CRITERIA FOR STRUCTURAL DESIGN OF AUTOMOTIVE COMPONENTS FORMED FROM THIN SHEET

ANGEL GUERRERO CASTELLS, XAVIER AYNETO GUBERT

ST MECÁNICA APLICADA, S.L.

ABSTRACT

The pressure of an increasingly competitive market is encouraging a widespread cost-cutting and weight reduction policy. For many structural automotive components, these two factors are linked to the thickness of the sheets of materials from which they are formed.

When it comes to conceiving components formed from thin sheet, designers must be aware that the usual strength or stiffness criteria may prove insufficient given the marked complexity of their structural behaviour. The mechanical behaviour of this type of components is markedly non-linear, so that the problems posed for design of this type of parts often escapes from the designer's intuition.

This complexity results, on one hand, from their propensity to present notably non-linear phenomena and instability under service loads, and on the other hand from their sensitivity to factors associated with the physical reality of each particular specimen (initial imperfections, prior stresses, etc.). As a consequence of this, calculations are often incomplete, ambiguous or, at the very least, with a high degree of uncertainty, which makes it advisable to combine both analytical and experimental approaches.

The objective of this paper is to outline, arising from the experience of ST Mecánica Aplicada within this area, the physical bases of the behaviour of this type of components, and to draw attention to the hazards involved in an over-simplistic analysis.

INTRODUCTION

Since several different causes of components failure exist, it is important to correctly identify the ones that may apply to a given design, so that the appropriate analysis methods can be chosen to predict the behaviour.

The failure modes are usually classified as either deformation or fracture. A *deformation failure* is a change in shape or size of a component that is sufficient for its function to be lost or impaired. Cracking to the extent that a component is separated into two or more pieces is termed *fracture*. Besides these, other failure modes may be relevant such as *corrosion*, or the loss of material due to chemical action, and *wear*, or the surface removal due to abrasion or sticking between solid surfaces that touch.

The basic types of failure that are classified as either deformation or fracture are indicated in the figure below.

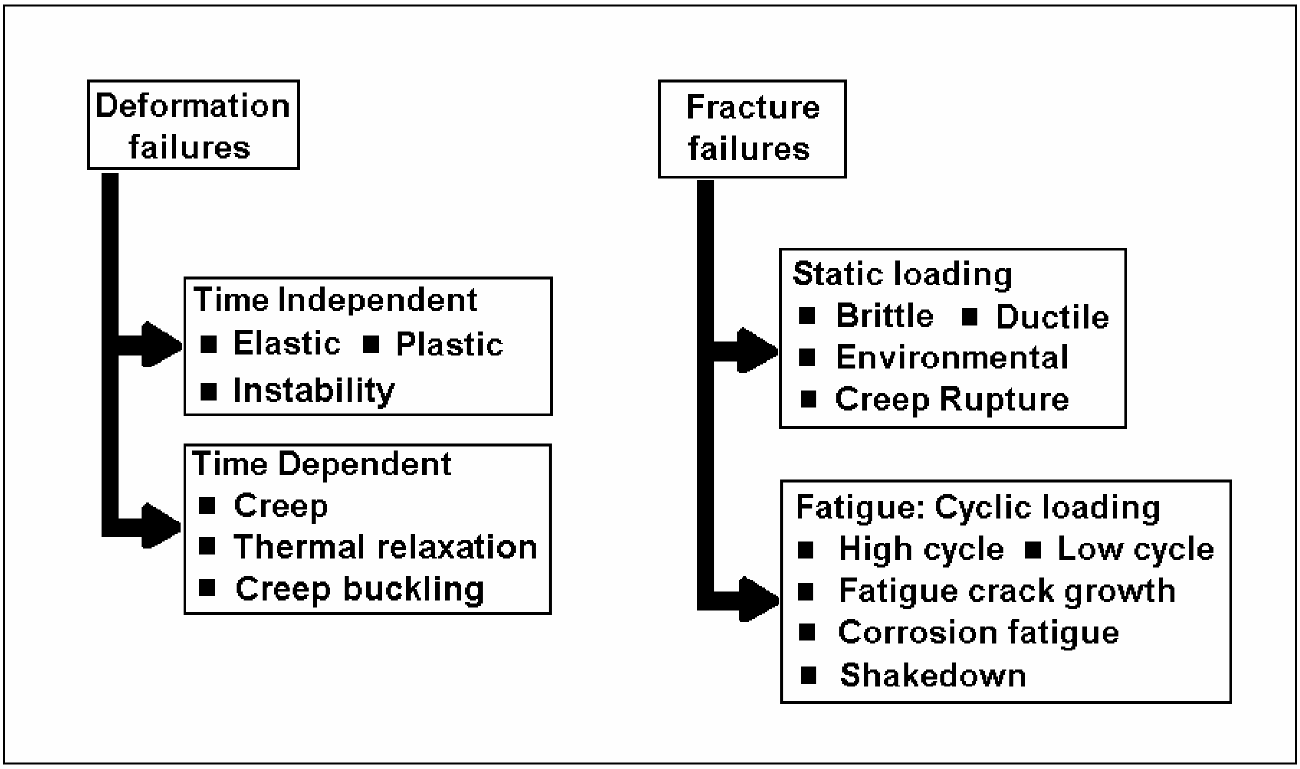


Figure 1: *Basic Types of deformation and fracture failures*

Within this paper, the relevance of a deeply-thought choice of the component or structure failure modes to be evaluated during the design process will be outlined. Attention will be specially drawn to the design of components formed from thin sheet since their markedly non-linear real behaviour may diverge significantly from predictions based on mere strength or stiffness criteria.

As an of example of the stated above, an optimization of the weight of a spot welded Hat section formed from thin sheets will be presented for several arrangements and design criteria, in order to show the importance of taking instability as an ultimate state condition to be checked sistematically when designing this type of components.

INSTABILITY, AN UNAVOIDABLE ULTIMATE STATE CONDITION

The phenomenon of instability or buckling basically consists in a disproportionate increase in displacements resulting from a small increase in the applied load.

This change arises due to a sudden change in the distribution of internal stresses in the part when a critical load value is reached. The compressive membrane stresses, uniform throughout the thickness, are suddenly replaced by flexural and/or torsion stresses of very much higher values.

This shift of behaviour can be either reversible, remaining within the elastic field, case in which its immediate effects can disappear, or it irreversible leading to permanent plastic deformation. Instability is therefore associated with compressive membrane stresses whose critical value depend on the intrinsic stiffness of the material and the geometry of the part. Designers must therefore pay special attention to this situation.

Instability can be either overall or local (see figure 2). Where overall, the component presents overall strength failure, while local instability only leads to partial failure. A typical example of overall instability would be the collapse of a bar under compression (large wavelengths within the deformed shape), while an example of local instability would be the crippling of the zone between two spot welds that are spaced too far apart (short wavelengths), as it will be shown later.

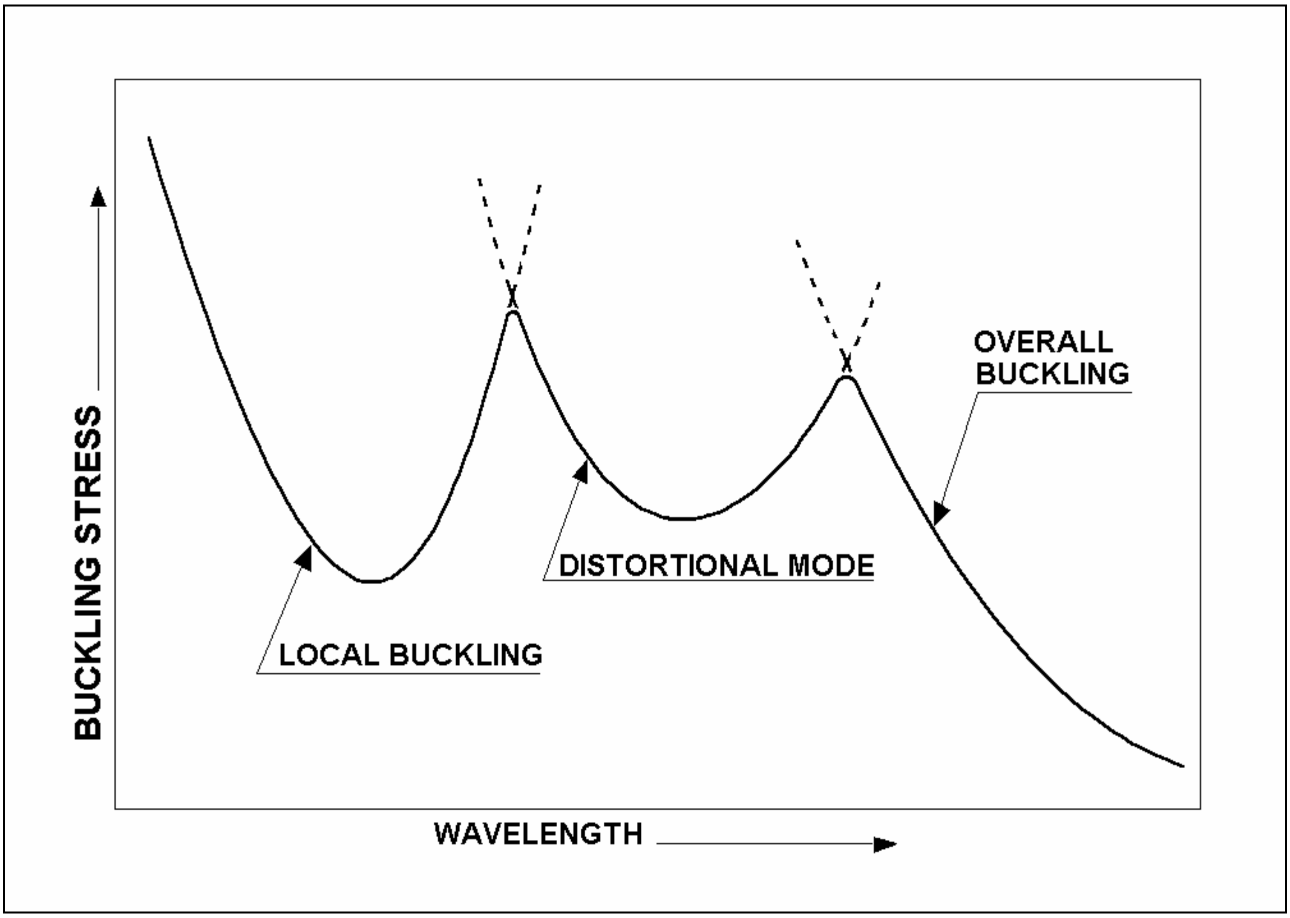


Figure 2: *Evaluation of the buckling stress*

The instability point can be reached without existing any prior non-linear behaviour, or it can arise after a zone of high non-linear behaviour. The calculations to be applied in each case are considerably different.

When perfect components are statically analysed, there can be found two phenomena usually referred to as buckling: the collapse at the maximum point of the load-displacement curve or equilibrium path and the bifurcation of the equilibrium.

Figure 3 shows several options for equilibrium paths. Following an initial behavioural phase which may be linear or non-linear, a point of bifurcation (λ_B) is reached at which a geometrically perfect component can evolve in two ways: either towards a maximum of the initial equilibrium path (collapse load λ_P) or according to a second equilibrium path which can be stable or unstable. Where this secondary path is stable, the component will present a post-critical strength reserve, while if it is unstable it will show collapse. In this second case the component is said to be sensitive to imperfections.

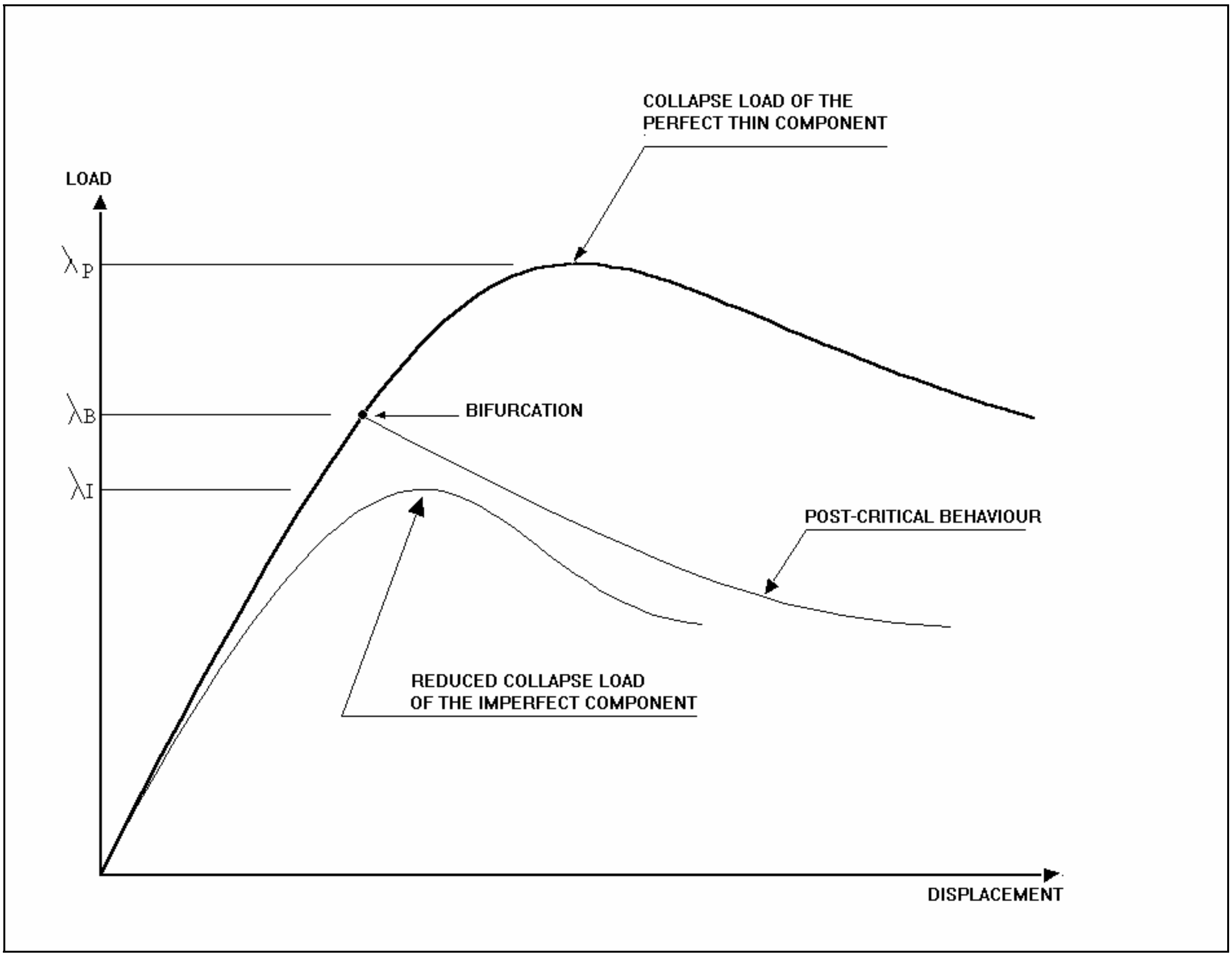


Figure 3: *Several equilibrium paths (I)*

A real component, which has imperfections, presents a reduced collapse load (λ_I) lower than the bifurcation load of the perfect component (λ_B), as it can be seen in last figure.

The difference between the bifurcation load of a perfect component (λ_B) and the reduced collapse load of a real component (λ_I) will depend on initial imperfections and on the shape of the secondary equilibrium path of the perfect component

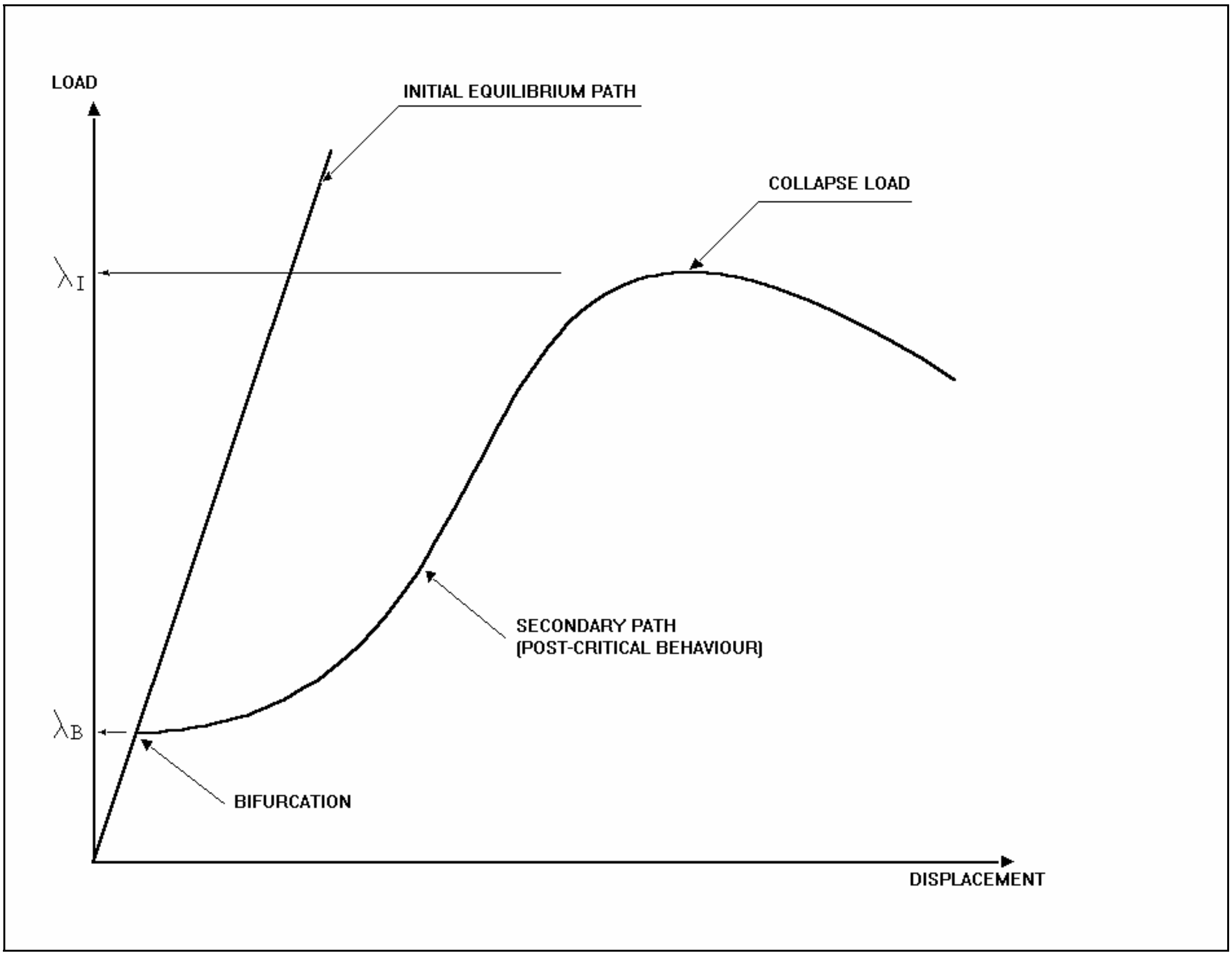


Figure 4: *Other possible forms of behaviour (II)*

Figure 4 presents another possible form of behaviour of a component: after an initial linear phase, two possible equilibrium paths arise at the bifurcation point. A real component would follow the secondary path, which in this case is stable due to its positive slope. In this case the bifurcation load λ_B characterises a local buckling with little effect on overall strength. The collapse load λ_I is finally reached.

When weight plays a fundamental role, the several parts of a structure are allowed to exceed the local buckling loads, in order to take advantage of their post-critical strength. (Note here that if this capacity is to be taken into account in the optimisation process of a design, the utmost caution must be taken when making physical simplifications or component behavioural models - when, for example, geometrical imperfections are brought into calculations).

Another equally risky approach lies in designing so that overall and local instabilities arise simultaneously (the load capacity of structures with simultaneous local and overall buckling can be especially sensitive to initial imperfections).

On components submitted to cyclical loadings in which plastic fields arise, instability can appear in a delayed manner due to the progressive accumulation of plastic deformation, possibility to be taken into account when determining the lifetime of the component. It should be noted here that the appearance of parasitic or secondary stresses during crippling can give rise to fatigue fracture even in the elastic field.

INSTABILITY OF SPOT-WELDED THIN PROFILES

Resistance spot welding is the most widely used joining method in automobile manufacture. The number, location and quality of welds are some of the factors that influence the performance of welded subassemblies and body panel structures.

When a spot welded joint is subjected to external load the maximum local stresses generally occur on the inside of the plate at the edge of the weld spot nugget. The stress increase at the weld spot is caused primarily by the concentration of the force flow on to the jointing face of the weld spot, flow concentration both in the plane of the plate and in planes perpendicular to it, the latter particularly with tensile shear load (see figure 5).

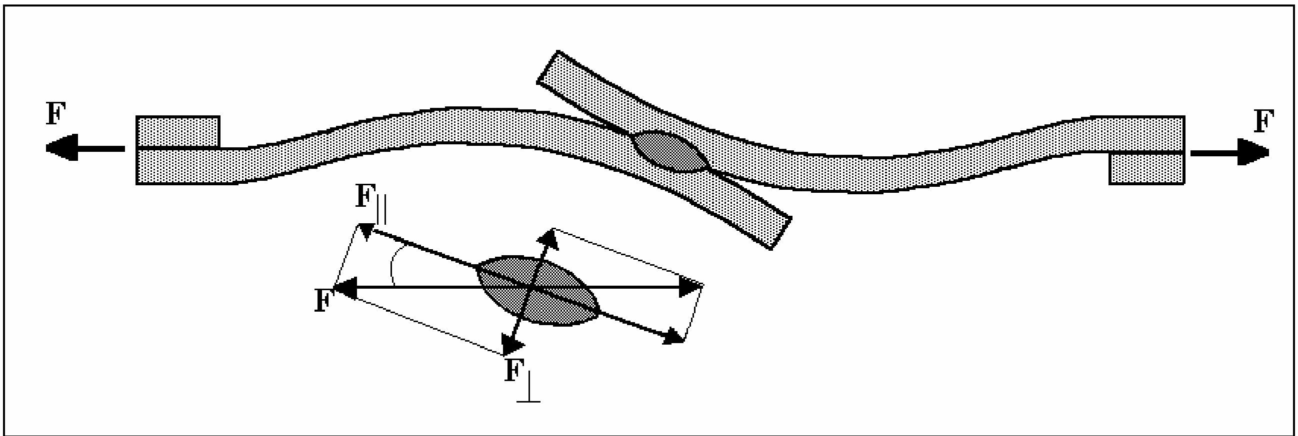


Figure 5: *Slanting of weld spot subjected to tensile shear, superimposed cross tension by slanting*

Non-linear effects in the form of large deflections (large relative to plate thickness) and large rotations occur particularly with thin sheet metal. The large deflections, mainly in the immediate vicinity of the weld spots may, after overcoming the gap width, cause the jointed plates to touch and support each other. On the other hand, they may cause buckling, primarily at some distance from weld spots. Assuming ideal geometry without predeformation, the buckling occurs as an instability phenomenon.

A characteristic feature for corresponding numerical models on the other hand is that predeformation, notably gaps and bending distortion, has to be simulated in the initial state, and that the numerical results (stresses, strains, deflections, rotations, reaction forces) depend non-linearly on the load level. The large rotations mentioned have an influence on the mechanical behaviour of weld spots subjected to tensile shear loading (see again figure 5) and also require simulation by a non-linear model.

If a bending load is applied to spot welded box section members manufactured from thin hat profiles, with spot welded flanges in the region of maximum bending stresses, the flanges are subjected to longitudinal compression and tension. Assuming small deformations, this does not cause any local increase in stress at the weld spots (by appart from the geometrical stress concentration). In reality though, large deflections with high bending stress do occur at the weld spot as well as in the middle of the flange section between two weld spots, specially when subjected to longitudinal compression.

The gap formation at the flanges under longitudinal compression, figure 6, can be interpreted as a rod or plate buckling process. In the case of a box section member manufactured from hat profile and cover plate, it is the cover plate particularly which is at risk of buckling. It deflects in the shape of a half wave between two weld spots even when subjected to relatively low longitudinal stresses.

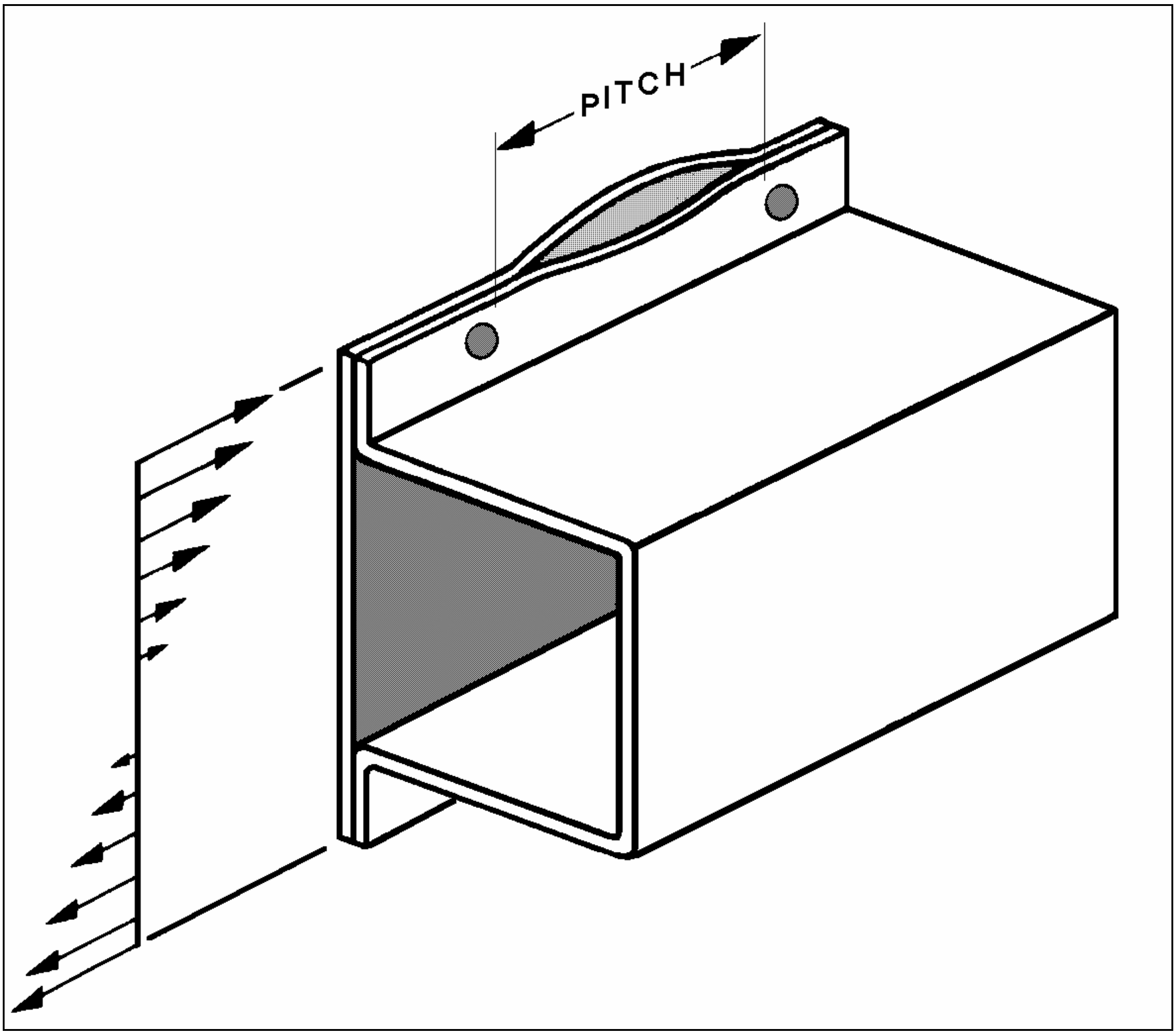


Figure 6: *Flange gap in spot welded Hat sections with cover plate under bending*

As a practical application of the stated above, an optimization analysis of the weight of a spot welded Hat section formed from thin sheets has been worked out for several arrangements and design criteria, in order to show the importance of taking instability as an ultimate state condition to be checked sistematically when designing this type of components.

For this purpose, the finite elements method has been used, having determined a priori the ranging limits of the design variables by means of rough approximate formulations.

OPTIMIZATION ANALYSIS OF A SPOT WELDED HAT SECTION THIN PROFILE

A spot welded aluminium hat section formed from thin sheets has been optimized to achieve a minimum-weight component that accomplishes the specified design criteria under a bending load. In order to enhance the relevance of a sensible choice of the component or structure failure modes to be evaluated within the design process, optimizations have been worked out for two different failure modes, this is to say,

- Avoiding local and overall instabilities under loads lower than $\gamma_f \times$ (service load) (where γ_f is a partial safety factor for loadings, at a value of 1.5)
- Limiting stresses under the service load to a value corresponding to the material yielding point reduced by a partial safety factor of 1.1.

Three different hat section arrangements often used in automotive applications have been analysed (see figure 7): two single hat profile specimens and one peaked hat profile specimen.

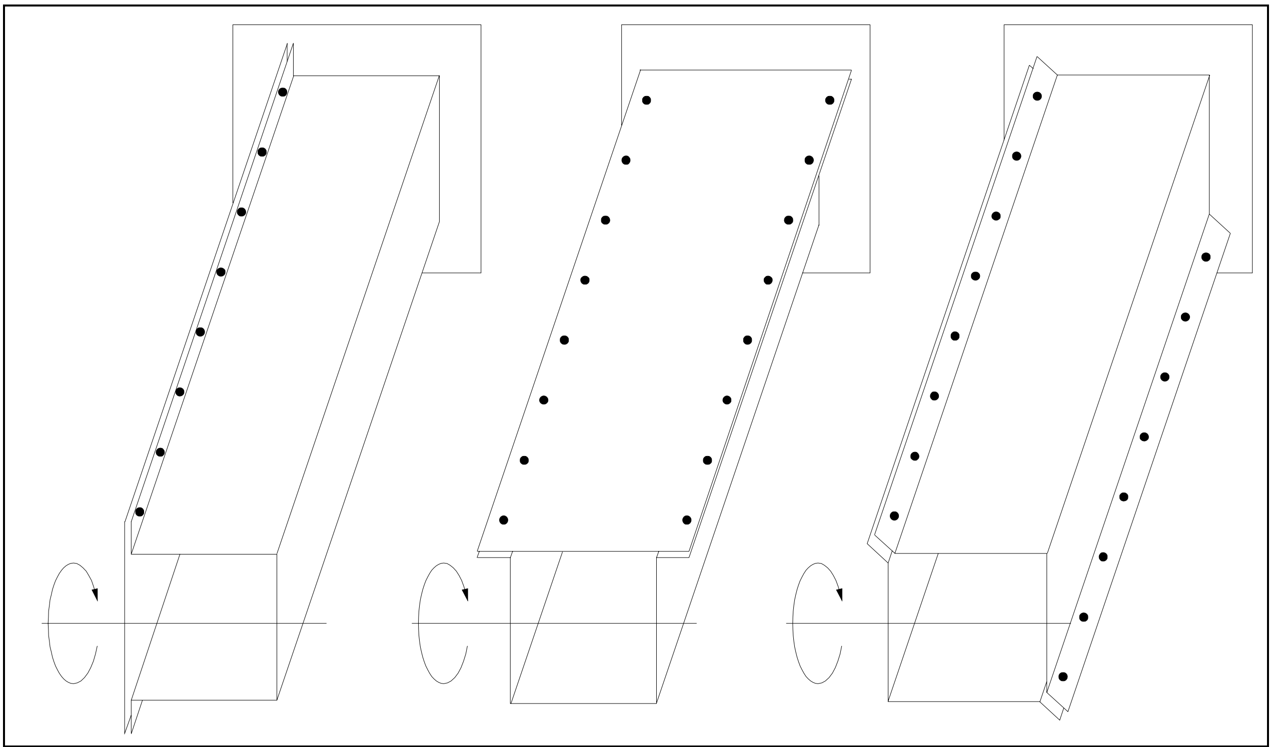


Figure 7: *Three different spot welded hat section arrangements*

Design parameters of spot welded hat sections include plate thickness, plate width, spot diameter, number of spots, spot arrangement, distance between spots, nature and level of imperfections and the cross sectional shape of the plate. As the present analysis is posed for showing plainly to the reader the relevance of considering the appropriate design criteria when optimizing a thin profile, only two design parameters have been taken into account for each hat section arrangement (see table 1):

DESIGN PARAMETERS	Initial value	Lower limit	Upper limit
Sheet thickness (uniform in all the cross section)	1 mm	0.5 mm	1.5 mm
Distance between spots (the pitch)	50 mm	10 mm	90 mm

Table 1: *Design parameters for the optimization and ranging limits*

The search for the minimum-weight design parameters values such to assure that no instabilities will arise under 1.5 × (service load), has been performed through 'linear buckling' finite elements calculations, resulting in the load factors at which instabilities appear for each of the design parameters combinations analysed. On the other hand, for the stress-based optimization, linear-elastic finite elements calculations have been developed.

(Note: These over-simplistic calculations have been set to assure an easy understanding of the concerning basic concepts. A realistic optimization process would have to be based on non-linear calculations involving elasto-plastic material behaviour, large displacements and rotations, and taking into account the effect of initial imperfections and plates contacts.)

The results of the optimization processes are the ones shown in table 2:

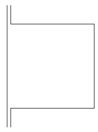
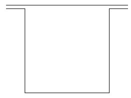
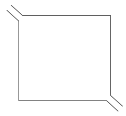
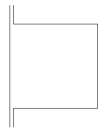
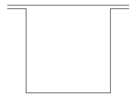
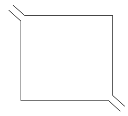
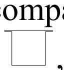
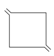
	STABILITY OPTIMIZATION			**STRESS-BASED OPTIMIZATION**		
ARRANGEMENT ➔						
Pitch (mm)	65	28.1	**90**	67.4	**90**	90
Thickness (mm)	1.17	0.96	**0.82**	0.9	**0.78**	0.85
Weight (Kg/m)	1.891	1.494	**1.345**	1.465	**1.295**	1.402

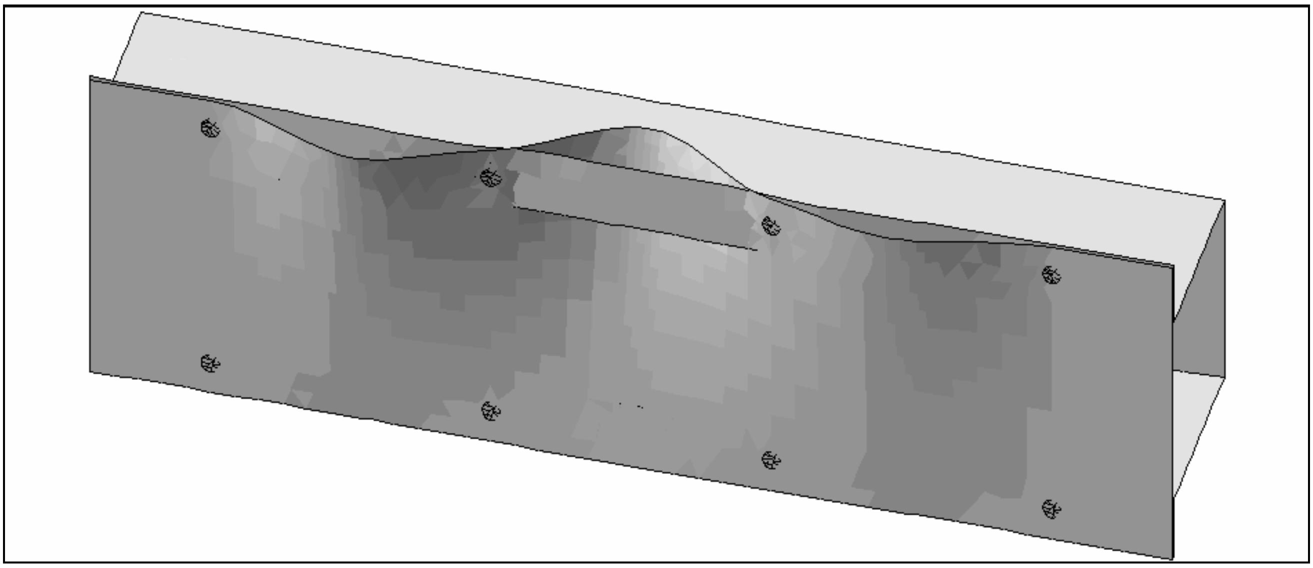
Table 2: *Design parameters combinations resulting from the optimizations*

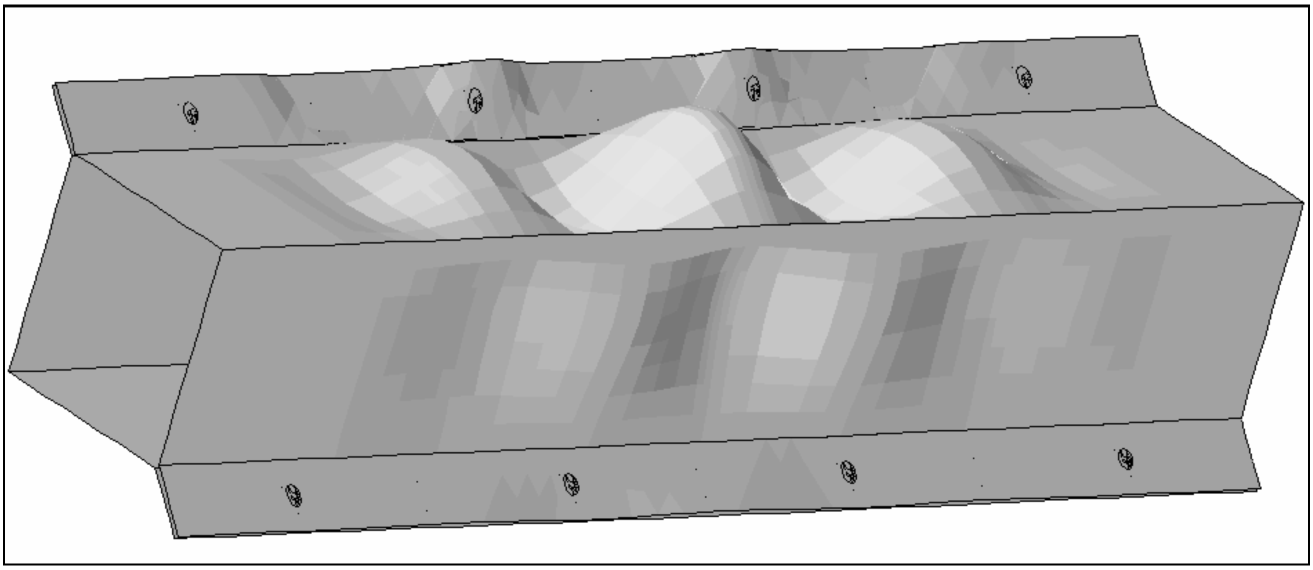
Arising from these results the reader will find that the best combination of the design parameters values and arrangement, under stability constraints, does not match that obtained from the stress-based optimization (even the optimum spot welds arrangement is different according to each optimization). A more representative optimization process would combine both stability and stress groups of constraints in an unique analysis to obtain a final product that met the whole set of design requirements with a minimum weight.

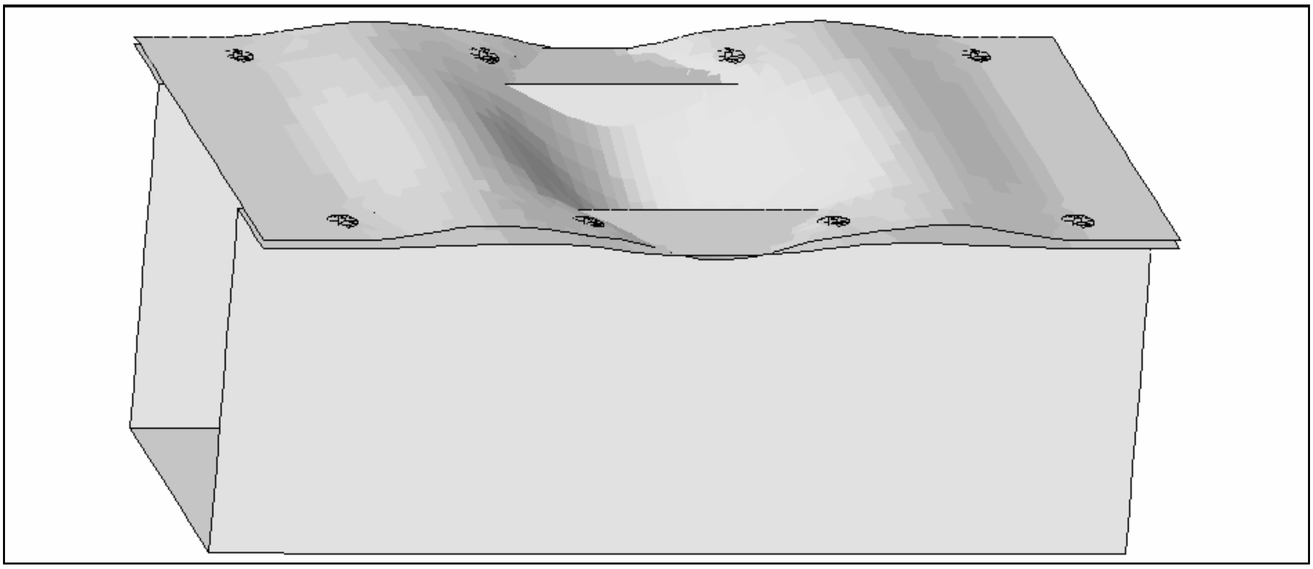
In this analysis, the stability optimization generally meets convergence when considering non-slender components (low pitch and high thickness) if compared to those stress-based. Indeed, the optimum profile obtained according to the latter, , does not reach the settled minimum buckling load so it would fail even when subjected to relatively low longitudinal stresses (in this arrangement, the slenderest element of the hat profile, the cover plate, is uniformly compressed at the maximum stresses). As a consequence of the stated above, the stability constraints have turned out to be a more severe design criteria than mere yielding.

For the peaked hat arrangement () stress restrictions have resulted slightly harder than the stability ones, mainly due to non-existing a plain slender cover plate (the corner and the welding flange stiffen the compressed upper web, decreasing its propensity to be unstable).

Next figures 8 to 10 show the different types of instabilities that arise from the finite elements stability optimization at each one of the arrangements studied under bending. (The reader must remember the simplifications stated before, such as non-simulating the plates contacts).







Figures 8 to 10: *D*ifferent types of instabilities arising from the finite elements stability optimization at each one of the arrangements studied under bending

CONCLUDING REMARKS

When conceiving components formed from thin sheets, the designer must take the utmost care in order to avoid over-simplistic analysis that may lead to faultily-optimized components. Some of the cautions to be held during the design process are the following ones:

- þ Take instability as an ultimate state condition to be checked systematically.
- þ Limit the distance between supports of compressed components.
- þ Take control of the free length of compressed members between restrictions such as spot welds. The designer must take in consideration enabling formal or added stiffeners on compressed members prone to local buckling under service loads.
- þ Check the imperfections level within final product and its material quality.
- þ Reduce as much as possible compressive load eccentricities.
- þ Take into account possible stress redistributions that may overload compressed components in case of appearing local yielding.
- þ Check the possible generation of cyclic cripplings that may produce increasing strains or fatigue fracture.
- þ Let the choice of the types of calculations be based on a deeply-thought analysis of the component failure modes and of its expected behaviour: abusing of linear calculations leads to faulty designs, while abusing of non-linear calculations leads to unbearable over-expensive analysis.

REFERENCES

- Brush, D.O., and B.O. Almroth: “Buckling of Bars, Plates, and Shells”, Mc Graw-Hill Book Company, 1975
- Kollár, L., and E. Dulácaka: “Buckling of Shells for Engineers”, English transl. edited by G. R. Thompson, John Wiley & Sons, Inc., 1984
- Bushnell, D: “Computerized Buckling Analysis of Shells”, Kluwer Academic Publishers,1985
- Allen, H. G., and P. S. Bulson: “Background to buckling”, Mc Graw-Hill Book Company, 1980