COMPUTACIÓN APLICADA A LA INDUSTRIA DE PROCESOS

Actas del 7º Congreso Interamericano de Computación Aplicada a la Industria de Procesos, CAIP'2005

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PRÓLOGO

El Séptimo Congreso Interamericano de Computación Aplicada a la Industria de Procesos, CAIP' 2005, tuvo por sede la Universidad de Tras-os-Montes e Alto Douro, ubicada en Vila Real, Portugal. Los congresos anteriores a este se realizaron en La Serena - Chile en 1992, Santiago -Chile en 1994, Villa María - Argentina en 1996, San José - Costa Rica en 1999, Campos de Jordão - Brasil en 2001 y Puebla - México en 2003.

Este séptimo congreso ha sido organizado y patrocinado por la Universidad de Tras-os-Montes e Alto Douro y el Centro de Información Tecnológica de La Serena – Chile teniendo como objetivo el de que la comunidad académica y profesional Iberoamericana intercambie ideas, establezca contactos profesionales e académicos, discuta sus ideas respecto al desarrollo y la investigación en el tema de la computación aplicada a la Industria.

Para este evento se recibieron 145 resúmenes ampliados de los cuales 104 se aceptaron, después de su arbitraje por pares, para ser incluidos en este libro. En este proceso de evaluación y selección participaron 41 árbitros, conocidos científicos en sus áreas respectivas, de 10 países. El primero objetivo del CAIP'2005, era el de interesar investigadores de España y Portugal y tener un incremento de trabajos y congresistas de estos dos países. Ese objetivo fue conseguido pues hay alredor de 40 congresistas españoles y portugueses.

Los trabajos de los investigadores de 12 países, publicados en estas actas muestran un amplio temario y un alto nivel de la investigación que se realiza en Ibero-América. Estos trabajos se presentan en 7 capítulos: 1) Simulación de Procesos, 2) Procesos en Ingeniería Mecánica y Metalúrgica, 3) Energía y Ambiente, 4) Automatización y Control de Procesos, 5) Sistemas expertos y *Soft Computing*, 6) Transferencia de Calor y Materia y 7) Enseñanza de la Computación Aplicada.

Esperamos que la cantidad e calidad de trabajos presentados en este congreso motive a investigadores de Iberoamérica a mantener a CAIP dentro de los eventos privilegiados y respectados de la región.

Por último, en nombre del Comité Organizador, se expresa un profundo agradecimiento a todos los integrantes del Comité Internacional, a todos los árbitros y en especial al Dr. José O. Valderrama, cuya asesoría fue fundamental en el desarrollo del congreso.

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ANALYSIS OF PIPING ELBOWS FOR IN PLANE BENDING USING TWO DIFFERENT NUMERICAL MODELS

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Abstract

The piping systems are used in industrial plants subjected to any efforts. These components are constituted by straight and curved or elbow elements. Elbows attached to nozzle or straight pipes produce a stiffening effect due to the restraint of ovalization provided by the adjacent components. These elements have a small thickness and when submitted to any efforts, the excessive oval shape may reduce the structural resistance and can lead to structural collapse. The complexity of this analysis will need some powerful and efficient numerical techniques with high computing performance. This work presents the development of two different numerical models of a finite pipe element with two nodal sections for mechanical analysis. The formulation presented is based on shell displacement theory, where the displacement field is based in different polynomial functions or trigonometric functions, for rigid beam displacement, and based in development Fourier series to model warping and ovalization of pipe cross section, reported in Fonseca (2005), Thomson (1980) and Melo (1992). Finite elements models presented in this work have been compared with some numerical examples produced by other author (Thomas, 1981). Finally, some conclusions are presented about the efficiency of these finite pipe element models.

Finite Element Formulation

The deformation field of a pipe element refers to membrane strains and shell curvature variations. The following assumptions, referred in Fonseca (2005), Thomson (1980) and Melo (1992), were considered in the present analysis: the curvature radius is assumed much larger than the section radius; a semi-membrane deformation model is adopted and neglects the bending stiffness along the longitudinal direction of the toroidal shell but considers the meridional bending resulting from ovalization.

The geometric parameters (figure 1) considered for this element definition are: the arc length (s), the mean curvature radius (R), the thickness (h), the mean section radius of the pipe (r) and the central angle (α). The displacements u, v e w are calculated in shell surface from the structural element, function of a displacement field under mean line arc (U, W and φ), figure 2. Those parameters are related through simple differential equations from beam bending theory. U is the tangential displacement, W the transversal displacements and φ represent the rotation in z direction.

Two different models will be presented for the displacement field calculation in curved pipe elements.

First: a high order formulation should be used and six parameters are necessary to define the displacement field. From this, U can be approached by the following fifth order polynomial (5P):

$$U_{(s)} = a_o + a_1 s + a_2 s^2 + a_3 s^3 + a_4 s^4 + a_5 s^5 \tag{1a}$$

The transverse displacement and the rotation can be calculated:

$$W_{(s)} = -R\left(a_1 + 2a_2 s + 3a_3 s^2 + 4a_4 s^3 + 5a_5 s^4\right) \tag{1b}$$

$$\varphi_{(s)} = -R\left(2a_2 + 6a_3 s + 12a_4 s^2 + 20a_5 s^3\right) \tag{1c}$$

The coefficients are determined as a function of imposed boundary conditions under the curved referential.

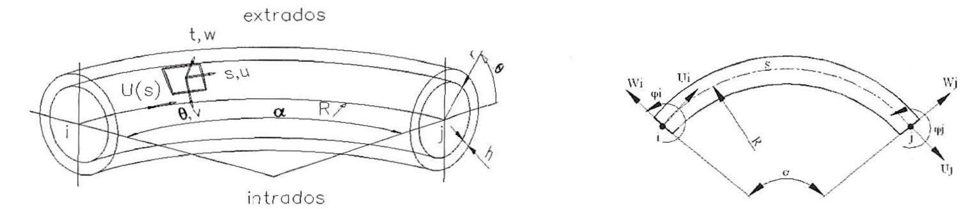


Figure1: Geometric parameters and degrees of freedom for in plane finite pipe element.

The second model is a formulation based on trigonometric functions (TF) and four parameters are necessary to define the displacement field. U can be approximated by the following function:

$$U_{(s)} = b_1 \cos\left(\frac{s}{R}\right) + b_2 \sin\left(\frac{s}{R}\right) + b_3 \cos\left(2\frac{s}{R}\right) + b_4 \sin\left(2\frac{s}{R}\right) \tag{2a}$$

The transverse displacement can be calculated:

$$W_{(s)} = -R\frac{dU}{ds} = b_1 \sin\left(\frac{s}{R}\right) - b_2 \cos\left(\frac{s}{R}\right) + b_3 2\sin\left(2\frac{s}{R}\right) - b_4 2\cos\left(2\frac{s}{R}\right) \tag{2b}$$

For the rotation field a linear polynomial may be used:

$$\varphi_{(s)} = b_5 + b_6 S \tag{2c}$$

The coefficients are determined using the same imposed boundary conditions under the curved referential.

For straight pipe elements a formulation based in third order polynomial (3P) was used with Hermitian shape functions.

The shell finite element displacement field resulting from the superposition of displacement rigid beam (equations 1 or 2) and the complete Fourier expansion for ovalization and warping terms:

$$u = U_{(s)} - r\cos\theta\varphi_{(s)} + u(s,\theta) \tag{3a}$$

$$v = -W_{(s)} \sin\theta + v(s,\theta) \tag{3b}$$

$$w = W_{(s)} \cos\theta + w(s,\theta) \tag{3c}$$

where $w(s,\theta)$ is the surface displacement in radial direction results from ovalization, $v(s,\theta)$ the meridional displacement due to ovalization and $u(s,\theta)$ is the longitudinal displacement due to warping section pipe effect.

The mechanical deformation model considers that pipe undergoes a semi-membrane strain field with the equation 4, where ε_{ss} is the longitudinal membrane strain, $\gamma_{s\theta}$ the shear strain and $\chi_{\theta\theta}$ the meridional curvature form ovalization.

$$\underset{\sim mec}{\varepsilon} = \begin{Bmatrix} \varepsilon_{ss} \\ \gamma_{s\theta} \\ \chi_{\theta\theta} \end{Bmatrix} = \begin{bmatrix} \dfrac{\partial}{\partial s} & -\dfrac{\sin\theta}{R} & \dfrac{\cos\theta}{R} \\ \dfrac{1}{r}\dfrac{\partial}{\partial\theta} & \dfrac{\partial}{\partial s} & 0 \\ 0 & -\dfrac{1}{r^2}\dfrac{\partial}{\partial\theta} & \dfrac{1}{r^2}\dfrac{\partial^2}{\partial\theta^2} \end{bmatrix} \begin{Bmatrix} u \\ v \\ w \end{Bmatrix} \tag{4}$$

The application of the virtual work principle gives finally the system of algebraic equations to be solved. The matrix force-displacement equation for this finite element pipe is:

$$[K]\{\delta\} = \{F_{mec}\} \tag{5}$$

δ is a nodal unknown displacement vector and F_{mec} the applied nodal forces. The element stiffness matrix K is calculated from the matrix equation:

$$K_{global} = [T]\left(\int_s [B]^T [D][B]dS\right)[T]^T \tag{6}$$

where $dS = rdsd\theta$, $[T]$ is the transpose matrix for global system, $[B]$ results from the derivative of the shape functions for the finite pipe element and the elasticity matrix $[D]$ appears with a simple algebraic definition, dependent of the elastic modulus E, the pipe thickness h and Poisson's ratio ν.

In the formulation of the pipe element here presented, a Gaussian integration was used along variable s while one exact was used along the circumferential direction θ.

The total number of degrees of freedom for this element is $2(3+2N_\theta)$, where N_θ is the number of trigonometric terms used in Fourier expansions equal 8 terms.

Study case: Flexibility Determination for in-Plane Bending in Different Types of Piping Elbows and Nozzle Constraints

Figure 3 show the geometry created, for in-plane bending. One-dimensional mesh was used for each model and our numerical results are compared with solution referred in Thomas (1981), when considering the pipe factor equal $\bar{h} = hR/r^2 = 0.1304$.

The piping steel elbow is subjected to a bending moment (M=73450Nm). For simulating the stiffening effect of a nozzle, a different length of constraint X is considered as well as a built-in-end extremity.

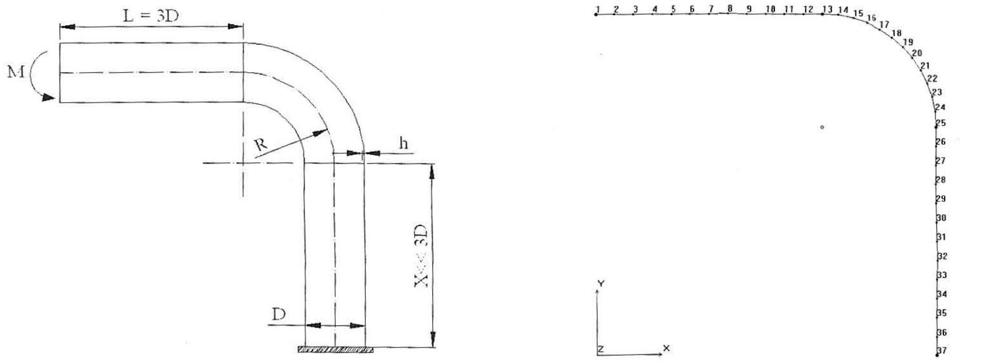
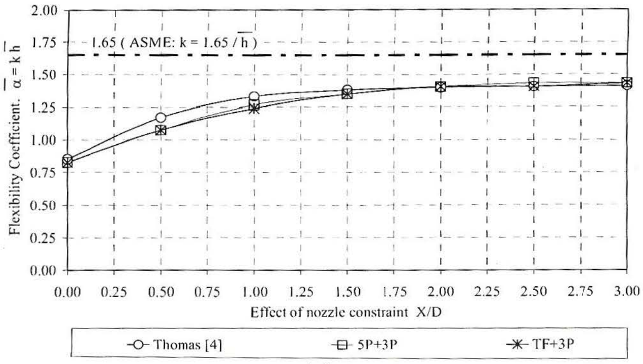


Figure 3: Piping elbows geometry and one-dimensional mesh used for in-plane bending.

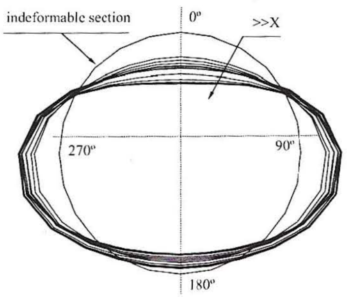
The curves shown in figure 4.a represent the flexibility coefficient, $\overline{\alpha} = 2Ert^2\varphi_{(node13)}/M$, being affected by the presence of nozzle constraint and obtained with different loading types.

Our numerical results using the two different models are in good agreement with the reference (Thomas, 1981).

Figure 4.b represent the ovalization problem obtained with the numerical results calculated at the surface shell, for any different X length, at the transverse section-elbow at mid-span.



(a)



(b)

Figure 4: Flexibility coefficient and the effect of nozzle constraint. Ovalization at mid-span elbow.

Conclusions

A computational program based on the finite element method was presented to study the mechanical behaviour of steel piping elbows. It was shown the elbow with a rigid flange at one end and a pipe on the other is about twice as stiff as an isolated elbow. The presence of a rigid restraint near one end of the elbow shifts the maximum ovalization from the mid-span of the elbow.
These conclusions are in good agreement with the observations of Thomas, when used finite thin-shell element. Two different numerical models with two-node finite pipe element have shown good capabilities for analysis of mechanical problems.
The solution here proposed appears as a simple and easy-to-handle alternative to the use of current meshes dealing with shell finite elements, which can not only demand more expensive pre-processing programs, but also generate a higher number of unknowns. It's a simple finite element, easy to operate and avoid the pre-processing mesh generation for shell definition surface, considered as an advantage for the engineering designers.

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