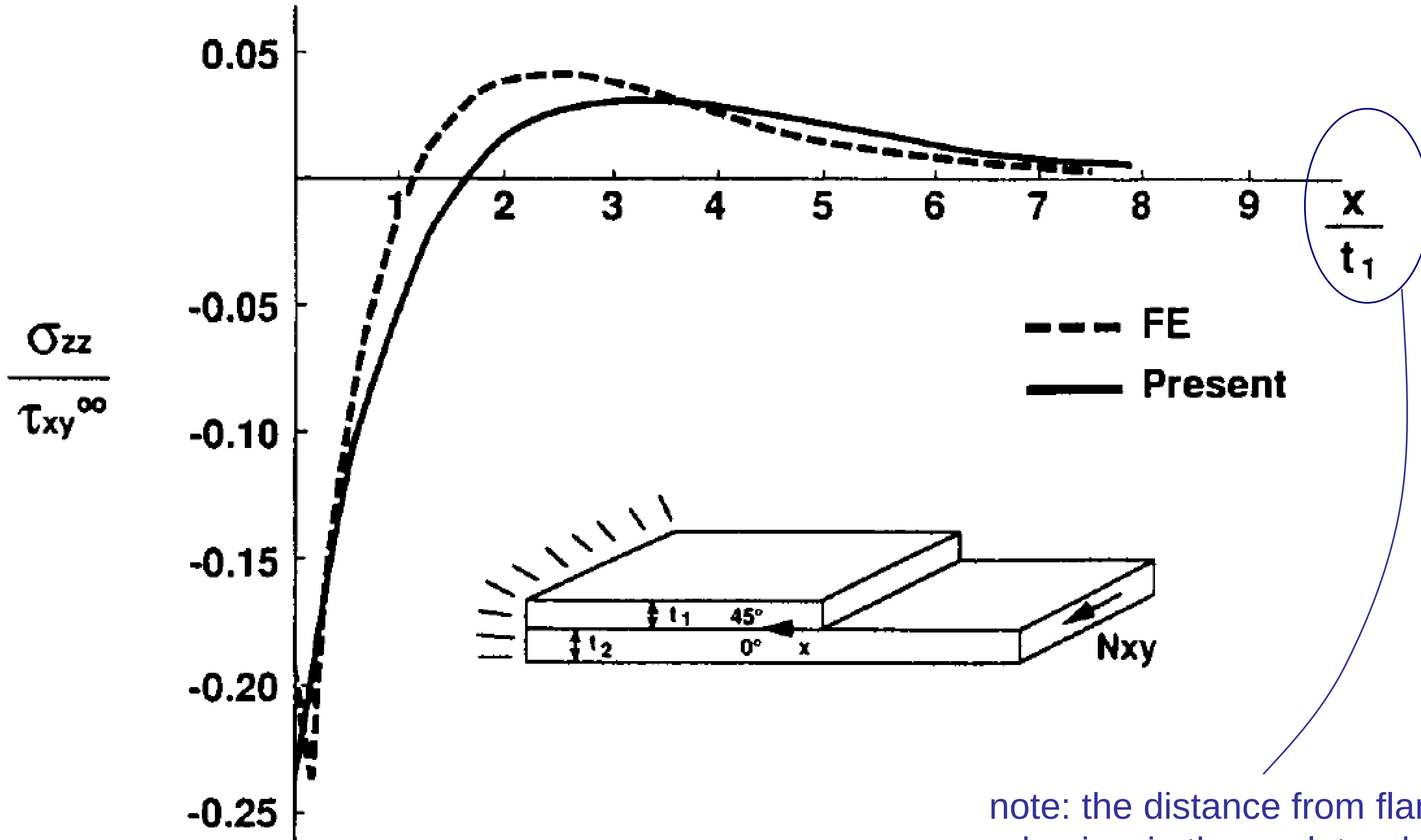
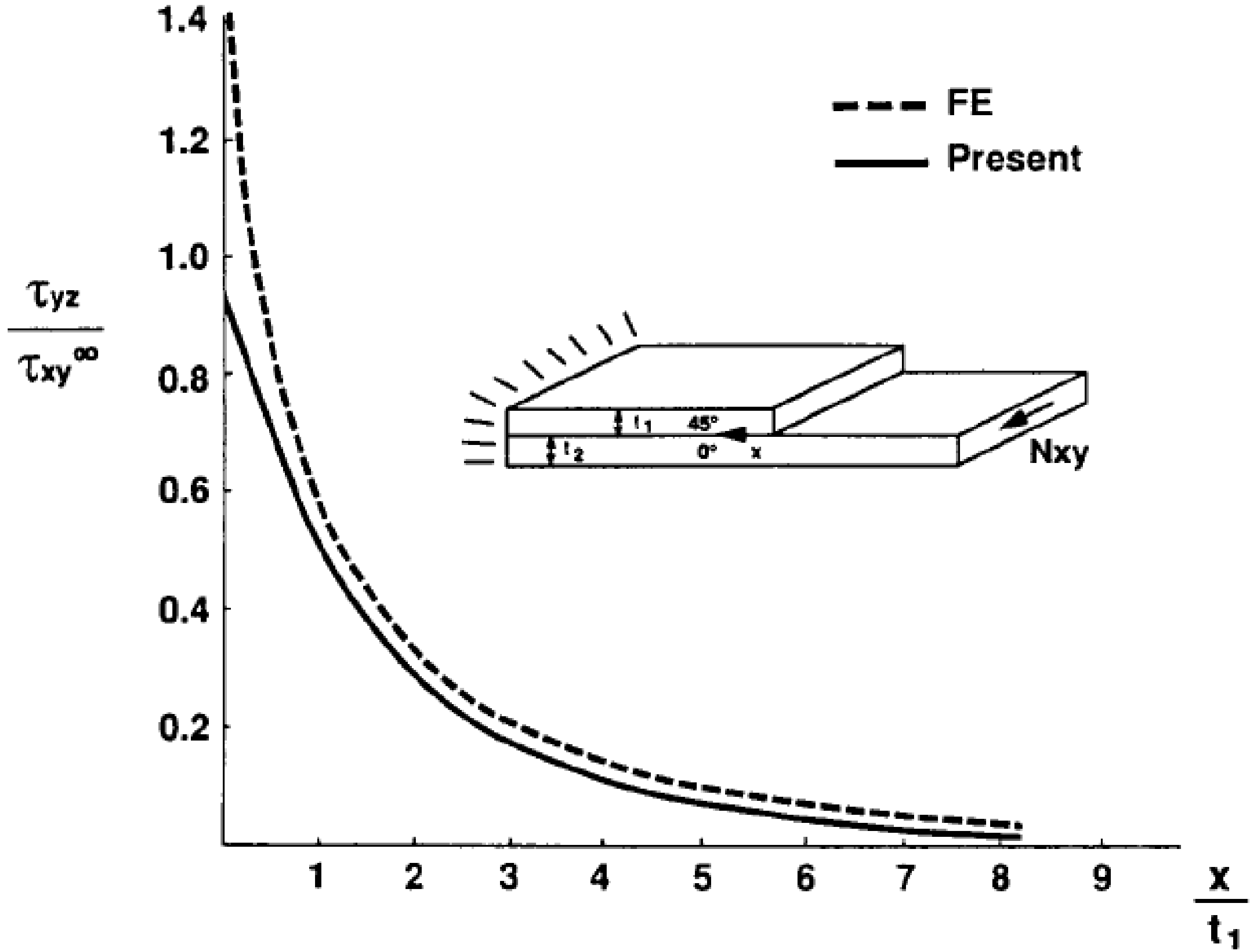
Comparison of solution to FE

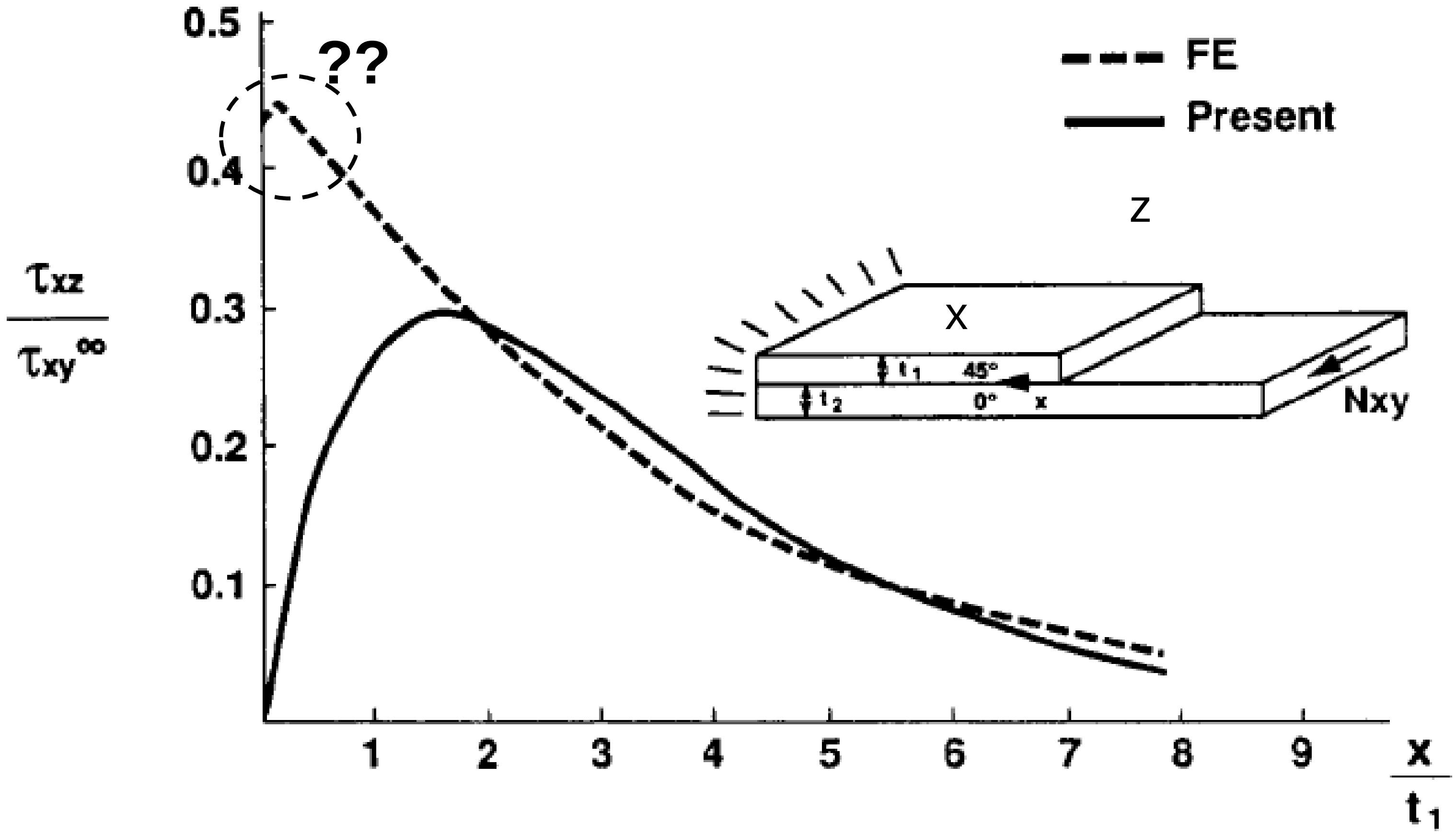


note: the distance from flange edge is x in these plots while it was y in the derivation !!!

Comparison of solution to FE



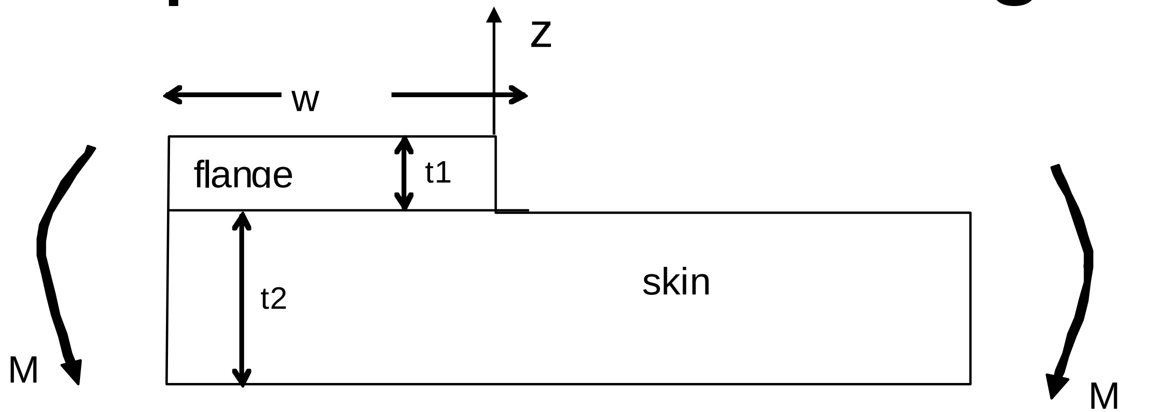
Comparison of solution to FE



More elaborate solutions to the skin-stiffener separation problem

- Cohen, D. and Hyer, M.W., "Calculation of skin-stiffener interface stresses in stiffened composite panels" NASA CR 184682, 1987

Skin-stiffener separation: Implications for design



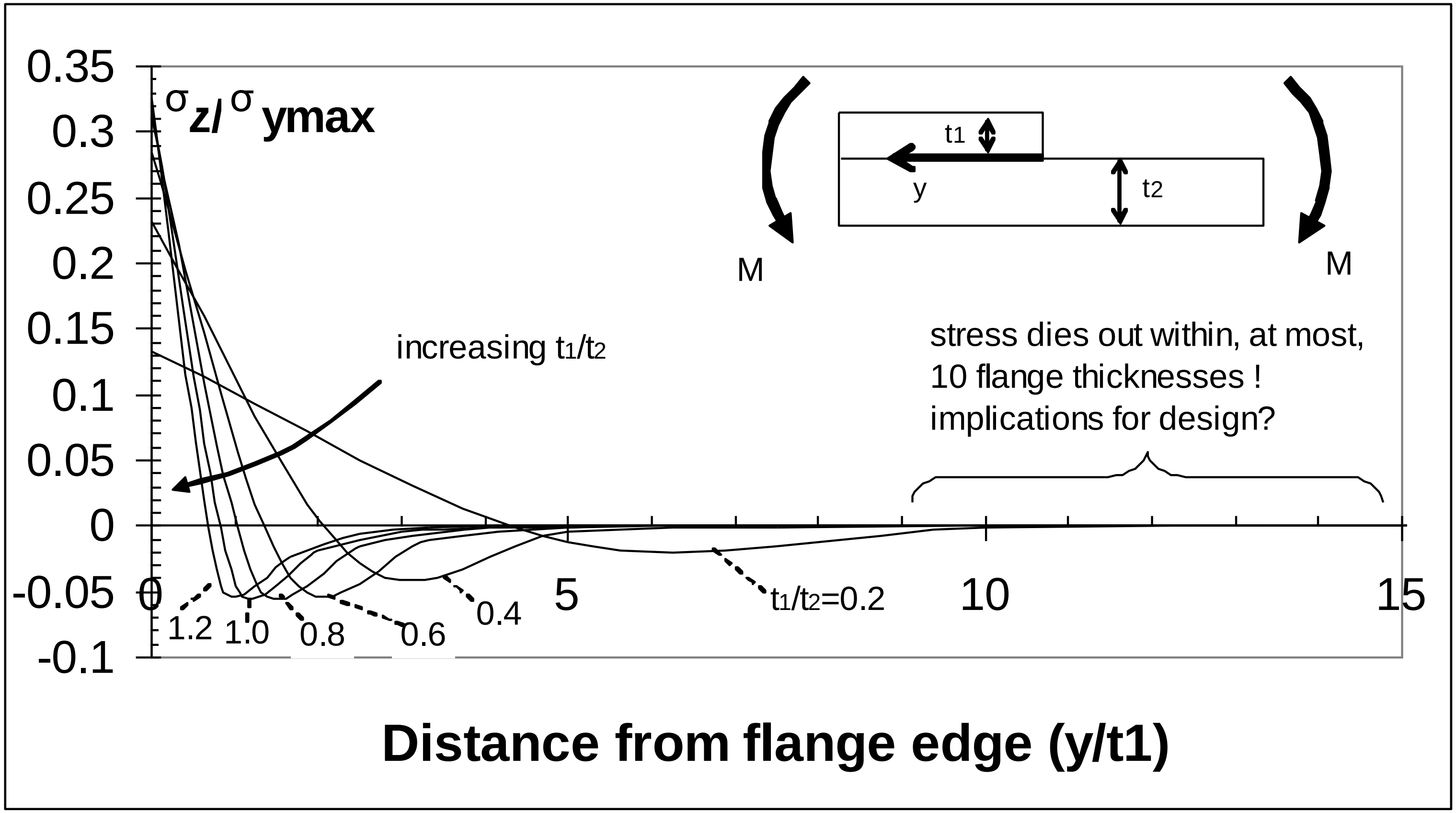
results apply to other load cases as well!

Material properties:

Ex=137.9 GPa	Ez=11.03 GPa
Ey=11.03 GPa	Gxz=4.826 GPa
Gxy=4.826 GPa	Gyz=3.447 GPa
v_{xy}=0.29	vxz=0.29
tply=0.152 mm	vyz=0.4

Effect of flange/skin thickness ratio

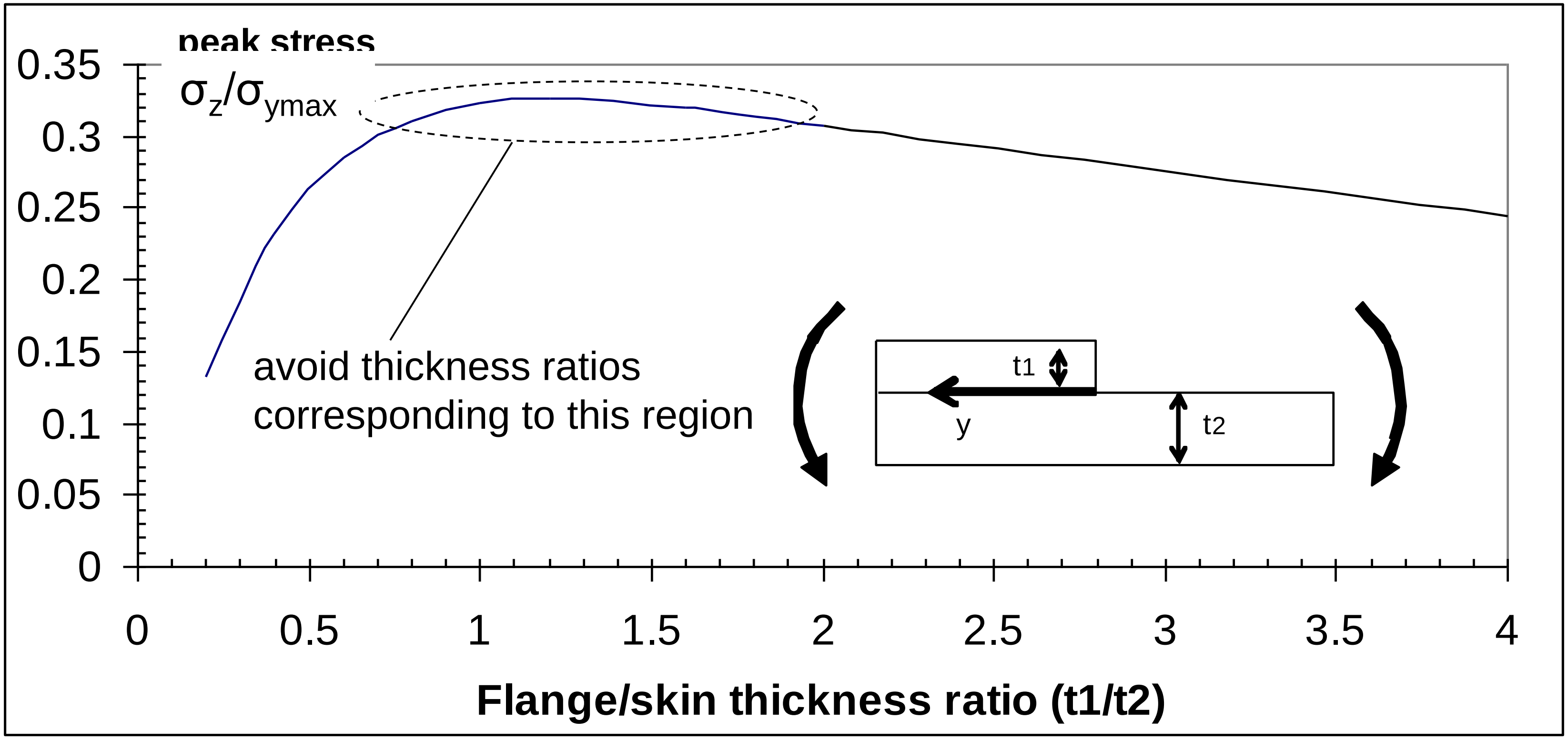
skin and flange have **same** layup: [45/-45/-45/45]n



- stresses die out within 10 (or less) flange thicknesses

- as flange thickness increases, peak stress increases (BUT see next Figure)

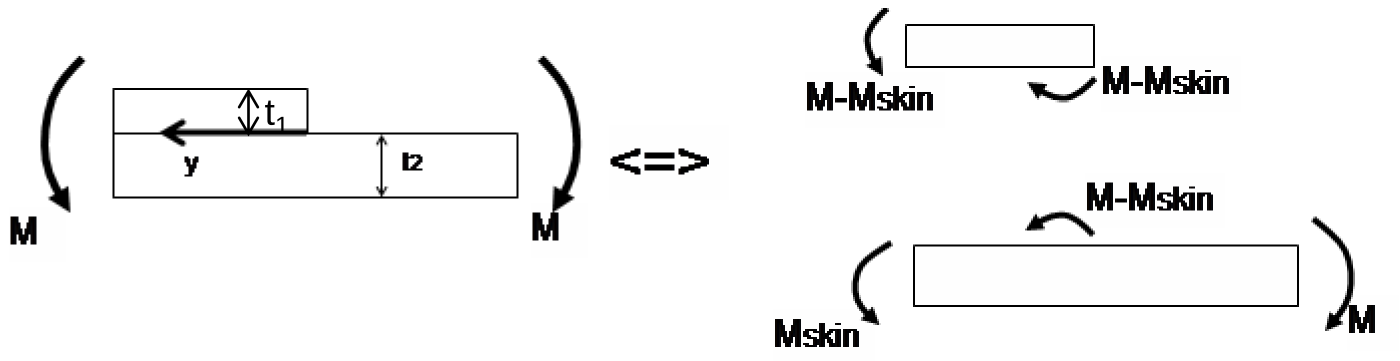
Effect of flange/skin thickness ratio on peak interlaminar stress



- Flange thickness should **not** be close to skin thickness!!

Why should we have $t_1 \neq t_2$?

- free-body diagram of structure



- the interlaminar normal stress σ_z gives rise to $M\text{-}M_{skin}$
- the higher $M\text{-}M_{skin}$ the higher the force couple created by σ_z and the higher the peak value of σ_z

Why should we have $t_1 \neq t_2$?

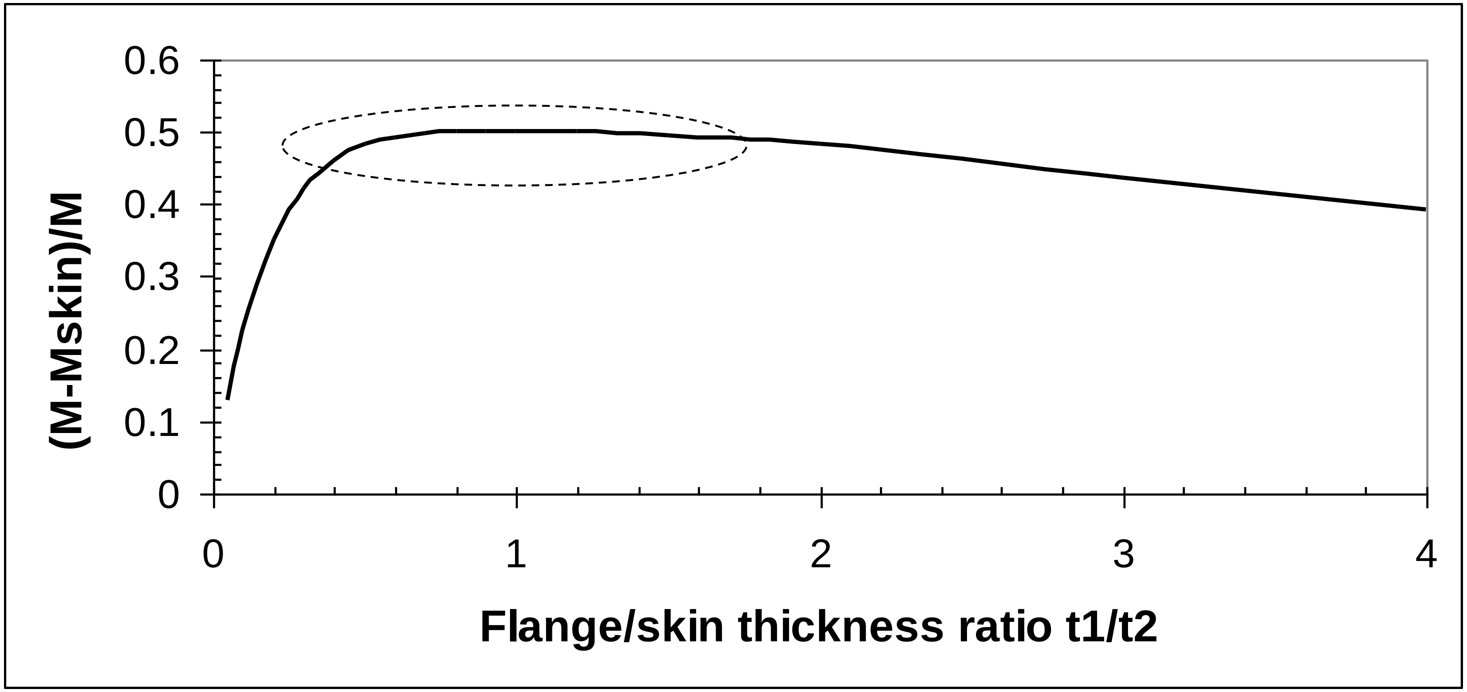
- from beam theory,

$$M - M_{skin} = M\left(1 - \frac{\frac{t_1}{t_2}\left(\left(\frac{t_1}{t_2}\right)^2 + 3\right)}{\left(1 + \frac{t_1}{t_2}\right)^3}\right) \quad for \quad t_2 < t_1$$

$$= M\left(1 - \frac{\left(3\left(\frac{t_1}{t_2}\right)^2 + 1\right)}{\left(1 + \frac{t_1}{t_2}\right)^3}\right) \quad for \quad t_2 > t_1 \qquad (5.4.3.36)$$

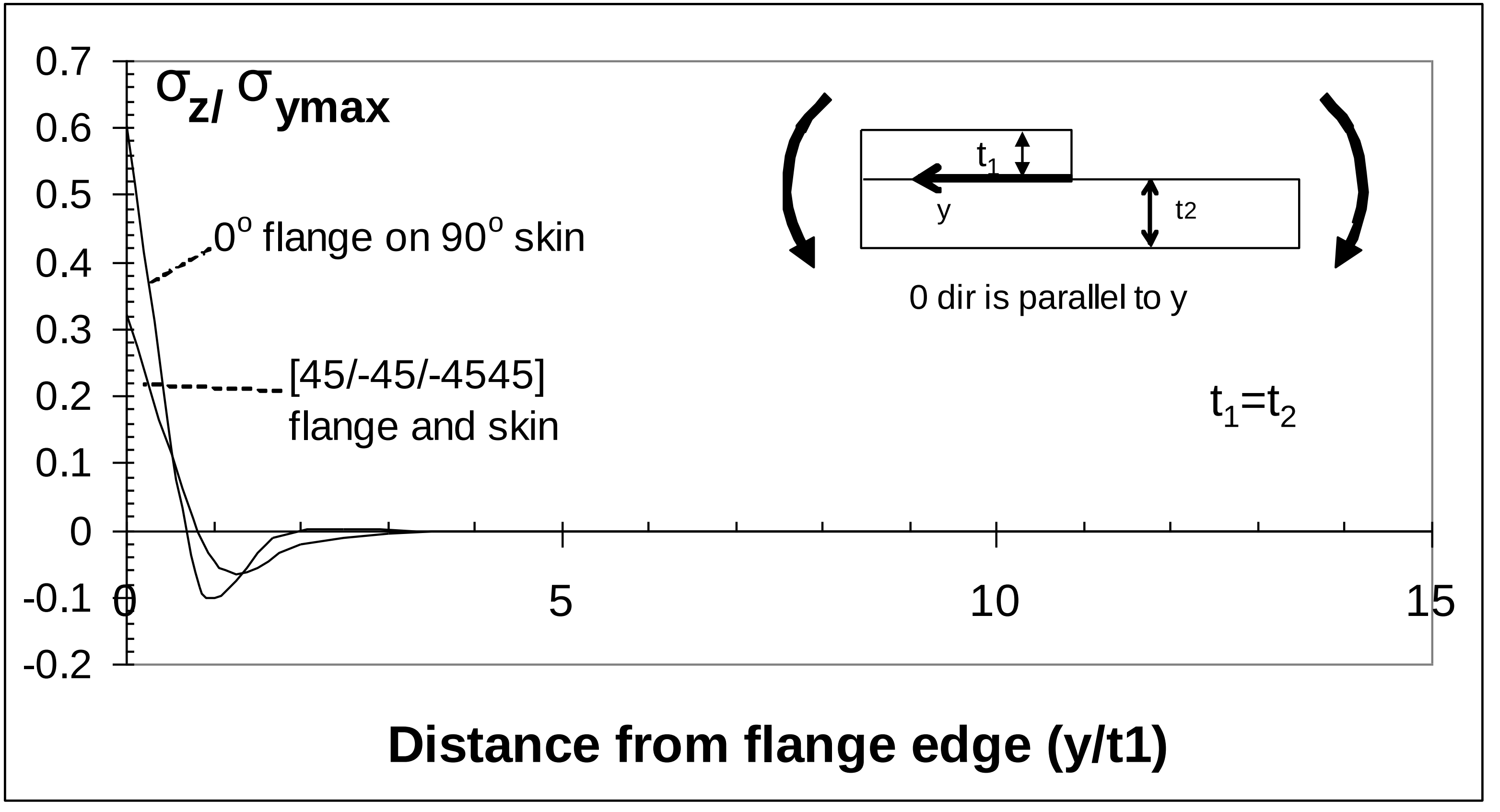
Why should we have $t_1 \neq t_2$?

- plot $M\text{-}M_{skin}$ as a function of t_1/t_2



- M-Mskin is maximized when $t_1 \approx t_2$!!

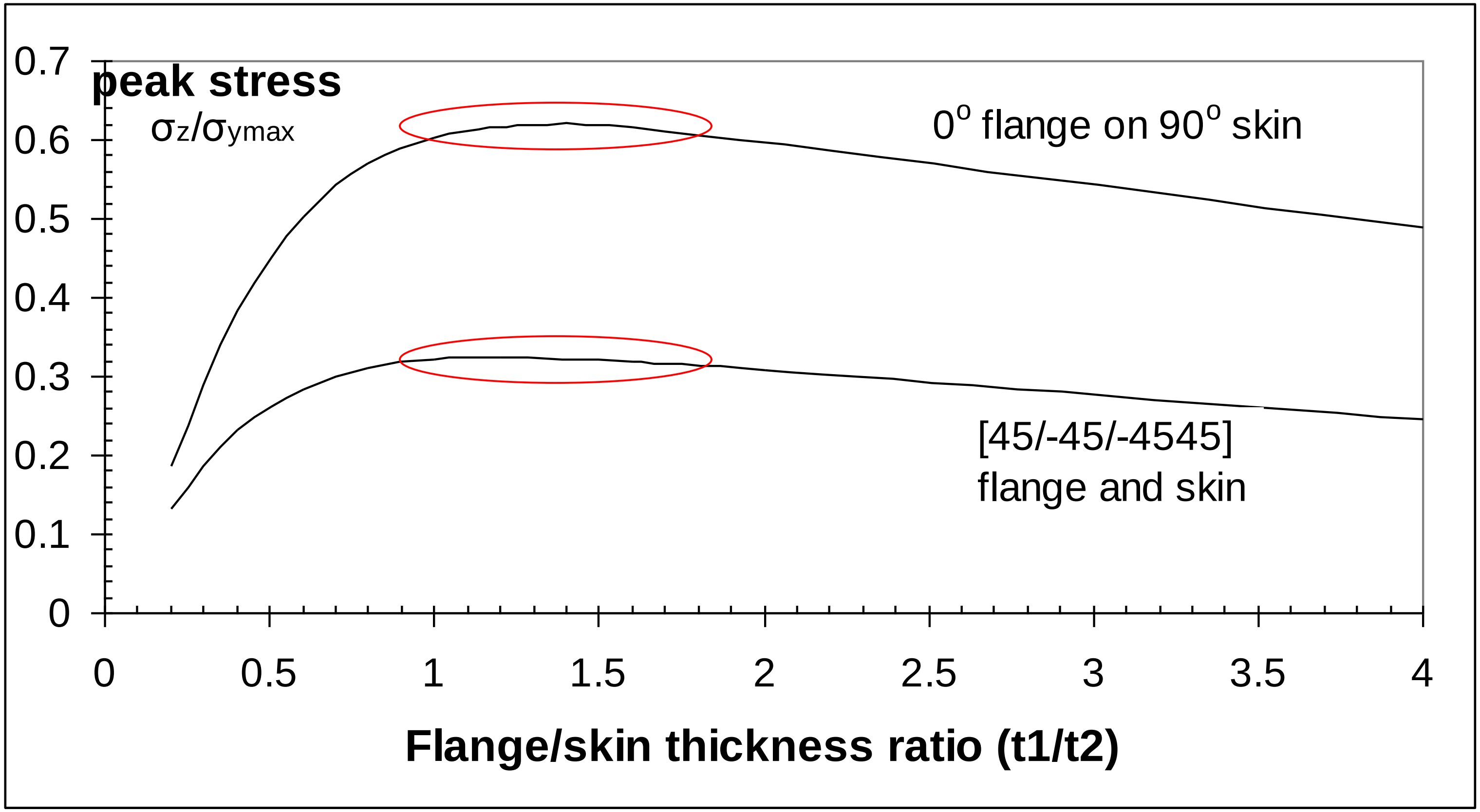
Effect of layup on skin-stiffener separation stresses



• 0° flange on 90° skin is worst mismatch: peak stress is twice the value of [45/-45/-45/45] flange and skin

• the higher the peak stress the faster the rate of decay (why?)

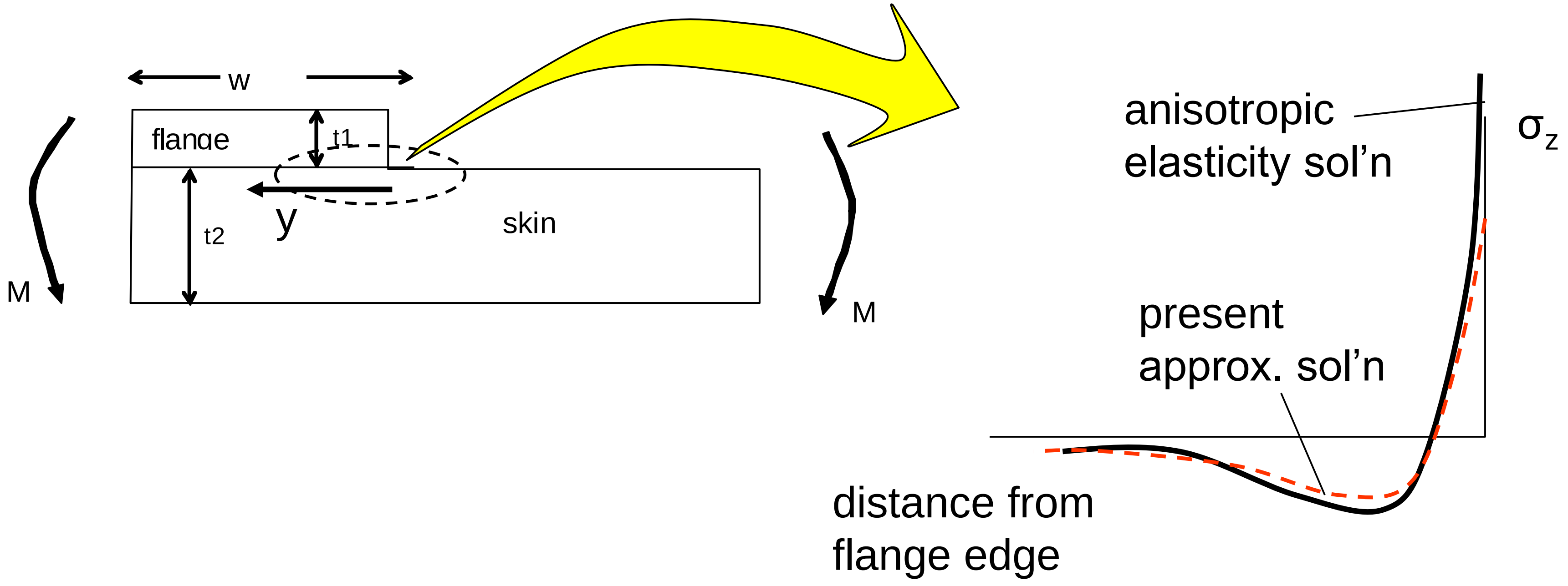
Effect of stiffness mismatch



- the stiffness mismatch almost doubles the interlaminar stresses
- roughly equal flange and skin thicknesses must still be avoided

Brief discussion on stress singularity

- comparisons based on the peak stress can be misleading
- the exact value at the flange edge is unknown; a full 3-D anisotropic elasticity solution would predict the stress is singular there



Brief discussion on stress singularity

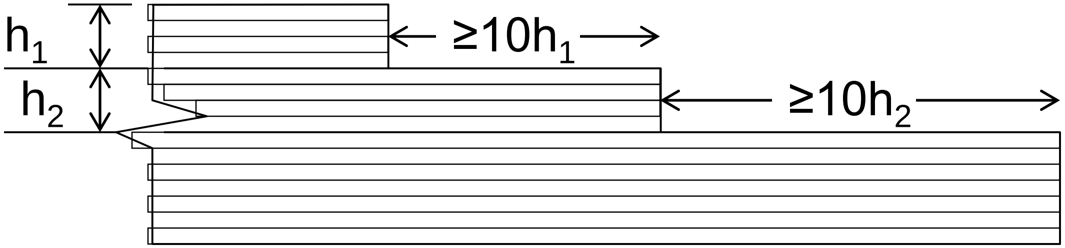
• however, the elasticity solution is based on assuming homogeneous plies (matrix and fiber are not explicitly accounted for)

• the strength of the singularity is weak (logarithmic or less) implying that the singularity is significant over distances from the flange edge that are of the order of a few fiber diameters **where the homogeneity assumption breaks down anyway;** so singularity is of no consequence in design (but still need to decide what to do with peak stresses calculated)

• can combine the peak stress or some average stress or stress at a distance with an onset of delamination criterion

Skin/stiffener separation – Summary of findings

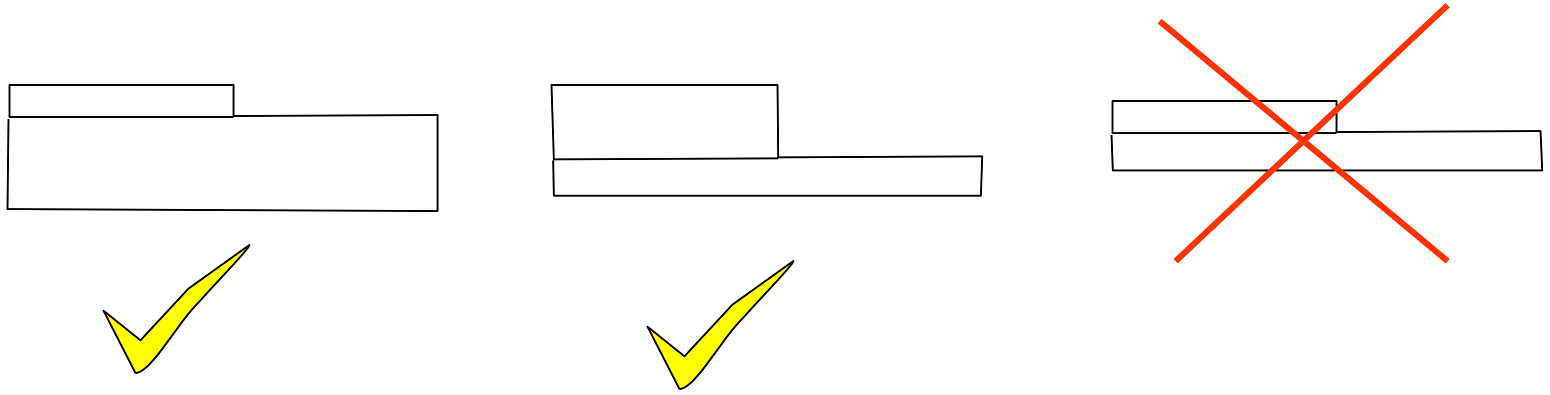
- two factors tend to make the interlaminar stresses worse:
  - flange thickness; in general, the higher it is the higher the interlaminar stresses
  - stiffness mismatch; the higher it is the higher the interlaminar stresses
- interlaminar stresses die out within ~10 flange thicknesses (important for successive or staggered plydrops):



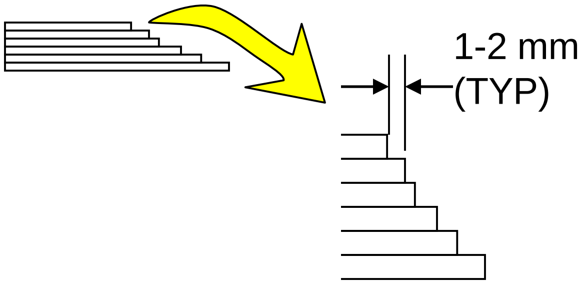
(similar conditions are used for internal plydrops)

Skin/stiffener separation – Summary of findings

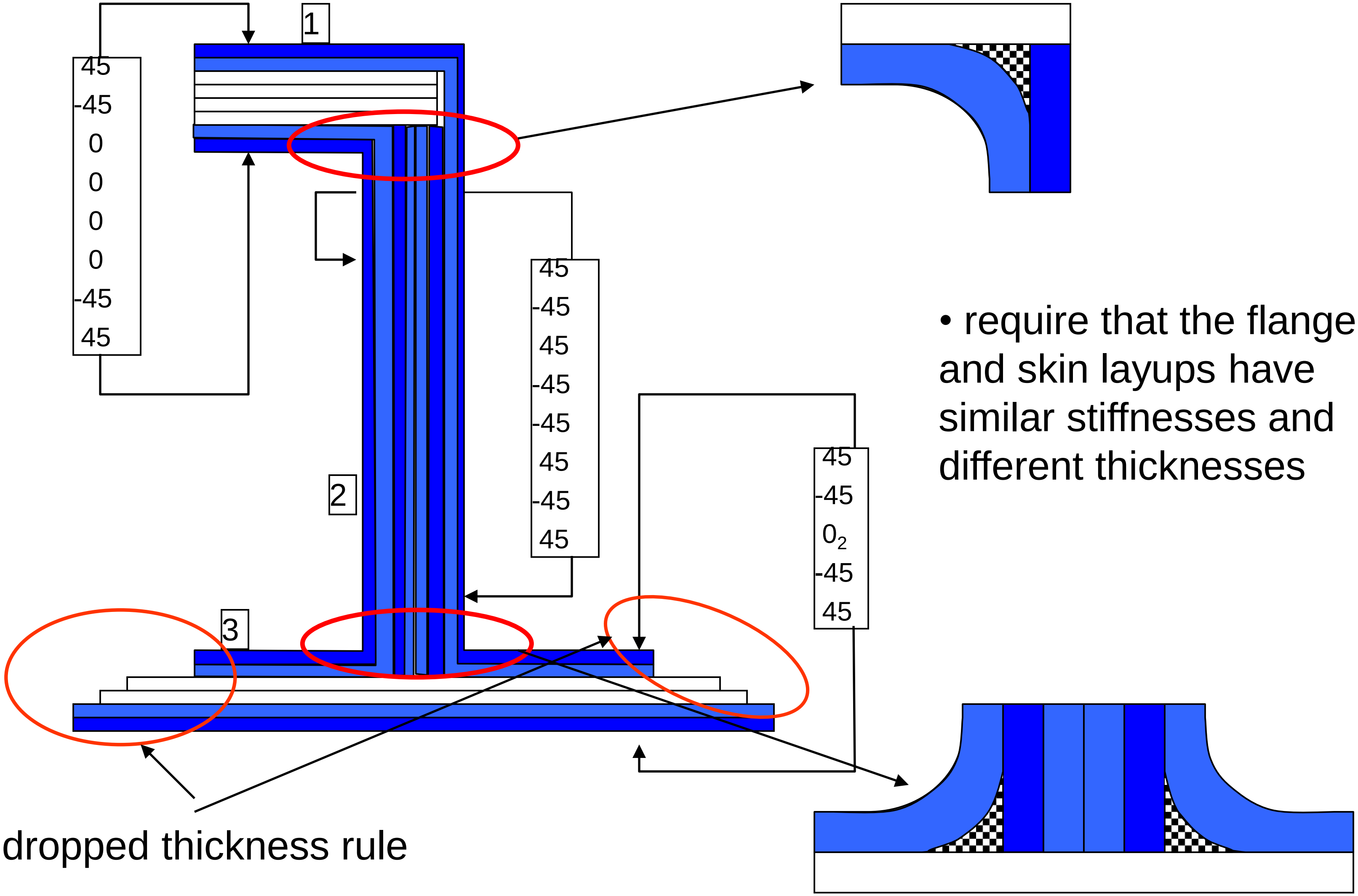
• in general, the flange and skin thickness should not be close to one another



• one final note: if one ply is dropped, 10(tply) is only 1-2 mm. Placing/dropping with such accuracy in production is difficult (and expensive)



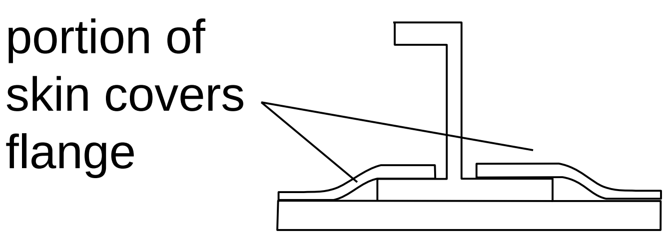
Revisiting the stiffener cross-section we have been considering all along



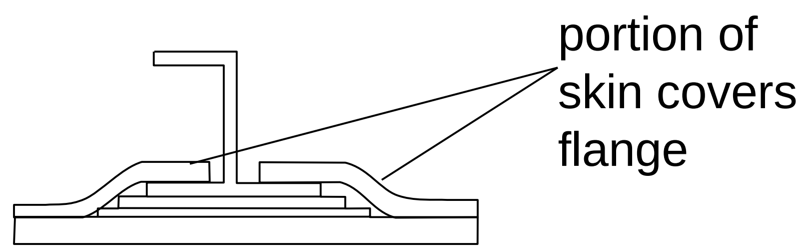
- require that the flange and skin layups have similar stiffnesses and different thicknesses

10 x dropped thickness rule

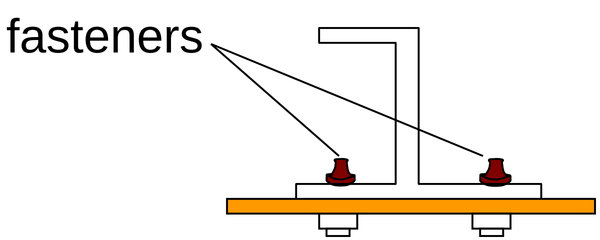
Other options for delaying skin/stiffener separation



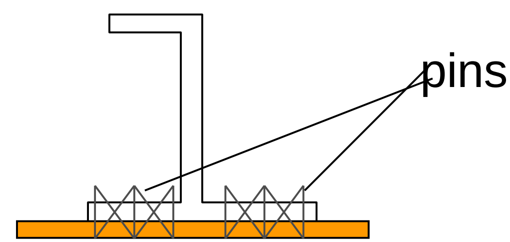
expensive!



expensive!

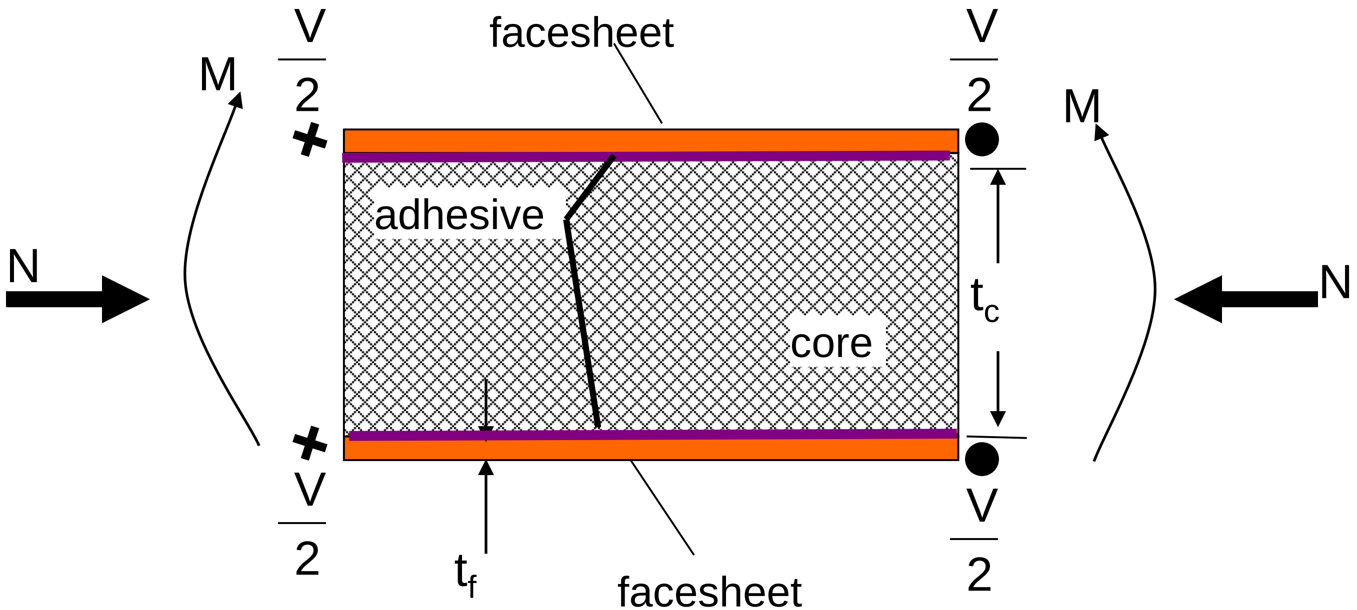


expensive!



expensive!

Sandwich Structure



See: Plantema, F.J., *Sandwich Construction*, John Wiley & Sons, NY, 1966

Sandwich structure components

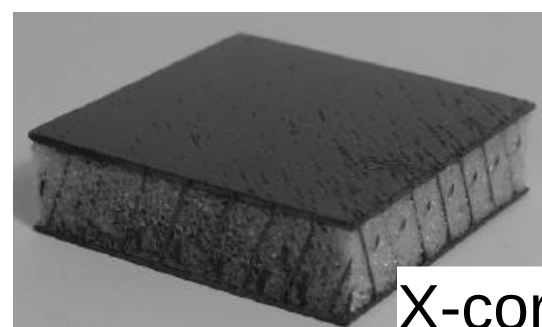
• facesheet: any load-carrying composite material

– preferred to have the outer-most ply as fabric to minimize damage extent due to impact

– layup of each facesheet does not have to be symmetric (even though it would be preferred) as long as the entire sandwich layup is symmetric

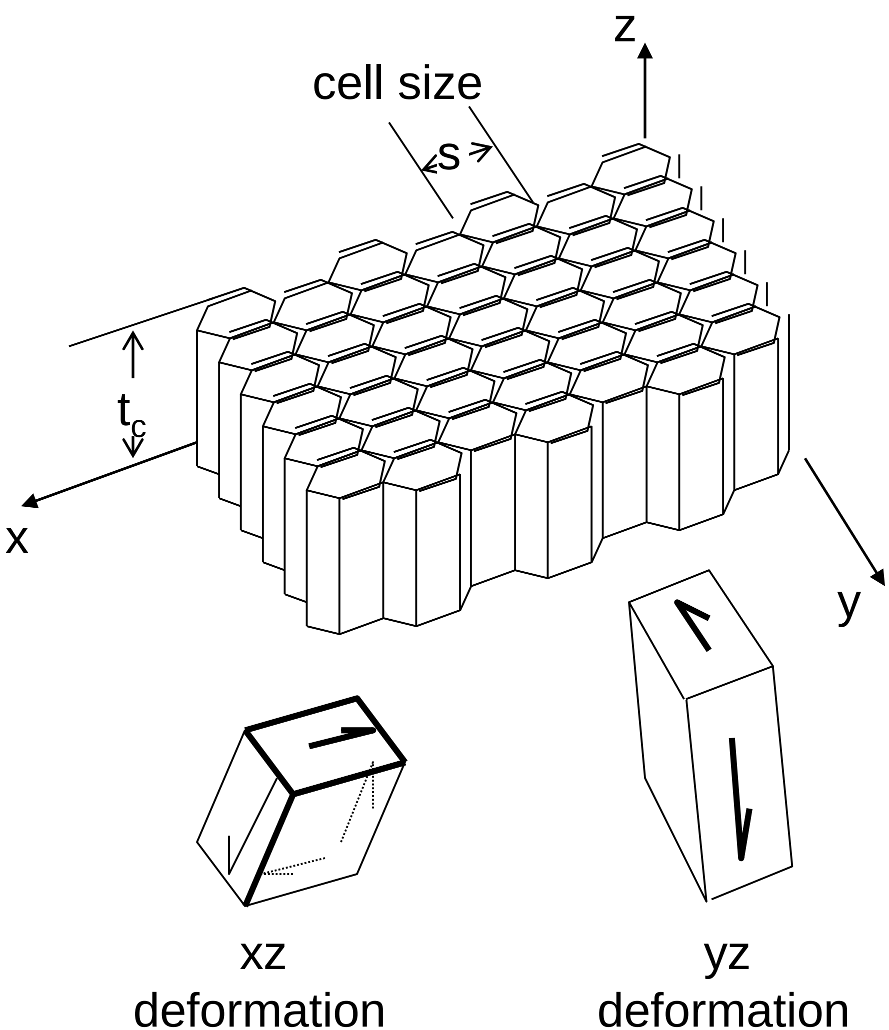
• core: honeycomb, foam, pins (X-cor™, K-cor™)…; required to have "sufficient" flatwise strength and stiffness and two transverse shear strengths and stiffnesses

• adhesive: film adhesive, at least 0.08 mm thick; in some cases (e.g. X-cor™) it can be omitted



X-cor

Honeycomb Core properties of importance



- transverse shear moduli, G_{xz}, G_{yz}
- transverse shear strengths F_{xz}, F_{yz}
- out-of-plane stiffness E_z
- out-of-plane tension and compression strengths F_z^t, F_z^c

Sandwich Structure

- by moving material away from the neutral axes, core increases drastically the bending stiffness of the structure=> increase in buckling load

- at the same time, for cores thicker than 6-7 mm, transverse shear effects become significant

- sandwich bending stiffness:

$$D_{ij} = 2(D_{ij})_f + 2(A_{ij})_f \left(\frac{t_c + t_f}{2} \right)^2 \qquad (5.5.1)$$

subscript “f” denotes single facesheet

of course, one can also get Dij by defining entire laminate with core stiffness values=0

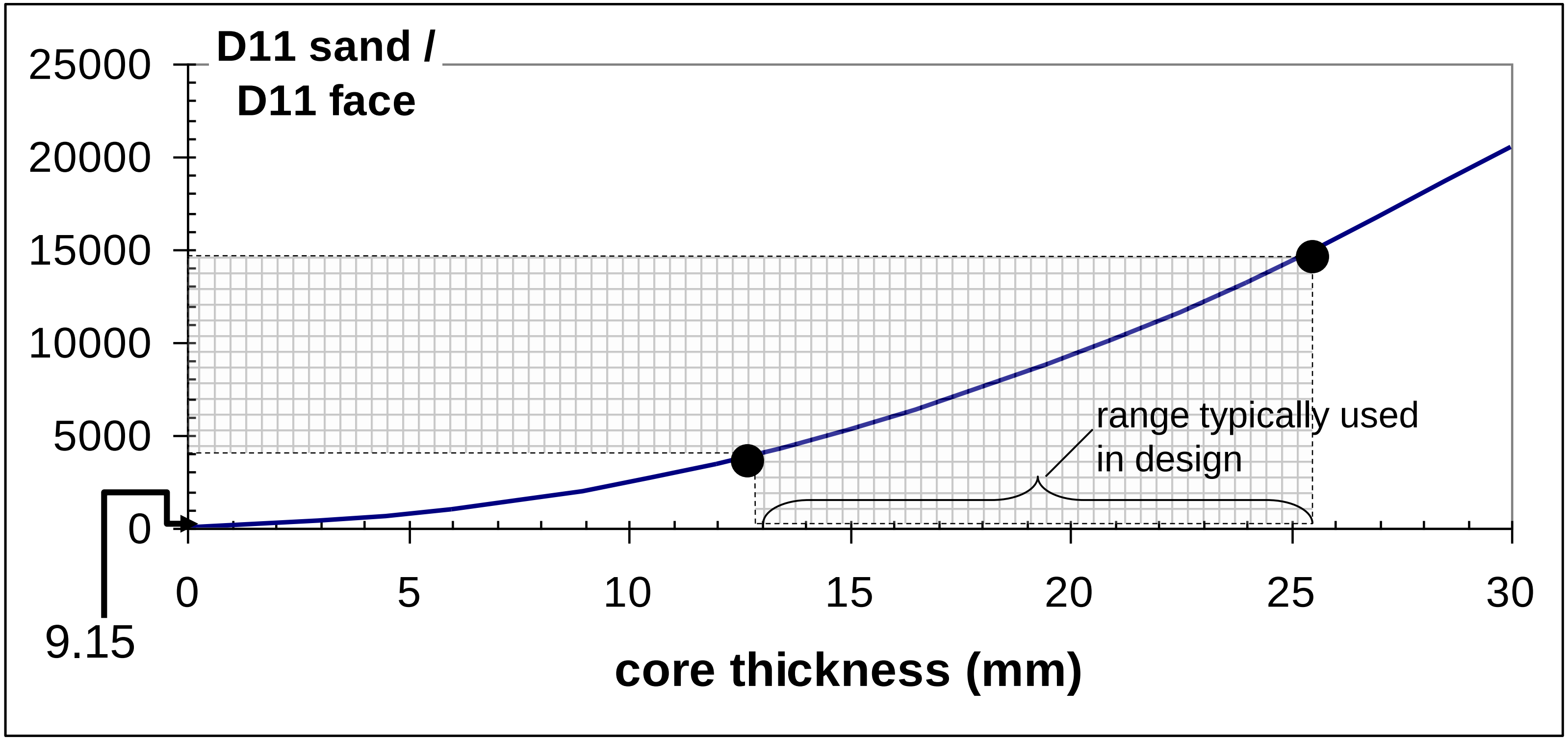
Effect of core thickness on bending stiffness of sandwich

- example we saw before; facesheet: (±45)/(0/90)/(±45)
- facesheet thickness: 0.5715 mm

A_{11}	28912.44	N/mm
A_{12}	12491.43	N/mm
A_{22}	28912.44	N/mm
A_{66}	13468.58	N/mm

D_{11}	659.7	Nmm
D_{12}	466.9	Nmm
D_{22}	659.7	Nmm
D_{66}	494.0	Nmm

Effect of core thickness on bending stiffness of sandwich

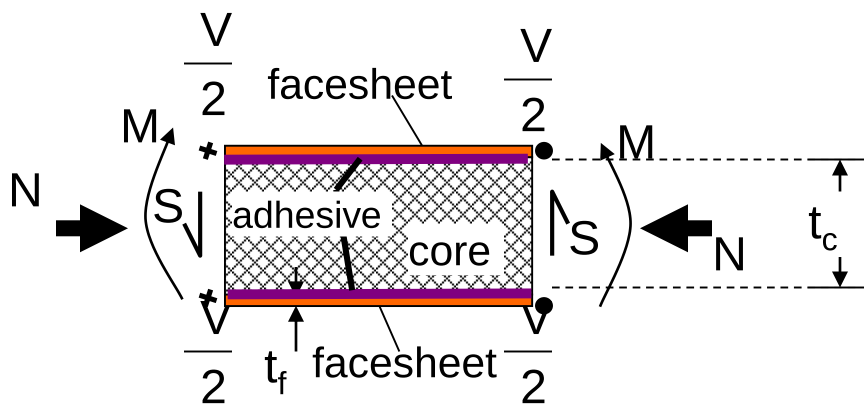


Everything comes at a price…

• the huge increase in bending stiffness and buckling load would make the sandwich the ideal structural element, BUT

- – more failure modes must be accounted for; every component (facesheet, adhesive, core) can fail and in more than one ways
- – transitioning to adjacent structure (rampdowns) is not easy
- – attachments usually require inserts which can be expensive
- – susceptibility to moisture absorption and freeze-thaw cycles

Standard practice for analysis



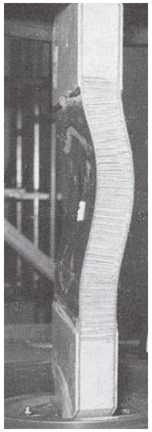
- all loads shown are taken by the facesheet except transverse shear S which is taken by the core
- moments are resolved as force couples:

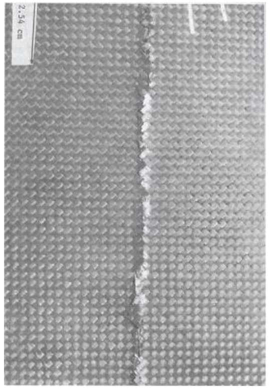
$$F_{face} = \frac{M}{t_c + t_f} \quad (5.5.2.1)$$

Exceptions to the rule

- the assumption that all loads (except for transverse shear) are carried by the facesheet is valid as long as the core is very soft, which is typical of most applications; if the core used has stiffness comparable to that of the facesheet (e.g. solid aluminum or carbon core) the assumption is not valid

Failure analysis

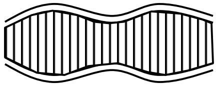




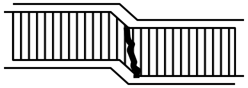
• panel buckling (buckling of sandwich panel as a whole)

• facesheet strength failure (tension, compression, shear)

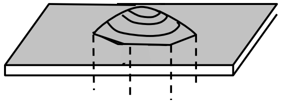
• facesheet wrinkling (local buckling of facesheet on elastic foundation)



• shear crimping (precipitated by core shear failure usually after facesheet antisymmetric wrinkling)



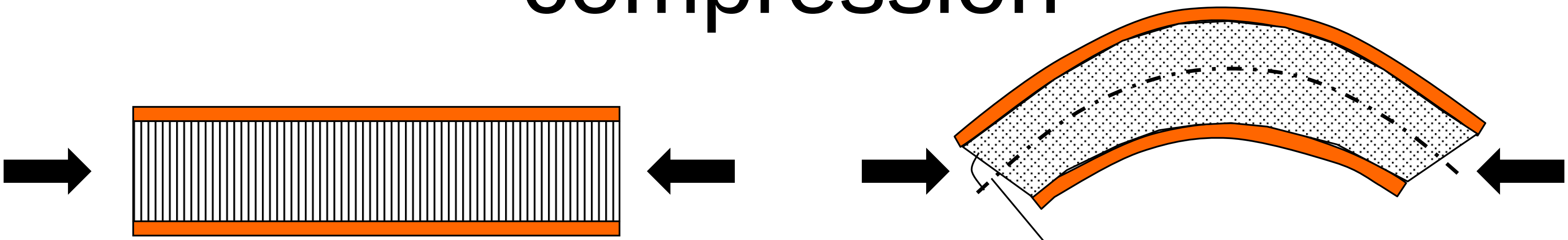
• facesheet dimpling or intra-cellular buckling (facesheet buckling between cell boundaries)



• adhesive strength failure (tension, shear)

• core strength failure (tension, compression, shear)

Sandwich panel buckling - compression



• **unless the core is very thin, transverse shear effects are important!** Plane sections remain plane but are no longer perpendicular to the mid-plane

• treat the sandwich as a wide column; then from [1], the buckling load (per unit width) for an isotropic beam is given by

$$N_{crit} = \frac{N_{Ecrit}}{1 + \frac{k N_{Ecrit}}{t_c G_c}} \qquad (5.5.3.1.1)$$

N_{Ecrit}=Buckling load without transv shear effects

k=shear correction factor

G_c=(core) shear modulus (in dir of loading)

t_c=(core) thickness

1. Timoshenko, SP, and Gere, JM, *Theory of Elastic Stability*, McGraw-Hill, NY, 1961, p 351…

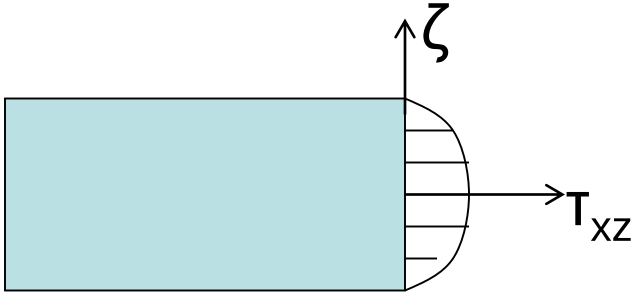
Note on shear correction factor k

• inconsistency between derived and assumed through-the thickness strain distributions

• derived:

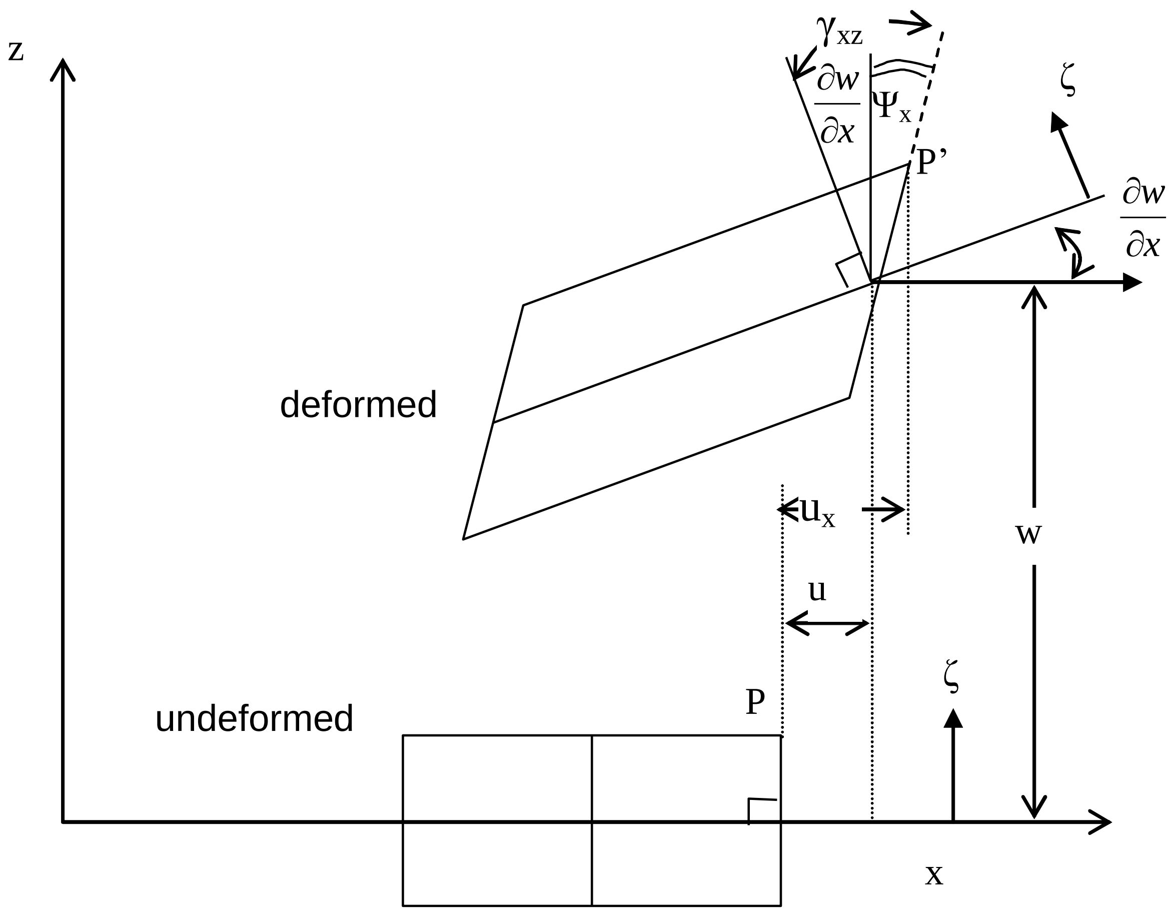
$$\tau_{xz} = \frac{3}{2}\frac{Q_x}{h}\left[1-\left(\frac{\zeta}{h/2}\right)^2\right]$$

quadratic in ζ => strain γ_{xz} is also quadratic in ζ



Note on shear correction factor k

- assumed in first order shear deformation theory:

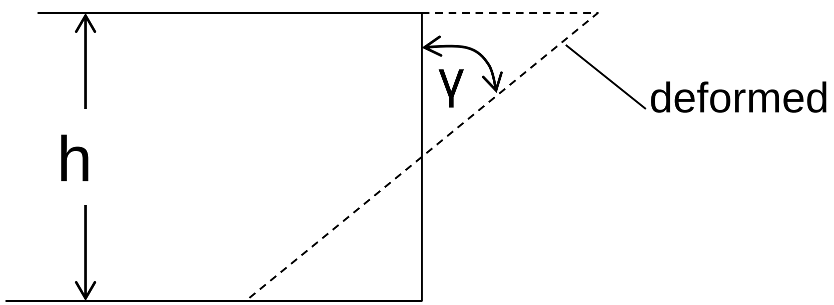


$$\gamma_{xz} = \Psi_x + \frac{\partial w}{\partial x}$$

independent of ζ!

Note on shear correction factor k

- reconcile inconsistency by making sure the work done is the same for both approaches



In general, work done:

$$W = \int_{-h/2}^{h/2} \tau_{xz}\gamma_{xz}dz$$

1st order theory

$$\gamma_{xz} = const = \gamma \Rightarrow W = \gamma \underbrace{\int_{-h/2}^{h/2} \tau_{xz}dz}_{\text{shear force } Q_x}$$

Kirchoff quadratic distribution

$$\tau_{xz} = \frac{3}{2}\frac{Q_x}{h}\left[1-\left(\frac{\zeta}{h/2}\right)^2\right] \Rightarrow W = \int_{-h/2}^{h/2} \frac{\left[\frac{3}{2}\frac{Q_x}{h}\left(1-\left(\frac{\zeta}{h/2}\right)^2\right)\right]^2}{G_{xz}} d\zeta = \frac{6}{5}\frac{Q_x^{\ 2}}{hG_{xz}}$$

Note on shear correction factor k

1st order theory

$$W = Q_x \gamma$$

Kirchoff quadratic distribution

$$W = \frac{6}{5}\frac{Q_x^{\,2}}{hG_{xz}}$$

shear correction factor k

$$Q_x = \frac{5}{6} G_{xz}\gamma h = \frac{5}{6}\tau_{xz} h$$

instead of

$$Q_x = \tau_{xz} h$$

Sandwich panel buckling - compression

- for a sandwich, the through-the-thickness shear distribution is (very nearly) uniform and k≈1; then, rearranging eq. (5.5.3.1.1):

why?

$$N_{crit} = \frac{t_c G_c}{\dfrac{t_c G_c}{N_{Ecrit}} + 1} \qquad (5.5.3.1.2)$$

- N_{Ecrit} for uni-axial compression was found before in eq. (5.2.3.1):

$$N_{Ecrit} = \frac{\pi^2\left[D_{11}m^4 + 2(D_{12} + 2D_{66})m^2(AR)^2 + D_{22}(AR)^4\right]}{a^2 m^2} \qquad (5.5.3.1.3)$$

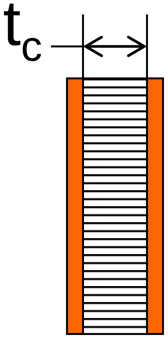
with a the panel length (load // a) and Dij given by (5.5.1)

Sandwich panel buckling - Example

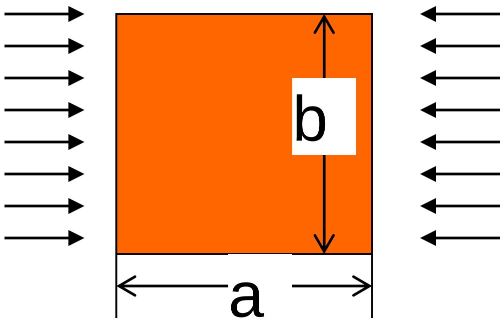
(±45)/(0/90)/ (±45) facesheet with
Nomex HRH-10 1/8-3.0 core

D11	659.7	Nmm
D12	466.9	Nmm
D22	659.7	Nmm
D66	494.0	Nmm
A11	28912	N/mm
A12	12491	N/mm
A22	28912	N/mm
A66	13469	N/mm

facesheet properties

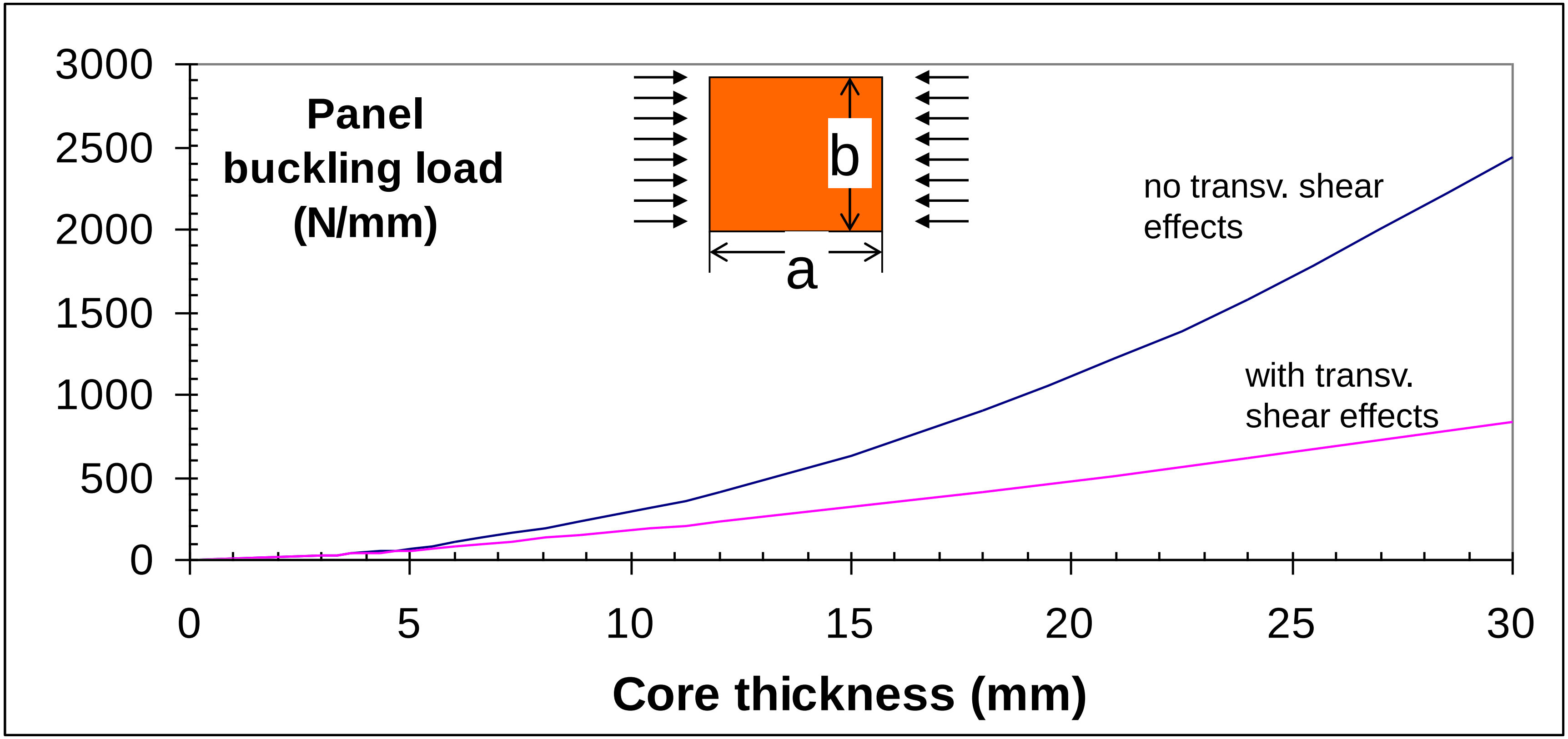


Core shear stiffness G_{LT}=42.1 N/mm^2



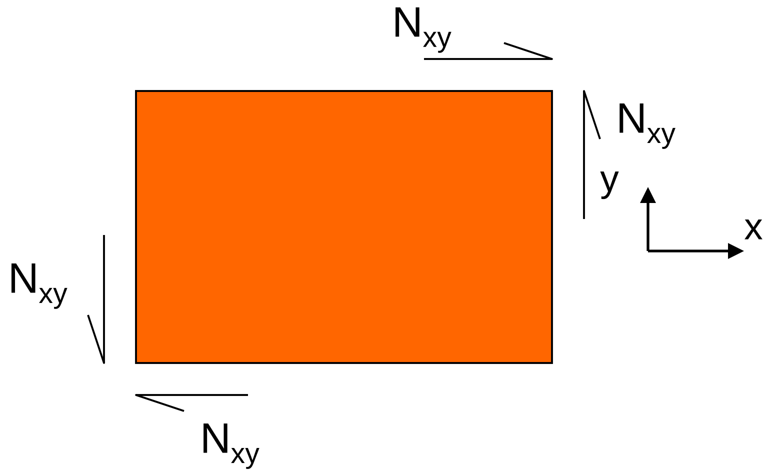
a=b=508 mm

Transverse shear effect on sandwich buckling load



at t_c=3 mm the difference in buckling loads with and without shear effects is already 21%

Sandwich panel buckling under shear



$$N_{xycrit} = \frac{(G_{xz} + G_{yz})t_c}{\frac{(G_{xz} + G_{yz})\ t_c}{N_{xyEcr}} + 2} \qquad (5.5.3.1.4)$$

• where N_{xyEcr} is the buckling load under shear for simply supported plate without shear correction

Sandwich panel under shear

- for N_{xyEcr} can use the expression derived at the very beginning of the course:

$$N_{xyEcr} = \frac{9\pi^4 b}{32a^3}\left(D_{11} + 2(D_{12} + 2D_{66})\frac{a^2}{b^2} + D_{22}\frac{a^4}{b^4}\right) \times 0.79$$

factor introduced to correct the expression derived by 2x2 eigenvalue problem

- or (see Advanced Composites Design Guide, DoD/NASA, 1983)

$$N_{xyEcr} = \frac{\dfrac{\pi^4 b}{a^3}}{\sqrt{\dfrac{14.28}{D1^2} + \dfrac{40.96}{D1D2} + \dfrac{40.96}{D1D3}}}$$

$$D1 = D_{11} + D_{22}\left(\frac{a}{b}\right)^4 + 2(D_{12} + 2D_{66})\left(\frac{a}{b}\right)^2$$

$$D2 = D_{11} + 81D_{22}\left(\frac{a}{b}\right)^4 + 18(D_{12} + 2D_{66})\left(\frac{a}{b}\right)^2$$

$$D3 = 81D_{11} + D_{22}\left(\frac{a}{b}\right)^4 + 18(D_{12} + 2D_{66})\left(\frac{a}{b}\right)^2$$

for $0.5 \le a/b < 1$

Sandwich panel under shear

- for 0≤a/b<0.5, interpolate between the value for a/b=0 given by

$$N_{xyEcr} = \left(\frac{2}{a}\right)^2 \left[D_{11}{}^3 D_{22}\right]^{1/4} + \left(8.125 + \frac{5.05\left(D_{12}+2D_{66}\right)}{\sqrt{D_{11}D_{22}}}\right)$$

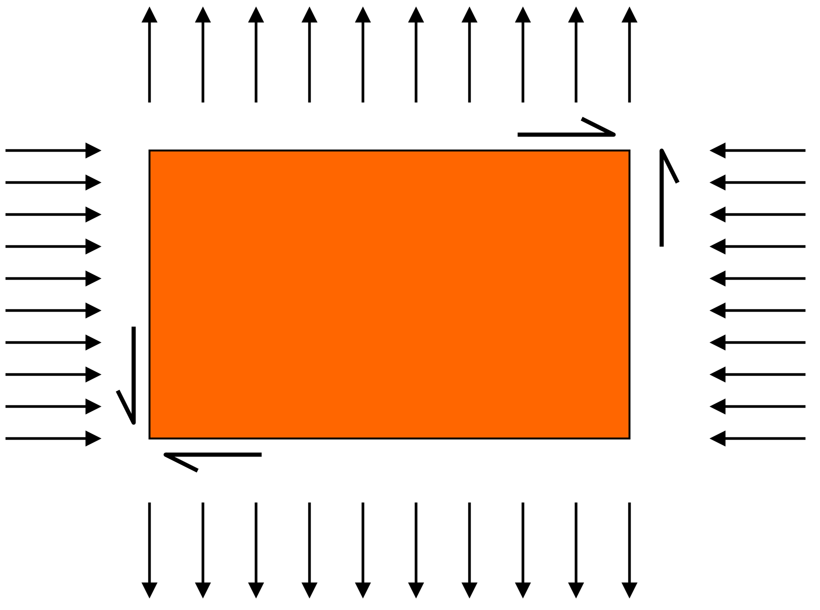
$$if \quad \frac{\sqrt{D_{11}D_{22}}}{\left(D_{12}+2D_{66}\right)} > 1$$

$$N_{xyEcr} = \left(\frac{2}{a}\right)^2 \sqrt{D_{11}\left(D_{12}+2D_{66}\right)}\left[11.7 + 0.532\frac{\sqrt{D_{11}D_{22}}}{\left(D_{12}+2D_{66}\right)} + 0.938\frac{D_{11}D_{22}}{\left(D_{12}+2D_{66}\right)^2}\right]$$

$$if \quad \frac{\sqrt{D_{11}D_{22}}}{\left(D_{12}+2D_{66}\right)} < 1$$

- and the value for a/b=0.5 given in the previous page

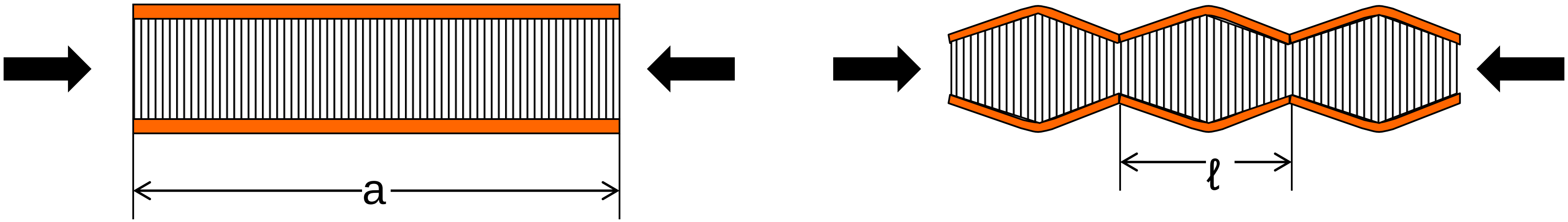
Sandwich panel buckling under combined loads



• use the interaction curves we had before but correct the individual buckling loads for for transverse shear effects

Wrinkling[1]

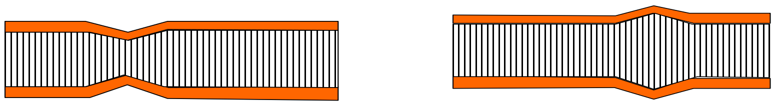
- wrinkling is a local buckling phenomenon where the facesheet buckles over a characteristic half-wave length ℓ unrelated to the panel length or width



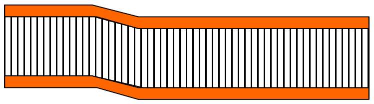
(1) Hoff, N.J.,Mautner, S.E., “The Buckling of Sandwich-Type Panels”, J Aeronautical Sciences, July 1945, pp 285-297

Wrinkling

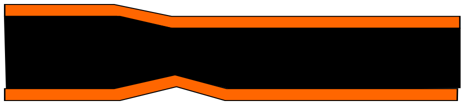
- two different failure modes:
  - symmetric[1]



  - antisymmetric

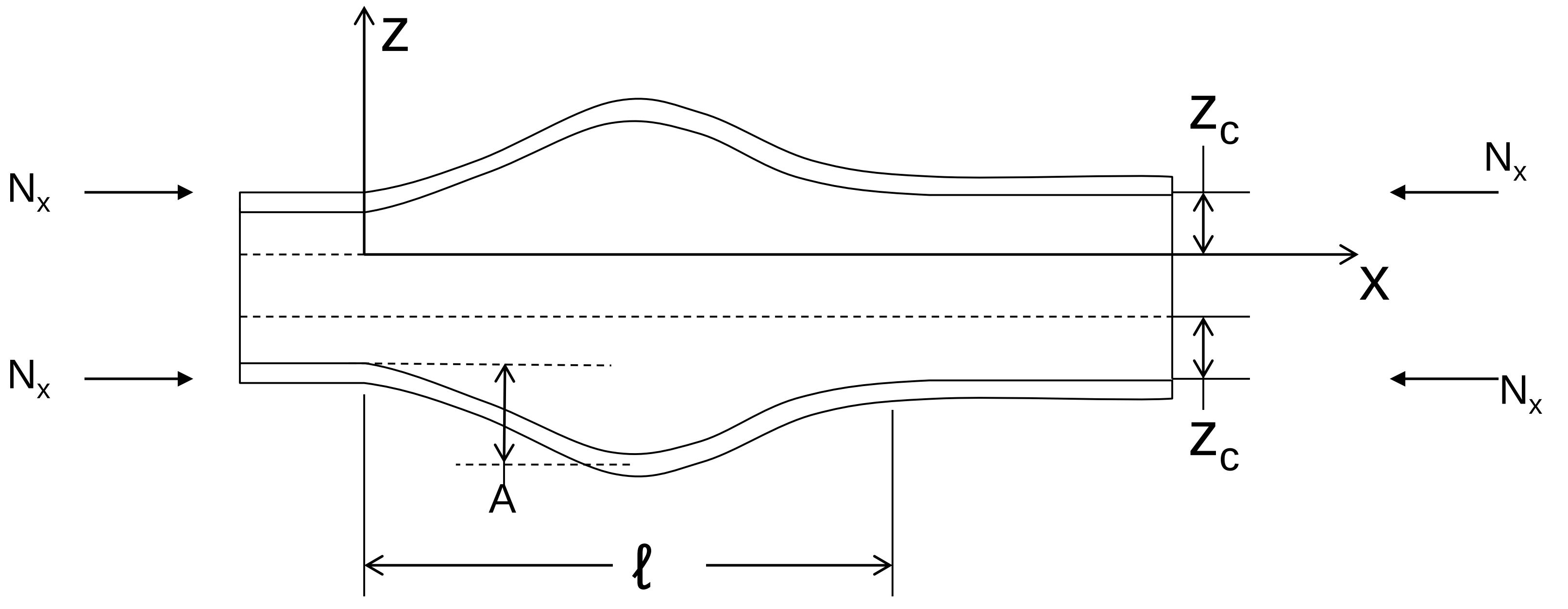


  - mixed modes are also possible



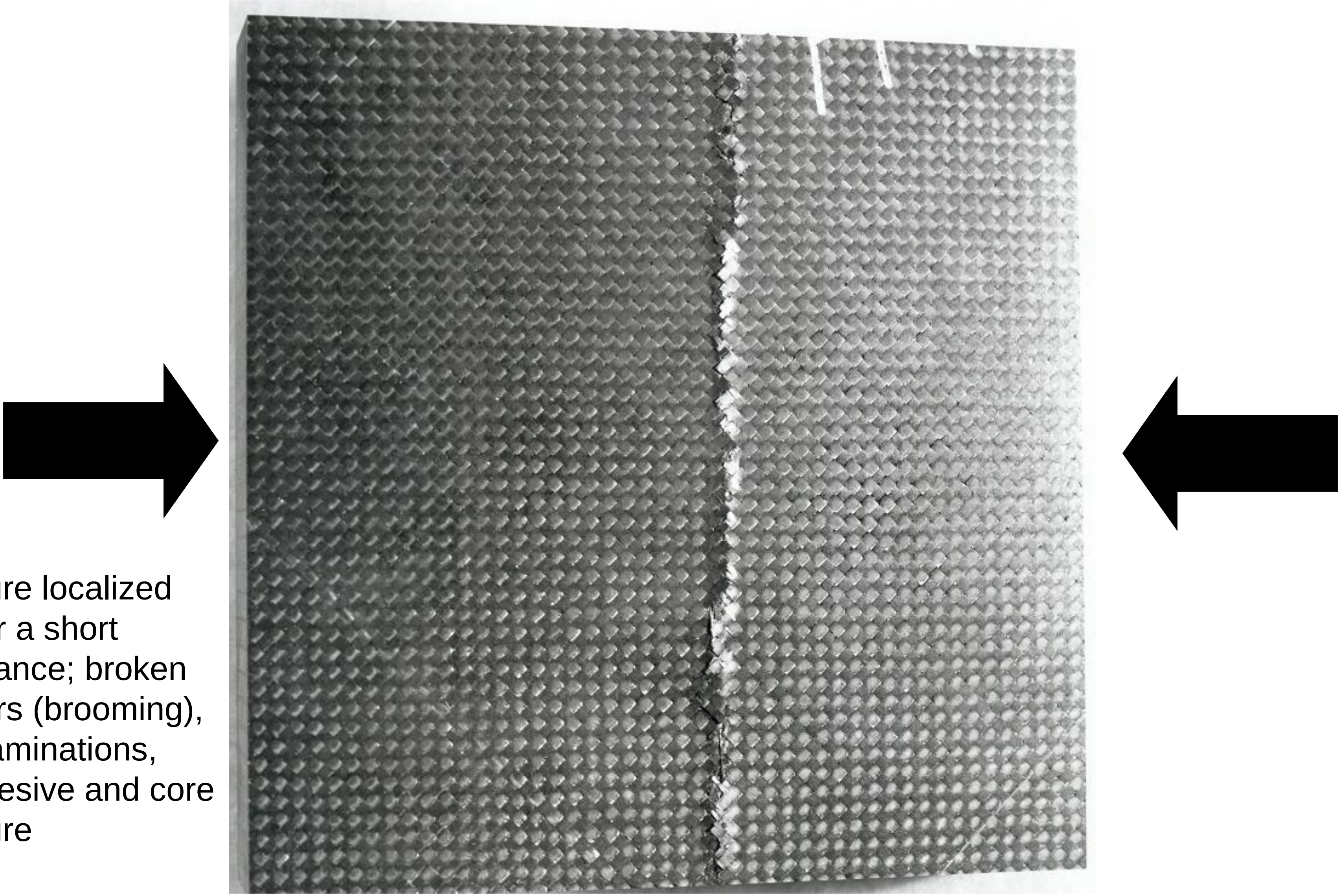
(1) symmetry refers to the local half-wave and is about an axis perpendicular to the plane of the panel

Symmetric Wrinkling



- sandwich is infinite in y direction
- facesheet buckles locally over a half-wavelength ℓ with simply supported BC's
- deflections vary linearly with z over a portion z_c of the core

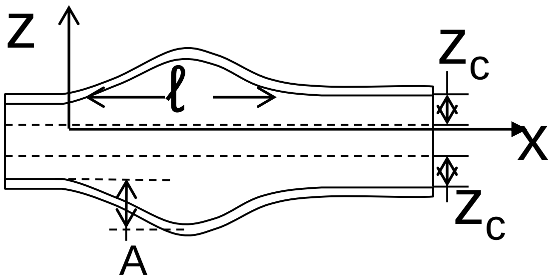
Symmetric Wrinkling



failure localized over a short distance; broken fibers (brooming), delaminations, adhesive and core failure

Symmetric Wrinkling

• assume deflections of facesheet and “active” portion of the core are given by



satisfies the z & x BC's on w

$$w = A\frac{z}{z_c}\sin\frac{\pi x}{\ell} \qquad (5.5.3.2.1)$$

• determine the wrinkling load by energy minimization

• total energy (=potential-work done) per unit width:

$$\Pi_c = 2U_f + U_c - 2W \qquad (5.5.3.2.2)$$

where U_f is energy stored in each facesheet, U_c is energy stored in the core, and W is work done by applied force N_x per facesheet

Symmetric Wrinkling

- assume that deformations in the plane of the facesheet (u,v) are negligible (=> $\varepsilon_{xo}=\varepsilon_{yo}=\gamma_{xyo}=0$)

- strains and stresses in the facesheet are given by

$$\left.\begin{aligned} \varepsilon_x &= -z\frac{\partial^2 w}{\partial x^2} \\ \sigma_x &= E_f \varepsilon_x \end{aligned}\right\} \Rightarrow \quad \sigma_x \varepsilon_x = E_f z^2 \left(\frac{\partial^2 w}{\partial x^2}\right)^2 \qquad (5.5.3.2.3)$$

where E_f is the facesheet (membrane) Young's modulus appropriately calculated

- then, per unit width,

$$U_f = \frac{1}{2}\iint E_f z^2 \left[\left(\frac{\partial^2 w}{\partial x^2}\right)^2\right]_{z=z_c} dz dx = \frac{1}{2} E_f \bar{I} \int_0^\ell \left[\left(\frac{\partial^2 w}{\partial x^2}\right)^2\right]_{z=z_c} dx = \frac{\left(\overline{EI}\right)_f}{2} \int_0^\ell \left[\left(\frac{\partial^2 w}{\partial x^2}\right)^2\right]_{z=z_c} dx \qquad (5.5.3.2.4)$$

where $\bar{I}=t_f^3/12$ and $(\overline{EI})_f \approx (D_{11})_f$

Symmetric Wrinkling

- strains and stresses in the core are given by

$$\varepsilon_z = \frac{\partial w}{\partial z}$$

$$\gamma_{xz} = \frac{\partial w}{\partial x} + \frac{\partial u}{\partial z}$$

u≈0 by previous assumption

$$\sigma_z = E_c \varepsilon_z$$

$$\tau_{xz} = G_{xz} \gamma_{xz}$$

$$\sigma_z \varepsilon_z = E_c \left(\frac{\partial w}{\partial z} \right)^2$$

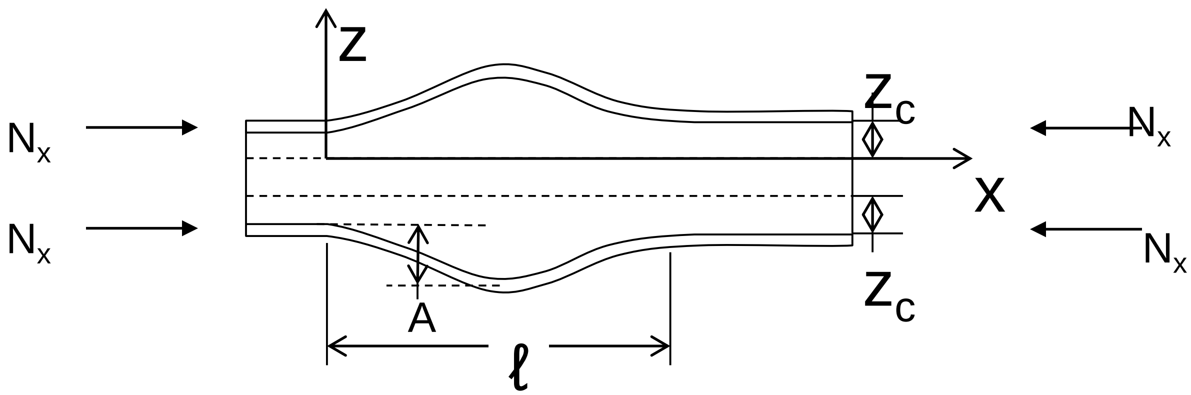
$$\tau_{xz} \gamma_{xz} = G_{xz} \left(\frac{\partial w}{\partial x} \right)^2 \qquad (5.5.3.2.5)$$

where E_c and G_{xz} are core axial and shear moduli respectively

- then, per unit width,

$$U_c = \frac{1}{2} \int_0^{\ell} \int_{-z_c}^{z_c} \left(E_c \left(\frac{\partial w}{\partial z} \right)^2 + G_{xz} \left(\frac{\partial w}{\partial x} \right)^2 \right) dz dx \qquad (5.5.3.2.6)$$

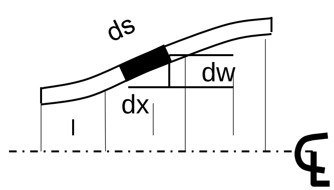
Symmetric Wrinkling



- work done per facesheet (per unit width):

$$W = N_x \delta$$

$$\delta = \ell - \int_0^{\ell} dx$$



$$(dx)^2 + (dw)^2 = (ds)^2 \Rightarrow dx = ds\sqrt{1-\left(\frac{dw}{ds}\right)^2}$$

$$\sqrt{1-\left(\frac{dw}{ds}\right)^2} \approx 1 - \frac{1}{2}\left(\frac{dw}{ds}\right)^2 \quad for \quad small \quad \left(\frac{dw}{ds}\right)^2$$

$$\frac{dw}{ds} \approx \frac{dw}{dx} \quad for \quad small \quad w$$

$$W = \frac{N_x}{2}\int_0^{\ell}\left(\frac{\partial w}{\partial x}\right)^2\Bigg|_{z=z_c} dx \qquad (5.5.3.2.7)$$

Symmetric Wrinkling

- use (5.5.3.2.1) to calculate necessary derivatives:

$$\left(\frac{\partial w}{\partial x}\right)^2 = \frac{A^2 z^2}{z_c{}^2}\frac{\pi^2}{2\ell^2}\left(1+\cos\frac{2\pi x}{\ell}\right)$$

$$\left(\frac{\partial w}{\partial z}\right)^2 = \frac{A^2}{z_c{}^2}\frac{1}{2}\left(1-\cos\frac{2\pi x}{\ell}\right) \qquad (5.5.3.2.8)$$

$$\left(\frac{\partial^2 w}{\partial x^2}\right)^2 = \frac{A^2 z^2}{z_c{}^2}\frac{\pi^4}{2\ell^4}\left(1-\cos\frac{2\pi x}{\ell}\right)$$

- substitute in (5.5.3.2.2) and carry out the integrations

$$\Pi_c = \frac{\pi^4}{2\ell^3}(\overline{EI})_f\, A^2 + \frac{1}{2}\left[\frac{E_c\ell}{z_c} + \frac{1}{3}G_{xz} z_c \frac{\pi^2}{\ell}\right]A^2 - N_x \frac{A^2\pi^2}{2\ell} \qquad (5.5.3.2.9)$$

Symmetric Wrinkling

• to determine the wrinkling load N_{xwr}, minimize the energy (differentiate Π_c with respect to A and set the result equal to zero):

$$\frac{\partial \Pi_c}{\partial A} = 0 \Rightarrow 2A\left[\frac{\pi^4}{2\ell^3}(\overline{EI})_f + \frac{1}{2}\left[\frac{E_c \ell}{z_c} + \frac{1}{3}G_{xz} z_c \frac{\pi^2}{\ell}\right] - N_x \frac{\pi^2}{2\ell}\right] = 0 \Rightarrow$$

$$N_{xwr} = \frac{\pi^2 (\overline{EI})_f}{\ell^2} + \frac{E_c \ell^2}{\pi^2 z_c} + G_{xz}\frac{z_c}{3} \qquad (5.5.3.2.10)$$

Euler column buckling load

contribution from beam on elastic foundation; compare with result from section 5.3.2

elastic foundation contribution when it consists of torsional springs

$$K_{mm} = \frac{\pi^2 EI}{L^2}\left(m^2 + \frac{kL^4}{\pi^4 (EI) m^2}\right)$$

same when m=1 and $k=E_c/z_c$

Symmetric wrinkling

- the expression for the wrinkling load, eq. (5.5.3.2.10) is still in terms of two unknown constants, ℓ and z_c
- since we are looking for the lowest buckling load, determine the value of ℓ that minimizes N_{xwr}

$$\frac{\partial N_{xwr}}{\partial \ell} = 0 \Rightarrow \ell = \pi \left(\frac{(\overline{EI})_f}{E_c} z_c \right)^{1/4} \qquad (5.5.3.2.11)$$

- substituting in (5.3.2.10):

$$N_{xwr} = \frac{2\sqrt{E_c (\overline{EI})_f}}{\sqrt{z_c}} + \frac{G_{xz} z_c}{3} \qquad (5.5.3.2.12)$$

Symmetric wrinkling

- in a similar manner, the "active" portion of the core, z_c can be determined,

$$\frac{\partial N_{xwr}}{\partial z_c} = 0 \Rightarrow z_c = 3^{2/3}\left(\frac{E_c(\overline{EI})_f}{G_{xz}^{\ 2}}\right)^{1/3}$$

- and substituting for $\overline{I}_f = t_f^3/12$

$$z_c = \frac{3^{2/3}}{12} t_f \left(\frac{E_c E_f}{G_{xz}^{\ 2}}\right)^{1/3} = 0.91 t_f \left(\frac{E_c E_f}{G_{xz}^{\ 2}}\right)^{1/3} \qquad (5.5.3.2.13)$$

- use this expression to substitute in (5.5.3.2.11) to get ℓ

$$\ell = \frac{\pi 3^{1/6}}{12^{1/3}} t_f \left(\frac{E_f}{\sqrt{E_c G_{xz}}}\right)^{1/3} = 1.648 t_f \left(\frac{E_f}{\sqrt{E_c G_{xz}}}\right)^{1/3} \qquad (5.5.3.2.14)$$

Symmetric wrinkling

• use (5.5.3.2.13) and (5.5.3.2.14) to substitute in (5.5.3.2.12) to get

$$N_{xwr} = 0.91 t_f \left(E_f E_c G_{xz} \right)^{1/3} \qquad (5.5.3.2.15)$$

per facesheet!

• this expression has been derived by many people making different assumptions and using different methods; typically, the only thing that changes as the approach and assumptions change is the coefficient; for a good review of most of the methods, see: Ley, R.P., Lin, W., and Mbanefo, U., “Facesheet Wrinkling in Sandwich Structures”, NASA/CR-1999-208994, January 1999