Vibro-acoustic Analysis of the Brazilian Vehicle Satellite Launcher (VLS) fairing

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Abstract
During flight missions, space vehicles are subjected to a severe dynamic pressure loading when their rocket-propulsion systems are operated. This loading may be critical for the vehicle components, as well as for the payload such as satellites, which are usually very soft structures. The success of a satellite launching is determined, amongst other measures, by the satellite resistance to the fairing internal acoustic pressure.

This paper describes a numerical analysis of the dynamic response of the mechanical structure and the fairing inner acoustic cavity of the Brazilian Vehicle Satellite Launcher (VLS). Finite Element (FE) and Boundary Element (BE) methods are used for the low-frequency analysis with emphasis on the vibro-acoustic coupling effects between the fairing structural vibrations and the inner cavity acoustic pressures. The high-frequency vibro-acoustic behaviour is analyzed using a Statistical Energy Analysis (SEA) model.

1 Introduction

The Brazilian Vehicle Satellite Launcher (VLS) is a conventional launcher with four stages, launched from an earth platform [1]. Since 1980, the Brazilian government has already invested around US $ 280 million, including research and development in this project. In lift-off phase, VLS has a length of 19 meter, with a total mass of 50 tons and a thrust of 100 kN. Initial propulsion is assured by solid propellant rocket motors, in all stages, with a total amount of propellant mass of 41 tons. The VLS allows putting satellites ranging from 100 to 350 kg in circular orbits at altitudes ranging from 250 to 1000 km.

In flight missions, launch vehicles are subjected to various broadband loads. In terms of acoustic solicitation, critical instants during a rocket launching, such as lift-off, transsonic flight and maximum dynamic pressure phases, must be studied. These excitations, which have a broadband and random nature, expose the launcher’s upper parts to extreme pressure loadings. During lift-off, the jet noise generated by solid rocket boosters, is reflected by the launch platform and re-injected into the payload compartment. Such very severe acoustic levels, estimated at about 160 dB sound pressure level, can damage sensitive parts of the payload, destroying the satellite mission and wasting considerable amount of money. As such, a survey of the VLS fairing vibro-acoustic environment must be carried out, to determine its inner acoustic pressure levels. In this respect, it is very important to have reliable numerical tools that can predict the vibro-acoustic response of launch vehicle structures to acoustic loads encountered in flight and that enable noise control engineers to optimize the vehicle design [2-4].

This paper discusses the use of deterministic FE and BE vibro-acoustic techniques for low-frequency analysis (up to 150 Hz) as well as the use of Statistical Energy Analysis (SEA) for high-frequency analysis (up to 8 kHz) of the Brazilian VLS fairing.
2 Model description

2.1 VLS fairing description

Figure 1 shows the Brazilian VLS fairing structure [1]. The fairing has a hammerhead type geometry, with a maximum nominal diameter of 1.2 m and a height of 3.5 m. The fairing structure consists of aluminum shells, reinforced by beams. The exterior fairing surface is lined with cork material. No acoustic lining is provided inside the fairing cavity. The fairing has a total weight of 150 kg, including the weight of the aluminum structure, the weight of some functional components such as the electric and pyrotechnic components of the ejection system, as well as the exterior cork material.

2.2 modelling methodology

2.2.1 low-frequency techniques

In view of analysing the sound insulation properties of the fairing structure and predicting the operational fairing cavity noise levels, both the dynamic displacements of the fairing structure as well as the acoustic pressure fields at both the interior and the exterior side of the fairing should be considered. In this study, however, the fluid-structure coupling interaction between the structural displacements and the exterior acoustic pressure field is neglected. The exterior acoustic pressure is assumed to be a known external excitation for the vibro-acoustic system, consisting of the fairing structure and the internal acoustic cavity.

The FE and BE methods are the most appropriate numerical techniques for the (low-frequency) dynamic analysis of this type of vibro-acoustic systems.

FE based models for vibro-acoustic problems are most commonly described in an Eulerian formulation, in which the fluid is described by a single scalar function, usually the acoustic pressure, while the structural components are described by a displacement vector. The resulting combined FE/FE model in the unknown structural displacements and acoustic pressures at the nodes of, respectively, the structural and the acoustic FE mesh are [5],

\[
\begin{pmatrix}
K_S & K_C \\
0 & K_A
\end{pmatrix} + j \omega \begin{pmatrix}
C_S & 0 \\
0 & C_A
\end{pmatrix} \omega^2 \begin{pmatrix}
M_S & 0 \\
0 & M_A
\end{pmatrix} \begin{pmatrix}
\nu_i \\
\nu_a
\end{pmatrix} = \begin{pmatrix}
F_{Si} \\
F_{Sa}
\end{pmatrix}
\]

In comparison with coupled FE/FE models, the acoustic part in FE/BE models has a smaller size. Nevertheless, this feature does not result in a higher computational efficiency, since acoustic BE matrices are fully populated, complex and frequency dependent.

In deterministic models, the dynamic variables within each element are expressed in terms of nodal shape functions, which are usually based on low-order (polynomial) functions that are no local solutions of the governing dynamic equations. Since these low-order shape functions can only represent a restricted spatial variation, a large number of elements are needed to accurately represent the oscillatory wave nature of the dynamic response. A general rule of thumb states that at least 10 (linear) elements per

\[
\begin{pmatrix}
K_S + j \omega C_S - \omega^2 M_S \\
\frac{D}{\rho_c \omega^2} \frac{\partial^2}{\partial x^2} - \rho C_A
\end{pmatrix} \begin{pmatrix}
\nu_i \\
\nu_a
\end{pmatrix} = \begin{pmatrix}
F_{Si} \\
F_{Sa}
\end{pmatrix}
\]
wavelength are required to get reasonable prediction accuracy. Since wavelengths decrease for increasing frequency, the model sizes and the subsequent computational efforts and memory requirements increase also with frequency. As a result, the use of FE and BE models is practically restricted to low-frequency applications. In comparison with uncoupled structural or acoustic problems, this practical frequency threshold becomes significantly smaller for coupled vibro-acoustic problems, since a structural and an acoustic problem must be solved simultaneously. Moreover, as mentioned above, the matrices in a coupled deterministic model are no longer symmetrical, so that less efficient non-symmetrical solvers must be used. As a consequence, the computational load, involved with the use of coupled FE/FE and FE/BE models for real-life vibro-acoustic engineering problems, such as the considered fairing problem, becomes already prohibitively large at very low frequencies.

In order to obtain coupled vibro-acoustic response predictions within reasonable computational efforts, the dimensions of the coupled FE/FE problem (1) have to be substantially reduced. The most commonly applied technique for such a model reduction is the modal superposition technique, which expresses the unknowns of the considered system in terms of a modal base, resulting in a set of unknown modal participation factors, whose size is much smaller than the size of the original set of unknowns. The most appropriate choice for the base functions is the modes of the coupled vibro-acoustic system. Again, the determination of these coupled modes with a non-symmetric eigensolver is a very time consuming procedure, which makes it for most vibro-acoustic problems a practically impossible calculation. The most commonly used alternative is a modal expansion in terms of uncoupled structural and uncoupled acoustic modes, which result from computationally efficient symmetric eigenvalue problems. However, the fact that uncoupled acoustic modes have a zero displacement component, normal to the fluid-structure coupling interface, implies that a large number of high-order uncoupled acoustic modes is required to accurately represent the normal displacement continuity along the fluid-structure interface. Hence, the benefit of a computationally efficient construction of the modal base is significantly reduced by the smaller model size reduction, obtained with an uncoupled modal base.

In the present FE/FE study, a modal expansion in terms of uncoupled structural and uncoupled acoustic modal bases has been used. On the one hand, structural wavelengths are usually much smaller than acoustic wavelengths, so that the structural FE mesh of the fairing should be finer than the acoustic FE mesh of the inner cavity. On the other hand, due to the continuity of the normal structural and fluid displacements along the fluid-structure coupling interface, both meshes should have comparable mesh densities, at least in the region of the fluid-structure coupling interface. In view of these two considerations and in order to keep the computational efforts within reasonable limits, the following modelling methodology has been adopted. A fine FE mesh of the fairing is used for the construction of the uncoupled structural modal base. The resulting modes are then projected onto a rougher FE mesh of the fairing structure. For the acoustic cavity mesh, the same mesh density is used along the fluid-structure coupling interface as the rough mesh of the fairing structure, while the mesh density has been slightly decreased towards the central axis of the cavity. The uncoupled modes, resulting from this acoustic FE mesh, together with the projected structural modal base of the fairing structure, have then be used in a coupled FE/FE model. Note that the rough structural mesh contains only the shell area of the fairing structure, while all reinforcing beams are omitted, since it is assumed that these stiffeners have no significant effect on the fluid-structure coupling interaction, while their presence would increase the computational load of the modelling process.

For the case of the FE/BE model, the modal expansion cannot be used, since the frequency dependency of the matrix coefficients in the acoustic part prohibits a standard eigenvalue calculation. As a consequence, a semi-modal approach is adopted, in which the structural displacements are expanded in terms of an uncoupled structural modal base, while the acoustic part of the FE/BE model remains unchanged. Due to the disadvantageous computational properties of BE models, the rule of thumb of 10 (linear) elements per wavelength becomes prohibitive for the actual fairing study. Therefore, the density of the structural mesh for the coupled FE/BE model had to be relaxed, compared with the density of the structural FE mesh, used in the coupled FE/FE model. Note that, as in the FE/FE model, the uncoupled modes in the structural modal base have been obtained from the fine
fairing FE mesh and have then been projected onto the latter rough mesh.

**structural models**

The fairing has been divided in five surfaces, as shown in figure 2. The surface areas are discretized into 4-noded quadrilateral shell elements, while 2-noded beam elements are used for the circumferential and the axial stiffeners. To account for the mass loading effect of the cork blanket on the exterior fairing surface, a distribution of concentrated mass elements are attached to the fairing surface nodes. Table 1 lists the properties of both the fine and the rough structural mesh, used for the FE/FE analysis as well as the rough mesh, used for the FE/BE analysis.

![figure 2a: surfaces of the fairing](image)

![figure 2b: location of the reinforcing beams](image)

Shell surface 1 has a thickness of 3mm and is made of aluminum (E=72 GPa, \( v=0.29, \rho=2750 \) kg/m\(^3\)), while the other four surfaces are 0.8 mm thick and made of an aluminum alloy (E=72 GPa, \( v=0.29, \rho=7000 \) kg/m\(^3\)). Note that surface 1 is not shown on figure 1.

The FE fine mesh contains at least 6, 8 and 10 elements per structural wavelength in the frequency range up to, respectively, 350 Hz, 220 Hz and 150 Hz. For the FE rough mesh, these frequency limits are 200 Hz, 120 Hz and 80 Hz.

<table>
<thead>
<tr>
<th>mesh</th>
<th># shell el.</th>
<th># beam el.</th>
<th># mass el.</th>
<th># nodes</th>
</tr>
</thead>
<tbody>
<tr>
<td>FE fine</td>
<td>S1 4000</td>
<td>240</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>S2 6000</td>
<td>1080</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>S3 2000</td>
<td>360</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>S4 10000</td>
<td>1800</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>S5 12000</td>
<td>1672</td>
<td></td>
<td></td>
</tr>
<tr>
<td>total</td>
<td>34000</td>
<td>5152</td>
<td>30200</td>
<td>34200</td>
</tr>
<tr>
<td>FE rough</td>
<td>S1 2250</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>S2 3000</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>S3 750</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>S4 6000</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>S5 7500</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>total</td>
<td>19500</td>
<td></td>
<td>19650</td>
<td></td>
</tr>
<tr>
<td>FE roughest</td>
<td>S1 1000</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>S2 1500</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>S3 500</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>S4 2500</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>S5 3000</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>total</td>
<td>8500</td>
<td></td>
<td>8600</td>
<td></td>
</tr>
</tbody>
</table>

table 1: structural mesh properties

The FE roughest mesh, used in the coupled FE/BE calculations, contains at least 3, 4 and 5 elements per structural wavelength in the frequency range up to, respectively, 175 Hz, 110 Hz and 75 Hz.

The bottom edge of surface 1 is assumed to be clamped. A total of 174 structural modes in a frequency range up to 220 Hz have been identified using the fine fairing mesh. The first fairing mode, shown in figure 3, is identified at 38.6 Hz.

**acoustic models**

The FE mesh for the fairing acoustic cavity consists of 119,577 nodes and 110,238 elements (106,050 8-noded hexahedral elements and 4,188 6-noded pentahedral elements). The cavity air has a mass density \( \rho = 1.225 \) kg/m\(^3\) and a speed of sound \( c = 340 \) m/s. The bottom face of the cavity is assumed to be acoustically closed. A total of 80 acoustic modes in a frequency range up to 566 Hz have been identified using the acoustic mesh. As mentioned earlier, a proper approximation of the displacement continuity along the fluid-structure coupling interface requires that the uncoupled acoustic modal base extends up to a higher frequency range (566 Hz) compared with the structural modal base (220 Hz). Apart from the acoustic rigid body mode at 0 Hz, the first
acoustic mode, shown in figure 4, is identified at 63.5 Hz.

Figure 3: structural mode at 38.6 Hz

Figure 4: acoustic mode at 63.5 Hz

Tables 2 and 3 describe the first structural and acoustic modes, respectively.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>first bending</td>
<td>38.637</td>
</tr>
<tr>
<td>first breathing</td>
<td>77.821</td>
</tr>
<tr>
<td>second breathing</td>
<td>92.694</td>
</tr>
<tr>
<td>first longitudinal</td>
<td>108.749</td>
</tr>
<tr>
<td>first torsional</td>
<td>125.772</td>
</tr>
<tr>
<td>second bending</td>
<td>150.735</td>
</tr>
</tbody>
</table>

Table 2: structural modes

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>rigid body</td>
<td>0.000</td>
</tr>
<tr>
<td>first longitudinal</td>
<td>63.491</td>
</tr>
<tr>
<td>second longitudinal</td>
<td>112.129</td>
</tr>
</tbody>
</table>

Table 3: acoustic modes

The acoustic BE mesh is a 2-D mesh, which contains the same 8500 structural 4-noded quadrilateral elements, described in table 1, plus 1155 4-noded elements that close the top and bottom surfaces.

**model excitation**

In contrast with the aerodynamic noise during flight ascent, the nature of the lift-off acoustic pressure loading is close to a diffuse field excitation, having a (nearly) uniform pressure distribution ([2-4]). A uniform exterior pressure loading is simulated by applying a normal point force on all nodes of the fairing shell elements. The force value is defined such that the total load is equivalent to a uniform pressure loading of 160 dB.

### 2.2.2 high-frequency technique (SEA)

The computational efforts involved with deterministic techniques, such as FE and BE methods, put a severe restriction to their use for high-frequency analysis. In addition, a characteristic of high-frequency analysis is the uncertainty in modal parameters. The resonance frequencies and mode shapes show great sensitivity to small variations of geometry, construction and material properties. In light of these uncertainties, it is more appropriate for high-frequency modelling to consider a (large) population of nominally identical systems and to provide information on the ensemble-averaged dynamic response (and the associated confidence levels), as is done in Statistical Energy Analysis (SEA) [7]. In this technique, complex vibro-acoustic systems are modelled as a composition of (weakly coupled) subsystems. The dynamic system response is described in terms of the frequency- and space-averaged energy response levels for each subsystem. These subsystem energy levels result from an SEA model that expresses energy balances for the various subsystems,

\[
\begin{bmatrix}
P_1 \\
\vdots \\
P_n
\end{bmatrix} = \omega [L] \begin{bmatrix}
E_1 \\
\vdots \\
E_n
\end{bmatrix}
\]

where \( E_i (i=1..n) \) is the time-averaged energy of subsystem \( i \) in the considered frequency band with centre frequency \( \omega \) and where \( P_i (i=1..n) \) is the time-averaged input power into subsystem \( i \) in that frequency band. The coefficients in the \((nxn)\) total loss factor matrix \([L]\) are related to the internal loss factors \( \eta_i (i=1..n) \) of the subsystems and to the coupling loss factors \( \eta_{ij} (i,j=1..n, i \neq j) \) between the various subsystems.

In contrast with the element based methods, the size and subsequent computational effort of an SEA model are very small. Moreover, it is quite
fair to assume that the frequency- and space-averaged results (for complex vibro-acoustic systems) give adequate information on the ensemble-averaged results.

**SEA structural fairing model**

The VLS fairing body has been divided into four surfaces, as shown in figure 5. The SEA structural fairing model considers connected plates and beams (edge, longitudinal and circular). On the one hand, conical surfaces 2 and 4 are modelled with 16 connected trapezoidal plates. On the other hand, cylindrical surfaces 1 and 3 are modelled as 16 rectangular plates, connected to each other. Surfaces 2, 3 and 4 are modeled as rib-stiffened plates comprising circumferential stiffeners (Cbeam). Between each rib-stiffened plate, a longitudinal simple beam (Lbeam) is assigned to represent the axial stiffeners.

![figure 5: SEA fairing model](image)

Circumferential beams (Ebeam) are assigned along the edges of the various rib-stiffened surfaces. The structural model has 176 elements (16 simple plates, 48 rib-stiffened plates, 64 longitudinal beams and 48 edge beams).

Only the inertial effects of the cork lining, attached to the exterior of surfaces 2, 3 and 4, have been taken into account by defining a combined equivalent aluminium/cork density for the structural surface components.

**SEA acoustic fairing model**

The acoustic cavity of the fairing is modelled as a single 3-D acoustic volume (2.86 m³), filled with air (sound velocity 340 m/s and mass density 1.225 kg/m³).

**SEA connections**

All the 177 elements (176 structural and 1 acoustic elements) of the complete SEA fairing model are connected with line type connections (plate-plate connections, plate-Ebeam and plate-Lbeam connections) and point type connections (connections beam-beam). The connection between the plates and the acoustic volume are provided by area type connections. As such, a total amount of 342 connections provide the energy exchange between elements in the SEA model of the VLS fairing.

**excitation**

The overall pressure levels during the lift-off phase are estimated at about 160 dB. This noise has a (nearly) diffuse character. Since it is assumed that only elements with large surface areas are susceptible to this acoustic excitation, a diffuse pressure field excitation is applied to each of the plates in the SEA structural fairing model.

The complete fairing model, with 176 structural elements, 1 acoustic 3-D volume, 64 diffuse pressure field excitations and 342 connections is shown in figure 6.

![figure 6: complete SEA fairing model](image)
3 analysis results

3.1 low-frequency techniques

3.1.1 FE/FE response calculations

A modal expansion in terms of 174 uncoupled structural and 80 uncoupled acoustic modes is used for the coupled response calculations. A modal damping of 1% is assigned to each structural mode. The calculations are performed with a frequency resolution of 1 Hz. Figure 7 plots the acoustic pressure spectra in three cavity positions and a radial displacement spectrum at a fairing shell structure position for the case of a uniform exterior pressure loading. It can be seen that the low-frequency cavity pressure is dominated by the first longitudinal pressure mode around 63.5 Hz (see figure 4) and the second longitudinal mode around 112 Hz. It seems also that the coupling effects are rather small, since the resonance frequencies, observed in the coupled response, are close to the uncoupled resonances, as listed in tables 2 and 3.

3.1.2 FE/BE response calculations

A modal expansion in terms of 174 uncoupled structural modes is used for the coupled structural response calculations. As mentioned before, due to the frequency dependency of the BE model, an acoustic modal base cannot be used. A modal damping of 1% is assigned to all structural modes. All calculations are performed with a frequency resolution of 2 Hz. Figure 8 presents a comparison of the space-averaged inner cavity acoustic pressure, obtained from the FE/FE and FE/BE models.

The calculations have been performed on a HP-C3000 Unix-Workstation (400 MHz single processor, 2.5 GB memory). The uncoupled structural modal base is constructed with MSC/NASTRAN, while all other calculations are performed using LMS/SYSNOISE Rev 5.5. With respect to computational effort, the construction of the structural and acoustic modal bases took approx. 28 minutes and 9 hours and 45 minutes of CPU time, respectively. The subsequent FE/FE coupled response calculations in the frequency range from 5 Hz to 150 Hz with a resolution of 1 Hz took 1h14', while the FE/BE coupled response calculations in the frequency range from 5 to 150 Hz with 2 Hz resolution took 68h45'. Since it is assumed that the exterior acoustic pressure field is known a priori, the performance of the coupled FE/FE model for this interior vibro-acoustic problem is superior to the coupled FE/BE model.

3.2 SEA response calculations

The fairing SEA model has been solved using LMS/SEADS in third-octave bands, ranging from 5 Hz to 8 kHz. Figure 9 shows the resulting space- and frequency-averaged acoustic pressure spectrum in the fairing cavity.
3.3 analysis procedures

The interest of this study is to determine a vibro-acoustic analysis procedure of the VLS fairing in the frequency range from 5 to 8000 Hz. Deterministic element based models are applied for the analysis up to 200 Hz, while SEA is applied up to 8000 Hz. Figure 10 presents the prediction results of the fairing cavity acoustic pressure over the entire frequency range of interest, obtained from the deterministic models (space-averaged) and from the SEA model (space- and frequency-averaged).

Figures 11 compare the space- and frequency-averaged structural displacement in surface 4 and the fairing cavity acoustic pressure. Due to the violation of the basic SEA assumption of high modal density for each component, the low-frequency SEA results deviate highly from the FE results. Nevertheless, these comparisons indicate that the FE and SEA results tend to converge to the same values in the high-frequency range, but that it is advisable to try to increase the FE and BE mesh densities, such that reliable deterministic predictions can also be obtained above 200 Hz.

4 summary

This paper studies the dynamic behaviour of the fairing of the Brazilian Vehicle Satellite Launcher using both deterministic and stochastic vibro-acoustic modelling techniques. For the coupled FE/FE model, a modal expansion is used, in which the dynamic response is expressed in terms of the uncoupled modes of the fairing structure and the uncoupled acoustic cavity modes. In the coupled FE/BE model, a semi-modal approach is adopted, in which only the structural displacements are projected onto an uncoupled modal base. A uniform exterior pressure loading is applied on the fairing structure in both the FE/FE and the FE/BE models. No significant differences have been observed between the low-frequency (<150 Hz) results from these deterministic models, but the computational efficiency of the coupled FE/FE technique is significantly higher.

An SEA model has been built for the high-frequency analysis of the fairing vibro-acoustic behaviour. A comparison between the low-frequency element based results and the high-frequency SEA results indicates that there is still a
frequency twilight zone (say between 150 Hz and 300 Hz), for which the computational efforts involved with deterministic models become highly demanding, but for which the basic SEA assumption of high component modal density is not yet fully satisfied.

In the next phase, the fairing response for a real diffuse pressure excitation will be studied using coupled FE/BE models and some finer FE and BE meshes will be created in an attempt to provide reliable deterministic predictions up to higher frequencies. In addition, an extensive measurement campaign on a dedicated test-rig is planned to validate both the low- and high-frequency prediction results.

5 acknowledgements

The research work of Rogerio Pirk is financed by a grant of “Fundacao Coordenacao de Aperfeicoamento de Pessoal de Nivel Superior – CAPES”, Brazil. The research work of Bert Pluymers is financed by a scholarship of the Institute for the Promotion of Innovation by Science and Technology in Flanders (IWT). Wim Desmet is a Postdoctoral Fellow of the Fund for Scientific Research – Flanders (Belgium) (F.W.O.).

6 references
