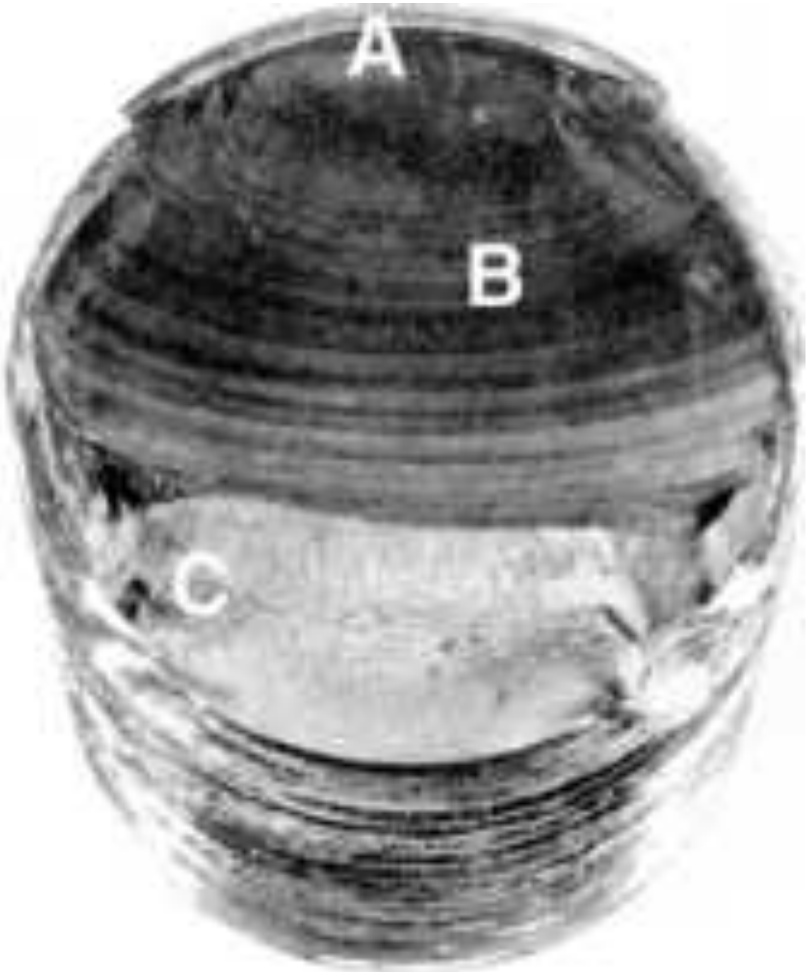
Introduction to Fatigue in Metals

- Loading produces stresses that are variable, repeated, alternating, or fluctuating
- Maximum stresses well below yield strength
- Failure occurs after many stress cycles
- Failure is by sudden ultimate fracture
- No visible warning in advance of failure

Stages of Fatigue Failure

- ***Stage I:*** Initiation of micro-crack due to cyclic plastic deformation
- ***Stage II:*** Progresses to macro-crack that repeatedly opens & closes, creating bands called *beach marks*
- ***Stage III:*** Crack has propagated far enough that remaining material is insufficient to carry the load, and fails by simple ultimate failure



Fatigue Fracture Example

- AISI 4320 drive shaft
- B: crack initiation at stress concentration in keyway
- C: Final brittle failure

Fatigue-Life Methods

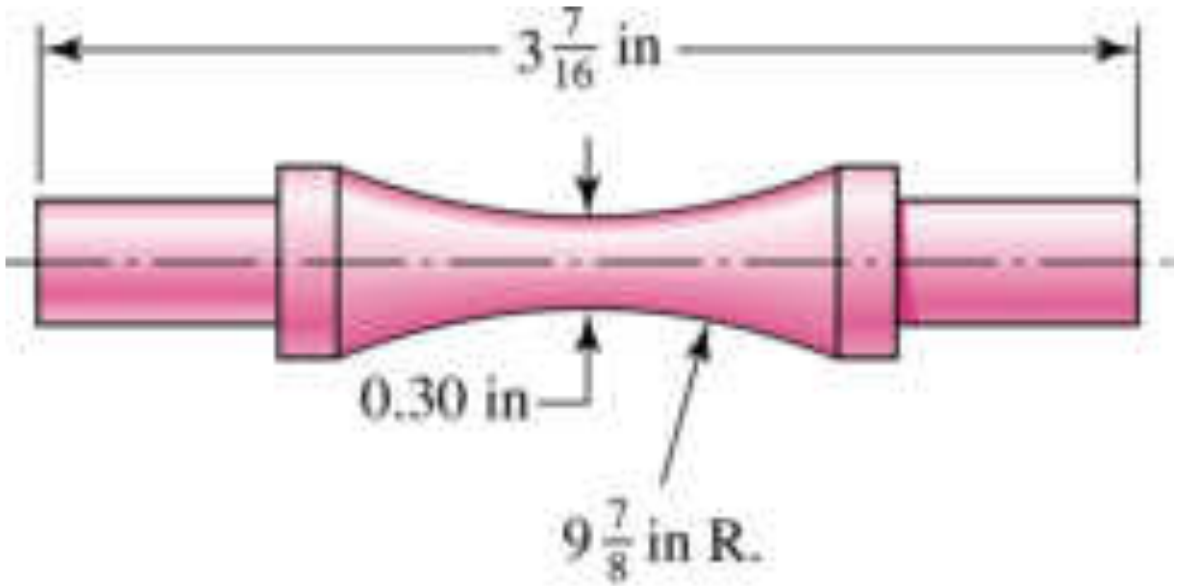
- Three major fatigue life models
- Methods predict life in number of cycles to failure, *N*, for a specific level of loading

Fatigue-Life Methods

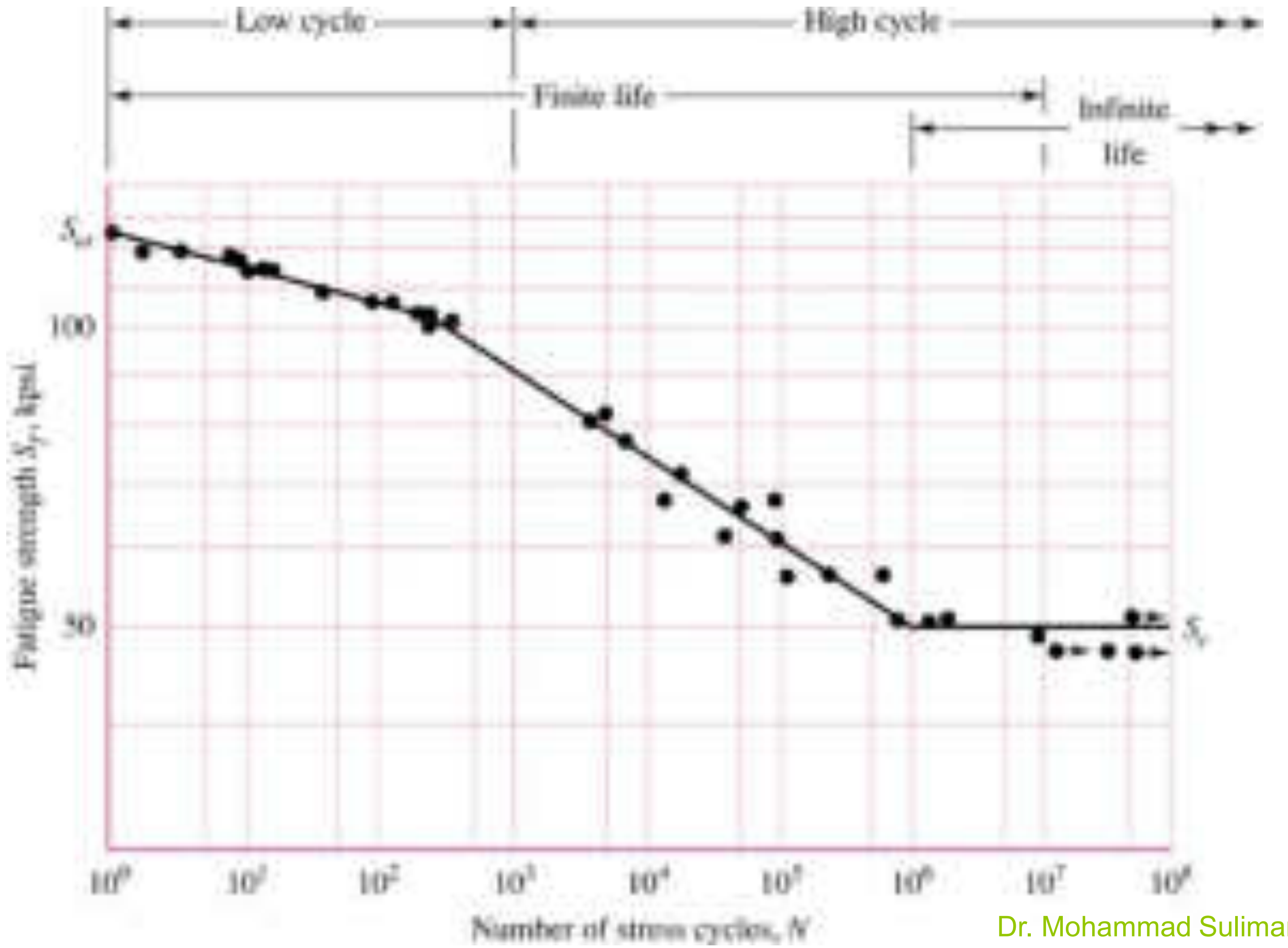
1. Stress-life method
2. Strain-life method
3. Linear-elastic fracture mechanics method

1. Stress-Life Method

- Test specimens subjected to repeated stress while counting cycles to failure
- Pure bending with no transverse shear
- *completely reversed* stress cycling



S-N Diagram

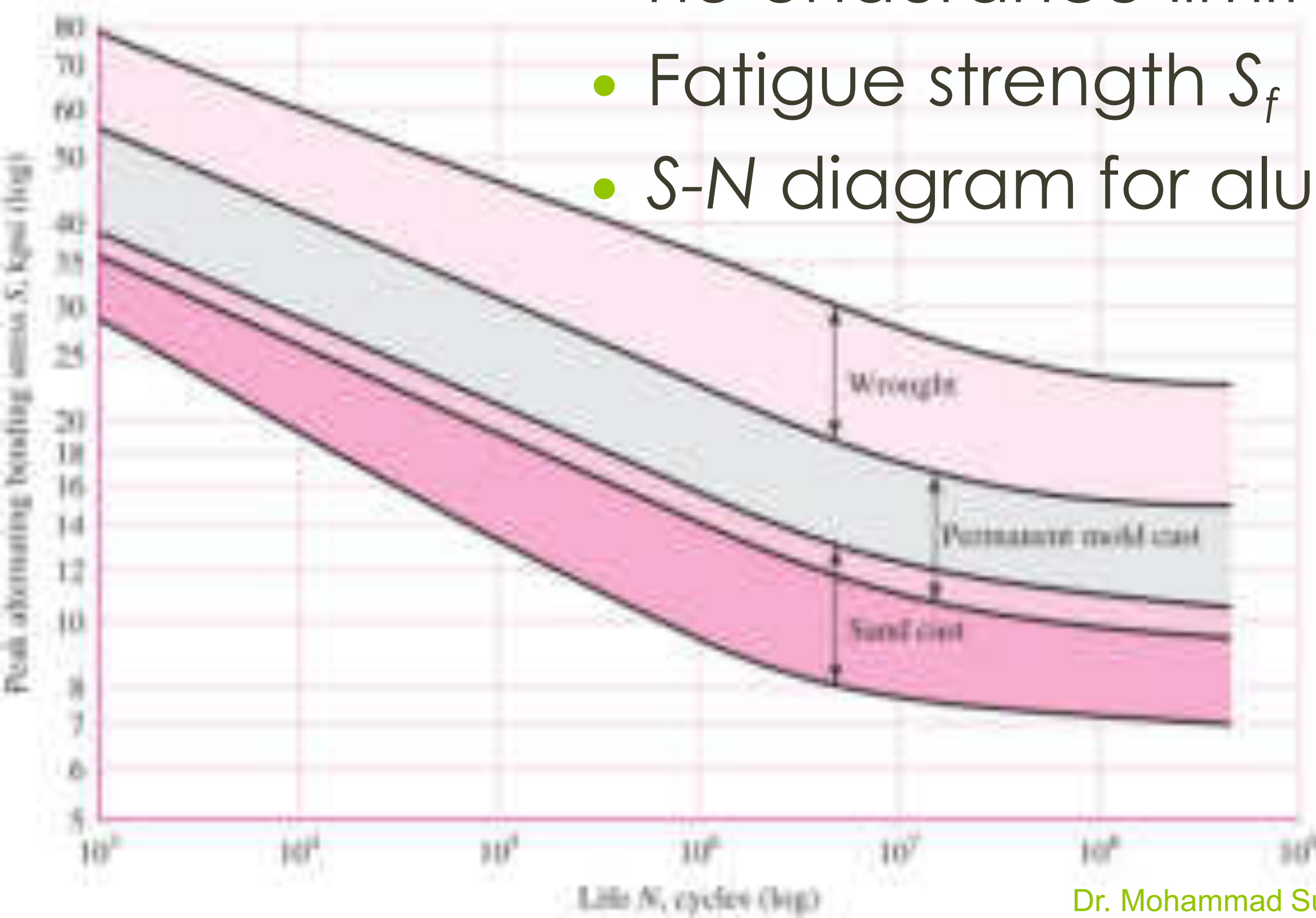

Low cycle
High cycle
Finite life
Infinite life
S_{ut}
100
50
S_e
Fatigue strength S_f, kpsi
10^0 10^1 10^2 10^3 10^4 10^5 10^6 10^7 10^8
Number of stress cycles, N

S-*N* Diagram for Steel

- Stress levels below S_e predict infinite life
- 10^3 to 10^6 cycles: finite life
- Below 10^3 cycles: *low cycle*
  - ✓ quasi-static
  - ✓ Yielding usually occurs before fatigue

S-N Diagram for Nonferrous Metals

- no endurance limit
- Fatigue strength S_f
- *S-N* diagram for aluminums



2. Strain-Life Method

- Detailed analysis of plastic deformation at localized regions
- Compounding of several idealizations leads to significant uncertainties in numerical results
- Useful for explaining nature of fatigue

2. Strain-Life Method

- Fatigue failure begins at a local discontinuity
- When stress at discontinuity exceeds elastic limit, plastic strain occurs
- Cyclic plastic strain can change elastic limit, leading to fatigue

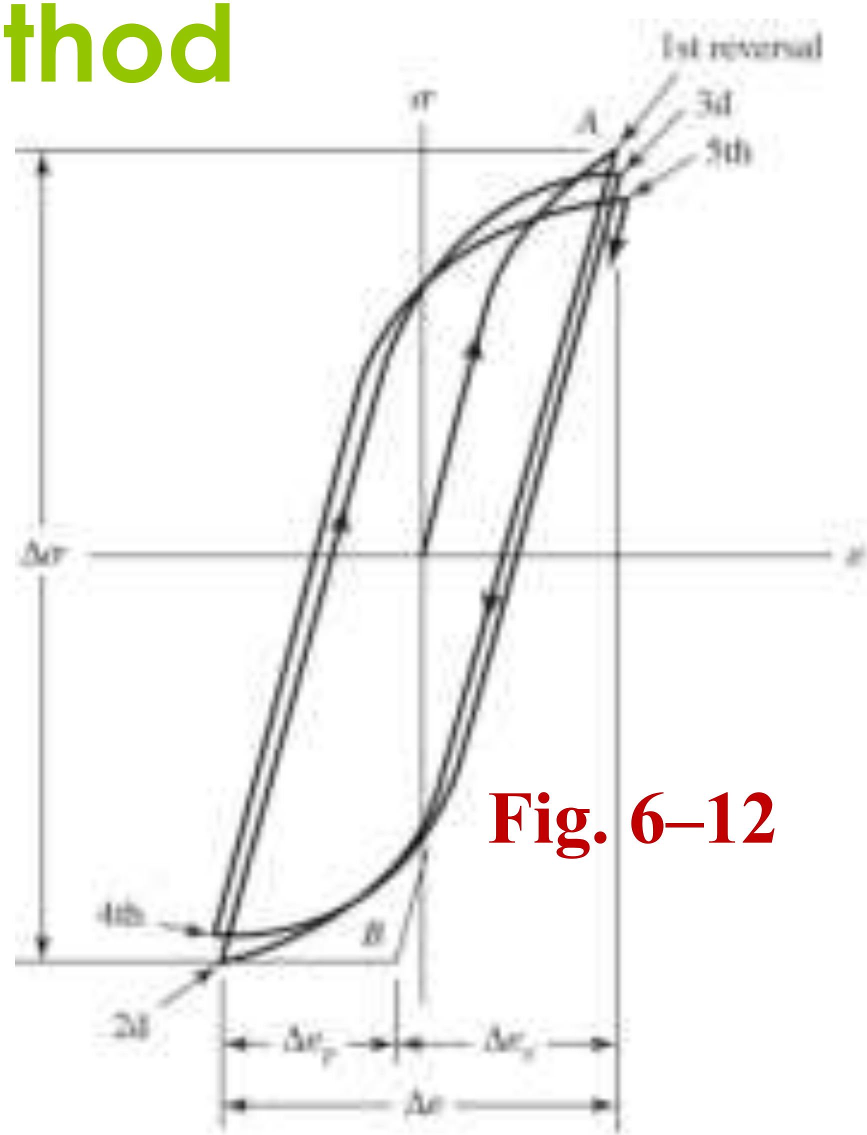
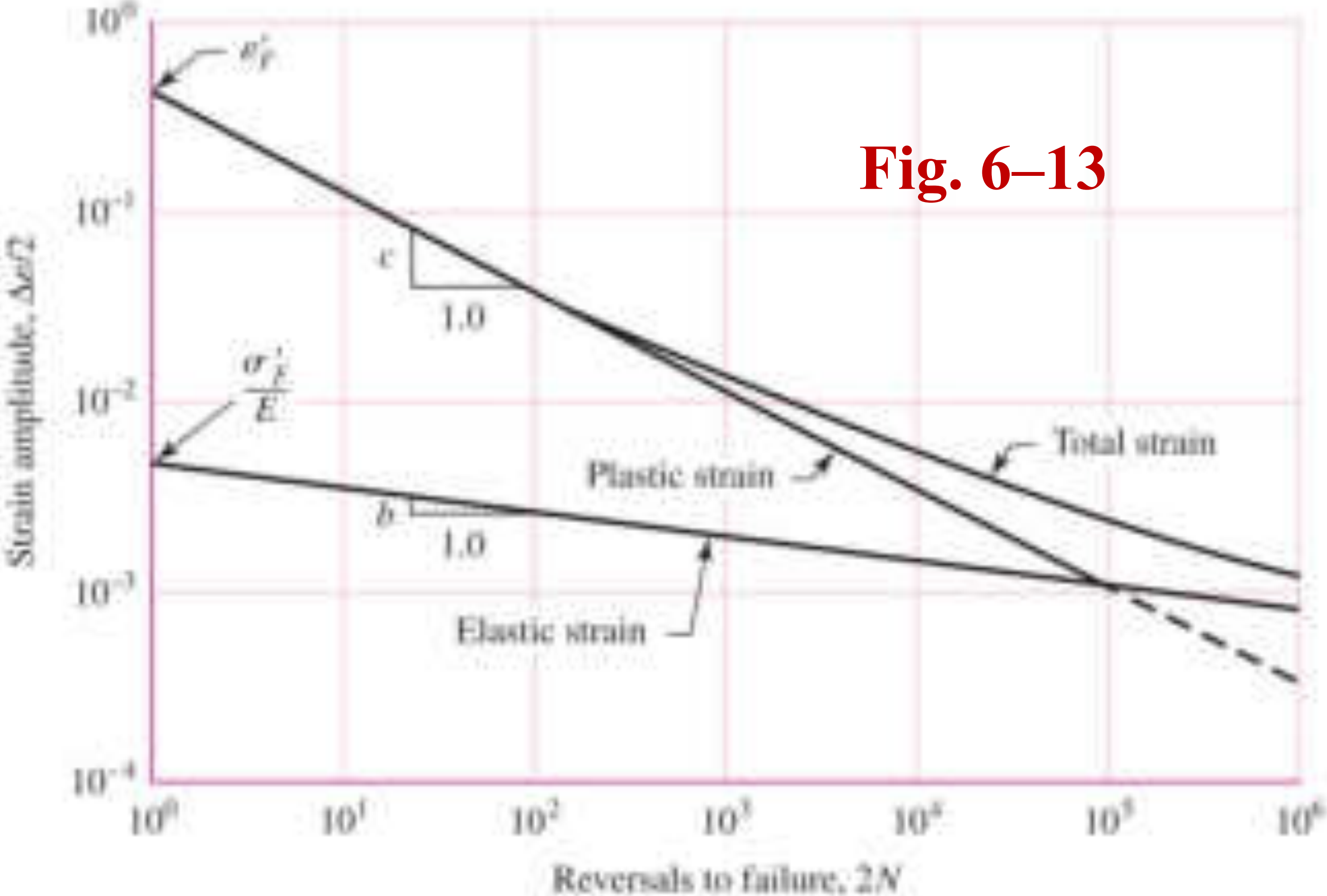


Fig. 6–12

Relation of Fatigue Life to Strain

- **Figure 6–13:** relationship of fatigue life to true-strain amplitude
- ***Fatigue ductility coefficient*** ε'_F = true strain at which fracture occurs in one reversal (point A in Fig. 6–12)
- ***Fatigue strength coefficient*** σ'_F = true stress corresponding to fracture in one reversal (point A in Fig. 6–12)

Relation of Fatigue Life to Strain



Relation of Fatigue Life to Strain

- Equation of plastic-strain line in Fig. 6–13

$$\frac{\Delta\varepsilon_p}{2} = \varepsilon'_F (2N)^c \qquad (6\text{–}1)$$

- Equation of elastic strain line in Fig. 6–13

$$\frac{\Delta\varepsilon_e}{2} = \frac{\sigma'_F}{E}(2N)^b \qquad (6\text{–}2)$$

$$\frac{\Delta\varepsilon}{2} = \frac{\Delta\varepsilon_e}{2} + \frac{\Delta\varepsilon_p}{2}$$

$$\frac{\Delta\varepsilon}{2} = \frac{\sigma'_F}{E}(2N)^b + \varepsilon'_F (2N)^c \qquad (6\text{–}3)$$

Relation of Fatigue Life to Strain

- ***Fatigue ductility exponent c*** = slope of plastic-strain line
- 2*N* stress reversals = *N* cycles
- ***Fatigue strength exponent b*** = slope of elastic-strain line

Relation of Fatigue Life to Strain

$$\frac{\Delta\varepsilon}{2} = \frac{\sigma'_F}{E}(2N)^b + \varepsilon'_F(2N)^c \qquad (6\text{–}3)$$

- **Manson-Coffin:** relationship between fatigue life and total strain
- **Table A–23:** values of coefficients & exponents
- Equation has **limited use for design**
  - ✓ Values for total strain at discontinuities are not readily available

The Endurance Limit

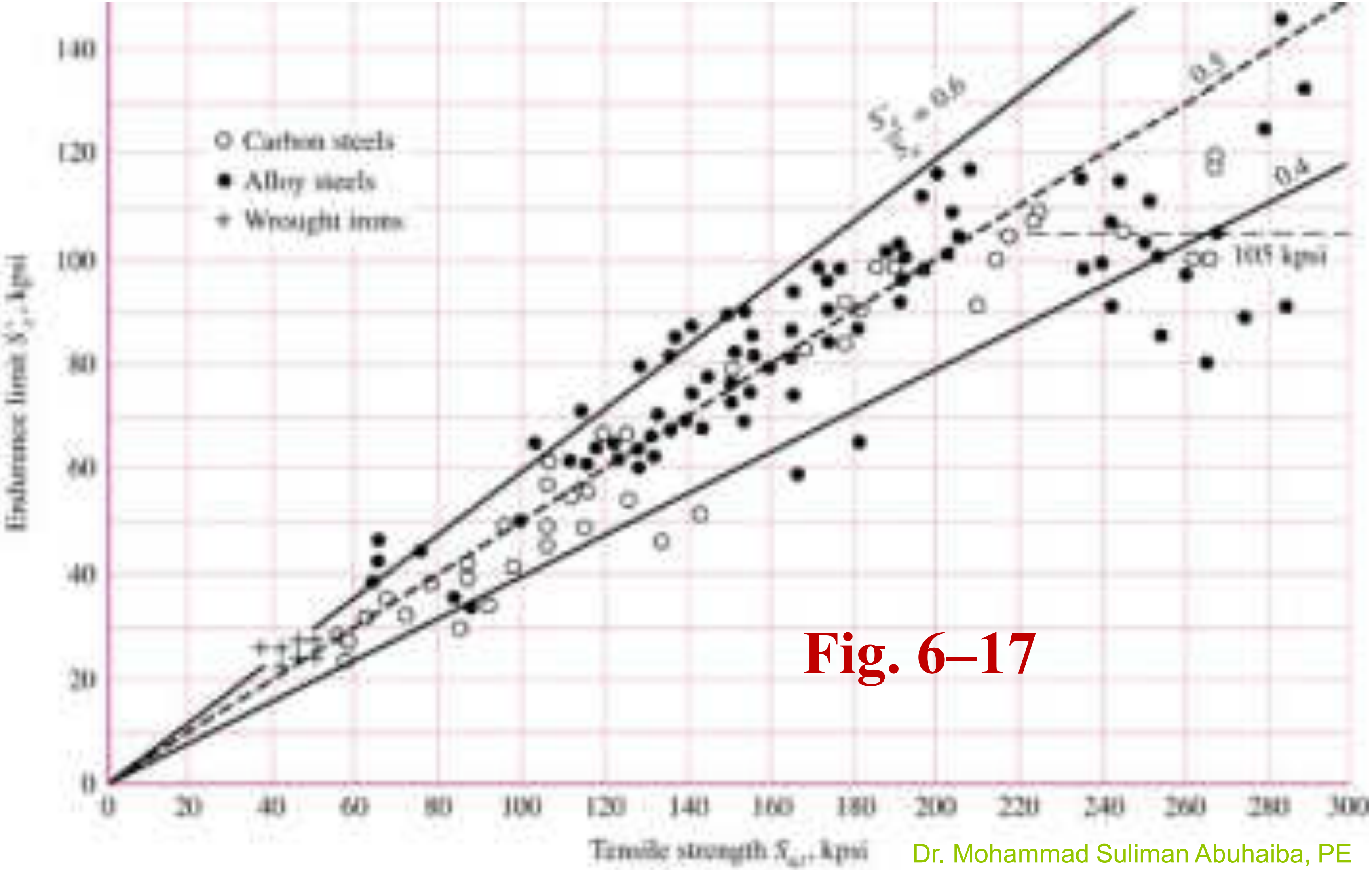


Fig. 6–17

The Endurance Limit

Simplified estimate of endurance limit for steels for the rotating-beam specimen, S'_e

$$S'_e = \begin{cases} 0.5S_{ut} & S_{ut} \le 200 \text{ kpsi } (1400 \text{ MPa}) \\ 100 \text{ kpsi} & S_{ut} > 200 \text{ kpsi} \\ 700 \text{ MPa} & S_{ut} > 1400 \text{ MPa} \end{cases} \qquad (6\text{–}8)$$

Fatigue Strength

Dr. Mohammad Suliman Abuhaiba, PE

- For design, an approximation of idealized *S-N* diagram is desirable.
- To estimate fatigue strength at 10^3 cycles, start with Eq. (6-2)

$$\frac{\Delta\varepsilon_e}{2} = \frac{\sigma'_F}{E}(2N)^b \qquad (6\text{-}2)$$

- Define specimen fatigue strength at a specific number of cycles as

$$(S'_f)_N = E\Delta\varepsilon_e/2$$

$$(S'_f)_N = \sigma'_F(2N)^b \qquad (6\text{-}9)$$

Fatigue Strength

- At 10^3 cycles, $(S'_f)_{10^3} = \sigma'_F (2 \cdot 10^3)^b = f\, S_{ut}$
- f = fraction of S_{ut} represented by $(S'_f)_{10^3}$

$$f = \frac{\sigma'_F}{S_{ut}} (2 \cdot 10^3)^b \qquad (6\text{–}10)$$

- SAE approximation for steels with $H_B \leq 500$ may be used.

$$\sigma'_F = S_{ut} + 50 \text{ kpsi}$$

$$\sigma'_F = S_{ut} + 345 \text{ MPa}$$

Fatigue Strength

- To find *b*, substitute endurance strength and corresponding cycles into Eq. (6–9) and solve for *b*

$$b = -\frac{\log\left(\sigma'_F / S'_e\right)}{\log\left(2N_e\right)} \qquad (6\text{–}12)$$

Fatigue Strength

$$(S'_f)_N = \sigma'_F (2N)^b \qquad (6\text{–}9)$$

$$f = \frac{\sigma'_F}{S_{ut}} (2 \cdot 10^3)^b \qquad (6\text{–}10)$$

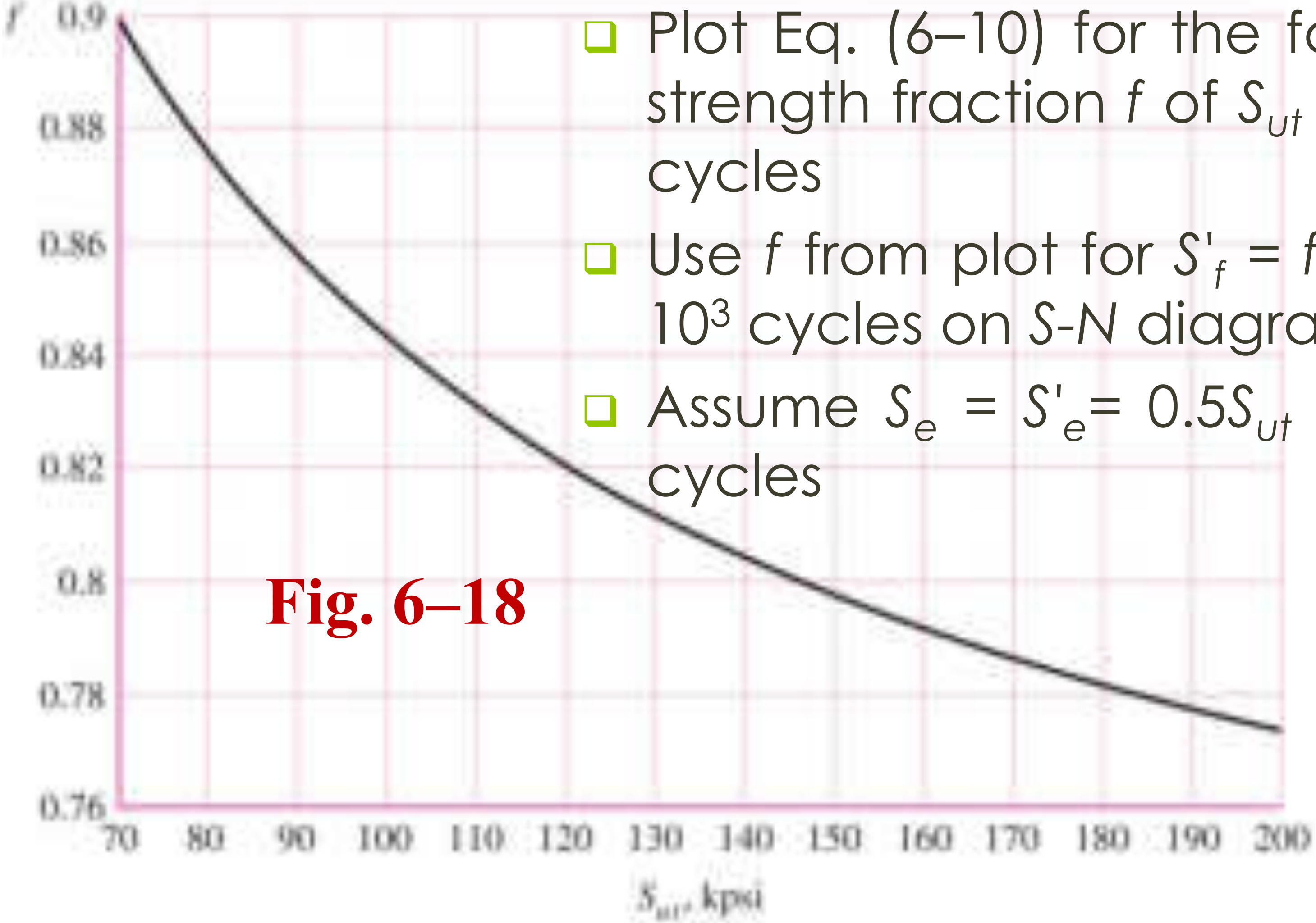
$$\sigma'_F = S_{ut} + 50 \text{ kpsi}$$

$$\sigma'_F = S_{ut} + 345 \text{ MPa}$$

$$b = -\frac{\log(\sigma'_F / S'_e)}{\log(2N_e)} \qquad (6\text{–}12)$$

Substitute Eqs. 6–11 & 6–12 into Eqs. 6–9 and 6–10 to obtain expressions for S'_f and f

Fatigue Strength Fraction *f*



- Plot Eq. (6–10) for the fatigue strength fraction *f* of S_{ut} at 10^3 cycles
- Use *f* from plot for $S'_f = f\ S_{ut}$ at 10^3 cycles on *S-N* diagram
- Assume $S_e = S'_e = 0.5S_{ut}$ at 10^6 cycles

Fig. 6–18

Equations for *S-N* Diagram

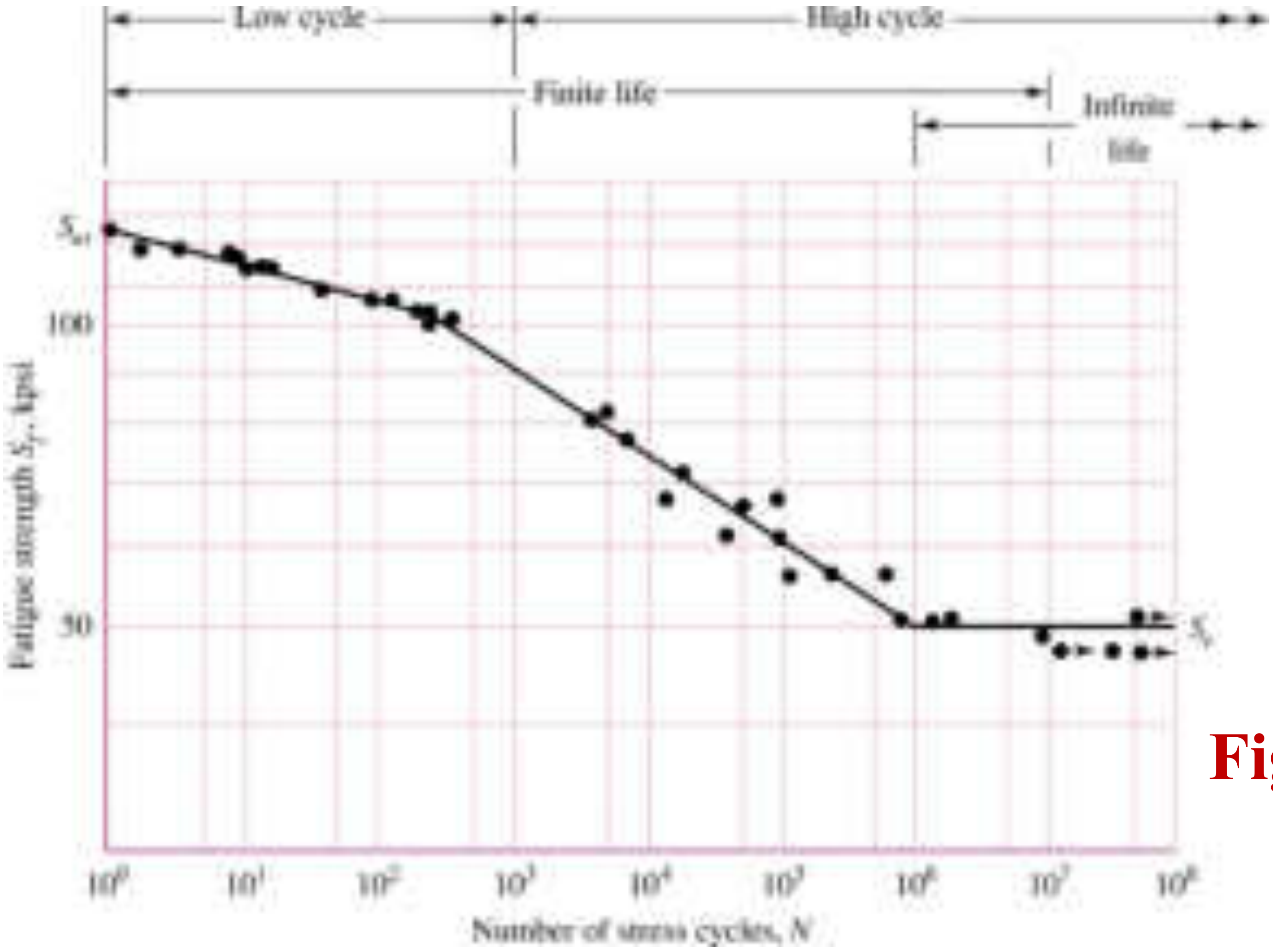


Fig. 6–10

Equations for *S-N* Diagram

- Write equation for *S-N* line from 10^3 to 10^6 cycles
- Two known points
  - ✓ At $N = 10^3$ cycles, $S_f = f\,S_{ut}$
  - ✓ At $N = 10^6$ cycles, $S_f = S_e$
- Equations for line:

$$S_f = a\,N^b \qquad (6\text{–}13)$$

$$a = \frac{(f\,S_{ut})^2}{S_e} \qquad (6\text{–}14)$$

$$b = -\frac{1}{3}\log\left(\frac{f\,S_{ut}}{S_e}\right) \qquad (6\text{–}15)$$

Low-cycle Fatigue

- $1 \le N \le 10^3$
- On the idealized *S*-*N* diagram on a log-log scale, failure is predicted by a straight line between two points (10^3, $f\, S_{ut}$) and (1, S_{ut})

$$S_f \ge S_{ut} N^{(\log f)/3} \qquad 1 \le N \le 10^3 \qquad (6\text{–}17)$$

Example 6-2

Given a 1050 HR steel, *estimate*

a. the rotating-beam endurance limit at 10^6 cycles.
b. the endurance strength of a polished rotating-beam specimen corresponding to 10^4 cycles to failure
c. the expected life of a polished rotating-beam specimen under a completely reversed stress of 55 kpsi.

Endurance Limit Modifying Factors

- Endurance limit S'_e is for carefully prepared and tested specimen
- If warranted, S_e is obtained from testing of actual parts
- When testing of actual parts is not practical, a set of *Marin factors* are used to adjust the endurance limit

Endurance Limit Modifying Factors

$$S_e = k_a k_b k_c k_d k_e k_f S'_e \quad (6\text{–}18)$$

k_a = surface condition modification factor

k_b = size modification factor

k_c = load modification factor

k_d = temperature modification factor

k_e = reliability factor[11]

k_f = miscellaneous-effects modification factor

S'_e = rotary-beam test specimen endurance limit

S_e = endurance limit at the critical location of a machine part in the geometry and condition of use

Surface Factor k_a

- Surface factor is a function of ultimate strength
- Higher strengths are more sensitive to rough surfaces

$$k_a = aS_{ut}^b \qquad (6\text{–}19)$$

Table 6–2

Parameters for Marin Surface Modification Factor, Eq. (6–19)

Surface Finish	Factor a S_{ut}, kpsi	Factor a S_{ut}, MPa	Exponent b
Ground	1.34	1.58	−0.085
Machined or cold-drawn	2.70	4.51	−0.265
Hot-rolled	14.4	57.7	−0.718
As-forged	39.9	272.	−0.995

Example 6-3

A steel has a min ultimate strength of 520 MPa and a machined surface. Estimate k_a.

Size Factor k_b rotating & Round

- Larger parts have greater surface area at high stress levels
- Likelihood of crack initiation is higher For bending and torsion loads, the size factor is given by

$$k_b = \begin{cases} (d/0.3)^{-0.107} = 0.879d^{-0.107} & 0.11 \le d \le 2 \text{ in} \\ 0.91d^{-0.157} & 2 < d \le 10 \text{ in} \\ (d/7.62)^{-0.107} = 1.24d^{-0.107} & 2.79 \le d \le 51 \text{ mm} \\ 1.51d^{-0.157} & 51 < d \le 254 \text{ mm} \end{cases} \qquad (6\text{–}20)$$

Size Factor k_b rotating & Round

- Applies only for round, rotating diameter
- For axial load, there is no size effect,
  - ✓ $k_b = 1$

Size Factor k_b not round & rotating

- An **equivalent round rotating** diameter is obtained.
- Volume of material stressed at and above 95% of max stress **=** same volume in rotating-beam specimen.
- Lengths cancel, so **equate areas**.

Size Factor k_b not round & rotating

- For a **rotating round section, the 95% stress area** is the area of a ring,

$$A_{0.95\sigma} = \frac{\pi}{4}[d^2 - (0.95d)^2] = 0.0766d^2 \qquad (6\text{–}22)$$

- Equate 95% stress area for other conditions to Eq. (6–22) and solve for *d* as the equivalent round rotating diameter

Size Factor k_b round & not rotating

- For non-rotating round,

$$A_{0.95\sigma} = 0.01046d^2 \qquad (6\text{–}23)$$

- Equating to Eq. (6-22) and solving for equivalent diameter,

$$d_e = 0.370d \qquad (6\text{–}24)$$

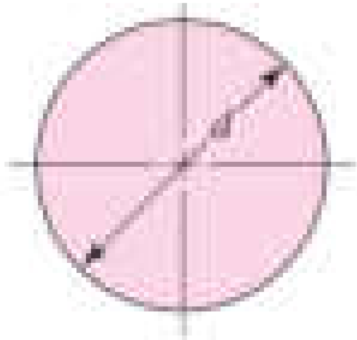
Size Factor k_b not round & not rotating

- For rectangular section h x b, $A_{95\sigma}$ = 0.05 hb. Equating to Eq. (6–22),

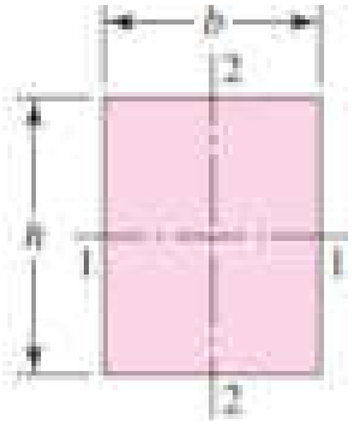
$$d_e = 0.808(hb)^{1/2} \qquad (6\text{–}25)$$

Size Factor k_b

Table 6–3: $A_{95\sigma}$ for common non-rotating structural shapes undergoing bending



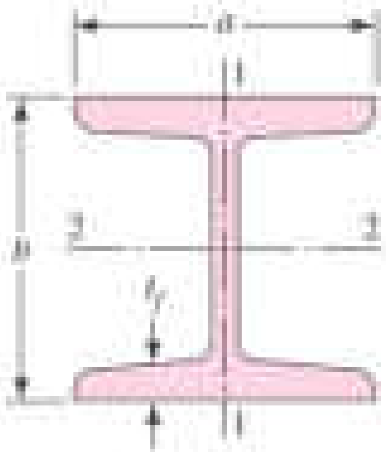
$$A_{0.95\sigma} = 0.01046d^2$$
$$d_e = 0.370d$$



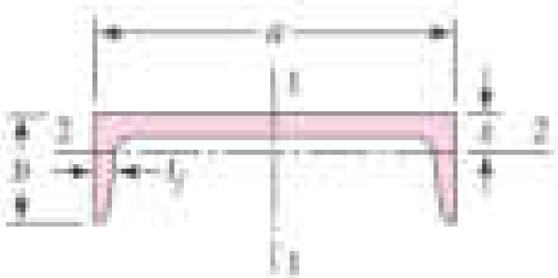
$$A_{0.95\sigma} = 0.05hb$$
$$d_e = 0.808\sqrt{hb}$$

Size Factor k_b

Table 6–3: $A_{95\sigma}$ for common non-rotating structural shapes undergoing bending



$$A_{0.95\sigma} = \begin{cases} 0.10\,a t_f & \text{axis 1-1} \\ 0.05\,b a \quad t_f > 0.025a & \text{axis 2-2} \end{cases}$$



$$A_{0.95\sigma} = \begin{cases} 0.05\,ab & \text{axis 1-1} \\ 0.052\,xa + 0.1\,t_f(b - x) & \text{axis 2-2} \end{cases}$$

Example 6-4

A steel shaft loaded in bending is 32 mm in diameter, abutting a filleted shoulder 38 mm in diameter. The shaft material has a mean ultimate tensile strength of 690 MPa. Estimate the Marin size factor k_b if the shaft is used in

a. A rotating mode.
b. A nonrotating mode.

Loading Factor k_c

- Accounts for changes in endurance limit for different types of fatigue loading.
- Only to be used for **single load types**.
- Use **Combination Loading method** (Sec. 6–14) when **more than one load type** is present.

$$k_c = \begin{cases} 1 & \text{bending} \\ 0.85 & \text{axial} \\ 0.59 & \text{torsion}^{17} \end{cases} \qquad (6\text{–}26)$$

Temperature Factor k_d

- Endurance limit appears to maintain same relation to ultimate strength for elevated temperatures as at RT
- **Table 6–4:** Effect of Operating Temperature on Tensile Strength of Steel.* (S_T = tensile strength at operating temperature (OT); S_{RT} = tensile strength at room temperature; $0.099 \leq \hat{\sigma} \leq 0.110$)

Temperature Factor k_d

Table 6–4

Temperature, °C	S_T/S_{RT}	Temperature, °F	S_T/S_{RT}
20	1.000	70	1.000
50	1.010	100	1.008
100	1.020	200	1.020
150	1.025	300	1.024
200	1.020	400	1.018
250	1.000	500	0.995
300	0.975	600	0.963
350	0.943	700	0.927
400	0.900	800	0.872
450	0.843	900	0.797
500	0.768	1000	0.698
550	0.672	1100	0.567
600	0.549		

Temperature Factor k_d

- If ultimate strength is known for OT, then just use that strength. Let $k_d = 1$.
- If ultimate strength is known only at RT, use Table 6–4 to estimate ultimate strength at OT. With that strength, let $k_d = 1$.
- Use ultimate strength at RT and apply k_d from Table 6–4 to the endurance limit.

$$k_d = \frac{S_T}{S_{RT}} \qquad (6\text{–}28)$$

Temperature Factor k_d

- A fourth-order polynomial curve fit of the data of Table 6–4 can be used in place of the table,

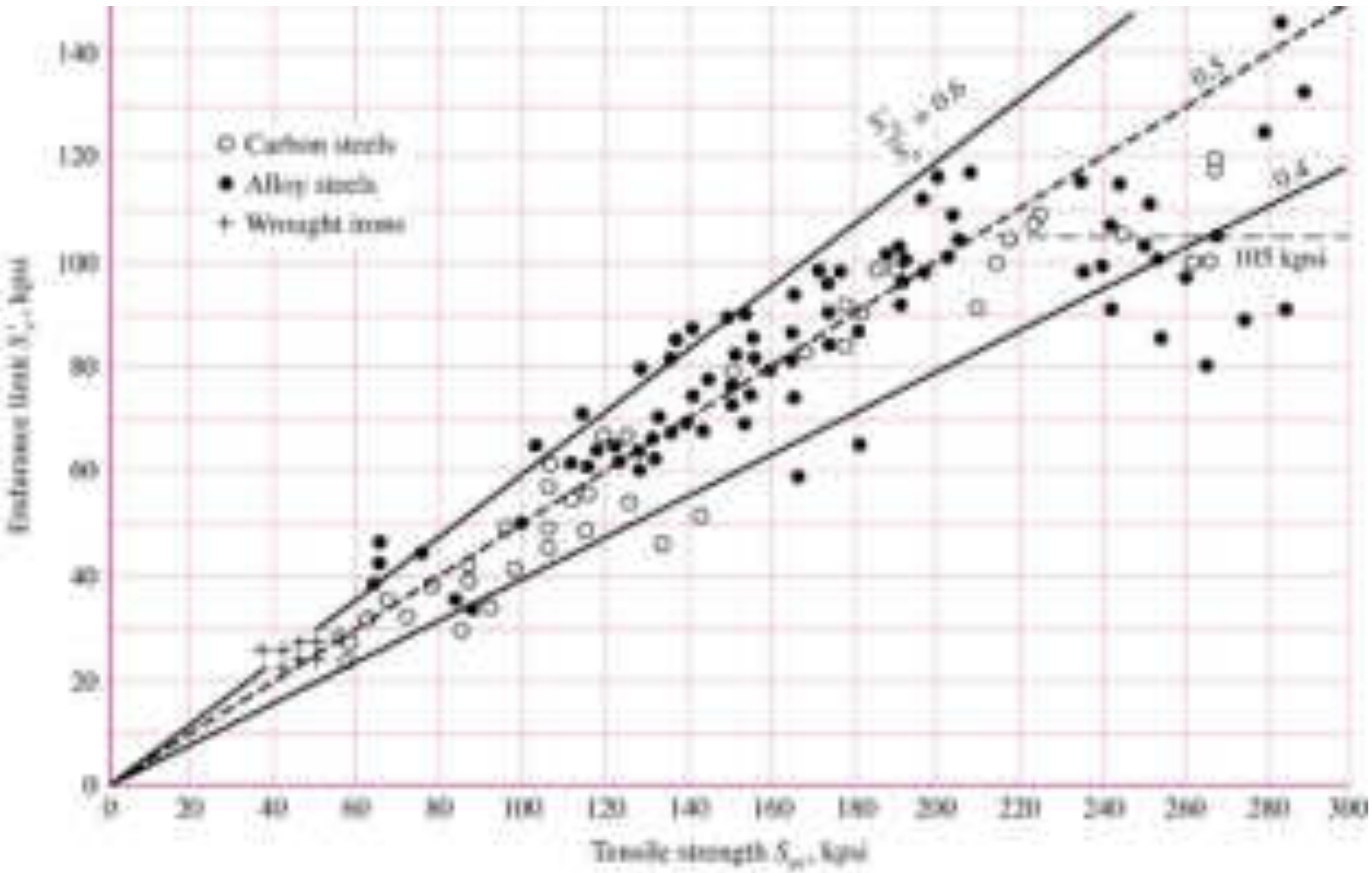
$$k_d = 0.975 + 0.432(10^{-3})T_F - 0.115(10^{-5})T_F^2 + 0.104(10^{-8})T_F^3 - 0.595(10^{-12})T_F^4 \qquad (6\text{–}27)$$

Reliability Factor k_e

- Fig. 6–17, $S'_e = 0.5\ S_{ut}$ is typical of the data and represents 50% reliability.
- Reliability factor adjusts to other reliabilities.

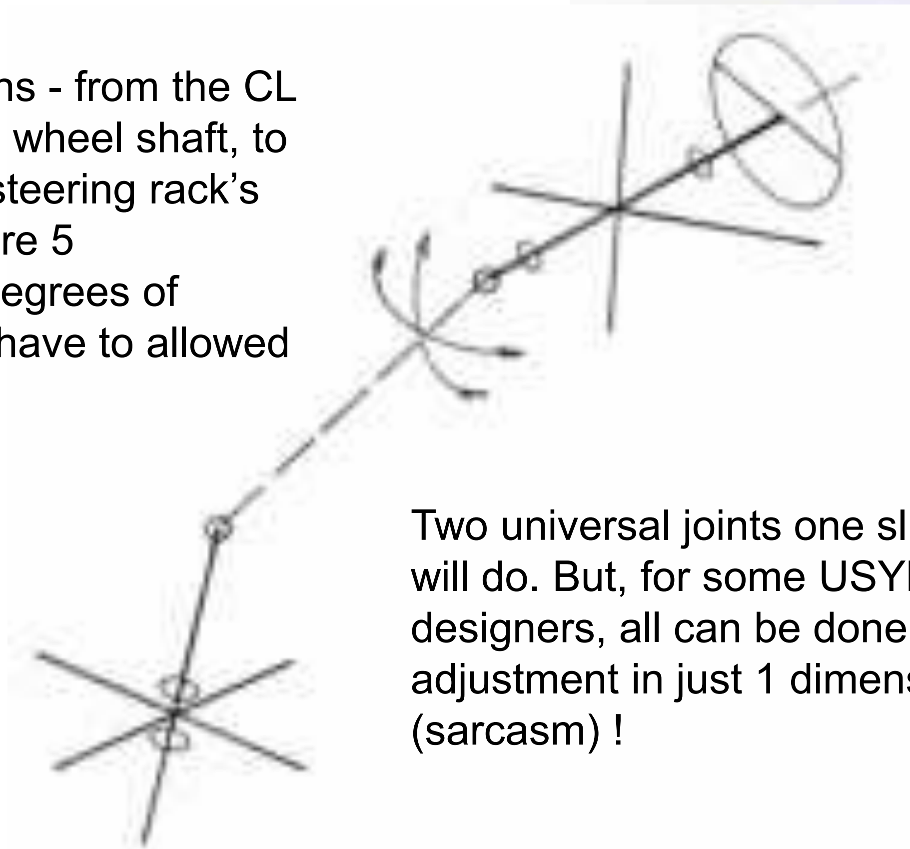
Reliability Factor k_e

Fig. 6–17



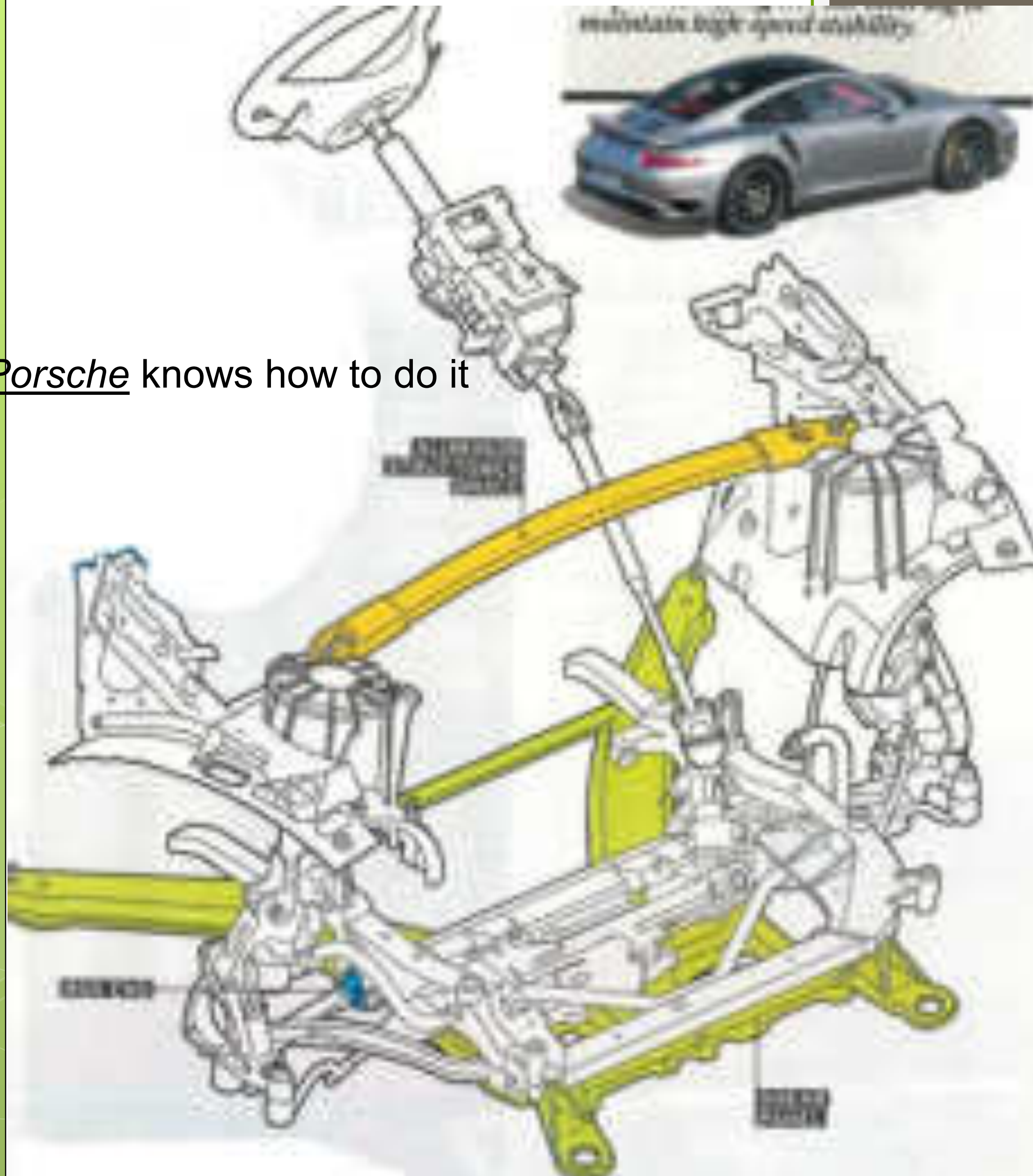
2 Dimensional misalignments

In 3 Dimensions - from the CL of the steering wheel shaft, to the CL of the steering rack's pinion, there are 5 independent degrees of freedom, that have to allowed for, exactly !



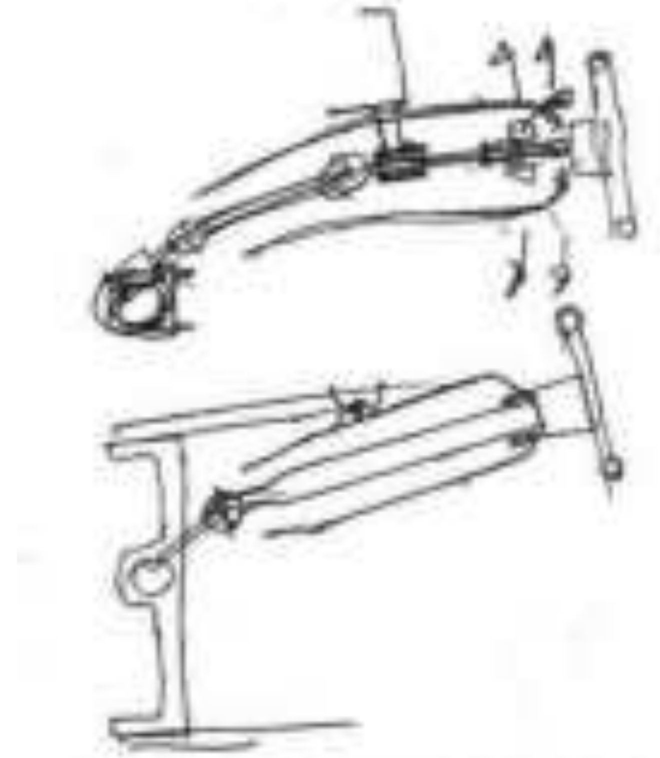
In 2016 for the first time, we used the coupling shown here, in our steering

Two universal joints one sliding joint will do. But, for some USYD designers, all can be done with an adjustment in just 1 dimension (sarcasm) !

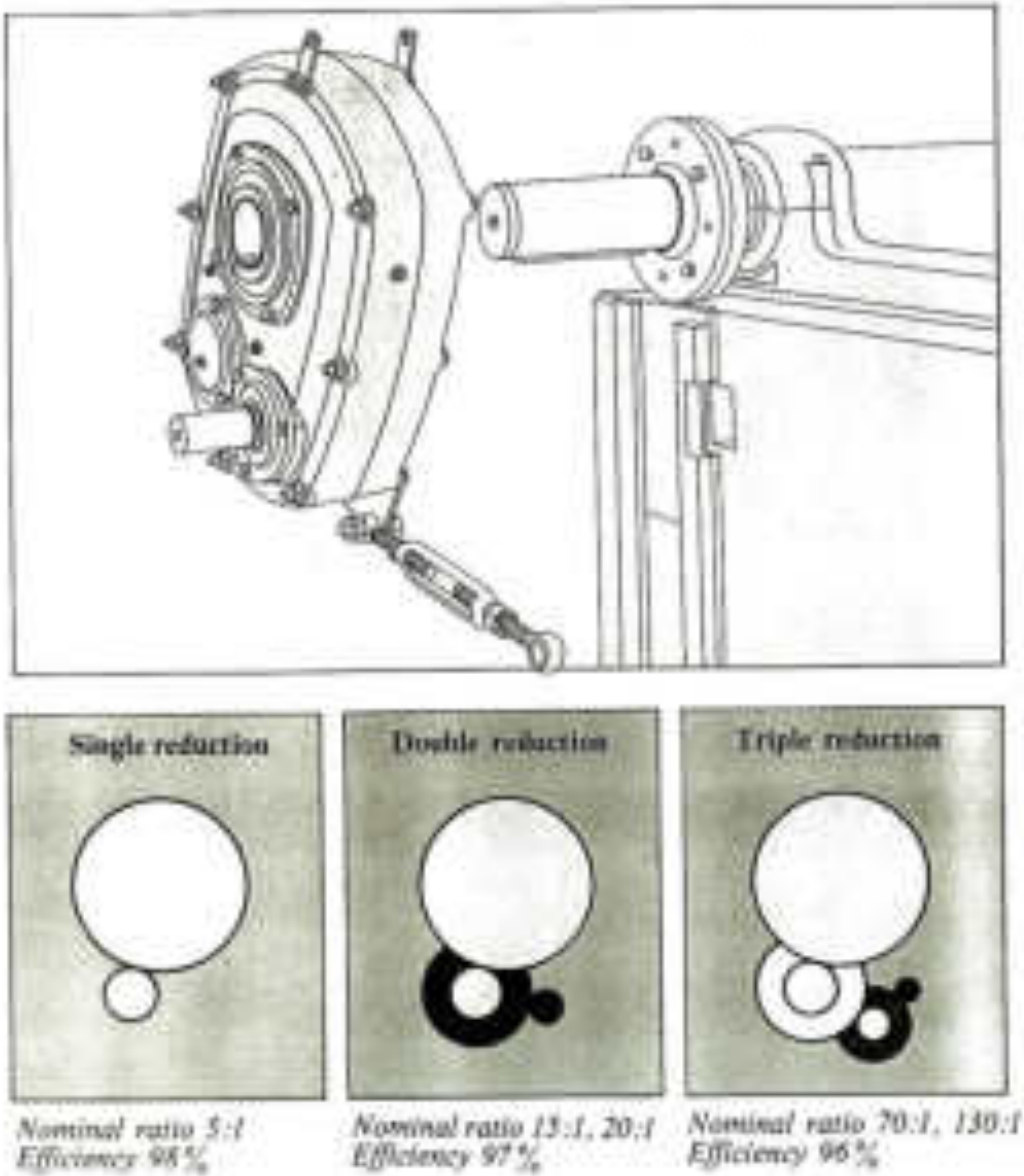


Porsche knows how to do it

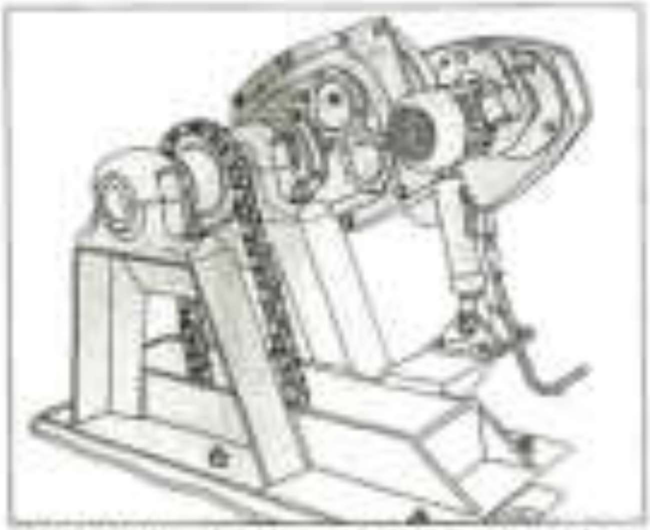
Even *Gordon Murry* an Englishman, for the McLaren F1 team understood the problem and the solution.



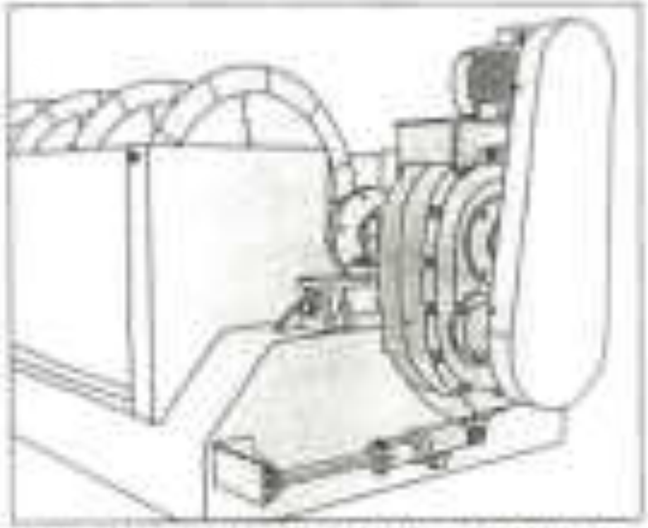
In F1 $12000 is spent to save 1 kg (the Economist magazine).



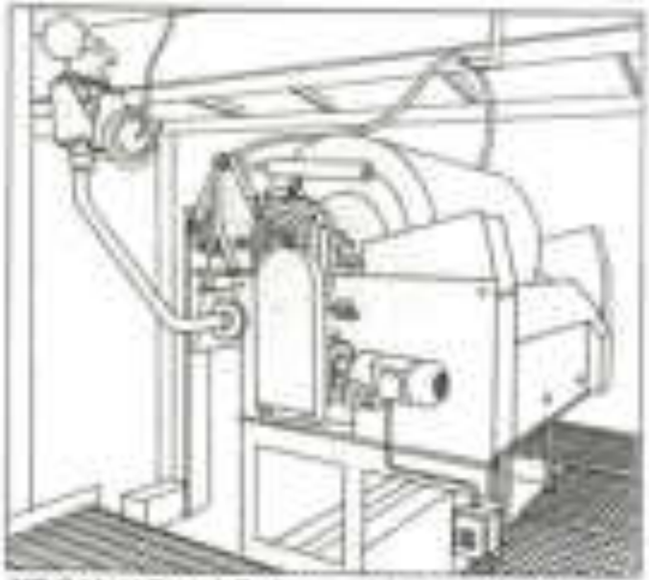
The use of shaft mounted gearbox as shown here, have become very popular. This is largely because if the box and the shaft are not mounted separately their alignment is difficult to guarantee. The weight due to the box typically causes an insignificant increase in shaft stresses.



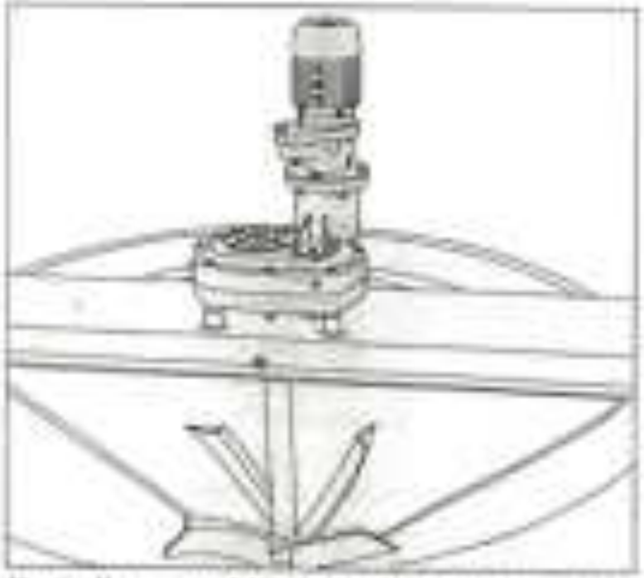
NT-SALA Speed Reducer TV 72 SC with overload release driving a chain scraper in a water treatment plant.



Screw classifier with NT-SALA Speed Reducer TV 201 and overload release.

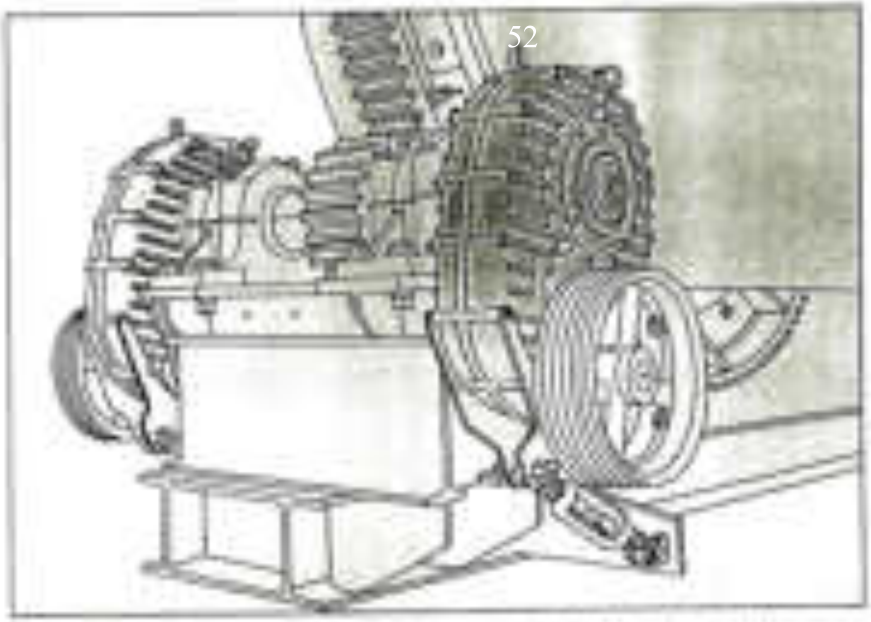


NT-SALA Speed Reducer with variator for filter drive in the mining industry.

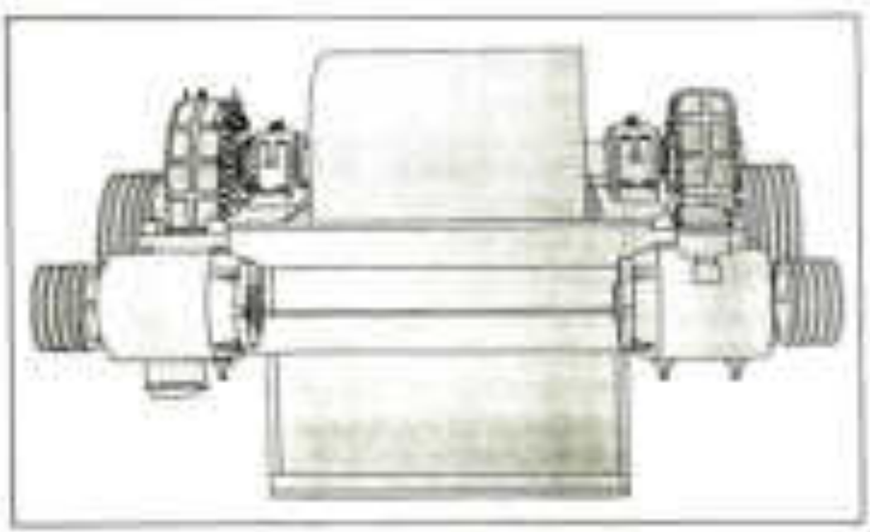


Vertically mounted NT-SALA Speed Reducer with flange connected gear motor used on agitator.

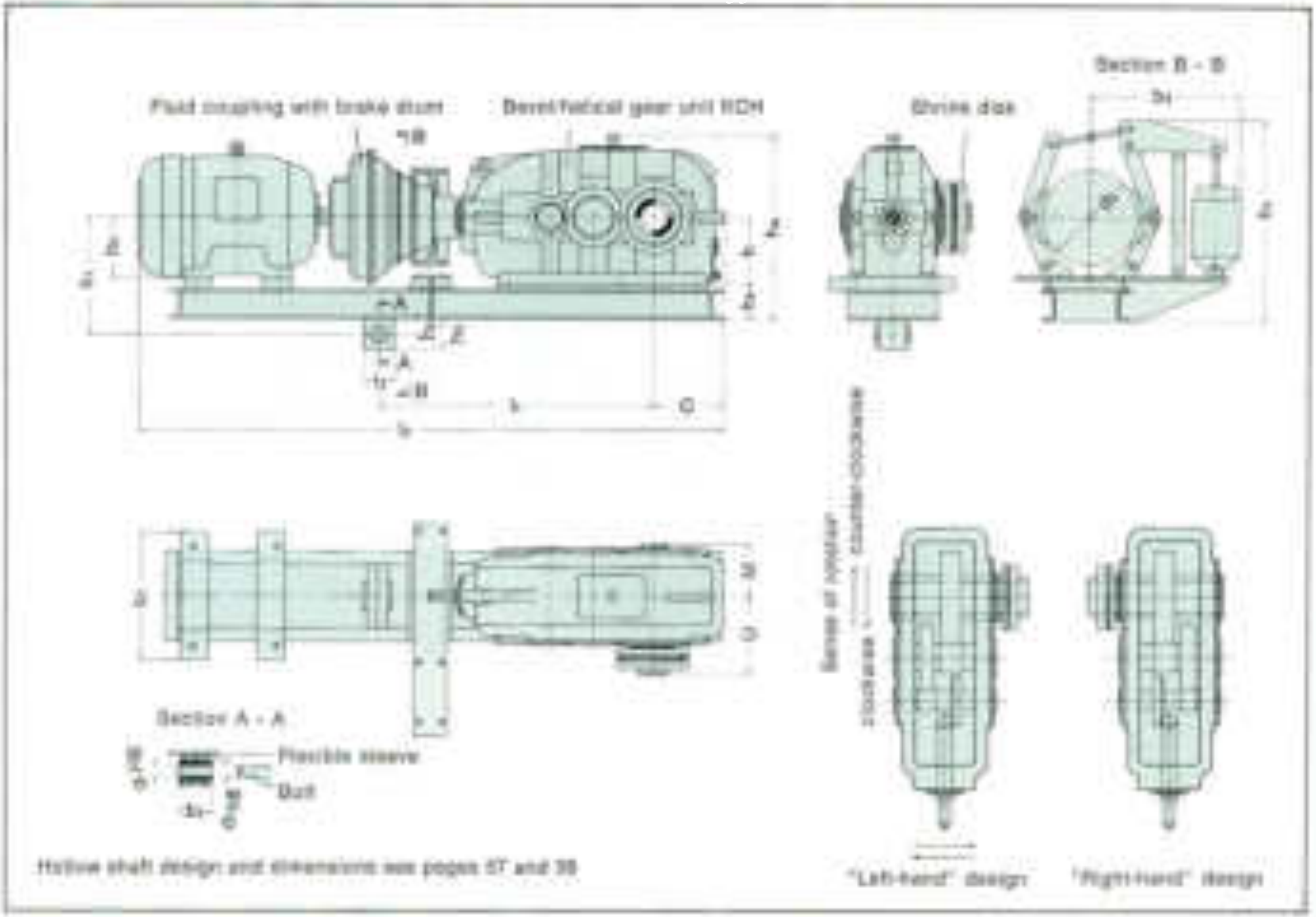
Dual drive



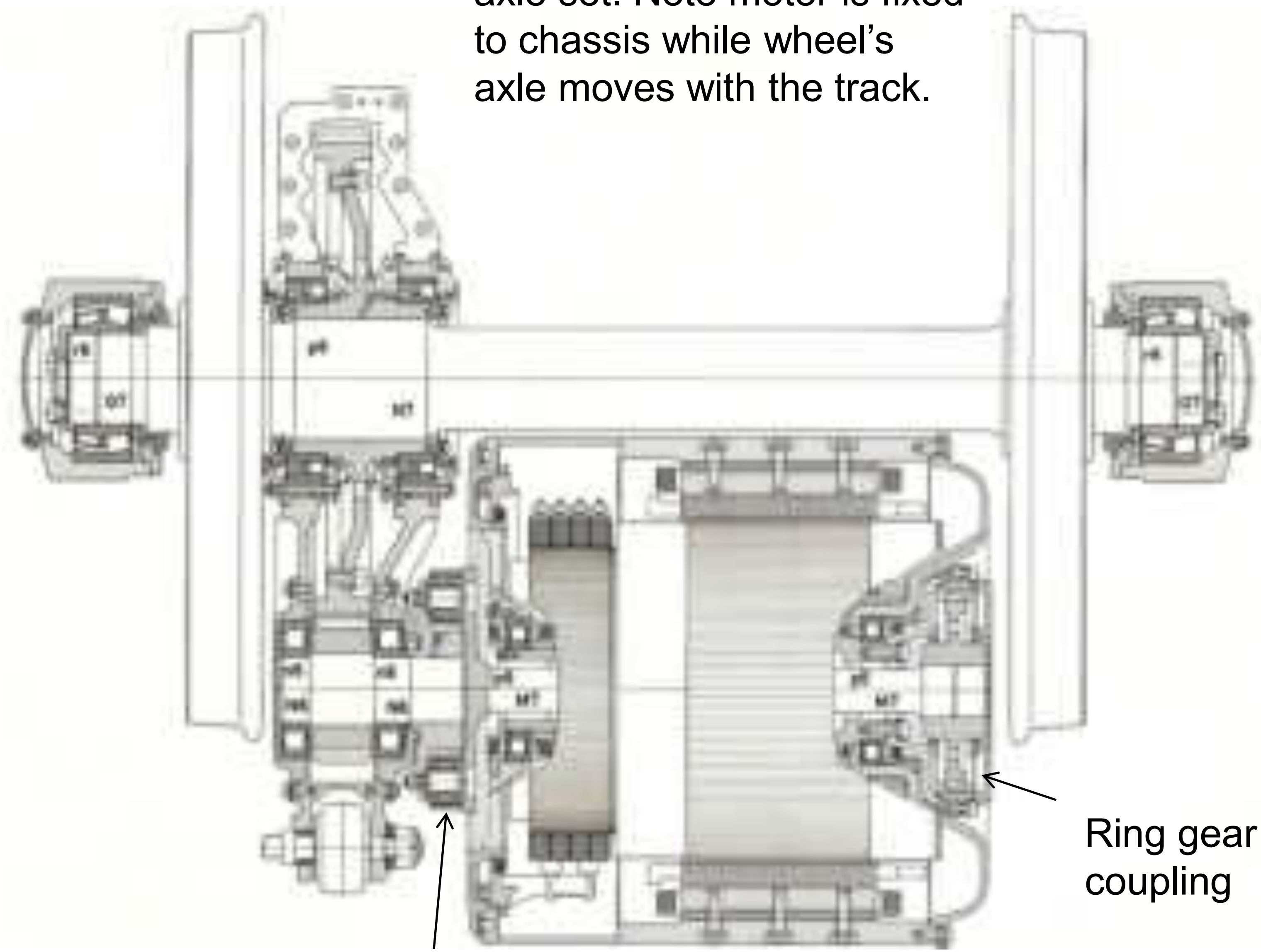
Two NT-SALA TV 802 Speed Reducers with dual 37 kW 1470 RPM electric motors, driving a 30 m long, 2.5 m diameter drying drum.



Typical dual drive for conveyor using two NT-SALA TV 402 Speed Reducers.

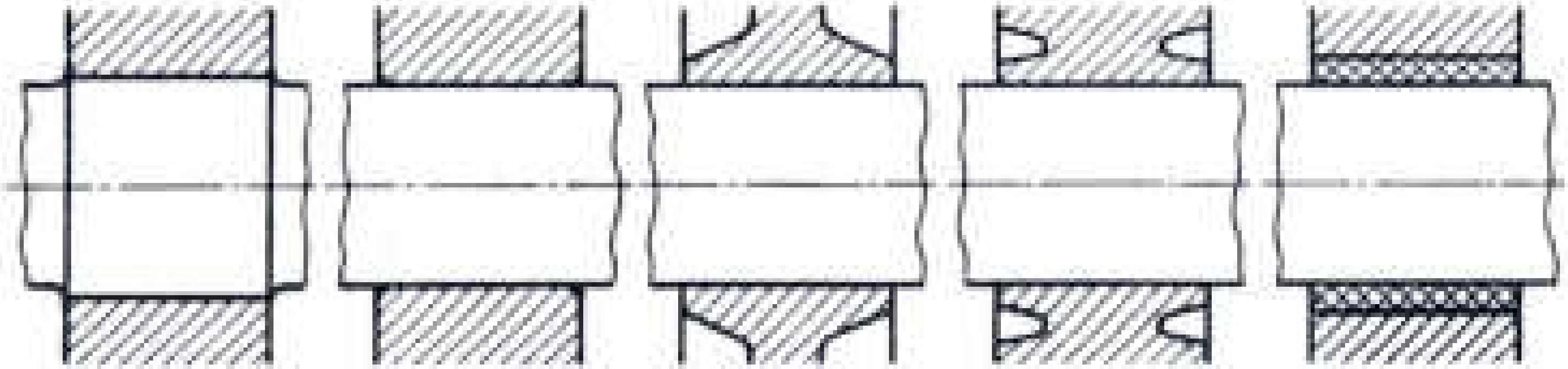

Fluid coupling with brake drum
Bevel/helical gear unit RDH
Shrink disc
Section B - B
Section A - A
Flexible sleeve
Bolt
Sense of rotation
clockwise
counter-clockwise
"Left-hand" design
"Right-hand" design

Locomotive drive motor and axle set. Note motor is fixed to chassis while wheel's axle moves with the track.

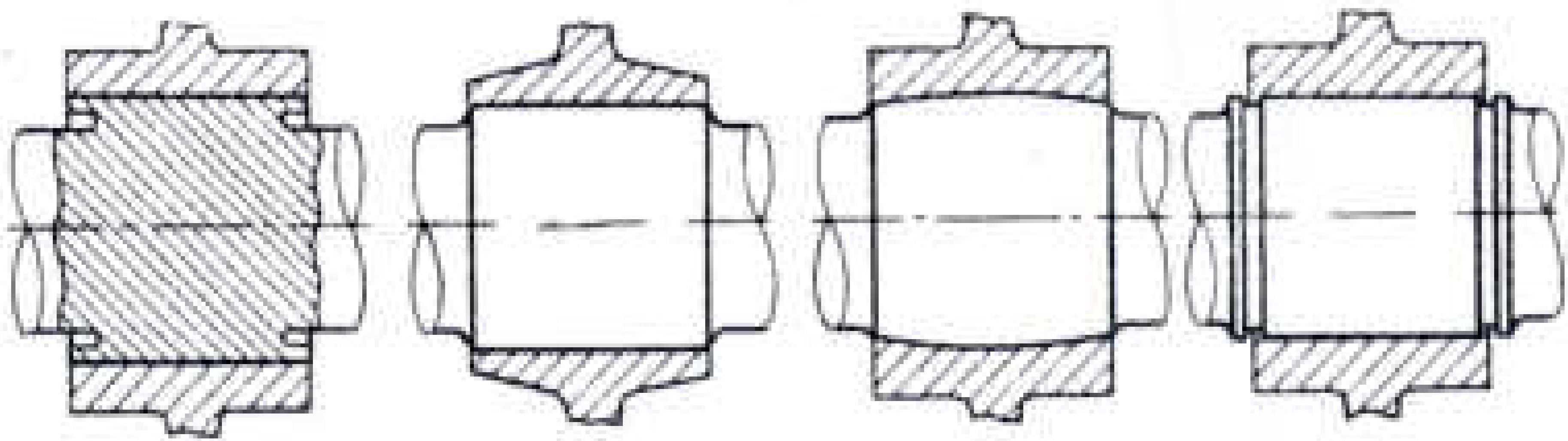


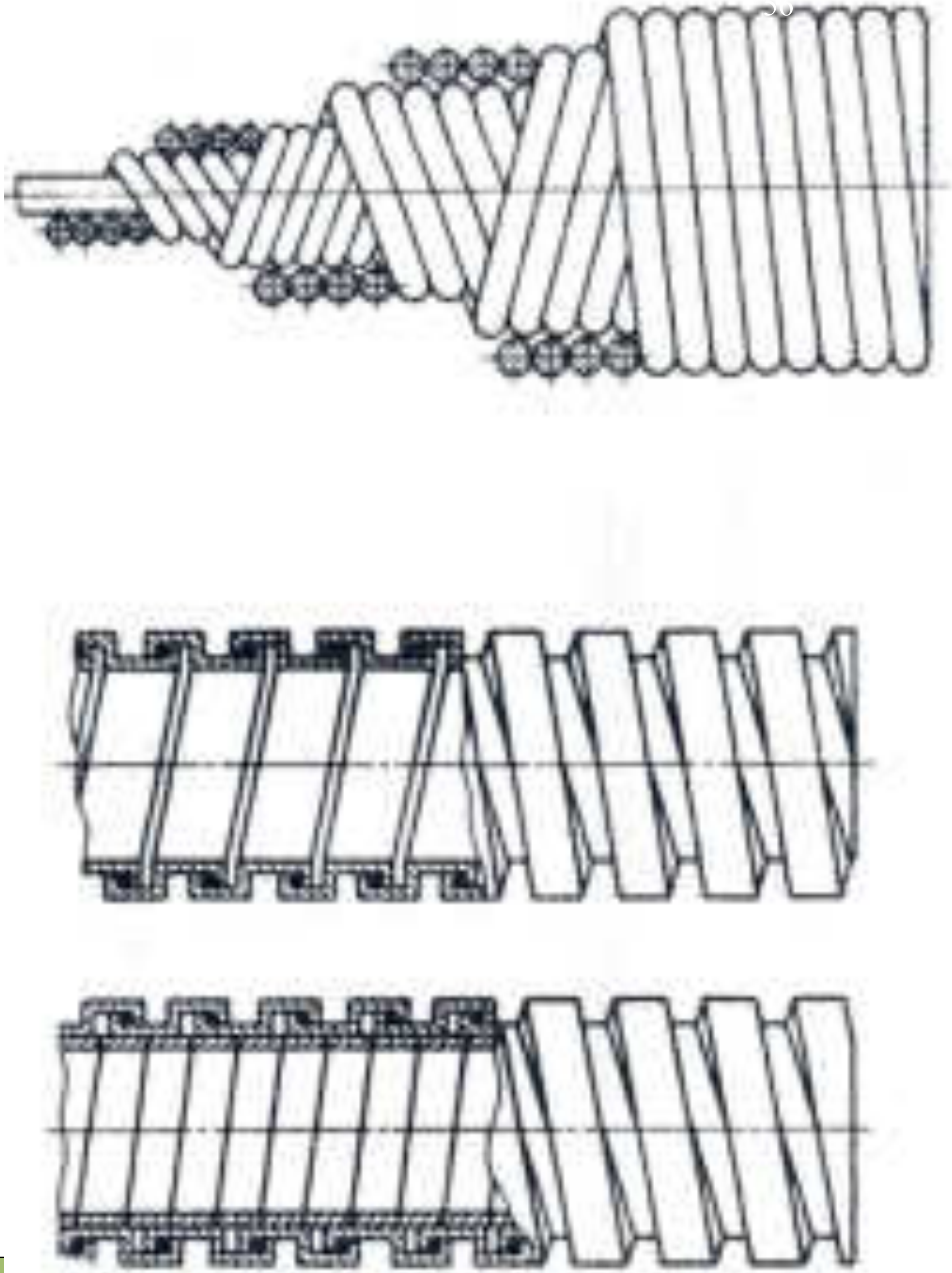
Ring gear coupling

Pin and rubber coupling



Means of reducing bending moments and stress concentration where a shaft enters a boss or housing





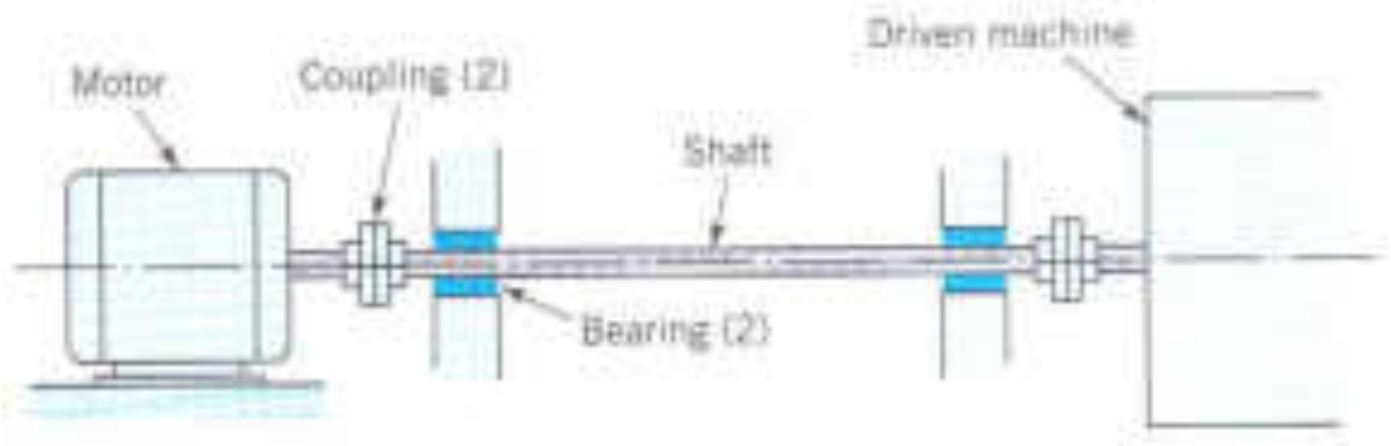
In nearly all shafts deflection is closely constrained, yet there are instances where the shaft is designed to be rigid in torsion but not in bending.

In some instances some specific bending is part of the shaft function, and allowed in its analysis.

Design of Shaft

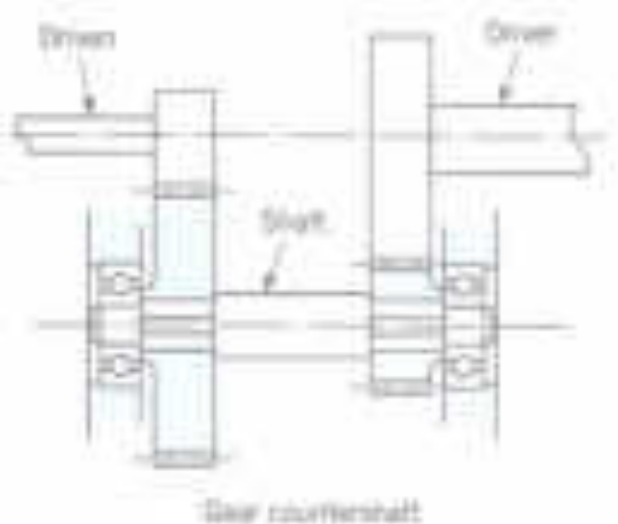
- A shaft is a rotating member usually of circular cross-section (solid or hollow), which transmits power and rotational motion.
- Machine elements such as gears, pulleys (sheaves), flywheels, clutches, and sprockets are mounted on the shaft and are used to transmit power from the driving device (motor or engine) through a machine.
- Press fit, keys, dowel, pins and splines are used to attach these machine elements on the shaft.
- The shaft rotates on rolling contact bearings or bush bearings.
- Various types of retaining rings, thrust bearings, grooves and steps in the shaft are used to take up axial loads and locate the rotating elements.
- Couplings are used to transmit power from drive shaft (e.g., motor) to the driven shaft (e.g.

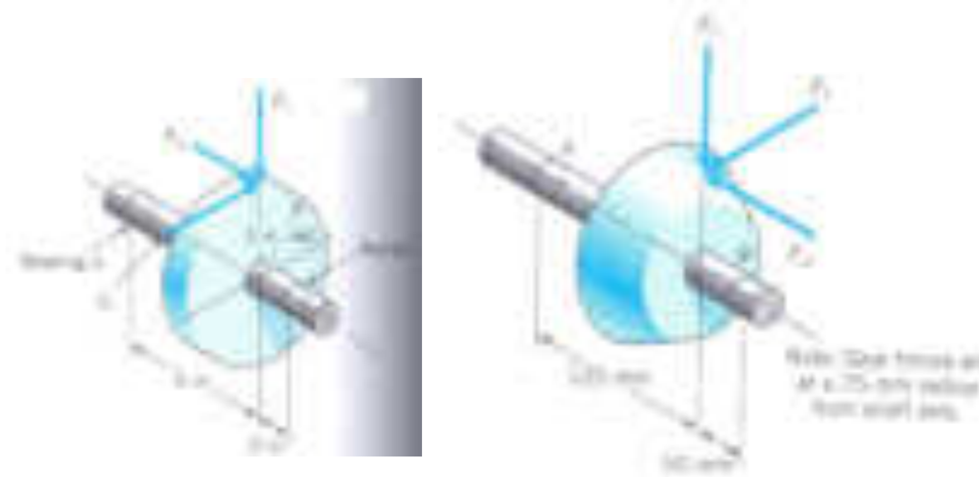
The connecting shaft is loaded primarily in torsion.



(a) Connecting shaft

Combined bending and torsion loads on shaft: Shaft carrying gears.





From power and rpm find the torque (T), which gives rise to shear stress.

From Torque (T) and diameter (d), find $F_t = 2T/d$. From F_t and pressure angles of gears you can find F_r and F_a.

F_r and F_t are orthogonal to each other and are both transverse forces to the shaft axis, which will give rise to normal bending stress in the shaft. When shaft rotates, bending stress changes from tensile to compressive and then compressive to tensile, ie, completely reversing state of stress.

F_a will give rise to normal axial stress in the shaft.

Loads on shaft due to pulleys

Pulley torque (T) = Difference in belt tensions in the tight (t_1) and slack (t_2) sides of a pulley times the radius (r), ie

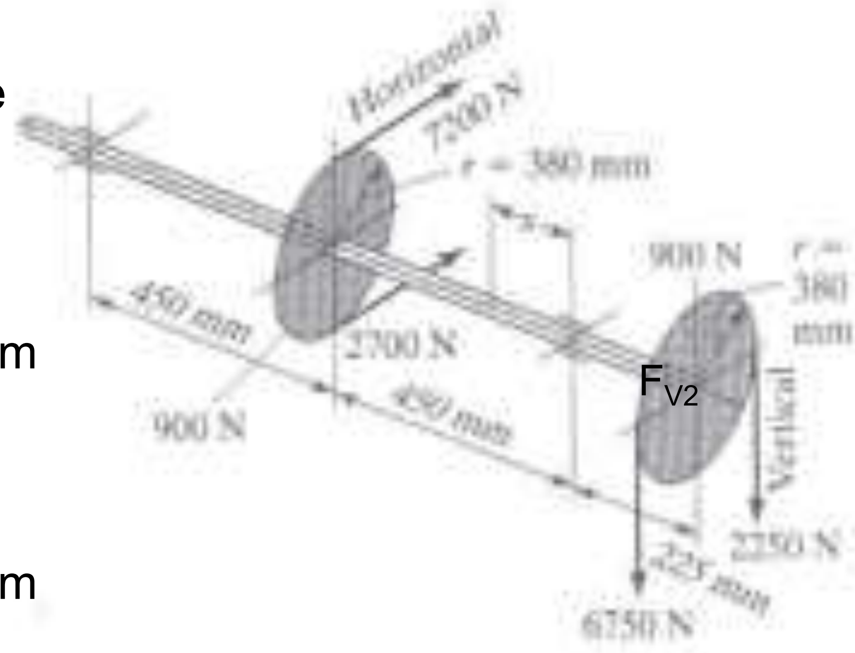
$T = (t_1 - t_2) x r$

Left pulley torque

T_1 = (7200-2700)x380=1,710,000 N-mm

Right pulley has exactly equal and opposite torque:

T_2 = (6750-2250)x380=1,710,000 N-mm

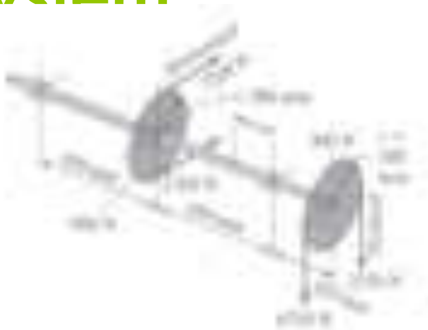


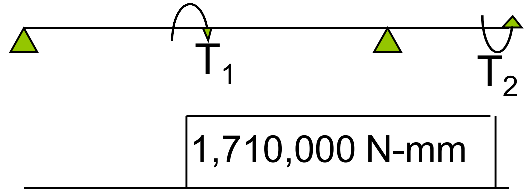
Bending forces in vertical (F_v) and horizontal (F_H) directions:

At the left pulley: F_{V1}=900N; F_{H1}=7200+2700 = 9900N

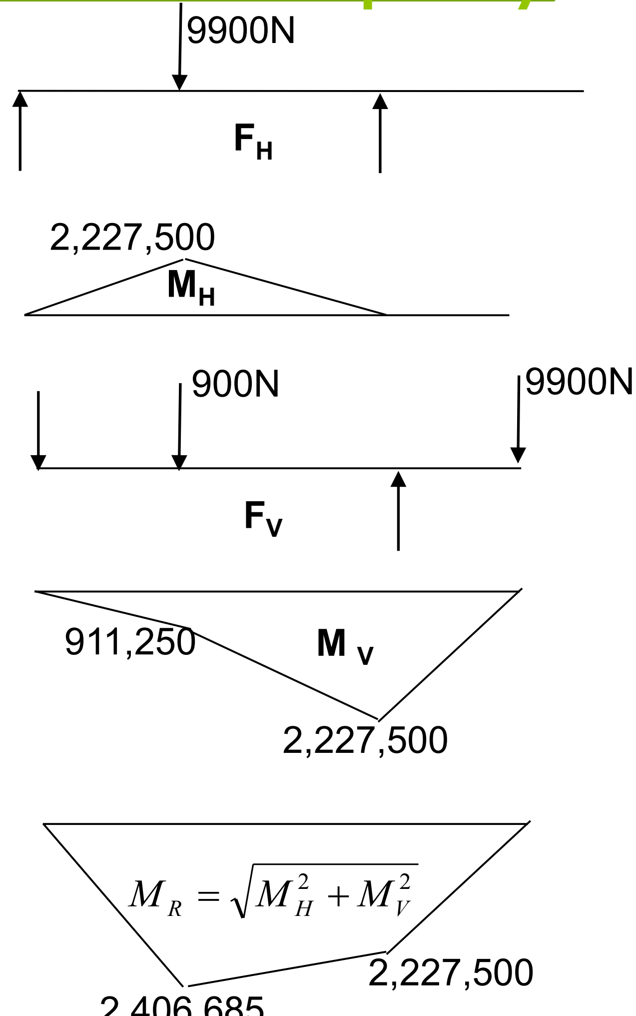
At the right pulley: F_{V2}=900+6750+2250=9900N; F_{H2}=0

Torque and Bending moment diagrams for the pulley system





Torque diag.



Resultant bending moment

From Horizontal forces (F_H) and vertical forces (F_v), Bending moments M_H & M_V are drawn separately.

Then the resultant moments at various points on the shaft can be found from

$$M_R = \sqrt{M_H^2 + M_V^2}$$

The section of shaft where the left pulley is located has obviously the highest combination of Torque (1,710,000 N-mm) and Bending moment (2,406,685 N-mm)

Power, toque & speed

For linear motion:

Power = F.v (force x velocity)

For rotational motion

Power P = Torque x angular velocity

= T (in-lb).ω (rad/sec) in-lb/sec

= T.(2 π n/60) in-lb/sec [n=rpm]

= T.(2 π n/(60*12*550)) HP [HP=550 ft-lb/sec]

= T.n/63,025 HP

or, T= 63,025HP/n (in-lb), where n = rpm

Similarly, T= 9,550,000kW/n (N-mm), where n = rpm

Shear (τ) and bending (σ) stresses on the outer surface of a shaft, for a torque (T) and bending moment (M)

For solid circular section:

$$\tau = \frac{T}{J} r = \frac{T}{(\pi d^4 / 32)} \frac{d}{2} = \frac{16T}{\pi d^3}$$

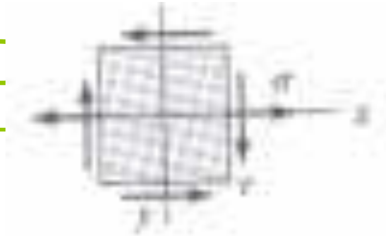
$$\sigma = \frac{M}{I} r = \frac{M}{(\pi d^4 / 64)} \frac{d}{2} = \frac{32M}{\pi d^3}$$

For hollow circular section:

$$\tau = \frac{T}{J} r = \frac{T}{(\pi (d_0^4 - d_i^4)/32)} \frac{d_o}{2} = \frac{16T}{\pi (d_0^4 - d_i^4)} d_o = \frac{16T}{\pi d_0^3 (1-\lambda^4)} \quad where, \lambda = \frac{d_i}{d_o}$$

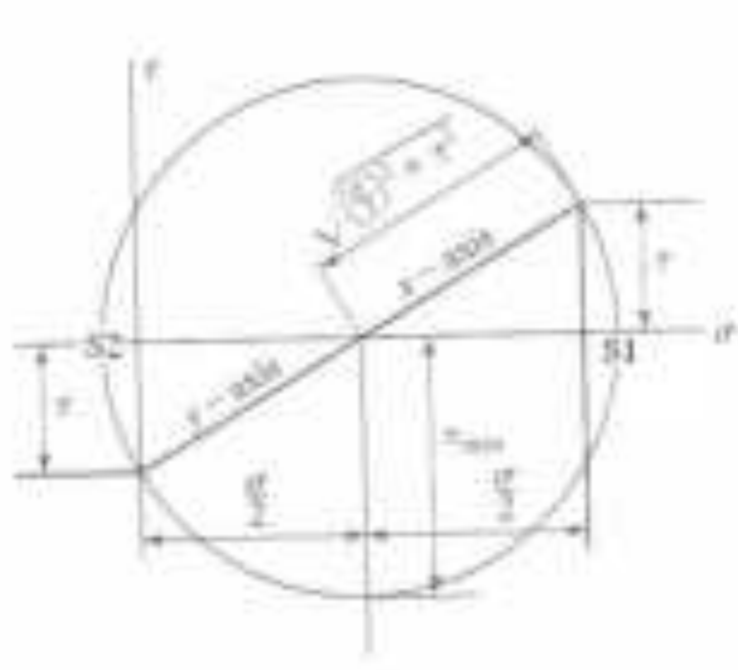
$$\sigma = \frac{M}{I} r = \frac{M}{(\pi (d_0^4 - d_i^4)/64)} \frac{d_o}{2} = \frac{32M}{\pi (d_0^4 - d_i^4)} d_o = \frac{32M}{\pi d_0^3 (1-\lambda^4)} \quad where, \lambda = \frac{d_i}{d_o}$$

Principal Normal Stresses and Max D
Energy Failure criterion for non-rota



The stress at a point on the shaft is normal stress (s) in X direction and shear stress (t) in XY plane.

From Mohr Circle:



$$S_1 = \frac{\sigma}{2} + \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2} \; and \; S_2 = \frac{\sigma}{2} - \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

Max Distortion Energy theory:

$$S_1^2 + S_2^2 - S_1 S_2 \leq \left(\frac{S_{yp}}{N_{fs}}\right)^2$$

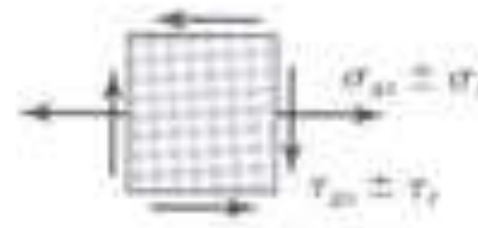
Putting values of S_1 & S_2 and simplifying:

$$\sigma^2 + 3\tau^2 \leq \left(\frac{S_{yp}}{N_{fs}}\right)^2$$

This is the design equation for non rotating shaft

Design of rotating shafts and fatigue consideration

The most frequently encountered stress situation for a rotating shaft is to have completely reversed bending and steady torsional stress. In other situations, a shaft may have a reversed torsional stress along with reversed bending stress.



The most generalized situation the rotating shaft may have both steady and cyclic components of bending stress (s_{av},s_r) and torsional stress (t_{av},t_r).

From Soderberg's fatigue criterion, the equivalent static bending and torsional stresses are:

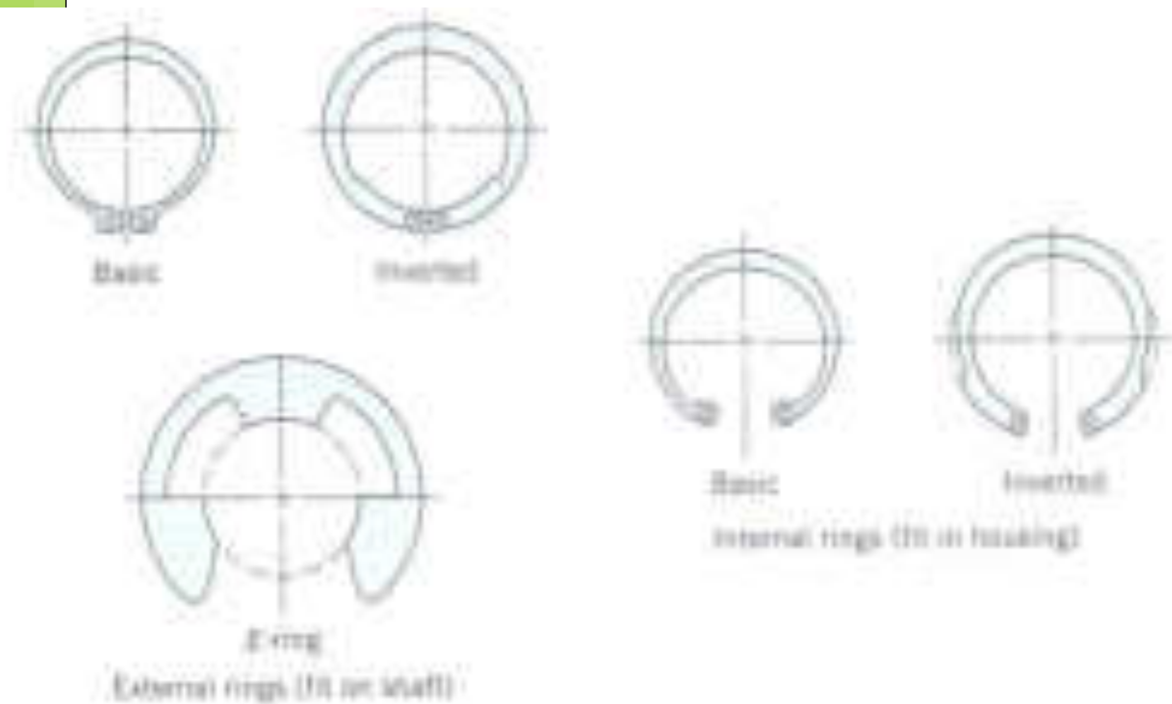
$$\sigma_{equivalent} = \sigma_{av} + \sigma_r K_{fb}\left(\frac{S_{yp}}{S_e}\right)$$

$$\tau_{equivalent} = \tau_{av} + \tau_r K_{ft}\left(\frac{S_{yp}}{S_e}\right)$$

Using these equivalent static stresses in our static design equation, the equation for rotating shaft is:

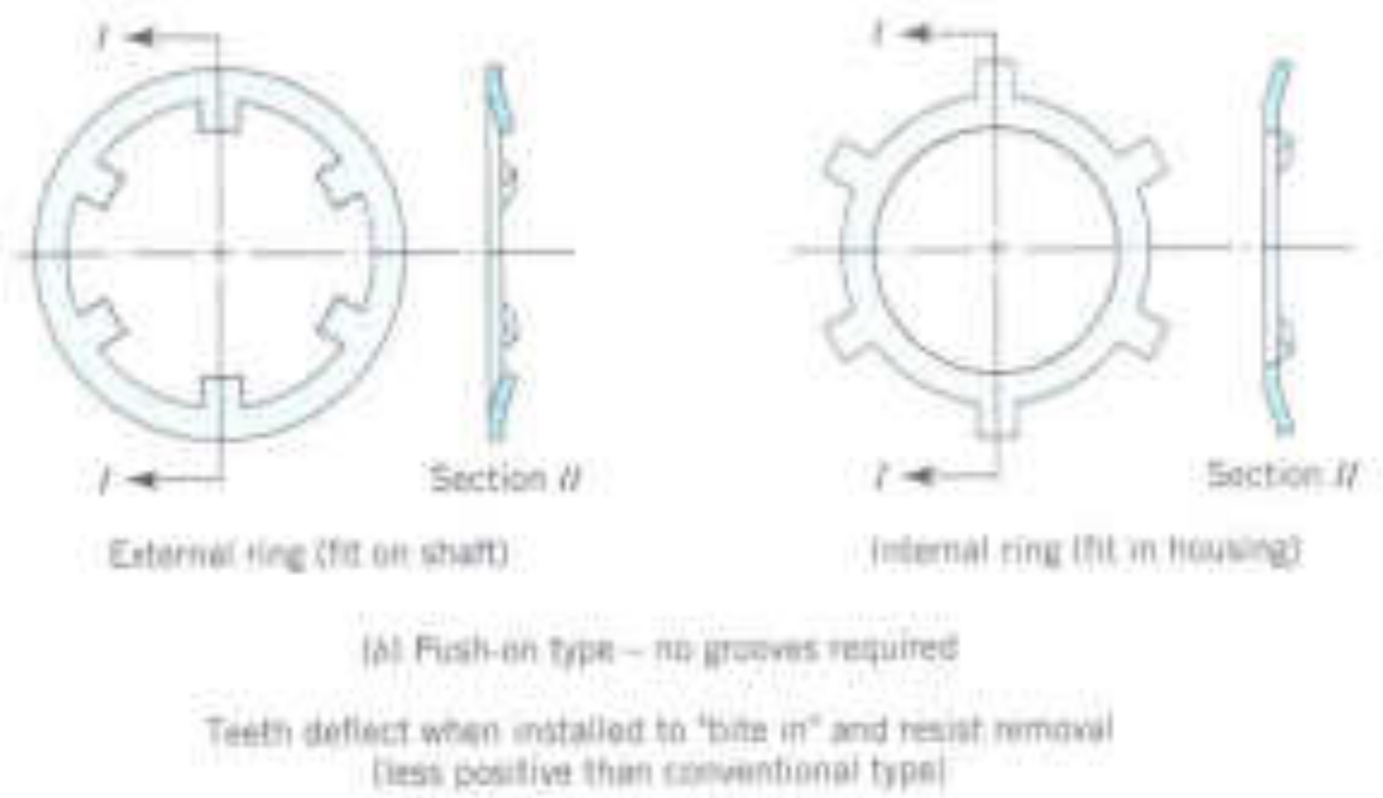
$$\left(\sigma_{av} + \sigma_r K_{fb}\left(\frac{S_{yp}}{S_e}\right)\right)^2 + 3\left(\tau_{av} + \tau_r K_{ft}\left(\frac{S_{yp}}{S_e}\right)\right)^2 \leq \left(\frac{S_{yp}}{N_{fs}}\right)^2$$

Conventional retaining (or snap) rings fit in grooves and take axial load, but groves cause stress concentration in shaft

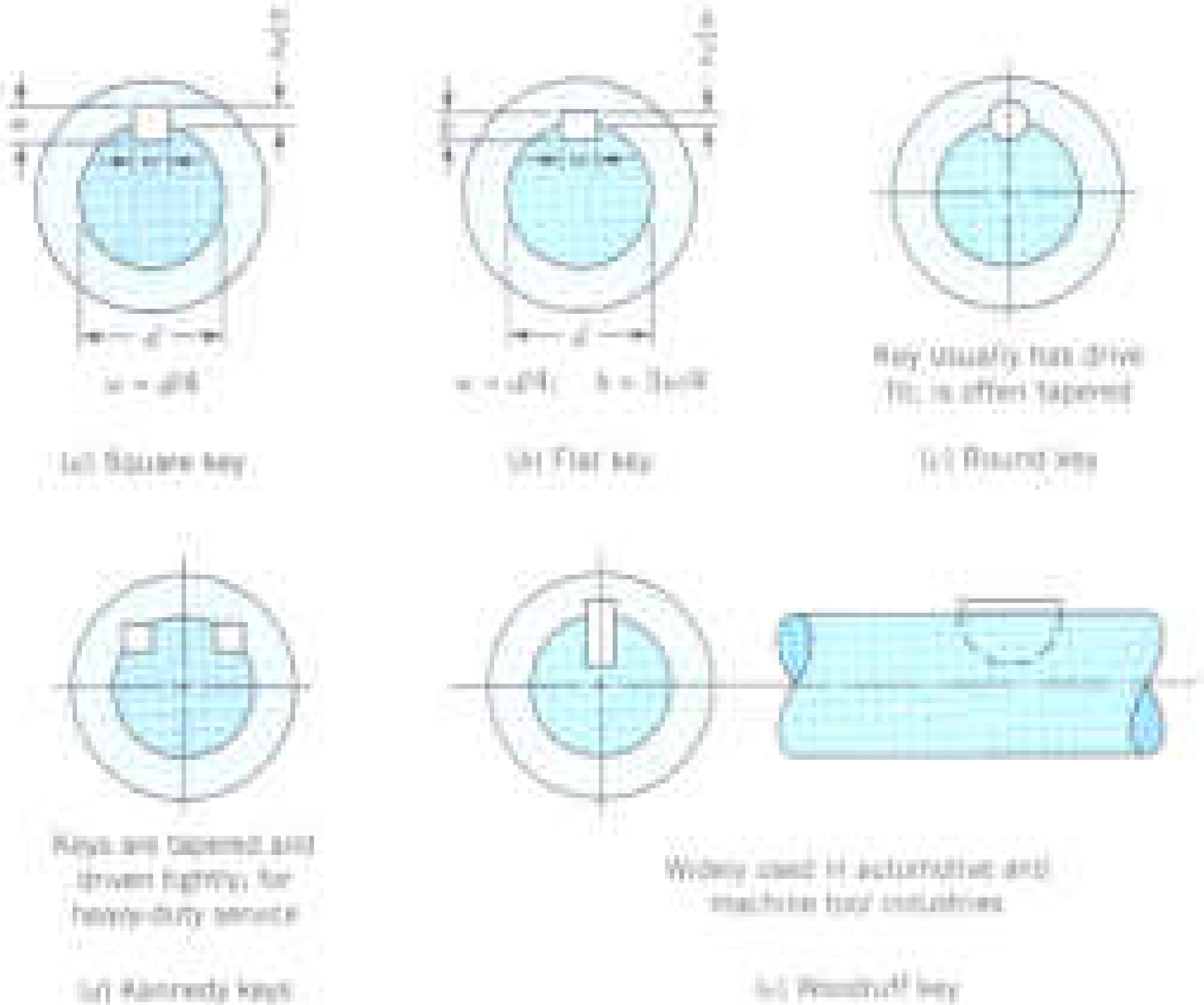


Retaining rings are standardized items, available in various standard sizes with various axial load capacities.

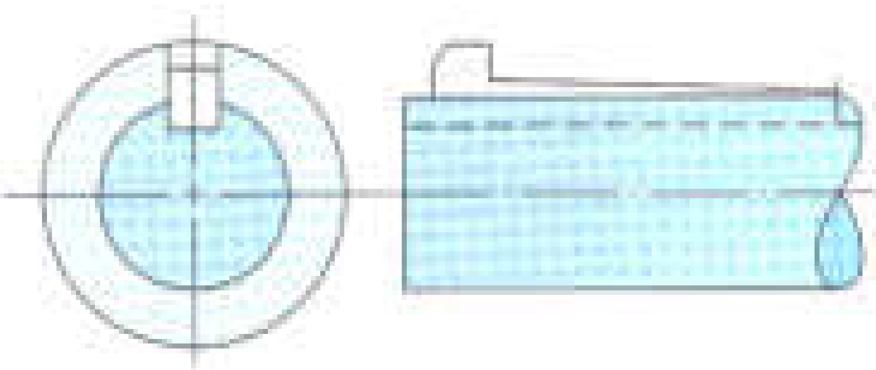
Push type retaining rings – no grooves required, less stress concentration, but less axial support



Various types of keys for transmitting torque

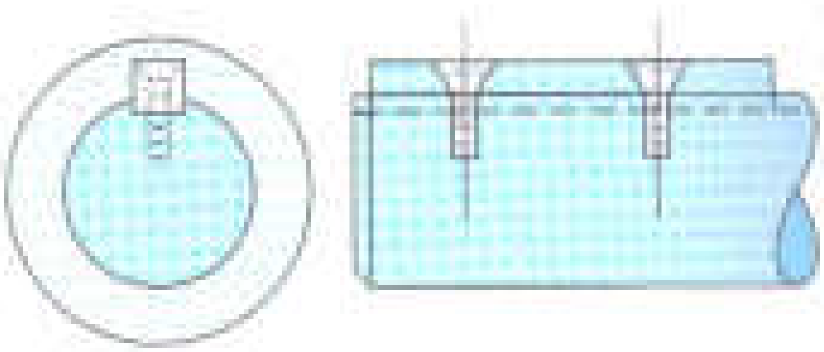


Other common types of keys



Usually tapered, giving tight fit when driven into place; gib head facilitates removal

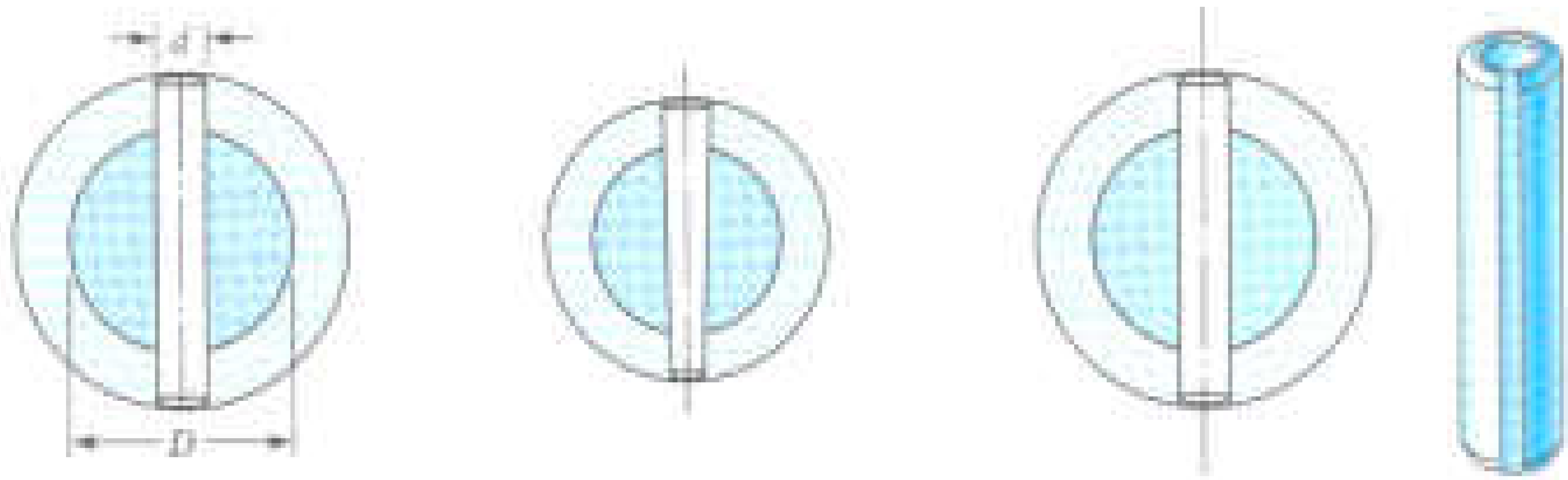
(f) Gib-head key



Key is screwed to shaft; hub is free to slide axially – easier sliding is obtained with two keys spaced 180° apart

(g) Feather key

Various types of collar pins

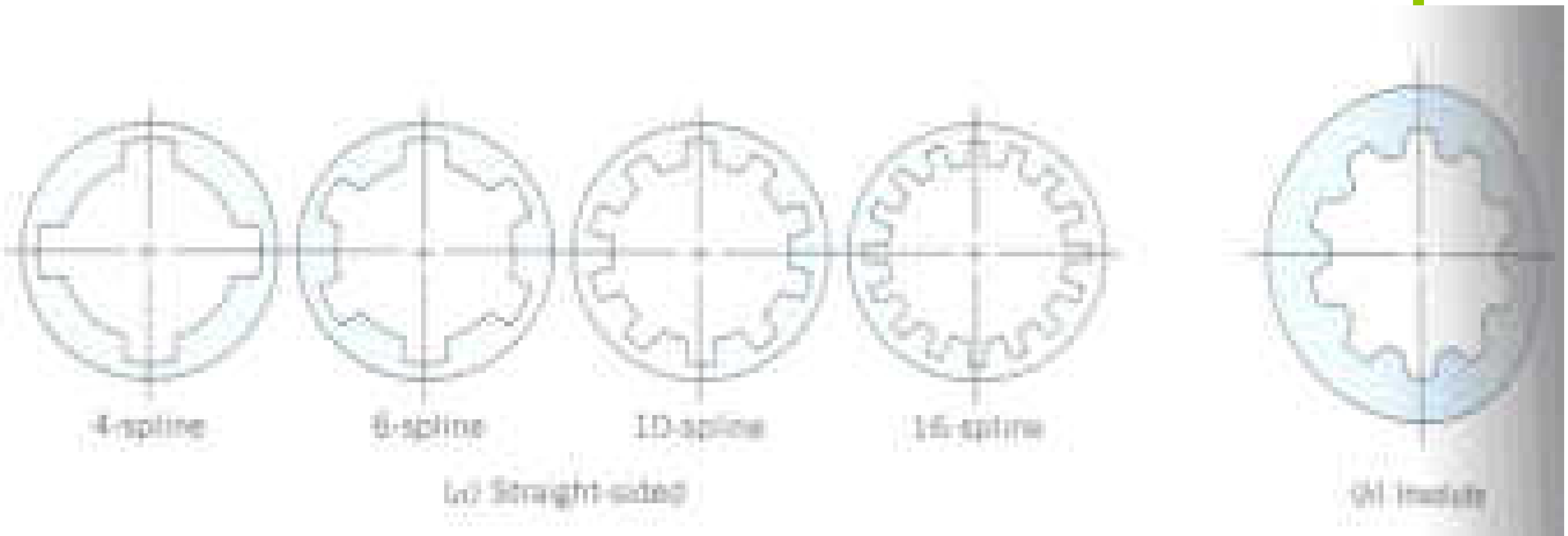


(a) Straight round pin

(b) Tapered round pin

(c) Split tubular spring pin

Integrated splines in hubs and shafts allow axial motion and transmits torque



Common types of splines.

All keys, pins and splines give rise to stress concentration in the hub and shaft