Lecture Note 01
Press-Fit (Interference-Fit) Design

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ME 1029 Mechanical Design 2

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Sichuan University - Pittsburgh Institute

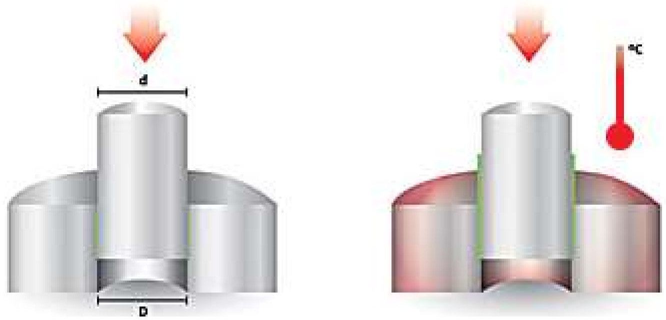
Technical Topics

- Thick-Walled Cylinders (Sec. 3-14)
  - Thick-Walled vs. Thin-Walled
  - Characteristics under Thick-Walled
- Characteristic Stresses of Thick-Walled Cylinders (Sec. 3-14)
- Generalization to Press Fits or Interference Fits (Sec. 3-16, 3-17)
- Limits and Fits (Sec. 7-8)

- Technical References:
  - R.G. Budynas, Advanced Strength and Applied Stress Analysis, McGraw-Hill Book Company.
  - Preferred Limits and Fits for Cylindrical Parts, ANSI B4.1-1967
  - Preferred Metric Limits and Fits, ANSI B4.2-1978

Press Fits or Interference Fits

- In a press/interference fit, the shaft is compressed and the hub (OD Cylinder) is expanded.
- Typical press/interference fits: **Press Fit** & **Shrink Fit**



Press Fit Shrink Fit

- Press fits, or interference fits, are similar in the way to pressurized cylinders:
  - placement of an oversized shaft in an undersized hub results in a radial pressure at the interface
- Design Interest: Calculate radial pressure at the interface

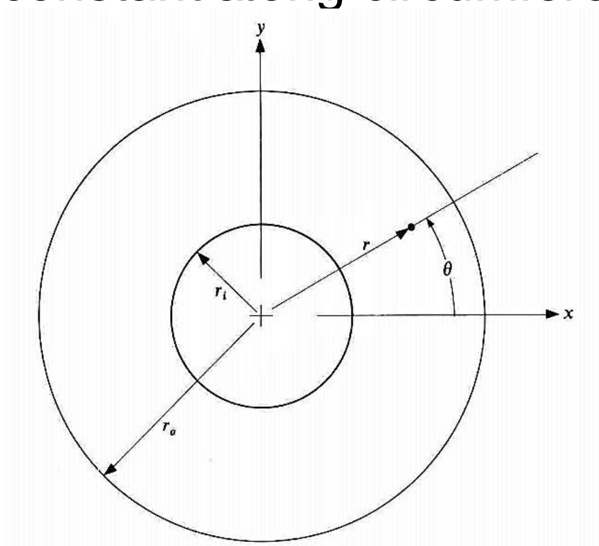
Characteristic Stresses of Thick-Walled Cylinders

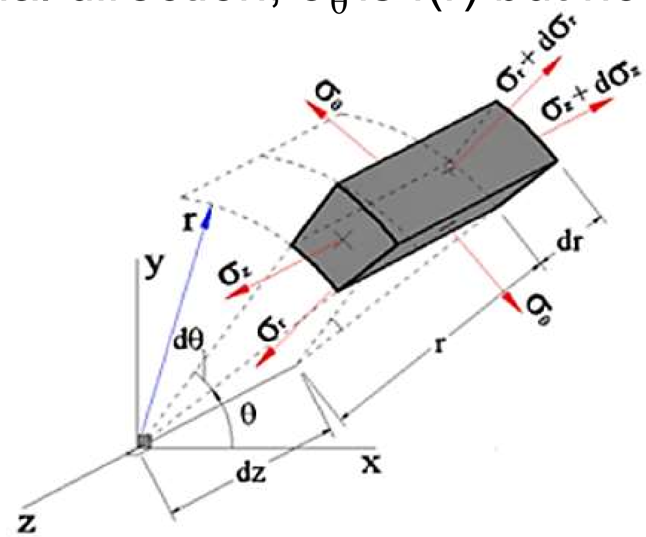
Characteristic stresses on a thick-walled cylinder are

- Circumferential (hoop) stress σ_θ,
- Radial stress σ_r, and
- Longitudinal (axial) stress σ_z

Assuming cylinder geometry is symmetric along axial axis Z,

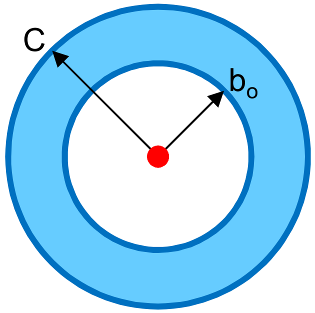
- Shear stress $\tau_{r\theta}$ will not develop; $\tau_{r\theta} = 0$
- σ_θ constant along circumferential direction; σ_θ is f(r) but not f(θ)



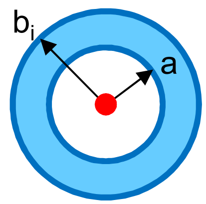


Characteristics of Press Fits

- The shaft is compressed and the hub is expanded.
- Pressures at the mating surfaces (P) are equal and opposite.
- The relative amount of compression and expansion depends on the stiffness (elasticity and geometry) of the two pieces.
- Sum of shaft compression and hub expansion equals the **interference** introduced.
- Assume both members have the same length



Hub: Outer Cylinder



Shaft: Inner Cylinder

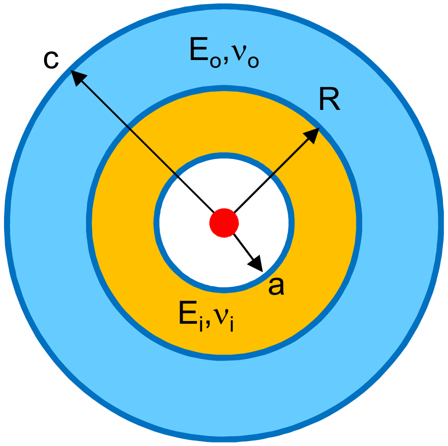
Radial Interference

$$\delta = b_i - b_o$$

Contact Pressure at Press Fit Interface

- Contact pressure (P) serves as outer surface pressure for the shaft (inner cylinder) and inner surface pressure for the hub (outer cylinder).
- Assume shaft and hub are of different materials.
- Since b_o and b_i are almost equal, let $b_o \approx b_i \approx R$
- Contact pressure P:

$$P = \frac{\delta}{R\left[\frac{1}{E_o}\left(\frac{c^2+R^2}{c^2-R^2}+\nu_o\right)+\frac{1}{E_i}\left(\frac{R^2+a^2}{R^2-a^2}-\nu_i\right)\right]}$$

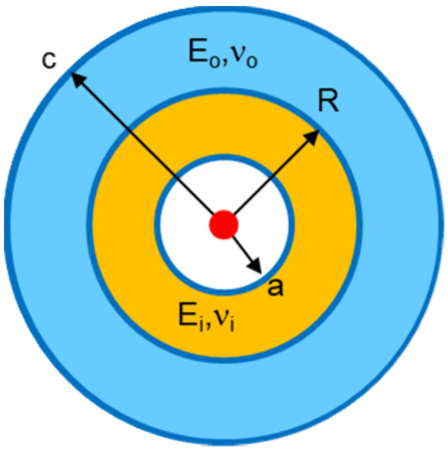


δ = radial interference, $\delta = b_i - b_o$
R: approximate radius at interface
c: hub outer radius
a: shaft inner radius
E_o , E_i Modulus of elasticity of outer & inner parts respectively
ν_o , ν_i Poisson’s ratio of elasticity of outer & inner parts respectively

Contact Pressure at Press Fit Interface

- Common Case: inner cylinder and outer cylinder are made of the same material ($E_o = E_i = E, \nu_o = \nu_i = \nu$)
- Contact pressure P:

$$P = \frac{E\delta}{2R^3}\left[\frac{(c^2 - R^2)(R^2 - a^2)}{c^2 - a^2}\right]$$



δ = radial interference, $\delta = b_i - b_o$
R: approximate radius at interface
c: hub outer radius
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E_o , E_i Modulus of elasticity of outer & inner parts respectively
ν_o , ν_i Poisson's ratio of elasticity of outer & inner parts respectively

Stresses at Press Fit Interface

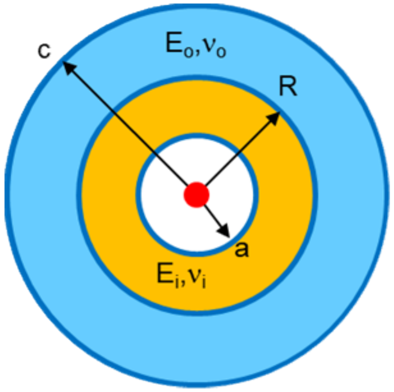
- Radial Stress σ_r

$$(\sigma_r)_{b_i} = (\sigma_r)_{b_o} = -P$$

- Hoop Stress σ_θ

$$(\sigma_\theta)_{b_i} = -P\frac{R^2 + a^2}{R^2 - a^2}$$

$$(\sigma_\theta)_{b_o} = P\frac{c^2 + R^2}{c^2 - R^2}$$



Note that:

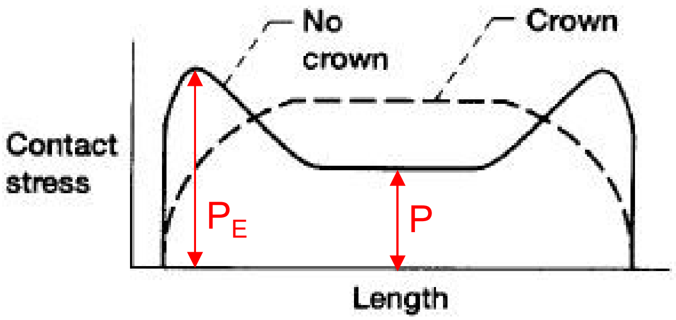
- Radial stress is compressive on both hub and shaft
- Hoop stress is tensile on hub ID and compressive on shaft OD
- Absolute magnitude of radial stress is less than hoop stress

Question: Which stress component on which part is the limiting factor in terms of press-fit design?

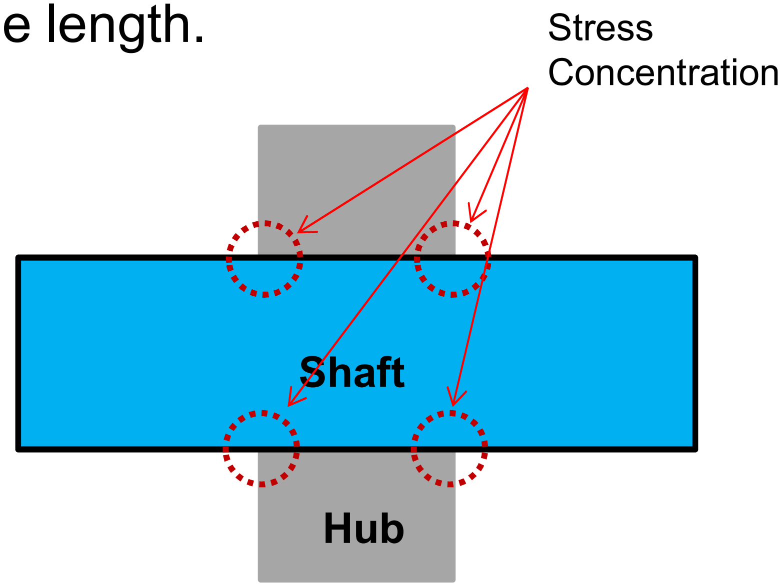
Hoop stress on **hub ID** surface is the limiting factor.

Missing from Calculated Contact Pressure

- Assumed both members have the same length.



$$Stress\ Concentration\ \ K = \frac{P_E}{P}$$



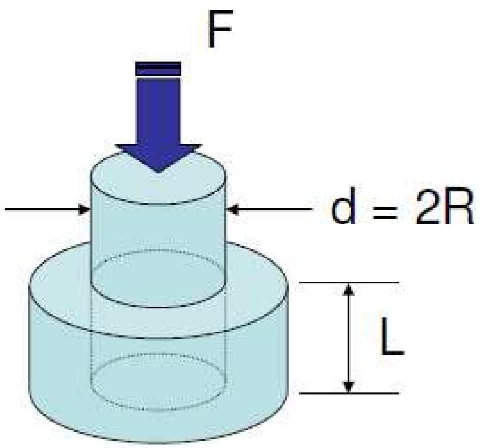
- In the case of a hub that has been press-fitted onto a shaft, this assumption would not be true, and there would be an increased pressure at each end of the hub.
- Stress concentration factor (K) depends upon the contact pressure and the design of the female member, but its theoretical value is seldom greater than 2.

Torque Transmission by Press Fit

Required force in order to press part thru

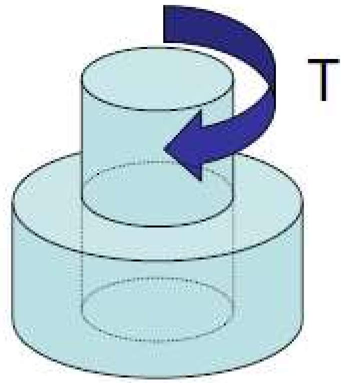
$$F_{max} = \mu(\pi\ d\ L)\ P$$

μ: Coefficient of Friction



Limiting capacity for torque resistance

$$Torque = \mu(\pi\ d\ L)\ P\ R$$

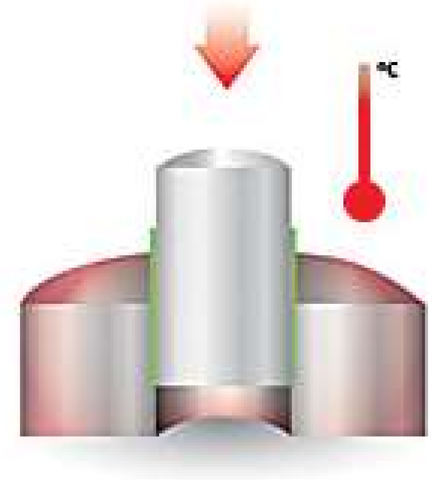


Shrink Fits

If heating or cooling a part is used to achieve a shrink fit, the required radial interference is:

$\delta = \alpha\, b\, \Delta T$

- b: approximate radius at interface
- α: Coefficient of Thermal Expansion
- ΔT: Applied Temperature Change

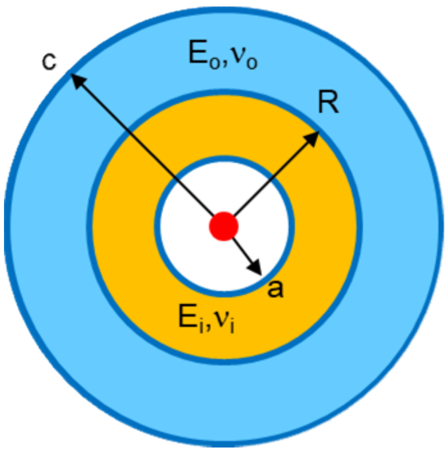


Bonded Shrink Fit

Table 3–3 Coefficients of Thermal Expansion

Material	Celsius Scale (°C^{-1})	Fahrenheit Scale (°F^{-1})
Aluminum	$23.9(10)^{-6}$	$13.3(10)^{-6}$
Brass, cast	$18.7(10)^{-6}$	$10.4(10)^{-6}$
Carbon steel	$10.8(10)^{-6}$	$6.0(10)^{-6}$
Cast iron	$10.6(10)^{-6}$	$5.9(10)^{-6}$
Magnesium	$25.2(10)^{-6}$	$14.0(10)^{-6}$
Nickel steel	$13.1(10)^{-6}$	$7.3(10)^{-6}$
Stainless steel	$17.3(10)^{-6}$	$9.6(10)^{-6}$
Tungsten	$4.3(10)^{-6}$	$2.4(10)^{-6}$

Barring material selections, primary consideration in press fit design is the decision of interference, which not only depends on **nominal dimensions,** but also their **tolerances**, of the mating parts.



7-8 Limits and Fits

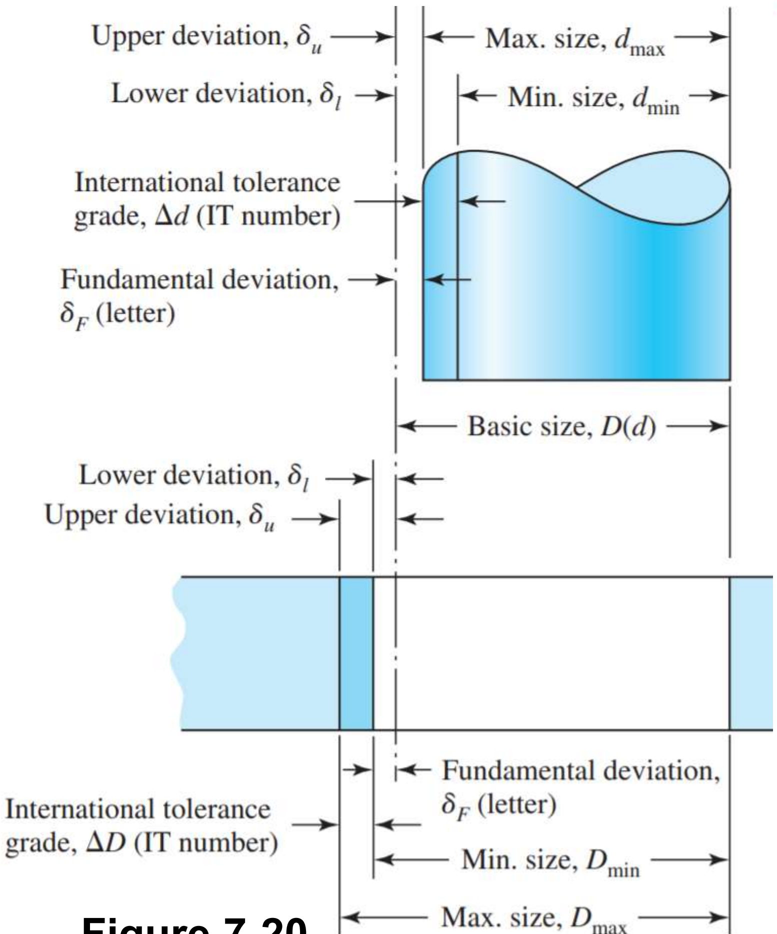


Figure 7-20

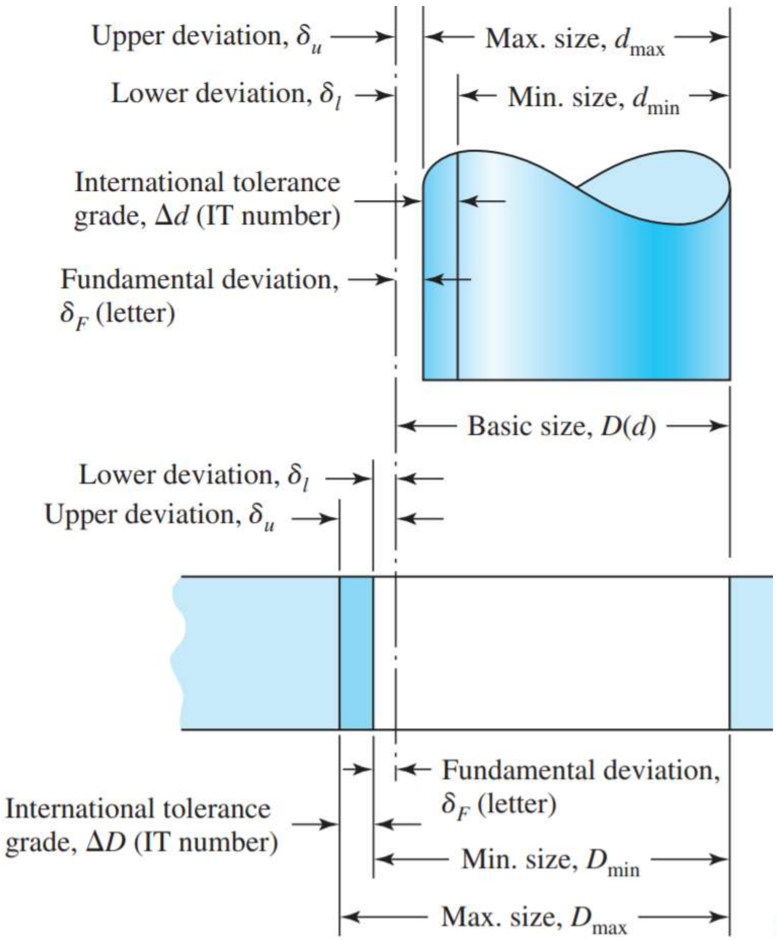
Table 7-20

Type of Fit	Description	Symbol
Clearance	*Loose running fit:* for wide commercial tolerances or allowances on external members	H11/c11
	Free running fit: not for use where accuracy is essential, but good for large temperature variations, high running speeds, or heavy journal pressures	H9/d9
	Close running fit: for running on accurate machines and for accurate location at moderate speeds and journal pressures	H8/f7
	Sliding fit: where parts are not intended to run freely, but must move and turn freely and locate accurately	H7/g6
	Locational clearance fit: provides snug fit for location of stationary parts, but can be freely assembled and disassembled	H7/h6
Transition	*Locational transition fit:* for accurate location, a compromise between clearance and interference	H7/k6
	Locational transition fit: for more accurate location where greater interference is permissible	H7/n6
Interference	*Locational interference fit:* for parts requiring rigidity and alignment with prime accuracy of location but without special bore pressure requirements	H7/p6
	Medium drive fit: for ordinary steel parts or shrink fits on light sections, the tightest fit usable with cast iron	H7/s6
	Force fit: suitable for parts that can be highly stressed or for shrink fits where the heavy pressing forces required are impractical	H7/u6

- Also Tables A-11 to A-14

Preferred Limits and Fits for Cylindrical Parts, ANSI B4.1-1967. Preferred Metric Limits and Fits, ANSI B4.2-1978.

7-8 Limits and Fits



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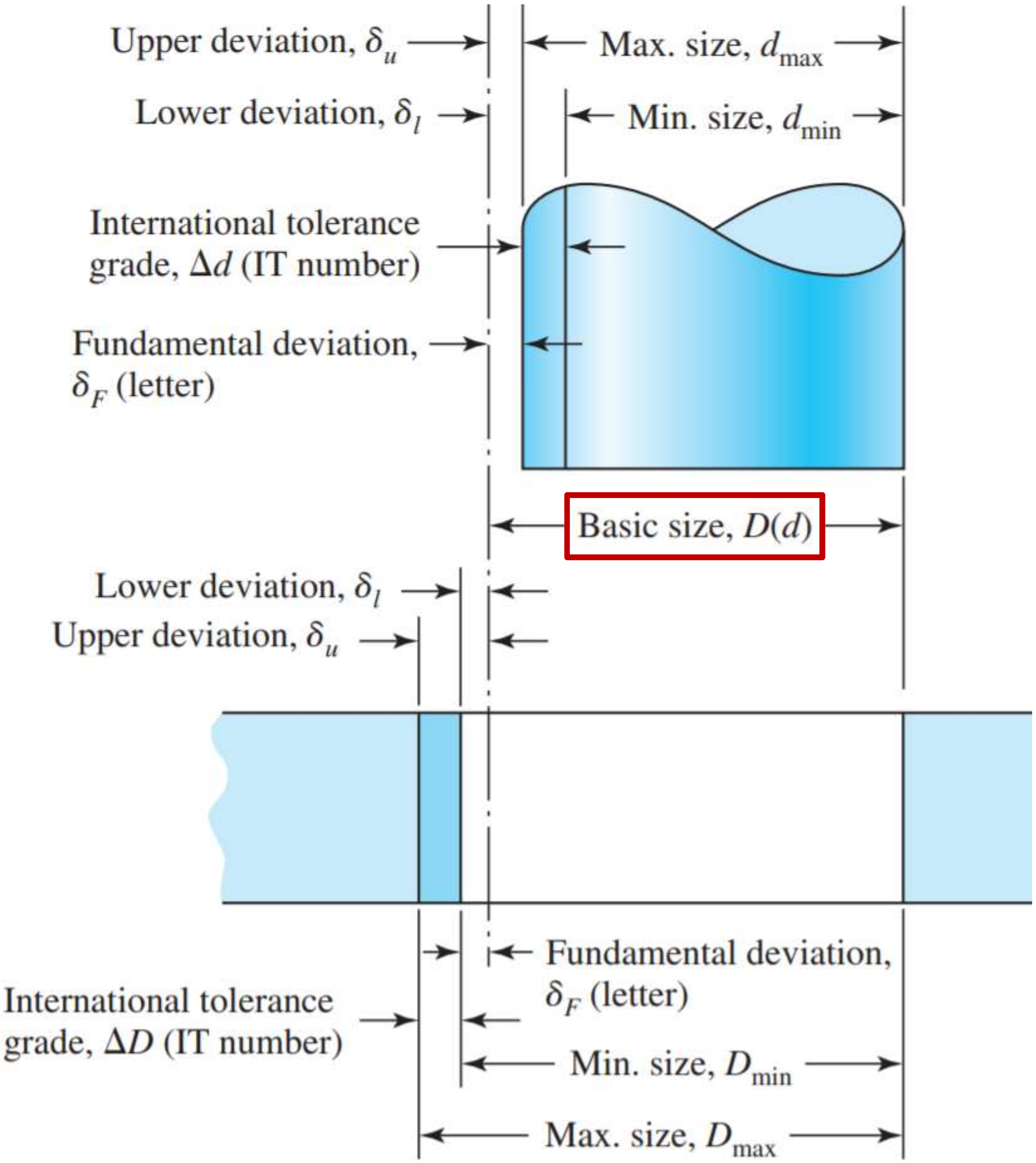
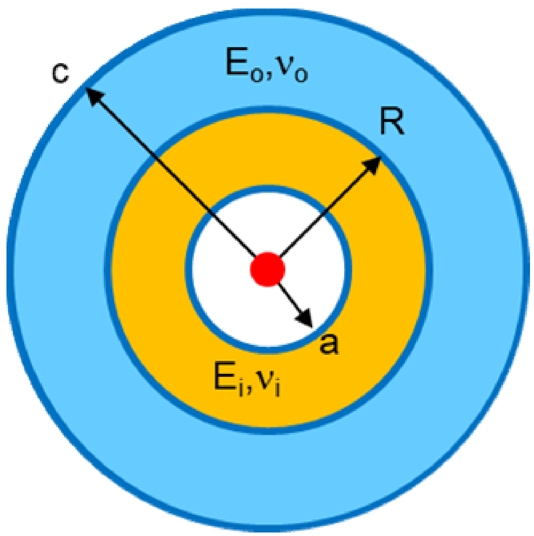
Capital letters always refer to the hole; lowercase letters are used for the shaft.

7-8 Limits and Fits





Example 7-7 Clearance Fit

Find shaft and hole dimensions for a loose running fit with a 34-mm basic size

Type of Fit	Description	Symbol
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Design per **H**11/**c**11 spec

(Capital letter: Hole; Lowercase letter: Shaft)

Example 7-7 Clearance Fit

Find shaft and hole dimensions for a loose running fit with a **34-mm** basic size

International Tolerance (IT)

Table A–11

A Selection of International Tolerance Grades—Metric Series (Size Ranges Are for *Over* the Lower Limit and *Including* the Upper Limit. All Values Are in Millimeters)

Source: Preferred Metric Limits and Fits, ANSI B4.2-1978. See also BSI 4500.

Basic Sizes	Tolerance Grades IT6	IT7	IT8	IT9	IT10	IT11
0–3	0.006	0.010	0.014	0.025	0.040	0.060
3–6	0.008	0.012	0.018	0.030	0.048	0.075
6–10	0.009	0.015	0.022	0.036	0.058	0.090
10–18	0.011	0.018	0.027	0.043	0.070	0.110
18–30	0.013	0.021	0.033	0.052	0.084	0.130
30–50	0.016	0.025	0.039	0.062	0.100	0.160
50–80	0.019	0.030	0.046	0.074	0.120	0.190
80–120	0.022	0.035	0.054	0.087	0.140	0.220
120–180	0.025	0.040	0.063	0.100	0.160	0.250
180–250	0.029	0.046	0.072	0.115	0.185	0.290
250–315	0.032	0.052	0.081	0.130	0.210	0.320
315–400	0.036	0.057	0.089	0.140	0.230	0.360

Per A-11, tolerance grade of H11 $\Delta D=0.160$mm

tolerance grade of c11 $\Delta d=0.160$mm

Hole diameter: $D_{max}=D+\Delta D=34+0.160=34.160$mm; $D_{min}=D=34$mm

Example 7-7 Clearance Fit

Find shaft and hole dimensions for a loose running fit with a **34-mm** basic size

Δd=0.160mm

For shaft with c11, δ_F =-0.120mm

Max shaft dia:
$d_{max}=D+\delta_F$
=34+(-0.120)
=33.880mm

Min shaft dia:
$d_{min}=D+\delta_F-\Delta d$
=34+(-0.120)-Δd
=33.720mm

Table A–12 **Fundamental Deviations for Shafts**

Fundamental Deviations for Shafts—Metric Series
(Size Ranges Are for *Over* the Lower Limit and *Including* the Upper Limit. All Values Are in Millimeters)
Source: Preferred Metric Limits and Fits, ANSI B4.2-1978. See also BSI 4500.

Basic Sizes	Upper-Deviation Letter c	d	f	g	h	Lower-Deviation Letter k	n	p	s	u
0–3	−0.060	−0.020	−0.006	−0.002	0	0	+0.004	+0.006	+0.014	+0.018
3–6	−0.070	−0.030	−0.010	−0.004	0	+0.001	+0.008	+0.012	+0.019	+0.023
6–10	−0.080	−0.040	−0.013	−0.005	0	+0.001	+0.010	+0.015	+0.023	+0.028
10–14	−0.095	−0.050	−0.016	−0.006	0	+0.001	+0.012	+0.018	+0.028	+0.033
14–18	−0.095	−0.050	−0.016	−0.006	0	+0.001	+0.012	+0.018	+0.028	+0.033
18–24	−0.110	−0.065	−0.020	−0.007	0	+0.002	+0.015	+0.022	+0.035	+0.041
24–30	−0.110	−0.065	−0.020	−0.007	0	+0.002	+0.015	+0.022	+0.035	+0.048
30–40	−0.120	−0.080	−0.025	−0.009	0	+0.002	+0.017	+0.026	+0.043	+0.060
40–50	−0.130	−0.080	−0.025	−0.009	0	+0.002	+0.017	+0.026	+0.043	+0.070
50–65	−0.140	−0.100	−0.030	−0.010	0	+0.002	+0.020	+0.032	+0.053	+0.087
65–80	−0.150	−0.100	−0.030	−0.010	0	+0.002	+0.020	+0.032	+0.059	+0.102
80–100	−0.170	−0.120	−0.036	−0.012	0	+0.003	+0.023	+0.037	+0.071	+0.124
100–120	−0.180	−0.120	−0.036	−0.012	0	+0.003	+0.023	+0.037	+0.079	+0.144
120–140	−0.200	−0.145	−0.043	−0.014	0	+0.003	+0.027	+0.043	+0.092	+0.170
140–160	−0.210	−0.145	−0.043	−0.014	0	+0.003	+0.027	+0.043	+0.100	+0.190
160–180	−0.230	−0.145	−0.043	−0.014	0	+0.003	+0.027	+0.043	+0.108	+0.210
180–200	−0.240	−0.170	−0.050	−0.015	0	+0.004	+0.031	+0.050	+0.122	+0.236

Diametral Clearance:
$\delta_{max}=D_{max}-d_{min}$=34.160-33.720=0.440mm=$\Delta D-\delta_F$
$\delta_{min}=D_{min}-d_{max}$=34.000-33.880=0.120mm=$-\delta_F$

Example 7-7 Clearance Fit

Find shaft and hole dimensions for a loose running fit with a **34-mm** basic size (ANSI B4.2-1978 for detailed H11/c11 fit dimensions)

Basic Size		Loose Running			Free Running			Close Running			Sliding			Locational Clearance		
		Hole H11	Shaft c11	Fit[†]	Hole H9	Shaft d9	Fit[†]	Hole H8	Shaft f7	Fit[†]	Hole H7	Shaft g6	Fit[†]	Hole H7	Shaft h6	Fit[†]
25	Max	25.130	24.890	0.370	25.052	24.935	0.169	25.033	24.980	0.074	25.021	24.993	0.041	25.021	25.000	0.034
	Min	25.000	24.760	0.110	25.000	24.883	0.065	25.000	24.959	0.010	25.000	24.980	0.007	25.000	24.987	0.000
30	Max	30.130	29.890	0.370	30.052	29.935	0.169	30.033	29.980	0.074	30.021	29.993	0.041	30.021	30.000	0.034
	Min	30.000	29.760	0.110	30.000	19.883	0.065	30.000	29.959	0.020	30.000	29.980	0.007	30.000	29.987	0.000
40	Max	40.160	39.880	0.440	40.062	39.920	0.204	40.039	39.975	0.089	40.025	39.991	0.050	40.025	40.000	0.041
	Min	40.000	39.720	0.120	40.000	39.858	0.080	40.000	39.950	0.025	40.000	39.975	0.009	40.000	39.984	0.000
50	Max	50.160	49.870	0.450	50.062	49.920	0.204	50.039	49.975	0.089	50.025	49.991	0.050	50.025	50.000	0.041
	Min	50.000	49.710	0.130	50.000	49.858	0.080	50.000	49.950	0.025	50.000	49.975	0.009	50.000	49.984	0.000
60	Max	60.190	59.860	0.520	60.074	59.900	0.248	60.046	59.970	0.106	60.030	59.990	0.059	60.030	60.000	0.049
	Min	60.000	59.670	0.140	60.000	59.826	0.100	60.000	59.940	0.030	60.000	59.971	0.010	60.000	59.981	0.000
80	Max	80.190	79.850	0.530	80.074	79.900	0.248	80.046	79.970	0.106	80.030	79.990	0.059	80.030	80.000	0.049
	Min	80.000	79.660	0.150	80.000	79.826	0.100	80.000	79.940	0.030	80.000	79.971	0.010	80.000	79.981	0.000
100	Max	100.220	99.830	0.610	100.087	99.880	0.294	100.054	99.964	0.125	100.035	99.988	0.069	100.035	100.000	0.057
	Min	100.000	99.610	0.170	100.000	99.793	0.120	100.000	99.929	0.036	100.000	99.966	0.012	100.000	99.978	0.000
120	Max	120.220	119.820	0.620	120.087	119.880	0.294	120.054	119.964	0.125	120.035	119.988	0.069	120.035	120.000	0.057
	Min	110.000	119.600	0.180	120.000	119.793	0.120	120.000	119.929	0.036	120.000	119.966	0.012	120.000	119.978	0.000

Example 7-8 Interference Fit

Find the hole and shaft limits for a medium drive fit using a basic hole size of 2 in.

Type of Fit	Description	Symbol
Clearance	*Loose running fit:* for wide commercial tolerances or allowances on external members	H11/c11
	Free running fit: not for use where accuracy is essential, but good for large temperature variations, high running speeds, or heavy journal pressures	H9/d9
	Close running fit: for running on accurate machines and for accurate location at moderate speeds and journal pressures	H8/f7
	Sliding fit: where parts are not intended to run freely, but must move and turn freely and locate accurately	H7/g6
	Locational clearance fit: provides snug fit for location of stationary parts, but can be freely assembled and disassembled	H7/h6
Transition	*Locational transition fit:* for accurate location, a compromise between clearance and interference	H7/k6
	Locational transition fit: for more accurate location where greater interference is permissible	H7/n6
Interference	*Locational interference fit:* for parts requiring rigidity and alignment with prime accuracy of location but without special bore pressure requirements	H7/p6
	Medium drive fit: for ordinary steel parts or shrink fits on light sections, the tightest fit usable with cast iron	H7/s6
	Force fit: suitable for parts that can be highly stressed or for shrink fits where the heavy pressing forces required are impractical	H7/u6

Table A-13 International Tolerance (IT)

Basic Sizes	Tolerance Grades IT6	IT7	IT8	IT9	IT10	IT11
0–0.12	0.0002	0.0004	0.0006	0.0010	0.0016	0.0024
0.12–0.24	0.0003	0.0005	0.0007	0.0012	0.0019	0.0030
0.24–0.40	0.0004	0.0006	0.0009	0.0014	0.0023	0.0035
0.40–0.72	0.0004	0.0007	0.0011	0.0017	0.0028	0.0043
0.72–1.20	0.0005	0.0008	0.0013	0.0020	0.0033	0.0051
1.20–2.00	0.0006	0.0010	0.0015	0.0024	0.0039	0.0063
2.00–3.20	0.0007	0.0012	0.0018	0.0029	0.0047	0.0075
3.20–4.80	0.0009	0.0014	0.0021	0.0034	0.0055	0.0087
4.80–7.20	0.0010	0.0016	0.0025	0.0039	0.0063	0.0098
7.20–10.00	0.0011	0.0018	0.0028	0.0045	0.0073	0.0114
10.00–12.60	0.0013	0.0020	0.0032	0.0051	0.0083	0.0126
12.60–16.00	0.0014	0.0022	0.0035	0.0055	0.0091	0.0142

Tolerance grade H7 ΔD=0.0010”
Tolerance grade s6 Δd=0.0006”

Hole Diameter:
$D_{max}=D+\Delta D=2+0.0010=2.0010$”
$D_{min}=D=2.0000$”

Example 7-8 Interference Fit

For shaft with s6, δ_F =0.0017”

Min shaft OD=d_{min}=D+ δ_F =2+0.0017=2.0017”

Max shaft OD= d_{max}=D+ δ_F +Δd=2+0.0017+0.0006=2.0023”

Table A–14

Fundamental Deviations for Shafts—Inch Series (Size Ranges Are for *Over* the Lower Limit and *Including* the Upper Limit. All Values Are in Inches, Converted from Table A–12)

Basic Sizes	Upper-Deviation Letter					Lower-Deviation Letter				
	c	d	f	g	h	k	n	p	s	u
0–0.12	−0.0024	−0.0008	−0.0002	−0.0001	0	0	+0.0002	+0.0002	+0.0006	+0.0007
0.12–0.24	−0.0028	−0.0012	−0.0004	−0.0002	0	0	+0.0003	+0.0005	+0.0007	+0.0009
0.24–0.40	−0.0031	−0.0016	−0.0005	−0.0002	0	0	+0.0004	+0.0006	+0.0009	+0.0011
0.40–0.72	−0.0037	−0.0020	−0.0006	−0.0002	0	0	+0.0005	+0.0007	+0.0011	+0.0013
0.72–0.96	−0.0043	−0.0026	−0.0008	−0.0003	0	+0.0001	+0.0006	+0.0009	+0.0014	+0.0016
0.96–1.20	−0.0043	−0.0026	−0.0008	−0.0003	0	+0.0001	+0.0006	+0.0009	+0.0014	+0.0019
1.20–1.60	−0.0047	−0.0031	−0.0010	−0.0004	0	+0.0001	+0.0007	+0.0010	+0.0017	+0.0024
1.60–2.00	−0.0051	−0.0031	−0.0010	−0.0004	0	+0.0001	+0.0007	+0.0010	+0.0017	+0.0028
2.00–2.60	−0.0055	−0.0039	−0.0012	−0.0004	0	+0.0001	+0.0008	+0.0013	+0.0021	+0.0034
2.60–3.20	−0.0059	−0.0039	−0.0012	−0.0004	0	+0.0001	+0.0008	+0.0013	+0.0023	+0.0040
3.20–4.00	−0.0067	−0.0047	−0.0014	−0.0005	0	+0.0001	+0.0009	+0.0015	+0.0028	+0.0049
4.00–4.80	−0.0071	−0.0047	−0.0014	−0.0005	0	+0.0001	+0.0009	+0.0015	+0.0031	+0.0057
4.80–5.60	−0.0079	−0.0057	−0.0017	−0.0006	0	+0.0001	+0.0011	+0.0017	+0.0036	+0.0067
5.60–6.40	−0.0083	−0.0057	−0.0017	−0.0006	0	+0.0001	+0.0011	+0.0017	+0.0039	+0.0075
6.40–7.20	−0.0091	−0.0057	−0.0017	−0.0006	0	+0.0001	+0.0011	+0.0017	+0.0043	+0.0083
7.20–8.00	−0.0094	−0.0067	−0.0020	−0.0006	0	+0.0002	+0.0012	+0.0020	+0.0048	+0.0093
8.00–9.00	−0.0102	−0.0067	−0.0020	−0.0006	0	+0.0002	+0.0012	+0.0020	+0.0051	+0.0102
9.00–10.00	−0.0110	−0.0067	−0.0020	−0.0006	0	+0.0002	+0.0012	+0.0020	+0.0055	+0.0112
10.00–11.20	−0.0118	−0.0075	−0.0022	−0.0007	0	+0.0002	+0.0013	+0.0022	+0.0062	+0.0124
11.20–12.60	−0.0130	−0.0075	−0.0022	−0.0007	0	+0.0002	+0.0013	+0.0022	+0.0067	+0.0130
12.60–14.20	−0.0142	−0.0083	−0.0024	−0.0007	0	+0.0002	+0.0015	+0.0024	+0.0075	+0.0154
14.20–16.00	−0.0157	−0.0083	−0.0024	−0.0007	0	+0.0002	+0.0015	+0.0024	+0.0082	+0.0171

Diametral interference

δ_{max}=d_{max}-D_{min}

2.0023-2.000

=0.0023”

=δ_F +Δd

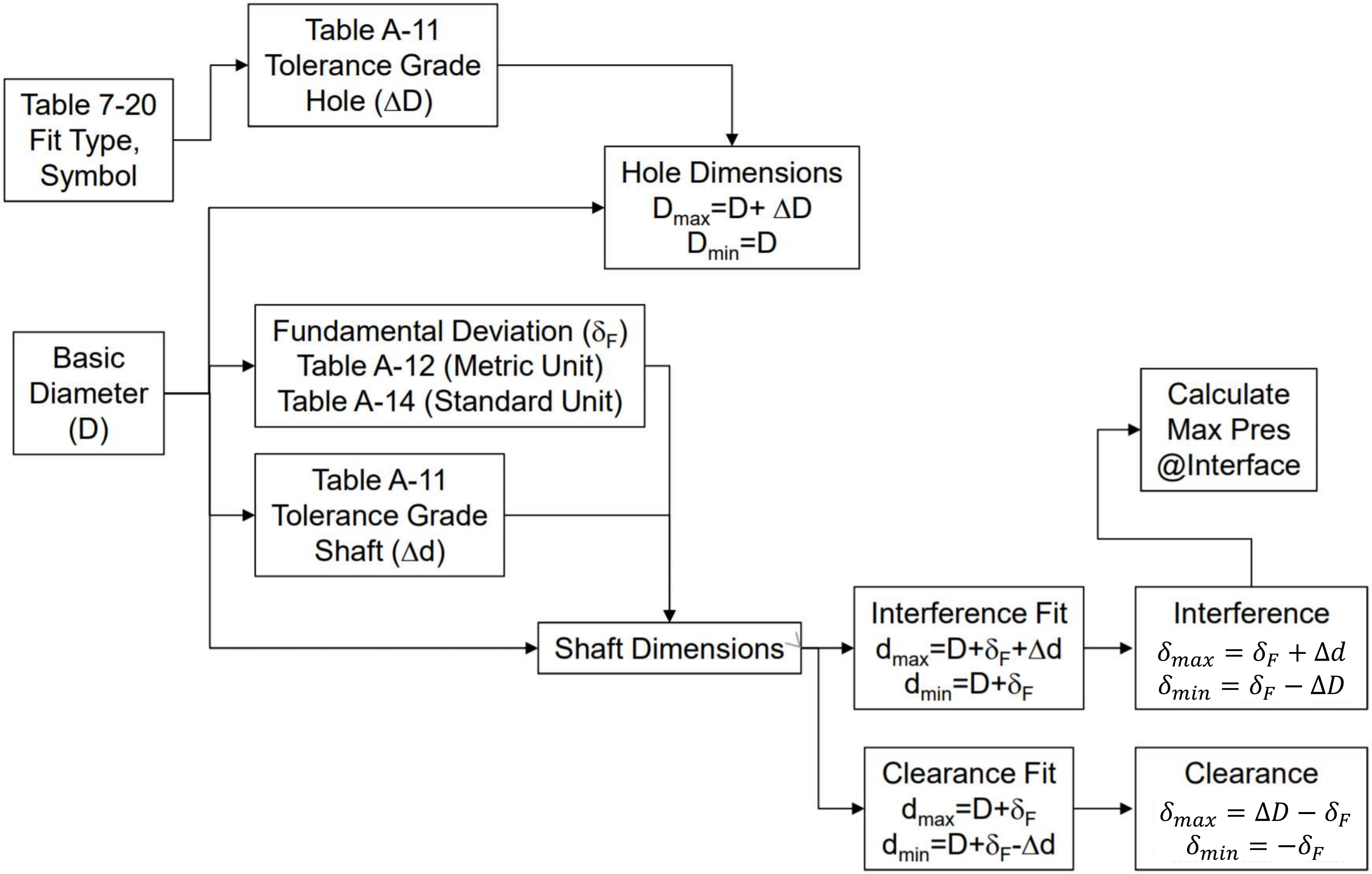
Min interference

δ_{min}=2.0017-2.0010

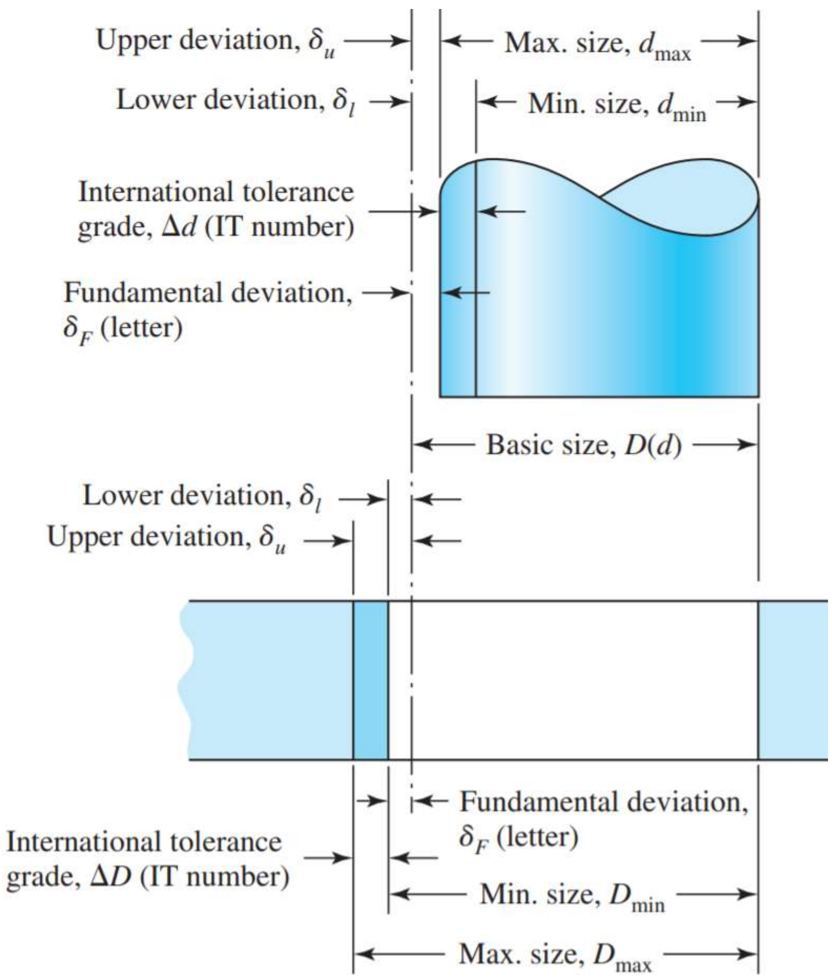
=0.0007”

=δ_F -ΔD

Press-Fit & Interference-Fit Design Workflow



7-8 Limits and Fits

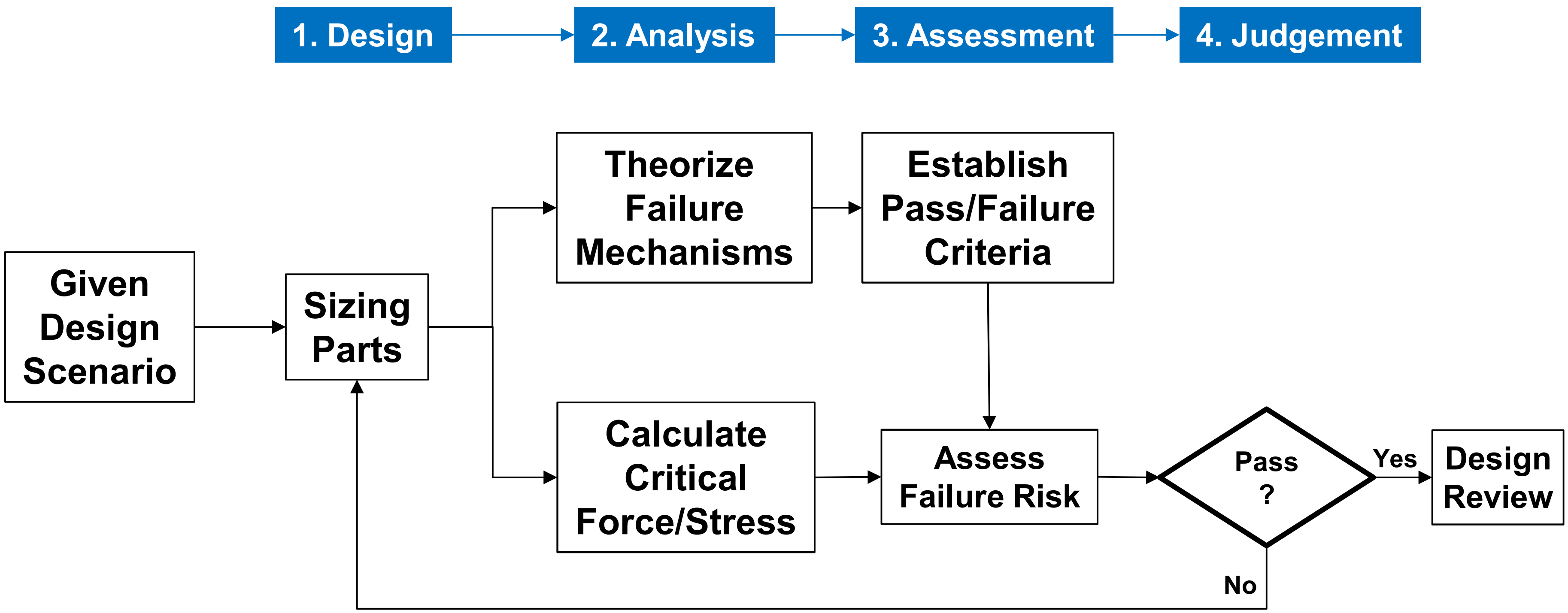


Type of Fit	Description	Symbol
Clearance	*Loose running fit:* for wide commercial tolerances or allowances on external members	H11/c11
	Free running fit: not for use where accuracy is essential, but good for large temperature variations, high running speeds, or heavy journal pressures	H9/d9
	Close running fit: for running on accurate machines and for accurate location at moderate speeds and journal pressures	H8/f7
	Sliding fit: where parts are not intended to run freely, but must move and turn freely and locate accurately	H7/g6
	Locational clearance fit: provides snug fit for location of stationary parts, but can be freely assembled and disassembled	H7/h6
Transition	*Locational transition fit:* for accurate location, a compromise between clearance and interference	H7/k6
	Locational transition fit: for more accurate location where greater interference is permissible	H7/n6
Interference	*Locational interference fit:* for parts requiring rigidity and alignment with prime accuracy of location but without special bore pressure requirements	H7/p6
	Medium drive fit: for ordinary steel parts or shrink fits on light sections, the tightest fit usable with cast iron	H7/s6
	Force fit: suitable for parts that can be highly stressed or for shrink fits where the heavy pressing forces required are impractical	H7/u6

Force fit grade is commonly used for driveshaft torque transmission.

Design Discipline

Every design we do in this class, we will follow this discipline…..



Application Notes:
Thick-Walled Cylinder and Its Stress Distribution

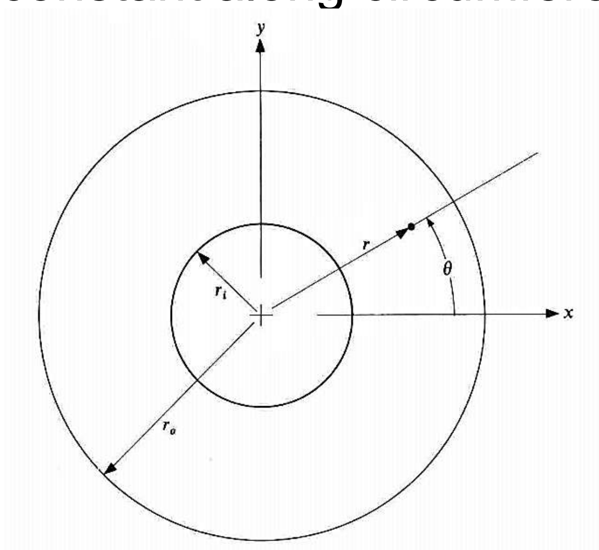
Characteristic Stresses of Thick-Walled Cylinders

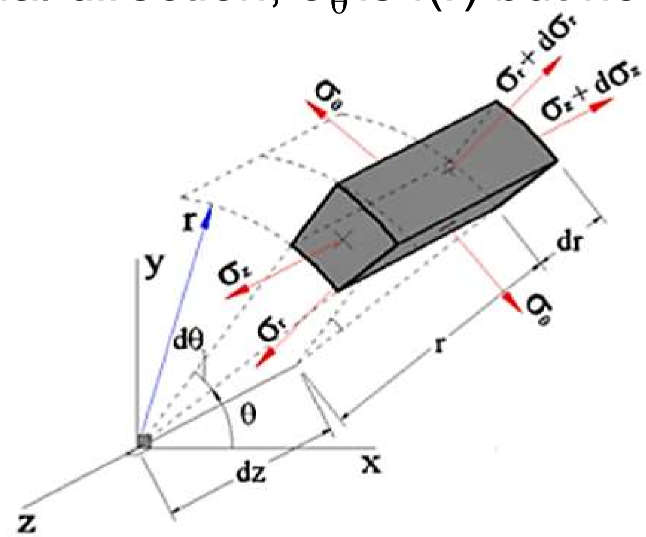
Characteristic stresses on a thick-walled cylinder are

- Circumferential (hoop) stress σ_θ,
- Radial stress σ_r, and
- Longitudinal (axial) stress σ_z

Assuming cylinder geometry is symmetric along axial axis Z,

- Shear stress $\tau_{r\theta}$ will not develop; $\tau_{r\theta} = 0$
- σ_θ constant along circumferential direction; σ_θ is f(r) but not f(θ)



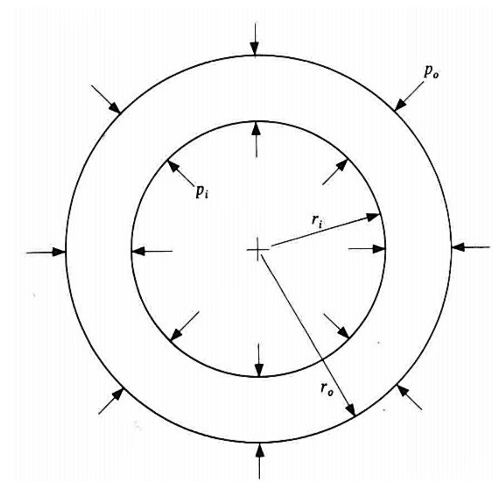


Stress Distributions on Thick-Walled Cylinders with Internal (pi) and External Pressures (po)

Per force equilibrium on elemental basis, generalized stress distributions on a thick-wall cylinder are:

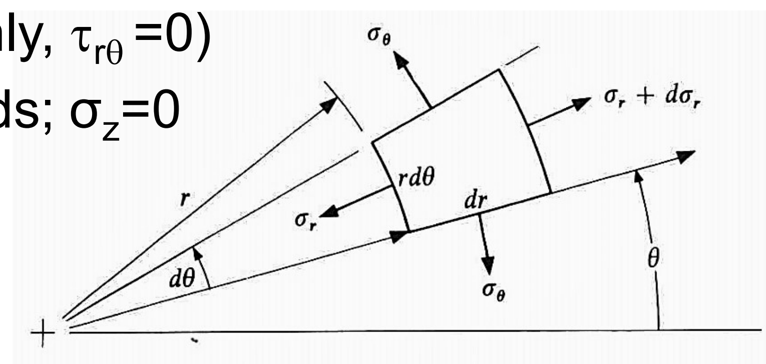
$$\sigma_r = \frac{p_i r_i^2 - p_o r_o^2 + (r_o r_i / r)^2 (p_o - p_i)}{r_o^2 - r_i^2}$$

$$\sigma_\theta = \frac{p_i r_i^2 - p_o r_o^2 - (r_o r_i / r)^2 (p_o - p_i)}{r_o^2 - r_i^2}$$



Assumptions imposed to derive the above equations:

- Cylinder symmetric along Z (σ_θ=f(r) only, $\tau_{r\theta}$ =0)
- Cylinder free of constraints on two ends; σ_z=0
- Cylinder rotation negligible; ω~0



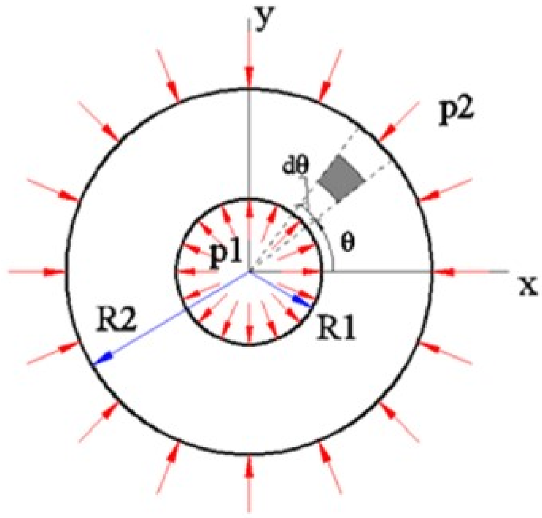
Thick-Walled Cylinder

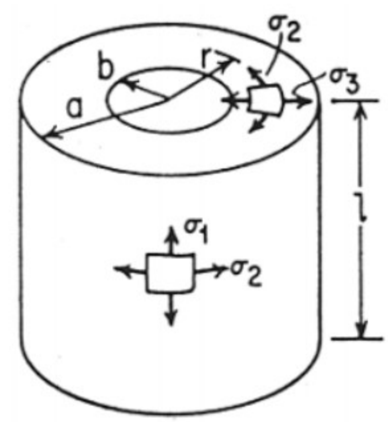
Thick-Walled vs. Thin-Walled

- Thick-Walled cylinder: wall thickness (t) greater than 10 percent of the average radius.
- Average Radius: $r_a = \frac{r_i + r_o}{2}$ $\quad t = r_o - r_i$
- For thick-walled cylinder: $t \geq \frac{r_a}{10}$

  , which implies $r_o \geq 1.11 r_i$

  Generally $r_o \geq 1.15 r_i$ at least
- ASME BPVC suggests $r_o \geq 1.25 r_i$ to qualify for thick-walled cylinder.
- Could be pressurized internally and/or externally





Effects of Cylinder OD Pressure on Radial Stress

$$\sigma_r = \frac{p_i r_i^2 - p_o r_o^2 + (r_o r_i / r)^2 (p_o - p_i)}{r_o^2 - r_i^2}$$

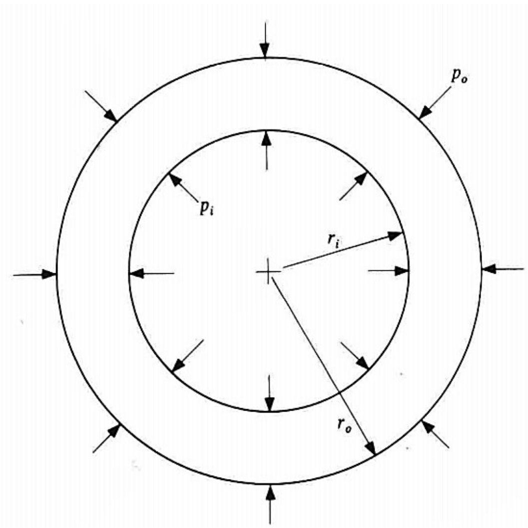
- If OD surface is unpressurized ($p_o = 0$), the radial stress at OD is

$$(\sigma_r)_{r=ro} = 0$$

- If OD surface is pressurized ($p_o \neq 0$), the radial stress at OD is

$$(\sigma_r)_{r=r} = -p_o$$

"-" indicates it is in compression



Lesson Learned

$$\sigma_r = \frac{p_i r_i^2 - p_o r_o^2 + (r_o r_i/r)^2 (p_o - p_i)}{r_o^2 - r_i^2}$$

$$\sigma_\theta = \frac{p_i r_i^2 - p_o r_o^2 - (r_o r_i/r)^2 (p_o - p_i)}{r_o^2 - r_i^2}$$

- Case 1: $p_o = 0$, $p_i \neq 0$ **Internal Pressure Only**

$$\sigma_r = p_i r_i^2 \frac{1 - (r_o/r)^2}{r_o^2 - r_i^2} \leq 0 \qquad \sigma_\theta = p_i r_i^2 \frac{1 + (r_o/r)^2}{r_o^2 - r_i^2} > 0$$

  – Radial stress is compressive, but hoop stress is tensile always

- Case 2: $p_o \neq 0$, $p_i = 0$ **External Pressure Only**

$$\sigma_r = p_o r_o^2 \frac{-1 + (r_i/r)^2}{r_o^2 - r_i^2} \leq 0 \qquad \sigma_\theta = p_o r_o^2 \frac{-1 - (r_i/r)^2}{r_o^2 - r_i^2} < 0$$

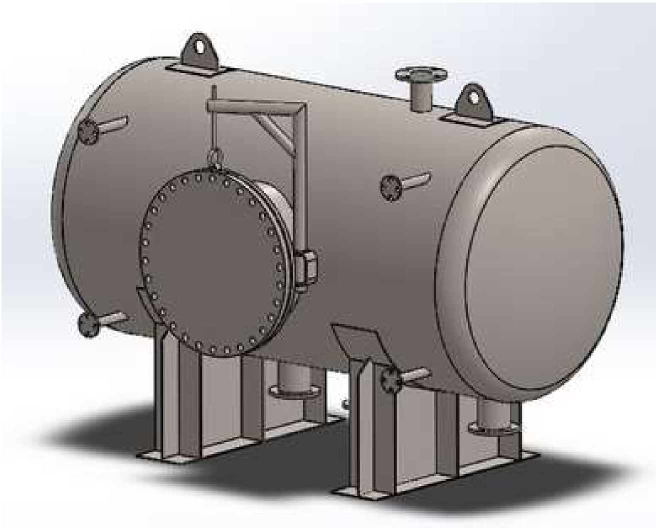
  – Both radial stress and hoop stress are compressive always

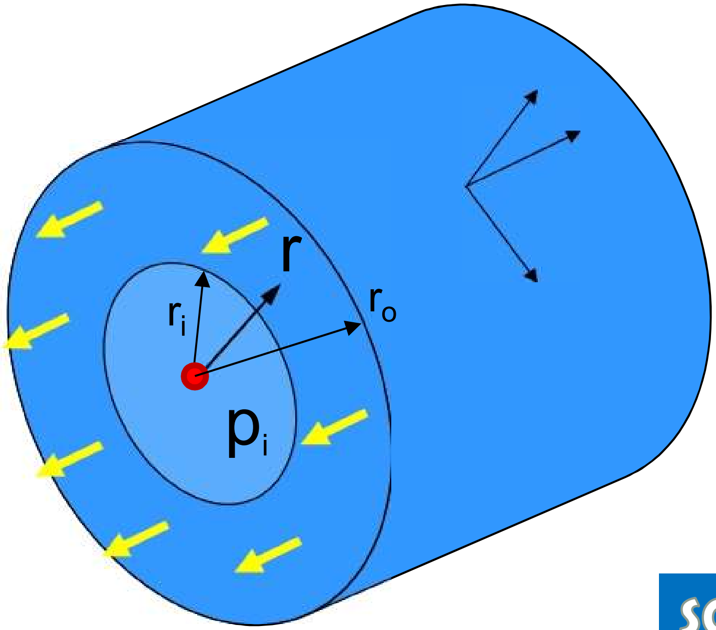
Longitudinal Stress

- Generally only considered for the case of internal pressurization ($p_o = 0$)
- Longitudinal stress is simply given by a Force/Area, where
  - Force $F = p_i \pi r_i^2$
  - Area = annular area of the cylinder cross section $A = \pi\left(r_o^2 - r_i^2\right)$

$$\sigma_z = \frac{F}{A} = \frac{p_i r_i^2}{r_o^2 - r_i^2}$$

An approximation of the average stress on end faces, not true stress distribution.



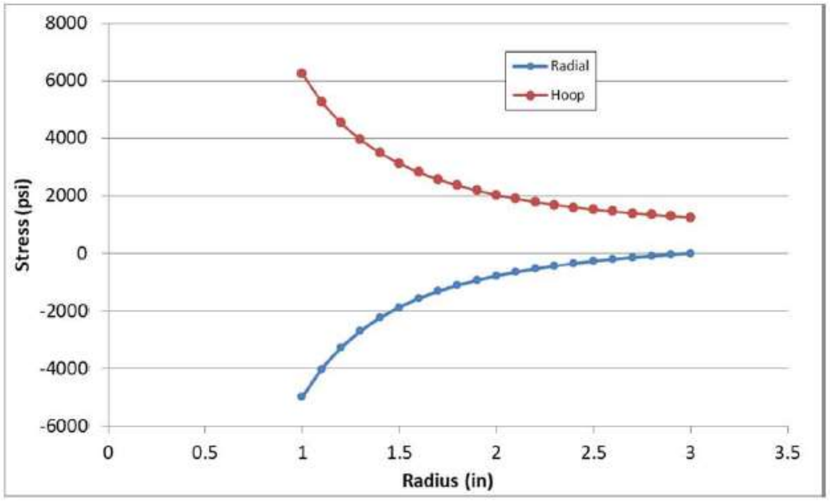


Example: Internal Pressure Only ($p_i \neq 0$, $p_o = 0$)

Determine the stress distribution in a cylinder with inner diameter of 2” and outer diameter of 6” with p_i=5000 psi and p_o=0 psi

$$\sigma_r = \frac{5000 * 1^2 + [\frac{(3*1)}{r}]^2(0-5000)}{3^2-1^2} = 625 - \frac{5625}{r^2}$$

$$\sigma_\theta = \frac{5000 * 1^2 - [\frac{(3*1)}{r}]^2(0-5000)}{3^2-1^2} = 625 + \frac{5625}{r^2}$$



Radial Stress

- On ID surface, σ_r= -5000 psi = $-p_i$
- On OD surface, σ_r= 0 psi = p_o

Hoop Stress

- Maximum at the inner surface, 6250 psi
- Lower, but not zero, at the unpressurized outer surface, 1250 psi.
- Magnitude of hoop stress is greater than radial stress

Longitudinal Stress

$$\sigma_z = \frac{p_i r_i^2}{r_o^2 - r_i^2} = \frac{5000 \cdot 1^2}{3^2 - 1^2} = 625\ psi$$

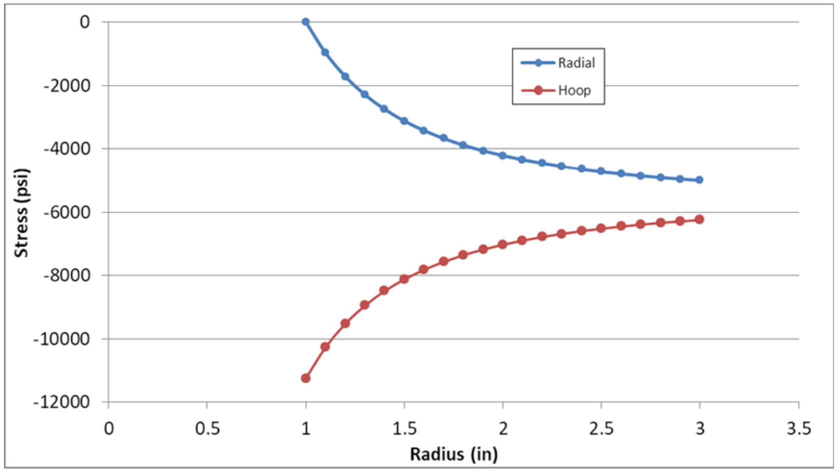
Longitudinal Stress σ_z = 625 psi, considered as a uniform, average stress across the thickness of the wall.

Example: External Pressure Only (p_i=0, $p_o \neq$ 0)

- Determine the stress distribution in a cylinder with inner diameter of 2" and outer diameter of 6" with p_i = 0 psi and p_o = 5000 psi

$$\sigma_r = \frac{-5000 * 3^2 + [\frac{(3*1)}{r}]^2(5000-0)}{3^2 - 1^2} = -5625 + \frac{45000}{r^2}$$

$$\sigma_\theta = \frac{-5000 * 3^2 - [\frac{(3*1)}{r}]^2(5000-0)}{3^2 - 1^2} = -5625 - \frac{45000}{r^2}$$



Radial Stress

- On ID surface, σ_r= 0 psi = p_i
- On OD surface, σ_r= -5000 psi = -p_o

Hoop Stress

- Maximum at the outer surface, -6250 psi
- Minimum at the unpressurized inner surface, -11250 psi.
- Absolute magnitude of hoop stress is greater than radial stress

Longitudinal Stress is not usually considered for external pressure.