In this paper, the thermo-acoustic behaviour of a rectangular panel fully immersed in a compressible fluid at rest is investigated. A boundary element method (BEM) has been employed taking into account the Kirchhoff-Helmholtz (K-H) integral equation for the acoustic pressure and with the fluid-plate interface boundary condition, the acoustic pressure jump over the panel is calculated. The thermal effects are considered regarding in the form of a uniform increment of temperature of the panel and are analysed in order to prevent the buckling phenomena. The deformation modes of the panel correspond to the vacuum case. Applying a collocation method for the panel equation, the natural frequencies are obtained. The effect of several geometric parameters regarding different thermal loads, on these frequencies are evaluated. Furthermore, the influence of the wave number for different temperatures of the panel on the acoustic damping ratio is evaluated, as well as the acoustic radiation efficiency for the different modes. The verification of the method is proven with other works.

Keywords: rectangular panel; thermo-acoustic behaviour; fluid-structure; natural frequencies; boundary element method.

1. Introduction

Fluid structure interaction problems are of great interest because of the wide variety of technological applications. Also the study of the dynamic behavior of plates or panels is interesting due to the wide variety of applications in technology found in aerospace, civil or naval engineering. This structures operates under different environments like thermal, moisture, etc, that affects its performance and stability. In satellite applications such as antennas or solar panels is interesting to know the effects of the thermal loads on its dynamic characteristics, due mainly to solar radiation that heats it severely, making impacts on the material properties, structure configuration, changing the panel stress state and altering the stiffness properties. Furthermore it is interesting to know the effects of the surrounding fluid on the dynamic vibration characteristics of these light and large structures in space-based applications that influences its performance efficiency as reflecting surfaces. In this way, the vibration essays in earth does not need to be made in vacuum chambers with the added difficulties, and simplifying these experimental processes, and the results can be extrapolated to the space environment (vacuum).

With the increased importance of thermal environment of space vehicles in aerospace industry, the structural dynamic behaviour of plates subjected to thermal loads is interesting. This is the case of sandwich panels that are widely used not only in aerospace, but also in automobile and railway applications. The thermal environment may change the stiffness and dynamic response of these structures. The early literature regarding structure dynamic behaviour under thermal environments date from 1950s, Boley et. al [1957]. Moreover, the plate interaction with an acoustic medium of radiating characteristics and considering thermal effects, involve some new difficulties. The dynamic response of structures affected by thermal environment loads has been studied for many years. Shu et al. [2000] studied the thermo-elasstical coupling free vibration behaviour of a clamped circular plate in which the dynamic response is obtained by solving a nonlinear time
dependent problem. Kim [2005] developed a theoretical method to investigate the vibration of functionally graded rectangular plates under thermal environments. Thermal stresses and temperature-dependent material properties were considered in the study. The results show the influence of the temperature, material composition and geometry on the vibration response. Jeyaraj et al. [2009] investigated the vibroacoustic behaviour of isotropic and composite plate subjected to thermal loads combining finite element and boundary element methods. Li and Yu [2015] analysed the forced vibrations of sandwich plate under acoustic and thermal environments, taking into account the critical temperature to prevent thermal load excess. The works of Li et al. [2017] and Liu & Li [2013] where the vibro-acoustic response of a clamped rectangular sandwich panel in thermal environment is investigated. Geng and Li [2012] studied the dynamic and acoustic characteristics of a flat plate under thermal environments, showing the thermal loads influence on the natural frequencies, mainly the fundamental frequency. Recently, many works made by Sharma et al [2017,2018] regarding the acoustic radiation properties of flat and curved composite vibrating structures subjected to an harmonic point excitation with different boundary conditions, employing a high order shear deformation theory, considering thermal and moisture environments, and baffled or unbaffled panel situations.

In addition, there are currently some interesting technological applications in the field of acoustics with respect to the acoustic-structure interaction in relation to passive and active control acoustics, as is the case of acoustic black holes, see Cao et al [2020], active acoustic metamaterials, see Kumar and Lee [2019], and in the thermoacoustic interaction of gases as in Rijke’s tube, see Xing et al [2020].

In this work the temperature and material properties are considered uniform over the panel, but in general are not uniform, for example for the case of functionally graded materials, as in the works of Sobhy [2018] and Zhou [2018].

The work of Gascón-Pérez [2018] where the vibroacoustic behaviour of a rectangular membrane is analyzed by the application of a BEM. Wallace [1972] presents a study of the radiation resistance or efficiency for the different modes of a rectangular panel baffled in an infinite plane by applying the Rayleigh’s integral for the acoustic pressure on one side of this plane, and not as in the present work where the acoustic pressure is obtained on the two sides of the panel submerged in the fluid by applying a BEM with the Helmholtz’s integral equation.

The thermo-acoustic behaviour of a simply supported rectangular panel is analyzed in this paper considering the influence of several parameters and the temperature on its free vibration response.

The novelty of the present work is highlighted in the results obtained for a simply supported unbaffled rectangular panel where the thermoacoustic behaviour is analyzed and the free vibration response is obtained calculating the natural frequencies as function of various geometrical parameters and temperatures of the panel, as well as the variation of the acoustic damping ratio with the reduced frequency (wave number) for different thermal loads, and also the acoustic radiation efficiency for the different modes with the wave number.

2. Analytical Formulation

The thermo-acoustic response of a rectangular panel with thermal stress due only to a uniform increment of temperature and fully immersed in the fluid is governed by the equation, see Wang et al. [2014] and Boley et al. [1997]:

\[ DV^4 u(x,y,t) + \frac{N_f}{1-\nu} \Delta u(x,y,t) + \rho \frac{\partial^2 u}{\partial t^2} = \delta p(x,y,t) \]  

(2.1)
With \( u \) the normal displacement of the plate, \( D = \frac{E \cdot t_h^3}{12(1 - \nu^2)} \) the flexural rigidity, \( E \) the Young’s modulus of the material, \( t_h \) the panel thickness, \( \nu \) the Poisson’s ratio and \( \rho_n \) the material density of the panel.

\( N_f \) is a thermal parameter regarding the in-plane forces of the panel associated to the thermal loads that is expressed as:

\[
N_f = \alpha t_h E \Theta
\]  

(2.2)

Being \( \alpha \) the thermal expansion coefficient of the panel material and \( \Theta \) the uniform temperature (increment of temperature) of the panel, where it has been considered the zero value for the reference temperature of the panel. All material properties are considered constant, that is, independent of temperature.

Fig. 1 represents the configuration of the panel submerged in an infinite fluid at rest.

The acoustic action over the panel due to the fluid is the pressure jump \( \delta p \) between its upper and lower sides.

Considering harmonic motion for both the fluid and structure, the solution takes this form:

\[
u(x, y, t) = \tilde{u}(x, y) \cdot e^{-i\omega t} \quad \text{and} \quad p(x, y, z, t) = \tilde{p}(x, y, z) \cdot e^{-i\omega t}
\]  

(2.3)

The deflection of the panel coupled with fluid is expressed in series of the deformation modes in vacuum:

\[
u(x, y) = \sum_n \sum_m \tilde{u}_n^m(x, y) = \sum_n \sum_m \varphi_n^m(x, y) \cdot \overline{\nu}_n^m
\]  

(2.4)

Where \( \overline{\nu}_n^m \) represent the weight coefficient associated to each mode, that takes into account the effect of the acoustic medium.

The modes in vacuum \( \varphi_n^m(x, y) \) for a simply supported panel, have the general expression Wang [2014].

\[
\varphi_n^m(x, y) = \sin \frac{m\pi x}{l_x} \cdot \sin \frac{n\pi y}{l_y}
\]  

(2.5)

Being \( m, n \) integer numbers associated to the number of nodes of null deflection.
Considering the wave equation for the fluid pressure, and the assumption of harmonic motion, the Helmholtz’s equation is deduced:

$$\Delta \tilde{p} + k^2 \tilde{p} = 0$$  \hspace{1cm} (2.6)

Where \( k = \frac{\omega}{a_e} \) is the wave number, being \( a_e \), the sound speed.

And considering the interface boundary condition of the momentum equation:

$$\frac{\partial \tilde{p}(x, y, z)}{\partial z} \bigg|_{z=0} = \frac{\rho_e \omega^2 \tilde{u}(x, y)}{\delta} + \frac{\rho_e \omega^2}{\kappa} \sum_{n} \sum_{m} \phi_n^m(x, y) \cdot \Pi_n^m$$  \hspace{1cm} (2.7)

at \( z = 0 \) and \( 0 < x < l_x \), \( 0 < y < l_y \).

### 2.1 Buckling Temperature

The buckling temperature is calculated finding the nonzero solutions for the thermo-elastic static problem defined by the following equation:

$$D \nabla^2 \tilde{u}(x, y) + \frac{N_T}{1 - \nu} \Delta \tilde{u}(x, y) = 0$$  \hspace{1cm} (2.8)

Taking into account the expression for the deformation, equation (2.4), it is deduced the expression:

$$D \left( \frac{m \pi}{l_x} \right)^2 + \left( \frac{n \pi}{l_y} \right)^2 + \frac{N_T}{1 - \nu} = 0$$  \hspace{1cm} (2.9)

Considering the expression for the thermal parameter, equation (2.2), and the flexural rigidity, the critical buckling temperature corresponding to the fundamental mode is:

$$\Delta T_{cr} = \Theta_{cr} = \frac{l_x^2 \pi^2}{12(1 + \nu) \alpha} \left[ \left( \frac{1}{l_x} \right)^2 + \left( \frac{1}{l_y} \right)^2 \right]$$  \hspace{1cm} (2.10)

This critical buckling temperature is considered to prevent any excess thermal loading in the analysis of the results.

### 2.2 Calculus of the pressure jump over the plate

For the panel immersed in the fluid, applying the K-H integral equation (from Refs. Crocker [2007] and Fahy [2001]) for the pressure field and considering the boundary condition (2.7), the following expression is deduced:

$$\rho_e \omega^2 \tilde{u}(x, y) = -\frac{1}{4\pi} \int_{\Sigma_{\rho}} \delta \tilde{p}(\xi, \eta) \frac{\partial^2}{\partial z^2} \left( \frac{e^{\delta z}}{\delta} \right) \bigg|_{z=0} d\xi d\eta$$  \hspace{1cm} (2.11)

Being \( \delta \), the distance between the field and source points and \( \Sigma_{\rho} \) the panel surface.

Let

$$Y = \left. \frac{\partial^2}{\partial z^2} \left( \frac{e^{\delta z}}{\delta} \right) \right|_{z=0}$$  \hspace{1cm} (2.12)

be the kernel of the integral equation. For the approximate solution of equation (2.11), a collocation method is applied.
Then the equation (2.11) can be expressed for the associated modes of deformation and pressure jump, and by applying the collocation method:

$$\rho \omega^2 \phi_m^*(x, y) \Pi_m^* = -\frac{1}{4\pi} \sum_{r=1}^{N_r} \sum_{s=1}^{N_s} \delta P_m^*(\zeta, \eta) \int \int \gamma(x - \xi, y - \eta, \zeta, \eta, \xi, \eta) d\xi d\eta$$

(2.13)

When evaluating the integral of the kernel in each rectangular element special care must be taken of the singularity when $\xi \to x_i$ and $\eta \to y_j$ so a principal value is taken, see Roussos [13], for more details see Gascón-Pérez [2018]:

Thus finally considering the equation (2.13), and solving for the modal pressure jump, it is obtained:

$$\left\{ \delta P_m^*(\zeta, \eta) \right\} = \rho \omega^2 \left[ \Gamma_m^* \right]^{-1} \left\{ \phi_m^*(x, y) \right\} \Pi_m^*$$

(2.14)

### 2.3 Calculus of the natural frequencies of the panel

Applying the generalized work equation of the $mn$ modal forces of the panel equation associated to the $uv$ modal panel deflection and making variation of the modal numbers $m, n, u, v = 1, 2, \ldots, N$, the following system is obtained, leading to an eigenvalue problem:

$$\left[ [K] - \omega^2 \left( [M] + [M_f] \right) \right] \{\Pi\} = \{0\}$$

(2.15)

Where the coefficients of these matrices are expressed:

$$K_m^* = D \left[ \Delta \phi_m^* \right] \left[ \Delta \sigma \right] \{\phi_m^*\} + \frac{N_r}{1 - \nu} \left[ \Delta \phi_m^* \right] \left[ \Delta \sigma \right] \{\phi_m^*\}$$

(2.16.1)

$$M_m^* = \rho_c \left[ \phi_m^* \right] \left[ \Delta \sigma \right] \{\phi_m^*\}$$

(2.16.2)

$$M_f^* = \rho_f \left[ \phi_m^* \right] \left[ \Gamma_f^{-1} \right] \left[ \Delta \sigma \right] \{\phi_m^*\}$$

(2.16.3)

Where $[\Delta \sigma]$ is the matrix associated to the panel surface.

Because the fluid mass matrix depends of the wave number $k$, the solution is obtained by an iteration procedure in an easy way due to the fast convergence. The modes of the coupled fluid-structure can be obtained taking into account the eigenvector $\{\Pi\}$ and the corresponding modes in vacuum.

### 2.4 Radiation efficiency

The acoustic power is calculated from the following expression:

$$P_a = \text{Re} \left\{ \int \int \delta P \cdot \hat{u} \cdot d\sigma \right\}$$

(2.17)

And the acoustic radiation efficiency defined as:

$$\eta = \frac{P_a}{\rho_c \sigma \Sigma_p \langle \hat{u}^2 \rangle}$$

(2.18)

Being $\langle \hat{u}^2 \rangle$ the quadratic medium panel velocity that is calculated as:

$$\langle \hat{u}^2 \rangle = \frac{1}{\Sigma_p \Sigma_x} \int \int \hat{u}^2 \cdot d\sigma$$

(2.19)
After average integration of the acoustic power and the quadratic medium panel velocity over the time period for harmonic motion, the acoustic radiation efficiency associated to each mode have the expression:

\[ \eta_n^m = \text{Re} \left\{ -i k \frac{\phi_n^m \left[ \left[ \Gamma \right]^{-1} \right]^T \left[ \Delta \sigma \right] \phi_n^m}{\phi_n^m \left[ \Delta \sigma \right] \phi_n^m} \right\} \]  

(2.20)

3. Results and discussion

For validation of the method regarding the fluid structure interaction, the results obtained by Kwak [1996] for the frequencies of a vibrating simply supported rectangular panel in contact with water are compared with the present method (the whole procedure is similar to that of equation (2.1) without the thermal term) for the different modes, and are found to be in a good agreement, although the case studied by Kwak is not exactly the same because an infinite baffled thin rectangular plate in contact with water (incompressible fluid) by both sides is considered and not as in the present work where the panel is considered unbaffled, so the slight difference in the compared results, see Table 1.

Table 1. Natural frequencies (Hz) for a simply supported rectangular steel plate (in vacuum \( f_v \) and water \( f_f \)) immersed in fluid, of side lengths \( l_x = 5m \), \( l_y = 4m \) and thickness \( t_y = 1cm \)

<table>
<thead>
<tr>
<th>( m \times n )</th>
<th>( f_v )</th>
<th>( f_f ) (Kwak)</th>
<th>( f_f ) (present method)</th>
<th>Relative discrepancy (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 , 1</td>
<td>2,498</td>
<td>0,398</td>
<td>0,435</td>
<td>8,5</td>
</tr>
<tr>
<td>1 , 2</td>
<td>5,422</td>
<td>1,093</td>
<td>1,171</td>
<td>7,1</td>
</tr>
<tr>
<td>2 , 1</td>
<td>7,067</td>
<td>1,561</td>
<td>1,623</td>
<td>3,9</td>
</tr>
<tr>
<td>2 , 2</td>
<td>9,991</td>
<td>2,468</td>
<td>2,503</td>
<td>1,4</td>
</tr>
</tbody>
</table>

The next results were obtained for the case of a simply supported composite rectangular panel submerged in air (compressible fluid), and the following properties are considered:

Side lengths \( l_x = 2m \), \( l_y = 1.5m \) , thickness \( t_y = 10mm \), material density \( \rho = 200 \frac{kg}{m^3} \), Young’s modulus \( E = 9 Gpa \), Poisson’s ratio \( v = 0.3 \), thermal expansion coefficient \( \alpha = 10^{-6} \frac{m}{K} \)

First the critical buckling temperature \( \Theta_{cr} \) is calculated with equation (2.10) in order to prevent any excess of this limit in the presented results, for the panel properties above considered, being \( \Delta \Theta_{cr} = \Theta_{cr} = 43.94 \degree C \)

Fig. 2 presents the ratio of frequency \( \gamma = \frac{f_f^m}{f_f^0} \) in vacuum of the panel as a function of the uniform temperature of the panel \( \Theta \) for different modes. The rate of decrease is higher for the fundamental mode and reaches a zero value for the temperature corresponding to the critical buckling value mentioned above.
Fig. 2. Variation of the ratio of frequency in vacuum with the temperature of the panel for different modal parameters

Fig. 3 presents the variation of frequency of the panel in vacuum \( f_v \) (Hz) and in contact with fluid (air) \( f_a \) as a function of the uniform temperature of the panel \( \Theta \) for the fundamental mode \( m=1, n=1 \). Both frequencies decrease with the temperature at an almost constant rate and this rate of decrease is higher as the temperature approaches to the corresponding buckling temperature where the frequencies reach a zero value. Furthermore, it is observed that the difference between the frequencies in air and vacuum is higher for lower temperature values.

Fig. 3. Variation of the frequency in vacuum and air with the temperature of the panel for the fundamental mode \( m = n = 1 \)

Fig. 4 presents the frequency coefficient (parameter) \( \Omega \) defined as \( \Omega = \sqrt{\frac{\rho_m l_y^2}{D}} \cdot \omega \), for the fundamental mode \( m = n = 1 \), as a function of the side length of the rectangular panel, for different values of the temperature \( \Theta \), assuming that the side length ratio \( \frac{l}{l_y} \) remains constant. This frequency coefficient decreases with the side length and the rate of decrease is higher, the
higher the values of the side length and the temperature. Although for the vacuum case, the frequency coefficient remains constant \( \Omega_{mn} \left( 0^{\circ}C \right) = \text{const} = 27.42 \), for the case \( \Theta = 0 \), as it is shown. It must be said that the frequency coefficient for the vacuum case decrease with the side length for temperatures other than \( 0^{\circ}C \).

Fig. 4. Variation of the frequency coefficient for the fundamental mode in air with the side length for different panel temperatures.

Fig. 5 presents the frequency coefficient \( \Omega \) for the fundamental mode \( m = n = 1 \), as a function of the panel aspect ratio \( \frac{y}{x} \) in the interval [0.5 – 1.3], for different values of the temperature \( \Theta \). This frequency coefficient decreases with the aspect ratio and the rate of decrease is higher, the lower the values of \( \frac{y}{x} \), but for the temperatures \( \Theta = 30^{\circ}C, 40^{\circ}C \) there is a threshold value of the aspect ratio for which there is a change in the slope of the curves and the rate of decrease of the frequency coefficient is higher the higher the values of the aspect ratio. The buckling situation corresponds to the zero value of the frequency coefficient.
Thermo-acoustic effects on the natural frequencies of vibration of an elastic rectangular panel

Fig. 5. Variation of the frequency coefficient for the fundamental mode in air with the aspect ratio for different panel temperatures.

Fig. 6 presents the frequency coefficient $\Omega$ for the fundamental mode $m = n = 1$, as a function of the fluid density in the interval [0 - 10] kg/m$^3$, for different values of the temperature. This coefficient decreases with the fluid density, but at a lower rate for higher values of the density. Although the frequency coefficient decreases with the temperature and the decrease rate is higher for lower values of the density. Furthermore, in the same figure the frequency coefficient of the panel appears in vacuum, that is for $\rho_\infty = 0$ kg/m$^3$.

![Graph showing variation of frequency coefficient with fluid density and temperature](image)

**Fig. 6.** Variation of the frequency coefficient for the fundamental mode in fluid with the density of the fluid for different panel temperatures.

In Fig. 7 the frequency coefficient $\Omega$ for the fundamental mode $m = n = 1$, is presented as function of the relative thickness $\frac{t}{h}$ of the panel, with $\frac{t}{h} = 0$ for zero thickness for different values of the temperature. It can be observed a great reduction in the frequency coefficient when the thickness tends to a small value, that in the limit, for the case $\Theta = 0^\circ C$ would correspond to the case of a diaphragm or membrane. Also in the same figure, the frequency coefficient for the panel in vacuum for the case $\Theta = 0^\circ C$ is presented, $\Omega_{\infty}(0^\circ C) = const = 27.42$. In this case the coefficient is independent of the panel thickness, therefore the corresponding curve shows the effect of the fluid on this coefficient. For the cases of temperature different from zero the frequency coefficient decrease rate is higher the smaller the value of the relative thickness due to the effects of fluid and temperature, and reaches a zero value at the corresponding buckling situation.
Figs. 8, 9, 10 and 11 presents the variation of the acoustic damping coefficient with the wave number $k$ for different temperatures of the panel. This coefficient presents a maximum at a certain value of $k$, and is calculated with the imaginary part of the fluid mass matrix, the critical damping and the frequency, see Gascón-Pérez [2018].

In Fig. 8 the variation of the damping ratio $\xi_{1}$ for the fundamental mode $1,1$ with the wave number is the same for the different temperatures, i.e. there is no variation of $\xi_{1}$ with $\Theta$. In Fig. 9 there is a similar variation of the damping ratio $\xi_{2}$ for the mode $1,2$ with the wave number for the different temperatures, but in this case $\xi_{2}$ varies decreasing with $\Theta$. In Fig. 10 the variation of the damping ratio $\xi_{1}$ for the mode $2,1$ with the temperature ($\Theta = 0^\circ C$ and $\Theta = 20^\circ C$) is higher than for the mode $1,2$ and decreasing with $\Theta$, but there is no variation with the temperature for the cases $\Theta = 20^\circ C$ and $\Theta = 40^\circ C$. Analogy in Fig. 11 the variation of the damping ratio $\xi_{2}$ for the mode $2,2$ with the temperature $\Theta$ for the values $\Theta = 0^\circ C$ and $\Theta = 20^\circ C$ is higher but there is a small variation with temperature for the values $\Theta = 20^\circ C$ and $\Theta = 40^\circ C$ and decreasing with $\Theta$ in both cases. It is observed in all the figures that there is no variation of the value of the wave number corresponding to the maximum value of the damping ratio, with the temperature.
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Fig. 8. Variation of the acoustic damping ratio for the mode 1,1 with the wave number for different panel temperatures

Fig. 9. Variation of the acoustic damping ratio for the mode 1,2 with the wave number for different panel temperatures
Fig. 10. Variation of the acoustic damping ratio for the mode 2,1 with the wave number for different panel temperatures.

Fig. 11. Variation of the acoustic damping ratio for the mode 2,2 with the wave number for different panel temperatures.

Figs. 12 and 13 presents the variation of the acoustic radiation efficiency for different modes of the panel with the wave number ratio \( \lambda = \frac{k}{k_m} \) where \( k_m \) is the modal wave number associated with the vacuum frequencies of the panel, defined as \( k_m = \sqrt{\frac{m \pi}{l_x} + \frac{n \pi}{l_y}} \). These Figures are represented in logarithmic scale and the curves presents a wavy form depending of the mode considered which is more intense the higher the modal number, and are analogous to the results obtained by the work of Wallace [1972], but not exactly the same because the case analysed by Wallace is regarding a simply supported rectangular panel in an infinite baffle with fluid on one side. The maximum value of the efficiency is reached for a wave number ratio value in the proximity of one, and the maximum value is higher the higher the values of the modal numbers considered. In this case, obviously there is no dependence with the temperature of the panel.
Thermo-acoustic effects on the natural frequencies of vibration of an elastic rectangular panel

Fig. 12. Variation of the acoustic radiation efficiency with the wave number ratio for different modes

Fig. 13. Variation of the acoustic radiation efficiency with the wave number ratio for different modes

Finally, in Table 2 the oscillation frequencies of the rectangular panel subjected to different thermal loads submerged in fluid (air) and in vacuum for the different modes are presented. It is observed the effect of the fluid and thermal loadings on the reduction of the frequency. These results are almost the same regardless considering the air compressible or incompressible.

Table 2. Natural frequencies (Hz) for the rectangular panel for the vacuum case and with air for different modes and different temperatures, with side lengths $l_x = 2m$ and $l_y = 1.5m$, and thickness $t = 1cm$

<table>
<thead>
<tr>
<th>$m$</th>
<th>$n$</th>
<th>$f_{00^\circ C}$</th>
<th>$f_{20^\circ C}$</th>
<th>$f_{40^\circ C}$</th>
<th>$f_{00^\circ C}$</th>
<th>$f_{20^\circ C}$</th>
<th>$f_{40^\circ C}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>22.14</td>
<td>17.53</td>
<td>16.71</td>
<td>12.94</td>
<td>6.63</td>
<td>5.25</td>
</tr>
<tr>
<td>1</td>
<td>2</td>
<td>46.06</td>
<td>39.23</td>
<td>40.71</td>
<td>34.67</td>
<td>34.54</td>
<td>29.42</td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>64.66</td>
<td>56.28</td>
<td>59.41</td>
<td>51.71</td>
<td>53.64</td>
<td>46.09</td>
</tr>
</tbody>
</table>
4. Conclusions

Thermo-acoustic behaviour of a rectangular panel under thermal loads and surrounded by fluid has been analysed. The pressure jump of the fluid over the panel is obtained by a BEM considering the K-H integral equation and with the panel equation of motion lead to obtain the natural frequencies of vibration. Furthermore it has been analysed the thermal buckling phenomena of the panel in vacuum and in contact with fluid.

The influence of various parameters of the rectangular panel on its thermo-acoustic response has been analyzed, in particular the variation of the frequency coefficient with the panel dimensions, geometric aspect ratio, thickness and fluid density, for different temperatures. This frequency coefficient decreases with the panel dimension, and also decreases with the aspect ratio, while the values of the frequency coefficient are lower for lower values of the relative thickness, and last this coefficient decreases with the fluid density. In all of these cases the frequency coefficient decreases with the thermal loads (temperature parameter).

Furthermore the variation of the acoustic damping ratio of the panel for different thermal loads, with respect to the wave number has been obtained as well as the acoustic radiation efficiency of the panel. The acoustic damping ratio presents a wave form variation with the wave number and remains constant with the temperature for the fundamental mode and decreases with the temperature for the rest of modes. The acoustic radiation efficiency presents a wavy form variation with respect to the wave number ratio for the different modes, with multiple peaks for higher values of the modal numbers.

Finally, results are presented for the rectangular panel submerged in air and compared regarding the vacuum case, subjected to different thermal loads.

References

Thermo-acoustic effects on the natural frequencies of vibration of an elastic rectangular panel


