DESIGN PROCEDURE FOR A COMPOSITE RUDDER

Troise, Rodrigo Tadeu Aparecido Malange
EMBRAER – PC180 Av. Brigadeiro Faria Lima, 2.170. CEP12227-901. São José dos Campos – SP - Brazil
rodrigo.troise@embraer.com.br

Almeida, Sérgio Frascino Muller de
ITA – Department of Mechanical Engineering. CEP12228-900. São José dos Campos – SP - Brazil
frascino@mec.ita.br

Riscado, Marcos Serafim
EMBRAER – PC180 Av. Brigadeiro Faria Lima, 2.170. CEP12227-901. São José dos Campos – SP - Brazil

Abstract. *The present work presents a design procedure for the structure, joints and actuator system of a composite material rudder of a regional jet. The main structural parts and supporting mechanism of the rudder are modeled as a multi body system. Several analyses are performed assuming the parts as rigid bodies. The preliminary analyses are important for the preliminary sizing of the actuator and joints. This simplified model serves as the basis for the development of a model including structural flexibility. This model accurately describes the mechanical behavior of the system and is used to verify the design.*

Keywords*: Mechanism, Structural Analysis, Composite Material, Rudder*

1. Introduction

Due to the strong competition currently observed in the aeronautical industry, the development and use of new technologies focusing on reducing production and operational costs are vital to increase the profitability and to reduce the final costs for the operators. Current requirements of the airlines obligate the manufacturers to produce aircrafts that are able to transport a large pay load with a longer range and spending less fuel. In order to reach this goal, composite materials can be considered as a tool to obtain this performance because they are lighter and stronger, comparing with the traditional materials used in aircraft manufacturing (Anonymous, 2003).

These materials are composed by a combination in a macroscopic scale of two or more materials in which can offer a superior performance. In fiber polymeric matrix composites, the resin is responsible for transferring the load to the fibers and to absorb impact. It is important to mention that the composite performance depends on the orientation of the fibers that defines the stiffness and strength of the component. The lay-up sequence and the fiber orientation are responsible for the anisotropy and possible mechanical couplings of the laminate (Daniel and Ishai, 2006).

Joining a component manufactured with composite material with a metallic part tends to be very complex. Special attention should be given when the link is between movable components, due the manufacturing and thermal expansion tolerances corresponding to each process. Also, the aeronautical agencies demand the use of fail safe design methodologies in order to guarantee the existence of redundant parts for, in the case of failure of any link between the parts, the remaining structure can support all the design loads making the aircraft safer to the passengers and crew members.

As a manner to improve the production and the efficiency of the aeronautical components, the present work is focused on the design and analysis of a composite rudder with emphasis on the design of its hinges and selection of the actuators. The main structural parts and supporting mechanism of the rudder are modeled as a multi body system and several analyses are performed assuming the parts as rigid bodies. A preliminary analysis is performed, demonstrating its importance to the preliminary sizing of the actuator and joints. This simplified model serves as the basis for the development of a model including structural flexibility. This model accurately describes the mechanical behavior of the system and is used to verify the design.

Also, as cost is important for aircraft manufacturing competitiveness, a cost estimate of the complete rudder will be made.

2. Hinges and rudder idealization model

The configuration of the hinges is presented in Fig. 1. Four hinges (from A1 to A4) are used to transfer all the loads applied to the rudder to the vertical empennage and the actuators. Hinges A2 and A3 are positioned close to the actuator hinges (AS and AI, respectively), to minimize bending and torsion effects on the spar. This configuration is based on FAR 25 regulation (FAR25, 1999) which establishes the necessity of having a fail safe structure supporting any failure (losing one hinge or one actuator), allowing a safe return of the aircraft to its base even when failure is detected in one hinge or one actuator.

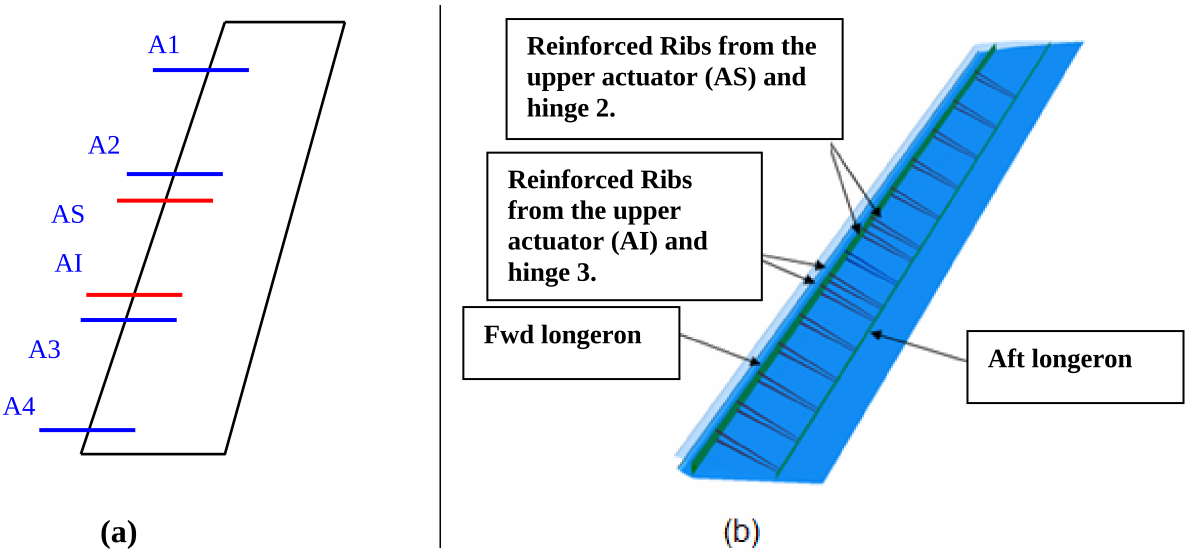


Figure 1. Schematic position of the hinges (a) and ribs position (b).

The basic rudder structure is composed by two spars and a certain number of ribs. The rudder structural analyses should demonstrate the capacity of the design to maintain the structure integrity or the necessity of adding reinforcers. If ribs are not used, a one shot cure process can be used for manufacturing the rudder minimizing the need for using bolted joints. This procedure tends to decrease the manufacturing time and produces parts with good quality in comparison with other well established processes. However, this work will focus only on the dimensioning of the hinges and actuators.

After the first modeling, which aims at studying the initial position of the hinges at both the vertical empennage and the rudder, special attention must be given to the actuator position in the rudder.

2.1. Loading estimation

The pressure distribution on the rudder surface will be assumed to correspond to a bi-linear form along the chordwise and lengthwise directions from, according to Fig. 2.

Interpolating $W_{\max}(z)$ based on W_1 and W_2, yields:

$$W_{\max}(z) = W_1 \cdot \frac{z}{b_f} + W_2 \cdot \left(1 - \frac{z}{b_f}\right) \tag{1}$$

According to Fig. 2, W varies with x from 0 to $W_{max}(z)$. Therefore, multiplying Eq. 1 by a value proportional to the x position, results:

$$W(x,z) = W_{\max}(z)\left(\frac{x - x_{\inf}}{x_{\max} - x_{\inf}}\right) \tag{2}$$

where:

$$x_{\max} = c_r - \frac{z}{tan(\beta)}; \quad x_{\inf} = -\frac{z}{tan(\gamma)}$$

So:

$$W(x,z)=\left(W_1\frac{z}{b_f}+W_2\left(1-\frac{z}{b_f}\right)\right)\left(\frac{x+\dfrac{z}{tan(\gamma)}}{c_r-\dfrac{z}{tan(\beta)}+\dfrac{z}{tan(\gamma)}}\right)$$

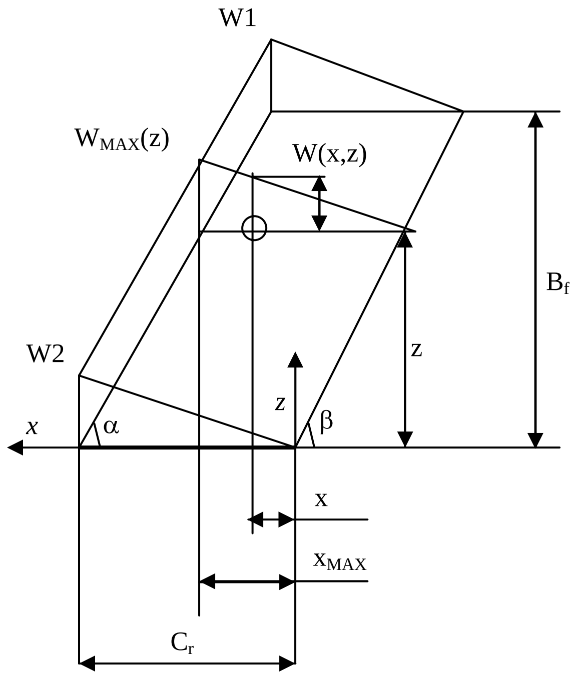


Figure 2. Idealized distribution of load at the rudder.

The total force, F, at the center of the rudder is:

$$F=\int_0^{b_f}dz\int_{-\frac{z}{tan(\gamma)}}^{c_r-\frac{z}{tan(\beta)}}W(x,z)dx \tag{3}$$

The values obtained using Eq. 3 will provide an equation having W_1 and W_2 as unknown variables. In order to obtain the second equation, the sum of the moments at the chord is done:

$$M_F=F\ Z_{CG}=\int_0^{b_f}z\ dz\int_{-\frac{z}{tan(\gamma)}}^{c_r-\frac{z}{tan(\beta)}}W(x,z)dx \tag{4}$$

The solution of Eqs 3 and 4 yields the values of W_1 and W_2.

2.2. Preliminary design of the actuators

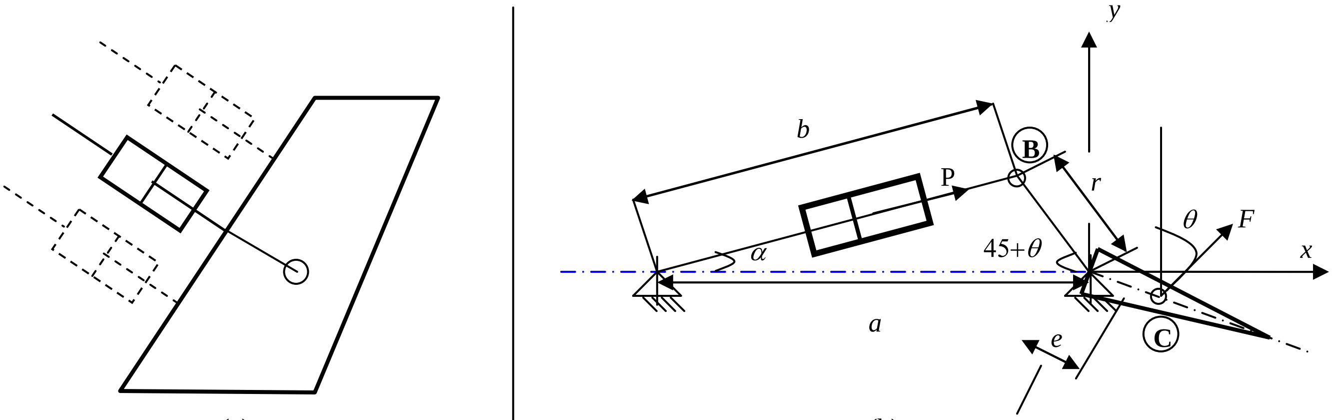
In this section the rudder will be analyzed as a rigid body, using the principle of virtual work (Megson, 2001). This will be extremely useful in order to obtain the necessary data for the preliminary sizing of the actuators. The process consists in replacing the actuators by a single one (because of the rigid body assumption), aligned with the pressure center of the rudder as illustrated in Fig. 3. This figure also describes the geometrical variables used in the analyses.

Using the principle of the virtual displacements, $\delta V = 0$ with $\delta W = Q_\theta\ \delta_\theta$, results:

$$Q_\theta = \vec{P}\,\frac{\partial \vec{r}_b}{\partial \theta} + \vec{F}\,\frac{\partial \vec{r}_c}{\partial \theta} \tag{5}$$

Solving Eq. 5, results:

$$P = \frac{e\,F}{r\,(\cos(\alpha)\,sen(45^{o}+\theta) + sen(\alpha)\,\cos(45^{o}+\theta))} \tag{6}$$



(a) (b)

Figure 3. (a) Idealized single actuator aligned with the pressure center and (b) variables used in the formulation.

Equation 6 yields the actuator force to counteract the resultant load F. This value can be used for the preliminary sizing of the actuators and its hinges saving time in the development stage. In order to validate the selection of the actuator, the pressure at the hydraulic line (210 bar) and the nominal diameter of the actuator (50 mm) were adopted to use the verification tables provided by the actuator manufacturer(Rexroth, 2003).

Table 1. Actuator dimensions with respect to the nominal diameter.

AL Ø (mm)	*ØCX* (mm)	*EP* h15 (mm)	*EX* (mm)	*LT* min (mm)	*MS* max (mm)	*XC* +/-1.25 (mm)	*XO* +/-1.25 (mm)
25	12 -0.008	8	10 -0.12	16	20	127	130
32	16 -0.008	11	14 -0.12	20	22.5	147	148
40	20 -0.008	13	16 -0.12	25	29	172	178
50	25 -0.008	17	20 -0.12	31	33	191	190
63	30 -0.008	19	22 -0.12	38	40	200	206

where the symbols in the table are described in Fig. 4.

Table 2, obtained using Fig. 3, the adopted pressure and actuator nominal diameter (*AL*) together with Eq. 6, presents the result of the first step of the actuator sizing, showing a comparison between the necessary internal area of the actuator to push (A_{1R}) and pull (A_{3R}) with the internal area of the actuator to push (A_1) and pull (A_3), respectively.

As a second step, an actuator with nominal diameter equal to 63 mm will be adopted as presented in the Table 3.

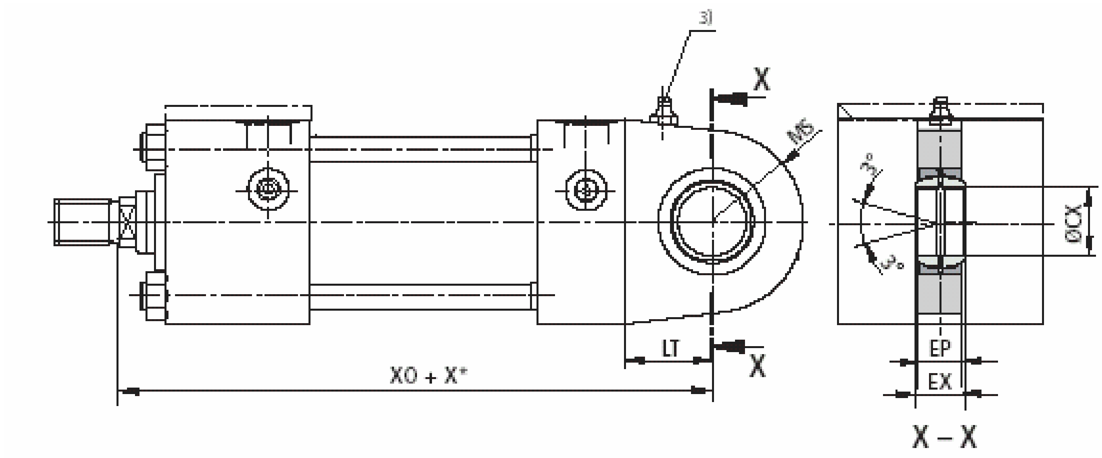


Figure 4. Actuator dimensions

Table 2. First step of the actuator sizing.

θ (º)	β (º)	*a* (mm)	*r* (mm)	*e* (mm)	*b* (mm)	α (rad)	*P* (total)(N)
15	56.502	313.11	50	232.41	280.0	0.093	45253
-15	56.502	313.11	50	232.41	280.0	0.150	28099
A_1 (cm^2)	12.56						
A_3 (cm^2)	10.02						
A_{1r} (cm^2)	13.77	not OK					
A_{3r} (cm^2)	22.17	not OK					

Table 3. Second step of the actuator sizing.

θ (º)	β (º)	*a* (mm)	*r* (mm)	*e* (mm)	*b* (mm)	α (rad)	*P* (total)(N)
15	56.502	313.11	50	232.41	280.0	0.093	45253
-15	56.502	313.11	50	232.41	280.0	0.150	28099
A_1 (cm^2)	31.17						
A_3 (cm^2)	25.01						
A_{1r} (cm^2)	13.77	OK					
A_{3r} (cm^2)	22.17	OK					

2.3. Initial finite element modeling of the rudder

The initial finite element model will enable the sizing of the laminate using as sizing criteria the strength analysis (Tsai-Wu failure index) and buckling, considering some possible failure condition cases, as detailed in Table 4.

These failure cases impose the necessity of the structure of being fail safe and support the design loads without buckle and catastrophic failure. These models were constructed with Nastran® (MSC, 2000) basically with CQUAD4 elements. As a start point, the hinges were modeled as rigid body elements (MPC) enabling the fast sizing of the rudder laminates before defining the hinge shape. At the verification phase of the development, the hinges were modeled using quadratic tetrahedron elements (Etrekin) having the same loads found at the first step. Figure 5 depicts the finite element model of the rudder.

In order to verify the preliminary sizing of the actuators and hinges, a comparison between the rigid body model, FEM model with hinges as rigid elements (MPC) and FEM model with hinges modeled with solid elements is presented. Fig. 6 depicts the finite element model of the hinges.

Table 4. Possible failure conditions.

Failure condition	Description
EO	Both hinges and actuators intact
AIF	Failure of the lower actuator
ASF	Failure of the upper actuator
2AFD01	Failure of the hinge number 01
2AFD02	Failure of the hinge number 02
2AFD03	Failure of the hinge number 03
2AFD04	Failure of the hinge number 04

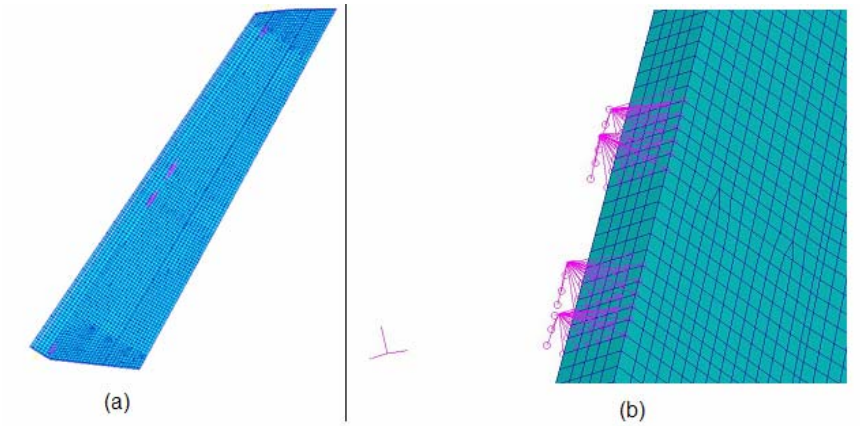


Figure 5. (a) FEM model of the rudder and (b) hinges modeled as rigid bodies (MPC).

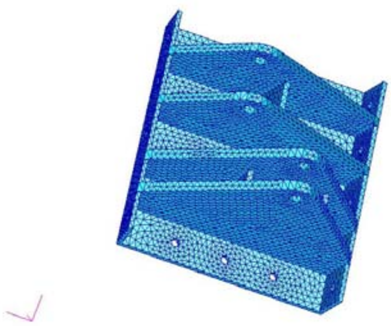


Figure 6. Hinge model.

Table 5 presents the material properties of the carbon/epoxy layers. The material properties of the aluminum are: (1) modulus of elasticity E = 71 GPa, (2) Poisson ratio ν = 0.33 and (3) yield strength F = 45.5 MPa.

Figure 7 presents the failure index analysis based on the Tsai-Wu failure criterion (Daniel and Ishai, 2006); Fig. 8 shows a buckling analysis for one load case. The structural analyses for all load cases demonstrate that the proposed structural design satisfies both the failure and buckling requirements. Figure 7 shows that the maximum failure index is 0.35, that is, from the fracture point of view the structure is oversized. On the other hand, Fig. 8 demonstrates that buckling requirement is dominant in the structural design. It must be emphasized that the emphasis of this work is the sizing of the actuators; structural optimization of the rudder is out of the scope of this work.

Table 5. Carbon/epoxy material properties (Daniel and Ishai, 2006).

Property	Value
Longitudinal modulus of elasticity, E_{11}	77 GPa
Transverse modulus of elasticity, E_{22}	75 GPa
In plane shear modulus , G_{12}	6.5 GPa
In plane Poisson ratio, ν_{12}	0.06
Longitudinal tension strength, F_{1t}	963 MPa
Transverse tension strength, F_{2t}	856 MPa
Longitudinal compression strength, F_{1c}	900 MPa
Transverse compression strength, F_{2c}	900 MPa
In plane shear strength, F_6	71 MPa

3. Results

Initially, an estimation of the sum forces for the actuators will be performed considering the rudder and hinges as rigid bodies. Table 6 presents a comparison between the three finite element models.

For the design of the hinges, a similar procedure to the one presented for the actuators design was followed based on the results of the rudder modeled with rigid hinges. Table 7 summarizes the comparison between the resulting forces at the hinges using as reference the FEM model with rigid hinges (MPC).

The analysis of the results will be presented at the next section.

Table 6. Comparison of loads at the actuators.

Model	Force at the actuators (N)	Difference with respect to the rigid model (%)
Rudder as rigid body	67548	-
Rudder as flexible body, hinges as rigid bodies (case EO+15º)	66305	1.84
Rudder and hinges as flexible bodies (case EO+15º)	66065	2.20

Table 7. Comparison of loads at the hinges.

Case	Actuator force (N) Case EO+15º Rudder: flexible Hinges: rigid	Actuator force (N) Case EO+15º Rudder: flexible Hinges: flexible	Difference (%) with respect to: Rudder: flexible Hinges: rigid
A1	1611	1863	13.52
A2	31724	32843	3.41
A3	34236	31481	-8.75
A4	1756	2016	12.89
AS	31683	33265	4.76
AI	34622	32800	-5.55

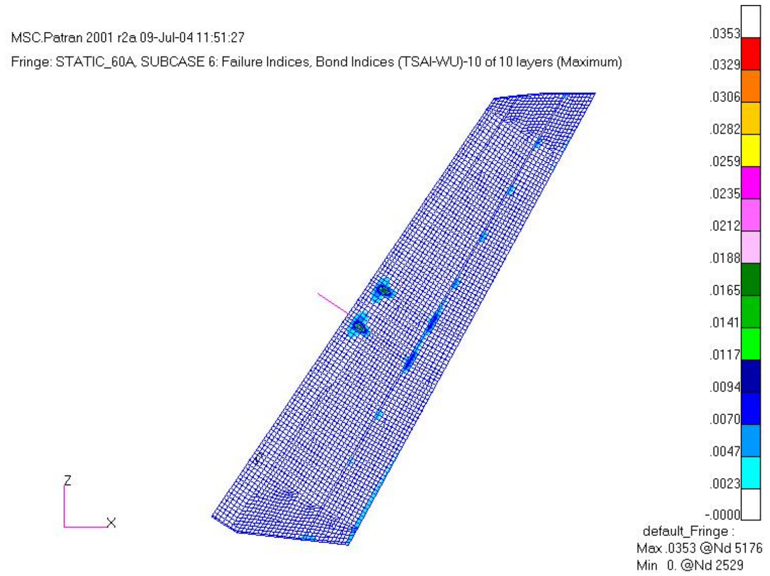


Figure 7. Tsai-Wu failure index results.

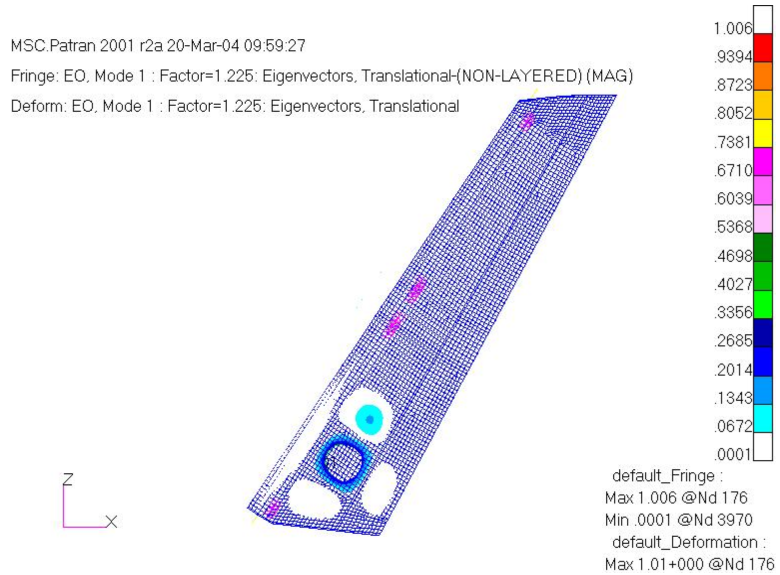


Figure 8. Buckling results using Lanczos method (Sundar, 2000).

4. Cost estimation

A cost estimation analysis is very difficult to be done due to the necessity to obtain accurate detailing of the costs involved in manufacturing and assembly process, inspection methodology, logistic and stock of raw material, production rate and other factors not included in this work. Therefore, based on the available data, a preliminary analysis of the costs involved including raw material, tooling and manufacturing process was done. All costs were obtained by means of specialized books (Daniel and Ishai, 2006) or values were adopted, only as reference, following the proposal to estimate the actual costs involved for a new development. The cost estimation, for the ruder structure, was based on the entire weight of the rudder multiplied by the unit weight cost of each prepreg. Table 8 presents the cost estimation for the raw material.

For the cost estimation of the tooling, an estimative was done of the amount of hours, necessary to the design and fabrication of the tooling. The cost per hour was adopted as U$ 50.00 for design (engineering) and U$ 25.00, for fabrication (operator). The required equipment for manufacture the parts were also included in these values.

The materials to be used in the tooling production were not included in the cost analysis since in this work the type and material of the tooling were not defined. Also, the cost of using a temperature control system necessary primarily for the use of the one shot process were not evaluated either because of the difficulty to obtain a homogenous temperature distribution between each mandrel during the curing in an autoclave.

Table 9 presents the estimated cost for the design and fabrication of the tooling, the rudder and the total cost of the entire process. The manufacturing cost to produce the rudder in serial production was estimated for each component, the amount of hours required for the task of cutting of the raw material, laminating, cure, inspection and assembly.

Table 8. Cost estimate of the raw material.

MATERIAL	UNIT	UNITARY COST	Quantity	Cost(US$)
Carbon fiber/Epoxy resin (pre-preg)	kg	40 US$	100 kg	4000.00
Rivets (Hi-Lite)	Piece	5 US$	100 PC	500.00
Hinges (Al7475-T7351)	Piece	250 US$	4 PC	1000.00
Metallic bonding plate	kg	20 US$	20 kg	400.00
TOTAL				5900.00

Table 9. Cost estimate of total cost.

<table>
<tr><th></th><th colspan="3">Number of hours</th><th colspan="3">Cost(U$)</th></tr>
<tr><th>COMPONENT</th><th>Design (Engineering)</th><th>Fabrication</th><th>Assembly</th><th>Design (Engineering)</th><th>Fabrication</th><th>Assembly</th></tr>
<tr><td>Rudder</td><td>800</td><td>200</td><td rowspan="2">20</td><td>40,000.00</td><td>5,000.00</td><td rowspan="2">500.00</td></tr>
<tr><td>Fittings</td><td>400</td><td>400</td><td>20,000.00</td><td>10,000.00</td></tr>
<tr><td>TOTAL(U$)</td><td>75,500.00</td><td colspan="5"></td></tr>
<tr><td colspan="4">TOTAL COST OF THE ENTIRE PROCESS(US$)</td><td>81,400.00</td><td colspan="2"></td></tr>
</table>

5. Conclusion and comments

The focus of this work is the design and analysis of a composite rudder and its hinges and actuators. The preliminary definitions of the rudder and hinge models were based on books and professionals with large aeronautical experience in preliminary design. The design consisted basically in detailing the rudder, actuators and hinges specifications. Each one of these elements has strong influence of at the system performance as a whole. Therefore, the design methodology for the system is intrinsically interactive. The preliminary actuator specification, based on a preliminary consideration of the rudder as a rigid body, enabled a realistic estimative of the reactions at the hinges, also considered as rigid. For the study of the rudder, initially the hinges were modeled as rigid bodies (MPC elements at Nastran®). This enabled the development of the hinges in parallel.

A final evaluation was made, based on comparing the reactions at the boundary of the model, with hinges modeled as rigid bodies and as flexible bodies. The analyses demonstrated a difference of approximately 8% at the main hinges (02, 03, AS e AI) and 15% at the secondary hinges (01 e 04). The values found at the secondary hinges do not invalidate the process because these hinges have to support lower loads in comparison with the main ones. Typically, they are considered oversized, because the optimized values would invalidate their fabrication.

It is important to remark that the load used for the actuators sizing presents an error of approximately 2% when compared to the flexible models. This allows the application of simultaneous engineering at the design of the rudder and its actuators and hinges. Another aspect is related to the cost evaluation in which the primary idea was to summarize the relevant components (raw material, manufacturing, engineering) under a qualitative point of view, demonstrating the necessity of consider all those items, together with others not considered, as the amount of pieces produced per month, manufacturing training, and others.

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