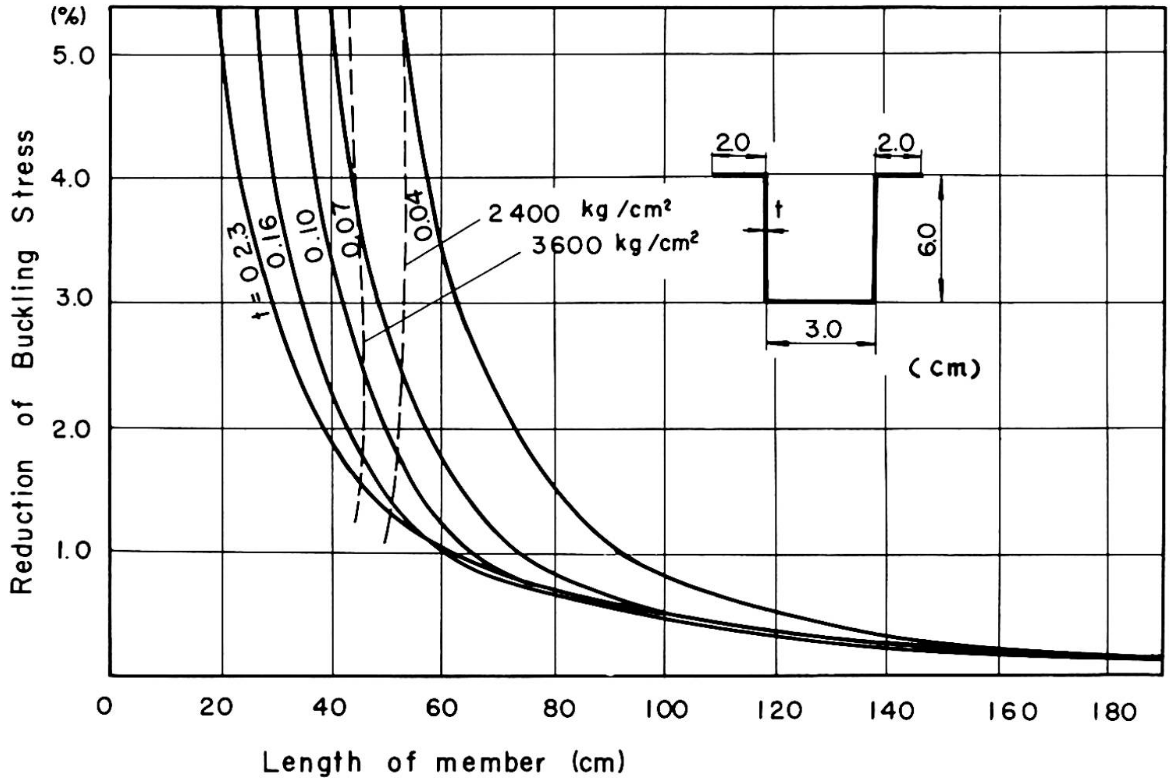

Fig. 5 Relation between reduction of buckling stress and length of member for hat-shaped column

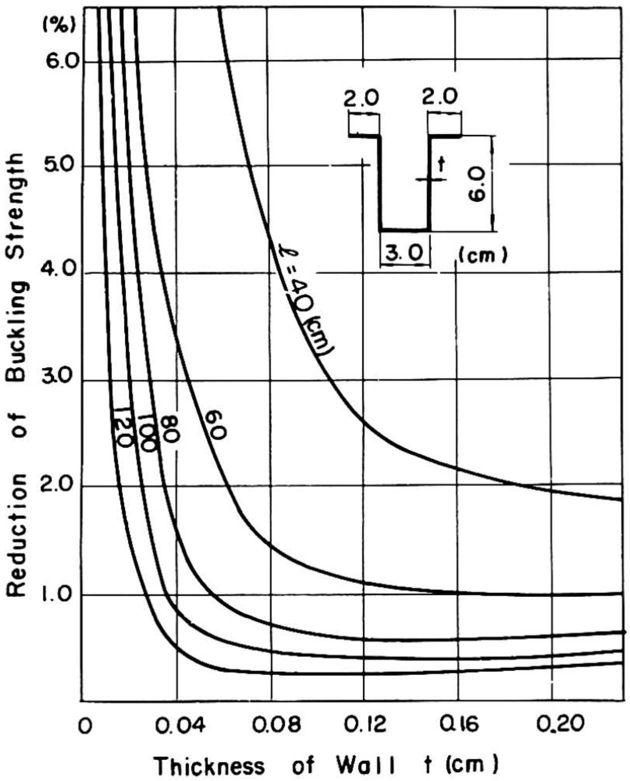

Fig. 6 Relation between reduction of buckling strength and wall thickness

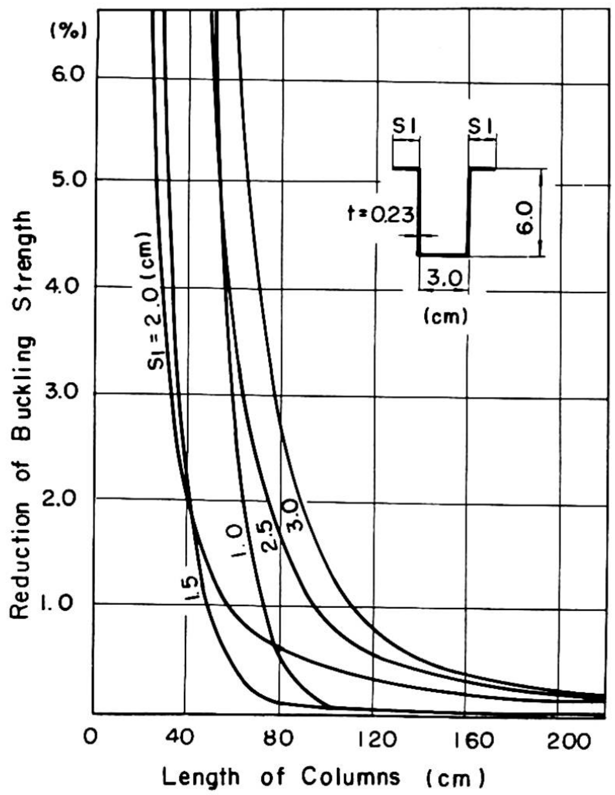

Fig. 7 Relation between reduction of buckling strength and size of lip plates

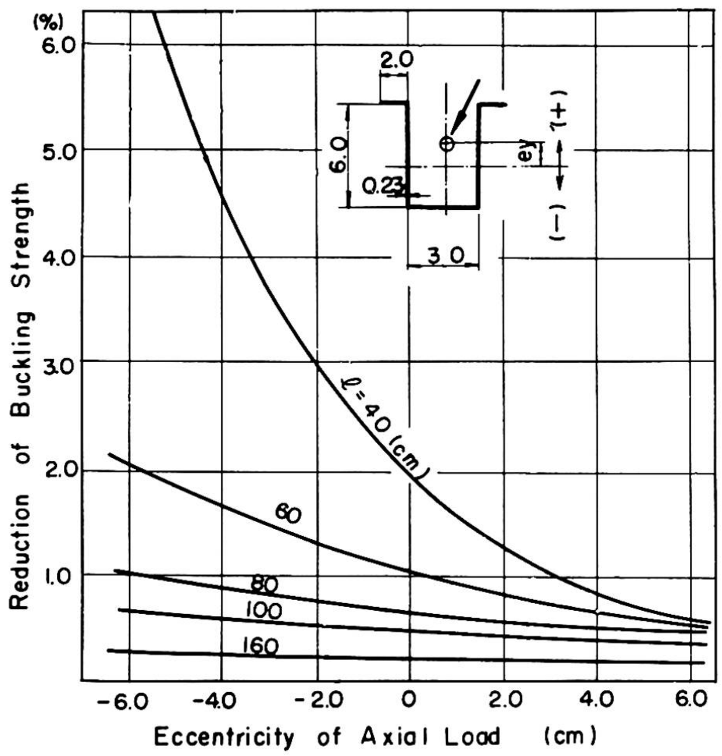

Fig. 8 Relation between reduction of buckling strength and eccentricity of axial force

Fig. 5 shows the change of reduction of the flexural-torsional buckling stress in relation to the length of column for some wall-thickness, when the hat-shaped column shown in the same figure is subjected to the axial compression only. For longer columns the reduction is generally very small and scarcely no more than 1%, but with the shortening of column length, acceleration of the reduction is noted; more acceleration is observed for a column with thinner cross section.

It is noticed that the results shown here are valid only for the elastic buckling, and this limits are shown in the figure by two broken lines for yield stresses of 2400 and 3600 kg/cm^2, and the curves on the lefthand side of these broken lines for each yield stress has to be modified to be meaningful by the theory of inelastic buckling. Consequently, it can be concluded that the reduction of the flexural-torsional buckling strength for a practical elastic column does not exceed a few percents.

Fig. 6 shows the relationship between the change of reduction and the change of the wall-thickness for the same column as for Fig. 5.

Fig. 7 shows the variation of reduction as a function of sizes of lip plates of the hat-shaped column under axial compression. It is presumed that there is a critical size of lip plates with which the minimum reduction may results.

Fig. 8 shows the relation between the reduction and the eccenntricity of the axial load. In the case of a longer member, the reduction is scarcely influenced by the eccentricity and it's magnitude is very small but in the case of a shorter member the reduction is largely changed by eccentricity, and moreover the reduction increases with increasing eccentricity to the direction for the top of the hat-shape and decreases with a eccentricity to the opposite direction.

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SUMMARY

An approximate method to estimate the influence of the cross-sectional distortion on the flexural-torsional buckling of open thin-walled members is presented based on the folded plate theory using the energy concept. From some numerical results on hat-shaped columns, it is shown that this influence is small and negligible for steel columns of practical length and section, but for such columns as with shorter length or much thinner wall thickness, the reduction of flexural-torsional buckling stress is highly enlarged.

RÉSUMÉ

Une méthode approximative d'estimation de l'influence de la déformation de la section sur le flambage combiné flexion -torsion d'une barre à section ouverte mince est développée à partir de la théorie des voiles prismatiques utilisant des considérations d'énergie. Quelques résultats numériques sur des profilés en U montrent que cette influence est minime pour des barres de dimensions raisonnables, mais que les tensions sont réduites sensiblement dans le cas de barres très courtes ou très minces.

ZUSAMMENFASSUNG

Aufgrund der Faltwerktheorie mittels Energiebetrachtung wurde eine Näherungsmethode entwickelt, um den Einfluss der Querschnittsverformung auf das Biegedrillknicken von offenen, dünnwandigen Stäben zu schätzen. Einige numerische Beispiele mit U-Profilen zeigen, dass dieser Einfluss verschwindend klein wird bei Stäben mit normalen Längen und Querschnitten, dass dagegen die Biegedrillknickspannung erheblich reduziert wird bei sehr kurzen oder sehr dünnwandigen Stäben.

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