3.3 SURFACE ELEMENTS

This section presents an overview of the behavior of three dimensional sheet components that cover a surface or area. They may be flat, curved about a single axis, or doubly curved. These surface elements may be subjected to lateral or in-plane loadings. The load can be carried by these elements by shear or by flexural or axial action.

This section gives a basic understanding of the expected structural behavior and presents the variables that govern that behavior. References furnish the more specific data that may be required.

Flat surfaces are discussed first. This is followed by singly curved surfaces and then doubly curved surfaces. This leads to a discussion of dent resistance.

3.3.1 FLAT SURFACES ....

Flat plates or panels subjected to loads normal to their surface have flexural deformation and flexural stresses just as linear members subjected to distributed lateral loads. However, flexural stresses exist in both directions at all points on a plate. Deflections and the stresses are functions of plate dimensions and edge conditions.

3.3.1.1 Flexure

The expression for maximum deflection of a rectangular plate subjected to a uniformly distributed load per unit area, **q**, can be written in the following form:

$$\frac{w_m}{t} = \frac{C_w q \left(\frac{L_1}{t}\right)^4}{E\left[\left(\frac{L_1}{L_2}\right)^2 + 1\right]^2}$$

Equation 3.3.1.1-1

where w_m = maximum plate deflection
t = plate thickness
L_1 = shorter of the rectangular plate dimensions
L_2 = larger of the rectangular plate dimensions
E = modulus of elasticity
C_w = deflection coefficient that varies with edge restraint and with $\frac{L_1}{L_2}$ ratio

Without considering a change in $\mathbf{C_w}$, it can be observed that there is a significant difference between a square panel in which $L_1/L_2 = 1$ and a very long narrow panel where $L_1/L_2 = 0$. The exact ratio of the center deflections of a square plate to that of a long narrow plate of the same $\mathbf{L_1}$ is 0.32.

This difference reflects the fact that in a square plate, the load spans both ways and the torsional plate stiffness (and resulting twisting moments developed) enhances the plate's load carrying capacity. The aspect ratio L_1/L_2 has an effect on the maximum stress in the plate, $\mathbf{f_{bm}}$ similar to the effect it has on deflection. This stress is:

$$f_{bm} = \frac{C_f\, q \left(\frac{L_1}{t}\right)^2}{\left(\frac{L_1}{L_2}\right)^2 + 1}$$

Equation 3.3.1.1-2

where $\mathbf{C_f}$ is a coefficient that is a function of edge restraint and aspect ratio L_1/L_2.

With simply supported edges, the stress $\mathbf{f_{bm}}$ in a square plate is 0.38 of the $\mathbf{f_{bm}}$ value in a long narrow plate. In a long narrow plate, the load spans across the short dimension $\mathbf{L_1}$ creating a one way bending situation except near the side supports. Values of $\mathbf{C_f}$ and $\mathbf{C_w}$ can be obtained for numerous variations of plate shape and edge condition from data provided in Reference 1.

The flexural stiffness of the plate can be improved very economically by simply corrugating the sheet. See Figure 3.3.1.1-1(a).

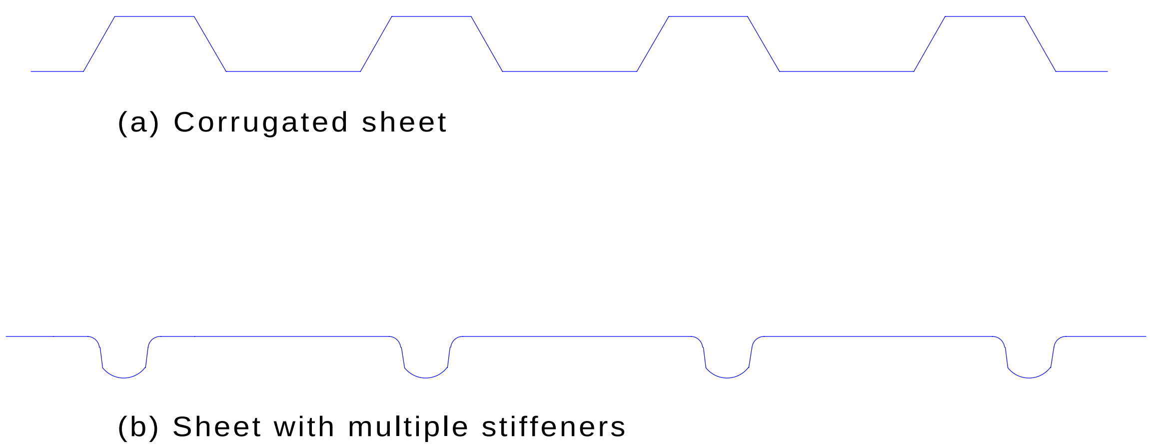

Figure 3.3.1.1-1 Ribbed and corrugated sheet

This produces a substantial stiffness and load capacity improvement in the direction of the ribs. Even stiffeners such as shown in Figure 3.3.1.1-1(b) can produce a significant improvement in flexural properties. The ribbed or corrugated plate basically exhibits one way flexural behavior, since the perpendicular direction retains the properties of a flat plate.

3.3.1.2 Membrane Behavior

The presence of the side support has a pronounced effect on plate behavior. The plate begins to develop significant in-plane stresses after deflecting an amount $\mathbf{w_m > t}$. If the span-to-thickness ratio is large, a plate will rely on in-plane or membrane stresses for most of its load carrying capacity.

The center deflection under combined bending and membrane stress, $\mathbf{w_o}$, can be expressed in the form

$$w_o = \frac{w_m}{1 + C_o\left(\frac{w_o}{t}\right)^2}$$ **Equation 3.3.1.2-1**

where w_m = bending deflection from Equation 3.3.1.1-1 or Equation 3.3.1.2-2
C_o = coefficient which is a function of rotational and in-plane edge restraint

The problem is now nonlinear in that the plate stiffness increases with increasing deflection. $\mathbf{C_o}$ is smallest if the edge of the plate is free to move radially. Values for circular plates of radius $\mathbf{a_p}$ are provided in Table 3.3.1.2-1.

Table 3.3.1.2-1 Deflection coefficients for circular plates

Rotational Edge Restraint	C_o		C'_w
	Free	Immovable	
Clamped	0.146	0.471	0.171
Simply Supported	0.262	1.852	0.696

$\mathbf{C'_w}$ in the Table 3.3.1.2-1 is the deflection coefficient for a circular plate. The center deflection of a circular plate with radius $\mathbf{a_p}$ under uniform pressure $\mathbf{q}$ can be expressed as:

$$\frac{w_m}{t} = \frac{C'_w q}{E}\left(\frac{a_p}{t}\right)^4$$ **Equation 3.3.1.2-2**

The tensile membrane force developed is restrained by an infinitely stiff compression ring in the case of the immovable edge. A compression ring is developed in the outer portion of the plate if the plate has a free edge. This is illustrated in Figure 3.3.1.2-1. It is possible to buckle the plate from the ring compression developed. In practice the edge usually has a stiffener (e.g. bent lip) which provides additional area for the ring and moment of inertia to resist ring buckling. Thus, the coefficient $\mathbf{C_o}$ usually ranges between the free and immovable values.

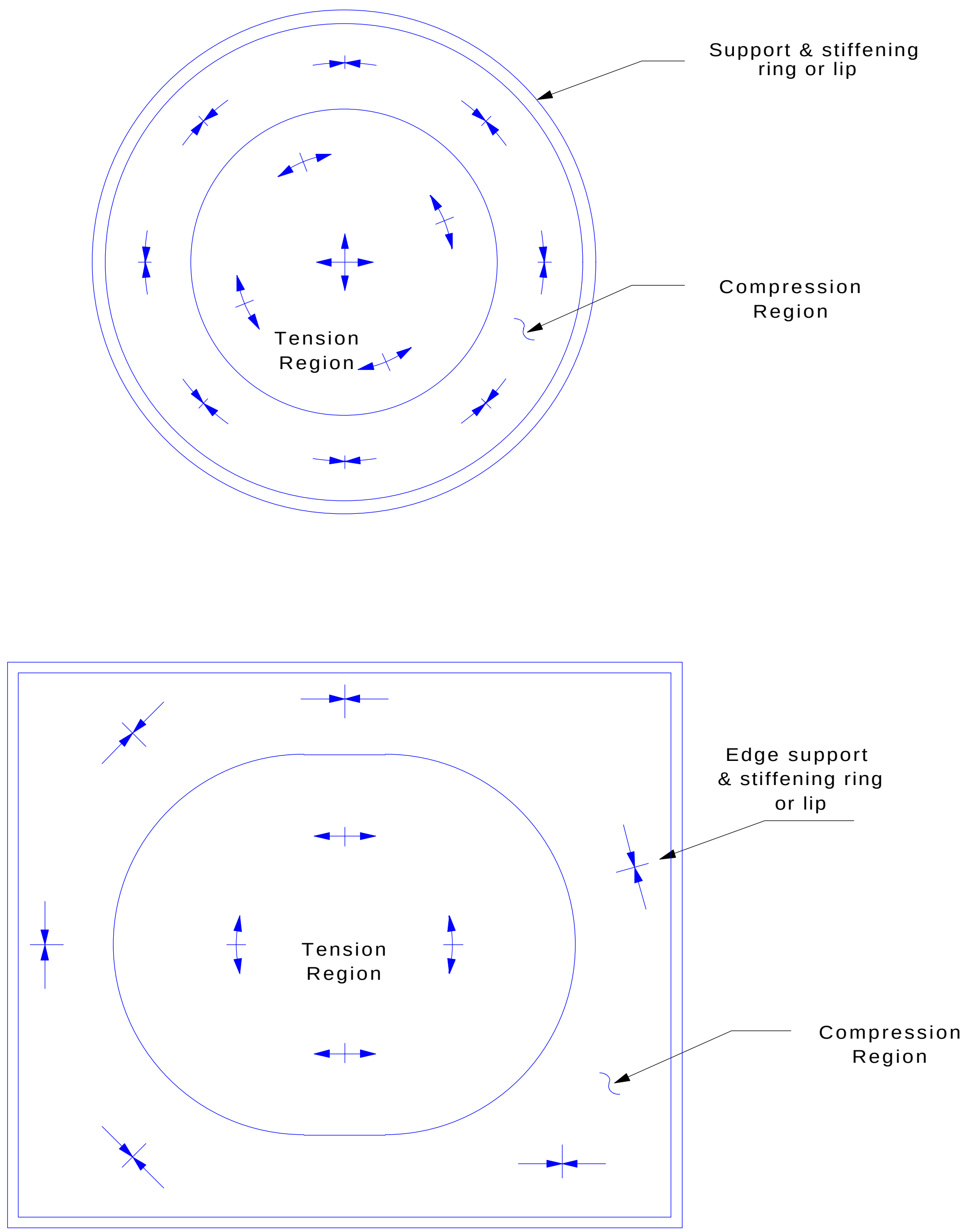

Figure 3.3.1.2-1 Membrane action in flat plate

The membrane and bending stresses developed are a function of the parameters shown in Table 3.3.1.2-2. Stress coefficients for circular plates are given in Reference 1.

Table 3.3.1.2-2 Membrane and bending stress parameters

Membrane	$E\left(\frac{w_o}{a_p}\right)^2$
Bending	$E\left(\frac{w_o t}{{a_p}^2}\right)$

The coefficients for circular plates can be used to approximate the situation for square plates or plates close to square by inscribing a circular plate over the actual plate. To be somewhat more conservative, the radius of the plate used in calculation should be taken as a bit larger than that of the inscribed circle. Tabulated information on membrane stresses and deflection of rectangular plates is given in Reference 2.

If the flat plate is very thin the flexural integrity of the plate can be completely ignored. For the case of a circular membrane clamped along the circumference under uniform pressure **q**, the center deflection can be evaluated as:

$$w_o = 0.662 a_p \sqrt[3]{\frac{q a_p}{E t}}$$ **Equation 3.3.1.2-3**

The corresponding tensile membrane stresses at the center and perimeter of the membrane are as follows:

$$(f_r)_{r=0} = 0.423 \sqrt[3]{E\left(\frac{q a_p}{t}\right)^2}$$ **Equation 3.3.1.2-4**

$$(f_r)_{r=a} = 0.328 \sqrt[3]{E\left(\frac{q a_p}{t}\right)^2}$$ **Equation 3.3.1.2-5**

Note that deflection is proportional to the cube root of the load intensity, but stress is proportional to the 2/3 power of the load intensity.

Stiffening beads are added to floor pans, dash panels and cargo box floors. In addition to increasing the flexural stiffness normal to the sheet, the beads are oriented to increase the load capacity for axial loadings imposed in crashes. (Beads oriented perpendicular to the direction of impact decrease the crush loads while beads oriented parallel increase crush loads.) The beads also suppress vibrations.

Design Procedure 3.3-2, located in Section 5.4, incorporates the procedure described in this section. This procedure and others shown in Section 5 are implemented in AISI/CARS.

3.3.1.3 Compressive Integrity

The integrity of the flat plate loaded in compression in one direction forms the basis of the effective width concept used for flat elements of cold formed members, as described earlier in this Chapter. The basic equation for the critical stress representing the buckling of the rectangular plate is:

$$f_{cr} = \frac{k\pi^2 E}{12\left(1-\mu^2\right)\left(\frac{w}{t}\right)^2}$$

Equation 3.3.1.3-1

where **w** and **k** are as shown in Figure 3.3.1.3-1 for the simply supported flat plate.

In the figure, **m** is the number of half since waves in the buckled configuration.

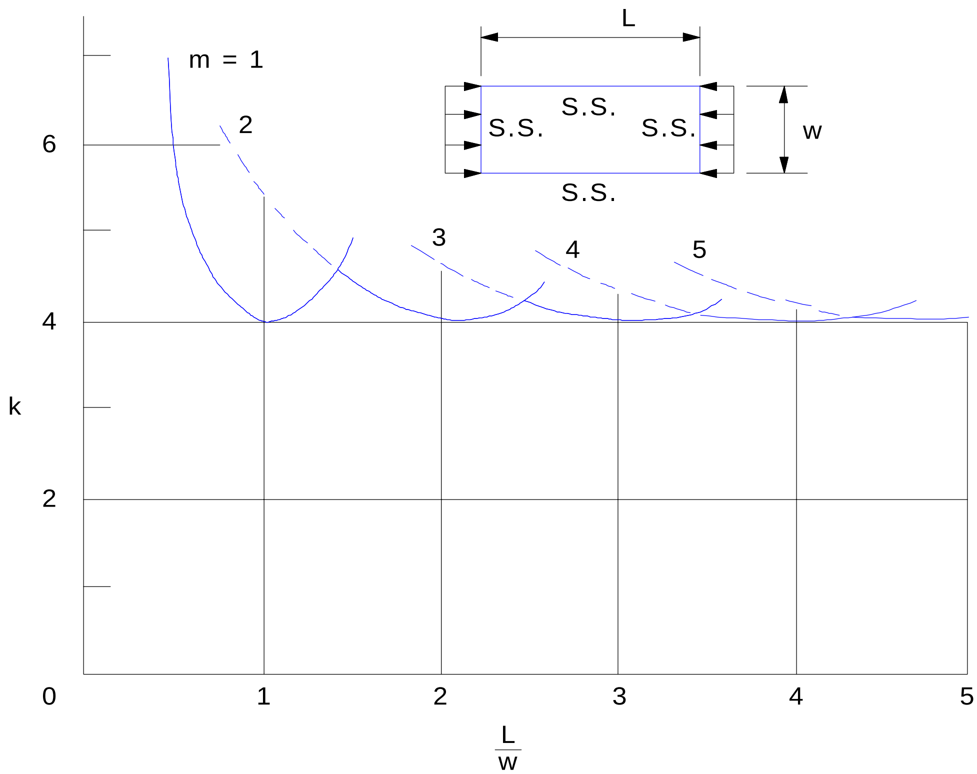

Figure 3.3.1.3-1 Buckling coefficient for simply supported flat plate

The expression in Equation 3.3.1.3-1 is applicable when **L** is equal to or greater than **w**. For **L** small relative to **w**, the buckling load will approach the Euler load for the plate. At **L = w**, the plate buckling load (four sides simply supported) is four times that of the plate with two sides free. Thus, the edge support is very beneficial.

Values of **k** for other edge conditions and other loads on a rectangular plate are given in Reference 3. Other plate buckling information for flat plates is tabulated in Reference 2.

Stiffening of a flat plate by attaching angles or other shapes to the flat plate can significantly improve the axial load capacity. The stiffening can also be achieved by corrugating or adding ribs to the plate as per Figure 3.3.1.1-1.

3.3.2 SINGLY CURVED SURFACES

The addition of a single parabolic or cylindrical curvature to a flat plate can produce a sizable increase in lateral load carrying capacity. This type of surface is the same as that shown in Figure 3.2.1-1. The lateral load is carried to the support primarily by means of in-plane compressive load. The plate bending is small relative to that of a flat plate on the same span.

The effective length for use in compression load design can be determined from Equation 3.2.1-1. Thus, in the elastic range, the critical buckling stress is

$$f_{cr} = \left[\left(\frac{2\pi}{\alpha}\right)^2 - 1\right]\left[\frac{E}{12\left(1-\mu^2\right)\left(\frac{a}{t}\right)^2}\right]$$

Equation 3.3.2-1

The ends of the curved plate must be restrained from translation in order for this buckling load to be achieved.

The bending induced depends on the type of loading. A uniform normal pressure produces no bending for a circular arch shape. A uniform pressure over a horizontal surface produces no bending on a parabolic arch shape. Any concentrated or non-uniform distributed load produces flexural stresses which must be evaluated in conjunction with the axial loads using Equation 3.1.3.7-7.

The addition of stiffening along the edges of a thin singly curved plate can provide a substantial improvement in the stiffness and strength of the plate. The edge support is estimated to provide about four times the critical stress obtained from Equation 3.3.2-1 provided that the stress remains in the elastic range. The addition of corrugations or ribs along the curved surface can further improve the integrity of the surface.

If the rise-to-span ratio is small, buckling can be in a symmetrical mode rather than the unsymmetrical mode shown in Figure 3.2.1-1.

3.3.3 DOUBLY CURVED SURFACES

The spherical shape is the most common doubly curved surface in that it has only one degree of curvature. A uniform pressure of **q** produces a membrane stress

$$f_m = \frac{qa}{2t}$$

Equation 3.3.3-1

where **a** is the spherical radius and **t** is the thickness of the spherical surface (See Figure 3.3.3-1(a)).

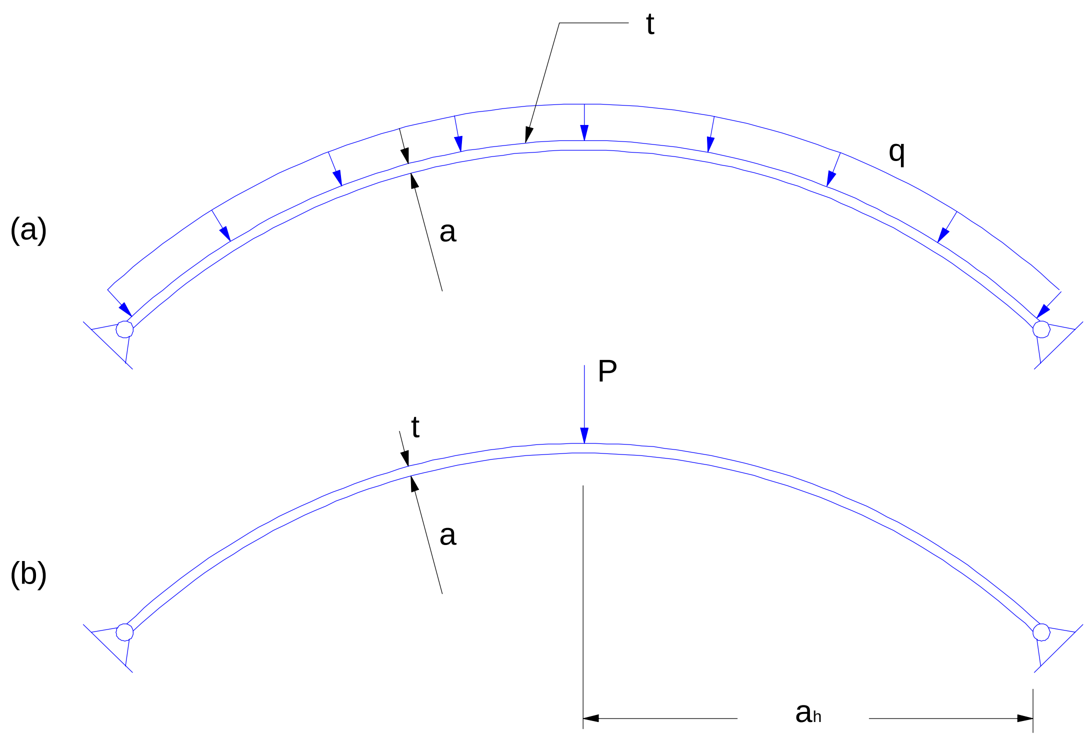

Figure 3.3.3-1 Spherical surfaces subjected to lateral load

An external pressure **q** of sufficient magnitude will lead to a buckling load which can be represented by the following equation.

$$q_{cr} = CE\left(\frac{t}{a}\right)^2$$ **Equation 3.3.3-2**

The theoretical value of **C** is approximately 1.2. However, the spherical shape is extremely sensitive to imperfections. So the achievable value of **C** is typically no more than 0.3 to 0.4. The amount of imperfection and edge effects (usually edge deflections) can reduce the value of **C** even further.

If the surface is not of a single thickness **t**, but stiffened by ribs in some manner, Equation 3.3.3-2 can be extended to the form

$$q_{cr} = CE\left(\frac{t_m}{a}\right)^2\left(\frac{t_B}{t_m}\right)^{3/2}$$ **Equation 3.3.3-3**

where $t_m = \frac{A_s}{S_s}$, equivalent membrane thickness evaluated as the stiffener area divided by the stiffener spacing **Equation 3.3.3-4**

where $t_b = \sqrt[3]{\frac{12 I_s}{S_s}}$, equivalent bending thickness evaluated from the stiffener moment of inertia **Equation 3.3.3-5**

If the in-plane buckling stress exceeds the proportional limit of the material, the above buckling expression should have **E** replaced by $\sqrt{E_s E_t}$ where $\mathbf{E_s}$ and $\mathbf{E_t}$ are the secant and tangent moduli of elasticity, respectively. See Figure 3.3.3-2 for the definition of $\mathbf{E_s}$ and $\mathbf{E_t}$. This can significantly reduce the buckling load. Use of a high yield strength material can produce a marked improvement in buckling resistance and allow more post buckling deformation prior to the onset of permanent deformations. Sharp yielding steels can have a proportional limit almost equal to the yield strength, while gradually yielding steels can have a proportional limit of only 70 percent of yield, or less. The cold working of the steel that occurs in achieving the spherical shape may increase the proportional limit.

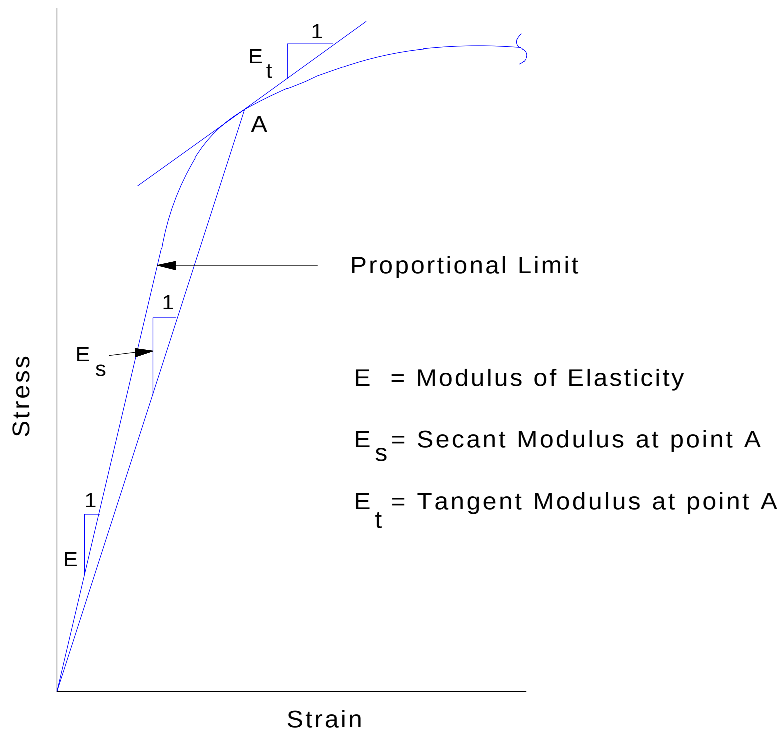

Figure 3.3.3-2 Compression stress strain curve illustrating various moduli

Additional information on shell stability can be found in References 4 and 5. One example of note in Reference 4 is that of a concentrated load on a spherical shell as shown in Figure 3.3.3-1(b). The load **P** will produce an inward dimple at the load as buckling occurs. The following expression approximates the theoretical buckling load for this case.

$$P_{cr} = \left[\frac{Et^3}{a}\right]\left[\sqrt{0.152\,(\gamma + 74.9)} - 2.88\right] \quad \text{for } 20 < \gamma < 100$$

$$P_{cr} = \left[\frac{Et^3}{a}\right]\left[\sqrt{0.093\,(\gamma + 11.5)} - 0.94\right] \quad \text{for } 100 < \gamma < 500$$

Equation 3.3.3-6

$$\text{where } \gamma = \frac{(a_h)^4}{\left(a^2 t^2\right)}$$

Equation 3.3.3-7

a_h = horizontal radius of spherical shell (Figure 3.3.3-1(b))

Bending stresses and membrane stresses produced by a concentrated load or a load over a small circular area are given in Reference 2. Also given are expressions for evaluating the stresses induced in a spherical shell from edge displacements and edge loads.

Design Procedure 3.3-3, located in Section 5.4, incorporates the procedure described in this section. This procedure and others shown in Section 5 are implemented in AISI/CARS.

3.3.4 DENT RESISTANCE

Body panels are typically formed into doubly curved shapes for styling and to achieve the most economical panel. The surface stiffness, the critical oil canning load as well as the denting energy are all considered in the design.

The expressions in this section for stiffness, dent resistance and oil canning load are based on technical papers published in the 1970's and early 1980's, which address empirical or theoretical approaches to a very complex issue. Stiffness, dent resistance and oil canning of body panels are known to be very sensitive to many variables, such as material properties, panel dimensions, panel curvature, edge conditions, sample fixturing and speed of loading. The following discussion is based on point loads, which is representative of small projectiles such as hail damage. However, it is not representative of palm denting. The designer is cautioned that testing is recommended to establish the actual performance of any body panel.[6]

The theoretical stiffness of a spherical shape under a concentrated load is of the form

$$K = \frac{P}{\delta} = \frac{C'Et^2}{a\sqrt{1-\mu^2}}$$

Equation 3.3.4-1

where a = spherical radius
C' = a constant

Since the curvature will seldom be the same in the two orthogonal directions, the expression above is generalized by replacing the curvature ($1/a$) with

$$\frac{1}{a} = \frac{\left(\frac{L_1^2}{R_1}\right) + \left(\frac{L_2^2}{R_2}\right)}{2\,L_1 L_2}$$

Equation 3.3.4-2

where L_1, L_2 = the rectangular panel dimensions as shown in Figure 3.3.4-1 with $L_1 \le L_2$
R_1, R_2 = the panel radii of curvature in the two orthogonal directions

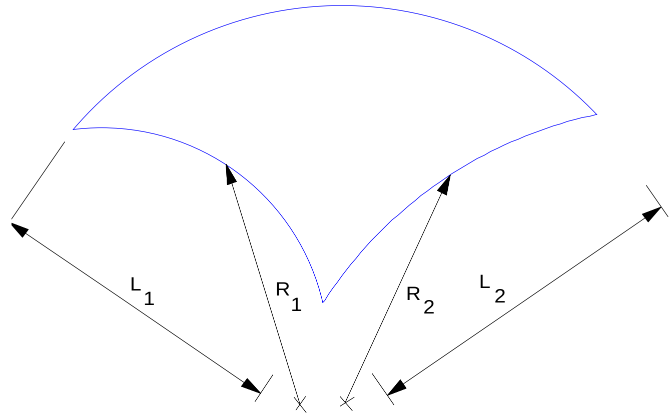

Figure 3.3.4-1 Doubly curved surface

The theoretical shell stiffness expression can be written in the following form[7, 8]

$$K = \frac{9.237\,Et^2\,H_c\,\pi^2}{k\,L_1\,L_2\,\sqrt{1-\mu^2}}$$ **Equation 3.3.4-3**

$$\text{where } H_c = \text{ crown height } = \frac{L_1^2}{8R_1} + \frac{L_2^2}{8R_2}$$ **Equation 3.3.4-4**

$$k = \text{spherical shell factor} = A\alpha^2$$ **Equation 3.3.4-5**

$$\alpha = 2.571\sqrt{\frac{H_c}{t}} \text{ for Poisson's ratio } = 0.3$$ **Equation 3.3.4-6**

A is obtained from Table 3.3.4-1.

$\frac{H_c}{t}$ has an upper limit of 60.

Table 3.3.4-1 Alpha vs. A

Alpha (α)	A	Alpha (α)	A
0	1.000	10	0.069
1	0.996	11	0.053
2	0.935	12	0.044
3	0.754	13	0.037
4	0.506	14	0.032
5	0.321	15	0.028
6	0.210	16	0.025
7	0.148	17	0.022
8	0.111	18	0.019
9	0.085	19	0.017
		20	0.016

A simplified expression for spherical shell factor **k** in steel is:

$$k = 8.06 - 0.088\frac{H_c}{t} \quad \text{for} \quad 4 \le \frac{H_c}{t} < 20$$
$$= 6.3 \quad \text{for} \quad 20 \le \frac{H_c}{t} \le 60$$

Equation 3.3.4-7

Both methods of determining the spherical shell factor, **k**, are plotted as a function of the crown height ratio, H_c/t, in Figure 3.3.4-2.

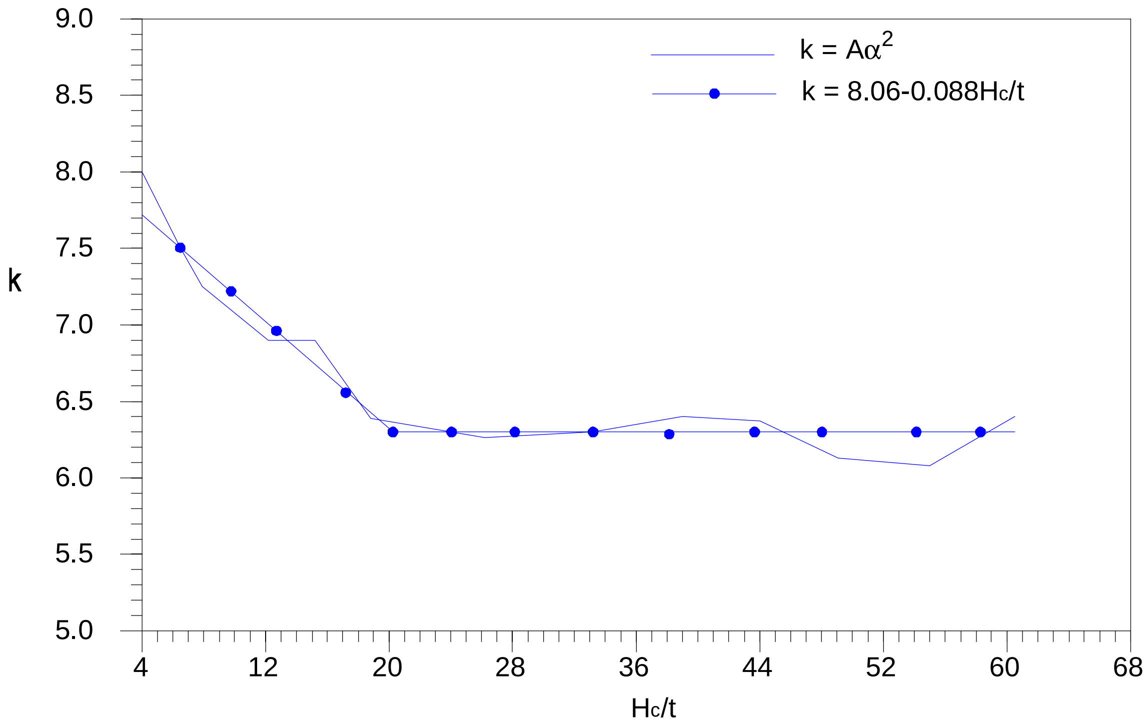

Figure 3.3.4-2 k vs. H_c/t

Alternatively, an empirical expression for curved panel stiffness is

$$K = C_2 t$$

Equation 3.3.4-8

where $\mathbf{C_2}$ is a function of panel geometry which was derived using steel panels (Reference 7). The designer should determine the minimum stiffness required.[9]

The denting resistance is evaluated in terms of minimum denting energy. The designer should also determine the minimum denting resistance required. It has been shown (Reference 10) that the dent resistance is proportional to $\mathbf{F_{yd}t^2}$ if $\mathbf{F_{yd}}$ is evaluated at the appropriate dynamic strain rate rather than being taken as the typical "static" value obtained from a test machine.

The dynamic strain rate corresponding to denting (10 to 100 per second) is many orders of magnitude higher than the typical static tensile test strain rate (0.001 per second). The ratio of dynamic to static yield strength of steels at strain rates of up to 100/sec are shown in Figure 3.3.4-3, based on the data given in Reference 11. It is up to the designer to select an appropriate factor from Figure 3.3.4-3 to evaluate $\mathbf{F_{yd}}$ or to use an appropriate value of $\mathbf{F_{yd}}$ obtained from another source. The influence of strain rate on yield strength is also discussed in Section 2.13.

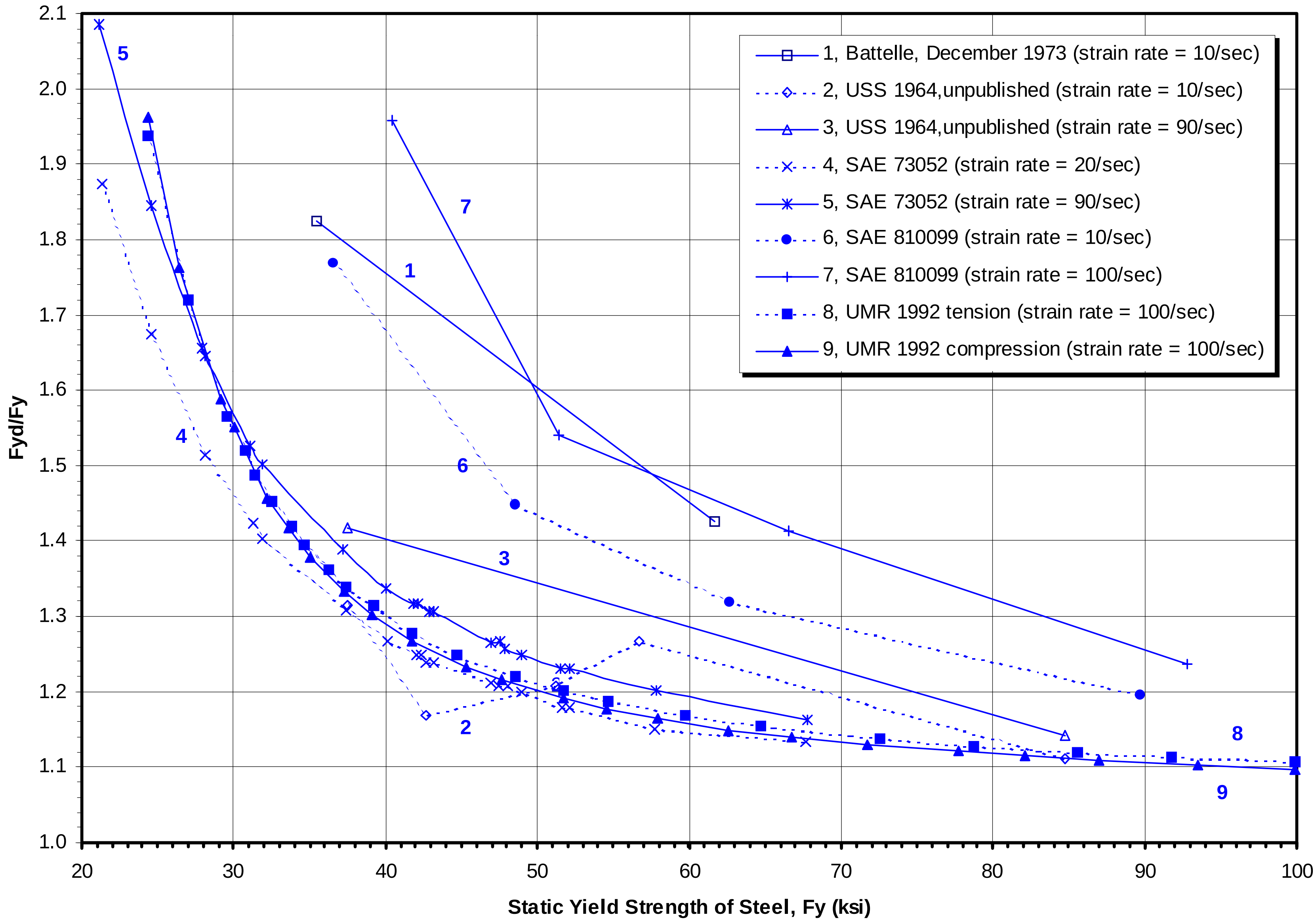

Figure 3.3.4-3 Fyd/Fy versus Fy for steel

Grade 35XF, 50XF and 100XF materials were tested at strain rates of 0.0001, 0.01 and 1.0 per second in a research program performed at the University of Missouri-Rolla.[11, 12, 13] Both tension and compression tests were performed. A method was developed to predict the yield and ultimate strengths of these steels for strain rates up to 1000 per second.

The denting energy based on an empirical curve fit (References 7 and 8) can be expressed as

$$W = 56.8 \frac{\left(F_{yd} t^2\right)^2}{K}$$

Equation 3.3.4-9

where K = panel stiffness per Equation 3.3.4-3
t = panel thickness
F_{yd} = yield strength of the panel at high strain rate

This formula defines a dent as 0.001 inch (0.025 mm) permanent deformation in the panel. The stiffness often tends to control over the dent resistance except when the panel size is small.

Oil canning load often controls panel designs over both stiffness and dent resistance. Of concern are the degree of oil canning, which determines the likelihood of a panel buckling measured by the parameter $\boldsymbol{\lambda}$, and the critical buckling load $\mathbf{P_{cr}}$ at which the panel would collapse and reverse its curvature. The critical oil canning load (References 9 and 14) is given as

$$P_{cr} = \frac{C R_{cr} \pi^2 E t^4}{L_1 L_2 \left(1-\mu^2\right)}$$

Equation 3.3.4-10

$$\text{where } C = 0.645 - 7.75\text{x}10^{-7} L_1 L_2 \text{ in millimeter units}$$
$$= 0.645 - 0.0005\, L_1 L_2 \text{ in inch units}$$

Equation 3.3.4-11

$$R_{cr} = 45.929 - 34.183\lambda + 6.397\lambda^2$$

Equation 3.3.4-12

$$\text{with } \lambda = 0.5 \sqrt{\frac{L_1 L_2}{t} \sqrt{\frac{12\left(1-\mu^2\right)}{R_1 R_2}}}$$

Equation 3.3.4-13

This expression is valid for

$$\frac{R_1}{L_1} \text{ and } \frac{R_2}{L_2} > 2$$

$$\frac{1}{3} < \frac{L_1}{L_2} < 3$$

$$L_1 L_2 < 1200 \text{ in}^2 \left(0.774 \text{ m}^2\right)$$

The designer should determine the minimum critical oil canning load required. Since oil canning is controlled by geometry and modulus of elasticity, it appears that there is no advantage in using high strength steels. However, high strength is necessary to maintain the elastic behavior under large deformations so that an elastic recovery is more likely to occur.

Example 3.3-1 shown in Section 6.2.3.1 illustrates the use of the equations for stiffness, denting, and oil canning in the design of a body panel.

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3.4 CONNECTIONS

There are four basic systems that are commonly used to join sheet steel body members with other members: welding, mechanical fasteners, mechanical fastening by deformation of the parent metal and adhesive bonding. Each system offers advantages and disadvantages, which are discussed in Sections 3.4.1 through Section 3.4.6. In some cases, the systems are combined to optimize the performance of the joint, such as weldbonding, which is discussed in Section 3.4.7.

The following checklist will assist in identifying the factors to be weighed in selecting the optimum system.

1. Assess the Application and Operating Environment.

   1.1 Determine the type and intensity of applied load.

   Type of load is essential; typical types are long-term or continuous, occasional, fatigue, and impact (high strain rate).

   Sources may be anticipated or accidental. Anticipated loads, which are usually quantified, include those computed from suspension G loads; low-speed bumper impacts; rough road tests; and door, hood and deck lid slam.

   Accidental loads, which are not readily quantified, include high speed collision, rollover, and low speed (parking lot) collisions.

   1.2 Assess potential problem areas

   Potential problem areas may include long term structural degradation, potential for rattles, galvanic and atmospheric corrosion, differential rates of thermal expansion, and repairability.

   1.3 Design the joint

   Select the system or combination that will meet the design requirements.

   Evaluate the shape and size of the contact areas of the members to be joined and the relationship between the type and direction of load and the joint configuration.

   1.4 Determine the side benefits

   Examine the side benefits that may be derived, such as galvanic separation, moisture barrier, and effects of load distribution that may eliminate or require local reinforcements and stiffeners on all affected members.

   1.5 Review the cosmetic requirements

   Witness marks are generally prohibitive on appearance surfaces, but may be tolerable on other surfaces.