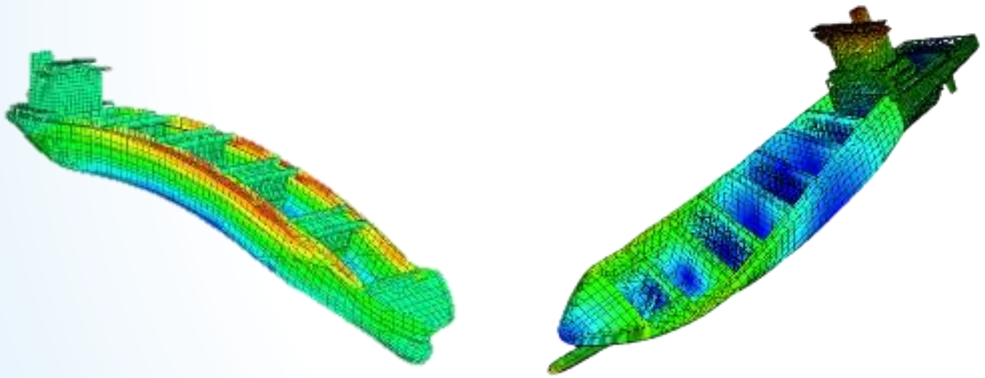
Topics in Ship Structures

(Advanced Local Structural Design & Analysis of Marine Structures)



* Long Clamped Plates (Topic 6)

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Adv. Local Structural Design & Analysis of Marine Structures (Overview)

[Theory of Plates and Grillages]

[Part I] Plastic Design of Structures

- Plastic theory of bending (Topic 1)
- Ultimate loads on beams (Topic 2)
- Collapse of frames and grillage structures (Topic 3)

[Part II] Elastic Plate Theory under Pressure

- Basic (Topic 4)
- Simply supported plates under Sinusoidal Loading (Topic 5)
- **Long clamped plates (Topic 6)**
- Short clamped plates (Topic 7)
- Low aspect ratio plates, strength & permanent set (Topic 7A)

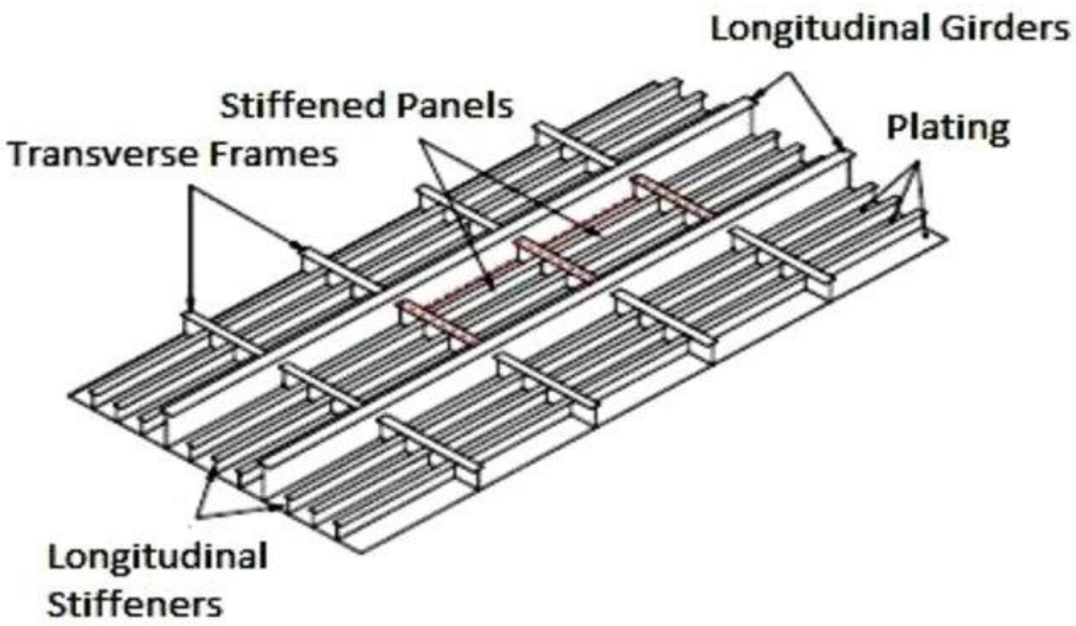
[Part III] Buckling of Plate & Stiffened Panels

- Failure modes (Topic 8)
- Tripping (Topic 9) + Post-buckling strength of plate (Topic 9A)
- Post-buckling behaviour (Topic 10)

Objectives

The aim of this lecture is:

- To equip you with the knowledge and understanding of long plate theory



Picture from:
Raviprakash A. V., Bala P., and Natarajan A. (2012). “Effect of location and orientation of two short dents on ultimate compressive strength of a thin square steel plate”, *International Journal of Computer Applications in Technology* 45(1) DOI: 10.1504/IJCAT.2012.050134

Learning Outcome

At the end of this lecture, you should be able to:

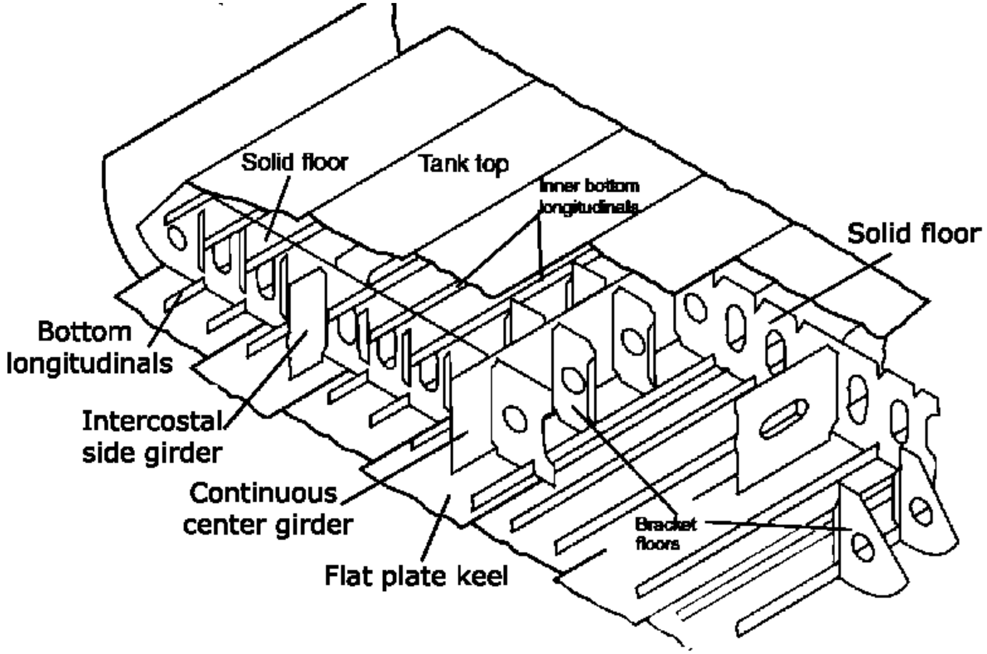
- Describe plate behaviour under lateral pressure and explain the effects of edge conditions on plate design.
- Perform elasto-plastic and plastic analysis of plates undergoing large deflections.
- Design under uniform pressure to meet various design criteria.
- Predict the performance of plate under uniform pressure.

Introduction (1/3)

- Ship plates are either longitudinally framed or transversely framed.

- The spacing of secondary members such as longitudinals, beams and ordinary frames are smaller than those of primary members such as deep transverses and girders.

- This results in the aspect ratios of the plates being Greater than 1.0.



Picture from:
https://www.marineinsight.com/naval-architecture/design-of-ships-bottom-structure/

Introduction (2/3)

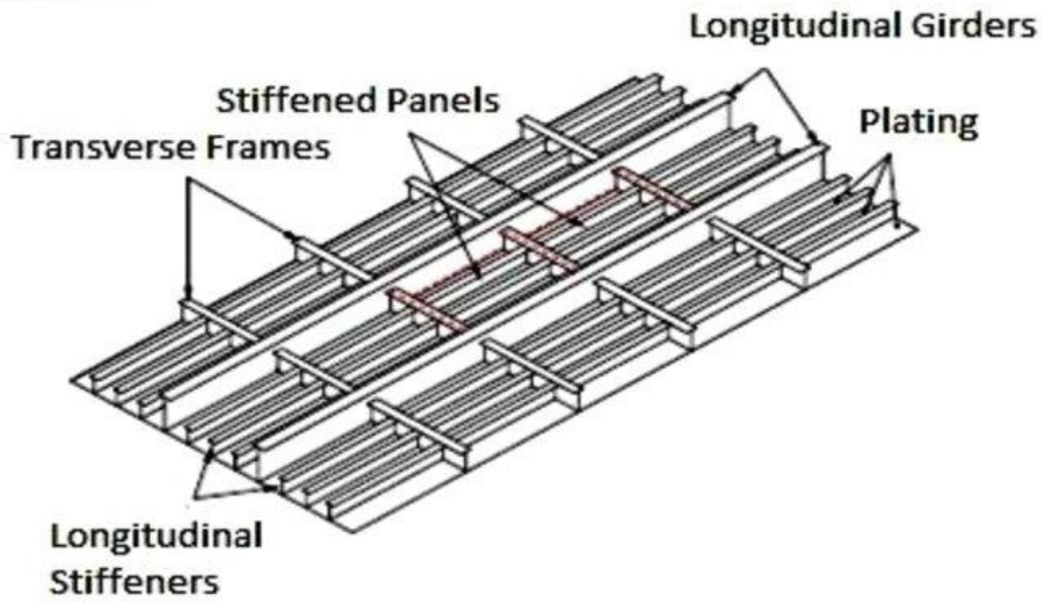
- In the case of large aspect ratio, i.e. “long” plate, the perturbation at the short edges is ignored and the plate is assumed to bend in one plane only as a beam.

- It follows therefore that linear theory will apply for rather greater deflections which need only to be considered small compared with the width of the plate so that the simple linear expression for curvature can be used with sufficient accuracy.

- The first three assumptions for linear plate theory are still made for long plate theory.

Introduction (3/3)

- The advantages of studying long plate theory, quite apart from its instructional value and simplicity, are:
  - Non-linear effects can be more readily examined, as can a variety of boundary conditions.
  - It is relatively easily extended to cover elasto-plastic behaviour.
  - It is useful in design in so far as long plate data is easily presented
  - The reduction in thickness required for short plates is available for any aspect ratio and a variety of theories and boundary assumptions.



*Topic 6.1:

Long Clamped Plates under Uniform Pressure

Long Clamped Plates: Uniform Pressure (1/5)

- A plate may be considered "long" if its primary direction of bending (i.e. that leading to the maximum stresses) leads to moments and stresses virtually identical to that of an infinitely long plate between parallel supports of distance b apart.

- The analysis is referred to as "cylindrical bending" and is identical (apart from Poisson's effects) to beam bending analysis.

- If a is the length of the plate, the maximum central deformation is $$\boxed{w_{\max} = k\left(w_{\max}\right)_{a=\infty}}$$

- When the aspect ratio a/b exceeds 2, $k \rightarrow 1$, a plate may generally be considered long if its aspect ratio is greater than about

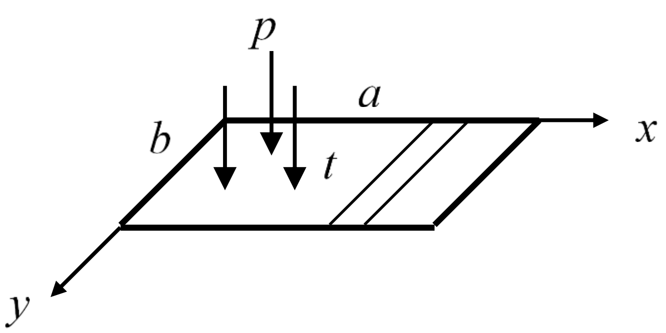
  2.5 to 3 for elastic theories

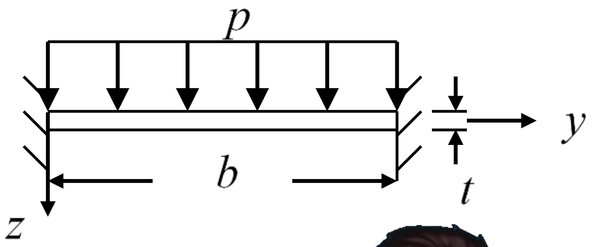
  2 to 2.5 for elasto-plastic or plastic theories

- For long clamped plate we can isolate a typical 'unit strip' and consider its behaviour under uniform pressure p.

Long Clamped Plates: Uniform Pressure (2/5)

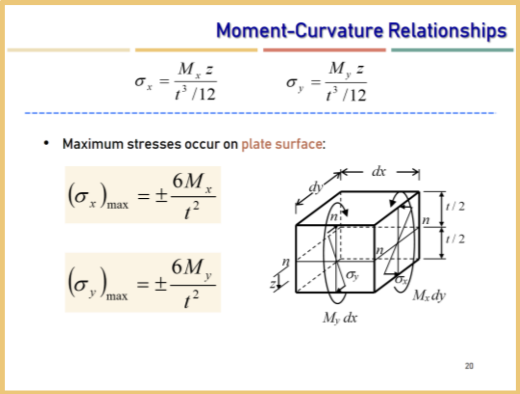
- The edges are assumed to be clamped and held rigidly.
- The rigid assumption means that the plate strip CANNOT move in any direction and in particular resists in-plane sliding that may occur as large deflection built-up.





- In this case we will need to allow for "membrane" tension which arise because of the rigid edge.

- For any given moment M_y per unit length, the bending stress in y-direction is



$$\sigma_y = \frac{6\boxed{M_y}}{t^2}$$

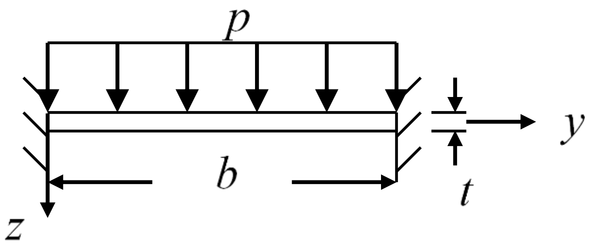
- This is maximum at the edge where $\boxed{M_y = pb^2/12}$ (i.e. fixed end moment for UDL)

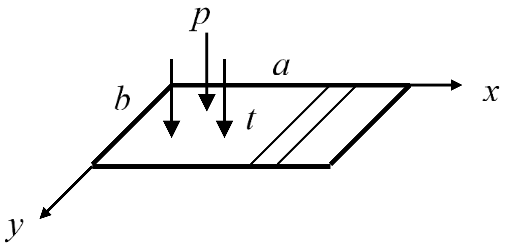
$$(\sigma_y)_{max} = \frac{p}{2}\left(\frac{b}{t}\right)^2$$

Note: UDL = uniformly distributed load

Long Clamped Plates: Uniform Pressure (3/5)

$$\left(\sigma_y\right)_{\max} = \frac{p}{2}\left(\frac{b}{t}\right)^2$$





- This result can also be obtained from the expression for σ_m in clamped plate under uniform pressure when k_2 equals to 1 as a / b exceeds 2.

- The yield pressure at which the edge stress first equals to the yield stress is

$$p_{yield} = 2\sigma_{yield}\left(\frac{t}{b}\right)^2$$

without Poisson's effect.

- Because of Poisson's effect there is also a stress in the long x-direction which is approximated by assuming the strain in x-direction is zero.

$$\varepsilon_x = \frac{1}{E}\left(\sigma_x - \nu\sigma_y\right) = 0$$

Then $\sigma_x = \nu\sigma_y$

Long Clamped Plates: Uniform Pressure (4/5)

- To find the pressure at which yield first occurs, using von-Mises criteria of yielding, requires

$$\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2 = \sigma_{yield}^2$$

where σ_1 and σ_2 are principal stresses.

- Let $\sigma_1 = (\sigma_x)_{\max} = \nu(\sigma_y)_{\max}$ and $\sigma_2 = (\sigma_y)_{\max}$

- For yield failure $(\sigma_y^2)_{\max}(1-\nu+\nu^2) = \sigma_{yield}^2$ → $\sigma_{y\max} = \dfrac{\sigma_{yield}}{\sqrt{1-\nu+\nu^2}}$

- In case of steel $\nu = 0.3$, we have → $(\sigma_y)_{\max} = 1.125\sigma_{yield}$

- Hence, the yield pressure should be modified to account for 12.5% increase in the stress value at which yielding really occurs.

$$\boxed{p_{yield} = 2 \times 1.125\sigma_{yield}\left(\frac{t}{b}\right)^2 = \boxed{2.25\sigma_{yield}\left(\frac{t}{b}\right)^2}}$$

with Poisson's effect

https://en.wikipedia.org/wiki/Von_Mises_yield_criterion

Long Clamped Plates: Uniform Pressure (5/5)

- If the central deflection is required, this is given with analogue to beam results.

- Maximum deflection

$$w_m = \frac{pb^4}{384\,D} \text{ for Plate}$$

$$where,\ D = \frac{Et^3}{12\left(1-\nu^2\right)} \qquad w_m = \frac{qL^4}{384\,EI} \text{ for beam}$$

$$\therefore\ \boxed{w_m} = \frac{\left(1-\nu^2\right)pb^4}{32Et^3} = \boxed{0.0284\frac{pb^4}{Et^3}} \text{ for steel}$$

- Comparing this result with the expression for the maximum central deflection in clamped plate under uniform pressure, the value of k_1 approaches to 0.0284 as a/b exceeds 2.

Clamped Plates: Uniform Pressure

- Maximum deflection

$$w_m = k_1\frac{p_o b^4}{Et^3}$$

- Maximum bending stress in y-direction

$$\sigma_m = k_2\frac{p_o}{2}\left(\frac{b}{t}\right)^2$$

- Values of k_1 and k_2 according to two reliable theories are plotted on the following page for $\nu = 0.3$.

a/b	1.0	1.2	1.4	1.6	1.8	2.0	∞
k_2	0.3078	0.3834	0.4356	0.4680	0.4872	0.4974	0.5000
k_1	0.0138	0.0188	0.0226	0.0251	0.0267	0.0277	0.0284

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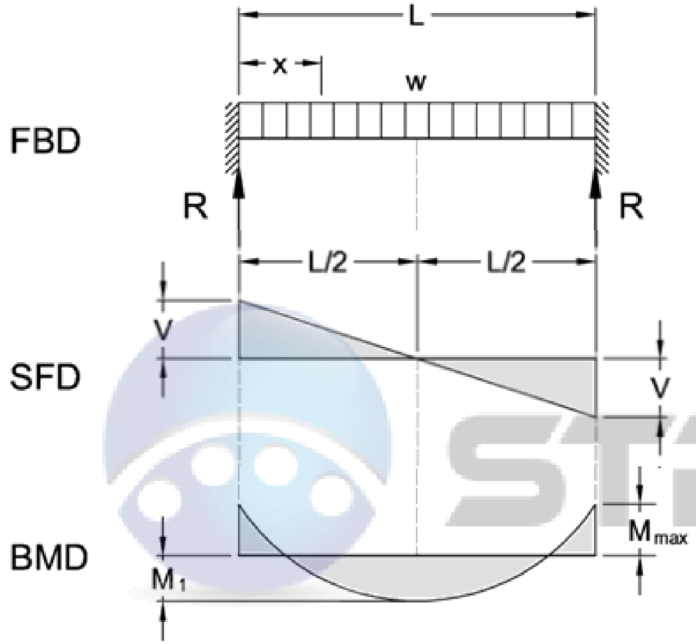
Long Clamped Plates: For your study (1/2)

(a) Sketch shear force diagram (SFD) and bending moment diagram (BMD).
(b) Calculate maximum deflection of the beam.



$R = V \ldots = \frac{wL}{2}$

$V_x \ldots = w\left(\frac{L}{2} - x\right)$

M_{max} (at ends) $\ldots = \frac{wL^2}{12}$

M_1 (at centre) $\ldots = \frac{wL^2}{24}$

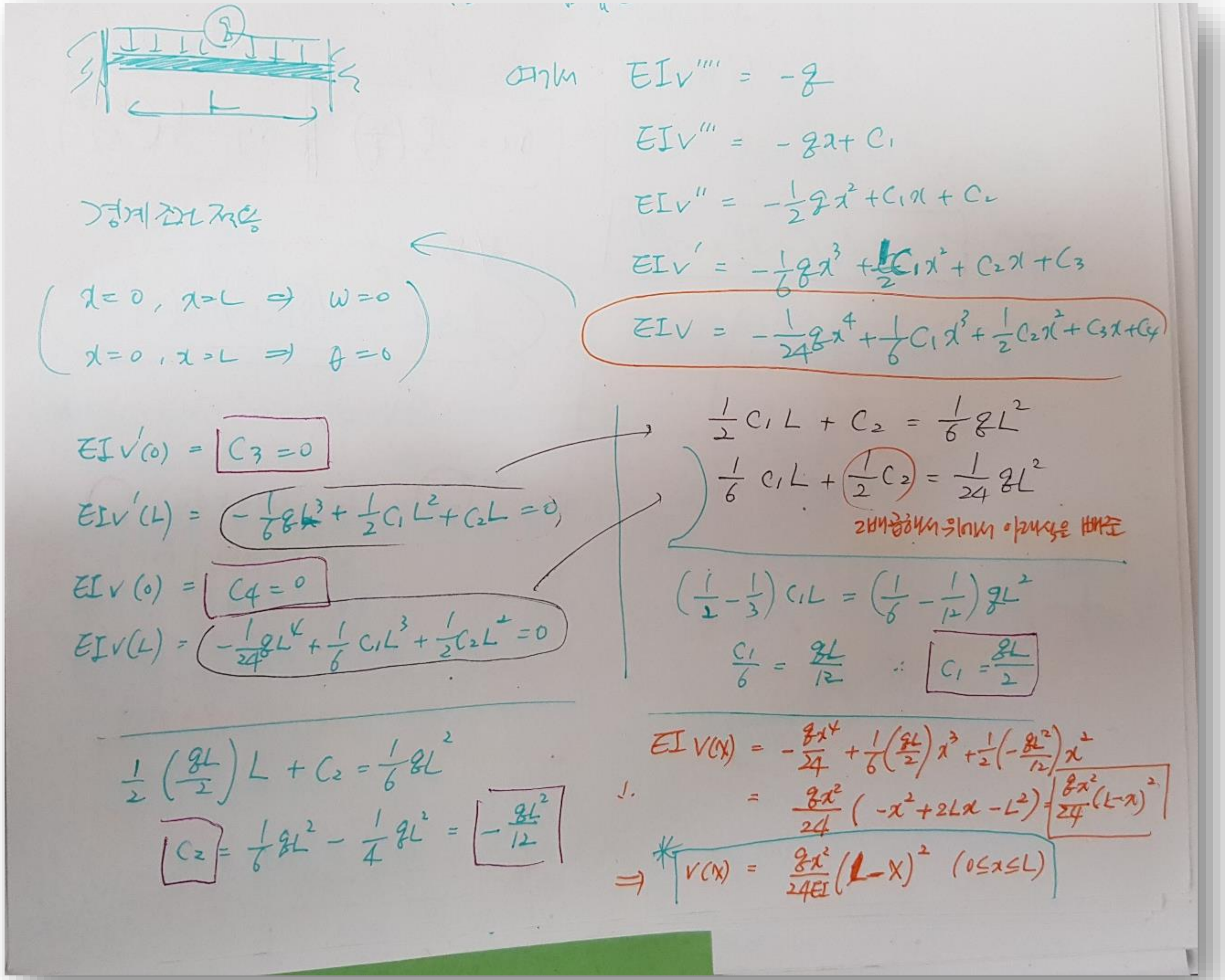
$M_x \ldots = \frac{w}{12}(6Lx - L^2 - 6x^2)$

Δ_{max} (at centre) $\ldots = \frac{wL^4}{384EI}$

$\Delta_x \ldots = \frac{wx^2}{24EI}(L - x)^2$

x (points of contraflexure) $\ldots = (\sqrt{3} - 3)L$

Long Clamped Plates: For your study (2/2)



$$v(x)\ or\ w(x) = deflection(m)$$

$$\theta(x) = \frac{dv(x)}{dx} = v'(x) = slope(rad)$$

$$M(x) = EI\frac{d^2v(x)}{dx^2} = EIv''(x) = moment(Nm)$$

$$V(x) = EI\frac{d^3v(x)}{dx^3} = EIv'''(x) = shear(N)$$

$$q(x) = EI\frac{d^4v(x)}{dx^4} = EIv''''(x) = Load(N/m)$$

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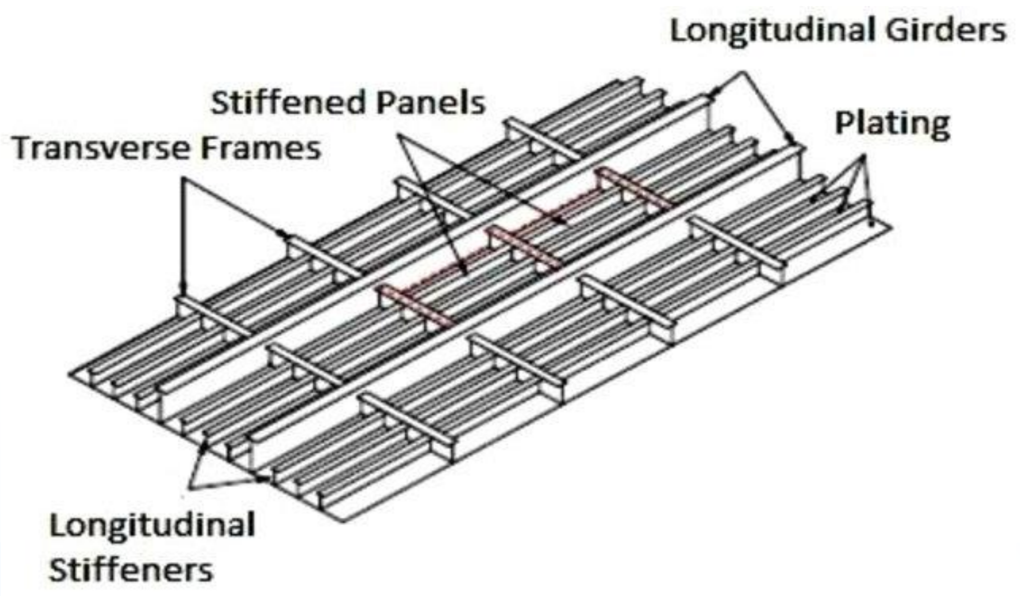


*Topic 6.2:

Elasto-plastic Bending of Long Clamped Plates

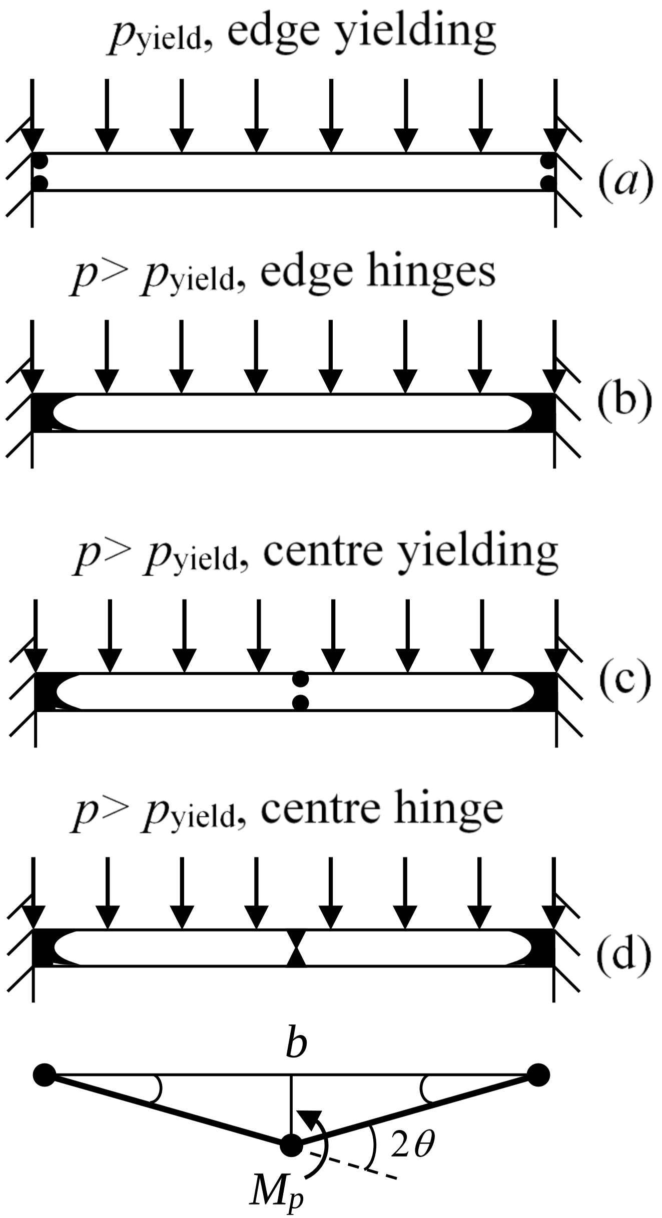
Long Clamped Plates: Elasto-plastic Bending (1/2)

- As pressure increases, the elastic bending moment diagram indicates that the yield first occurs at the edges (STAGE a) and bending moment increase at the mid-span.

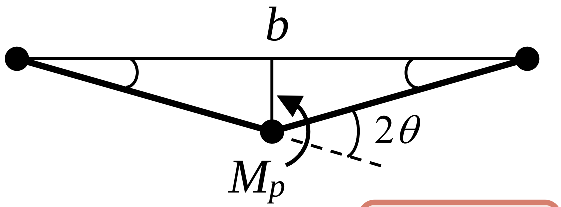
- Then it progresses until fully developed hinges occur at the edges (STAGE b).

- Pressure continues to increase in the bending moment at mid-span until it yields (STAGE c) and finally forms a hinge (STAGE d).

- The above progress is just the same as a plastic beam. In the final STAGE (d), three hinges have now formed a collapse mechanism.



Long Clamped Plates: Elasto-plastic Bending (2/2)



- Let p_u = collapse pressure
  M_p = plastic moment per unit length

- Internal work = $\boxed{4 M_p \theta}$
- External work = $2\left(\frac{p_u b}{2} w_p\right) = 2\left(\frac{p_u b}{2}\frac{b\theta}{4}\right) = \boxed{\frac{p_u b^2 \theta}{4}}$

- Equating them gives $\boxed{p_u = \frac{16 M_p}{b^2}}$

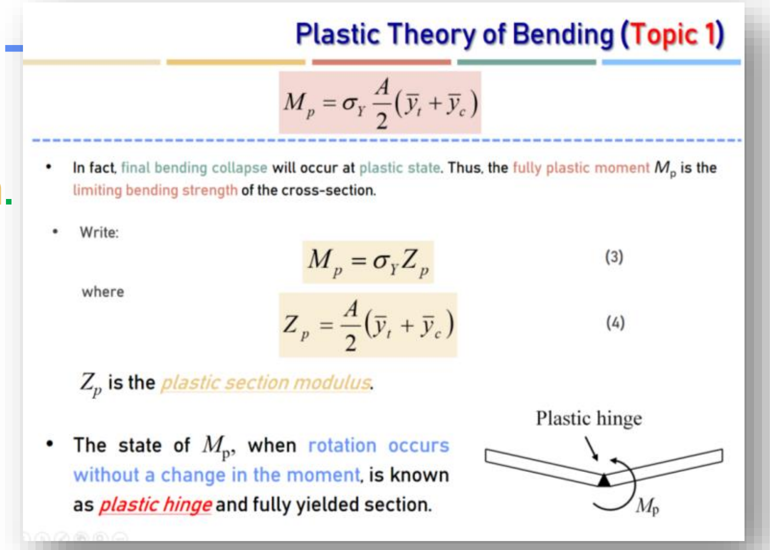
- Recalling $M_p = \sigma_{yield} Z_p$ where the plastic section modulus $\boxed{Z_p = t^2/4}$ per unit length.

- Hence,

$$\boxed{\begin{aligned} p_u &= 4\sigma_{yield}\left(\frac{t}{b}\right)^2 \\ p_u &= 4.5\sigma_{yield}\left(\frac{t}{b}\right)^2 \end{aligned}}$$

W/O Poisson's effect.

WITH Poisson's effect.



- When Poisson's effect is present, the maximum stress will increase by 12.5%.
- This collapse pressure is derived on the basis that the clamped edges are free to slide.

Reminder → Plastic analysis (Topic 2)

Ultimate Loads on Beams

(c) Fixed end beam with uniform distributed load

- Internal work $= 4\,M_p\,\theta$
- External work $=$

$$2\left(\frac{w_p L}{2}\delta\right) = 2\left(\frac{w_p L}{2}\frac{L\theta}{4}\right) = \frac{w_p L^2}{4}\theta$$

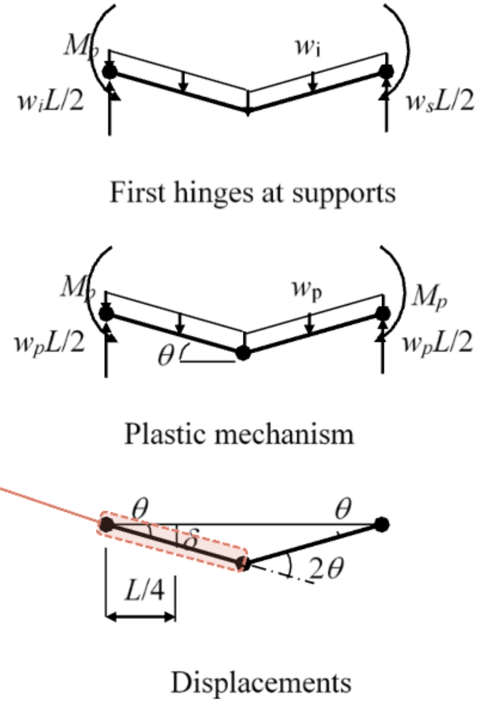
- By the principle of virtual work,

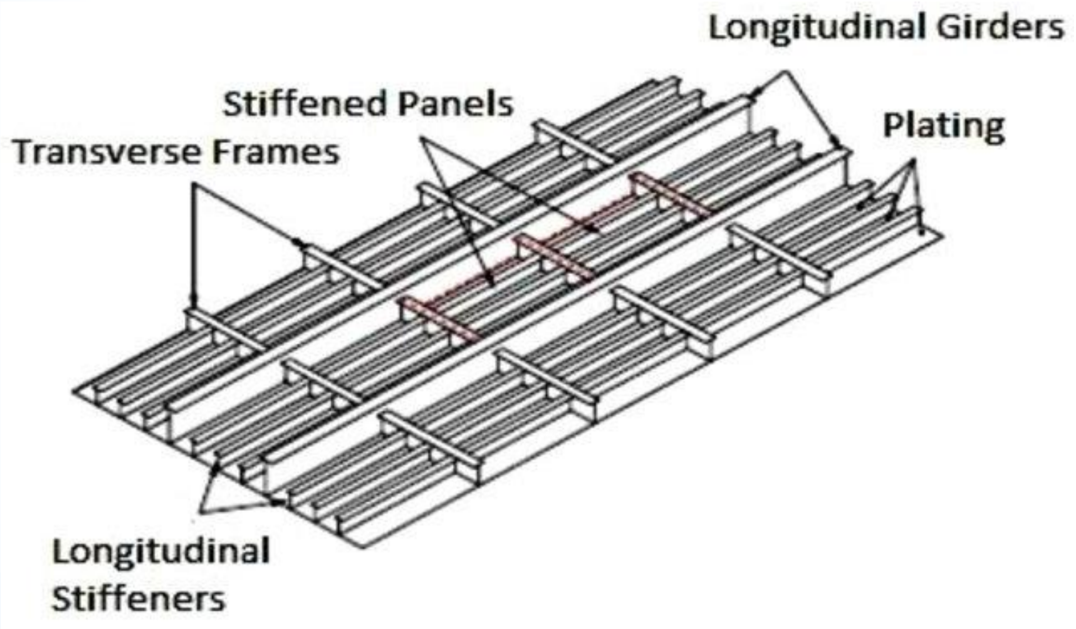
$$\frac{w_p L^2}{4}\theta = 4M_p\theta$$

- Thus, the ultimate load

$$w_p = \frac{16M_p}{L^2}$$

$$w_Y = \frac{12M_Y}{L^2}$$



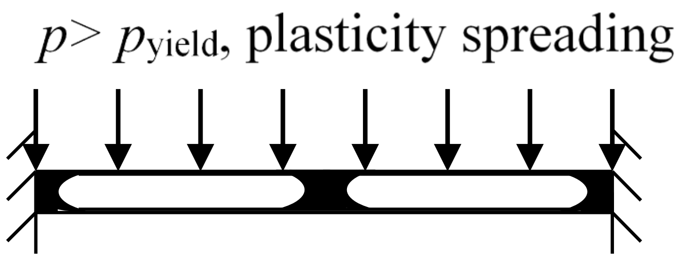


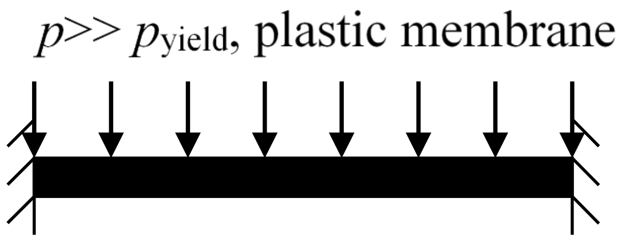
*Topic 6.3:

Long Plates at Rupture

Long Plates at Rupture (1/5)

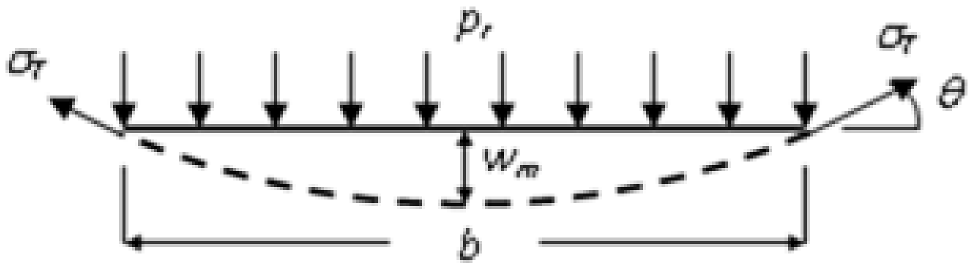
- If the edges are held rigidly, p_u derived above WILL NOT represent collapse.
- This is because membrane tension σ_T can develop in the plate, adding appreciable stiffness to the plate.
- In this case, plasticity will finally spread everywhere until $\sigma_T = \sigma_{yield}$.
- The plate will have become a fully plastic membrane in which it has NO REMAINING ability to carry a bending moment by sustaining the load by membrane alone.





Long Plates at Rupture (2/5) : Elastic Membranes

Elastic Membranes



- Consider a long thin plate in which the bending resistance is low.

- We can gain some insight into behaviour by assuming zero bending rigidity
- Thus, the only resistance to pressure is through membrane tension forces.

- For equilibrium,

$$p_r b = 2\sigma_T t \sin\theta$$

- For small θ,

$$\sin\theta \approx \theta \approx \frac{b/2}{R}$$

- Therefore,

$$p_r = \frac{\sigma_T t}{R}$$

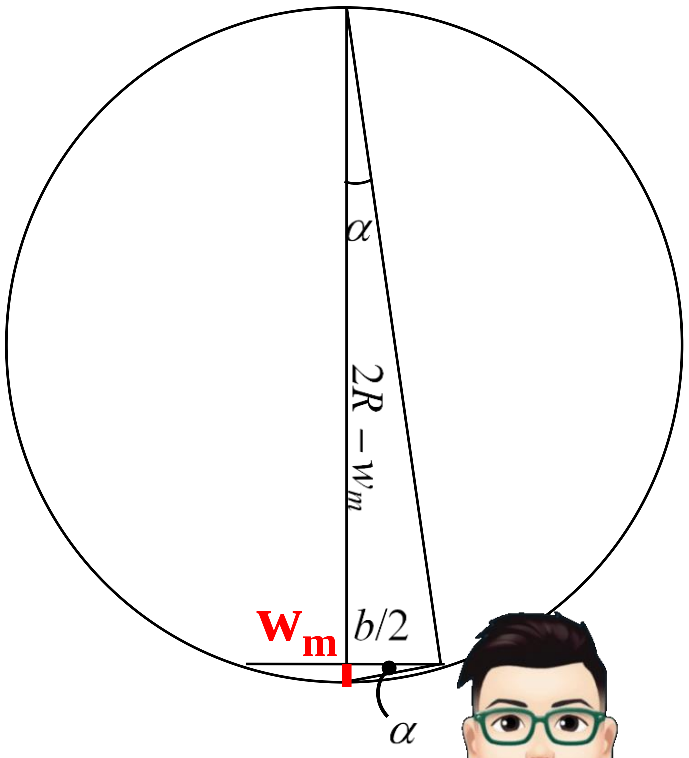
where R is the unknown radius of membrane, which can be expressed in terms of the central deflection w_m from geometry.

Long Plates at Rupture (3/5): Elastic Membranes

- From the geometry of a circle as shown,

$$\boxed{\frac{w_m}{b/2}} = \frac{b/2}{2R - w_m} \approx \boxed{\frac{b}{4R}} \quad \text{for } R \gg w_m \quad \therefore \boxed{\frac{1}{R} = \frac{8w_m}{b^2}}$$

R = unknown radius of membrane which can be expressed by central deflection w_n from geometry



- Hence,

$$p_r = \frac{\sigma_T t}{R} = \frac{8tw_m}{b^2}\sigma_T$$

- From stress-strain relation, we have

$$\varepsilon_T = \frac{\sigma_T}{E}\left(1 - \nu^2\right)$$

$$\sin\theta \approx \theta \approx \frac{b/2}{R}$$

$$\boxed{b/2 = R\sin\theta}$$

- From the geometry of a circle as shown,

$$\varepsilon_T = \frac{R\theta - \boxed{b/2}}{\boxed{b/2}} = \frac{\theta - \sin\theta}{\sin\theta} = \frac{\theta - \theta + \theta^3/6 - \cdots}{\theta - \theta^3/6 + \cdots} \approx \frac{\theta^3/6}{\theta} \approx \frac{\theta^2}{6}$$

$$\therefore \varepsilon_T = \frac{\theta^2}{6} = \frac{b^2}{24R^2} = \frac{8}{3}\left(\frac{w_m}{b}\right)^2$$

Long Plates at Rupture (4/5): Elastic Membranes

$$\boxed{p_r} = \frac{\sigma_T t}{R} = \boxed{\frac{8t\boxed{w_m}}{b^2}\sigma_T} \quad 2$$

$$\boxed{\varepsilon_T} = \frac{\theta^2}{6} = \frac{b^2}{24R^2} = \boxed{\frac{8}{3}\left(\frac{w_m}{b}\right)^2} = \frac{\sigma_T}{E}\left(1-\nu^2\right)$$

$$p_r = \frac{8tw_m}{b^2}\boxed{\sigma_T} = \frac{8tw_m}{b^2}\boxed{\frac{E\cdot\varepsilon_T}{\left(1-\nu^2\right)}} = \frac{8tw_m}{b^2}\boxed{\frac{E}{\left(1-\nu^2\right)}\frac{8}{3}\left(\frac{w_m}{b}\right)^2}$$

- Hence, we have rupture pressure p_r as a function of central deflection w_m

$$p_r = \frac{8tw_m}{b^2}\frac{E}{\left(1-\nu^2\right)}\frac{8}{3}\left(\frac{w_m}{b}\right)^2 \qquad \therefore \quad p_r = \frac{64E}{3\left(1-\nu^2\right)}\frac{t\boxed{w_m^3}}{b^4} \quad 1$$

or as a function of membrane stress by $\boxed{\text{eliminating } w_m}$

$$\frac{(2)^3}{1}$$

$$p_r = \frac{8t\sigma_T}{b^2}b\sqrt{\frac{3\sigma_T\left(1-\nu^2\right)}{8E}} \qquad \therefore \quad p_r = \frac{t}{b}\sqrt{\frac{24\left(1-\nu^2\right)\sigma_T^3}{E}}$$

- Taking $\sigma_T = \frac{2}{3}\sigma_{yield}$ as a limiting stress gives

$$p_r = \frac{8}{3}\sqrt{1-\nu^2}\,\frac{t}{b}\sqrt{\frac{\sigma_{yield}^3}{E}}$$

Long Plates at Rupture (5/5): Plastic Membranes

$$\boxed{p_r} = \frac{\sigma_T t}{R} = \boxed{\frac{8t\,w_m}{b^2}\sigma_T}$$

Plastic Membranes

- If we ignore the elastic behaviour and assume the material is rigid plastic, then the pressure is only resisted by membrane action at constant yield stress

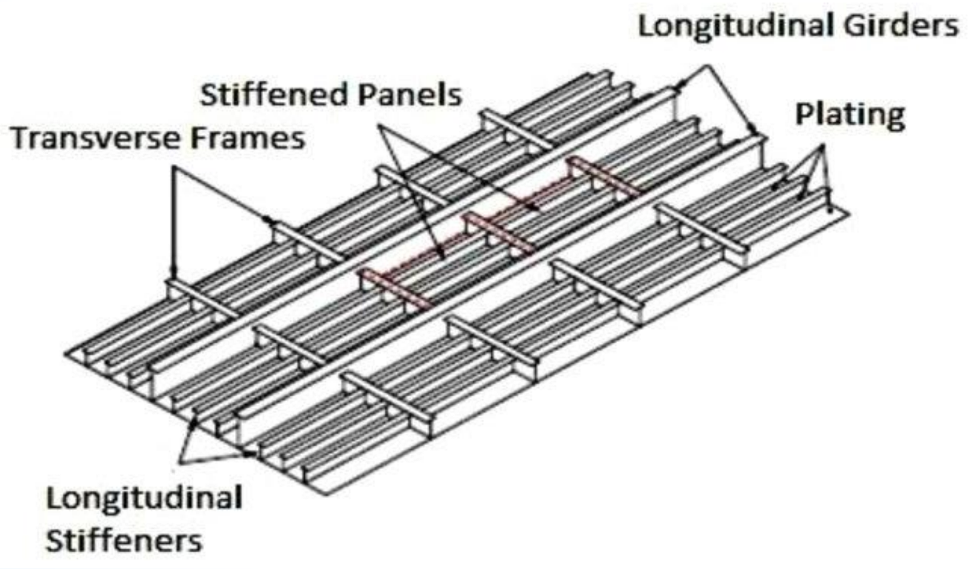
$$\sigma_T = \sigma_{yield}$$

- Hence, from the elastic membrane analysis, we have

$$p_r = \frac{8tw_m}{b^2}\sigma_{yield}$$ without Passion's effect

or

$$p_r = \frac{9tw_m}{b^2}\sigma_{yield}$$ with Passion's effect (12.5% increased)



*Topic 6.4:

Clarkson's Criteria for Design Limits

Clarkson's Criteria for Design Limits (1/2)

- Working for the Royal Navy in the late 50's and early 60's Clarkson (Trans RINA 1956) developed an Elasto-plastic analysis for long and square plates having clamped rigid boundaries
- It allowed for the growth of membrane stresses and plasticity.
- The effect of rotational constraints was shown to have little effect once permanent sets developed.
- He suggested three criteria for design limits.

Clarkson's Criteria for Design Limits (2/2)

i. For stocky plates $\beta \le 2.5$ the formation of the centre plastic hinge with a load factor of 1.0 (i.e. design load = failure load) should be considered, which clearly approaches to:

$$p_u = 4.5\sigma_{yield}\left(\frac{t}{b}\right)^2$$

ii. For slender plates $\beta > 2.5$ the membrane stress Does Not Exceed $2/3\ \sigma_{yield}$. This approaches to

$$p_r = \frac{8}{3}\sqrt{1-\nu^2}\,\frac{t}{b}\sqrt{\frac{\sigma_{yield}^3}{E}}$$

Since permanent sets arise from welding, and also from concentrated loads such as wheel loads (from aircraft, helicopters, fork-lift trucks, cars, lorries, etc), Clarkson also proposed an alternative design criterion based on an allowable permanent set δ (i.e. allowable initial deformation).

in which plate slenderness

$$\beta = \frac{b}{t}\sqrt{\frac{\sigma_{yield}}{E}}$$

Practical Boundary Conditions

- In most stiffened plate structures, the nature of the in-plane conditions varies from one plate element to another throughout the grillage.
- Near deck edges there is usually little resistance to in-plane movement.
- At the centre of deck, the continuity of plating suggests edges-held-rigidly apart might apply.
- However, in-plane compressive forces, which exist from grillage bending and/or hull bending actions, cause shear slide in the edge and increase lateral deformation.

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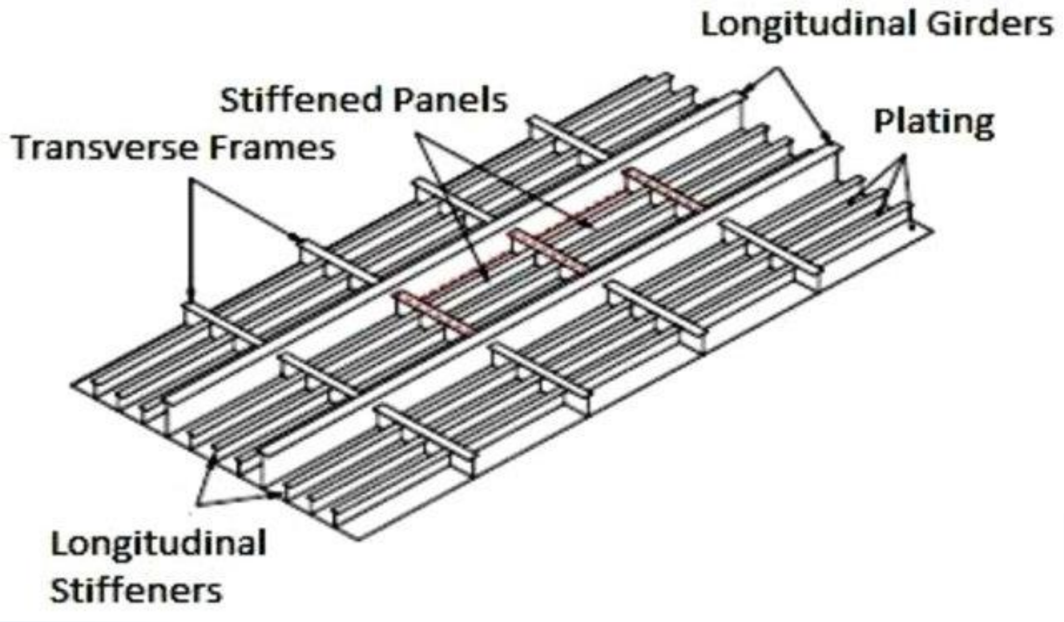


*Topic 6.5:

Design Methods for Long Clamped Plates

Design Methods for Long Clamped Plates (1/6)

- We now consider design methods for long clamped plates under uniform lateral pressure. The problem is given the uniform pressure p, the breadth b and material properties (yield stress σ_{yield}, Young's modulus of elasticity E and Poisson's ratio v) to determine the plate thickness t.

- For any given t, we can calculate the following quantities:

1. Calculate Elastic stress and deflection p, when the edges of the plate are free to slide
2. Calculate a value of p to cause initial yield
3. Calculate value of p to plastic collapse, if the edges are free to slide
4. Calculate the maximum permissible set δ
5. Calculate the value of p to cause rupture

$$\sigma_m = \frac{p}{2}\left(\frac{b}{t}\right)^2 \qquad w_m = 0.0284\,\frac{pb^4}{Et^3}$$

$$p_{yield} = 2\sigma_{yield}\left(\frac{t}{b}\right)^2$$

$$p_u = 4\sigma_{yield}\left(\frac{t}{b}\right)^2$$

Design Methods for Long Clamped Plates (2/6)

1 Calculate Elastic stress and deflection p, when the edges of the plate are free to slide

- [From 1] Determine a suitable thickness t, given a working stress σ_m.

$$t = b\sqrt{\frac{p}{2\sigma_m}}$$

2 Calculate a value of p to cause initial yield

- [From 2] Determine a suitable thickness t, given the yield stress σ_{yield}, with a first yield condition

$$t = b\sqrt{\frac{p}{2\sigma_{yield}}}$$

3 Calculate value of p to plastic collapse, if the edges are free to slide

- [From 3] Determine a suitable thickness t, given the yield stress s_{yield}, with an ultimate strength condition

$$t = b\sqrt{\frac{p}{4\sigma_{yield}}}$$

4 Calculate the maximum permissible set δ

- [From 4] there are design charts.

5 Calculate the value of p to cause rupture

- The rupture value, in 5, is not often used because of uncertainties concerning the actual ductility.

Design Methods for Long Clamped Plates (3/6)

- The thickness selected depends upon:

1. Boundary conditions and especially the response of the supports to membrane forces.

2. The criterion of failure. This depends upon the use to which the structure is put.
   - Maximum stress
   - Yield stress
   - Plastic collapse

Design Methods for Long Clamped Plates (4/6)

- A blast wall in an offshore structure for example would only have to withstand the loading once and for all
- If rupture were avoided, large permanent deformations would be quite acceptable.
- The deck of a Ro-Ro ship would have to withstand the same loading many times, WITHOUT any permanent deformation, large deflections, or overstressing.
- Fatigue would also have to be considered.
- Different cases need different criteria. These different cases need different design curves.
- It is useful and more general to develop such curves on a non-dimensional basis. Consider case 2 above.
- Then $\boxed{\dfrac{b}{t} = \sqrt{\dfrac{2\sigma_{yield}}{p}}}$
- Hence, $\boxed{\dfrac{b}{t}\sqrt{\dfrac{\sigma_{yield}}{E}}} = \sqrt{\dfrac{2\sigma_{yield}^2}{pE}}$

β = plate slenderness ratio

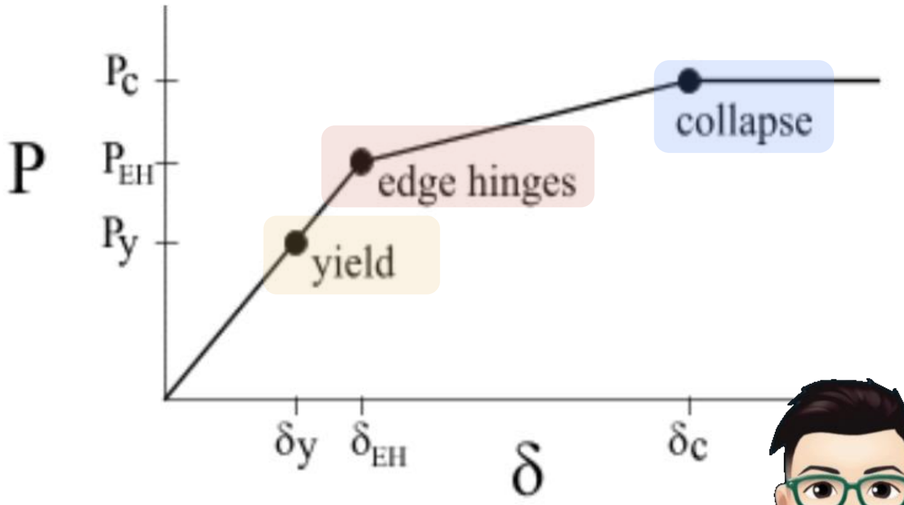
- It is thus convenient to plot $\boxed{\dfrac{pE}{\sigma_{yield}^2}}$ against $\boxed{\dfrac{b}{t}\sqrt{\dfrac{\sigma_{yield}}{E}}}$
- Such a plot can be applied to a range of materials.

Design Methods for Long Clamped Plates (5/6)

[Summary] Bending of Long Plates

Load	Deflection
$P_y = 2.25\sigma_y \left(\frac{t}{b}\right)^2 = 2\sigma_{yp}\left(\frac{t}{b}\right)^2$	$\delta_y = \frac{1}{384}\frac{P_y b^4}{D}$
$P_{EH} = 3.375\sigma_y \left(\frac{t}{b}\right)^2 = 3\sigma_{yp}\left(\frac{t}{b}\right)^2$	$\delta_y = \frac{1}{384}\frac{P_{EH} b^4}{D}$
$P_C = 4.5\sigma_y \left(\frac{t}{b}\right)^2 = 4\sigma_{yp}\left(\frac{t}{b}\right)^2$	$\delta_y = \frac{2}{384}\frac{P_C b^4}{D}$

- Plotting the load vs deflection, we get:



- Nomenclature

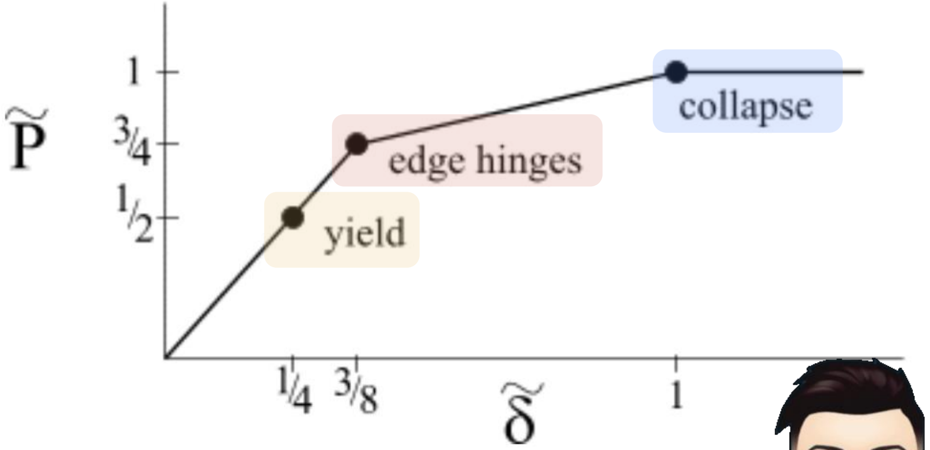
P_y = First yield pressure
P_{EH} = Pressure to form first plastic hinges
P_C = Collapse pressure
σ_y = Normal material yield stress
$\sigma_{yp} = 1.125\sigma_y$ (enhanced material yield stress)
δ_y = plate central deflection = w_m

Design Methods for Long Clamped Plates (6/6)

- If we **normalize** the **loads** P_c and the **deflections** by δ_c we get,

Load	Deflection
$\tilde{P}_y = \frac{P_y}{P_C} = 0.5$	$\tilde{\delta}_y = \frac{\delta_y}{\delta_C} = \frac{1}{4}$
$\tilde{P}_{EH} = \frac{P_{EH}}{P_C} = 0.75$	$\tilde{\delta}_{EH} = \frac{\delta_{EH}}{\delta_C} = \frac{3}{8}$
$\tilde{P}_C = \frac{P_C}{P_C} = 1$	$\tilde{\delta}_C = \frac{\delta_C}{\delta_C} = 1$

- Plotting the load vs deflection, we get:



- Nomenclature

P_y = First yield pressure
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- Thus, we see that:

P_y is 50% P_c & P_{EH} is 75% of P_c

δ_y is 25% δ_c & δ_{EH} is 37.5% of δ_c

Learning Outcomes (Review)

- We have investigated the Elastic Plate Theory.

- Now we are able to:
  - Describe plate behaviour under lateral pressure and explain the effects of edge conditions on plate design.
  - Perform elasto-plastic and plastic analysis of plates undergoing large deflections.
  - Design under uniform pressure to meet various design criteria.
  - Predict the performance of plate under uniform pressure.

- Details can be referred to topics 6 in the lecture notes.

Adv. Marine Structures / Adv. Structural Design & Analysis (Next class)

[Theory of Plates and Grillages]

[Part I] Plastic Design of Structures

- Plastic theory of bending (Topic 1)
- Ultimate loads on beams (Topic 2)
- Collapse of frames and grillage structures (Topic 3)

[Part II] Elastic Plate Theory

- Basic (Topic 4)
- Simply supported plates under Sinusoidal Loading (Topic 5)
- Long clamped plates (Topic 6)
- $\boxed{\text{Short Clamped plates (Topic 7)}}$
- Additional (Low aspect ratio plates, strength & permanent set)

[Part III] Buckling of Stiffened Panels

- Failure modes (Topic 8)
- Tripping (Topic 9)
- Post-buckling behaviour (Topic 10)

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Kam Sa Hab Ni Da

감사합니다

Thank you!

Questions?

Aerial View of Korean Presidential Archives in Sejong city
(Construction Completed in 2014)