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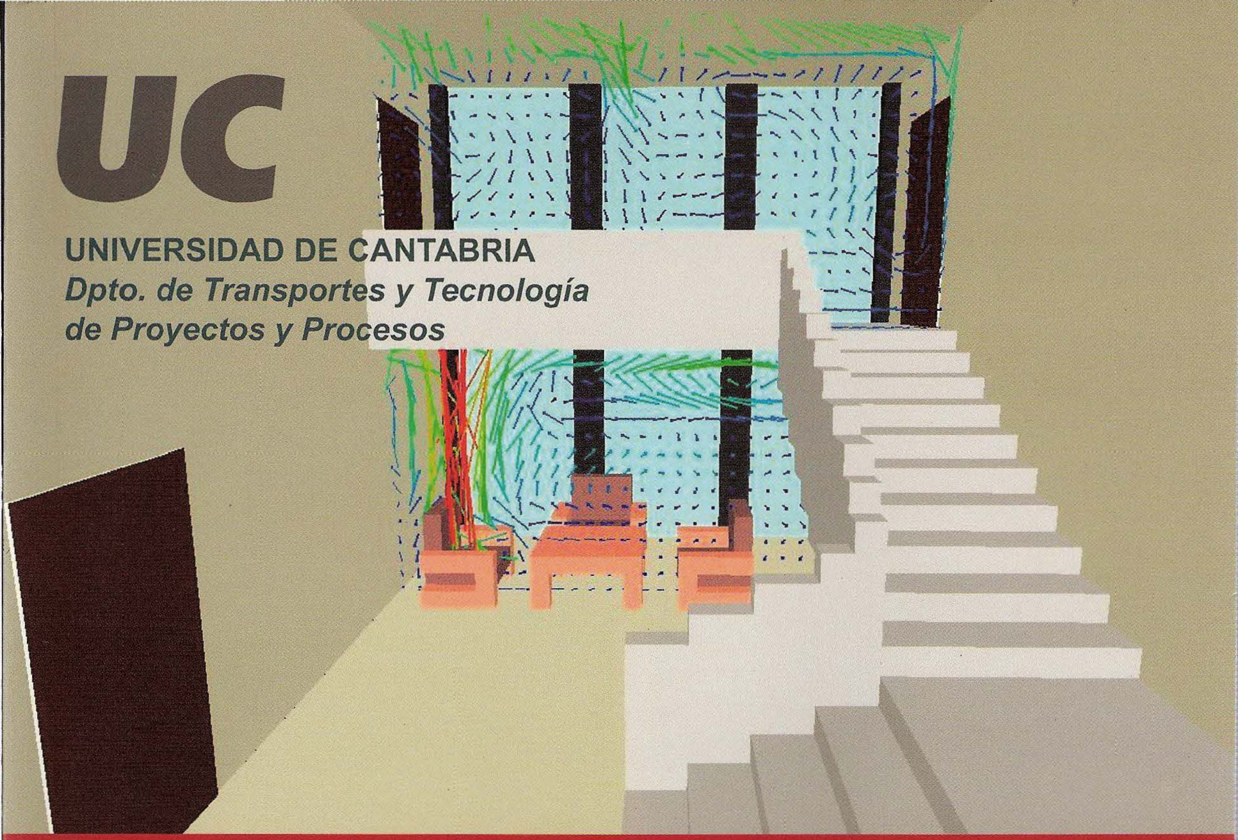

JORNADA TÉCNICA INTERNACIONAL

LIBRO DE PONENCIAS

Los Modelos de Simulación Computacional en la Ingeniería y la Investigación de Incendios

Computational Simulation Models in Fire Engineering and Research

20 de Octubre de 2004

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University of Edinburgh (UK)

Politecnico di Bari (Italia)

University of California, Berkeley (USA)

University of Southern Mississippi (USA)

Jornada Técnica Internacional

Los Modelos de Simulación Computacional en la Ingeniería y la Investigación de Incendios

Computational Simulation Models in Fire Engineering and Research

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Presentación

La Jornada Técnica Internacional sobre "LOS MODELOS DE SIMULACION COMPUTACIONAL EN LA INGENIERIA Y LA INVESTIGACION DE INCENDIOS", cuyo **Libro de Ponencias** presentamos, se celebró en la Escuela de Ingenieros Industriales y de Telecomunicación de la Universidad de Cantabria, Santander, Cantabria, España el 20 de Octubre de 2004.

Ha sido una oportunidad única para investigadores, ingenieros especializados en seguridad contra incendios, analistas de riesgos de incendios, y otras muchas disciplinas afines a la Ciencia y la Tecnología del Incendio para aproximarse más a lo que sin duda vienen siendo las herramientas mas validas en la Ingeniería de la Seguridad contra Incendios: LOS MODELOS DE SIMULACION COMPUTACIONAL.

A la Jornada se presentaron 23 ponencias en representación de **11** países (Estados Unidos, Reino Unido, Francia, Bélgica, Japón, Canadá, Rusia, Noruega, Italia, Portugal y España) y se contó con la Conferencia Magistral sobre "Fire Modeling: Development and Applications" impartida por el ilustre Dr. Carlos Fernández -Pello, Catedrático del Dpto. de Ingeniería Mecánica de la Universidad de California, Berkeley (USA).

Queremos expresar un especial reconocimiento por el trabajo desarrollado en la selección de las ponencias al Comité Científico de la Jornada integrado por los destacados Profesores e Investigadores - Dr. Jorge A. Capote (GIDAI – Universidad de Cantabria), Dr. Luis Villegas (GTED – Universidad de Cantabria), Dr. Carlos Fernández Pello (University of California, Berkeley – USA), Dr. Carlos Santolaria Morros (Universidad de Oviedo), Dr. Chris Shaw (Georgia Institute of Technology – USA), Dr. Francisco Hernández Olivares (Universidad Politécnica de Madrid), Dr. Francisco Jiménez Peris (Universidad de Córdoba), Dr. José A. Fraguela Formoso (Universidad de A Coruña), Dr. José L. Torero (University of Edinburgh – UK), Dr. Juan C. López (Universidad Politécnica de Cataluña), Dr. Noureddine Benichou (CRNC-NCR – Canadá), Dr. Pedro. J. Martínez (Universidad de Málaga), Prof. Piero Masini (Politécnico di Bari – Italia) y Dr. Tulio Sulbaran (University of Southern Mississippi – USA). Destacamos su gran aporte para lograr una Jornada con la más alta calidad científico-técnica.

Nuestras más expresivas gracias a los autores y ponentes quienes han dedicado su tiempo y esfuerzo para traernos en sus presentaciones, sus experiencias y metodologías en la aplicación de la simulación computacional a este apasionante campo de la Ingeniería y la Investigación de Incendios.

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20 de Octubre de 2004

The Thermal Modelling of Structural Piping Systems Under Fire Conditions

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ABSTRACT

This work presents a finite element formulation to model the thermal behaviour of structural piping systems under fire conditions. The study of steel structures at elevated temperatures needs the thermal action characterisation and the non-linear material behaviour, according to the Eurocodes standards. The surrounding fire temperature in structural systems is in accordance with ISO 834. Structural piping systems may have also, internal voids filled with air that needed to be simulated. In this domain, the internal temperature may be calculated with some simplified formulas obtained from heat transfer equations. For small values of the pipe thickness-mean radius ratio, the thermal behaviour may be determined with high accuracy using one dimensional mesh approach, for axisymmetric thermal boundary conditions, across pipe sections. The transient temperature evolution in piping systems and the internal temperature in the voids will be calculated. Conclusions are presented regarding the importance of the temperature field obtained in structural piping systems using one and two dimensional finite element meshing and results will be discussed for various study cases. Numerical results will be compared with other finite element commercial code, Cosmos/M, for the same situation.

1 INTRODUCTION

A finite element code has been developed to model the thermal behaviour of structural piping systems exposed to fire conditions [1-3]. A finite element formulation for heat conduction in solids is presented, with particular attention to material non-linearity problem, modelled by an iterative procedure based on the modified Newton-Raphson method. The simplified heat conduction equation included in Eurocode 3 – EC3 [4], is also presented. Structures may have internal voids filled with air (hollow columns, profile sections thermally insulated, tubular structures,...). In the presence of these cavities, the internal air temperature will be determined with some simplified formulas presented herein [5-7]. The results of this formulation will be presented for several studied cases. The transient temperature evolution in tubular structures subjected to fire condition will also be calculated, using one and two dimensional finite element modelling.

2 THE HEAT CONDUCTION EQUATION AND BOUNDARY CONDITIONS

The equation for transient heat conduction calculation, in domain Ω, takes the form:

$$\frac{\partial}{\partial x}\left(\lambda\frac{\partial\theta}{\partial x}\right)+\frac{\partial}{\partial y}\left(\lambda\frac{\partial\theta}{\partial y}\right)+\frac{\partial}{\partial z}\left(\lambda\frac{\partial\theta}{\partial z}\right)+\dot{Q}=\rho C\frac{\partial\theta}{\partial t} \tag{1}$$

where: λ is the thermal conductivity, $\dot{Q}$ the heat generated/unit volume, ρ the material specific mass, C the specific heat, θ the temperature and t the time.

The temperature field which satisfies this equation satisfy the following boundary conditions: prescribed temperatures $\overline{\theta}$; specified heat flux $\overline{q}$; heat flux by convection, heat flux by radiation and the environment at the temperature θ_∞.

The convection global effect is calculated by this equation:

$$q_c = h_c(\theta-\theta_\infty) \text{ on } \Gamma_c \tag{2}$$

where h_c is the heat transfer coefficient by convection.

The heat radiation flux through a part Γ_r of the boundary at the temperature θ and the environment at the absolute temperature θ_a is represented the following equation:

$$q_\mathrm{r} = \beta\varepsilon(\theta^4-\theta_a^4) = \underbrace{\beta\varepsilon(\theta^2+\theta_a^2)(\theta+\theta_a)}_{h_r}(\theta-\theta_a) = h_r(\theta-\theta_a) \text{ on } \Gamma_r \tag{3}$$

β is the Stefan-Boltzmann constant, ε is the emissivity and h_r is the heat transfer coefficient by radiation.

If the heat flux occurs simultaneously by convection and radiation and if in particular $\theta_\infty=\theta_a$, equation (2) and equation (3) may be rewritten as follows:

$$q_{cr} = q_c+q_r = h_c(\theta-\theta_\infty)+h_r(\theta-\theta_a) = h_{cr}(\theta-\theta_\infty) \tag{4}$$

$h_{cr}=h_c+h_r$ is the combined convection and radiation heat transfer coefficient.

For θ_∞ calculation we have used a standard temperature-time curve ISO 834, represented in figure 1, according Eurocode 1 – EC1 [8] with the following expression:

$$\theta_\infty = 20 + 345\log_{10}(8t+1) \tag{5}$$

t is the time in minutes [min].

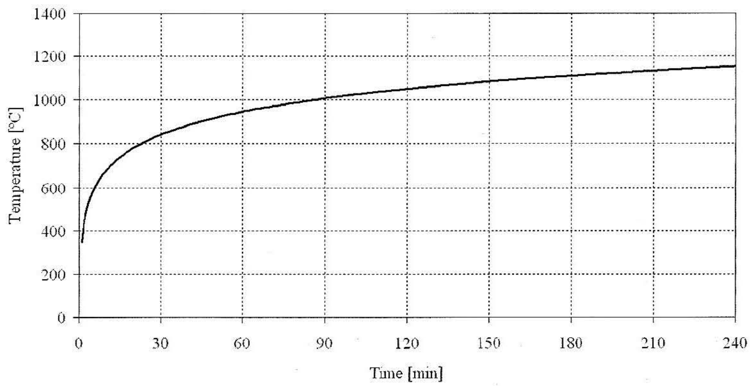


Fig. 1 – Standard temperature-time curve.

Using finite elements Ω^e to discretize the domain Ω, a weak formulation weigh functions based on the Galerkin Method is used, giving a system of differential equations:

$$\sum_{j=1}^{m}\left[\int_{\Omega e}\left(\frac{\partial N_i}{\partial x}\lambda\frac{\partial N_j}{\partial x}+\frac{\partial N_i}{\partial y}\lambda\frac{\partial N_j}{\partial y}+\frac{\partial N_i}{\partial z}\lambda\frac{\partial N_j}{\partial z}\right)d\Omega^e + \int_{\Gamma h^e} h_{cr}N_iN_j d\Gamma^e - \int_{\Gamma\theta^e} N_i\lambda\frac{\partial N_i}{\partial n}d\Gamma_\theta{}^e\right]\boldsymbol{\theta}_j +$$

$$\sum_{j=1}^{m}\left[\int_{\Omega^e}\rho C N_i N_j d\Omega^e\right]\dot{\boldsymbol{\theta}}_j = \int_{\Omega^e} N_i\dot{Q}d\Omega^e - \int_{\Gamma q^e} N_i\bar{q}d\Gamma_q^e + \int_{\Gamma h^e} N_i h_{cr}\theta_\infty d\Gamma_h^e \tag{6}$$

m is the total number of elements, N_i and N_j represents shape functions for one or two dimensional problems.

Using a finite difference technique to time discretization, the system of ordinary differential equations (6) results in the recurrence formula:

$$(\mathbf{K}_{n+\alpha} + \frac{\mathbf{C}_{n+\alpha}}{\alpha\Delta t})\boldsymbol{\theta}_{n+\alpha} = \mathbf{F}_{n+\alpha} + \frac{\mathbf{C}_{n+\alpha}}{\alpha\Delta t}\boldsymbol{\theta}_n \tag{7}$$

Solving the system of equations for $\boldsymbol{\theta}_{n+\alpha}$ at time $t_{n+\alpha}$, the value of $\boldsymbol{\theta}$ at the end of the time interval Δt, at time t_{n+1}, is given by the following equation.

$$\boldsymbol{\theta}_{n+1} = \frac{1}{\alpha}\boldsymbol{\theta}_{n+\alpha} + \left(1-\frac{1}{\alpha}\right)\boldsymbol{\theta}_n \tag{8}$$

α is a constant parameter used for several time integration schemes [1].

For non-linear problems, where the thermal material properties are temperature dependent, the system of equations (6) can generally be written as:

$$\mathbf{K}(\theta,t)\boldsymbol{\theta}(t)+\mathbf{C}(\theta,t)\dot{\boldsymbol{\theta}}(t)=\mathbf{F}(\theta,t) \tag{9}$$

In order to fully satisfy these non-linear problem conditions, it is necessary to employ an iterative procedure in each time step. In this algorithm a modified Newton-Raphson method is adopted [1].

2.1 Temperature Determination Inside the Void

Based on equation (6) for the heat transfer and considering the following hypotheses, the product of the specific heat and the air density can be neglected; also the air thermal conductivity may be neglected when compared with steel thermal conductivity of the structural element [4-6].

The following equation makes possible the evaluation of the temperature inside the void:

$$\theta_{Void}=\sum_{e=1}^{N_{FR}}\int_{\Gamma_h^e} N_l h_{cr}\theta_l d\Gamma_h^e \Big/ \sum_{e=1}^{N_{FR}}\int_{\Gamma_h^e} N_l N_m h_{cr} d\Gamma_h^e \tag{10}$$

where N_{FR} is the number of boundary elements at void region and θ_l is the calculated temperature at each element node.

At any time the fictitious temperature inside void will be considered uniform, determined by the heat convective and radiactive fluxes received from all the elements surrounding that region, represented in figure 2, where E_i represents the finite element $_i$ and N_i is the node at internal surface boundary.

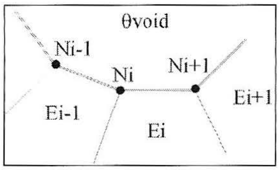


Fig. 2 - An internal void scheme.

2.2 Steel Properties According EC3

The thermal properties of steel are a function of the temperature and may be determined from EC3 [4]. The unit mass of steel ρ may be considered a temperature independent parameter and the value adopted is $\rho=7850$ [kg/m^3]. The specific heat of steel C is represented in figure 3 and may be determined from the EC3 formulas.

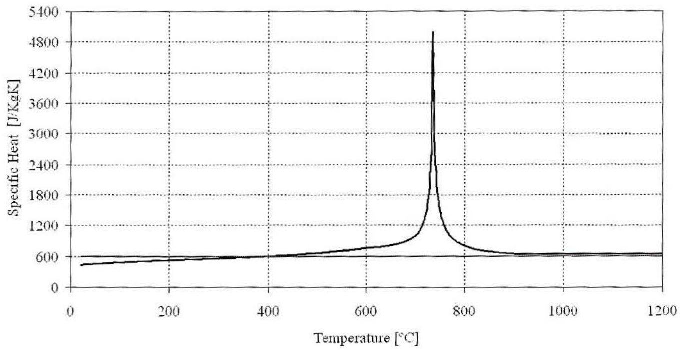


Fig. 3 – Specific heat variation.

The thermal conductivity of steel λ is represented in figure 4, according equations from EC3.

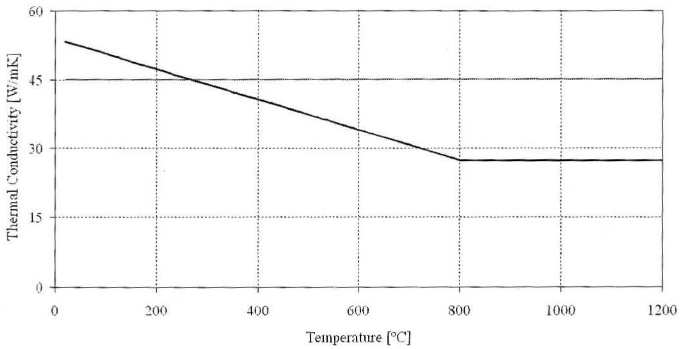


Fig. 4 – Thermal conductivity variation.

2.3 Simplified Equation According EC3

For an equivalent uniform temperature distribution in the cross-section, the increase of temperature $\Delta\theta_{a,t}$ in an unprotected steel member during a time interval Δt is determined from the simplified equation from EC3 [4].

$$\Delta\theta_{a,t} = \frac{A_m / V}{C\rho} \dot{h}_{net,d} \Delta t \tag{11}$$

where A_m / V is the section factor for unprotected steel members [m^{-1}], A_m is the exposed surface area of the member per unit length [m^2/m], V the volume of the member per unit length [m^3/m], $\dot{h}_{net,d}$ the design value of the net heat flux due to convection and radiation given by the following expression.

$$\dot{h}_{net,d} = \gamma_{n,c}\dot{h}_{net,c} + \gamma_{n,r}\dot{h}_{net,r} \text{ [W/m}^2\text{]} \tag{12}$$

$\gamma_{n,c}$ and $\gamma_{n,c}$ are equals 1.0, $\dot{h}_{net,c}$ should be calculated according to:

$$\dot{h}_{net,c} = h_c(\theta_g - \theta_m) \ [\mathrm{W/m^2}] \tag{13}$$

h_c may be taken as 25[W/m^{2}K], θ_m is the surface temperature of the member, θ_g is the gas temperature of the surrounding environment member in fire exposure done by equation 5.

$\dot{h}_{net,r}$ is the design value of the heat flux due to radiation given by expression:

$$\dot{h}_{net,r} = \Phi \cdot \varepsilon_{res} \cdot 5.67 \times 10^{-8} \cdot [(\theta_r + 273)^4 - (\theta_m + 273)^4] \ [\mathrm{W/m^2}] \tag{14}$$

Φ is the configuration factor, which should be taken equal unity, θ_r is the radiation temperature of the environment of the member usually taken as $\theta_r = \theta_g$, Δt time interval, which should not be taken as more than 5 seconds, $\varepsilon_{res} = 0.5$ is the resultant emissivity.

3 CASE STUDY 1: TEMPERATURE VARIATION ACROSS PIPE THICKNESS UNDER FIRE CONDITIONS

The temperature field in structural pipe system is presented, using two and one dimensional finite element modelling, figure 5.

Only one quarter of section are analysed due to the symmetry of boundary condition and geometry, when we used two dimensional meshes. For one dimensional mesh we have used the entire length pipe. The piping system is subjected to external fire conditions, using the standard fire curve ISO 834. According to the EC1 [8], in the surfaces not exposed to fire, the convection coefficient is equal a 9[W/m^{2o}K] and the radiation effect is neglected. In the surfaces exposed to fire the convection coefficient is equal 25[W/m^{2o}K] and the emissivity equal to 0.5.

Some tubular sections have been studied with different relations between the thickness and the section mean radius h/r [9].

The results are obtained using the developed code and compared with the solution using the simplified EC3 equation. We considered different situations. First, we used the internal void theory, second the internal region was considered insulated and finally the internal cavity was filled with air.

The figures 6, 7 and 8 represents the temperatures field obtained in thick curved pipes at two different times, using a two dimensional mesh.

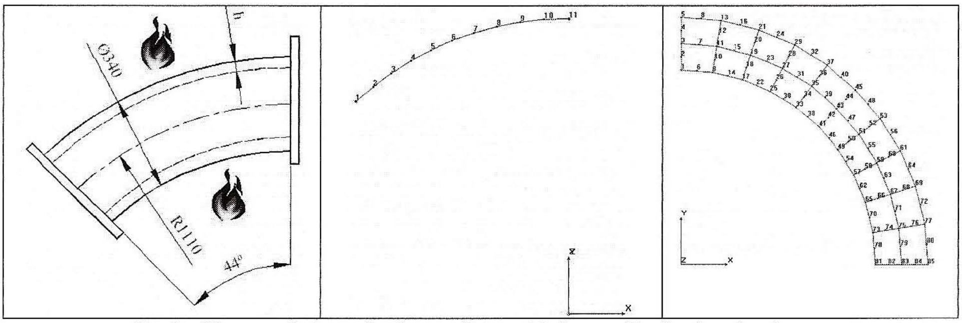


Fig. 5 – The curved pipe under fire conditions. Meshes used in developed code.

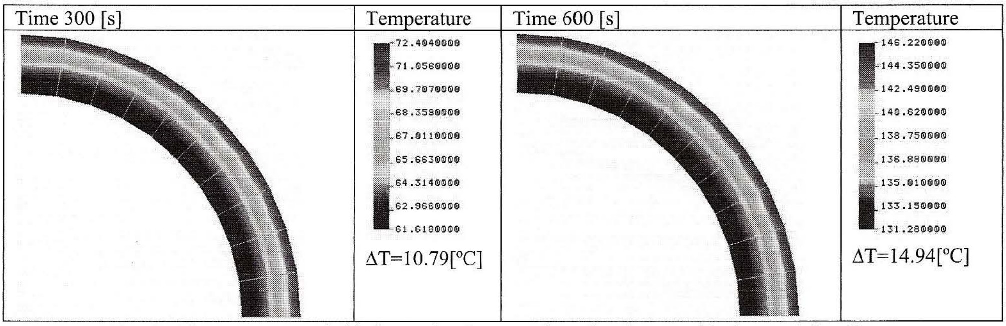


Fig. 6 – The temperature field obtained with internal insulated region (thickness 40 [mm]).

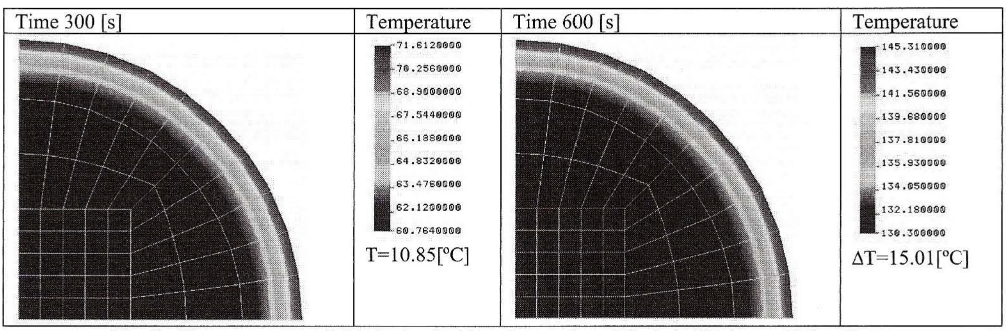


Fig. 7 – The temperature field obtained with an internal modelled air (thickness 40 [mm]).

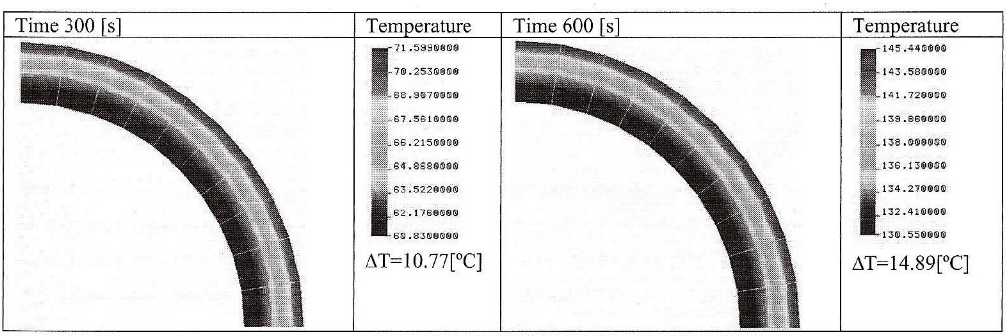


Fig. 8 – The temperature field obtained with internal void (thickness 40 [mm]).

Changing the relation h/r, the temperature cross the pipe thickness varies also and the simplified equation does not correspond to a real value. The same is applied for the one dimensional mesh, because the temperature calculation is determined for the medium pipe surface. So the one dimensional modelling may be applied to thin structural piping systems giving good results. Figures 9 and 10 show the temperature evolution for two of different relation tubes h/r, obtained using the two dimensional and one dimensional meshes at node of mean radius. The results are compared with the simplified heat equation.

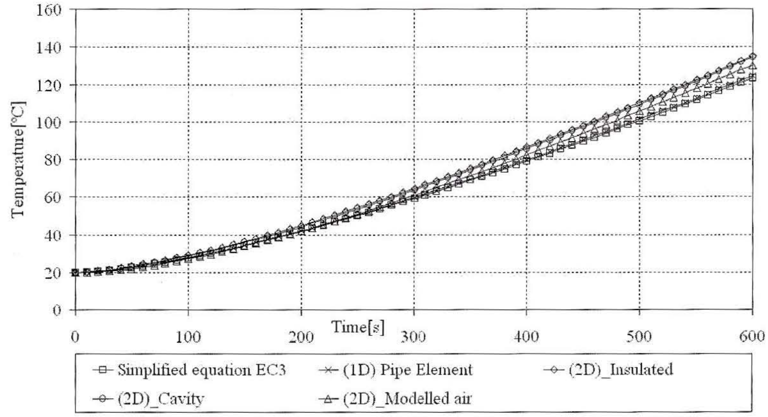


Fig. 9 – Time history of temperature of tube with the relation $h/r = 0.235$.

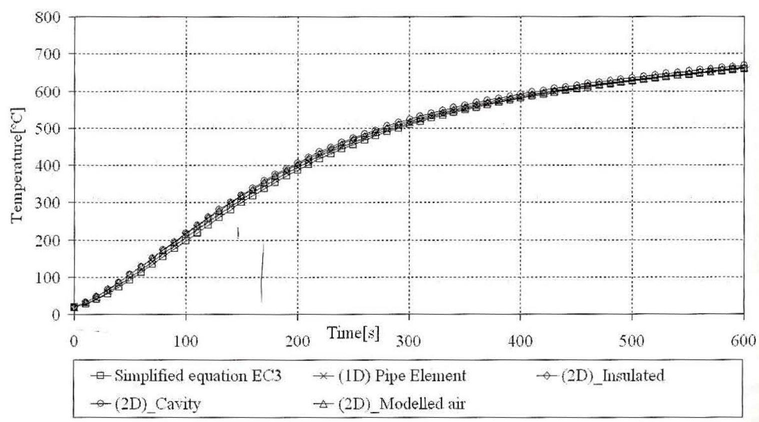


Fig. 10 – Time history of temperature of tube with the relation $h/r = 0.007$.

for thin structural pipes, the temperature variation across thickness is small and does not increase with fire action. for higher piping structure thickness the curves are not coincident, because there is a considerable temperature variation across thickness, as can see using two dimensional models. Figure 11 represents the temperature variation across thickness for different relation h/r at two different instant times.

In circular pipe sections due the axisymmetrie, considering the some boundary condition along the section radius, the inside surface temperature is uniform. As temperature is uniform inside voids, heat flux by radiation should not be considered, neglecting this type of heat transfer in structural piping systems. Temperature gradient across pipe thickness decrease has been verified for thin structural pipes and a uniform temperature may be considered as well the simplified heat equation EC3 proposes. So it is possible to obtain the temperature field for thin structural piping systems using one dimensional finite element model.

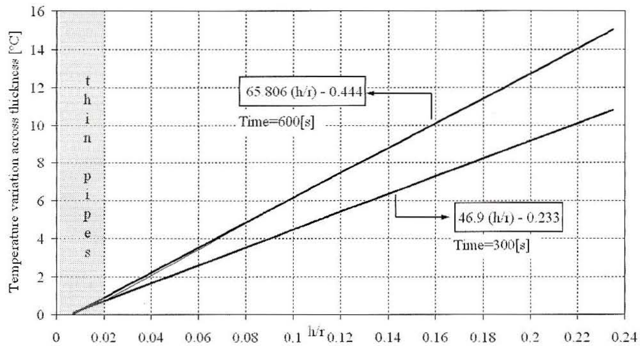


Fig. 11 – Temperature variation across pipe thickness.

4 CASE STUDY 2: TEMPERATURE CALCULATION OF A STRUCTURAL PIPING SYSTEM UNDER FIRE CONDITIONS

The structural piping system presented is composed by six elbows of 90° ASTM A234 and seven straight pipes ASTM A106 subjected to external fire conditions, figure 12. The mean radius is equal to 44.62 [mm] and the thickness is 4 [mm]. The curvature radius is 57.2 [mm]. All pipes have the same length equal to 100 [mm].

The material considered is steel and the all properties are according to EC3. A one dimensional mesh will be used, based on the new finite element formulation developed [9]. Comparison with a three dimensional finite shell element will be presented based on a commercial finite element programme Cosmos/M.

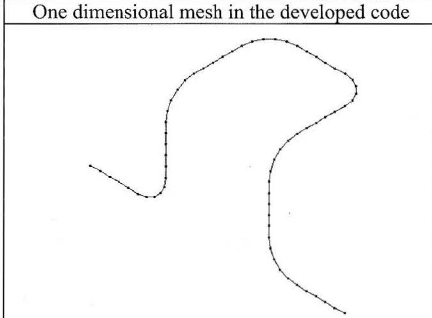
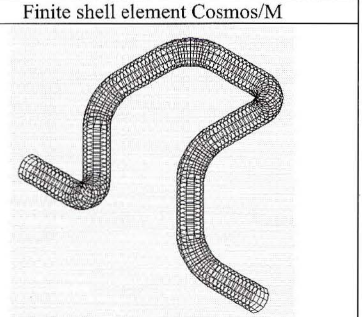
One dimensional mesh in the developed code	Finite shell element Cosmos/M
	

Fig. 12 – Finites elements used.

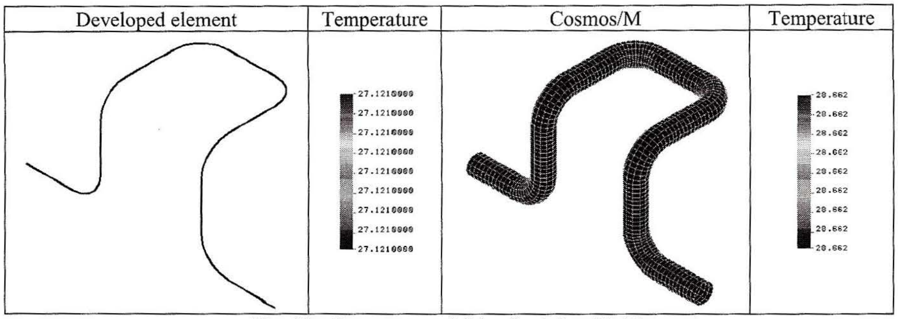


Fig. 13 – Temperature field at final time 20 [s].

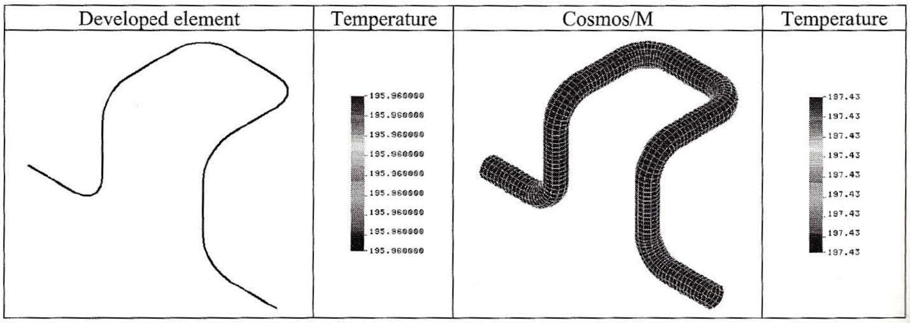


Fig. 14 – Temperature field at final time 200 [s].

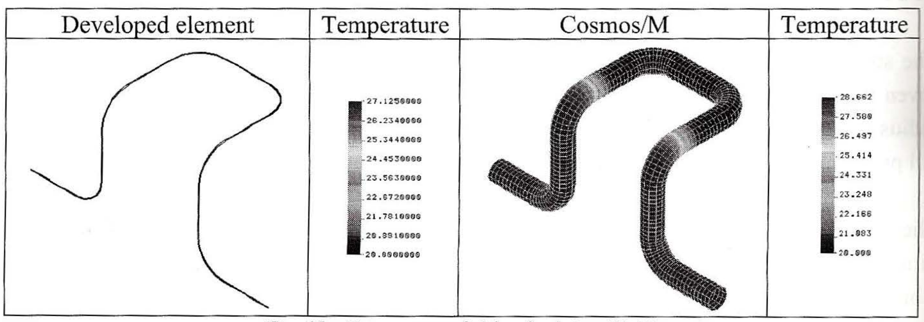


Fig. 15 – Temperature field at final time 20 [s]

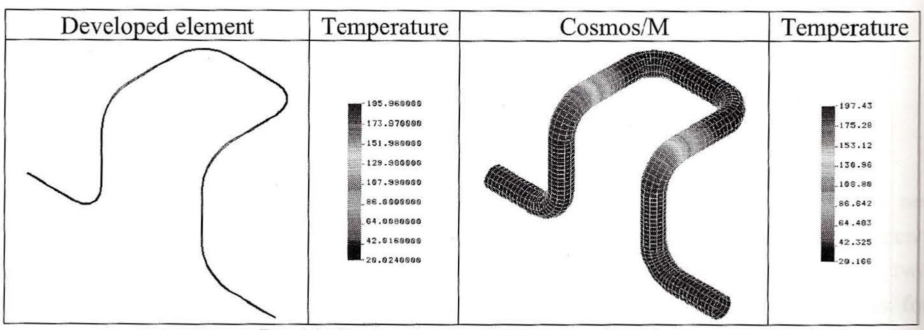


Fig. 16 – Temperature field at final time 200 [s].

For this fire condition; temperature distribution is considered constant across section mean radius and along piping system, because we have considered all the piping system exposed to this thermal action. The figures 13 and 14 represent the nodal temperatures at two different time steps using the referred models.

A new analysis will be presented for the same structural piping system, considering an insulated upper zone with other parts subjected to fire conditions. In figures 15 and 16 a comparison between the new formulation and the commercial code is presented for two distinct instant times. The results present good agreement and the insulated zone maintain the uniform initial temperature at 20 [ºC].

5 CONCLUSIONS

A computational program based on the finite element method was presented to study thermal model behaviour of piping systems subjected to fire conditions. The internal void temperature calculation in a structural element may be calculated using simplified formulas and reducing the computing effort when compared with other methods. The results of a transient temperature field obtained with the developed code have been compared with the results obtained with the simplified heat conduction equation and with the results obtained from 3D meshes derived from commercial codes. Based in the analysed cases studies, it may be concluded that for thin piping systems the temperature field can be obtained with less computational effort using one dimensional mesh for an external axisymmetric boundary condition.

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