



BUCKLING ANALYSIS OF THE INDUSTRIAL FACTORY MODEL BY FINITE ELEMENT METHOD

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Abstract. Buckling is a subject that has been discussed for a long time, however, it's still being studied and developed due to its practicality. The following article introduces two methods that are used to solve the problems involving buckling of the beam, shell and solid with an I shape cross-section having different cases of boundary load. The theory used in this article is Euler's formula and Eurocode 3 standard. The analytical results by ANSYS commercial software are compared with the theoretical results and results from Eurocode 3 standard. The authors based on the reliability of the calculation results to simulate buckling of the industrial factory model with different cases of load conditions. The simulation results show a general view of buckling cases.

Keywords: finite element method, buckling, Euler's critical load, Eurocode 3, industrial factory.

Classification numbers: 4.10.4, 5.4.3.

1. INTRODUCTION

History of overall beam stability theory has been developed for more than 100 years. The first authors were probably Prandtl and Michell with studies published in 1899 about the overall stability problem of simple beams with narrow rectangular sections and pure bending [1]. Next, Timoshenko set up and solved the stability problem of simple I-section beams subjected to pure bending in 1905 [2]. Timoshenko continued to develop the beam stability theory, the results gathered in the monograph were supplement and reprint many times. In the 1930s, Wagner formulated the stability theory of section I beams with a symmetry axis and formulated a parametric formula to show the degree of asymmetry of the section [3]. In 1940, Vlasov formulated the general theory of thin wall calculation including the general stability theory of flat horizontal bending beams [4]. Vlasov was the first author to introduce the concept of restraint twisting, the fan coordinates and the fan inertia moment of the thin-wall section. Bleich used an energy method in which the total potential of the beam is subjected to a load equal to the sum of the linear and external elastic deformation potential generated by the load on the displacements when the beam is unstable [5]. Timoshenko and Gere established stable equations when considering the equilibrium of an infinitesimal element combining the equilibrium of a beam [2]. In the present paper, two methods that are used to solve the problems involving

buckling of the beam, shell and solid with an I shape cross-section having different cases of boundary load.

The two buckling models for I-beam structure subjected to bending and compression will be analyzed with the use of different types of elements, one-dimensional beam elements, shell element and three-dimensional element. The purpose is to evaluate the advantages and disadvantages when using these types of elements in modeling bending and torsional instability problems. Simultaneously, the simulation results will be compared with analytical solutions referenced from Eurocodes standards and errors will be assessed carefully. Through the verification of the reliability of the simulation results of the beams and compressive instability beams, the industrial workshop instability model will be analyzed with different load cases. This helps to increase the safety in designing industrial buildings.

2. THEORY OF BUCKLING [1]

Buckling is a behavior of structure or when a structural system suddenly is deformed and is shifted out of the planeload set. This occurs even when stress inside is less than the critical stress of the material. When increasing load in structures such as beam, column, etc. enough to make the structural components become unstable, that is defined as buckling. Buckling is therefore an instability that leads to structural failure.

2.1. Buckling of a radial compressed bar

Consider the case as follows: a long thin bar with a constant cross-section having one head fixed and the other head is free and being under an axial force.

According to [2], when the \vec{P} impacting on the vertical axis of the bar is still not significant, apply an \vec{R} force impact on the horizontal axis of the bar, the result in the shifting of the bar out of the equilibrium position. The bar will shift back to the equilibrium position when it is no longer impacted by the \vec{R} force. This is the bucking of the bar.

Increasing \vec{P} to the critical value \vec{P}_{th} , the structure will be different. When the \vec{R} force is not applied, the bar is only compressed and still balanced with force \vec{P}_{th} . When the \vec{R} force is applied, the bar will deviate from the equilibrium position and when the \vec{R} force is removed, the bar does not return to its original position. The bar is still not destroyed because of $P \leq P_{th}$. However, the bar is at an unstable state and can be destroyed if we continue to apply force to the bar as described in Figure 1.

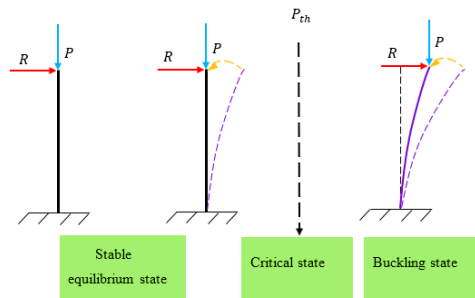


Figure 1. The states of radial compression bars.

2.2. Bending Buckling

Bending Buckling includes transposition of u and v of the shaft system, and includes both flexural stiffness components is EI_x and EI_y . According to [3], bending buckling is caused by bending moment due to the force applied multiplied by transposition u or v . Bending Buckling may occur for a beam element, a beam system or a frame as described in Figure 2.

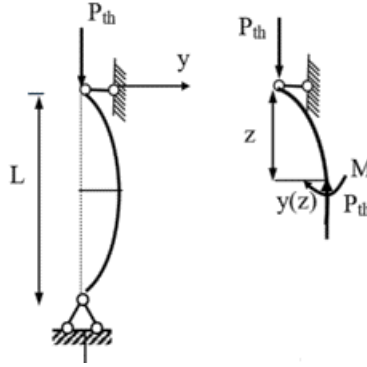


Figure 2. The compression bar model in the Euler problem.

3. THEORETICAL APPROACHES TO SOLVE COMPRESSION AND BENDING INSTABILITY PROBLEMS

3.1. Compression instability

Considering a double-jointed straight bar, subjected to compression by the critical force P_{th} . According to [2], when there is noise, the bar will be bent and we have a general formula (Euler's critical load) for critical load for radial compressed beam with different boundary conditions:

$$P_{th} = \frac{\pi^2 EI}{(\mu L)^2} \quad (1)$$

3.2. Bending instability

The standard system EUROCODES [6, 7] is a set of standards for construction structures prepared by the Technical Committee CEN / TC250 and issued by the European Standardization Committee (CEN) for general application to European Union countries. By the mid-1980s, the first standards of structural construction under the standard system Eurocodes were introduced. So far these standards have evolved into a system consisting of 10 main standards and divided into 4 groups.

The general formula for determining critical moment according to EUROCODE 3 is:

$$M_{th} = C_1 \frac{\pi^2 EI_z}{(kL)^2} \left[\sqrt{\left(\frac{k}{k_w} \right) \frac{I_w}{I_z} + \frac{(kL)^2 GI_t}{\pi^2 EI_z} + (C_2 z_g - C_3 z_j)^2} - (C_2 z_g - C_3 z_j) \right] \quad (2)$$

where: C_1, C_2, C_3 : the factors depending on the type of load and the condition of linking the two beam heads (with a picture attached), Different types of bending stability are shown in Figure 3.

k, k_w : the calculated length factors of beams when bending around the z-axis and when twisted.

The quantities z_j, z_g are determined as:

$$z_j = z_s - \frac{1}{2I_y} \int_A z(y^2 + z^2) dA = z_s - \frac{1}{2I_y} \left(\int_A z^3 dA + \int_A zy^2 dA \right) \quad (3)$$

$$z_g = z_a - z_s \quad (4)$$

The quantity I_w is determined by the following formula:

$$I_w = \frac{I_{fc} I_{ft}}{(I_{fc} + I_{ft})^2} I_z h_{fk}^2 \quad (5)$$

I_{fc} : The inertial moment of the compressive wing against the vertical axis of the section;
 I_{ft} : The inertial moment of the tensile wing against the vertical axis of the section; h_{fk} : Distance between the cut center of the tensile wing and the compressive wing.


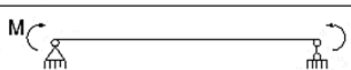

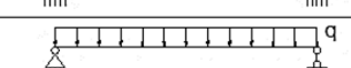
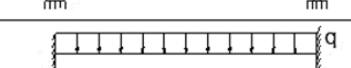
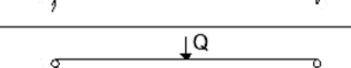
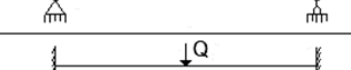
Types of buckling load	k = 1			k = 0,5		
	C_1	C_2	C_3	C_1	C_2	C_3
	1.00	-	1.00	1.00	-	1.14
	1.32	-	0.99	1.51	-	2.27
	1.88	-	0.94	2.15	-	2.15
	1.13	0.46	0.53	0.97	0.30	0.98
	1.28	1.56	0.53	0.75	0.65	1.07
	1.36	0.55	1.73	1.07	1.43	3.06
	1.56	1.27	2.64	0.94	0.71	4.80

Figure 3. Quantities C_1, C_2, C_3 .

4. CALCULATION OF THE CRITICAL LOAD OF THE STRUCTURE

4.1. Compressive structure

The parameter of material is given in Table 1 with length of 8 m.

Table 1. The parameter of the material.

Material	Cross section (cm)					
$E = 2.10^{11} N/m^2$	a	b	c	t ₁	t ₂	t ₃
$\nu = 0,3$	20	30	20	2	1	2

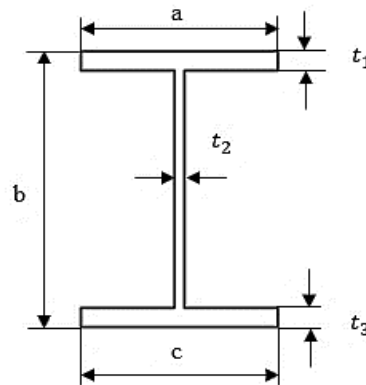


Figure 4. Model of a compressive I shape beam.

Using Euler's critical load in theory to calculate critical load with different boundary conditions. Numerical results are showed in Table 2 and the percentage of error are listed in Table 3. Two buckling mode shapes of compressive problem are shown in Figure 5.

Table 2. Comparison between analytical and numerical solution of buckling problem when using different types of element (unit N).

Boundary condition	FIXED – FREE ENDS	PIN – PIN ENDS	FIXED – PIN ENDS	FIXED – FIXED ENDS
μ	2	1	0.7	0.5
Euler	2.058e+05	8.231e+05	1.680e+06	3.293e+06
Solid Element	2.058e+05	8.213e+05	1.678e+06	3.277e+06
Beam Element	2.057e+05	8.219e+05	1.678e+06	3.272e+06
Shell Element	2.058e+05	8.210e+05	1.674e+06	3.290e+06

Table 3. The percentage of error between numerical and analytical solutions.

Boundary condition	FIXED – FREE ENDS	PIN – PIN ENDS	FIXED – PIN ENDS	FIXED – FIXED ENDS
Solid Element	0.009	0.224	0.116	0.483
Beam Element	0.035	0.152	0.104	0.61
Shell Element	0.024	0.316	0.33	0.197

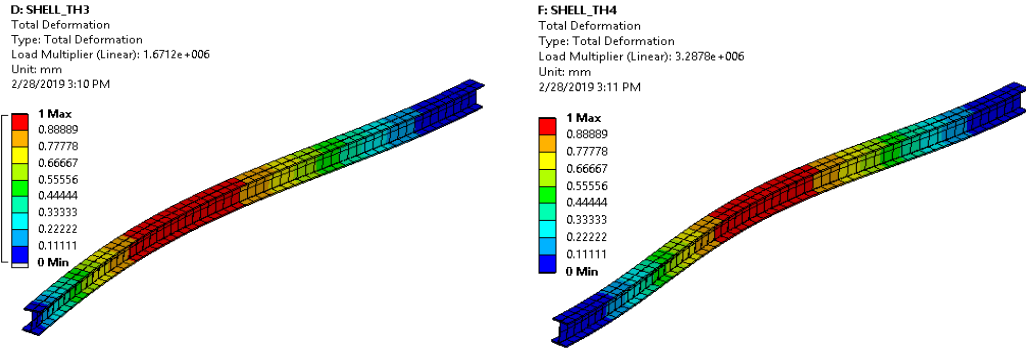


Figure 5. Two types of buckling mode shape when using shell element.

4.2. Bending structure

The parameter of material is used as problem 4.1.

Table 4. Comparison between analytical and numerical solution of buckling problem when using different types of element.

	Problem 1	Problem 2
Eurocode 3	2.77e+06 (N)	610.827 (N.mm)
Shell Element	2.79e+06 (N)	595,010 (N.mm)
Solid Element	2.82e+06 (N)	598,240 (N.mm)
Beam Element	2.75e+06 (N)	616,510 (N.mm)

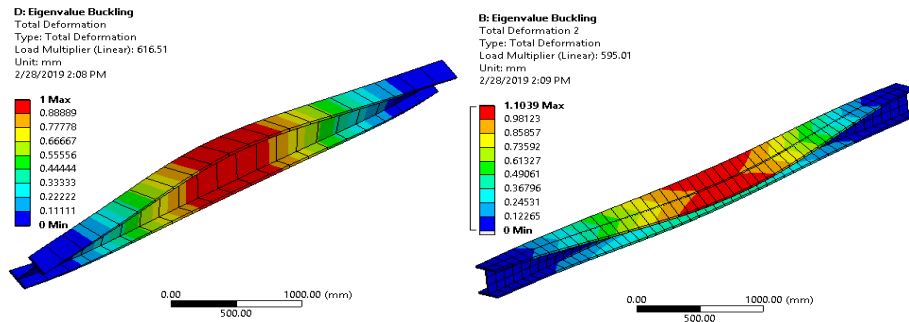


Figure 6. Two types of buckling mode shape for the bending structure.

Table 5. The percentage of error between numerical and analytical solutions.

	Problem 1	Problem 2
Shell Element	0.77	2.59
Solid Element	1.95	2.06
Beam Element	0.61	0.92

Problem 1: Model I shape with length 5 m, concentrated force in the middle with both ends fixed.

Problem 2: Model I shape with length 5 m, uniformly distributed pressure with both ends fixed.

Numerical results are showed in Table 4 and the percentage of error are listed in Table 5. Two buckling mode shapes of bending problem are shown in Figure 6

5. CALCULATION AND SIMULATION OF BUCKLING OF INDUSTRIAL FACTORY

The industrial factory has dimensions: length 37 m, width 20 m, height 14 m. Overhead crane using in the industrial factory is designed to crane of a maximum weight of 5 tons. The model of the factory is built based on 2 types of elements: column and rafter is shell element, the purlin is beam element.

Using the shell element instead of using solid element in the model help save time and resources of the computer in the calculating process. In addition, simulating the examples as above results in shell element error better than solid element.

Table 6. Critical load results in 6 simulations.

Crane's positions	Case	Load's position	Critical load (N)	Strain	Stress (MPa)
Crane at the middle of the house space	1	Concentrate force at the position of the trolley moving at the front of the crane	1.2102e6	2.311e-04	45.667
	2	Concentrate force at the position of the trolley moving at the middle of the crane	1.4925e6	1.856e-04	36.764
Crane at the front of the house space	3	Concentrate force at the position of the trolley moving at the front of the crane	1.1165e6	2.216e-04	44.248
	4	Concentrate force at the position of the trolley moving at the middle of the crane	1.4891e6	1.494e-04	29.852
Crane at ¼ of the house space	5	Concentrate force at the position of the trolley moving at the front of the crane	1.272e6	1.33e-5	2.66
	6	Concentrate force at the position of the trolley moving at the middle of the crane	8.0641e5	3.3254e-6	0.17918

This study takes a simulation of 6 cases of load-put to define the influence of the position to the critical load value of the industrial factory.

The results of the simulation of 6 cases of the load are shown in Table 6 and two buckling mode shapes of industrial factory frame are shown in Figures 7 and 8.

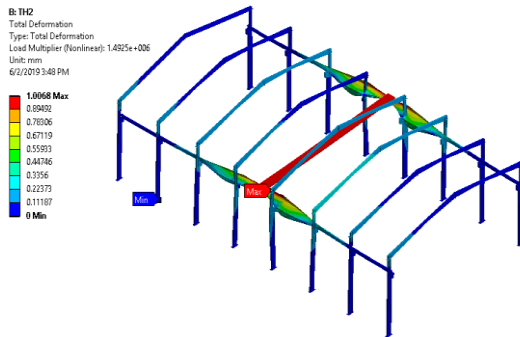


Figure 7. The simulation result of the industrial factory case 2.

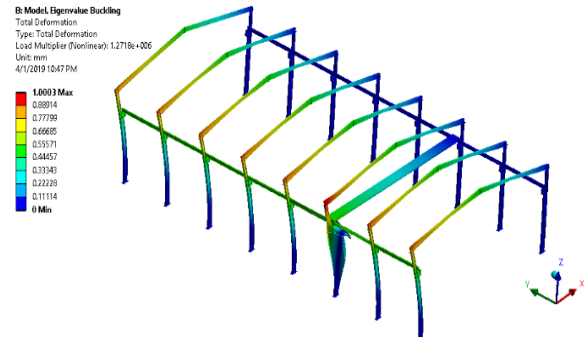


Figure 8. The simulation result of the industrial factory case 5.

Comment:

- The position of the crane with a high possibility of causing buckling for the factory is the front of the factory.
- The force concentrates at the location of the trolley moving in the middle of the crane in the cases cause buckling on two side beams of both sides of the factory, especially around the position of the bearing crane.
- The case in which the force concentrates at the location of the trolley moving at the front of the crane where the crane at 1/4 of the house space causes buckling at the location of the column close to the force's position.
- The force concentrates at the location of the trolley moving at the front and middle of the house space, causing buckling on a side of the beam bearing the load.
- According to the analysis results for both cases, mode shape 1 only has compressive buckling, from the mode of phase 2, there is a phenomenon of twisting buckling.

6. THE NATURAL FREQUENCY PROBLEM

The natural frequency is an important dynamic characteristic of the construction structure, which is closely related to the rigidity, mass, and damping capacity of the building. Determining the natural frequency of the system is one of the most important steps to identify structural dynamical characteristics to ensure stable operating conditions of the structure under the effects of loads.

In fact, the natural frequency of the structure when simulated is larger than reality cases because:

- The simplification of the model to reduce the number of calculations and simulations.
- The link between the factory and the ground actually is not mounting link, because the ground also has deformation.

The author solves the natural frequency problem with 3 types of deformation of factory structure corresponding to 5 modes in ANSYS software. The natural frequency results are listed in Table 7.

Table 7. Natural frequency calculation values.

Natural Frequency	Case 2	Case 4	Case 6
Mode 1	0.51588	0.51585	0.51585
Mode 2	1.0279	0.98245	1.0198
Mode 3	1.1223	1.1218	1.1219
Mode 4	1.1553	1.1524	1.1847
Mode 5	1.4614	1.4158	1.5767

7. TRANSIENT PROBLEM

Use the structural natural frequency simulation results to calculate the time-division step for transient problems. In a natural frequency problem, using a mode that gives results with deformation for each case of unstable structure.

When the crane carries objects with mass moving in the factory, to ensure stability and normal operating conditions, the car will be moved to the middle and bring the object to the desired location. Therefore, only the transient problem for 3 cases: case 2, case 4 and case 6 is solved and the chosen time steps are listed in Table 8.

Solving the transient problem with boundary conditions that are at the base of the columns.

The load of the problem is set as follows:

- From 0 to 4 seconds crane started lifting.
- From 4 to 8 seconds, the crane holds the object and moves at the constant speed $v = 50$ mm/s.
- From 8 to 10s of cranes drop the object gradually.

In the Figures 9 and 10, von Mises stress and deformation results vs time in the transient for case 2 are shown and the maximum stress and deformation displacement results are listed in Table 9.

Table 8. The chosen time steps for cases 2,4 and 6.

	Initial time step	Minimum time step	Maximum time step
Case 2	0.034	0.0034	0.34
Case 4	0.035	0.0035	0.35
Case 6	0.049	0.0049	0.49

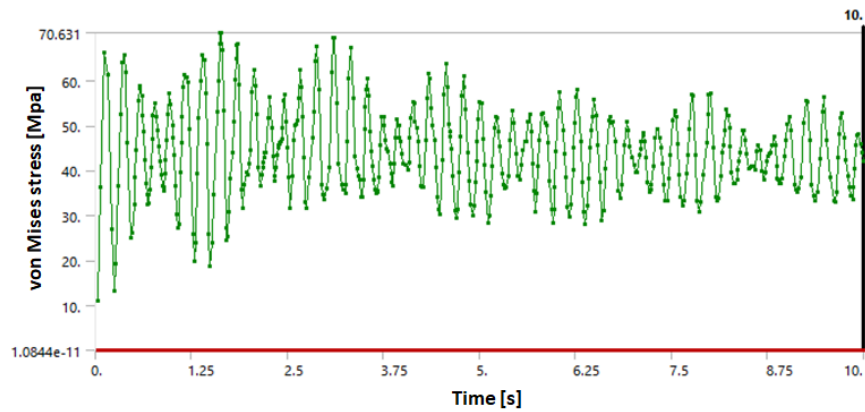


Figure 9. The von Mises stress graph in the transient problem of the case 2.

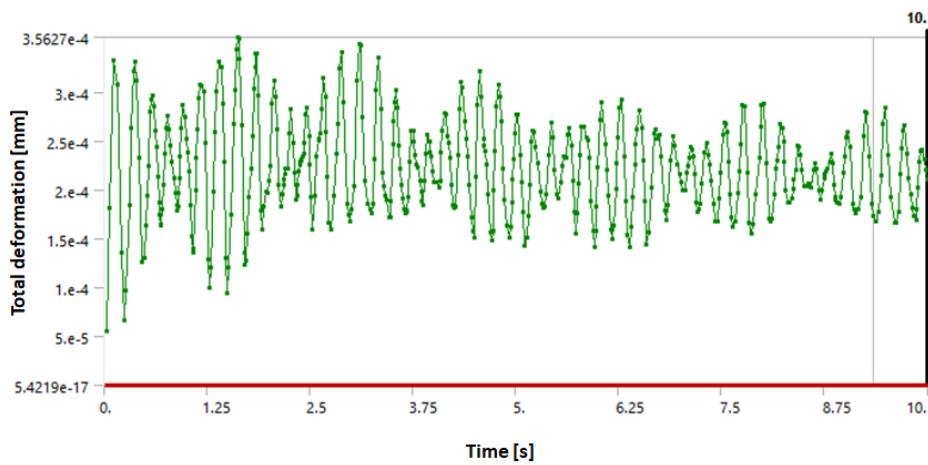


Figure 10. The total deformation graph in the transient problem of the case 2.

Table 9. The result of the maximum stress and deformation displacement.

	Maximum displacement	Maximum stress	Maximum deformation
Case 2	500	66.329	3.56e-04
Case 4	500	70.337	3.52e-04
Case 6	500	66.168	3.33e-04

8. CONCLUSION

With different time division steps, the displacement results for all 3 cases give the maximum displacement value when $t = 10$ s in the middle of the crane. The diagram of displacement increases linearly over time.

The maximum von Mises stress of case 2 is the largest in all 3 cases. The stress graph continuously changes in time from 0 to 10 s. The maximum stress in all 3 cases does not exceed the permissible stress value of the structure. So, the stable condition is satisfied.

The maximum deformation of case 2 is largest in all 3 cases. The strain graph is similar in shape to the stress graph. The deformation value does not exceed the allowable strain value of the structure. So, the rigidity condition is satisfied.

The critical force value is calculated in the case when buckling in the operating condition of the structure is greater than the designed load value of the crane.

The transition problem uses the largest load of the designed crane. The calculation results of displacement and stress show that the structure is stable under working conditions under the designed load.

When using a crane, users can use it to lift objects with loads greater than 5 tons. However, for a crane to operate with a load greater than 5 tons for a long time, it will cause wear down and tear down and may cause danger to people.

It should be regularly maintained and take appropriate measures so that the crane can always operate in the best conditions and the structure is always stable to ensure the safest for people.

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