Imperial College London

AE3-406 Airframe Design

Coursework Report

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1. Question 1

We aim in this problem to design a canard for a combat aircraft. The bearings are located on the rotation axis of the canard and must handle the load induced by the lift. Since the lift load line is located on the same axis as the rotation axis of the canard, we can resume the problem to a two dimension one, as shown on the figure 1.1, s being the total length of the canard, L the distance between the two bearings, F_1 and F_2 the load taken by the bearings, F_L the total lift force on the wing.

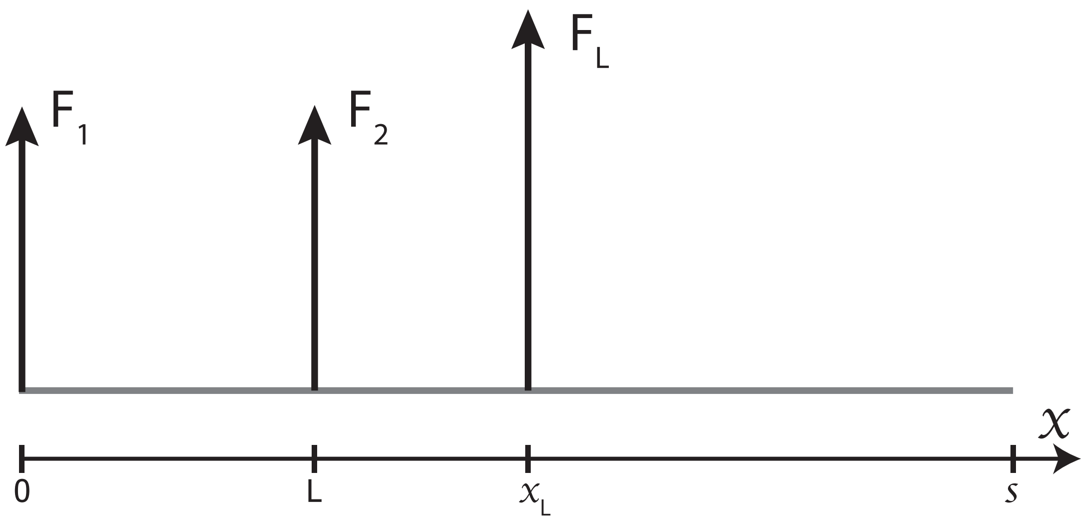

Figure 1.1: Parameters of the problem

We take the wing as the physical system to study. Three loads are applying on it : the two loads coming from the bearings and the load coming from the lift. The trapezoidal lift distribution is supposed to contain the weight of the wing. We also ignore the drive couple on the system.

The canard must be still if the bearings are correctly sized to handle the lift load. The sectional lift factor l is such that $\int_0^s l(-\frac{2}{3}\frac{x}{s}+1)dx = F_L$, which gives $l = \frac{3F_L}{2s}$. The static equilibrium and the momentum equilibrium at $x = 0$ using the lift distribution and writing $x_L = \frac{5}{12}s$ allows us to get:

$$\begin{cases} F_1 + F_2 + F_L = 0 \\ LF_2 + \int_0^s lx(-\frac{2}{3}\frac{x}{s}+1)dx = LF_2 + l\frac{5}{18}s^2 = 0 \end{cases}$$

$$\begin{cases} F_1 = (\frac{x_L}{L} - 1)F_L \\ F_2 = -\frac{x_L}{L}F_L \end{cases}$$

Hence, since we have the mass of the bearings given by $M_b = 0.05Fkg$ with F the loading force on the bearing in kN and $M_s = (5 + 20L)kg$, we can minimize the weight of the spigot and the bearings, not forgetting the absolute values and assuming $L \leq x_L$:

$$\frac{d(M_s + M_{b1} + M_{b2})}{dL} = 0$$

$$L_{optimum} = \sqrt{0.05F_L\frac{2x_L}{20}}$$

This allows us to get the optimal spacing between the bearings and the mass of the bearings and the spigot. We have as expected $L \leq x_L$.

Weight bearing 1 M_{b1}	Weight spigot M_s	Weight bearing 2 M_{b2}	Distance L
2.49 kg	14.8 kg	4.9 kg	0.49 m

Table 1.1: Design results for the bearings of the Canard

While this design process was really simple, one could argue that it is too simple to provide a good optimization of the weight of the bearings and the spigot:

- The lift distribution of the canard changes with the incidence of the flow. While we optimized the bearings to handle the wing at zero angle incidence, the lift is greater when the wing is tilted, and the impact on the bearing loads should be assessed during the design process.
- We assumed that the lift distribution contained the weight distribution of the wing. Still, the sizing process and the final weight given to the spigot and the bearings should impact the lift distribution. An iteration of the design process could be useful here.
- We neglected the way internal loads are spread during the sizing process. This simplifies greatly the design process, but for a combat aircraft, one may want to study extensively the impact of all forces and dynamic loads on the structure to optimize the spacing and the mass of the bearings and spigot using CAD software.

2. Question 2

2.1 Introduction to the problem

In this part we study a section of fuselage of a small tourism aircraft. In order to design this section, assumed without frame, we have been given three loads applied on the tail structure:

- A constant linear distribution of $8kN$ on the fuselage part accounting for all the weight in the structure.
- A vertical tail load of $20kn$, unevenly distributed on the two sides of the tail wing, and which can be either positive or negative.
- A load of $5kn$ on the vertical stabilizer than can be either positive or negative as well.

This gives us 8 cases of loading on the tail of the plane. The moment of those forces on the three axis can be expressed at the center of the X-X section O. This section is assumed to be circular. We neglect the pressurization of the cabin since it is a small tourism aircraft. We will also neglect the in-plane load created by the weight of the structure on itself as we are studying a 2D section.

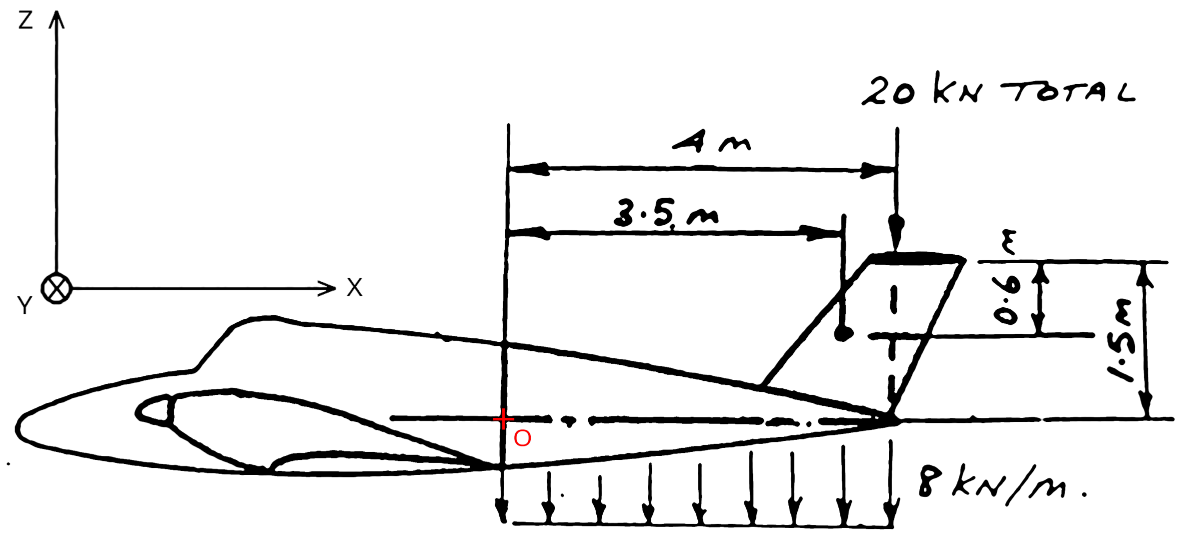

Figure 2.1: Parameters of the problem

The moment $\vec{M}$ of a load applied at a point P to the center of the section X-X can be expressed by:

$$\vec{M} = \vec{OP} \wedge \vec{F}$$

Using the distributions, loads and distances given on figure 2.1 we can thus get the sum of the moments at point O. We then use the following notations: $M_x = (\sum \vec{M}).\vec{e_x}$, $M_y = (\sum \vec{M}).\vec{e_y}$, $M_z = (\sum \vec{M}).\vec{e_z}$. The value obtained are summarized in the following table, with the sign of the load given by the first three columns.

Tail load location	15kn tail load location	Gust load location	M_x	M_y	M_z
+	+	+	5.5kNm	-144kNm	17.5kNm
+	+	-	14.5kNm	-144kNm	-17.5kNm
+	-	+	-14.5kNm	-144kNm	17.5kNm
+	-	-	-5.5kNm	-144kNm	-17.5kNm
-	+	+	-14.5kNm	16kNm	17.5kNm
-	+	-	-5.5kNm	16kNm	-17.5kNm
-	-	+	5.5kNm	16kNm	17.5kNm
-	-	-	14.5kNm	16kNm	-17.5kNm

Table 2.1: Moments created by the different load cases

The worst case is when the bending moment created by M_y and M_z on the section is the greatest, and when the twist created by M_x is the greatest. The values of the moments taken for the simulation are thus:

M_x	M_y	M_z
14.5kNm	-144kNm	-17.5kNm

Table 2.2: Moments corresponding to the worst case

The section will consist of a skin, assumed to take all the shear load generated by M_x, and of stringers that will stiffen the skin and are assumed to take all the compression loads generated by M_y and M_z.

Next is explained the strategy used to design the fuselage section from this moment given on the section center. It is to be noted that the code developed allows to play on the skin thickness, the number of stringers and the way they are distributed over the fuselage in order to find an optimal solution for this problem.

2.2 Load taken by the stringers

We assume we have N stringers evenly distributed over the fuselage to stiffen the skin. The stringers are noted from 1 to N, and the first one has an angle of α_{offset} with the z axis. The modelization can be seen on figure 2.2.

We choose classic Z stringers such that the stringers have an area $A_s = tb$, a thickness equivalent to the thickness of the skin $t_s \approx 1.25t$, and are such that $d/h = 0.4$. The appearance of the stringer is shown on figure 2.3.

We want to find the load σ_x taken by the stringers. This load is given by the classic Engineer's Theory of Bending equation, assuming the structure behaves linearly with small displacements:

$$\sigma_x = \frac{M_y I_{zz} - M_z I_{yz}}{I_{yy} I_{zz} - {I_{yz}}^2} z - \frac{M_z I_{yy} - M_y I_{yz}}{I_{yy} I_{zz} - {I_{yz}}^2} y$$

Accounting for the symmetry of the section we have $I_{xy} = 0$ and thus this equation simplifies to:

$$\sigma_x = \frac{M_y}{I_{yy}} z - \frac{M_z}{I_{zz}} y \tag{2.1}$$

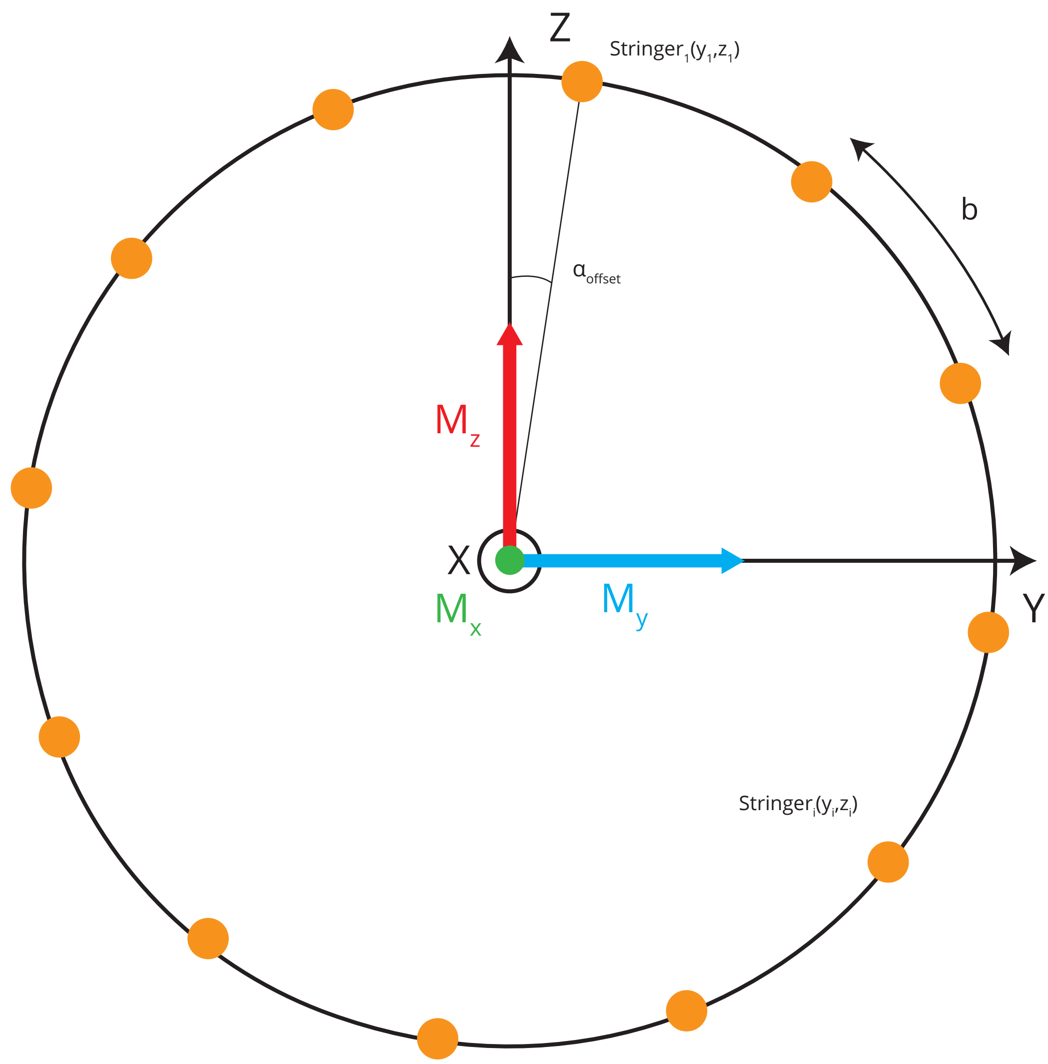

Figure 2.2: Parameters of the 2D section fuselage

In order to find the second moments of inertia, we need to find the distribution of matter in the fuselage respective to the neutral axis y and z. We will use an equivalent area for the stringers and the stiffened panel that we place at the position of stringers in order to make the calculus easy.

We consider that all the panels are assumed flat and ideal, which is not a too constraining assumption considering the number of panels and their relatively low curvature. Each of this panel, of thickness t and effective length $p = b - 30 * t_s$, supports a $\sigma_x(y, z)$ stress distribution. The equivalent boom to such panel is shown figure 3.2.
The boom areas B_1 and B_2 are given by:

$$\begin{cases} B_1 = \frac{tb}{6}(2 + \frac{\sigma_2}{\sigma_1}) \\ B_2 = \frac{tb}{6}(2 + \frac{\sigma_1}{\sigma_2}) \end{cases}$$

The total equivalent area for the stringer i, accounting for the equivalent boom area located at the same spot from the panels i and i-1, is thus:

$$A_i = A_s + \frac{tb}{6}(2 + \frac{\sigma_x^{i+1}}{\sigma_x^i}) + \frac{tb}{6}(2 + \frac{\sigma_x^i}{\sigma_x^{i-1}})$$

Which, using equation 2.1 and the fact that $I_{yy} = I_{zz}$ from the symmetry of the section,

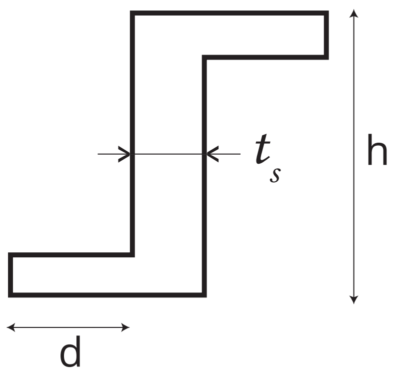

Figure 2.3: Dimensions of the Z-stringers

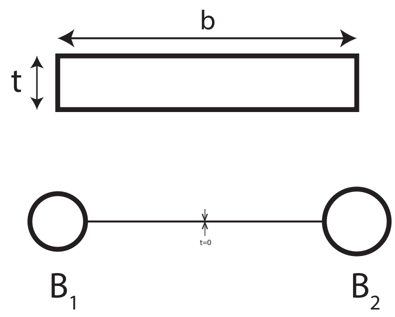

Figure 2.4: Equivalent boom to a panel

simplifies to:

$$A_i = A_s + \frac{tb}{6}(2 + \frac{M_y z_{i+1} - M_z y_{i+1}}{M_y z_i - M_z y_i}) + \frac{tb}{6}(2 + \frac{M_y z_i - M_z y_i}{M_y z_{i-1} - M_z y_{i-1}})$$

We then have $I_{yy} = I_{zz} = \sum A_i z_i^2$. We can finally obtain the load seen by the stringers with the equation 2.1.

2.3 Load taken by the skin

The skin is uniformally twisted by the moment M_x. We have a shear flow q_{twist} in the skin linked to the moment M_x by $q_{twist} = \frac{M_x}{2A_{total}}$, with $A_{total} = \pi(\frac{D}{2})^2$.

We finally have a shear on the skin of $\tau = \frac{q_{twist}}{t}$

2.4 Failure of the fuselage section

Having the value of the loads carried by both the stringers and the skin, we must now check if the structure can handle the different loading cases of the aircraft. The failure will depend on the Young modulus of the materials chosen for the structure.

	Skin	Stringer
Alloy	Aluminum 2024-T6	Aluminum 7075-T6
E	72.4 GPa	71.7 GPa

Table 2.3: Materials chosen for the skin and the stringers

We assume the stringer-skin panel fails at initial buckling, before it yields. The correspondent failure stress for a long plate under compression is $\sigma_{x,cr} = KE_{stringer}(\frac{t}{b})^2$, with K a correction factor for the presence of stringers stiffening the skin. The relevant correction factor $K = 5.2$ is chosen using an ESDU datasheet (see appendix A.1) for the geometry of the stringer used. We can then define the compressive stress ratio $R_c = \frac{\sigma_x}{\sigma_{x,cr}}$.

For a long flat panel in shear, the critical buckling shear stress can be expressed by $\tau_{cr} = 4.86E_{skin}(\frac{T}{p})^2$, with p the effective with of the panels and $T = t + \frac{A_s}{p}$ the effective thickness. We can then define the shear stress ratio $R_s = \frac{\tau}{\tau_{cr}}$.

The fuselage will thus fail if we have a panel-stringer system that fails. The condition of failure of a panel-stringer system is:

$$R_s{}^2 + R_c > 0.99$$

This is the failure criterion used in the matlab code to check if the fuselage section holds the worst load case.

2.5 Final design of the fuselage section

A matlab program as been developed to help design the section. The entry parameters are:

- The number of stringers
- The thickness of the skin
- The offset angle of the first stringer α_{offset} (not used but that could have help enhance the design)

We can compute, for different number of strings and thickness of skin, the failure or not of the structure using the criterion developed earlier. The results are shown figure 2.5. We can see that for a skin too thin or for a number of stringers too low the structure fails. From this we know we aim to find an optimal combination of those two values.

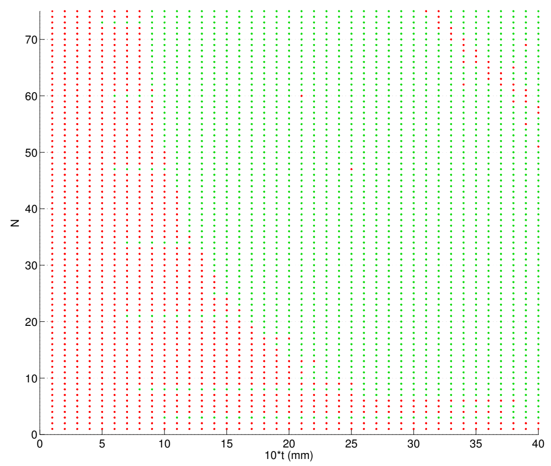

Figure 2.5: Failure (red) or not (green) of the structure

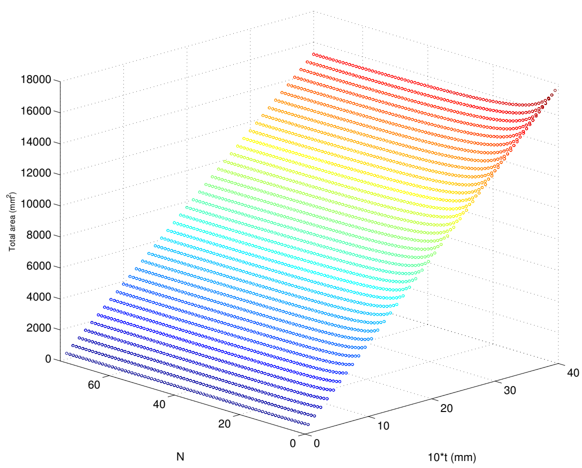

Figure 2.6: Total area of the structure

The area of the structure is linked to its weights. To a certain extent we seek to minimize it. The results are shown on the figure 2.6. We can see that the area depends greatly on the thickness of the skin. We thus seek a skin that is not too thick, but not too small such it does not fails and is also craftable. The number of stringers does not have a great impact on the area. We thus choose to take a reasonable number of stringers of $N = 36$. This number allows to get lower values of the skin thickness as can be seen on figure 2.5.

Finally, we plot for this chosen number of stringers a thickness of skin using the corresponding curve from the failure criterion. With this figure 2.7, we finally choose a thickness of $t = 1.5mm$, in order to leave a certain margin to the failure criterion.

We can sum up the final design in the table 2.4. A rendering of the section can be found on figure 2.8.

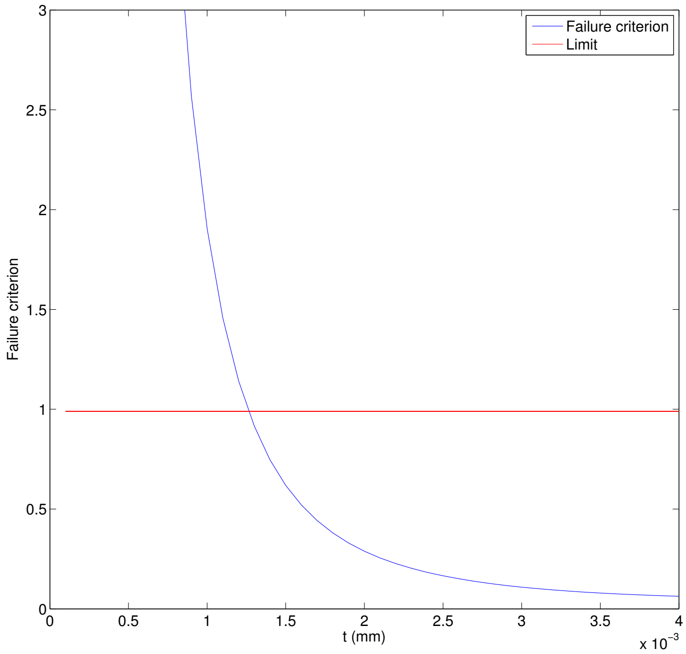

Figure 2.7: Failure criterion vs thickness

A_s	d	h	b	t_s	N	t
150 mm^2	1.8mm	4.65mm	10.8cm	1.87mm	36	1.5mm

Table 2.4: Structure design summary

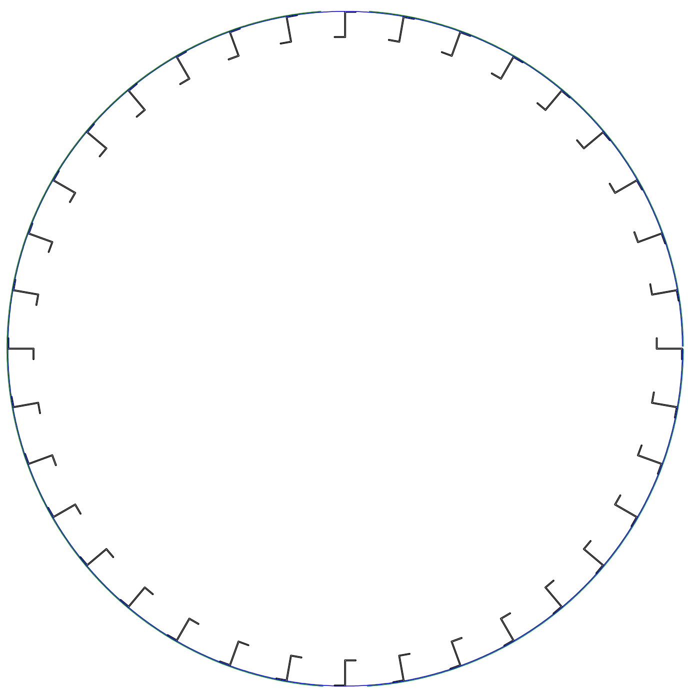

Figure 2.8: Final section design

3. Question 3

We study here an impact lander that would be use on Mars. The purpose of the lander is to use a crusher to absorb the energy of the impact while a cone would root into the surface of Mars, allowing a set of instrument to analyze the surface and the atmosphere of the planet.

3.1 The crusher

We aim to limit the deceleration caused by the impact of the lander at 200g. The crushing distance to perform such results can be calculated by applying the kinetic energy theorem and Newton's second law of motion to the lander from the instant of the impact to the instant the vessel is still:

$$\begin{cases} \Delta KE = \sum W \\ m\mathbf{a} = \sum \mathbf{F} \end{cases}$$

Noting that $\sum W = \int_x^0 \sum \mathbf{F}.d\mathbf{u} = \int_x^0 m\mathbf{a}.d\mathbf{u}$, assuming a perfectly vertical landing on mars and placing ourselves in the worst case of a $200g_{earth}$ deceleration constant until immobility we get:

$$x = \frac{v_i^2}{400g_{earth}} = 0.41m$$

The theoretical crush stress can be obtained from Newton's second law of movement:

$$ma = -mg_{mars} + \sigma_{crush}S_{contact}$$

$$\sigma_{crush} = \frac{ma - mg}{S_{contact}} = m\frac{200g_{earth} - g_{mars}}{\pi(0.9^2 - 0.258^2)/4} = 134.3kPa$$

The crushing distance seems achievable in practice since we have $x \leq 0.45m$ which respects the design specifications given for the lander. The crushing stress seems reasonable.

3.2 The platform material

The crushing stress at the instant of max deceleration will be entirely transmitted to the platform because of the principle of reaction and the cylindrical shape of the crushable sandwich. We want for the platform a material resistant to handle the crash without damaging the instruments but also light since the cost of launching an object into space is mostly linked to its weight.

We have chosen the aluminium alloy 7178-T6, which is an aerospace grade high strength material as well as being light [1]. Its characteristics are the following:

ρ	E	Tensile yield strength	Ultimate tensile strength
2850 kg/m^3	71.7 GPa	538 MPa	607MPa

Table 3.1: Characteristics of the alloy aluminium 7178-T6

3.3 Design of the platform

We use the CAD software Creo to design the platform from the dimensions given in the requirements. The strategy to design the platform is clearly iterative. We first design a simple platform, choose the number of ribs, check if the ribs does not buckle nor the ultimate strength reached, and then go back to the drawing board to try to make the platform lighter by removing matter.

The stress analysis has been performed using shells instead of a volumic solving. This allows to gain a considerable time in the simulation. It required to place the shells at the correct positions corresponding to the lines of applications of the loads between the different parts of the platform.

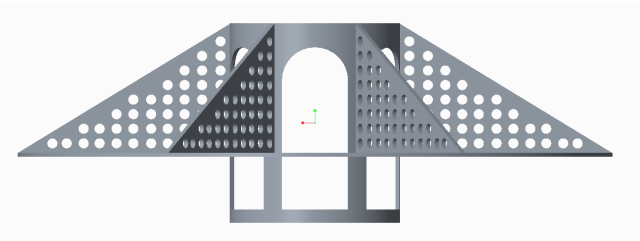

Figure 3.1: Side view of the final design

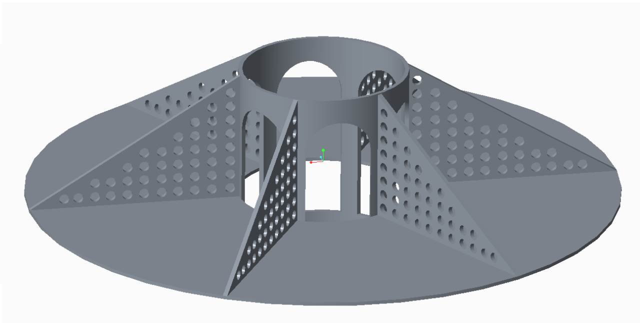

Figure 3.2: General view of the final design

We have chosen after several testings to use 6 ribs on the design, as 4 was not enough to prevent the free sides of the platform to move under the load, and as 8 was judged too much. The final design can be seen on figure 3.1. One can see that the cylindrical center has lost most if its matter between the ribs. This shape was chosen after seeing that most of the cylinder was useless to handle the impact load, the max stress being at the top of the ribs. The rounded top form of the hole is due to that, not to weaken the junction between the ribs and the cylinder.

A close up view of the ribs reveal that the structure is simply a triangle with circular holes in it. The reason for that is that the global weight of the structure was more influenced by the shape of the central cylinder than that of the ribs. It was decided not to make a big hole in the ribs in order not too put a too big load on the cylinder at the top of the ribs. But in order to gain weight on the ribs, some holes were added.

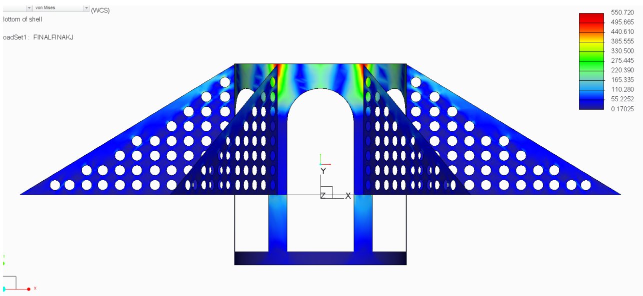

Figure 3.3: Von mises stress on the structure (side view)

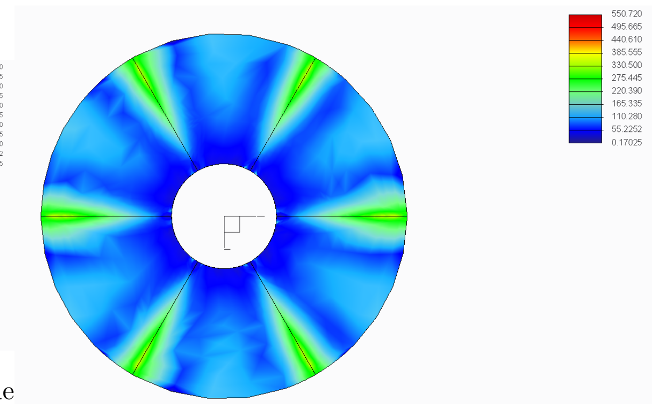

Figure 3.4: Von mises stress on the structure (top view)

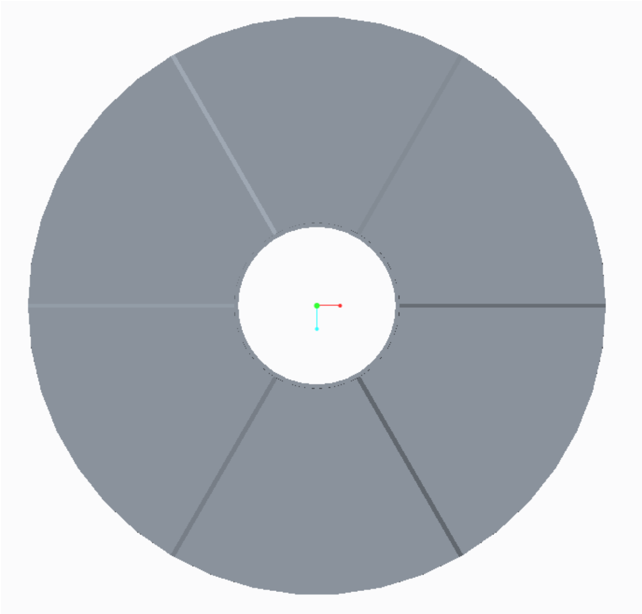

Figure 3.5: Top view of the final design

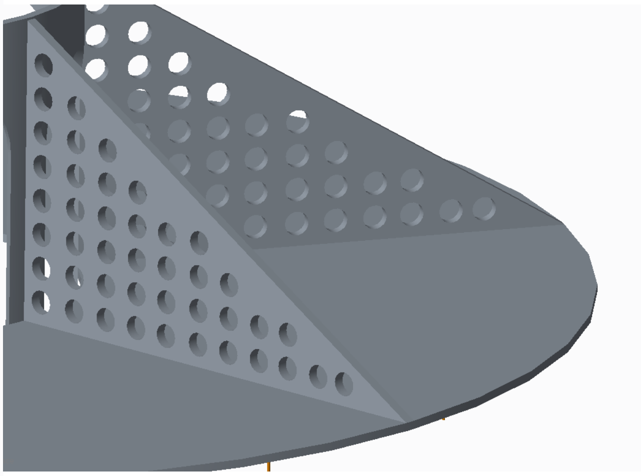

Figure 3.6: Close-up view of the ribs

The stress analysis confirms that for this design we do not have the creation of plasicity and the platform does not brake. However, one could have said that it would have been great to take some risks and allow the platform to plastify by letting the maximum stress being superior to the maximum tensile stress in order to gain some weight. As we are sending a scientific mission to mars, we decided not to jeopardize the project in order to win some kilograms, as a plastic deformation of the platform could damage the instruments.

The following captures attest for the final results of our design.

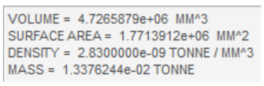

Figure 3.7: Screen capture of the final mass

Figure 3.8: Screen capture of the final B.L.F.

Mass	B.L.F.	Max stress
13.3kg	2.75	550 MPa

Table 3.2: Results of the design

The final weight of our design seems reasonable, and the final platform could be crafter easily from the cylindrical bloc of aluminium. We could have improved our design by using a different material, for instance titanium, but aluminium was chosen as making such a piece from a single bloc of titanium would cost a lot. The dimensions of the final design can be seen on the figure 3.9.

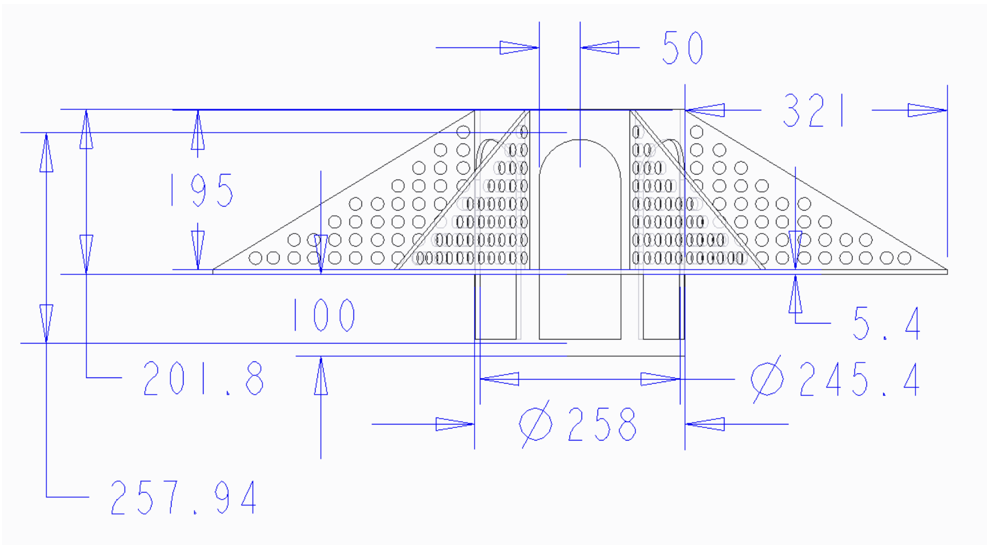

Figure 3.9: Final blueprint of the design

A. Appendices

A.1 Stringer-skin panel correction critical stress correction factor ($\frac{d}{h} = 0.4$)

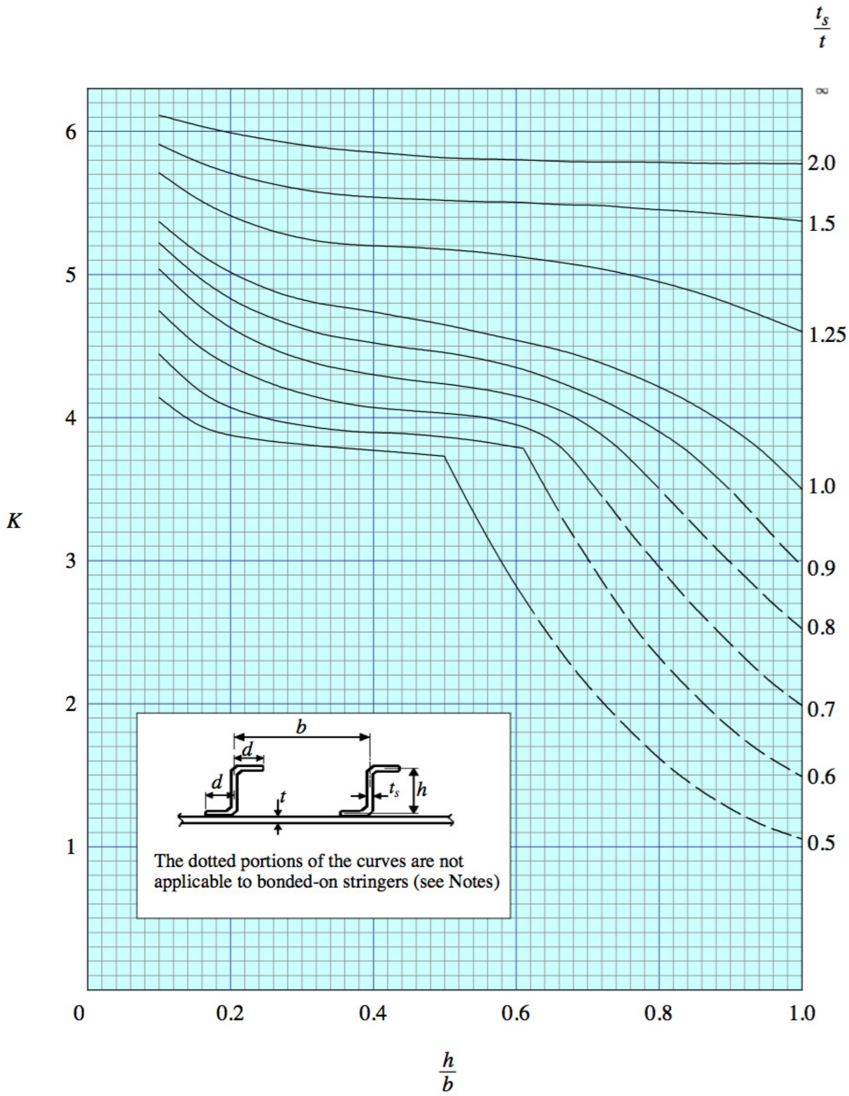

Figure A.1: ESDU datasheet for the correction factor for local buckling of panels with Z stringers

B. References

[1] Metals Handbook, Vol.2 - Properties and Selection: Nonferrous Alloys and Special-Purpose Materials, ASM International 10th Ed. 1990. Summary available here:
http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=MA7075T6 (Aluminum 7075-T6)
http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=MA2024t6 (Aluminum 2024-T6)
http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=MA7178T6 (Aluminum 7178-T6)