Course material from:

<u>Chapter 5 Materials</u>

-Read chapter (5-1 thru 5-20) for general info on materials properties and processing. Emphasize

5-1 static strength — S_y, S_u

5-4 hardness — Bhn ⟺ S_y

5-20 notch sensitivity

<u>Chapter 6 Steady Loading</u>

(failure theories)

Ch 6-1 thru 6-7 — MSST, DIST. ENERGY

Ch 6-9 and 6-10 — COULOMB-MOHR, MODIFIED-MOHR

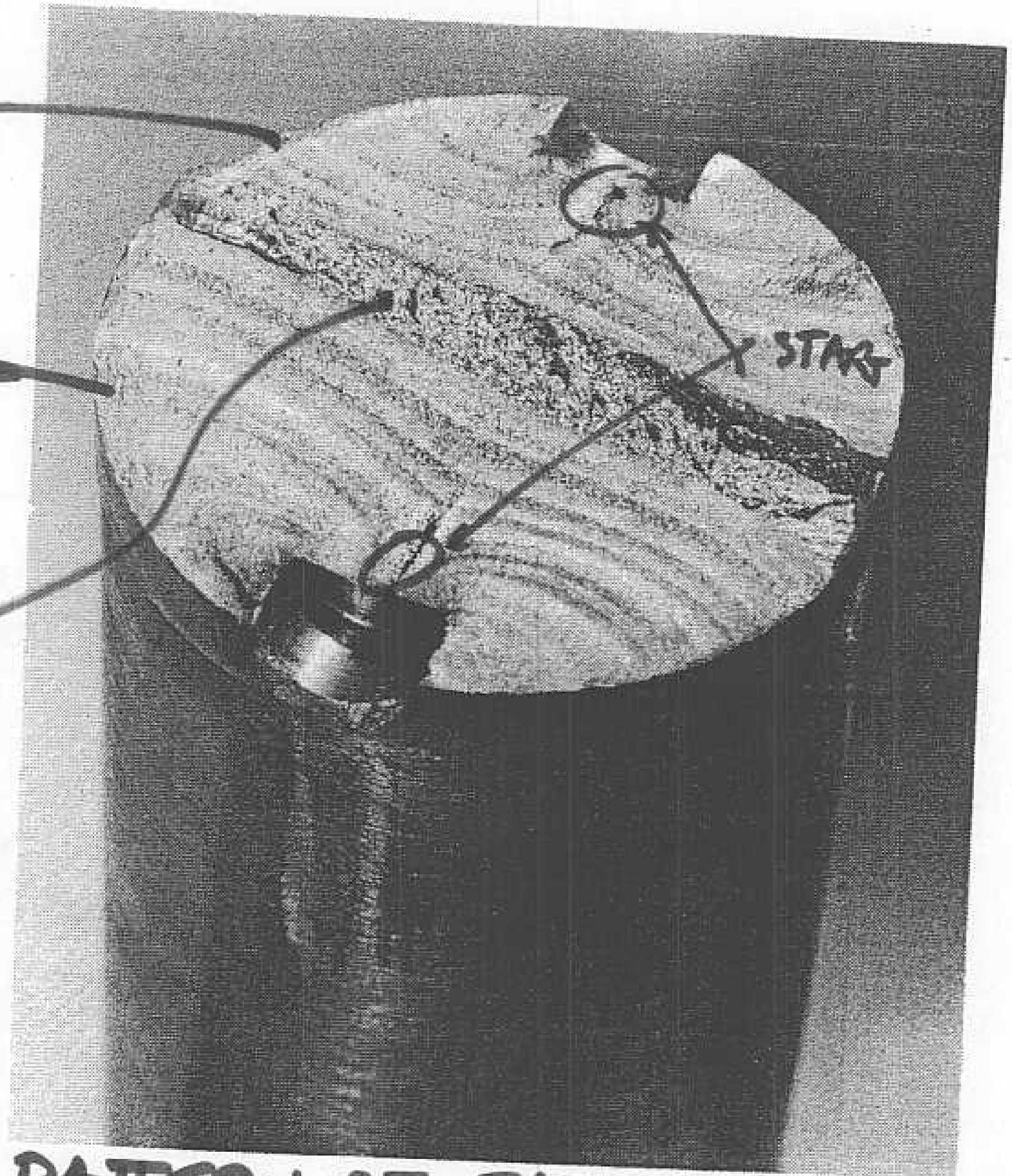
CH7 FATIGUE FAILURE

- STATIC TESTS SHOW YIELD STRENGTH AND ULTIMATE STRENGTH
- MACHINES LIKELY TO FAIL THROUGH S_y OR S_u WILL NORMALLY FAIL IN ROUTINE EVALUATION
- FATIGUE FAILURE RESULTS FROM DYNAMIC LOADS
- NOT OBVIOUS UNTIL SUDDEN FAILURE AFTER REPEATED CYCLES

- TECHNIQUES SUCH AS "MAGNA-FLUX" AND X-RAYS HAVE BEEN USED TO IDENTIFY CRACKING DUE TO FATIGUE - BEFORE A FAILURE TAKES PLACE
- STRESS LEVELS ARE BELOW YIELD
- TYPICALLY FAILURE BEGINS AT A STRESS CONCENTRATION POINT

FAILURE

- OCCURS WHEN AN "INITIAL" CRACK GROWS TOO LARGE
- REMAINING MATERIAL IS NOT SUFFICIENT TO SUPPORT THE LOAD AND FAILS



PATTERN OF FATIGUE FAILURE

Fatigue failure of a shaft. Failure started at the keyway where the transverse cracks grew to be so large that the remaining material could no longer sustain the load and the shaft finally broke. Note the shiny surface from the keyway to the break, which is granular in appearance. [Courtesy of Joseph T. Ryerson and Sons, Chicago, Ill.]

- FAILURE OCCURS OVERTIME AT STRESS LEVELS BELOW THOSE WHICH CAUSE STATIC FAILURE

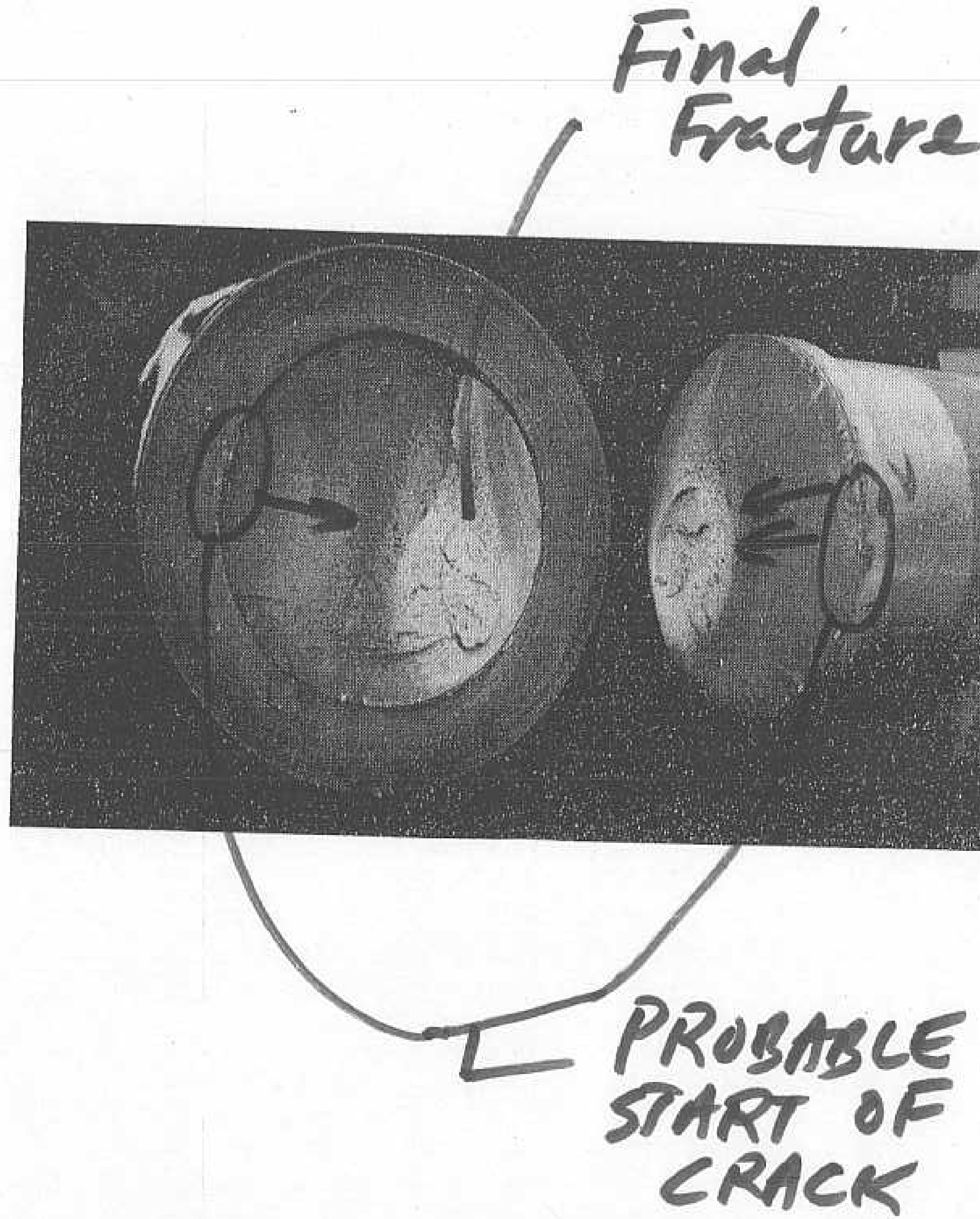
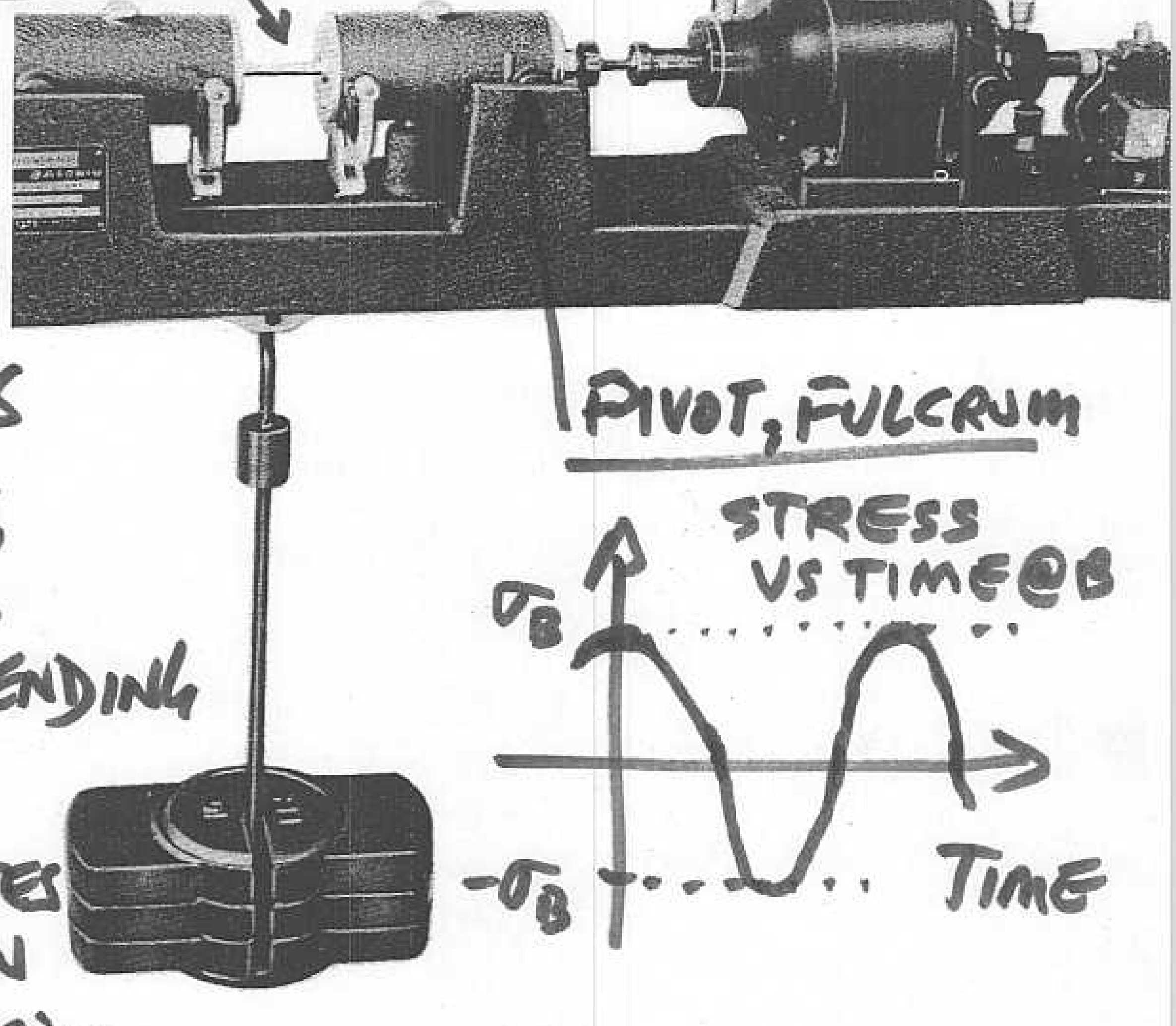


FIGURE 7-1

A fatigue failure of a $7\frac{1}{2}$-in-diameter forging at a press fit. The specimen is UNS G10450 steel, normalized and tempered nd has been subjected to rotating bending. *(Courtesy of The Timken Company.)*

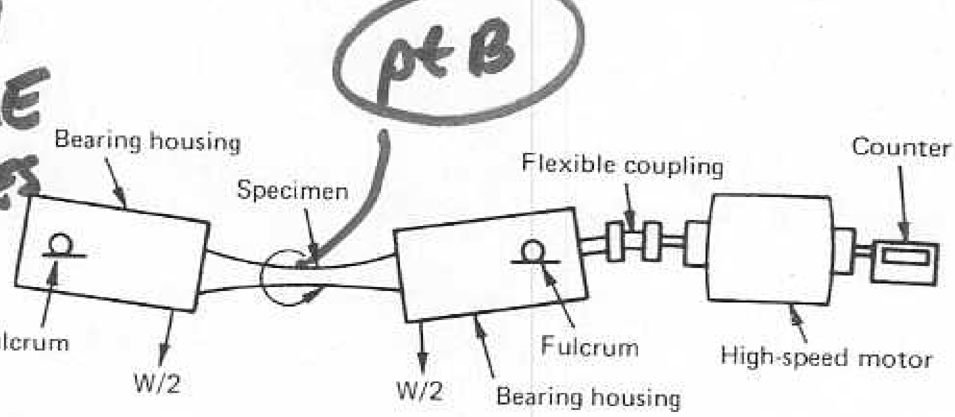


(b)

- SAMPLE ROTATES
- LOADING CREATES PURE BENDING
- STRESS OSCILLATES BETWEEN COMPRESSION & TENSION AS SAMPLE BEAM ROTATES

i.e.,

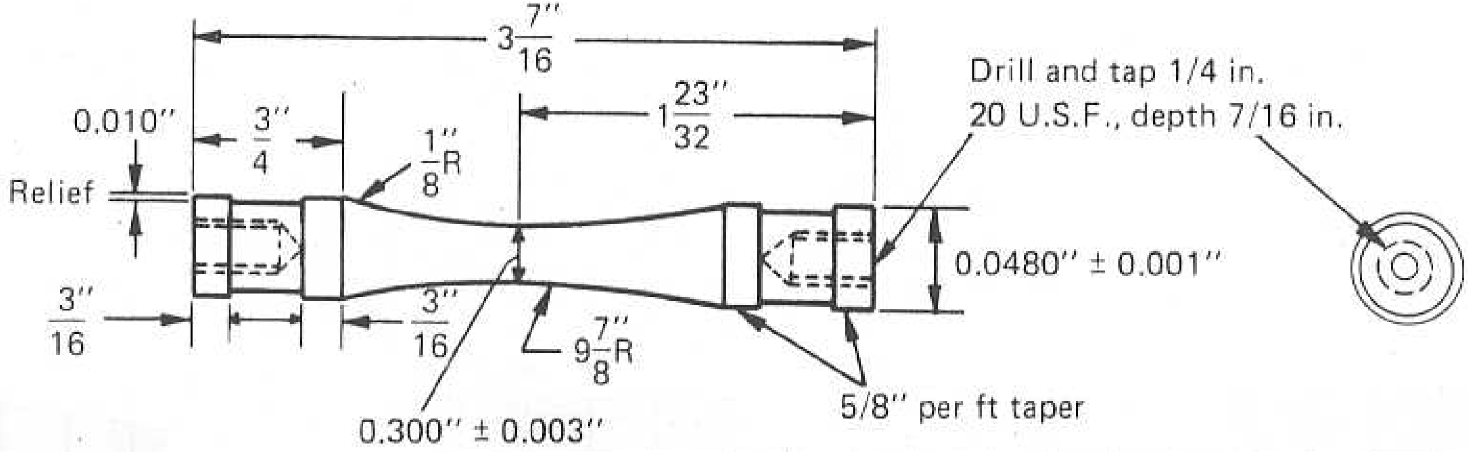
$$\pm \frac{Mc}{I} = \pm \sigma_B$$



(c)

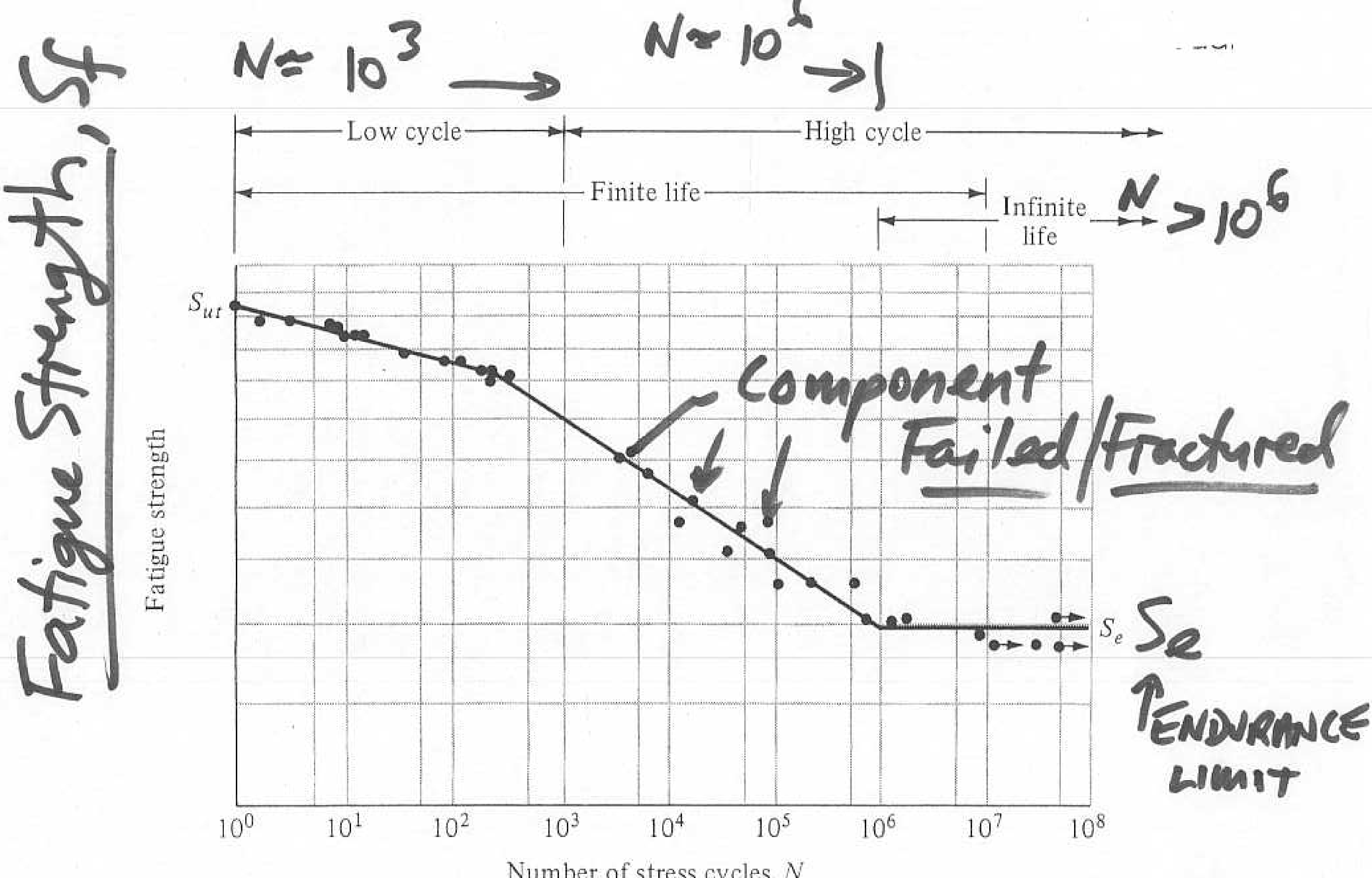
e testing machine. (c) Schematic of the R. R. Moore testing machine. (b) The R. R. Moore rotating f Satec Systems, Inc.]

STANDARD FATIGUE TEST APPARATUS



Standard Fatigue Sample

- 0.3 inch diameter
- HIGHLY POLISHED TO ELIMINATE/MINIMIZE SURFACE DEFECTS
- NO STRESS CONCENTRATION



Fatigue Strength, Sf
N ≈ 10^3 →
N ≈ 10^6 →|
N > 10^6
Low cycle
High cycle
Finite life
Infinite life
Fatigue strength
S_{ut}
Component
Failed/Fractured
S_e
Se
ENDURANCE LIMIT
10^0
10^1
10^2
10^3
10^4
10^5
10^6
10^7
10^8
Number of stress cycles, N
CYCLES, N

FIG 7-6

Terminology

S_f ⇒ Fatigue Strength (for some number of cycles) of a component

S_f' ⇒ Fatigue Strength of standard Sample

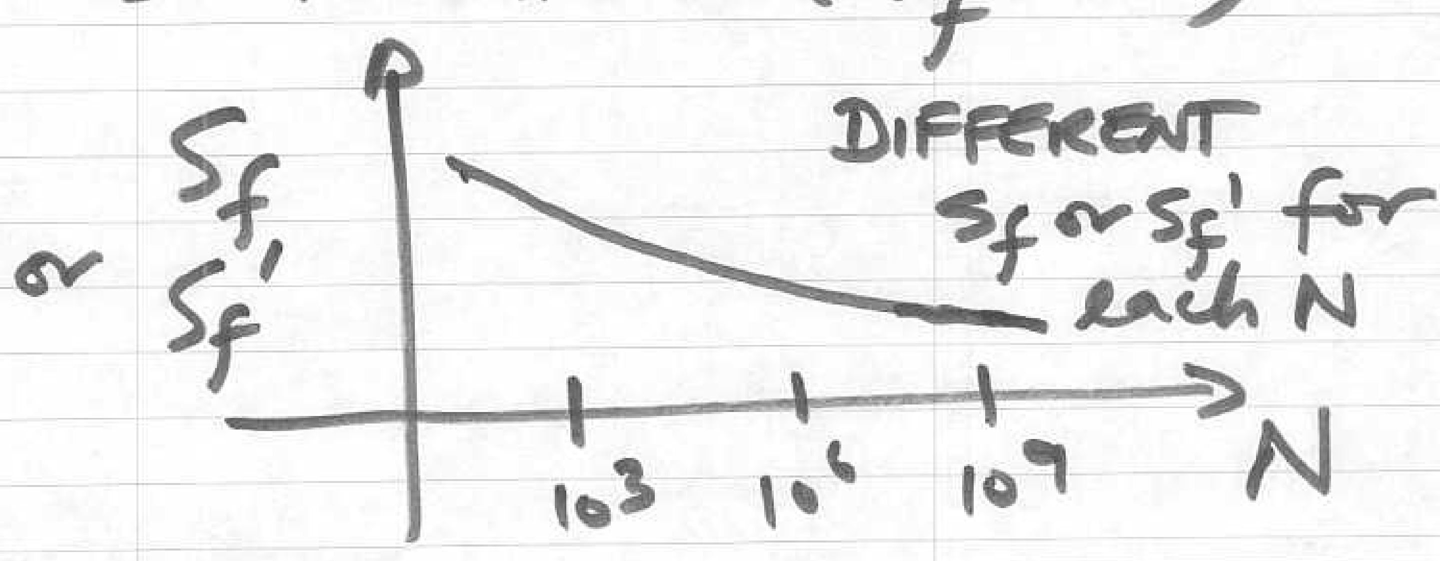
S_e ⇒ Endurance Limit of a component ∴ infinite life

S_e' ⇒ Endurance Limit of standard sample

- STEEL & IRONS USUALLY HAVE ENDURANCE LIMITS

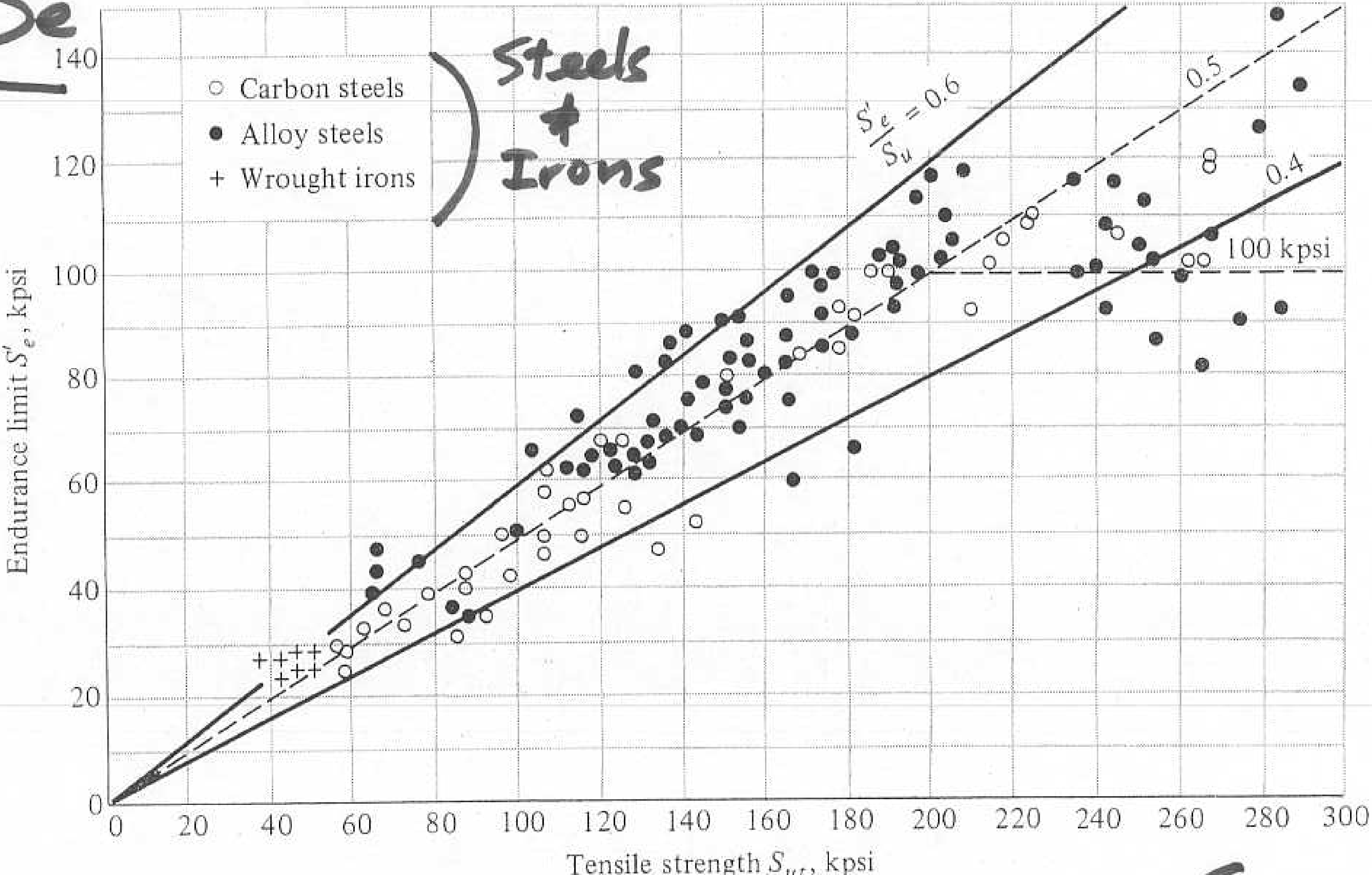
- S_e' can be related to S_{ut} ... i.e., no need to always do fatigue tests

- OTHER MTLS JUST HAVE S-N curves ($S_f \leftrightarrow N$)



Se'

FIG 7-7



Sut

RELATING ENDURANCE LIMIT TO ULTIMATE STRENGTH

Se' refers to std sample

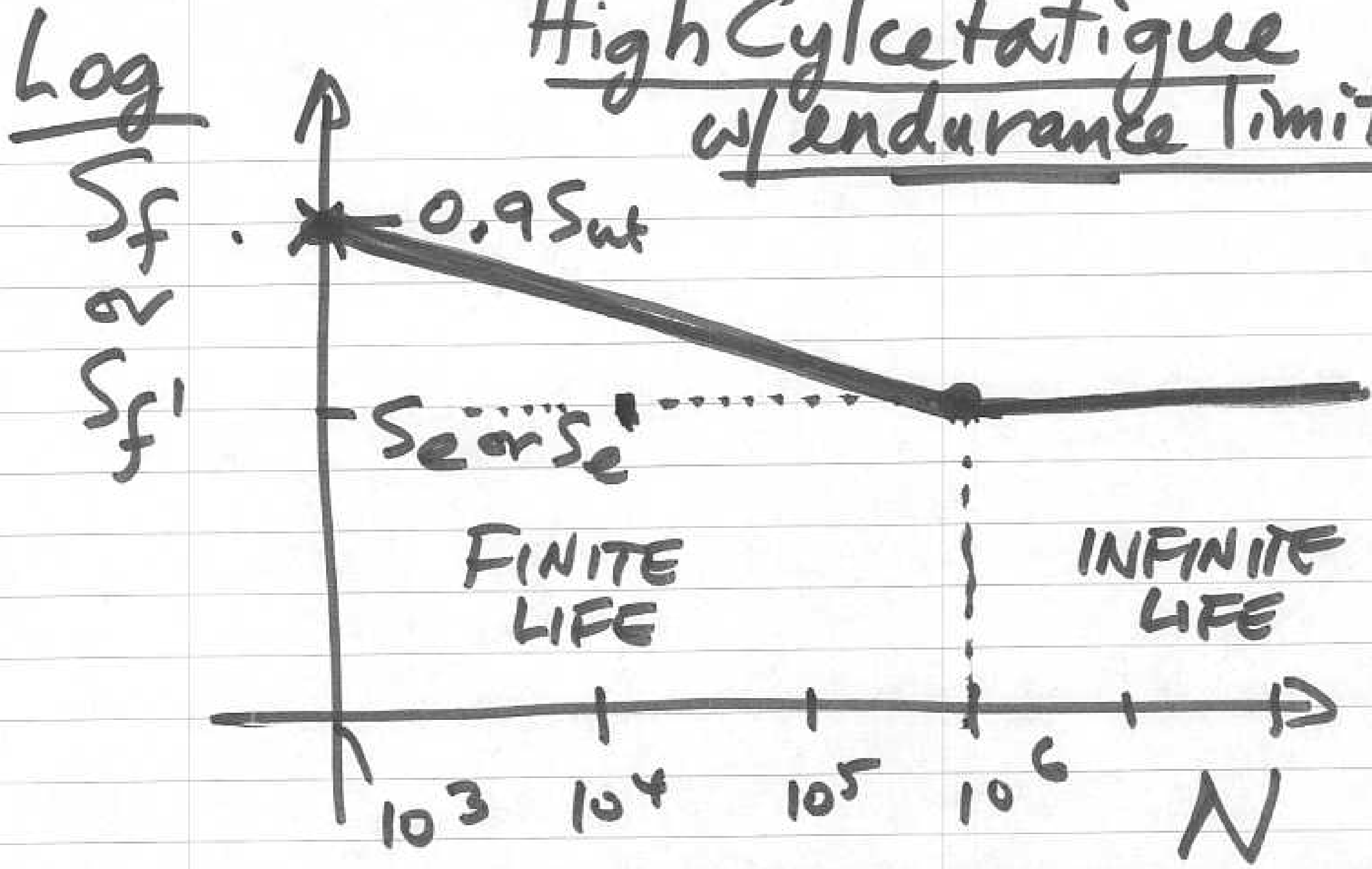
Relating S_e' to S_{ut}

$$S_e' = \underline{0.5\,S_{ut}} \quad \begin{cases} S_{ut} < 200\,\text{kpsi} \\ S_{ut} < 1400\,\text{MPa} \end{cases}$$

$$S_e' = \underline{100\,\text{kpsi}} \qquad S_{ut} > 200\,\text{kpsi}$$

$$S_e' = \underline{700\,\text{MPa}} \qquad S_{ut} > 1400\,\text{MPa}$$

High Cylce Fatigue w/ endurance limit



Need to define/calculate

S_f or S_f' for $10^3 < N < 10^6$

$$\log S_f = \log a + b \log N$$

OR

$$S_f = a N^b \; ; \; N = \left(\frac{S_f}{a}\right)^{1/b}$$

From straight line fit to log-log plot of S_f vs N:

$$a = \frac{(0.9 S_{ut})^2}{S_e}$$

$$b = -\frac{1}{3} \log\left(\frac{0.9 S_{ut}}{S_e}\right)$$

$$\left(10^3 < N < 10^6\right)$$

Factor of Safety for Fatigue for: { Alternating Stress σ_a

Infinite Life

$$n = \frac{S_e}{\sigma_a}$$

Finite Life

$$n = \frac{S_f}{\sigma_a}$$

Example:

Given: Steel with $S_{ut} = 1000\,MPa$

Stress fluctuates between $\pm 300\,MPa$

Determine: i) n for infinite life

ii) n for life of 10,000 cycles.

i). $n = \frac{S_e}{\sigma_a}$

$S_e = 0.5\, S_{ut} = \underline{500\, MPa}$

$$\boxed{n = \frac{500}{300} = 1.6}$$

ii) $n = \frac{S_f}{\sigma_a}$

use $S_f = aN^b$ eq. 7-5

$$a = \frac{(0.9\, S_{ut})^2}{S_e} = \frac{(0.9(1000))^2}{500}\ MPa$$

$a = \underline{1640\ MPa}$

$$b = -\frac{1}{3}\log_{10}\left(\frac{0.9 S_{ut}}{S_e}\right)$$

$$= -\frac{1}{3}\log\left(\frac{900}{500}\right) = -0.085$$

$$S_f = 1640(10{,}000)^{-0.085} \text{ MPa}$$

$$= \underline{749.6 \text{ MPa}}$$

$$n = \frac{S_f}{n} = \frac{749.6}{300}$$

$$\boxed{n = 2.50}$$

Consider also example when S_{ut}, n, and σ_a ($S_e < \sigma_a < 0.9 S_{ut}$) are given & you must find N.

ENDURANCE LIMIT MODIFYING FACTORS (7-8)

- ACTUAL ENDURANCE LIMIT IS SOUGHT FOR COMPONENTS & LOCATIONS (PTS. OF STRESS) THAT DO NOT CORRESPOND TO STANDARD SAMPLE

$$S_e = k_a k_b k_c k_d k_e S_e' \quad \ldots \; 7\text{-}2$$

- S_e: ACTUAL (ESTIMATED) ENDURANCE LIMIT
- k_a: Surface
- k_b: Size
- k_c: Load
- k_d: Temp
- k_e: MISC
  - ⋮
  - STRESS CONCENTRATION
  - ⋮
  - Corrosion

SURFACE FINISH

TABLE 7-4

SURFACE FINISH	FACTOR *a* kpsi	FACTOR *a* 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

$$k_a = a S_{ut}^{b}$$

Rougher surface

$\rightarrow$ smaller k_a

Stronger matl

$\rightarrow$ smaller k_a

SIZE FACTOR - Bending & Torsion

$$k_b = \begin{cases} \left(\dfrac{d}{0.3}\right)^{-0.1133} & 0.11 \le d \le 2 \text{ in} \\ \left(\dfrac{d}{7.62}\right)^{-0.1133} & 2.79 \le d \le 51 \text{ mm} \end{cases}$$

- for larger sizes, k_b is between 0.60 & 0.75
- For (uniform) axial stress

$$k_b = 1$$

- For non-circular X-sections use equivalent diameter, d_e

LOAD FACTOR, k_c

Load Factor k_c

The load factor is given by the equation

$$k_c = \begin{cases} 0.923 & \text{axial loading} \quad S_{ut} \leq 220 \text{ kpsi (1520 MPa)} \\ 1 & \text{axial loading} \quad S_{ut} > 220 \text{ kpsi (1520 MPa)} \\ 1 & \text{bending} \\ 0.577 & \text{torsion and shear} \end{cases}$$

... Note $k_c = 1$ FOR BENDING & AXIAL FOR $S_{ut} > 220$ kpsi

... 0.577 ⇒ "Shear Strength" as per distortion energy theory.

TEMPERATURE, k_d

... Table 7.5 ... for Steels

... Becomes significant,
e.g. $k_d = 0.8$, @ 475°C; 900°F

... Not impt below 300°C, 550°F

... if temp not mentioned,
Assume $k_d = 1$

Miscellaneous factor, k_e

- includes corrosion, fretting, coatings, other surface treatments, etc.
- MOST IMPT FOR US IS

"fatigue strength reduction factor"

$$k_e = \frac{1}{K_f}$$

K_f ⇒ fatigue stress concentration factor

- See sect 5-20 pp 217-218

K_f is related to

K_t ... Theoretical or geometric stress concentration factor Through "notch sensitivity" q.

$$q = \frac{K_f - 1}{K_t - 1}$$

∴ if $q = 1$ ⇒ "fully sensitive"

⇒ $K_t \Leftrightarrow K_f$

∴ if $q = 0$ $K_f = 1$ ⇒ "no strength reduction"

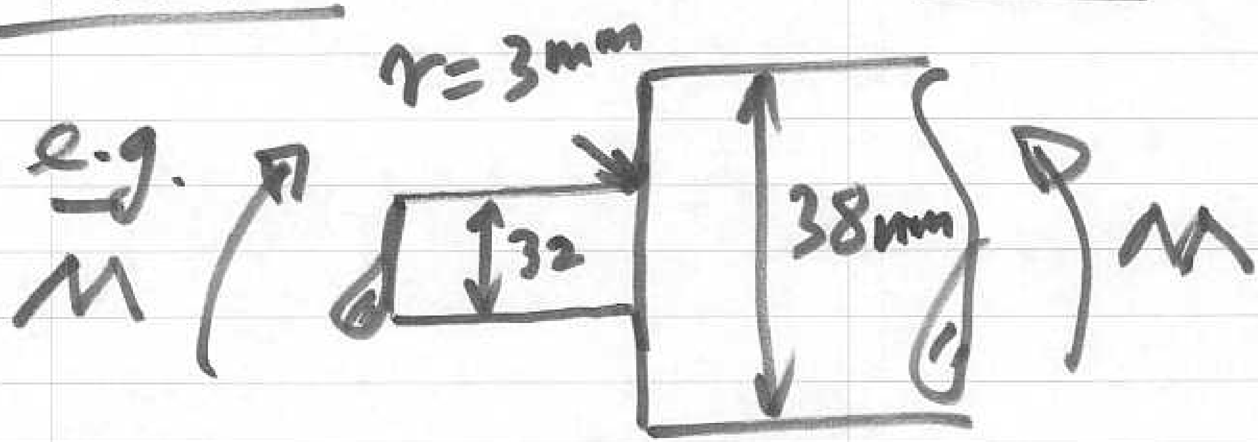
$$K_f = 1 - q(K_t - 1)$$

from 5-20

from tables A-15-...

EXAMPLE

e.g.



AISI 1050 Cold drawn steel

$S_{ut} = 690\,MPa$ ... Table A-20

$q = 0.82$ from Fig 5-16

... interpolating

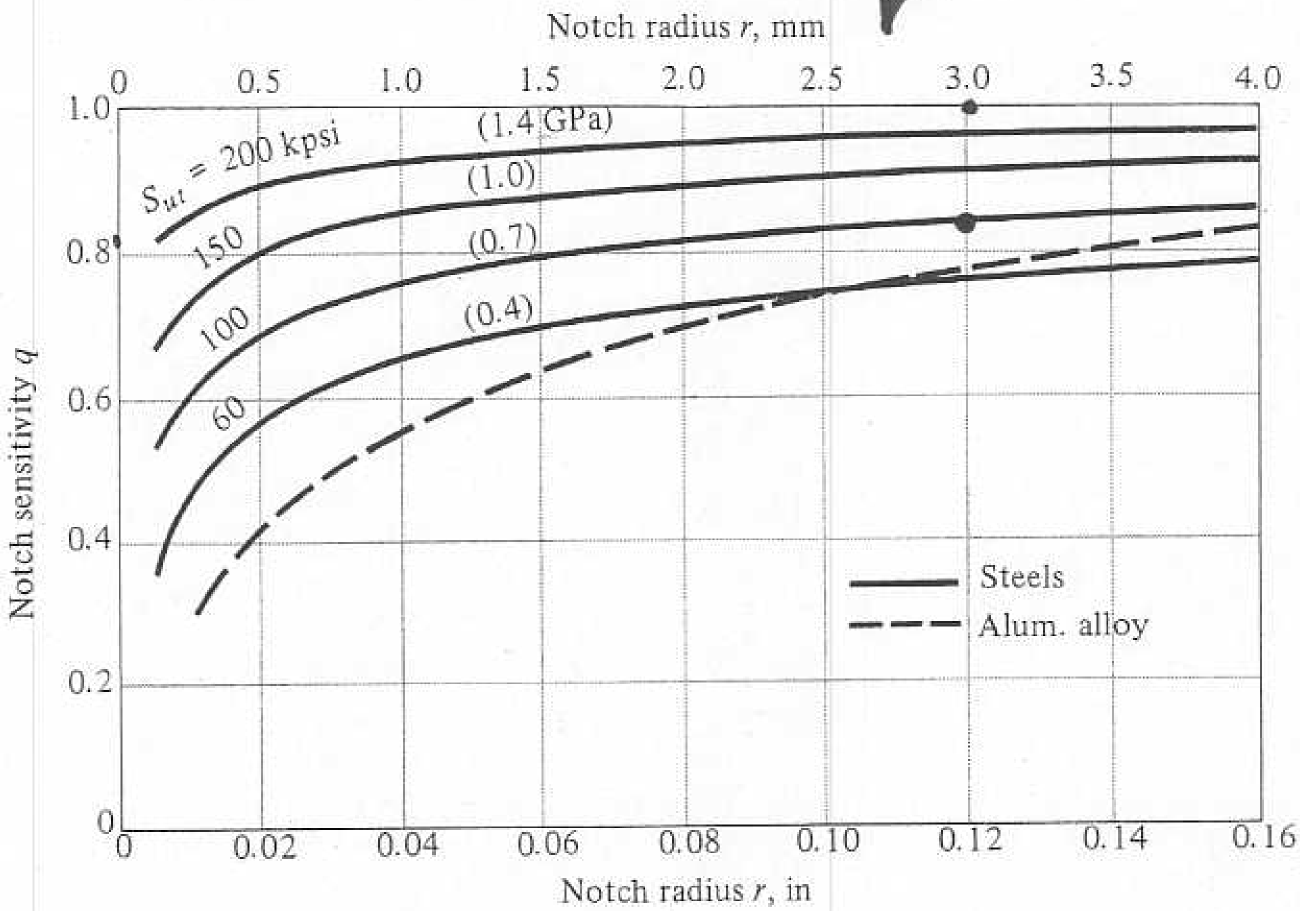
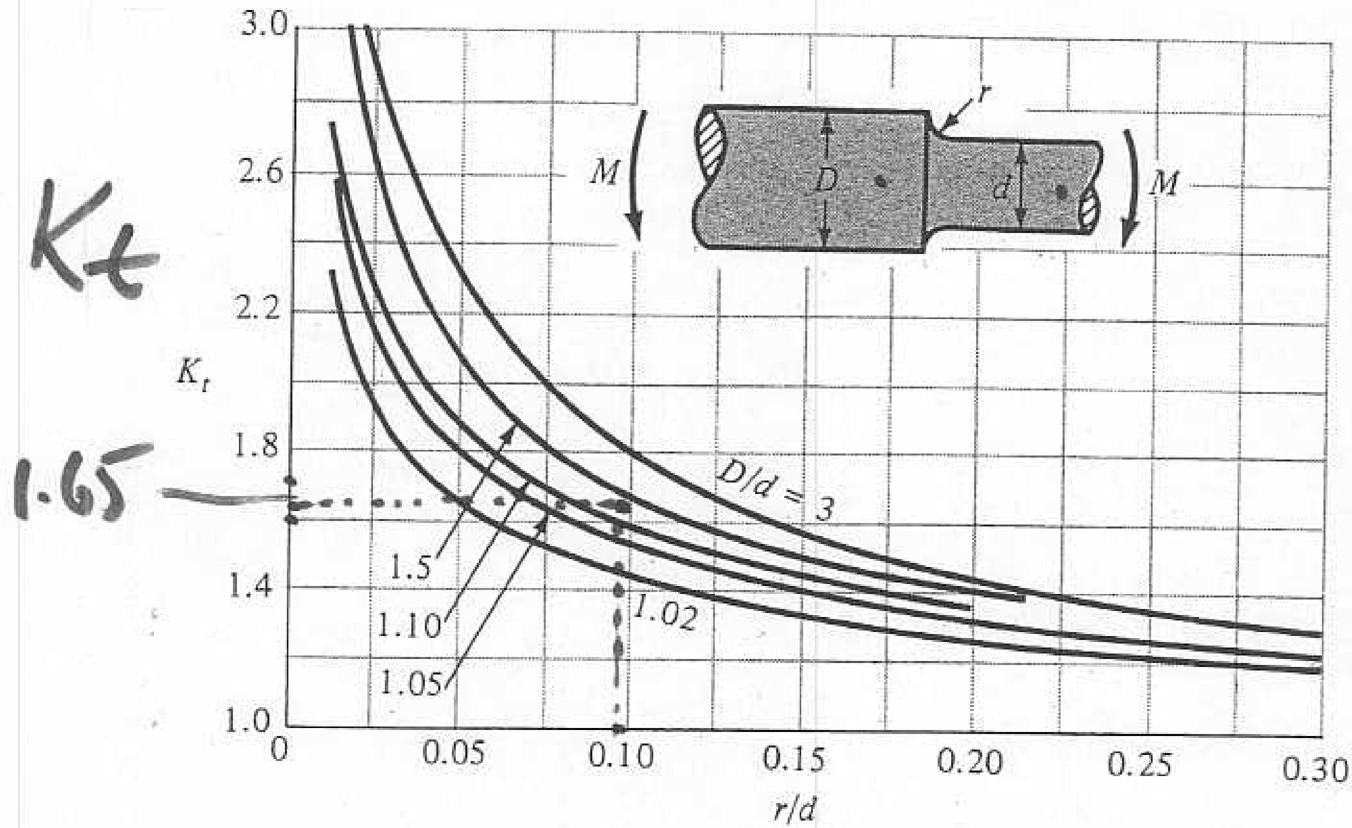


FIGURE 5-16

Notch-sensitivity charts for steels and UNS A92024-T wrought aluminum alloys subjected to reversed bending or reversed axial loads. For larger notch radii, use the values of q corresponding to $r = 0.16$ in (4 mm). *[Reproduced by permission from George Sines and J. L. Waisman (eds.), Metal Fatigue, McGraw-Hill, New York, 1959, pp. 296, 298.]*

A-15-



r/d

$r/d = 3/32 = \underline{.094}$

$D/d = 38/32 = \underline{1.19}$

$K_t = 1.65$

$K_f = 1 + q(K_t - 1) = 1 + 0.8(1.65 - 1)$

$\boxed{K_f = 1.53}$ ⇐

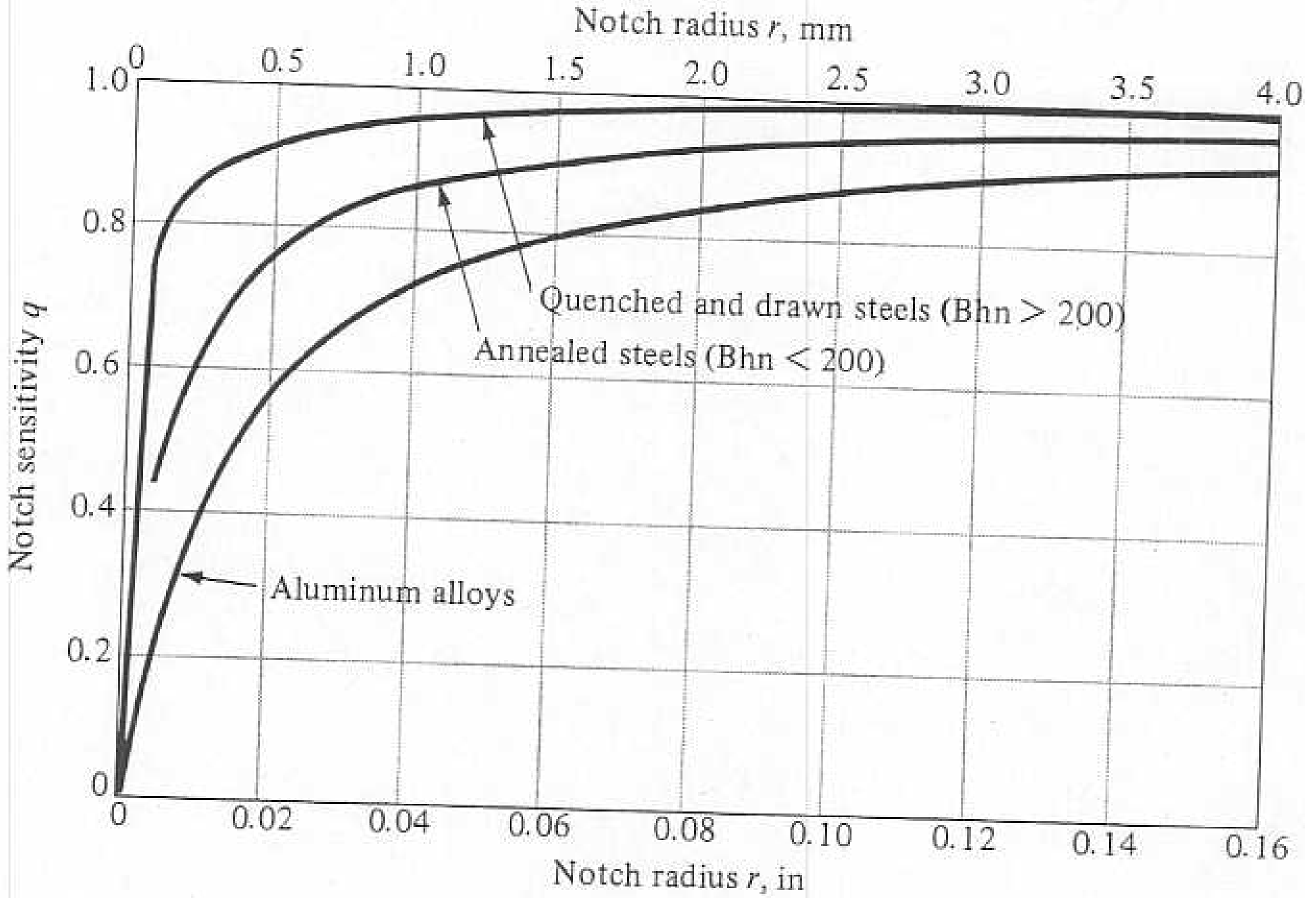


FIGURE 5-17

Notch-sensitivity curves for materials in reversed torsion. For larger notch radii, use the values of q corresponding to $r = 0.16$ in (4 mm).

Assignment #4
Due Mon Jan 17th

#1. 7-1 from text
#2. 7-5 (reversed bending, see sect 5-4 for tensile strength)
#3. 7-11 (infinite life and static failure (yielding))
#4. A 1.25 inch diameter hot-rolled steel rod has a 0.125 inch diameter hole drilled through it. Its (ultimate) tensile strength is 60 kpsi. The rod is subjected to a reversed (alternating) torque of 2000 in-lbs. Estimate the factor of safety for infinite life and against yielding. What would be the safety factor if a life of 20,000 cycles were needed? ... $S_y = 45$ kpsi

EXAMPLE 7-3 Figure 7-10a shows a rotating shaft supported in ball bearings at A and D and loaded by the nonrotating force F. Estimate the life of the part. ∞ or less

6.8kN = F .... Non-rotating force

shaft rotates

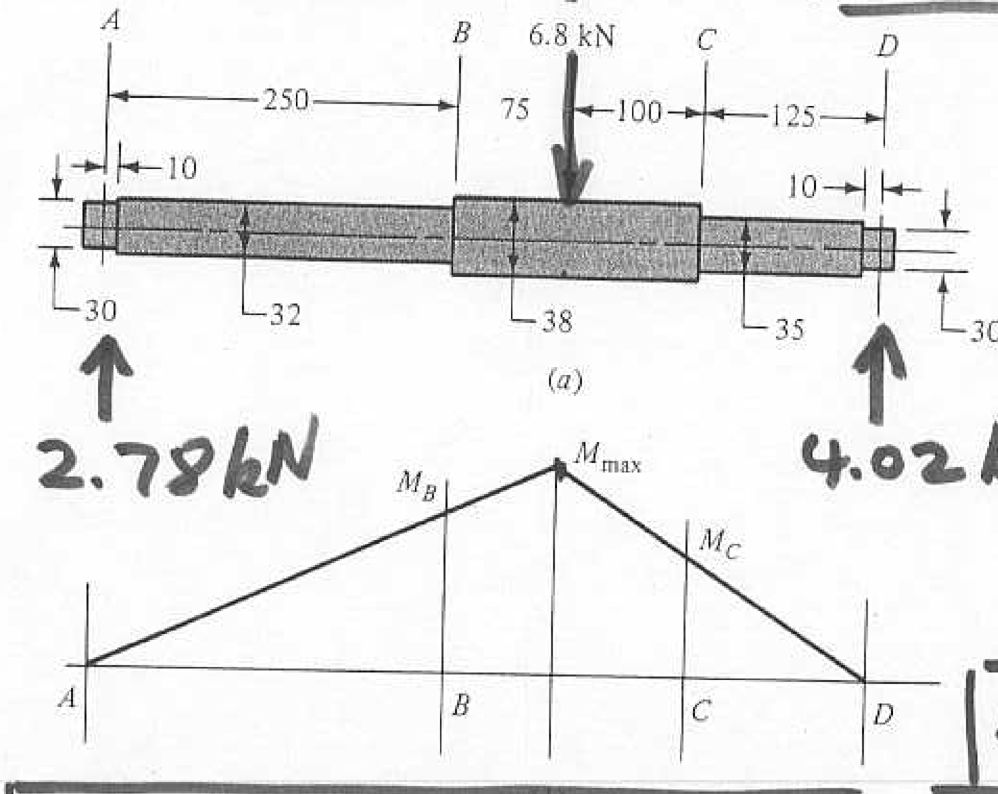


FIGURE 7-10

(a) Shaft drawing; all dimensions in millimeters; all fillets 3-mm radius. The shaft rotates and the load is stationary; material is machined from AISI 1050 cold-drawn steel. (b) Bending-moment diagram.

$M_C = 502.5$ N-m

$M_B = 2.78 \times 250 = 695$ N-m

$M_{max} = 903.5$ N-m

Fillet radii ⇒ 3 mm

AISI 1050 Cold Drawn ... Table A-20

$S_{ut} = 690$ MPa $S_y = 580$ MPa

- FAILURE IN REVERSED BENDING*
- BEGIN BY CHECKING FOR INFINITE LIFE
- LOCATE POINT OF MIN FACTOR OF SAFETY
- CANDIDATES

  PT B , ~~PT C~~, PT OF M_{max}

$$n = \frac{S_e}{\sigma_a} \Rightarrow \frac{Mc}{I}$$

PT B by inspection
- $M = 695$ N-m
- $d = 32$ mm
- Stress concentration

- Determine S_e'
- Apply modifying factors

$S_e' = 0.5\, S_{ut} = \frac{690\text{ MPa}}{2}$

$$\boxed{S_e' = 345\text{ MPa}}$$

$S_e = k_a k_b k_c k_d k_e S_e'$

$k_a = 0.798$ (Table 7-4)

$k_b = 0.85$ (eq 7-15)

$k_c = \underline{1}$ (eq 7-22)

$k_d = 1$ (Table 7-5)

$k_e = \frac{1}{K_f} = \frac{1}{1.53} = \underline{0.654}$ } From previous example

$\therefore\ S_e = 0.798(0.850)(.654)\ \underline{345}$

$$\boxed{S_e = 154\ MPa}$$

$M_B = \underline{695\ N\text{-}m}\ ;\ \sigma = \frac{Mc}{I}$

$\sigma = \frac{695}{I/c} = \underline{\underline{216\ MPa}}$

@d $\Leftarrow$

<u>OOPS</u> $\sigma > S_e$

$$\boxed{\therefore \text{ Finite Life}}$$

Check for finite life

$N = \left(\frac{\sigma_a}{a}\right)^{1/b}$ see p. 291

<u>$N = 188{,}000$ cycles</u> | WITH Factor of Safety = 1